Using the Energy Pulse Concept for Designing Better Wear Tests

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1. Introduction

When designing a bench test to model wear in practical contacts, considerations of scale are important. It is hazardous simply to attempt to define the "real life" conditions (load, speed, temperature etc.) and apply them to a small test piece on a test machine. The real concern is to do with the temperatures reached by the test specimens.

If large amounts of energy are being dissipated in small test specimens with supporting structures that do not allow the heat to escape then it is clear that the specimens will become very hot. Hence the bulk temperature may exceed what is experienced in practice and this will undoubtedly produce transitions in wear or frictional response. The temperature reached at the surface of the contact (the flash temperature) is influenced by the width of the contact, varying with the square root of contact width [1]. The bulk and flash temperatures are responsible for many wear and friction effects. Lim and Ashby [2] used these concepts to build their models for transitions on their wear mechanism maps for steel.

The purpose of tribological bench tests is to improve accessibility to the friction or wear processes and thus to facilitate a more detailed investigation than is possible with the real mechanism.

Most would accept that the most important criterion of "success" is that the test should reproduce the wear and/or failure mechanisms of the application. But is this a sufficient criterion?

2 Energy Dissipation And The Energy Pulse Concept

Wear occurs as the result of the dissipation of frictional energy in the contact and this is irresistibly accompanied by a rise in temperature. The energy can only be dissipated in heat and wear particle generation. The temperatures achieved in the contact drive the formation of oxides on the surfaces, the transformation of microstructure, the formation or break-down of lubricant additive or other tribochemical films, the melting of the surface (the PV limit of the material) or thermal stress induced failure.

The frictional energy is generated by the combination of load and sliding speed and its distribution and dissipation is influenced by other contacting conditions such as size and relative velocity. Different patterns of energy dissipation will give different wear patterns. These factors can usefully be summed up in two "global" parameters and these are shown to be valuable in defining the bench test.

2.1 Friction Power Intensity

The friction power intensity (FPI) as presented by Matveesky [3] is simply defined as of the amount of energy pumped into the rubbing surfaces as they pass through the contact zone. The temperature achieved in the contact and in the bulk is directly related to the FPI and the size and thermal characteristics of the materials and their supports.

$$\textit{Friction Power Intensity,}\ Q_F = \frac{\mu P v_s}{A} \qquad W/mm^2$$

μ is friction coefficient, P is the normal load, v_s is the sliding speed and A is the apparent area of contact. The FPI defines only the rate of energy generation and does not take into account the timescale over which this energy can be lost to the contacting materials. This timescale clearly has implications for the amount of damage caused in the contact.

2.2 Energy Pulse

The Energy Pulse (EP) as presented by Plint [4] is the product of the FPI and the contact transit time. The EP therefore takes into account the length of time during which the material is subjected to energy input during its transit of the contact zone.

$$\textit{Energy Pulse} \qquad EP = \left[\frac{\mu P v_s t_t}{2A}\right] \quad J/mm^2$$

Where t_t is the transit time and the factor of 2 makes the assumption that the energy generated in the contact zone is equally split between the two surfaces. In practice, the slower moving surface will have the higher Energy Pulse.

The EP can be seen to be analogous to the Archard Wear Law, but using the friction force rather than the applied load. This is perhaps more logical since it takes into account the actual rubbing conditions (but assumes that μ can be measured on the test machine).

$$\textit{Archard Wear Law} \qquad \Delta V = \left[\frac{k P v_s t_t}{A}\right] \qquad mm^3$$

Each Energy Pulse can be regarded as an incremental contribution to wear or surface damage in the contact. The sum of the Energy Pulses can be used as a measure of the total wear.

2.3 Defining the Energy Pulse

For the generic rolling/sliding line contact in Figure 1 with contact length *a* mm and width *b* mm and $v_2 > v_1$ the average FPI for the contact zone is:

$$Q_f = \mu P \frac{(v_2 - v_1)}{A}$$

The transit times for the contact are $t_t = (a/v_2).10^{-3}$ for the upper body and $(a/v_1).10^{-3}$ for the lower body. The E_p for the two bodies are:

$$upper\ body \quad EP = \mu P \frac{(v_2 - v_1)}{2bv_2} \cdot 10^{-3}$$

$$lower\ body \quad EP = \mu P \frac{(v_2 - v_1)}{2bv_1} \cdot 10^{-3}$$

The EP is independent, for a given applied load, of the load per unit width of contact. The smaller a, the greater FPI, but this is balanced by a reduction in the transit time. There are a number of other implications:

