

Wear and Thermal Effects in Low Modulus Polymer-Based Composite Friction Materials

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ABSTRACT: The wear properties of low modulus polymer-based friction materials were studied. The wear equation $W = K P^a V^b t^c$ was used to correlate the wear of polymer-based friction material sliding against cast iron with the wear coefficient (K), load (P), speed (V), and time (t). The parameters were determined experimentally by varying only one variable at a time and keeping the other

two variables constant. The wear rate of selected polymer-based friction material was compared with cast iron friction material. © 2005 Wiley Periodicals, Inc. *J Appl Polym Sci* 95: 1181–1188, 2005

Key words: friction material; polymer; wear; thermal

INTRODUCTION

When a vehicle is stopped, the kinetic energy of the moving vehicle is converted into heat by the brake, and it is then gradually dissipated into the atmosphere.¹ The last section of a brake system is a sliding friction couple consisting of a rotor connected to the wheel and a stator on which the friction material (pad, lining, or block) is mounted. The main disadvantages of cast iron friction materials are poor frictional performance in dry conditions, high tread wear, the problems associated with wear debris, thermal damage on counterface, noisy braking, and higher weight.¹ Because of the mentioned problems, many efforts have been made to replace this class of friction materials with polymer-based materials.

The main components of a polymer-based friction material are the polymer matrix, fibrous filler, particulate fillers of metal or mineral, and solid lubricants.² Fillers are used in the friction compounds as an auxiliary material to achieve the predetermined friction coefficient (μ), wear, and thermal properties. The compound used as friction material not only must have the specified friction coefficient (μ) and wear properties for the particular application, but also has to meet a number of other requirements such as resistance towards thermal cracking, which is the result of thermal accumulation and low wearing effects upon the opposing surface.^{2,3}

There are four possible mechanisms of wear at sliding surfaces:¹ (1) Adhesion of the two materials, followed by cohesion failure of one as the two slide past each other. (2) Ploughing or gouging of material by a fragment of another (harder) material. (3) Thermal or mechanical fatigue or melting, which permits solid pieces to become detached from the surface. (4) Oxidation and pyrolysis, leading to gas formation. Each of these mechanisms can reasonably be expected to be dependent on the conditions of sliding, that is, load, speed, and time. If the system does not reach a steady state, but goes through a transient state such as rising temperature due to friction, the relative contribution to the total wear of the individual wear mechanisms may vary with temperature.⁴

The wear, W , of friction materials can be described by the following wear equation:^{5,6}

$$W = K P^a V^b t^c \quad (1)$$

where K is the wear coefficient, P the normal load, V the sliding speed, t sliding time, and a , b , and c are a set of parameters for a given friction material-rotor pair at a given temperature. The objective of the present work is to study the wear properties of low modulus polymeric friction materials and to compare the wear rate of these materials with cast iron friction materials.

EXPERIMENTAL

Materials

The materials used in this work were SBR 1502 (BIPC Co., IRAN), phenolic resin (Novelac IP502) (Resitan

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TABLE I
The Composition of Polymeric Friction Materials

Components	Sample 1	Sample 2	Sample 3	Sample 4
SBR 1502	13	18	22	24
Phenolic resin	3	3	2.5	2.5
Iron powder	5	4.5	4.5	4.5
Steel wool	4	4	3.5	3.5
Calcium carbonate	15	14	13.5	13
Barium sulfate	25	23.5	22.5	22
Carbon black	11	10.5	10	9.5
Coke	7	6.5	6	6
Iron oxide	15	14	13.5	13
Sulfur	1	1	1	1
Vulcanization system	1	1	1	1

Co., IRAN), iron powder (Iran Metallurgy Powder Co., IRAN), calcium carbonate and barium sulfate (Iran Baryte Co., IRAN), carbon black N330 (Pars Carbon Black Co, IRAN), zinc oxide (Pars Neco Co., IRAN), MBTS (Bayer AG, Germany), and graphite (Cen Enterprise Co., China). A typical formulation, based on 13 weight percent of rubber (sample 1), is shown in Table I. In samples 2, 3, and 4, the percentage of rubber part were, respectively, 18, 22, and 24%, but the percentages of other components were kept constant.

