EFFECT OF CARBON MATERIALS ON THE THERMAL WEAR OF SLIDING SURFACES

Shadab Shaikh, University of Dayton, Dayton, OH, 45459, USA
Khalid Lafdi, University of Dayton, Dayton, OH, 45459, USA

Abstract

Prediction of surface temperature distribution and controlling the frictional heat causing thermal wear at the interface pose a big challenge in the design of modern tribo-systems. The present work involves a theoretical study on temperature distribution for sliding surfaces and a model for thermally induced wear process. The relations for surface and bulk temperatures for the rubbing bodies are formulated using the moving heat source theory and functional analysis approach for heat partition. The thermal wear model was developed based on the aspect of micro-asperity contact at the sliding interface. The model was used to evaluate the wear performance of carbon composite materials tested experimentally in different literature works. The results for temperature distribution and wear rate showed a good matching with experimental work from literature for different materials. The current study demonstrated the strong influence frictional heat can have on the wear process and its effects on the performance of a tribo-system. The influence of carbon composite materials in minimizing the effects of frictional heat and thermal wear were also emphasized.

Introduction

Rubbing or sliding action between two surfaces is a common friction and wear mode in industry, affecting many mechanical components, such as bearings, brakes, seals, cams, gears, etc. This process is affected by number of factors, including sliding speed, mechanical properties of the two bodies in contact, surface roughness, plastic deformation, temperature, etc. Also friction and wear is inconsistent over the range of energy conditions during sliding operation and is strongly dependent on the thermo-mechanical gradients. It was found by Lee 2001, that very high temperature gradients can be generated for the case of carbon composite materials when large amount of energy is absorbed in small amount of time. The very large thermal expansions when constrained produce large stress gradients, which make failure of the surface layers. Thus it can be said that in case of dry friction process, friction and wear mechanisms depend on sliding solid surfaces, thermo-mechanical material properties, and testing conditions. The test conditions include the temperature, sliding velocity, energy input, pressure, and time.

The *thermal wear* process can thus be treated as a dynamic process, depending on many parameters and the prediction of that process as an initial value problem. The wear rate may then be described by a general equation,

$$\frac{dh}{ds} = f\,(\textit{load, velocity, temperature , thermal properties, material-properties, and friction-coefficient})$$

where h is the wear depth, s is the sliding distance, and the friction coefficient is a function of temperature.

There are extensive reviews and original papers on the thermal analysis of sliding systems in the literature, since the pioneering work of Blok 1937 and Jaeger 1948. Blok and Jaeger assumed that heat source is applied at a single insulated region of the surface and had been acting for a sufficiently long time.

According to Archard's study on sliding wear of steel, the flash temperature in a contact area could reach 750–800 .C, which might induce phase transformations, soften materials, cause oxidation and thus decrease the wear resistance. The heat transfer and temperature distribution in a contact area have been widely studied. Flash temperature model was first proposed by Blok. In the model, the problem was considered as a semi-infinite body subject to a concentrated heat source.

Many others followed Blok's approach to solve different heat transfer problems (Ling 1959, Ling and Ng 1962, Ling and Simkins 1963). However, in realistic sliding processes, the heat generation and distribution are much more complicated. Ling and Pu 1964, introduced a stochastic model to estimate the temperature rise in a sliding contact area where a finite number of small contact spots was located stochastically.

Cameron et al. 1964, investigated the surface temperatures caused by the frictional heat source between two rubbing surfaces in rolling/sliding contact. Barber 1967, investigated the distribution of heat between metals of comparable hardness, sliding past one another. Berry and Barber 1984, examined the division of frictional heat by developing an alternative specimen geometry which permitted the division of heat between sliding solids of various materials experimentally. Francis 1970, developed an analytical solution for the steady-state interfacial temperature distribution within a sliding Hertzian contact. Floquet et al. 1977, considered the case of a dry bearing operating with a plastic liner. They applied the two-dimensional Fourier transform method developed by Ling to calculate the contact temperatures. They also found the partition coefficient between the stationary and the moving elements at the interface to vary along the contact length.

