MODELING OF FRICTION PERFORMANCE IN CARBON/CARBON BRAKE MATERIALS

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Introduction

The testing of carbon-carbon brake materials is done on various sizes of rotating friction dynamometers. The most direct performance tests are carried out on full sized machines designed for a particular aircraft qualification. Smaller scaled-down tests are conducted, but problems exist in trying to transfer results for any particular material from one test into a prediction for a different test. Traditionally, attempts are made to match some key parameters such as sliding speed, energy input rate, energy input per unit sample mass, etc. Attempts are then made to find empirical correlation factors to relate different sized tests. These attempts have been largely unsuccessful.

This report takes the view that all carbon materials behave essentially the same at the friction interface, and that behavior is best described by a local friction coefficient which is a smoothly varying function of Temperature, Pressure and sliding Speed. In other words, the local and instantaneous values of those variables determine the instantaneous friction coefficient. Then as a particular dynamometer test proceeds, the local conditions change in a manner characteristic to the machine, resulting in an average friction coefficient which depends on the track through those variables, which is machine dependent.

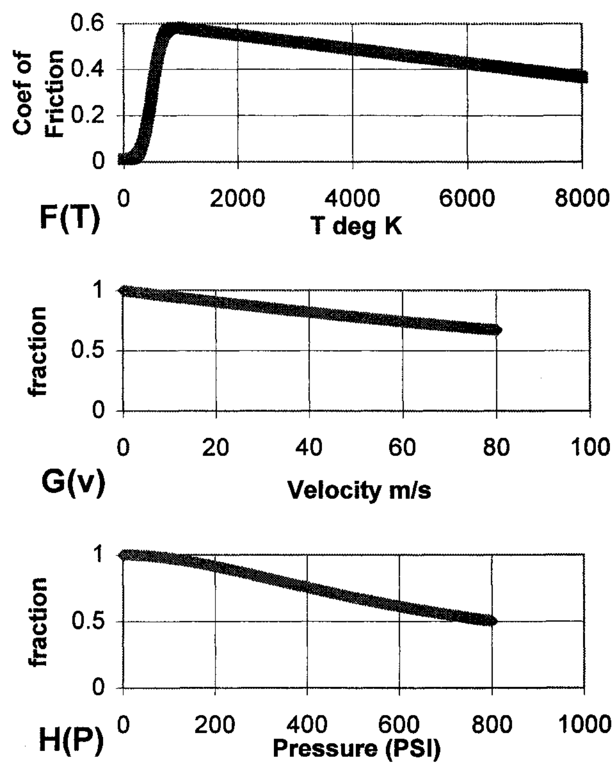
The Model

The friction coefficient is modeled as a function of T, v, and P:

$$\mu(\tau) = \mu_0 * F(T(t)) * G(v(t)) * H(P(t))$$

The friction coefficient model form assumes that the effects due to T, P, and v are separable, and that the variables can be changed independently. This is not always easy in an actual test without some special provisions. For example, if a higher interface pressure is used at a fixed speed, one would expect that a higher surface temperature would develop as a result. This would seem to make the temperature a dependent variable rather than an independent variable. These dependencies between T, P, and v are actually caused by the conditions of the test. In that sense, the test does constrain the variables in a unique way, and that constraint is the key to differences in dynamometer test machine results.

The functional forms for F(T), G(v), and H(P) are not known. We have adopted a model where the functional form is represented as in Figures 1-3.



Figures 1-3. Parametric friction model. One set of model parameters has been chosen resulting in the variations shown in these figures.

These are parametric models for which the slopes and temperature onset values can be altered by a few parameters. The forms are based on a variety of observations of friction materials and on carbon/carbon in particular. Now if the explicit time dependence of each of the three variables T,P,and V can be derived for the test, the friction performance can be evaluated.