To prepare the required test samples (100mm × 100mm × 30mm), all materials were compounded in a Banbury mixer. The mixture obtained from Banbury was ground in a high-speed grinder to form a powdery compound. The prepared compound was compression molded at 145°C and pressure of 100 kg/cm² for 20 min. The molded samples were post-cured at 180°C for 12 h.

Methods

Compression modulus measurement

The compression modulus of the test specimens was measured using a Rockwell hardness machine on which the ball was replaced by a cylindrical mandrel 13.3 mm in diameter. The minimum load was 10 kgf, and the maximum (total) load was 35 kgf. Before applying the load, the deflection of the test machine between minimum and maximum loads is first recorded and then the test piece is loaded sequentially. The minimum load is applied, and the dial is reset to zero (black graduated scale), with the test specimen placed in a central piston under the mandrel. The maximum load is then applied for 45 s, followed by a 10 s application of the minimum load. The dial is reset to zero, then the maximum load is applied again. The reading is then taken at the point when, after about

10 s, the needle shows a sudden deceleration. The deflection of the machine is then subtracted from this reading, and the net deflection expressed in graduations on the scale is multiplied by 2 to obtain the deflection Δh of the test specimen in μm . The modulus E is then calculated as the average of six measurements according to the following equation:⁷

$$E = \frac{(35 - 10) \times 9.8066}{\pi \times \frac{d^2}{4}} \times \frac{h}{\Delta h} \frac{N}{\text{mm}^2}$$

$$E = \frac{3.122 \times 10^5 \times h}{d^2 \times \Delta h} \frac{N}{\text{mm}^2} \quad (2)$$

where h is the height in mm, d is the diameter in mm, and Δh is the deflection of the test specimen in μm .

Hardness measurement

The hardness of the samples was tested according to method A in ASTM D 782-62 in which the minimum and maximum loads were, respectively, 10 and 60 kg.

Thermal conductivity measurement

The schematic of the apparatus used for thermal conductivity (λ) measurement is shown in Figure 1. The temperature reading was taken in the steady state, when the temperature variation in each of the copper blocks was not more than 0.1°C over a period of 10 min. The test samples were blocks (50mm × 50mm × 10mm) that were prepared from composite brake pads. During testing, ambient and sample temperatures were kept about 20 and 100°C, respectively. Thermal conductivity λ is then calculated from the equation:

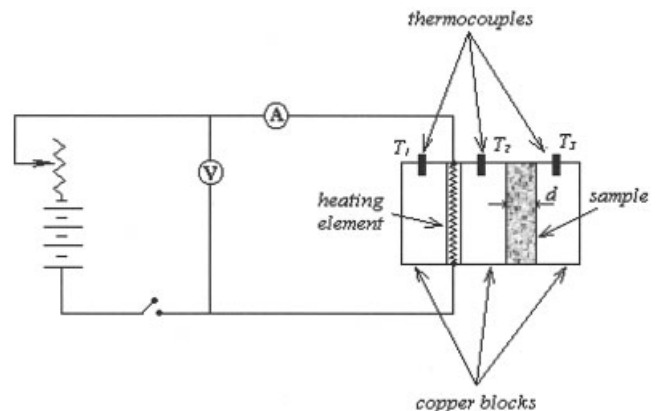


Figure 1 Schematic of the apparatus used for thermal conductivity measurement.

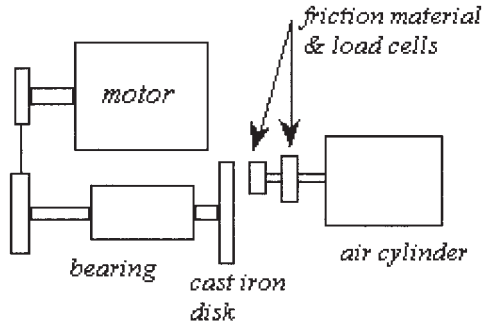


Figure 2 Schematic of the apparatus used for small specimen testing.

$$W = \frac{Q}{t}$$

$$J = \frac{W}{A} \quad \lambda = \frac{J}{\frac{\Delta T}{d}} \quad (3)$$

where Q is the heat in $W \cdot s$ or J , t is the time in s , W is the heat per time unit in W or J/s , A is the surface area in m^2 , J is the heat flux in W/m^2 , ΔT is the difference of temperature between the two sides of the sample in $^{\circ}C$, d is the thickness of the sample in m , and λ is the thermal conductivity in $W/m^{\circ}C$ or $J/ms^{\circ}C$.