Other researchers have focused on heat partition between moving and stationary heat source under quasi-steady-state conditions based on approximate equations for the maximum and average temperature rise on the rubbing surfaces (Greenwood 1991, Yuen 1988, Blok 1963). Komanduri and Hou 2001, investigated the variable heat partition between the two bodies in sliding contact was considered. They solved the thermal problem of a sliding system using the functional analysis approach originally proposed by Chao and Trigger 1955. Since the heat partition coefficient between two bodies in sliding contact is not a constant but varies along the interface, variable heat partition along the interface was considered using the functional analysis approach. They matched the temperature distribution at all points along the length of contact and determined the functions of the heat partition fractions as well as the temperature rise distributions at the sliding interface and with respect to depth

Thus, all these literature works have concentrated on the analysis of frictional heating, distribution of temperature between contacting bodies, and their surface plus bulk temperatures. However very few studies have concentrated on the thermal effects of this frictional heat directly on the wear rate or as defined before thermal wear of contacting bodies. The first part of this study included the use of surface temperature model for sliding bodies from the work of Hou and Komundari, which was correlated to formulate a thermal wear model based on the aspect of micro-asperity contact at the sliding interface. The second part of this work was concentrated on implementation of the thermal wear model to evaluate the wear performance of graphite based carbon composite materials tested experimentally in literature works.

Recent developments in the field of tribology and thermal managements have focused upon various friction materials which can withstand both the stress and the deleterious effects caused by the generation of heat. Carbon composite materials display several advantageous properties for a wide range of tribological and structural applications, Burchell 1999 and Newman et. al. 1986. This includes low density, good strength retention at high temperatures, high thermal and chemical stability in inert environments, high thermal shock resistance, high resistance to wear and good friction properties at high temperatures. However, there are certain unique attributes of carbon composite materials which make them a suitable candidate for wear applications.

The heat capacity of carbon composites is 2.5 times that of steel and they also posses an excellent coefficient of thermal conductance. These materials are refractory and do not melt at temperatures above 2000 $^{\circ}C$. They show consistent coefficients of friction in dry environment, which little variation with surface temperature. In addition to this they posses light weight which is a key factor in certain applications like aircraft brakes. All these factors have made the carbon composite materials an excellent choice as frictional materials.

Sliding temperature model

Hou and Komundari 2001 developed a mathematical analysis of tribological problems using the classical Jaeger's moving heat source theory. The analytical model presented represented a generic form to predict the surface temperature distribution for sliding contacts. This model could be modified to suit different sliding configurations pin on disk, sliding between to flat bodies, or sleeve on bearing type arrangement.

The heat source in the sliding system was approximated to an infinitely long band heat source, Figure 1. While the time for reaching the quasi-steady-state conditions could be very short for a moving body, it could be relatively long for a stationary body. Thus, it may require a long time to arrive at steady-state conditions for sliding. Consequently, the heat partition fractions for the two bodies may also vary for a long time before reaching steady-state values. It may be noted that the heat partition fractions depend on the relative velocity of the moving body, the interface contact length, and the thermo-physical properties of the two bodies in relative sliding or the Peclet number. If the heat partition fractions are considered uniform along the interface contact length, the resulting temperature distribution at the interface on either side is quite different and it would be difficult to match using Blok's ingenious heat partition principle in which the maximum temperatures are matched (or the subsequent work of Jaeger 1948 where average temperatures are matched) at the interface between the stationary and the moving bodies in sliding contact. As a consequence, the heat partition fractions for the two bodies should be non-uniform and variable heat partition fractions on both sides should be considered.

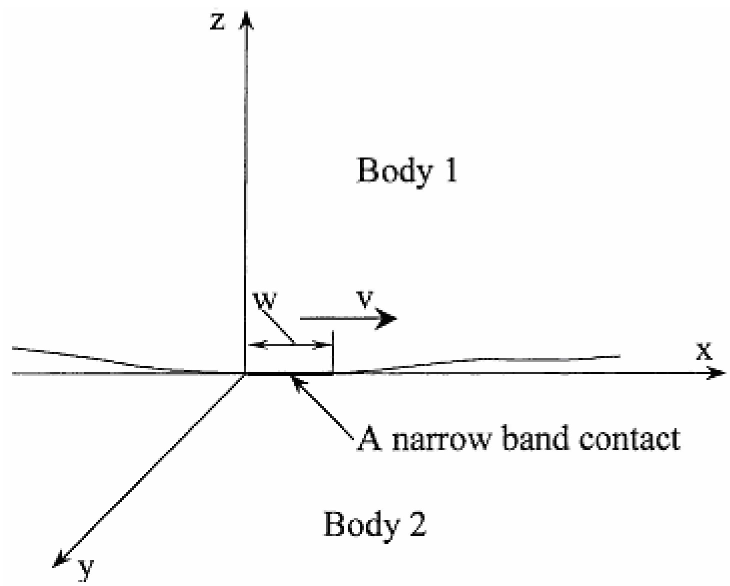

Figure 1. A general tribosystem showing schematic of an infinitely long band heat source formed due to the frictional heating at the interface between semi-infinite bodies