1. The energy pulse is always greater for the slower- moving surface than for the faster
2. If $v_1 = v_2$ (i.e. pure rolling) there is no energy pulse.
3. In any given mechanism under specified loading conditions, the level of the energy pulse is independent of running speed (it depends only on the sliding velocity)
4. If one surface is stationary relative to the point of contact, as in many tribology test machines, the energy pulse for that surface becomes infinite as the energy input is continuous

In machine components there can be very high FPI's but because the contact durations are short, the Energy Pulse in low and hence the incremental damage is low. In simple sliding test machines there is no Energy Pulse as such: one specimen receives a continuous flow of energy and therefore the rate of damage will be correspondingly high.

2.4 Expected values of the Energy Pulse

The Energy Pulse varies directly with the coefficient of friction in the contact. In typical engineering contacts such as gears and cam/followers this cannot be measured directly. Merritt [5] suggests a value of $\mu = 0.08$ as typical for spur gears, Bell and Colgan [6] assumed $\mu = 0.10$ for a cam follower contact for their modelling purposes. They also studied boundary lubricated friction under a range of loads and speeds in the TE 77 High Frequency Friction Machine and obtained values of μ from 0.06 to 0.08.

Figure 2 shows the magnitude of the Energy Pulse (maximum reached in the contact cycle) for the standard gear testing machines (assuming $\mu = 0.1$). At the maximum load the FZG machine imposes an EP of 0.0274 J/mm^2 which is lower than the other machines. It is well known that modern lubricants can pass FZG load stage 12 by a substantial margin, leading to the adoption of the Ryder configuration for more severe tests. An EP level of 0.0274 J/mm^2 would be sufficient to raise the temperature of a steel gear by about 75°C to a depth of 0.1 mm.

The figure also shows the EP for the standard sliding test machines used for load carrying studies. Since the point of contact is stationary on one specimen, the energy input is continuous: there is no Energy Pulse. In these cases the FPI parameter must be considered and these are shown for the same contacts in Figure 3. The Gear and Cam/Follower rigs all have similar FPI levels, but the bench-scale rigs have a wide range of FPI. When it is considered that all this power is being dissipated into small test pieces, it is perhaps understandable that they give different rankings for lubricants when compared with the larger component test rigs.

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It was noted earlier that wear transitions are related to the frictional energy dissipation. The use of the EP analysis is clearly on the basis that the Archard Wear Law is in operation. Can the EP also be used also to study the gross failure of surfaces?

3. Examples Of Tests Using The Energy Pulse

3.1 Cam and follower contact simulation

Bell and Colgan set out to find an explanation for the wear patterns commonly found in pivoted follower valve trains used in modern high-speed automotive engines. In these systems, wear is much more pronounced on the follower than the cam. The depth of wear along the path of contact of the follower was measured after running the engine at a camshaft speed of 1500 rpm for 100 hours.

Figure 4 shows such a wear pattern. It is clear that the most severe wear takes place at the ends of the path of contact, where the point of contact with the cam comes to rest and reverses its motion. There are secondary maxima, designated by R_1 and R_2, which correspond to the positions of the point of contact during passage of the opening and closing ramps of the cam.

Bell and Colgan made a systematic study of various parameters that might be expected to correlate with this wear pattern, including:

1. maximum Hertzian contact pressure
2. flash temperature rise, from the Blok equation
3. locations of minimum EHD oil film thickness

None of these criteria gave a correct prediction of the observed wear pattern. They then attempted a correlation using a derivative of the Archard Wear criterion, expressed in the form:

$$\delta = \left[\frac{k(h) P v_s t_t}{A} \right] \quad mm^3$$

where δ is the depth of wear and k(h) is a wear coefficient, which is a function of materials and lubricant and of the calculated oil film thickness h in the contact. This expression is evidently identical to Plint's expression for the Energy Pulse, with k(h) substituted for $\mu/2$. It was assumed that k(h) would increase as film thickness diminished, the same trend as would be expected for μ.

Figure 4 also shows the pattern of wear predicted for the follower using their criterion and a step-by-step computation, from which the close agreement is apparent. This confirmed that maximum damage occurred not at the point of minimum film thickness (although the wear is also high at that point), but at the point of maximum contact dwell time (Energy Pulse in the terms of this paper).

