Revisiting the Archard's wear equation.

Methods of data analysis for wear sliding tests.

A. Ramalho

Universidade de Coimbra – FCTUC-Dep. Eng. Mecânica

Pólo II, Pinhal de Marrocos, P-3030 Coimbra (Portugal)

Abstract

This paper reviews and critically discusses different methods suitable to data analyses of sliding wear tests and proposes new approaches. The Archard's wear equation was revisited and re-arranged in order to introduce the friction force instead of the normal load, resulting in a substantiation of the energetic approach of sliding wear. Analysing the results of the brass/hard steel sliding pair, it was demonstrated that the method used to analyse the results plays an important role in the scatter of specific wear rate values. As the majority of sliding tests reveal the existence of a first step where the wear is not linear with time, the assumption of a constant wear rate during the entire test could be a rough approximation. The energetic approach is a promising alternative method to characterise the wear behaviour of engineering materials.

Keywords: sliding wear, reliability of test results.

1. Introduction

Wear is a failure mode often identified as the main source of drawbacks of modern machinery. The response of materials science to minimize this problem includes composite materials, nanostructured metal alloys, new ceramics and a wide range of coatings. The increase of new solutions offered for applications subjected to high contact stresses superposed with tangential relative motion, leads to a significantly increase in the amount of new papers on the tribology areas, specially focusing the wear and friction of unlubricated sliding contacts.

The development in these research areas also provokes a dissemination of the parameters which are used to quantify the materials wear resistance. Nevertheless, the mass lost during the test, the variation in the area measured transversally to the sliding direction and the volume of the removed material are in the basis of the most used quantitative parameter, let us call them prime wear quantities. These prime quantities are in general used to calculated new parameters, which are intended to be less dependent of the test equipment and the experimental variables. Therefore, the prime wear quantities usually appear normalized to the sliding distance, the time, the normal load, the speed, the contact pressure, or simultaneously several of these variables. Let us call these new parameters as design-using parameters. The differences in the parameters usually used to express the results makes the comparison between published results very difficult and most of the published papers can hardly be used to compare with the performances of new trial products. In the limit, the main scientific principle of the repeatability of the experiments by other laboratories could be very difficult.

The present paper starts with a clarification of the Archard's wear equation in order to justify an energetic approach of wear and subsequently analyze experimental results applying different methods in order to discuss the accuracy of the final-using parameters.

2. Substantiation of the wear energetic approach

Based on experiments Archard [1] achieved the following conclusions:

- the material volume removed by wear is proportional to the sliding distance;
- the material volume removed by wear is proportional to the normal applied load;
- the materials display a wear amount inversely proportional to their hardness.

Considering the above conditions, the widely known Archard's equation can be easily established, equation (1).

$$V = K\frac{Nx}{H} \quad (1)$$

Where V is the wear volume, N is the normal load, x is the sliding distance, H is the hardness of the material and K is a non-dimensional wear coefficient. Czichos [2], proposed a new equation (2) where the material influences, H and K are grouped in a new parameter k, which is nowadays called specific wear rate.

$$V = kNx \quad (2)$$

The proportionality of wear volume with the sliding distance is easily verified as most systems reach a steady-state regime after a more-or-less small running in period. Thus, the wear appears as a cumulative phenomenon linearly proportional to the sliding distance.

The effect of the normal load is harder to explain. In fact, the normal load is perpendicular to the sliding direction during all the relative motion. Therefore, the work done by the normal load is zero during all the sliding action. Thus, this is an inconsistency of the Archard's equation, because this law establishes the proportionality between the wear volume and the normal load which do a nil work during the motion. Nevertheless, Archard's equation fits the results obtained in most of the experimental studies. The reason for this is related to the fact that the normal load determines the value of the friction force if a constant friction coefficient, μ, is assumed.

Considering the Amonton's friction equation (3), the tangential force appears proportional to the normal load.

$$F = \mu N \quad (3)$$

Using equations (2) and (3) to eliminate N, a new equation (4) could be derived.

$$V = \left(\frac{k}{\mu}\right)Fx = k'Fx \quad (4)$$

Where $k' = \left(\frac{k}{\mu}\right)$ is the specific wear rate function of the friction force.

This new equation is physically consistent because it establishes proportionality between an output of the sliding process, the wear amount, and the work done by the friction force, which is the main energy input in the tribosystem. Therefore, this new equation (4), establishes proportionality between the wear volume and the work done by the friction force along the sliding.

