

Study of the Prediction Method for Maximum Traction Coefficient

Toshinari Sano and Mitsuaki Tomita
Toyota Motor Corporation

Masashi Inoue, Yasuhiro Takeuchi and Muneo Yorinaga
Nippon Soken Inc

ABSTRACT

This report proposes a rheological model and a thermal analysis model for oil films, which transmit power through a variator, as a prediction method for the maximum traction coefficient, and then describes the application and verification of this method.

The rheological model expresses the conditions inside the contact ellipse using a combination of viscosity and plasticity.

The thermal analysis model for oil films was confirmed by comparison of previously obtained temperatures directly measured from the traction contact area of the four-roller experimental apparatus [1]. The measurement used a thin-film temperature sensor and the consistency between the calculated and measured values was verified in the estimation model by reflecting the precise thermal properties of the thin film. Most values were consistent with the calculated values for the middle plane local shear heating model inside the oil film. However, under some conditions, the values were closer to those calculated for homogeneous shear. Based on these results, this paper proposes a calculation method that combines both heating models within the contact ellipse.

In addition, the variator in a continuously variable transmission (CVT) was verified for the first time by using a test box to take direct temperature measurements from the transmission contact area in the CVT. This test verified the precision of the thermal analysis model, including the spin motion.

Finally, the maximum traction coefficient was estimated using these calculation models and compared with measured values.

CITATION: Sano, T., Tomita, M., Inoue, M., Takeuchi, Y. et al., "Study of the Prediction Method for Maximum Traction Coefficient," *SAE Int. J. Passeng. Cars - Mech. Syst.* 6(2):2013, doi:10.4271/2013-01-0366.

1. INTRODUCTION

The development of fuel efficient vehicles is being actively promoted as a result of growing awareness in environmental and energy-related issues. In addition to improving transmission efficiency, key measures for transmissions include increasing the number of shift speeds, introducing continuously variable transmissions (CVTs), and widening shift speed ranges.

Tribology technology plays a major role in the development of traction-drive type CVTs since such transmissions transfer power through a solid film (hyaloid). The maximum surface contact pressure of the transmission contact area that forms the oil film exceeds 4 GPa and requires a normal load of more than 1 ton. This large load is one factor that makes it difficult to reduce the size of the

CVT. Furthermore, relative rotation (generally referred to as "spin") is generated at the contact surface, which is one cause of efficiency reduction. The amount of spin differs in accordance with the geometrical structure of the variator. One such structure is referred to as a half-toroidal configuration, which can be designed with low spin at the transmission contact area.

According to a recent report, a double-cavity half-toroidal variator has been developed with a transmission efficiency of 97% or more in the overdrive ratio and an increased ratio range of 7.5 [2]. However, such a high transmission efficiency and wide ratio range requires excess normal load at the transmission contact area to be minimized. Accurately estimating the maximum traction coefficient ($\mu_{\max}$) is important to accomplish this goal.



2. CALCULATION MODEL PROPOSAL

2.1. Rheological Model and Calculation Method

Film shearing under high traction oil pressures is generally categorized as viscous, visco-elastic, or elasto-plastic. Estimations of the maximum traction coefficient require the use of comparatively high shear rate regions, which means that elastic terms can frequently be ignored. Consequently, the proposed method attempts to express the rheological model using viscosity and plasticity only. Eyring's non-linear model for viscous flow can be expressed using Equation (1).

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

Conventionally, the rheological parameter Eyring stress τ_0 is reverse-calculated from a measured traction curve [3,4], and the high-pressure viscosity η_p is estimated by fitting a model to values measured using a dedicated measurement apparatus [5]. The resulting maximum traction coefficient calculation is regarded to have a close correlation with actually measured values. However, when traction oils with enhanced properties are adopted, the rheological parameters have to be re-calculated. This requires many processes, including time-consuming measurements of high-pressure viscosity.

