

FRICTION OF CARBONS PART 2: SURFACE ANALYSIS

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Introduction

During any friction and wear process, a significant amount of heat is generated between the two sliding surfaces. Depending on the material properties, sliding velocity, contact nature, rubbing geometry and mass, the heat flux generated on the friction interface will be partitioned among the rotor and stator [1]. Determination of the heat partition factor is extremely complicated [2]. For identical materials of similar geometry, it is commonly considered that the heat is shared equally between the rotor and stator. Further simplification includes total conversion of kinetic energy to heat. In reality, kinetic energy can be converted by means of friction to heat, sound, vibration, chemical energy and wear of materials.

To model the temperature field of the sample, we need to determine how much heat is diffused into the sample. A conventional way to determine this heat partition is by assuming equal division. However, other alternative methods can be considered to approximate this heat more accurately. Since the mass of the stator sample used in the fast test is quite small, has high thermal conductivity and if the back of the sample is thermally insulated, the thermal gradient in the sample is expected to be small. This makes a general lumped capacitance method applicable for preliminary approximation for the heat flux. In general, the thermal energy in the sample can be represented as follows:

$$\dot{E}_{in} = \dot{E}_{out} + \dot{E}_{stored} \quad (a)$$

$$Q(t) = hA(T - T_{amb}) + \sigma \epsilon A(T^4 - T_{sur}^4) + \rho c_p V \frac{dT}{dt} \quad (b)$$

where Q is the heat absorbed by the sample, h is the convection coefficient of the ambient air, A is the contact area, σ is the Stefan-Boltzmann constant, ϵ is emissivity of the sample, T is the temperature recorded by the thermocouple, T_{amb} is the ambient temperature, T_{sur} is the surrounding temperature, and ρ , c_p and V is the density, specific heat and volume of the sample, respectively.

Knowing the thermal transport properties of the sample and the environmental conditions, a heat energy balance can be established. This energy balance will contribute to determine friction heat rate as function of time, $Q(t)$. Thermal properties of the sample such as thermal diffusivity and specific heat were measured experimentally in our laboratory and are illustrated in Figure 1.

Heat losses due to convection and radiation are also included in the estimation. By providing appropriate insulation at the back of the sample holder, conduction heat loss to the sample holder is minimized and therefore can be neglected. In our model, convection coefficient is taken to be 25 W/m²oC from literature [3]. Coke's emissivity is taken as 0.9, as for most carbon materials. The energy stored in the sample can be calculated from temperature history of the sample. In this work, a thermocouple is placed into the sample 2.5 mm from the rubbing surface. Carbon paint is used to provide good thermal contact between the junction and the sample. The response of the thermocouple is recorded using a data logger. Figure 1 shows the temperature history of the sample for the entire 90 minutes test along with its numerical gradient, dT/dt . By substituting all known parameters mentioned before into equation (b), friction heat rate to the sample can be calculated as a function of time. Figure 1 shows the heat flux diffusing into the sample as function of time.

The apparent contact area between the sample and disk observed to change during friction transitions. The degree of change can be estimated by the fingerprints or traces of the wear tracks that were created on the sliding surface. The heat flux tends to increase during the transition. It is also found that only 1% to 30% of the friction energy is converted into heat and absorbed by the sample. Most of the rest of the friction energy is absorbed by the rotor disk.

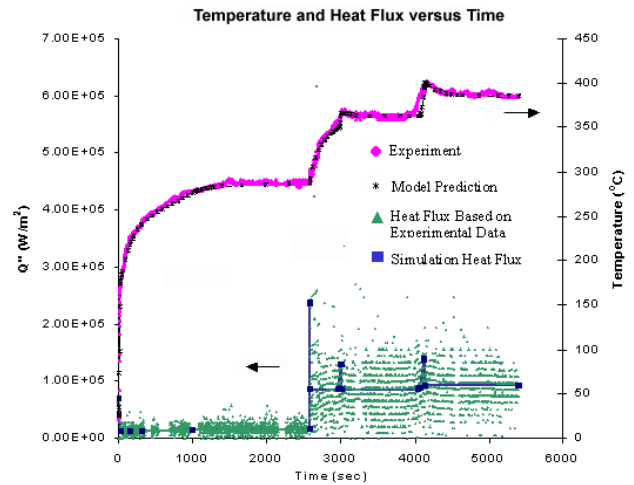
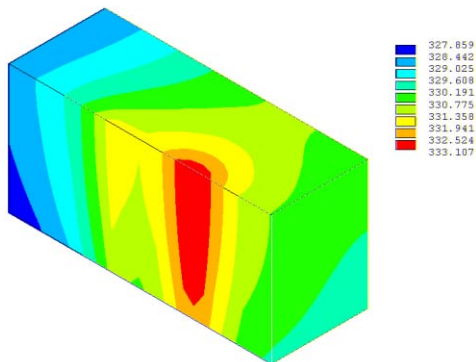


