Measurement and Analysis of High-Temperature
Friction Damper Properties

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1 INTRODUCTION

Given the current status of computer technology it is quite realistic to seek to predict the vibration characteristics of structures having non-linear components. Figure 1, for example, compares measured and predicted non-linear response levels for a two-cantilever beam assembly, with a friction damper bridging between them, [1]. The friction contact was non-linear but the rest of the structure was modelled with linear finite elements. The first-order harmonic balance method was used for these calculations.

A suitable representation of the non-linear elements is obviously required within the computer routine. This paper describes the measurement of friction contact properties used, for example, to predict vibration response characteristics of under-platform dampers, used to control vibration of gas turbine rotor blades.

It would be possible to represent non-linear properties by a look-up table, but it is usually preferable to determine material constants for a model which represents the non-linearity as an analytic function. A hybrid exponential function was used in [1], a simple square law based on a linear asperity model is explored here.

2 FRICTION DAMPER TESTING

The measurement of friction damper characteristics involves a deceptively simple experiment (Figure 2) in which an oscillatory force F is applied parallel to a contact surface with a normal load R, and the oscillatory response, δ, is measured. Assuming, as is quite reasonable, that F is unaffected by the (quite modest) sliding velocity, the data may be represented as a force/displacement hysteresis loop.

Friction vibration dampers have a two-phase cycle of operation:

(i) in the macro-slip phase there is gross sliding of one surface over the other and the damper force should be, at least roughly, proportional to the normal load, in accordance with Coulomb's Law of friction;

(ii) in the 'stuck', or micro-slip, phase there is always some relative slip in at least part of the contact surface, which makes the damper force/relative displacement relation non-linear and results in energy dissipation, even if the motion is entirely within the micro-slip regime.

The test rig used by the authors is sketched in Figure 3. Two test specimens are clamped to horizontal arms: one, the 'fixed' arm, being attached to a rigid support, the other to an electromagnetic shaker, the specimens being loaded normally by a dead-weight arrangement. The (identical) test pieces have 1mm-wide flats, oriented at 90°

so as to give a nominal contact area of $1mm^2$. A number of complicating factors had to be taken into account in setting up the tests:

(i) The excitation produced approximately sinusoidal forcing, whereas what was required was near-sinusoidal (displacement) motion. A substantial mass was interposed on the 'moving' side, which moved under the action of the sinusoidal input force and a (smaller) friction force to approximate the required sinusoidal response and to avoid multiple motion reversals;

(ii) in order to be relevant to turbine blade under-platform dampers, characteristics were required at representative temperatures. The specimens could be heated to temperatures in excess of 1000°C by using four mains-voltage halogen lamps, in close proximity to the specimens, in a fire-brick enclosure;

(iii) the vibration velocity of the hot, moving specimen was measured very close to the contact area, using a laser Doppler vibrometer (LDV) beam, directed through drilled holes in the fixed arm, as indicated. Numerical integration was applied, so as to give motion in displacement terms;

(iv) the damper force was measured as the sum of signals from two force transducers on either side of the 'fixed' arm, outside the heated area, and

(v) in practice, the 'fixed' specimen support was not totally rigid and additional movements were measured with the LDV directed at this specimen. An allowance was made, on this basis, for the 'fixed' side flexibility - of the order of $20x10^{-9}$m/N, depending on temperature.

3 A SIMPLE FRICTION-DAMPER MODEL

In some vibration applications, relative motion between contact friction damper surfaces may be quite large. The macro-slip phase is then dominant, and damper behaviour can be represented quite effectively by a rectangular force/displacement hysteresis loop. This is not the case for the gas turbine under-platform blade dampers where relative movements of the order of only 10 - 20 μm are to be expected.

Experimental hysteresis loops exhibited a noticeable contact stiffness and there was initially some uncertainty as to whether this was genuine, or whether it was the result of some unsuspected flexibility in the test rig, and confirmation was sought from theoretical predictions.

Damper contact surfaces generally exhibit significant roughness, or develop roughness by a process of fretting, and contact may be assumed to be made only at microscopic areas, where surface asperities come into contact. Several models of asperity contact have been devised: the one due to Burdekin *et al* [2] has previous experimental support and will be explored here in some detail.

In the Burdekin model, contact is assumed to be between a plane surface and a number of prismatic rods which all have the same individual normal and shear stiffnesses, k_{ni} and k_{si} respectively, and which obey Coulomb's law of friction, μ being the same for each rod. The rods are assumed to have heights graded linearly so that the number of rods in contact is proportional to the relative deflection of the surfaces measured in the normal direction.