These conditions were then modelled by using a modified Amsler test device [7] in which a block was reciprocated against a driven roller (Figure 5). By adjusting the stroke of the block, the contact dwell time could be adjusted to give values similar to that of the cam/follower system. In this way a simple test generates the same type of wear patterns as the complex mechanical system and thus provides a more controlled and accessible test environment.

3.2 Gear Simulation: scuffing in two roller contact

Bell's model test on the reciprocating Amsler produced a cyclic Energy Pulse. The more traditional method of working in EHD lubrication is a two-roller machine. Here two discs can be run with a fixed ratio of sliding to rolling and therefore a fixed value of Energy Pulse. The failure of the rollers can be linked to a critical level of EP

Bell, Dyson and Hadley [8] carried out extensive experiments on rollers having a radius of 38.1 mm and a width of 11 mm, under conditions closely resembling those at the start of gear engagement. The rollers where of steel to specification BS EN 34, (a low carbon 2% nickel case hardening steel) hardened to 870 VPN. The lubricant was a solvent refined paraffinic oil without extreme pressure additives, viscosity SAE 100 ; such an oil would, of course, be expected to show substantially lower scuff resistance than a modern gear oil. These tests covered a wide range of variables:

rolling speeds	0.24 to 11.25 m/s
sliding speeds	0.48 to 4.00 m/s
loads	232 to 1595 N/mm face width

Sufficient data is given to allow the value of the Energy Pulse to be calculated (Figure 6) The figure also shows which tests terminated in scuffing of the rollers and the correlation with the value of EP is evident. The results are also plotted in order of diminishing Friction Power intensity in Figure 7. Here there is a degree of correlation but the results are much more closely grouped and it would be difficult, on the basis of these tests, to specify a "safe" level of FPI .

It appears from these tests that , for this combination of materials and lubricant, a limiting safe value for EP is about 0.01 J/mm^2.

3.3 Interchangeability of EP and Surface Temperature in Lubricated Scuffing

The preceding analysis of the two roller contact leads to the conclusion that there us a link between the EP and scuffing in a lubricated contact. This effect was further examined by Plint et al. [9] in a test machine where one specimen was fixed and the other (a roller) was reciprocated (Figure 8). This roller was in turn driven by a mechanism to give produce a combined rolling and sliding motion with a range from pure rolling (Rolling Velocity Ratio, RVR = 1) to pure sliding (RVR = 0). Having the plate fixed meant that the bulk temperature of the contact could be varied very easily as an independent test variable.

A series of tests were carried out exploring the relation between the magnitude of the Energy Pulse and the tendency to scuff. The test conditions were as follows:

Roller:	NSOH B 01 gauge plate, hardened to 550 VPN, 20 mm diameter
Plate:	NSOH B 01 gauge plate, hardened to 750 VPN finished to 0.2 μm R_a
Stroke:	10 mm
Frequency:	5 Hz
Load:	250 N (Maximum Hertz Stress: 1.8 GPa)
Temperature:	ramped from 50°C to failure at 5°C/minute
Lubricant:	Synthetic oil, SAE 5W50, FZG Pass Level 9, drip fed at a rate of 10 ml/hour

The test procedure was to start the machine and, after a running-in period of 10 minutes, to increase the bulk temperature of the fixed specimen at a rate of 5°C/min. Friction force was recorded continuously and failure was indicated by a sudden increase in the friction force. The scuffing was seen on both surfaces, but was most severe on the moving specimen, the one subjected to the larger Energy Pulse.

Figure 9 plots the μ immediately before failure, scuffing temperature and derived Energy Pulse against RVR. Figure 10 shows the variation of failure temperature with EP. With no energy input from sliding the lubricated contact would cease to function at 300°C, the failure temperature falls steadily as the EP increases.

4. Conclusion

This presentation started with the assertion that wear in any contact can only arise as a result of the expenditure of frictional energy in the contact.

There have been attempts to quantify this effect, of which the most widely known and successful has been the Archard Wear Criterion. The Friction Power Intensity of Matveesky has also been used as a measure of the rate at which frictional energy is generated in the contact, but this clearly needs to be multiplied by the duration of that energy input to give a meaningful indication of the true severity of the contact conditions. Plint's Energy Pulse, while analogous to the Archard Criterion, is logically preferable since it makes explicit reference to the frictional forces involved.