Equations (3) and (4) lead to more or less the same result if a constant friction regime is attained early in the initial phase of the sliding. However, in a large number of cases, the friction force varies significantly along the entire test and, therefore, the Archard's equation no more fits the results. In these cases, it is difficult to calculate a reasonable value for the friction coefficient, therefore the equation (4) needs to be improved resulting in equations (5) and (6).

$$V = k'\int_x F(x)dx = k'v\int_t F(t)dt \qquad (5)$$

$$V = k'\sum_i \overline{F}x_i \Delta x_i = k'v\sum_i \overline{F}t_i \Delta t_i \qquad (6)$$

Where

- *F(x)* and *F(t)* are the instantaneous values of the friction force for the sliding distance *x* or the instant *t*;
- $\overline{F}x_i$ and $\overline{F}t_i$ are average values of the friction force during Δx_i or Δt_i.

These concepts can easily be applied to the analysis of tribological experiments because it involves only the value of the friction force along the sliding distance (or along the time). Usually discrete values F_i of the friction were acquired during the test with a period Δt between acquisitions; therefore, the equation (6) can be directly applied. Alternatively the arithmetical average of the friction force can be calculated applying this value in the equation (4), leading to the final value of the specific wear rate function of friction (*k'*).

3. Methods of data analysis for sliding wear tests

Assuming the specific wear rate as a suitable final-using parameter to express the wear performance of the materials, another important question remains: what can be done to assure a high reliability of the experimental results? Further than the quality of the test equipment, the care of the test procedures and the accuracy of the measurement techniques; the method used to derive the value of the final-using parameter from the test results could lead to significant differences. An experimental study will be used to test different approaches.

3.1 Experimental work

A set of tests was done using a sliding tribometer with unlubricated crossed cylinder contact, figure 1. The equipment includes a rotating cylinder (3) and a cylindrical stationary specimen (5). The normal load is applied by means of a spindle/spring system (4) and is measured by a load cell (1). The stationary specimen, which diameter is 10 mm, is supported by a free rotating system, which is equilibrated by a second load cell (2) used to measure the friction force.

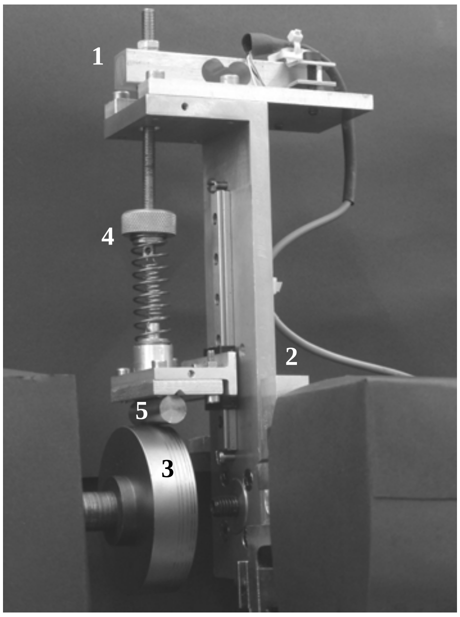

Figure 1- Crossed-cylinder wear test equipment.

This experimental study was designed to compare the efficiency of different methods which could be used to calculate the value of the specific wear rate of materials under sliding contacts. Table 1 summarizes the materials and the contact conditions.

Table 1 – Test conditions

	Rotating cylinder	Fixed Cylinder
Diameter (mm)	57	10
Material	α Brass (67%Cu 33%Zn)	Steel 34 CrNiMo6
Hardness (MPa)	1710	3650
Sliding speed (m/s)	0.5	
Normal load (N)	1, 2, 3, 4 and 5	
Sliding distance (m)	300, 600, 900, 1200 and 1800	

The normal load ranged from 1 to 5 N and for each value of the normal load the test sliding distance was varied from 300 to 1800 m; therefore 25 tests were carried out.

Before testing, the specimens were cleaned with ethyl alcohol. During the test, the friction force value was acquired periodically, at time intervals of Δt. In each acquisition, a set of several thousand of values was collected, corresponding to an acquisition time greater than the rotation period. Therefore, the average value of the friction force, $\bar{F}$, calculated from the acquired friction force data, corresponds to the average of the friction during a rotation.