Hata proposed estimation formulae for high-pressure viscosity and Eyring stress [6,7]. These are calculated in consideration of molecular structure and intermolecular interaction, based on Eyring's theory of fluid viscosity. These formulae are capable of accurately estimating rheological parameters and can also be calculated from easy-to-measure general properties such as molecular weight and atmospheric viscosity and density. Accurate calculation of the traction coefficient requires identification of the fluid characteristics used. Since Hata reported that the estimated rheological parameters closely matched actual measurements, it was determined to use Hata's formulae in this research. Hata's formulae are shown in Equations (2), (3), (4), (5) below. The term CED in the equations refers to the cohesive energy density. This is a scale for indicating the strength of the tendency of liquid molecules to cohere to each other.

$$CED = 2.04 \times 10^{-5} \cdot \theta_f \cdot \frac{\rho_{f0}}{MW} \cdot \ln(2.51 \times 10^6 \cdot \nu_{f0} \cdot MW) \quad (2)$$

$$\alpha_p = \frac{\alpha_0}{1 + \alpha_0 P(-0.6 \cdot CED + 0.108) \cdot \exp\{18(CED - 3.8)P\}} \quad (3)$$

$$\eta_p = \eta_{f0} \exp(\alpha_p P) \quad (4)$$

$$\tau_0 = 4.62 \times 10^{-6} \cdot \theta_f \cdot \frac{\rho_{f0}}{MW} \cdot \exp\{15.2(CED - 0.18)P\} \quad (5)$$

In the plastic region, oil is treated as an object that has been solidified by glass transition. Plastic flow is regarded as the shear fracture of a solid, and the generated stress is defined as the limiting shear stress τ_L . Although Tabor defined a qualitative theory for calculating the limiting shear stress of a solid metal [8], there is no theoretical equation for calculating the limiting shear stress of a traction oil with an amorphous structure. In this paper, the limiting shear stress is calculated by an empirical formula (Equation (6)), which was obtained by linking the temperature calculation model (See section 2.2) to data obtained from a four-roller experiment under high speed conditions using a synthetic naphthene oil developed by Idemitsu Kosan Co., Ltd.

$$\tau_L = (0.1518 - 0.0001577 \cdot \theta_f)P + 0.01459 \quad (6)$$

Figure 1 shows that the results obtained using the empirical formula closely correlate with the pressure and temperature dependence in the test data

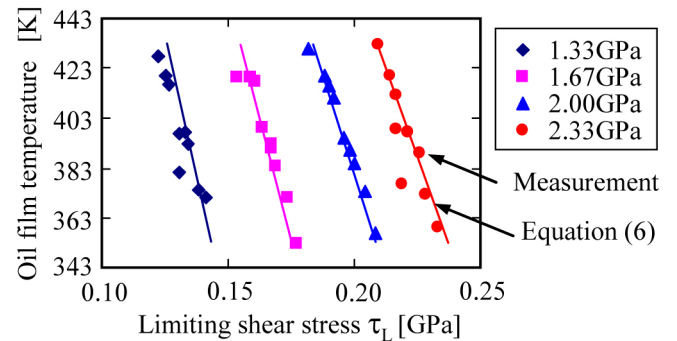


Figure 1. Fitting result of limiting shear stress model (measured data obtained under following conditions: Lubrication temperature 343, 373, and 403 K, rolling velocity 10, 20, and 26.18 m/sec, average Hertzian contact pressure 1.33, 1.67, 2.00, and 2.33 GPa)

The following section explains the usage of the viscosity and plasticity models.

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 was selected from the two rheological models for each mesh in accordance with the local conditions (such as the surface contact pressure inside the ellipse, oil film temperature, shear rate, and the like). The total integrated rolling direction component of the shearing force generated inside the meshes was defined as the traction force. The power that cannot be transmitted in the rolling direction between the contact surfaces forms heating loss.

The boundary for model selection is defined as the intersection of the shear stress in the viscous and plastic

models as shown on the left of Fig. 2. This can be interpreted as the boundary at which the viscosity term η_p in Equation (1) increases exponentially with respect to the pressure and the shear can no longer be expressed as fluid behavior. The model with the lowest stress value is selected. The pattern diagram on the right of Fig. 2 illustrates the surface contact pressure distribution inside the contact ellipse and the switching of the rheological models. 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. The shear rate $\dot{\gamma}$ is calculated using this film thickness.