Wear test

The schematic of the used dynamometer is given in Figure 2. This apparatus has been developed such that a small specimen of friction material could be brought in contact with a rotating cast iron disk, the speed of which is controlled by a motor.

A small friction specimen is pushed to the face of the rotating disk by a pneumatic cylinder. The normal force is measured with a load cell, and the friction force is evaluated via a calibrated strain gauge bridge mounted on the load arm. The temperature of the friction specimen was measured with a thermocouple embedded into the back of the specimen (included

TABLE II
The Values of Compression Modulus, Hardness, and Conductivity of Different Composite Materials*

Specimen	Compression modulus (N/mm ²)	Hardness (Rockwell X)	Conductivity (kJ/m-h- ^o C)
1	2302.9 ± 428.9	59.8 ± 0.45	3.75 ± 0.07
2	980 ± 61.3	38.14 ± 2.03	3.62 ± 0.05
3	493.4 ± 22.7	18.8 ± 2.6	3.44 ± 0.08
4	336.8 ± 18.87	4.8 ± 2.39	3.50 ± 0.10

* The compound was prepared from composition of Table I.

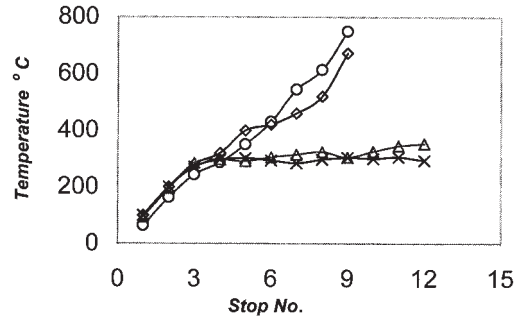


Figure 3 Maximum disc temperatures reached by samples (O) 1, (□) 2, (Δ) 3, and (×) 4 in Table II during stopping on a dynamometer from 700 rpm. Brake pressure = 10 kg cm⁻², braking time = 10 s.

back plate). Friction coefficient (μ) and wear of composite and metallic friction materials were tested with the dynamometer. The conditions were as follows: (a) rolling radius, 28.2 cm; (b) brake force, 70N. The samples were first cut to 1in × 1in × 1/4in. The rubbery surface was ground to a radius of curvature equal to that of the brake drum friction surface. Rotor temperature and friction coefficient were recorded during each stop. Sample weights were measured before the test and after each 20-stop interval. The wear rate of the samples was calculated according to the following equation:

$$\text{Wear rate}(\%) = \frac{w_1 - w_2}{w_1} \times 100 \quad (4)$$

where w_1 and w_2 are the sample weights before and after each test.

Scanning electron microscopy (SEM)

Scanning electron microscopy of the composite materials was performed on gold-coated samples using a

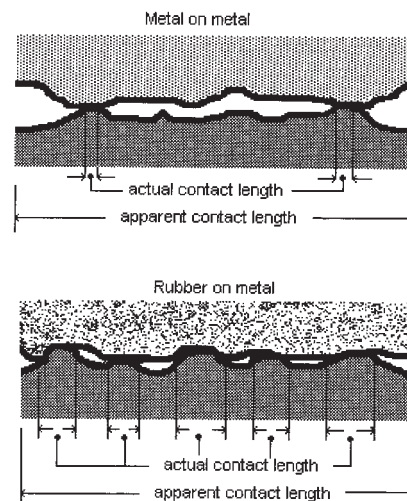


Figure 4 Contact between two surfaces.

