

Real Time Virtual Temperature Sensor for Transmission Clutches

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ABSTRACT

Many experiments have demonstrated that clutch overheating is a major cause of clutch deterioration. Clutch friction material deterioration not only leads to clutch failure, but also causes poor shift quality. Unfortunately, it is not practical to monitor each individual clutch temperature in a production vehicle due to high costs or technical challenges. This paper introduces a proposal for a virtual clutch temperature sensor to monitor the real time clutch temperature changes in Chrysler transmissions with PWM solenoid based control systems. Both vehicle and laboratory dynamometer (dyno) tests demonstrate that the model results match very closely with the thermocouple temperature measurements under many different driving conditions. The real time virtual temperature sensor provides a tool for clutch surface overheat protection and for design improvement and enhancement to shift quality.

INTRODUCTION

The automatic transmission has been using hydraulically applied clutches to shift planetary geartrains since its inception. The power-on shift mechanism is achieved through synchronized opening and closing of the clutches attached to different elements of the planetary systems. The clutch assembly used in automatic transmissions typically consists of multiple separators, friction plates and other components as shown in Fig.1. The friction plate is made of a low carbon steel core plate with friction material bonded on each side. These plates are splined to the input shaft while the separator plates are lugged to the clutch housing. The clutch housing links to either another shaft (rotating clutches) or to the transmission case (brake clutches).

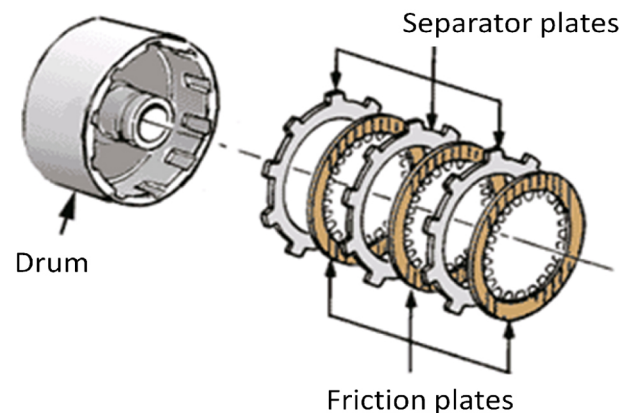


Figure 1. Clutch Assembly

Due to their desirable friction characteristics for shift quality and cost effectiveness, wet paper based friction materials have been primarily used as clutch facings for automatic transmissions in cars and brake materials for industrial vehicles. However, because of the lack of heat and chemical stability under extremely high temperature conditions, the paper-based cellulose fiber often breaks down. This may cause material break out, high friction lining thickness loss and eventually loss of appropriate friction characteristics. Many researchers [1,2,3,4] have concluded that thermal degradation of the friction material due to high temperatures are the major factor for determining the life cycle of wet clutches.

After the friction material is thermally degraded to a certain extent, the friction characteristics will be changed dramatically and performance will suffer. Generally, the least heat resistant composition in the friction material will be degraded first at elevated temperatures. Fig. 2 illustrates a sample weight loss as a function of temperature [4]. The first

rapid decline happens at about 380C and corresponds to the rate of cellulose degradation. The second rapid decrease occurs at about 550C and corresponds to inert contents. The figure indicates that the cellulose fibers start to carbonize near 200C and finish the degradation at about 420 C under the Thermal Gravimetric Analysis (TGA) test condition.

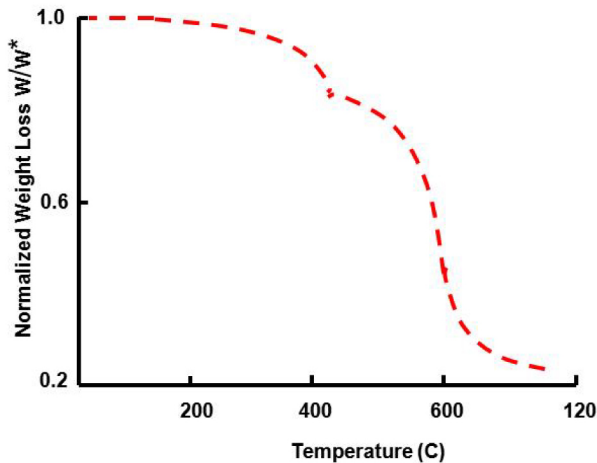


Figure 2. Typical Thermal Gravimetric Analysis weight loss curve of friction material

The damage that occurs in friction systems from the temperatures induced by high-energy shifts is cumulative. Each excursion to high temperatures continues to cause the damage. By reducing the peak interface temperatures and by reducing the time above the activation temperature, the damage to the friction material is reduced and the useful life of the friction system can be increased.