For the stationary specimen, the volume of the wear scar can be calculated assuming an imposed wear shape using the approximate expression (7) derived by Ramalho [3]. This simple equation is very accurate with errors of less than 0.2% [3].

$$V = \frac{\pi}{2} \times h^2 \times \sqrt{r_1 \times r_2} \qquad (7)$$

Where

r_1 – radius of the stationary specimen;

r_2 – radius of the rotating specimen;

h – depth of the scar.

Each scar is measured by taking the larger, *a*, and the smaller, *b*, dimensions of the wear surface. The value of scar depth can be calculated either by equation (8) or (9). In the present study, the average value of the wear depth, *h*, will be used, equation (10).

$$h_1 = r_1 - \sqrt{r_1^{\,2} - \left(\frac{a}{2}\right)^2} \qquad (8)$$

$$h_2 = r_2 - \sqrt{r_2^{\,2} - \left(\frac{b}{2}\right)^2} \qquad (9)$$

$$h = \frac{h_1 + h_2}{2} \qquad (10)$$

The sliding leads to a wear track on the rotating cylinder, the evaluation of the removed material was done by measuring the area of the transversal profile of the track. For each wear track five measurements were done in different radial positions, and the wear volume was calculated multiplying the average value of the area by the track length.

3.2 Data analysis

To analyse the accuracy of the results, different methods were used to calculate the value of the specific wear rate from the experimental results.

Method A- Each test considered separately

This is the most frequently used method. For each test considered separately, the wear volume measured at the end of the test is used to calculate the value of the specific wear rate applying the equation (2). Therefore, in the current study a total of 25 values were calculated, tables 2 and 3 show the results respectively for the brass rotating cylinder and the steel fixed pin.

Table 4 summarizes the average results obtained for the 25 test conditions and also the maximum and minimum values as well as the scatter. The scatter parameter was calculated as the range of the values normalized by the average.

As both sliding distance and normal value were varied, the effect of each of those test parameters can be object of separated analysis. Table 5 and 6 display the results obtained for both brass cylinder and steel pin.

The results reveal that the steel pin shows always higher scatter than the brass cylinder. This fact certainly occurs because the wear is measured by a direct way for the cylinder brass whereas for the steel pin the wear was calculated assuming a constant shape of the wear scar. Probably this assumption is rough for the present case where the pin is much harder than the cylinder. In general, the scatter decreases when the sliding distance or the applied load increase.

Table 2a) Brass-cylinder wear volume values (mm^3)

	Normal load (N)				
Sliding distance (m)	1	2	3	4	5
300	0.251	0.922	0.776	1.454	1.417
600	0.822	1.223	2.133	2.710	2.805
900	0.909	2.383	3.533	4.963	4.362
1200	1.381	3.088	4.857	5.528	7.077
1800	2.438	4.405	8.414	9.534	11.439

Table 2b) Brass-cylinder specific wear rate (mm^3/Nm)

	Normal load (N)				
Sliding distance (m)	1	2	3	4	5
300	8.38×10^{-4}	1.54×10^{-3}	8.62×10^{-4}	1.21×10^{-3}	9.45×10^{-4}
600	1.37×10^{-3}	1.02×10^{-3}	1.19×10^{-3}	1.13×10^{-3}	9.35×10^{-3}
900	1.01×10^{-3}	1.32×10^{-3}	1.31×10^{-3}	1.38×10^{-3}	9.69×10^{-4}
1200	1.15×10^{-3}	1.29×10^{-3}	1.35×10^{-3}	1.15×10^{-3}	1.18×10^{-3}
1800	1.35×10^{-3}	1.22×10^{-3}	1.56×10^{-3}	1.32×10^{-3}	1.27×10^{-3}

Table 3a) Steel-pin wear volume values (mm^3)

	Normal load (N)				
Sliding distance (m)	1	2	3	4	5
300	0.002	0.021	0.034	0.032	0.034
600	0.016	0.034	0.049	0.055	0.057
900	0.009	0.031	0.053	0.074	0.062
1200	0.012	0.043	0.063	0.073	0.084
1800	0.038	0.055	0.096	0.108	0.132

Table 3b) Steel-pin specific wear rate (mm^3/Nm)