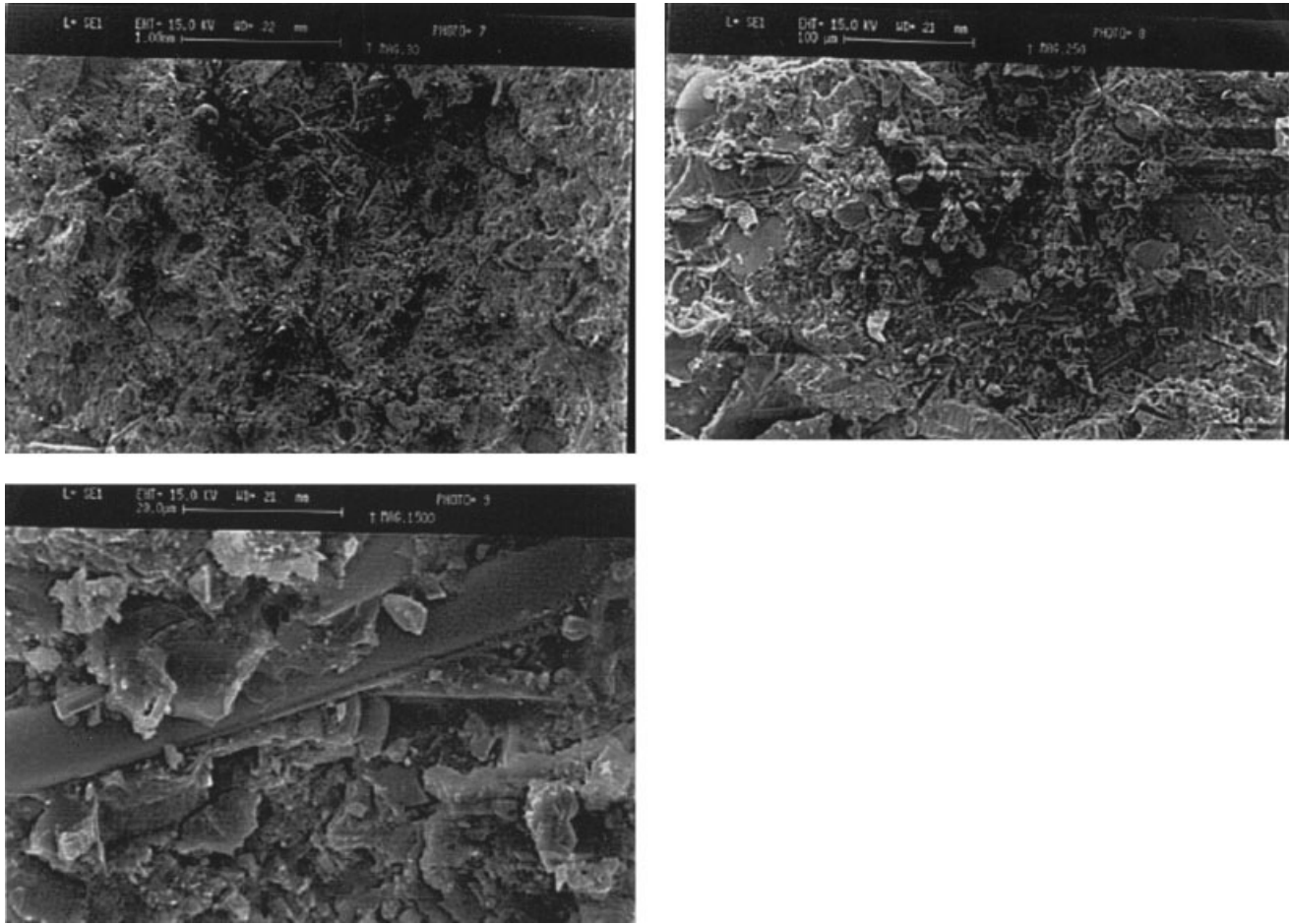


Figure 5 SEM micrograph of sample 3: (a) $\times 30$ magnification, (b) $\times 250$ magnification, and (c) $\times 1500$ magnification.

Polaron sputter coater. A Cambridge S-360 SEM operating at 10 kV was employed for morphological studies of the samples.