The use of the Energy Pulse then allows the experimenter to design appropriately scaled tests to study the wear and failure of contacts.

5. References

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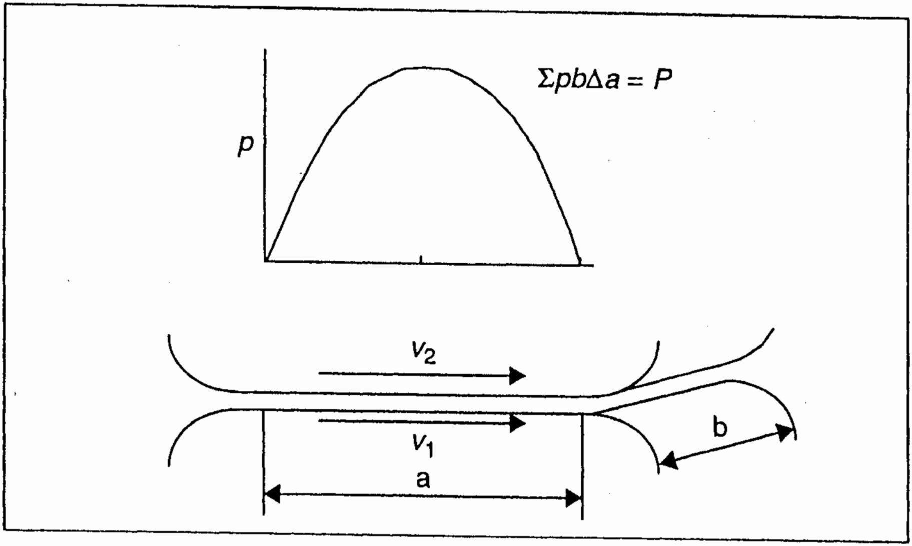