	Normal load (N)				
Sliding distance (m)	1	2	3	4	5
300	6.59×10^{-6}	3.47×10^{-5}	3.73×10^{-5}	2.69×10^{-5}	2.27×10^{-5}
600	2.67×10^{-5}	2.83×10^{-5}	2.70×10^{-5}	2.30×10^{-5}	1.90×10^{-5}
900	1.02×10^{-5}	1.73×10^{-5}	1.96×10^{-5}	2.06×10^{-5}	1.39×10^{-5}
1200	9.63×10^{-6}	1.77×10^{-5}	1.75×10^{-5}	1.52×10^{-5}	1.41×10^{-5}
1800	2.11×10^{-5}	1.54×10^{-5}	1.78×10^{-5}	1.50×10^{-5}	1.46×10^{-5}

Table 4- Values obtained using the results of each test separately.

	Specific wear rate mm^3/Nm			
	Average	Maximum	Minimum	Scatter
Brass cylinder	1.18×10^{-3}	1.56×10^{-3}	8.4×10^{-4}	± 30.4%
Steel pin	1.95×10^{-5}	3.73×10^{-5}	6.58×10^{-6}	± 78.5%

Table 5- Values of brass cylinder specific wear rate obtained by grouping the results for the same load or the same sliding distance.

	Normal load (N)					k	
Sliding distance (m)	1	2	3	4	5	Average (mm^3/Nm)	Scatter (%)
300						1.08×10^{-3}	± 32.4
600						1.13×10^{-3}	± 19.3
900						1.20×10^{-3}	± 17.1
1200						1.22×10^{-3}	± 8.1
1800						1.35×10^{-3}	± 12.4
Average k	1.14×10^{-3}	1.28×10^{-3}	1.25×10^{-3}	1.24×10^{-3}	1.06×10^{-3}		
Scatter (%)	± 23.22	± 20.27	± 27.79	± 10.07	± 15.85		

Method B- Wear rate calculated for each value of the normal load

For each normal load value, the wear rate was calculated as the slope of the straight line fitted to the experimental data points, figure 2. The specific wear rate was calculated dividing the wear rate by the normal load. The uncertainty Δk of the results could be estimated selecting the parallel straight line defined by the more distant points below and above the average line. The difference between the intercept of these two lines is the range of wear volume, ΔV, which could be divided by the range of the sliding distance covered by the test program, S_{range}, equation (11) normalized to the normal load.

Table 7 shows the results; the scatter was normalized by the average value of the specific wear rate.

$$\Delta k = \frac{\Delta V}{N(S_{range})} \quad (11)$$

Table 6- Values of steel pin specific wear rate obtained grouping the results for the same load or the same sliding distance.

	Normal load (N)					k	
Sliding distance (m)	1	2	3	4	5	Average (mm^3/Nm)	Scatter (%)
300						2.65 x10^{-5}	± 59.9
600						2.48 x10^{-5}	± 18.8
900						1.63 x10^{-5}	± 32,0
1200						1.48 x10^{-5}	± 27.3
1800	1,48E-05	2,27E-05	2,38E-05	2,01E-05	1,69E-05	1.68 x10^{-5}	± 19.3
Average k	1.48 x10^{-5}	2.27 x10^{-5}	2.38 x10^{-5}	2.01 x10^{-5}	1.69 x10^{-5}		
Scatter (%)	± 67.7	± 42.6	± 41.4	± 29.5	± 26.1		