RESULTS AND DISCUSSION

The physical and mechanical properties of different compounds prepared based on the formulations of

Table I are given in Table II. The best criterion to choose the material for the wear test is the maximum temperature reached by the friction material during contact with the disc. It has been shown that the controlling factor of this maximum temperature is the ratio of conductivity to modulus of the friction material.⁸

These samples, then, underwent thermal tests to determine the maximum reached temperature. The

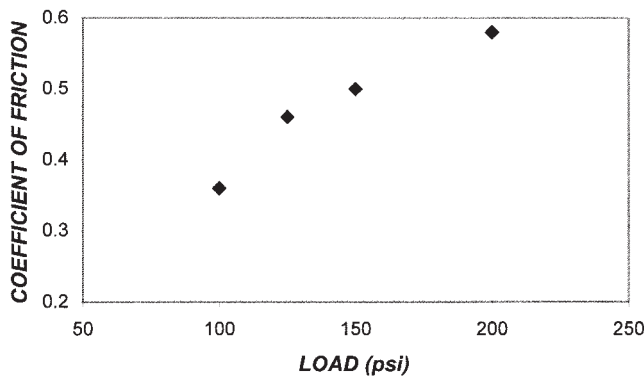


Figure 6 Variation of coefficient of friction against normal load ($V = 400$ rpm, $t = 12$ min, $T = 150^\circ\text{C}$).

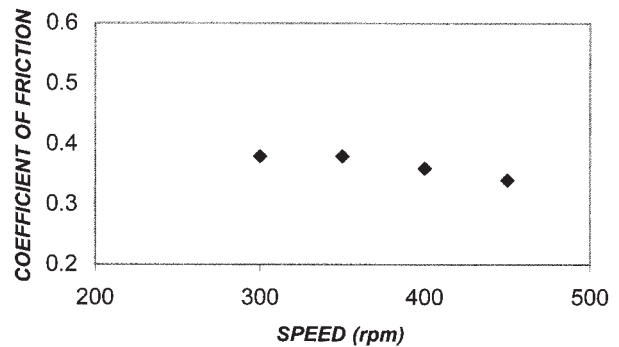


Figure 7 Variation of coefficient of friction against normal speed ($P = 150$ psi, $t = 12$ min, $T = 150^\circ\text{C}$).

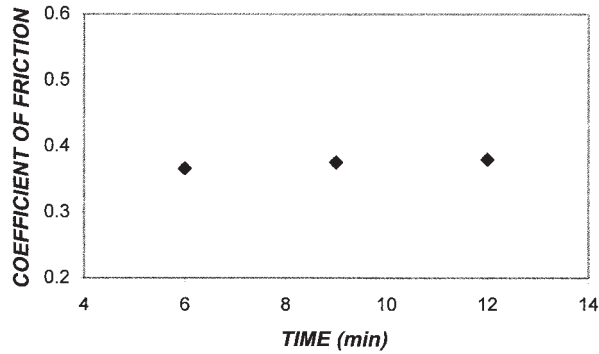


Figure 8 Variation of coefficient of friction against braking time ($P = 150$ psi, $V = 400$ rpm, $T = 150^\circ\text{C}$).

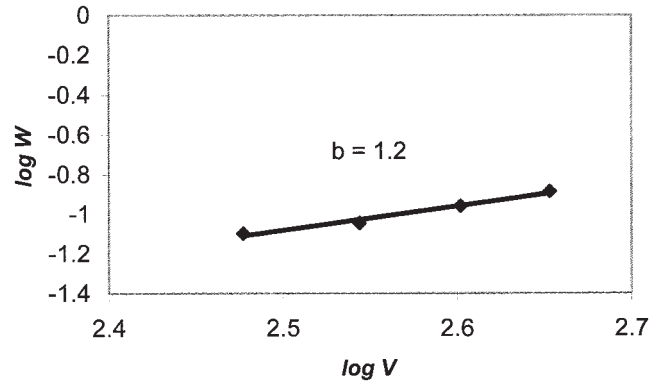


Figure 10 Dependence of wear on the speed ($P = 150$ psi, $t = 12$ min, $T = 150^\circ\text{C}$).

maximum temperatures are, respectively, shown in Figure 3. The obtained results clearly show that to avoid creating high temperatures on the sliding surfaces, the modulus of the friction materials must be kept at a minimum level.