Figure 1: Generic Sliding/Rolling Hertzian Line Contact

Figure 2: Energy Pulse Parameter for Common Test Machines

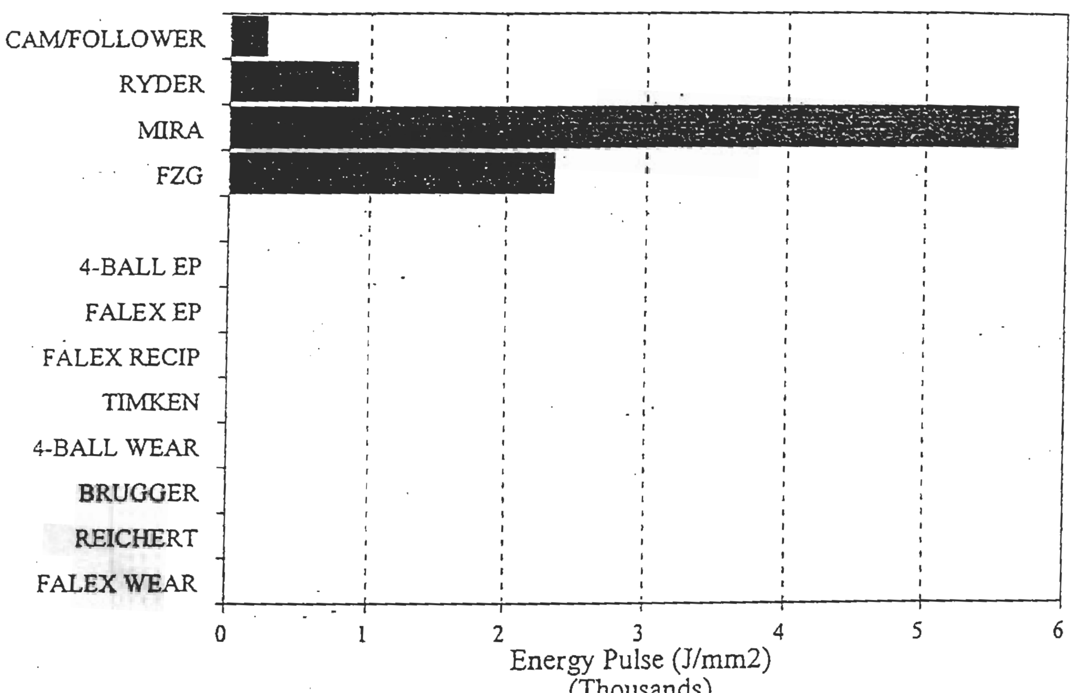
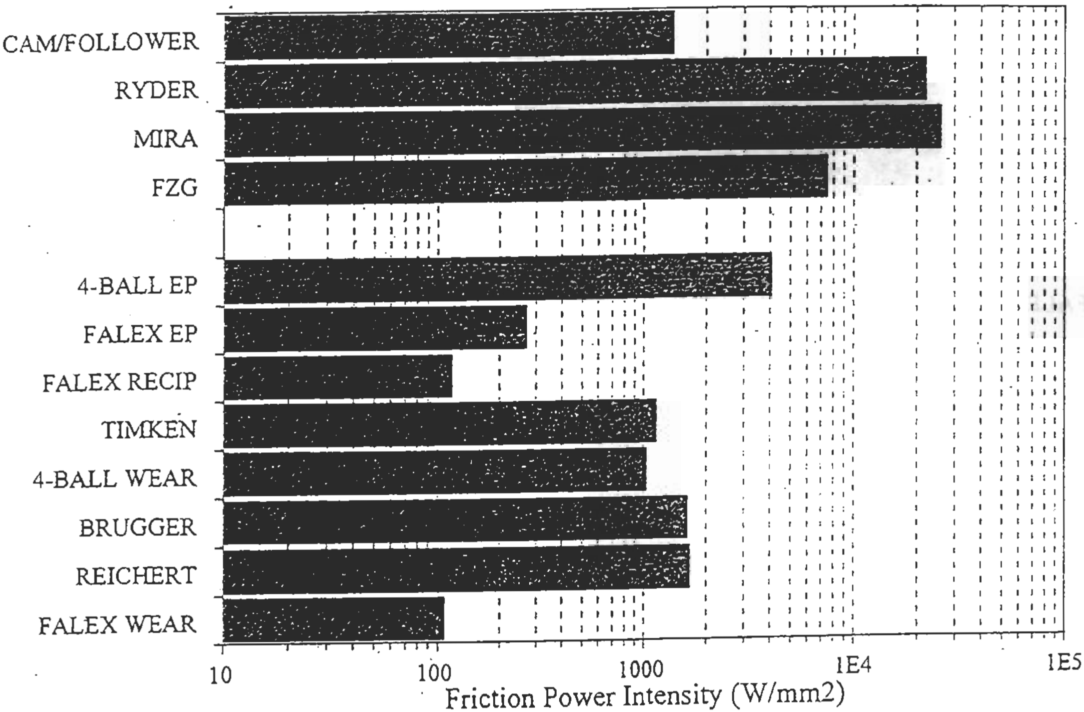
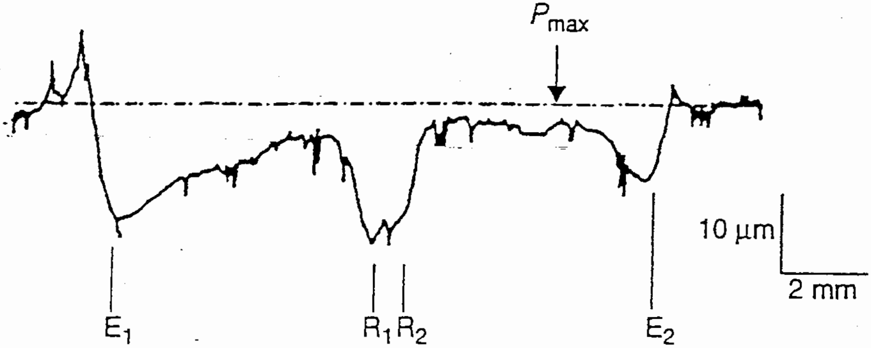


Figure 3: Friction Power Intensities for Common Test Machines



(a) Wear profile, showing predicted positions of maximum wear



(b) Predicted wear profile by derivative of Archard Wear Criterion

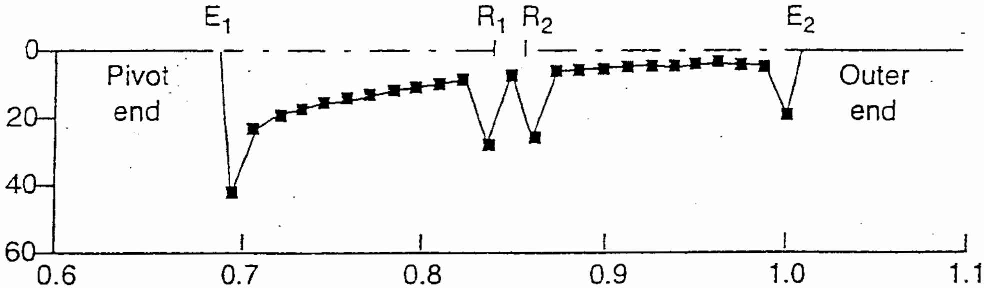


Figure 4: Wear of Finger Follower after Bell and Colgan [4]