The instantaneous velocity of the friction surface can be shown to be given by

$$v(t) = v_0 - \frac{r}{I}\int P(t)A\mu(t)rdt \qquad (1)$$

v(t) = instantaneous velocity
r = friction radius
I = dynamometer inertia
P = friction interface pressure
A = friction area
$\mu(t)$ = instantaneous friction coefficient
t = time

The derivation of surface temperature with time is much more involved. One derivation, which comes from a transient model of an infinite plate with heat input from two sides gives:

$$T(t) = T_0 + \frac{1}{\rho CL}\int q(t')dt' + \frac{2}{\rho CL}\sum_{n=1}^{\infty}\int_0^t q(t')e^{-\frac{\lambda\pi^2 n^2}{\rho CL^2}(t-t')}dt$$

where
ρ = density
C = heat capacity
L=1/2 thickness of disk
λ = thermal conductivity
q(t) = time dependent heat input rate

The heat input rate can further be derived to be:

$$q(t) = \frac{1}{2}P(t)\mu(t)\left[V_0 - \frac{r^2 A}{I}\int_0^t P(t')\mu(t')dt'\right] \qquad (3)$$

The pressure is often a controlled quantity in the test, but if the test is under torque control, the pressure is simply related to the fixed torque through the friction coefficient. These equations give a recursive formula for calculating the temperature, velocity and pressure at any time given a knowledge of the temperature and heat input rate at all previous times, and given the model for the dependence of the friction coefficient on T, P, and V.

These expressions can be evaluated numerically by starting at time zero, choosing a small time increment, and calculating the friction coefficient expected at the next time increment and so on until the flywheel velocity drops to zero.

Results and Discussion

Figures 4 and 5 show two simulated dynamometer tests which show the effect of thermal conductivity on the effective friction coefficient of the test. The difference in the test and the effective friction coefficient is entirely due to the different surface temperature tracks which are followed by the two materials.

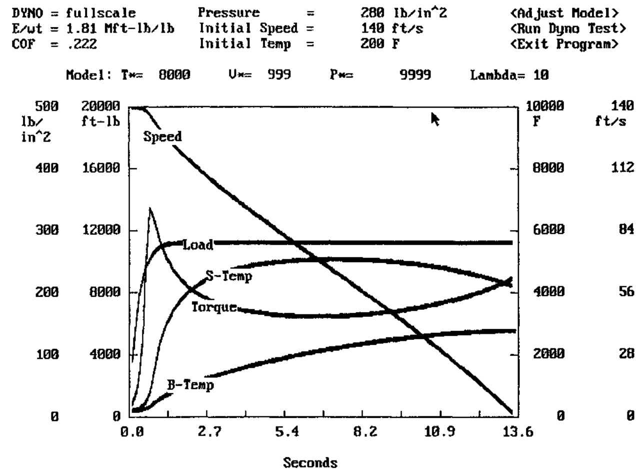


Figure 4. High energy stop for low thermal conductivity.

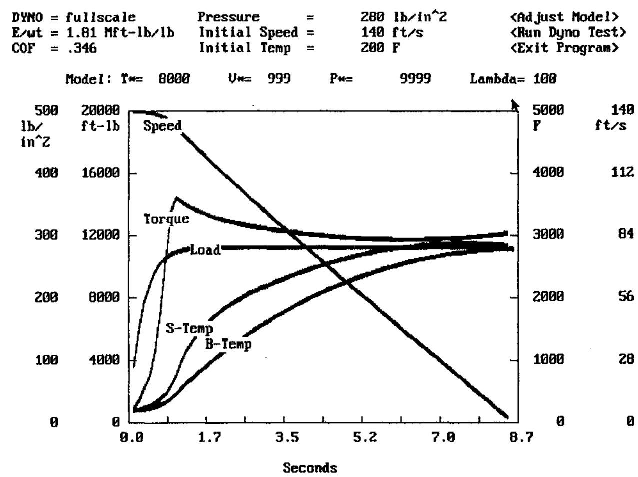


Figure 5. High energy stop for high thermal conductivity.

The significant dip in the torque curve (fade) is apparent for the lower thermal conductivity material. The resulting difference in apparent friction coefficient is due entirely to the different temperature history during the two tests.

Conclusions

A method for predicting different dynamometer test results has been presented based on mapping the varying temperature, velocity, and pressure as constrained by the dynamometer dynamics. Comparisons can be made using this simulation model to highlight expected performance differences between different materials and different tests.