The authors Hou and Komundari 2001 following the work of and Chao and Trigger 1955, considered variable heat partition between the two bodies in sliding contact. The thermal problem of a sliding system was solved using the functional analysis approach originally proposed by Chao and Trigger 1955. The temperature distribution at all points along the contact length as well as the functions that represented the heat partition fractions at the interface were closely matched. The temperature variation at the sliding contact interface as well as with respect to depth was, thus, determined.

The author derived the solutions for average surface temperature distribution for both stationary and moving bodies using the stationary and moving infinitely long band heat source theories.

Stationary body:

$$T_{MS} = \frac{B_{i1} q_{bd}}{\pi\lambda} \int_{x_i=0}^{w} dx_i \int_{u=(X-x_i)/\sqrt{4a_1 t}}^{t} \frac{e^{-u^2}}{u} du \tag{1}$$

Moving body:

$$T_{MM} = \frac{B_{i2} q_1}{\pi\lambda} \int_{x_i=0}^{w} e^{-(X-x_i)v/2a_2} K_0\left[\frac{v}{2a_2}\sqrt{(X-x_i)^2 + z^2}\right] dx_i \tag{2}$$

The function for the non-uniform distribution of heat partition fraction *Bi = f (xi/w)* was determined using the functional analysis approach. A convenient mathematical expression used for this function was a polynomial as it was relatively easy to adjust the parameters for matching the two temperature rise distribution curves at the interface on either side of the contact interface. While three terms would generally be adequate, by increasing the number of terms of the polynomial function, the two temperature rise distribution curves at the interface on both sides were matched to a high degree of accuracy.
The simplest forms of non-uniform distribution were a linear function given by,

$$B_{i1} = (B_1 + \Delta B) - 2\Delta B\left(\frac{x_i}{w}\right)^m \tag{3}$$

$$B_{i2} = (B_2 - \Delta B) + 2\Delta B\left(\frac{x_i}{w}\right)^m \tag{4}$$

The final functional relationship of the local heat partition factors was given by the following forms,

$$B_{i1} = (B_1 + \Delta B) - 2\Delta B\left(\frac{x_i}{w}\right)^m - C\Delta B\left(\frac{x_i}{w}\right)^m \tag{5}$$

$$B_{i2} = (B_2 - \Delta B) + 2\Delta B\left(\frac{x_i}{w}\right)^m + C\Delta B\left(\frac{x_i}{w}\right)^m \tag{6}$$

By substituting equations 5 and 6 in equations 1 and 2 respectively the average surface temperature distribution can be obtained for stationary and moving bodies for certain time duration.

Theoretical model for thermal wear

The first part of this work consists of modeling the wear phenomena based on the concept of frictional heating and energy dissipation, which can be correlated with the surface temperature equations 1 and 2 from literature work. Now consider in general the case of sliding contact between a stationary body and a moving body. A theoretical analysis of this mechanism can be made by assuming a uniform distribution of infinitely hard conical asperities, Booser and Wilcock 1976. The surfaces of the asperities are assumed to be inclined at an angle θ to the plane of surface.

Now as shown in figure-2 below let there be *n* equidistant cones, each of which penetrates to a depth *d* into the stationary body.