Fig. 1 Simulation heat fluxes and temperature result from modeling.

To simulate the temperature field in the sample, a finite element mesh comprised of brick elements with eight nodes and a single degree of freedom (temperature) was generated. In the model, the four outer faces of the model is subjected to convection. However, the back surface of the model (and the actual sample) is fully insulated. Various heat fluxes were applied to the friction surface. The temperature simulated at the location of the

thermocouple along with the heat fluxes applied to the model is shown in Figure 1.

It is clearly shown that temperature increases very fast in the beginning of friction process and at the initiation of a transition. Both the numerical simulation and experimental results compare favorably. The simulation has been made based on two combined effects: the rate of heat absorption as a function of time and the observed actual contact area calculated from the wear tracks. Figure 2 shows the predicted temperature field of the sample at specific times during the friction test. We notice that during the transition, heat is localised along the wear track due to the observed decrease in contact area, and its impact on the temperature magnitude becomes greater as the test progresses.



Time: 2737 sec (FrictionTransition)

Fig. 2 Temperature field of the sample from the modeling results.

From the stress analysis, it is found that at the beginning of the friction process the stress is fairly low. Higher stresses are located along the back bottom edge of the sample. As the time increases, stresses start to build up mainly due to the increasing normal pressure when friction film is formed. When the first friction transition occurs, stresses at the end of the wear track increase. As the transition progresses, maximum stress increases at the bottom edge of the friction surface. Comparing this value to the strengths of the sample, crack is expected to occur and propagate into the sample from this location. This crack is caused by the higher contact pressure and thermal stresses. Creation of the cracks will vary the nature (mechanical strength and structure) of the sample and causes the prediction of the stress to deviate from the real situation. Figure 3 below shows the cracks created at the end of wear tracks of the samples after friction tests.

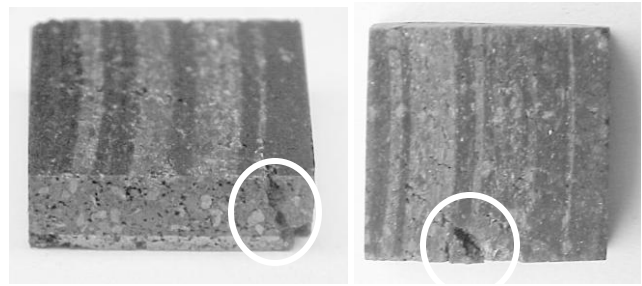


Fig. 3 Cracks generation in the samples of various testing conditions.

Conclusions

In general, we found that the evolution of a friction transition is closely related to the rate of the energy absorbed and the friction film. Stress may develop on the surface and within the bulk material. The friction film can be disrupted by this stress. The disruption of the friction film creates wear tracks, which made up of debris. The shearing of these particulates cause both friction and thermal instabilities. Friction surfaces of the sample were examined using various techniques. A structural change from graphitic to amorphous carbon has occurred on the surface of the specimen. During the run-in process, a friction film is built up. Apart from the water effect (water was desorbed at high power input), the shear stress within the film increases during a friction transition and causes an increase in the friction coefficient. The stresses increase until it disrupts the film and causes a sudden change in friction coefficient. However, the disruption of the film releases the wear particulates and caused a decrease in true contact area. Decrease of contact area keeps the friction coefficient high but decreases gradually as film is being built again. The shear stress may propagate and crack the bulk material.

References

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