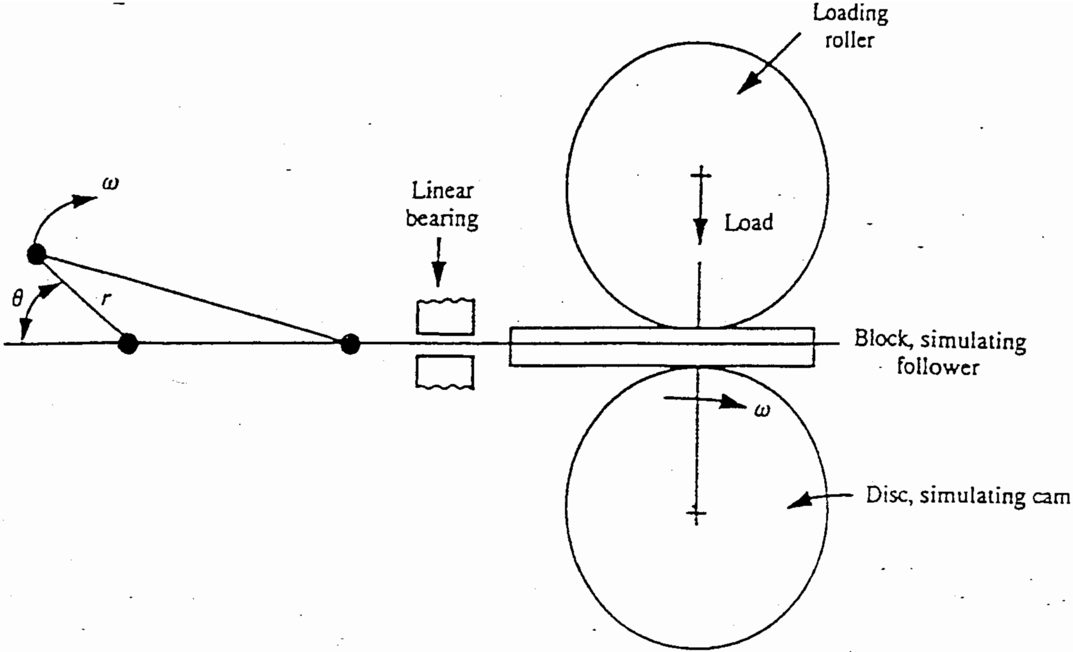


Figure 5: Basic Kinematic Design of Shell Reciprocating Amsler Rig after Bell [5]

Figure 6: Energy Pulse Parameter for Bell, Dyson and Hadley's Data
Shaded Tests Failed by Scuffing