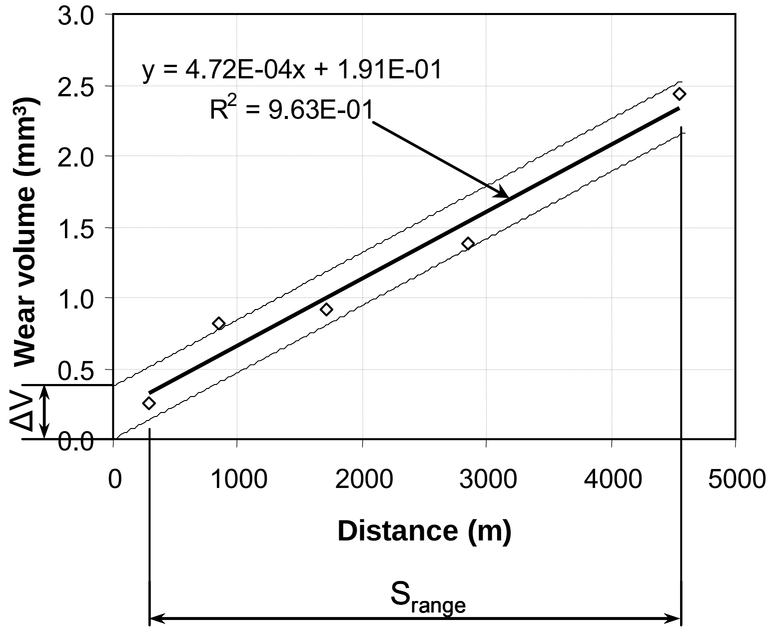

Figure 2- Example how to calculate the specific wear rate, average and scatter.

Comparing the results for constant normal load showed on the table 5 and 6 with those of the table 7, there are significant differences. The uncertainty of the results is much lower when the tests are analysed by the method B.

Table 7- Specific wear rate calculated by the method B.

	Specific wear rate k (mm^3/Nm)			
	Brass cylinder		Steel pin	
Normal load (N)	Average	Scatter (%)	Average	Scatter (%)
1	1.25 $x10^{-3}$	± 9.0	1.69 $x10^{-5}$	± 26.6
2	1.25 $x10^{-3}$	± 2.5	1.73 $x10^{-5}$	± 14.8
3	1.44 $x10^{-3}$	± 2.5	1.88 $x10^{-5}$	± 11.0
4	1.28 $x10^{-3}$	±1.3	1.65 $x10^{-5}$	± 11.0
5	1.18 $x10^{-3}$	± 1.6	1.48 $x10^{-5}$	± 6.4

Method C- Specific wear rate calculated from all the experimental data

The specific wear rate was calculated from all the experimental results as the slope of the straight line fitted to the wear volume plotted against the normal load x sliding distance.

The scatter of the results was calculated by a similar way as explained for the method B, but applying the equation (12).

$$\Delta k = \frac{\Delta V}{(LS)_{range}} \quad (12)$$

Table 8 show the results revealing low uncertainty for both brass and steel.

Table 8 - Specific wear rate calculated by method C.

	Specific wear rate mm^3/Nm	Scatter
Brass cylinder	$1.26x10^{-3}$	± 12.7%
Steel pin	$1.62x10^{-5}$	± 11.5%

Method D- Energetic approach

Taking into account the previously introduced definitions and as resumed in equations (4) to (6), the energy dissipated by friction, E_f, can be calculated based on the evolution of the friction force along the test, equation (13).

$$E_f = \sum_i \overline{F} x_i \Delta x_i \quad (13)$$

On all experiments the friction force was acquired along the test of each specimen, figure 3. Table 9 shows the values of the energy dissipated by friction along the test, calculated applying the equation (13) to the friction force values corresponding to each test.

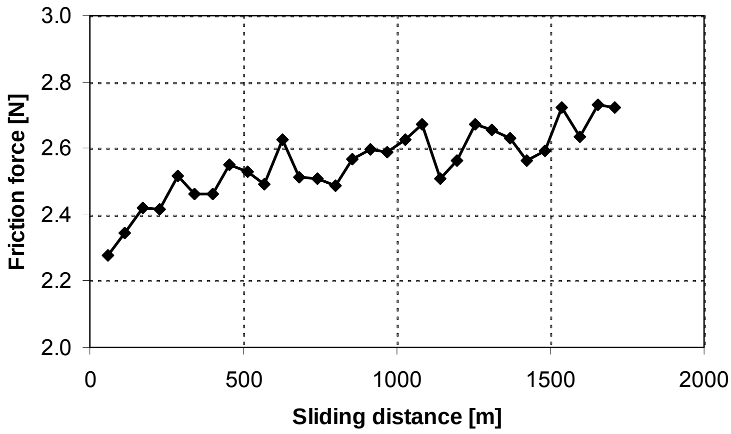

Figure 3- Evolution of the friction along the test (normal load: 4 N, Sliding distance: 1800 m).

Table 9 – Energy dissipate by friction, E_f (J).