To avoid high local temperature, one must attempt to design a friction material so that the frictional heat generated is uniformly distributed over its working area. The real contact area between the friction material and the counterface is much less than the nominal area of contact. The friction material will deform elastically and conform to some degree with the contacting surface of the disc. The less rigid the material of the pad, the greater will be the degree of conformity, as illustrated in Figure 4.⁹

Consequently, to keep the temperature at a low level, the contact area should be relatively large (the material needs to be soft) and the locus of the contact area has to traverse the whole area of the friction material once per each revolution of the disc. This dictates a low modulus for the material and conformability to the opposing surface. This is because the frictional heat could be dissipated mechanically, and the temperature of the contact area cannot increase more than a definite level.

Figure 5 shows the SEM micrographs of composite friction material number 3, in which the increase in the contact temperature is prevented by low elastic modulus of the friction material.

Coefficient of friction in isothermal tests

Figures 6 to 8 give the curves of the coefficient of friction plotted against load, speed, and time for sample 3 in Table II. The results are the average of three tests.

Contact between a polymer and a metal is often predominantly elastic. In this important respect, the friction of polymers differs fundamentally from that of metals. The ratio E/H , where E is the Young's modulus and H is the hardness of the material, determines the extent of plasticity in the contact region. For metals, the value E/H is typically 100 or greater, while for many of the softer (i.e., low modulus) polymers E/H is only about 10. The plasticity index for a soft polymer thus has only about one tenth of the value for a metal, and the contact is therefore almost completely elastic, except against very rough surfaces.¹⁰

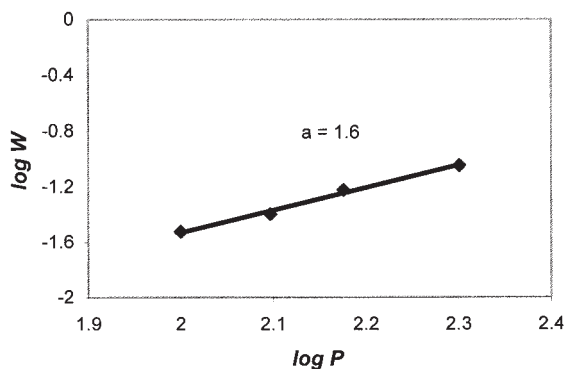


Figure 9 Dependence of wear on the load ($V = 400$ rpm, $t = 12$ min, $T = 150^\circ\text{C}$).

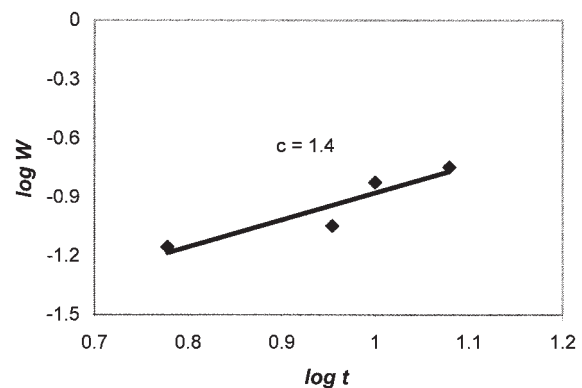


Figure 11 Dependence of wear on the time ($P = 150$ psi, $V = 400$ rpm, $T = 150^\circ\text{C}$).

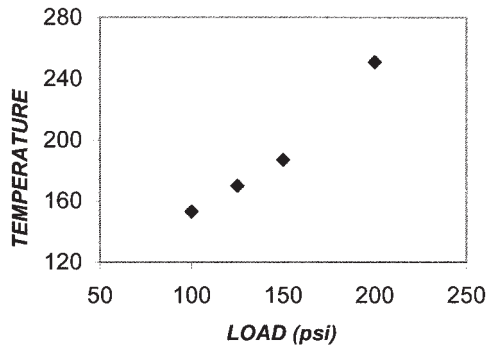


Figure 12 Dependence of disc temperature on the applied normal load. Sliding speed and braking time are 450 rpm and 12 min, respectively.