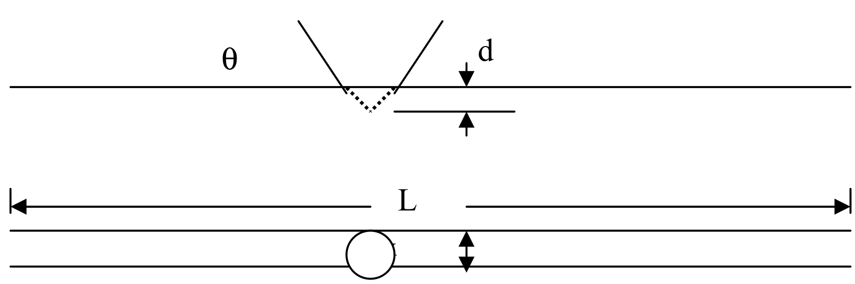

Figure 2. Enlarged view for a contact asperity contact

Let,
A, real projected area of contact
V, Volume of material displaced
L, distance of sliding
Then the projected real area of contact form figure-2 can be,

$$x = \frac{d}{\tan\theta} \tag{7}$$

$$A = n\pi x^2 = \frac{n\pi d^2}{\tan^2\theta} \tag{8}$$

Thus, the volume of material displaced is given by,

$$V = \frac{nLd^2}{\tan\theta} \tag{9}$$

From equation-8 *V* becomes,

$$V = \frac{AL\tan\theta}{\pi} \tag{10}$$

Now since rate of wear is the volume of material displaced during sliding, rate of wear is given by,

$$W = \frac{KAL\tan\theta}{\pi} \tag{12}$$

Now, if it is assumed that during penetration by a conical asperity, the real area of contact, *A*, is determined by plastic flow or the brittle fracture of stationary body. Also considering that the sliding distance is product of sliding velocity *v* and time *t*.

$$A = \frac{\text{ratio of load applied}}{\text{Hardness}} = \frac{P}{H} \tag{13a}$$

$$L = vt \tag{13b}$$

Substituting equations 13a and 13b in equation 12, rate of wear becomes,

$$W = \frac{KPvt\tan\theta}{\pi H} \tag{14}$$

where, the constant *K* may be interpreted physically as proportion of volume of material displaced which appears as lose debris.

Due to the rubbing action between the rotating and stationary body, frictional heat is generated at the interface. It is assumed that the product (*Pv*) which is the energy density parameter acts as weighing factor and can predict the rate of thermal wear occurring due to the frictional heating. Thus, the product *Pv* is taken as function of average surface temperature at the contact surface (equation 1), the equation for rate of wear of the stationary body can be then written as,

$$W = \frac{KPv(T_M)t\tan\theta}{\pi\mu H} \tag{15}$$

It is well known that the heat generated per unit area per second is given by relation,

$$Q = \frac{\mu Pv}{A_n} \tag{16}$$

The friction energy in the contact is dissipated through three processes: rise in temperature; wear particle generation; and entropy changes associated with material transformation at the interface.

For the sake of thermal analysis assuming that out of the total friction heat generated a fraction of heat diffuses into the stationary body, the rest goes into the rotating body. Some of the heat diffuses up the body of the pin the rest is lost by radiation. Considering a energy balance for the heat *Q*,

$$\frac{\mu Pv}{A_n} = c_p(T_{MS} - T_o) + \varepsilon(T_{MS}^4 - T_o^4) \tag{17}$$

Therefore,

$$Pv = \frac{A_n}{\mu}\left(c_p(T_{MS} - T_o) + \varepsilon(T_{MS}^4 - T_o^4)\right) \tag{18}$$

where T_o is the temperature of the heat sink at the other end of the stationary body. Thus by substituting the values of average surface temperature at different time intervals from equation 1, the temperature dependent values for the energy density parameter can be calculated. Finally, by substituting equation 18 into equation 15 the wear rate can be predicted for the sliding bodies.

The coefficient of friction in equation 18 is considered as a constant, however it can be considered as a variable parameter depending on the type of material tested and the test conditions.