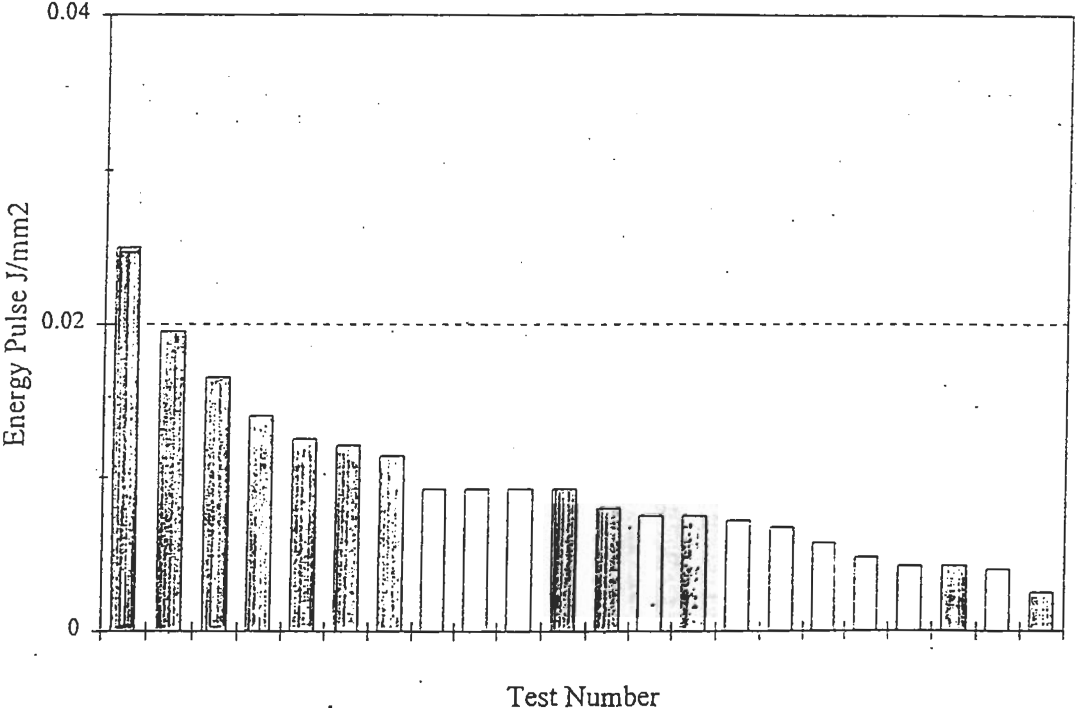


Figure 7: Friction Power Intensities for Bell, Dyson and Hadley's Data
Shaded Tests Failed by Scuffing