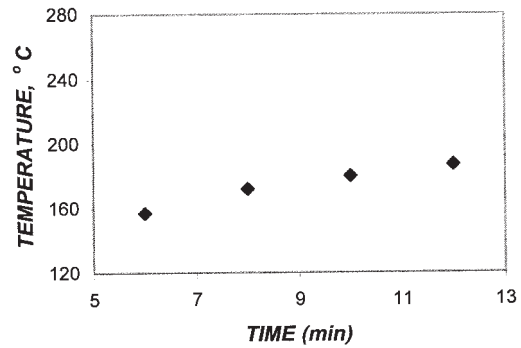


Figure 14 Dependence of disc temperature on the braking time. Sliding speed and normal load are 450 rpm and 150 psi, respectively.

Coefficients of friction between friction materials sliding against metals, which have been used in these experiments, lie in the range from 0.1 to 0.6. In these materials, μ varies with normal load, sliding speed, and time. Figure 6 indicates a substantial increase for the friction coefficient with normal load. However, this material exhibits a slight decrease in the coefficient of friction with sliding speed (Fig. 7). The obtained results show that the coefficient of friction is principally constant with sliding duration (Fig. 8).

The friction of polymers, like that of metals, can be attributed to two sources: a *deformation* term, involving the dissipation of energy in quite a large volume around the local area of contact, and an *adhesion* term, originating from the interface between the slider and the counterface.¹⁰

Wear of friction materials in isothermal tests

Eq. (1) can be rewritten as follows:

$$\log W = \log K + a \log P + b \log V + c \log t \quad (5)$$

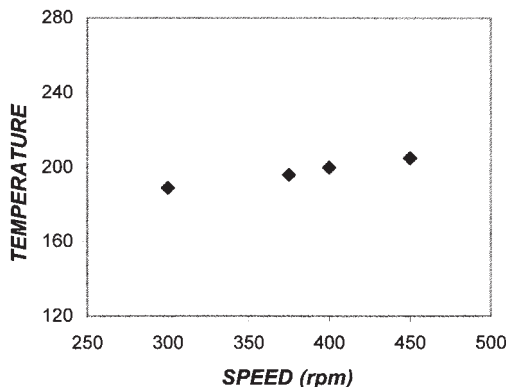


Figure 13 Dependence of disc temperature on the sliding speed. Normal load and braking time are 150 psi and 12 min, respectively.

If only one variable is varied, keeping the other variables constant, the parameters a , b , and c can be determined from the slope of lines obtained from plotting the $\log W$ against $\log P$, $\log V$, and $\log t$, respectively. Then K can be determined, based on the values of a , b , and c . The wear results are plotted in Figures 9–11.

Transient temperature tests

The variation of surface temperature with load, sliding speed, and time are shown in Figures 12–14. There is a considerable increase in the surface temperature with increasing normal load. On the other hand, temperature increases slightly with sliding speed or braking time. Figures 15–17 show the temperature variation of the friction coefficient of specimens at different conditions.

The coefficient of friction increases with increasing temperature, suggesting that deformation effects play an important role in increasing the friction at higher temperatures, since the polymer softens with increasing temperature. It is believed that the deformation effect is not due to elastic hysteresis loss or lowering by

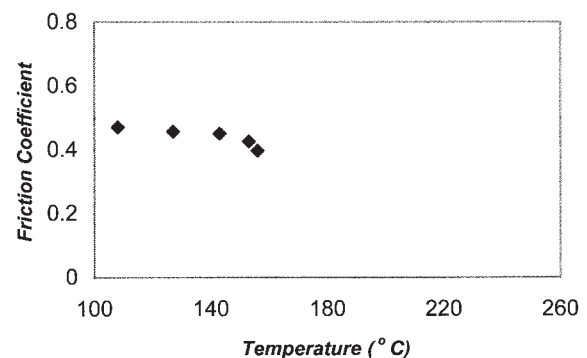


Figure 15 Relationship between friction coefficient and temperature. Load is 100 psi, sliding speed is 400 rpm, and braking time is 12 min.