	Normal load (N)				
Sliding distance (m)	1	2	3	4	5
300	54.0	246.7	444.7	673.0	788.0
600	313.1	506.1	960.0	1414.3	1681.7
900	361.7	896.9	1594.2	2211.4	2673.7
1200	577.0	1100.1	2069.7	2645.5	3652.8
1800	1010.2	2080.4	3276.4	4342.4	5825.1

The value of the specific wear rate function of the friction force, *k'*, can be calculated as the slope of the straight line fitted to the experimental points in a plot wear volume – energy, figure 4. The uncertainty of the results can be calculated applying similar concepts to those established in methods B and C. Table 10 summarizes the average values and the scatter of the specific wear rate as a function of the friction for both the brass cylinder and the steel pin.

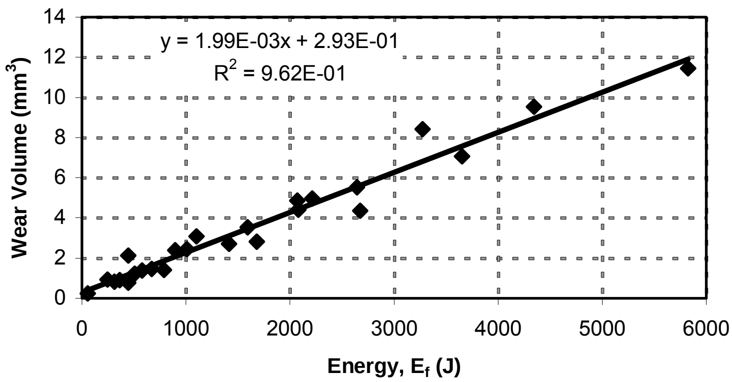

Figure 4- Wear volume as a function of the energy dissipated by friction.

Table 10 – Specific wear rate function of the friction, E_f (J).

	Specific wear rate function of the friction mm^3/Nm	Scatter
Brass cylinder	$1.99x10^{-3}$	± 12.3%
Steel pin	$2.08x10^{-5}$	± 16.4%

4. Discussion

Methods A to C use the same wear design-using parameter although calculated by different ways. Comparing the methods A to C, it can be concluded that the method C allows the lowest uncertainty of results. In fact, the method C leads to scatter values between 15% and 40% for the same parameters when calculated using the method A.

Focusing on the results obtained by method A, it can be observed that, in general, the scatter of the specific wear rate is much higher for the tests done with the lower sliding distances. It can also be observed that the steel pins show a decreasing average value of the specific wear rate when the sliding distance rises from 300 to 1800 meters. On the contrary the brass cylinders reveal an increasing tendency. These behaviours reveal a dependence of the wear rate with time, which affect the two materials in different ways. It is widely accepted that there is a typical evolution of the wear with time that includes 3 regimes. H. Czichos [4] identifies a first regime with the wear growing proportionally to the square root of time; followed by a steady state regime and finally a regime with an exponential growth of the wear. Thus, assuming this evolution, only during the second regime, the wear is proportional to the time, or the sliding distance under constant speed. Therefore, it is impossible to calculate a suitable value for the specific wear rate assuming a linear evolution from the origin.

The energetic approach seems be the best way to calculate a reliable parameter to be used on the design of mechanical components. Besides the reliability, another advantage of the energy approach is the fact that the evolution of the wear as a function of the energy dissipated by friction is always linear passing on the origin. In the present work the use of energetic approach leads to values of wear parameters with reasonably low scatter.

5. Concluding remarks

- Archard's wear equation for sliding systems could be re-arranged in order to introduce the friction force instead of the normal load.
- The methods used to analyze the test results play an important role in the scatter of the obtained specific wear rate.
- Calculating the specific wear rate of a material using fixed test parameters could lead to low reliable results, even if the test was repeated several times.
- The energetic approach is a promising alternative method to characterise the wear behaviour of materials.

References

[1] J.F. Archard, Contact .and Rubbing of Flat Surfaces," J. App. Phys.1953, 24, 981.

[2] H. Czichos, Tribology, a Systems Approach to the Science and Technology of Friction, Lubrication and Wear, Tribology Series 1, Elsevier, Amsterdam, 1978.

[3] A. Ramalho, "Micro-abrasive wear test of thin coated cylindrical surfaces", Tribology Letters, Vol.16 Nº1-2 (2004) 133-141.

[4] H. Czichos, Systematic diagnosis of friction and wear test data, in ASM Handbook: Friction, Lubrication, and Wear Technology, Vol 18, Ed. by PJ Blau, 1992.