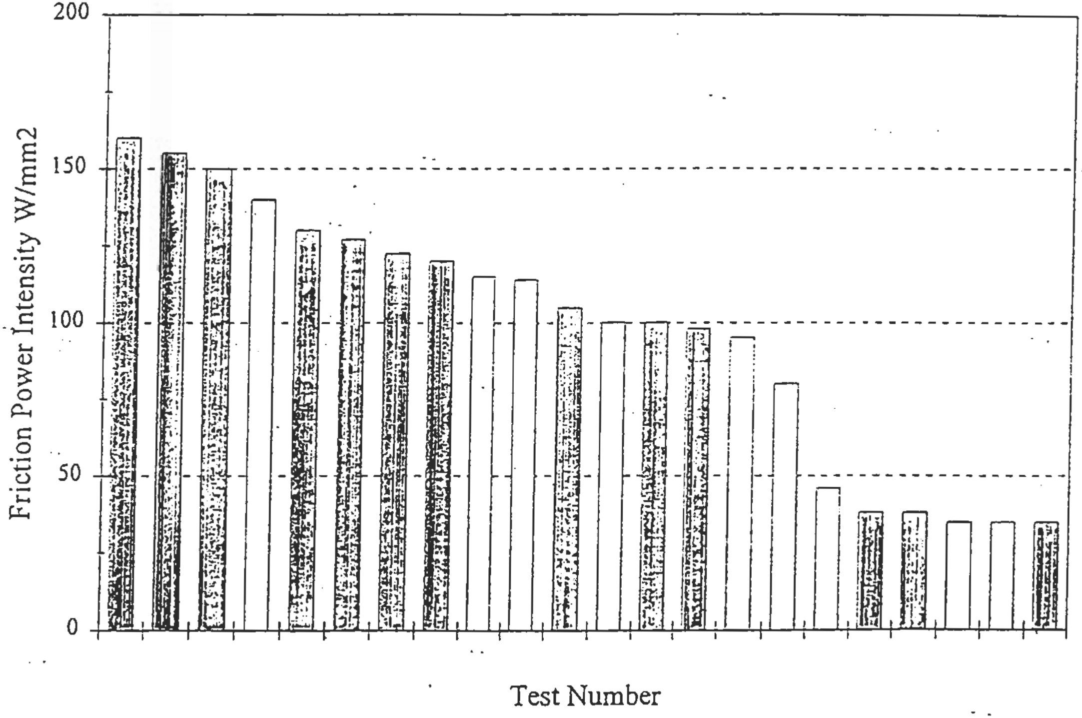


Figure 8: Schematic of Model Slide/Roll Contact

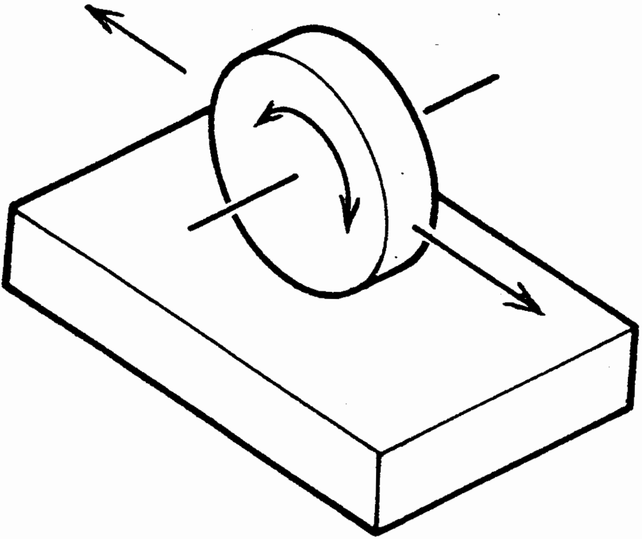


Figure 9

Relationship between Scuffing Failure Temperature, Energy Pulse and Coefficient of Friction in a Model Rolling/Sliding Test Configuration

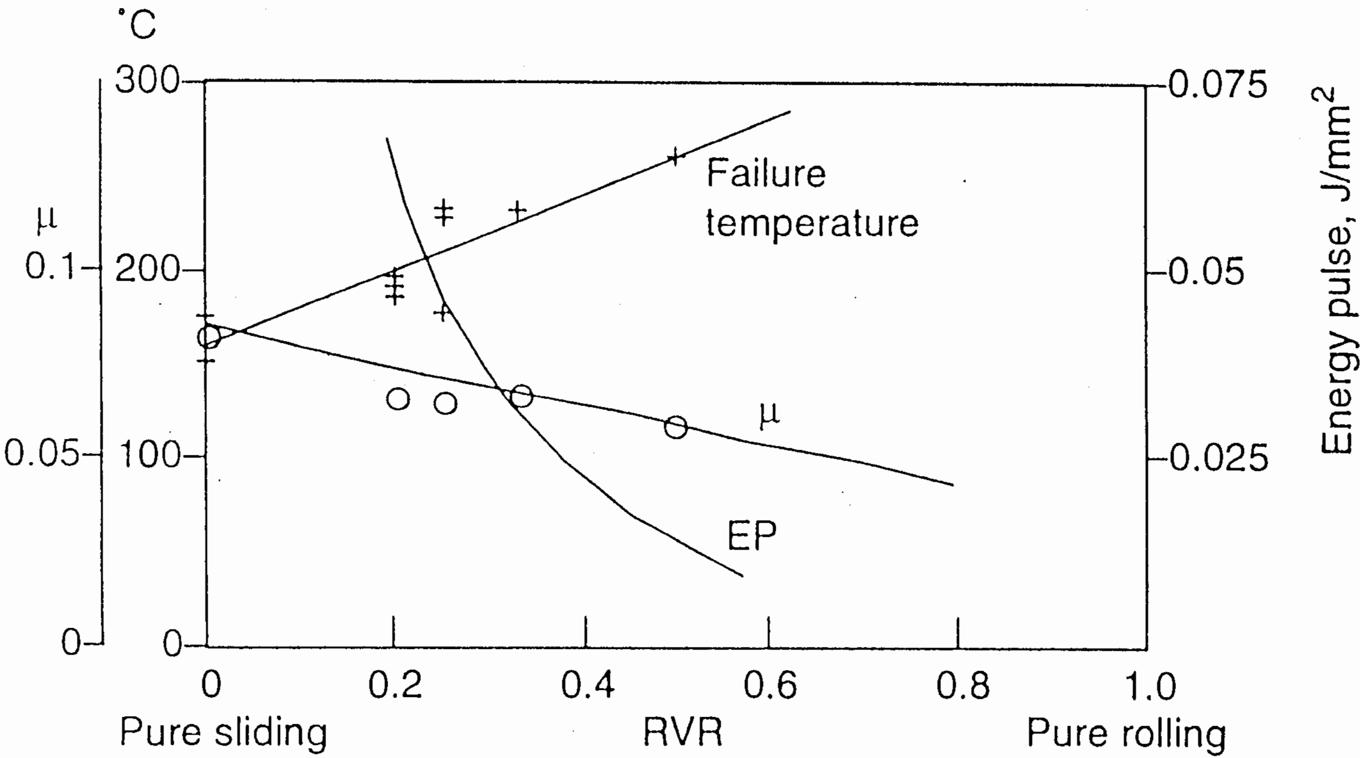


Figure 10: Relation between Scuffing Temperature and Energy Pulse