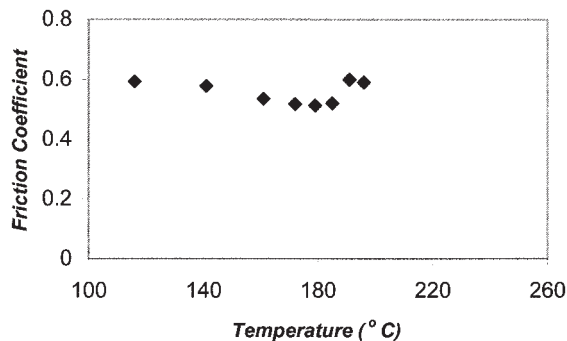


Figure 16 Relationship between friction coefficient and temperature. Load is 150 psi, sliding speed is 400 rpm, and braking time is 12 min.

asperities on the disk surface, but to the deformation of a wave-like surface deformation. The contribution of the hysteresis is generally very small and only a little lowering occurs on the softer polymer.¹⁰

Wear rate of cast iron and polymer-based friction materials

As can be seen in Table III, the wear rate of cast iron material is at least seven times more than the wear rate of composite material, giving considerable savings in direct costs. There are also many indirect savings as a result of reduced maintenance, and the removal of many of the problems associated with the wear debris from cast iron.

In the case of metals, there are several mechanisms that change in relative importance as the sliding conditions are varied. The main factors controlling the importance of the mechanisms are mechanical stresses, temperature, and oxidation phenomena. The complexity of sliding wear arises from the fact that all three controlling factors are interrelated, and may be influenced by both load and sliding velocity. Increasing the load leads directly to higher stresses, and these will result in more severe mechanical damage. Both

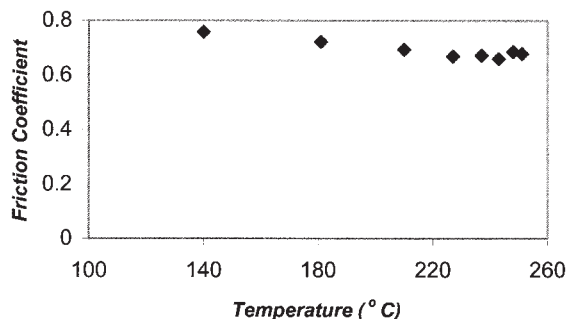


Figure 17 Relationship between friction coefficient and temperature. Load is 200 psi, sliding speed is 450 rpm, and braking time is 12 min.

TABLE III
Comparison Between Wear Rate of Cast Iron and Composite Friction Materials

Material type	Wear rate* (%)
Cast iron	5.34 ± 0.19
Composite	0.70 ± 0.13

* Sliding speed and braking time are 700 rpm and 9 min, respectively.

the load and the sliding velocity influence the interface temperature. Together, they control the power dissipated at the interface since that is the product of the sliding speed and the frictional force.

Oxidation phenomena are also important in sliding wear, since nearly all metals form oxide films in air. The rate of film growth depends, as in static oxidation, strongly on temperature; however, the local temperature at the sliding interface may be substantially higher than that of the surroundings, and may also be enhanced at the asperity contacts by transient “flashes” or “hot spots.”

Polymer-based composites are much more compliant than metals, and it is therefore reasonable to consider metallic counterfaces when sliding against polymers to act as rigid bodies. Nearly all the deformation due to contact or sliding takes place within the polymer, and the surface finishing the hard counterface has a strong influence on the mechanism of the resulting wear. If the counterface is smooth, the wear may result from adhesion between the surfaces, and involve deformation only in the surface layers of the polymer. On the other hand, if the counterface is rough, then its asperities will cause deformation in the polymer to a significant depth; wear then results either from abrasion associated with plastic deformation of the polymer, or from fatigue crack growth in the deformed region. These two classes of wear mechanisms (involving surface and subsurface deformation, respectively) have been termed interfacial and cohesive processes.¹⁰

CONCLUSION

The wear equation $W = K P^a V^b t^c$ was found to satisfactorily describe the wear of polymer-matrix friction materials in which $a = 1.6$, $b = 1.2$, $c = 1.4$, and wear factor K is 4.4×10^{-10} . The results of wear experiments showed that the polymer composite friction materials exhibit considerably higher wear resistance compared with the cast iron friction materials. In the case of polymer-based friction materials, to minimize the increase in temperature, the contact area should be relatively large, and therefore, the material needs to be soft enough. This means that the material should possess a low modulus and be conformable to

the opposing surface. This enables the generated frictional heat to be dissipated mechanically, which prevents the increase of temperature in the contact area.

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