Thermal wear in epoxy/exfoliated graphite composites

Literature work on wear performance of exfoliated graphite composites

There have been several attempts to use graphite flakes for a solid lubricant material within polymer composites. Li et al. 2004, studied epoxy and graphite composites on a ring-block type test rig. Samples were fabricated from 0-45% graphite by volume and tested against a steel counter surface at 50N load and 200 rpm. The results showed that the friction coefficient can drop from 0.48 to 0.25 as graphite content increases from 0 to 30 vol. %, but further addition of graphite will have little effect on modifying the friction coefficient. Xian and Zhang 2005, fabricated polyetherimide/graphite composites with graphite content varying from 0-20% by volume. In this particular study, wear measurements were performed by a pin-on-disc apparatus. The results showed a continual decrease in the wear rate with increasing graphite content. The wear rate decreases nearly two orders of magnitude from the baseline pure resin to its lowest value for 20 vol. % graphite. Zhang et al. 1999, observed the friction and wear behavior of polytetrafluoroethylene (PTFE)/graphite composites with constant graphite content (30 % by volume) while varying the load from 100-400 N. The friction and wear tests were performed with a ring-block wear tester using a steel ring. The results showed that the graphite reduces the friction coefficient of the PTFE composite, and the friction coefficient decreases with increasing load. The wear of the PTFE composites were shown to decrease by two orders of magnitude compared to that of pure PTFE. This study also showed that the wear of the graphite-filled PTFE composites increases with the increase of load.

Recently, Debelak and Lafdi 2007, studied the tribological properties of epoxy/exfoliated graphite composites. Exfoliated graphite (EXG) is an expanded form of graphite and a composite filler material prepared by first intercalating crystalline graphite flakes (about 0.4 mm) with bromine or sulphuric acid, followed by heating which causes the composite to expand from 20 to 200 times, changing from a flaky to a low density (0.003-0.03 g/cm^3), long wormlike material.

Debelak and Lafdi 2007 showed that exfoliated graphite has a tremendous potential as a filler material to create solid lubricant carbon nanocomposites. They prepared epoxy/exfoliated graphite nanocomposites by loading three sizes of exfoliated graphite flakes with different volume fractions into epoxy resin. The thermal conductivity of the composite material for different sizes of exfoliated graphite flakes was measured as shown in figure 3 below.

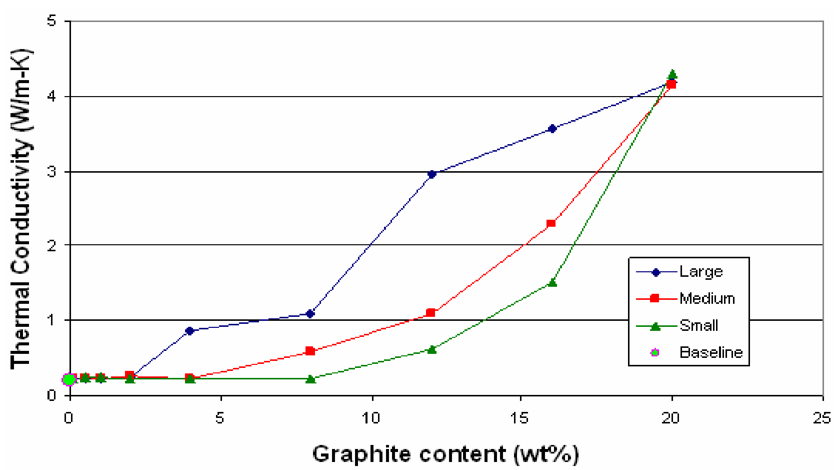

Figure 3. Thermal conductivity of exfoliated graphite filled polymers with different graphite flake sizes as a function of the graphite content.

As shown in Figure 3 considerable enhancement was observed in the thermal conductivity values of the epoxy/exfoliated composites as compared to the pure epoxy resin. The nanocomposites for the three types of graphite flakes with volume fraction between 0-12 % were tested for their wear performance on a pin-on-disc climate tribometer. The exfoliated graphite filled polymer nanocomposites were chosen as the material for the pin, and stainless steel was selected for the counter surface, or disk.

Debelak and Lafdi observed that the increase of the EXG concentration from 0 to 12% led to a significant decrease of the wear rate among the three sets of samples. For each of the three sets, the addition of graphite resulted in a nearly linear decrease of the wear rate. The tribological properties of these materials not only depend on the EXG concentration but also on their morphology and interfacial characteristics. The results indicated that the large graphite flake filled polymers provided the most wear resistance followed by the medium flake graphite. From these results they concluded that the higher aspect ratio of the large flake graphite directly contributed to greater wear resistance. The other important factors are the low thermal expansion and high thermal conductivity achieved by the higher aspect ratio particles. The greatest concentration of graphite particles coupled with the highest aspect ratio particles produced the most wear resistant material.