Theoretical and experimental studies from Yubo [4] showed that weight loss for the least temperature resistant component in the friction material studied, cellulose, is in a linear relationship to time and the reaction rate, or

$$\frac{W}{W_o} = 1 - \frac{\theta \cdot t}{W_o} \quad (1)$$

From the theoretical and experimental results, Hiroshi Osanai [1] also introduced the relationship between the maximum clutch surface temperature and critical cycles of engagement as shown in Figs. 3&4.

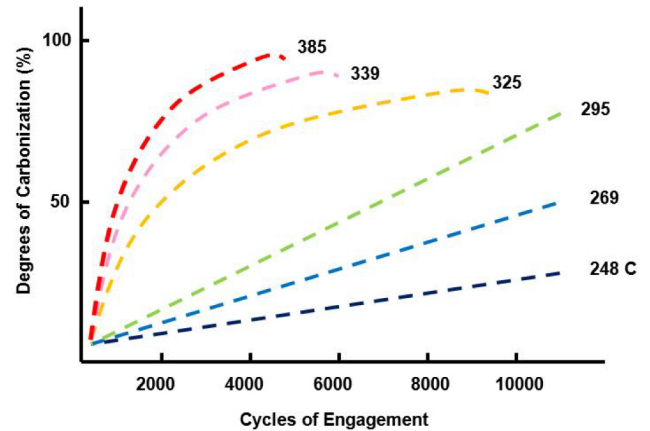


Figure 3. The Change of carbonization degree of repeated engagements under different conditions

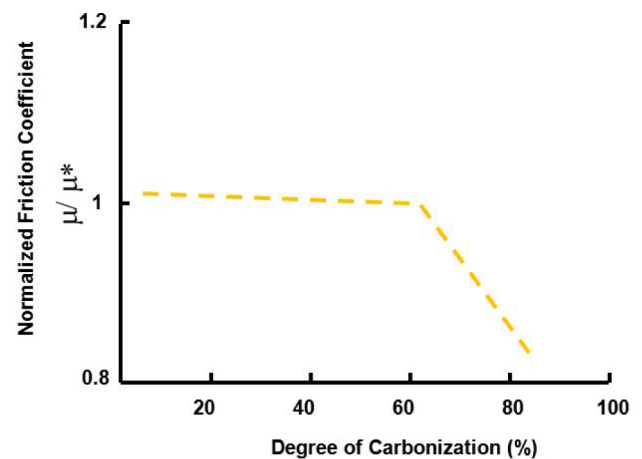


Figure 4. The relationship between dynamic friction coefficient and degree of carbonization

These results provide useful information to protect the clutch from failure or to project clutch material life under variable cycle conditions in any application environment as long as clutch temperature is known.

In addition to causing clutch failure, high temperatures also influences shift quality. It is well known that excellent shift quality depends on accurate clutch torque execution during a shift. Typically, clutch torque is a linear function of clutch friction coefficient, which is a strong function of temperature. The experimental results from literature [5] show the coefficient of friction (COF) is almost a linear function of the interface temperature. The COF could vary 20-30% over a 100 degree of temperature change as shown in Fig. 5. Without temperature compensation, shift quality is very difficult to maintain.

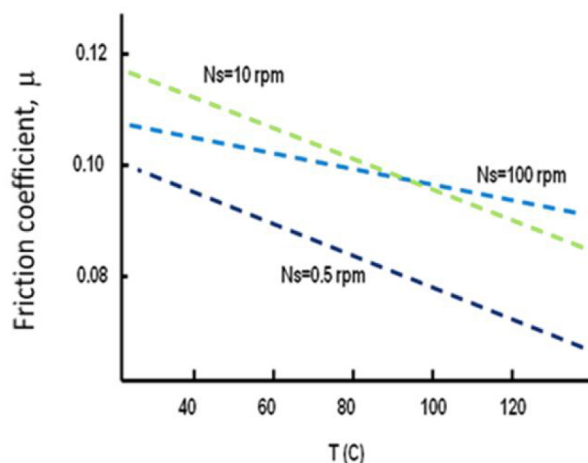


Figure 5. Experimental results and regression curve at applied force equal to 18 KN

During engagement, the separator plates and friction plates are forced together by pressure, and the kinetic energy of relative slip speed is converted to heat at the interfaces. With each shift, the clutch system absorbs energy generated by the slipping of the clutch during the engagement and is then removed from the friction system by the oil and heat conduction through the separator spline teeth. Thus, there are three elements to the friction system energy balance: energy-in, energy-stored, and energy-out.