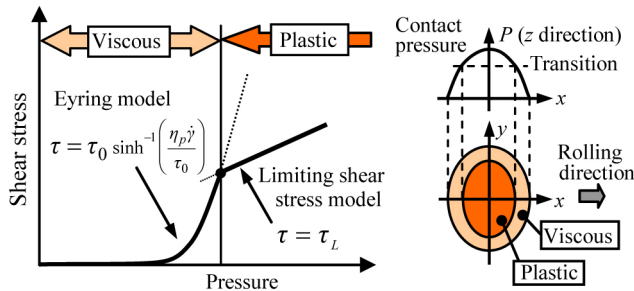


Figure 2. Transition from viscous to plastic region

2.2. Thermal Analysis Model for Contact Area

An estimation of the contact area temperature requires a calculation of the heating loss and the 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 in Section 2.1. These models are temperature functions. Traction drive is characterized by the relative sensitivity of stress values and stress limits as a factor of the temperature. Equation (7) shows the component terms of the oil film temperature θ_f .

$$\theta_f = \theta_0 + \Delta\theta_{inlet} + \Delta\theta_S + \Delta\theta_f \quad (7)$$

θ_0 is the rolling element temperature. The temperature rise at the contact film inlet zone $\Delta\theta_{inlet}$ uses Gohar's equation [9]. The maximum temperature rise of the rolling surface inside the contact area $\Delta\theta_S$ uses Carslaw and Jaeger's correction equation [10] for the rise in temperature due to a moving heat source. With respect to the maximum oil film temperature rise $\Delta\theta_f$, the two types of shear heating shown in Fig. 3 were studied using the heating and thermal conduction models. These models are middle plane shear heating, which is generated by local shear in a portion of the surface, and homogeneous shear heating, which is generated by uniform shear inside the oil film. The former is a case of solid shear fracture, the latter is liquid shearing, which may actually be in mixed condition at the boundary.

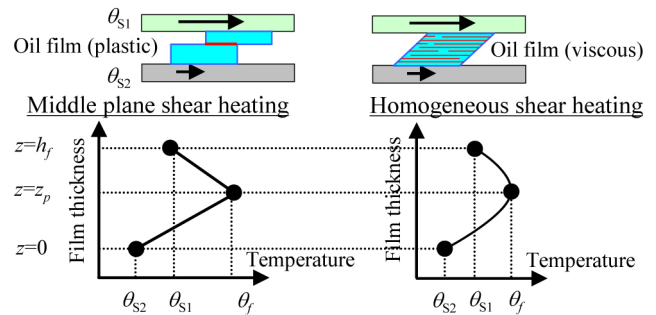


Figure 3. Shear heating types in oil film

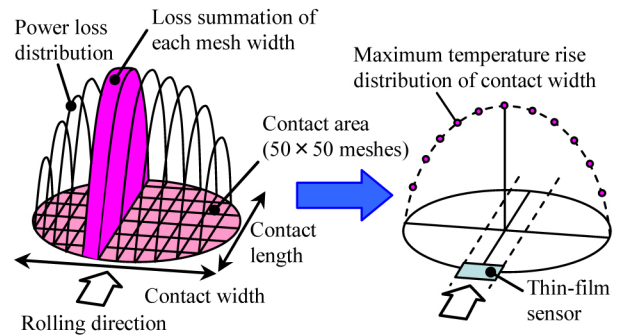


Figure 4. Loss summation method to calculate temperature distribution

The following section describes the temperature distribution calculation.

The rolling contact surface temperature rise $\Delta\theta_S$ in Equation (7) calculates the maximum temperature rise caused by the integrated heating loss within the range of the contact length of the moving heat source (in this paper, this refers to the contact length in the rolling direction). Therefore, it does not calculate the distribution in the contact length direction. To calculate the localized temperature at the passing position of the thin-film sensor (described below), this method integrates the heating loss in each of 50 equal meshes over the width of the ellipse (left of Fig. 4). This allows the temperature distribution in the contact width direction to be calculated (right of Fig. 4).

3. CONTACT AREA TEMPERATURE MEASUREMENT USING THIN FILM SENSOR

The thermal analysis model described above was verified by comparing its measurement accuracy with temperatures directly measured from the traction contact area. These measurements were obtained by thin-film technology using the four-roller experimental apparatus and are reported in Reference [1].

However, unlike the four-roller experimental apparatus, the transmission contact area of a half-toroidal variator has additional heating loss caused by spin motion. Therefore, this