Comparative study for thermal wear model

The thermal wear model formulated in equation 17 was used to predict the thermal wear in the three types of epoxy/exfoliated graphite composites analyzed by Debelak and Lafdi. The dimensions of the composite samples, their thermal properties, the rubbing time, the normal load and other tribo-parameters for the rubbing process were used from the work of Debelak and Lafdi. Equations 1 and 2 were first used to obtain the average surface temperature for the different samples (Figure 4). It can be observed from figure 4 that a significant reduction in surface temperature was obtained with the inclusion of high thermal conductivity EXG with different sized flakes. The large EXG flakes due to its high aspect ratio and surface area created the biggest drop in surface temperature.

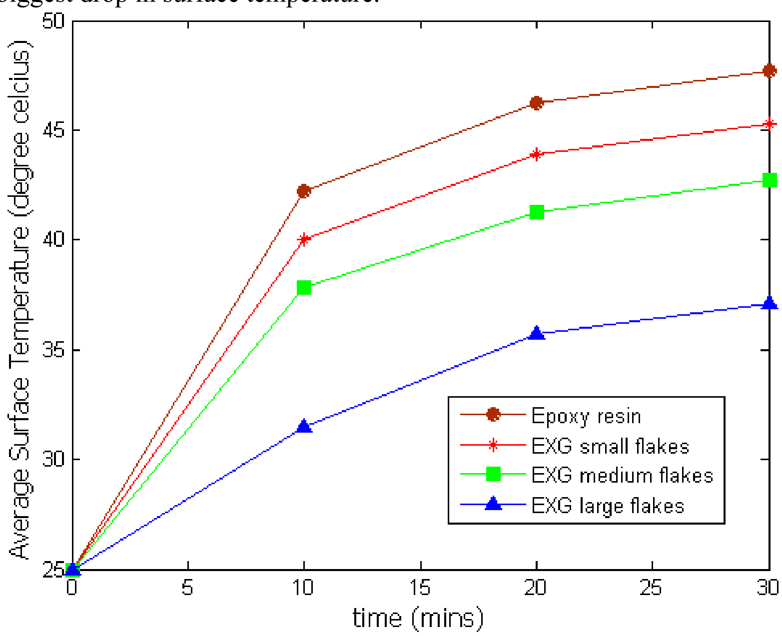

Figure 4. Predicted values for average surface temperature

The surface temperature values were then used in equation 18 to obtain the energy density parameter for all the samples. The value for constant K was obtained by measuring the volume of material which was collected as loose debris during sliding; the hardness H and slope θ values were experimentally measured. Finally the energy density values were used in equations 15 to obtain the wear rate. Figure 5 gives the normalized wear rate values for the three types of exfoliated graphite polymer composites predicted by the thermal wear model as compared with the experimental results from literature. As observed from the figure 5 a good agreement was found between the theoretically estimated and experimental values.