Energy-in occurs primarily during the engagement of the applying clutch for a power-on upshift and during disengagement of releasing clutch for a power-on downshift. The total energy of the shift is the integrated product of the slip speed and the clutch torque over time. The energy input rate (power) during a shift is typically very high relative to the energy removal rate. As a result, the temperature of separator and friction plates increases during shifts.

Energy is removed from the friction system primarily by the oil that flows past the separators and by conduction to the hub and housings through the spline teeth. While the clutch is not applied, there is cooling oil flowing to the clutch. If the clutch is rotating as an assembly while applied, cooling flow is being centrifugally forced through the grooves in the friction facings, carrying heat away from the hot surfaces. In the case of a grounded clutch, the oil feed is not being centrifugally driven through the grooves and a high percentage of the heat will be removed by conduction through the spline teeth.

The temperature rise at the interface during engagement is determined by the input energy, the heat dissipation rate through the separators and the heat removal rate of the transmission oil. The temperature history near the friction interface is important since it determines the performance characteristics, and especially, the thermal degradation rate of the friction material.

LITERATURE REVIEW

Over the years, extensive research has been conducted trying to gain knowledge of temperature distributions in a clutch disk. Early researches were mostly focused on clutch temperature related influencing factors through experimental investigations [6]. One of the earliest analytical and experimental works was conducted by Newcomb [7]. Expressions were derived to determine the torque capacity of a clutch and rate at which heat is dissipated by friction during a single engagement, together with equations to enable calculation of the temperatures at various depths inside the contacting bodies during the slipping period. However, Newcomb produced analytical solutions for the temperature rise in dry clutches. These solutions are not applicable to wet clutches, which have different physical configurations and boundary conditions.

Pacey took a simple numerical approach with the use of convective heat transfer correlations [8] with varying coefficients for the clutch engagement modes of engaged and slipping, engaged and not slipping, and disengaged. The model allows the prediction of clutch plate and oil temperature for a wide range of operating conditions. The model also allows for the evaluation and analysis of potential design changes such as oil flow rate, plate properties, and oil properties. However, the clutch temperature model basically depends on experimental data regression to obtain convective heat transfer coefficient and many details of heat transfer processes were neglected.

Lai [9] constructed a three-dimensional numerical clutch pack simulation using a finite volume method. This model was consistent with experimental data. Using a groove level of analysis, Payvar numerically determined the convective heat transfer coefficient for the oil in a radial groove and between the groove face and steel separator plate in the continuous slipping mode [10]. This effort quantified a significant oil film thickness between the separator plate and groove face as well as a recirculation zone for the oil with the groove itself.

Yubo [11] developed a more comprehensive model for simulating the engagement of both the traditional wet clutch and single sided wet clutch. It included many important phenomena which were not investigated before, such as non-uniform contacting, compressibility of friction lining, automatic transmission fluid flow into lining due to permeability, etc. The results had good correlation with the experimental measurements.

While sophisticated 3-D models are able to simulate clutch temperature profiles, most of them are offline simulations. These simulations have great use as a design tool to reduce field testing and to optimize existing or proposed designs.

Analysis of the clutch interface temperature is an important step in evaluating the durability of the friction system. By design, clutch thermal durability should not be an issue if the clutch will not be subjected to high energy shifts that are closely spaced in time. However, closely spaced shifts will cause the temperature to increase if the time between shifts is insufficient to return the friction system to sump oil temperature as shown in Fig.6. Even though the energy input for single shift can be calculated from estimated clutch torque and slip speeds, the cooling effect during and after clutch engagement/disengagement is time dependent and very difficult to estimate. Typical linear or non linear look-up tables and predefined constant temperature decay rates can't be used to estimate the real clutch temperature since the clutch temperature is an integrated result of input energy and cooling heat transfer.