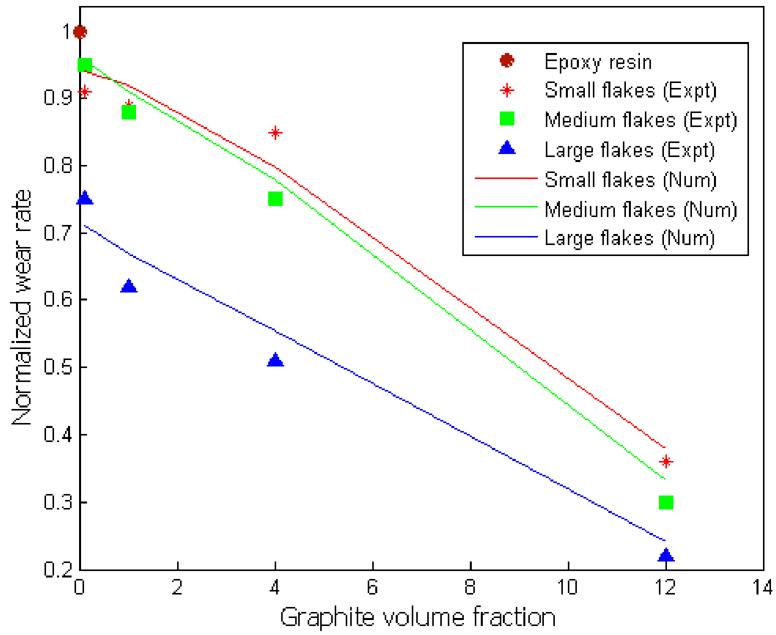

Figure 5. Comparison of thermal wear rate with literature

The results predicted by the thermal wear model emphasize the huge influence of frictional heating on wear rate and the role played by the high thermal conductivity, low thermal expansion exfoliated graphite material in minimizing this thermal wear. Nearly all of the frictional energy produced from sliding friction is dissipated in the form of heat. The heat dissipation is a cumulative and continuous process; therefore temperature gradients are formed within the sliding bodies, with the highest temperatures attained at the contact surface. The overall temperature rise in the bodies thus depends upon the rate of input of frictional energy, the thermal properties of the contacting bodies, and the sliding speed. When the temperature of a sliding body is increased, several effects may occur: its mechanical properties will change, its rate of oxidation may increase, and phase transformations may take place. Having a lower thermal expansion material embedded within the matrix, which can dissipate the heat away from the surface can reduce the destructive effects caused by frictional sliding and thus minimize the wear rate.

References

Archard J. F. 1959. The temperature of rubbing surfaces, *Wear* 2: 438–455.

Barber J. R. 1967. Distribution of heat between sliding surfaces, *Journal of Mechanical Engineering Science*. 9: 351–354.

Berry G. A, Barber J. R. 1984. The division of frictional heat—a guide to the nature of sliding contact, *Trans. ASME Journal of Tribology* 106: 405–415.

Booser and Wilcock. 1976. Assessment of carbon graphite wear properties.

Blok H. 1937. Theoretical study of temperature rise at surfaces of actual contact under oiliness lubricating conditions, in: Proceedings of the General Discussion on Lubrication and Lubricants, Institute of Mechanical Engineers, London 2: 222–235.

Burchell T. D. 1999. Carbon materials for advanced technologies (Pergamon).

Chao B. T, Trigger K. J. 1955. Temperature distribution at the chip–tool interface in metal-cutting, *Transaction of ASME* 77: 1107–1121.

Debelak B. and Lafdi K. 2007. Physical and tribological properties of exfoliated graphite based nanocomposites. Submitted to *Polymer Journal.*

Floquet A, Play D, Godet M. 1977. Surface temperatures in distributed contacts application to bearing design, *Transaction ASME Journal of Lubrication Technology* 99: 277–283.

Francis H. A. 1970. Interfacial temperature distribution within a sliding Hertzian contact, *ASLE Transaction.* 14: 41–54.

Greenwood J. A. 1991. An interpolation formula for flash temperatures, *Wear* 150: 153–158.

Hou Z. B, Komanduri R. 2001. Analysis of heat partition and temperature distribution in sliding systems. *Wear* 251: 925–938.

Jaeger J. C. 1942. Moving sources of heat and the temperature at sliding contacts, Proc. Royal Soc. NSW 76: 203–224.

Lee et. al. 2001. Experimental and numerical analysis of friction and wear behavior of polycarbonate. *Wear* 54: 383-390.

Ling F. F. 1959. A quasi-iterative method for computing interface temperature distributions, *Zeitschrift fur Angewandte Mathematik* 10: 461–474.

Ling F. F, Ng C. W. 1962. On temperatures at the interfaces of bodies in sliding contact, *Proceedings of the 4th US National Congress of Applied Mechanics, ASME, New York, NY*, 4: 1343–1349.

Ling F. F, Simkins T. F. 1963. Measurement of pointwise junction condition of temperatures at the interface of two bodies in sliding contact, *Transaction of ASME Journal of Basic Engineering* 85: 481–487.

Ling F. F, Pu S. 1964. Probable interface temperatures of solids in sliding contact, *Wear* 7: 23–34.

Li X, Gao Y, Xing J, Wang Y, and Fang L. 2994. Wear reduction mechanism of graphite and MoS_2 in epoxy composites *Wear*. 257: 279-283.

Xian G and Zhang Z. 2005. Sliding wear of polyetherimide matrix composites: II. Influence of graphite flakes. *Wear* 2005; 258: 783-788.

Yuen W. Y. D. 1988. Heat conduction in sliding solids, International Journal of Heat and Mass Transfer 31: 637–646.

Zhang Z. Z, Xue Q. J, Liu W. M, and Shen W. C. 1999. Friction and wear characteristics of metal sulfides and graphite-filled PTFE composites under dry and oil-lubricated conditions. Journal of Applied Polymer Science 72: 751-761.