Considering the fact that it is very difficult to measure the maximum clutch temperature in production vehicles due to high cost or technical challenges, the practical way to track clutch temperature is the use of a real time clutch temperature model. This paper illustrates a real time clutch model for in-vehicle applications. The model minimized some less critical complicated equations, but includes many important phenomena, such as heat conduction and heat convection due to oil flow both in an open clutch and in grooves, and oil vaporization. The goal is to develop a clutch temperature model with a manageable model size and computational time for real time transmission control applications.

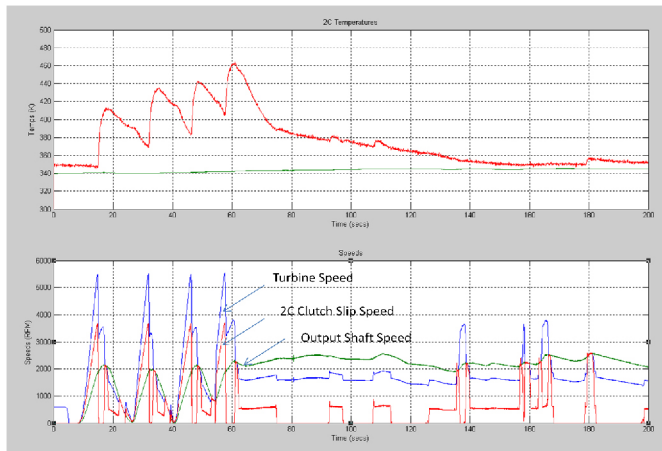


Figure 6. The effect of closely spaced shifts causing clutch temperatures to rise.

THEORETICAL MODEL

An idealized representation of the clutch is shown in Figure 7. Mathematically, the transient conduction for two dimensional heat transfer using cylindrical coordinates is

$$\frac{\partial T}{\partial t} - \frac{k}{\rho C} \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial Z^2} \right] = 0 \quad (2)$$

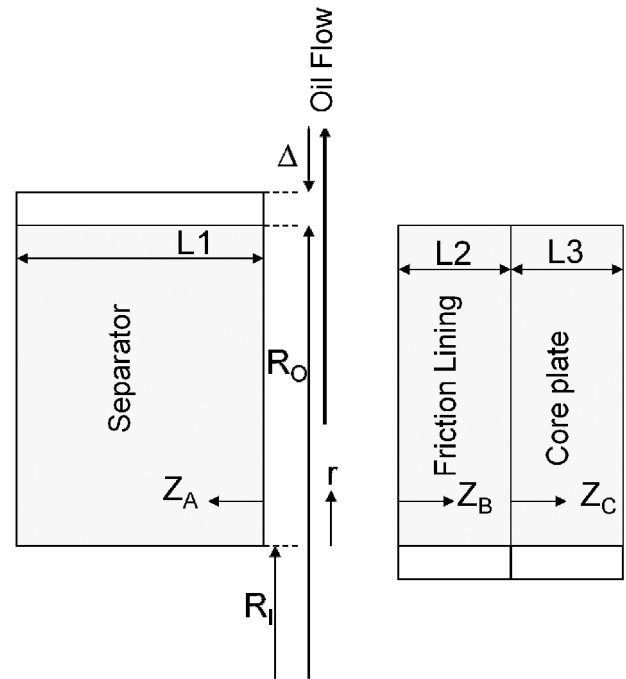


Figure 7. The schematics of separator and friction plates

Based on the results of numerical simulation from Yubo [11], the temperature inside the separator plate is typically close to uniform after one second due to the high conductivity. To simplify the problem for real time application in a transmission controller, a method of lumped-separate plate with two dimensional boundary conditions was applied.

$$2\pi \int_0^{L1} \int_{Ri}^{R0} \frac{\partial T}{\partial t} r dr dZ - \frac{k}{\rho C} 2\pi \left[\int_0^{L1} \int_{Ri}^{R0} \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) r dr dZ + \int_0^{L1} \int_{Ri}^{R0} \frac{\partial^2 T}{\partial Z^2} r dr dZ \right] = 0 \quad (3)$$

Boundary conditions,

$$\text{At } Z_A = L1 \quad \frac{\partial T}{\partial Z} = 0 \quad (4)$$

$$\text{At } Z_A = 0 \quad Q + k_A \frac{\partial T_A}{\partial Z_A} + k_B \frac{\partial T_B}{\partial r_B} - h(T_A - T_{mo}) = 0 \quad (5)$$