With a given steady normal load, R, on application of a transverse friction force, F, each asperity deforms elastically in the tangential direction until, if the movement is enough, it slips, the transverse force being equal to the limiting friction force for the individual asperity. One can obtain a simple relation for the friction force in the micro-slip phase of a hysteresis loop:

$$F(\delta) = k_c\,\delta - \frac{k_c^{2}\,\delta^{2}}{8\,\mu R} \qquad (1)$$

δ being the relative transverse deflection in the direction of the friction force, with the origin of coordinates at *L* in Figure 4.

The data required to evaluate Equation (1) are, simply: the coefficient of friction, μ, the normal load, *R*, and the contact stiffness, k_c (i.e. $\partial F/\partial\delta$ at the extreme displacement amplitudes, Figure 4).

According to this model, the micro-slip curve is thus a half-parabola, with the vertex at the point, *N*, Figure 4, of macro-slip break-away. This point occurs at a displacement of $4\mu F/k_c$, measured from *L*. The macro-slip phases involve constant friction forces $\pm\mu R$.

If the origin of coordinates is shifted to the macro-slip break-away point, *N*, then, in micro-slip,

$$F = -S\,A_a\,\delta^2 \qquad (2)$$

where *S* is a micro-slip parameter, dependent purely on contact surface properties i.e.

$$S = \frac{k_{si}^{2}\,C}{4\,\mu\,k_{ni}} \qquad (3)$$

A_a is the nominal or apparent area of contact and *C* is a constant, dependent on surface roughness.

S may be determined (from an individual measured hysteresis loop) by:

$$S = \frac{k_c^{2}}{8\,\mu R\,A_a}\;. \qquad (4)$$

Equation (2) implies that the micro-slip part of a hysteresis loop is independent of the normal load, i.e. if a set of loops are superimposed to coincide at macro-slip break-away points, their micro-slip curves will overlay. i.e. the micro-slip factor, *S*, and the coefficient of friction, μ, can be derived from a measurement at one <u>normal load</u>, and these data used to predict hysteresis loops at any other normal load. Also, using Masing's rule, [3], the hysteresis loop for any other <u>amplitude</u> may be constructed, as illustrated in Figure 5

Hence, if the linear asperity theory is obeyed in practice, vibration involving different contact areas, normal loads and amplitudes, cyclic variations in normal load and load position, and two-dimensional relative motion could all be predicted using just these two values.

4 TEST RESULTS

Figure 6 presents an example of a measured hysteresis loop that illustrates a persistent problem in the measurement process: friction force variation in the macro-slip phases. The problem was attributed, largely, to dynamic problems with the rig. The friction force inevitably had discontinuities at the micro/macro-slip transitions, twice per cycle, which, equally inevitably, excited natural modes of vibration of the rig. This unwanted vibration appeared to modulate the applied normal load, indicated by a non-uniform macro-slip force, or a high-frequency squeak, or unbalance between the two force signals. These effects were minimised by running tests at only 100Hz. Higher test frequencies would have been preferred, although these are superfluous if velocity insensitivity can be assumed.

Figure 7(a) illustrates the micro-slip phase of a hysteresis loop which was well modelled by the simple linear asperity model described in Section **3**. The loop in Figure 7(b) did not conform well to this model. Loop (a) was one of a set measured at different normal loads which exhibited micro-slip similarity, as predicted by equation (2), and illustrated in Figure 8(a). However, the second set of loops in Figure 8(b) clearly, do not show the same degree of conformity. The data in Figure 8(b) were measured at 1000°C, but this is incidental as deviations of this type were not obviously associated with a high test temperature. They illustrate a major problem in friction testing: in operation, friction dampers invariably experience wear, which implies the generation of debris and modification in the surface contour, both of which may modify the hysteresis loop randomly.

The environmental temperature can have other important effects on friction damper characteristics. With turbine blade/damper materials, both the coefficient of friction and the micro-slip parameter have been seen to decrease by 60% between room temperature and running temperature. Several temperature cycles may be necessary to reach a stable situation; they presumably affect surface finish, oxidisation and metallurgical state, and debris generation and composition - features which are not dealt with here.

5 CLOSING COMMENTS

In some large-amplitude applications a friction damper contact may, perhaps, be represented satisfactorily just by its coefficient of friction. In most cases, however, experimental evidence suggests that contact stiffness, curvature in the micro-slip force/displacement line, and the corresponding micro-slip energy dissipation have to be included in any mathematical model of a friction damper.

For some friction damper contacts the linear asperity assumptions appear to give as good a model for the contact hysteresis loop as could be expected for any model of such an intractable process. The use of this simple model implies that just one hysteresis loop measurement is sufficient to characterise the contact, and that only two quantities, the coefficient of friction and the micro-slip parameter, are sufficient to represent it in non-linear structural analysis. These advantages are so considerable that it is tempting to suggest that the model be widely adopted, even where the achievable curve-fit is quite poor. Friction-damped structural response predictions are, after all, always approximate because of unavoidable random variations in friction properties within real structures.

6 ACKNOWLEDGEMENTS

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7 REFERENCES

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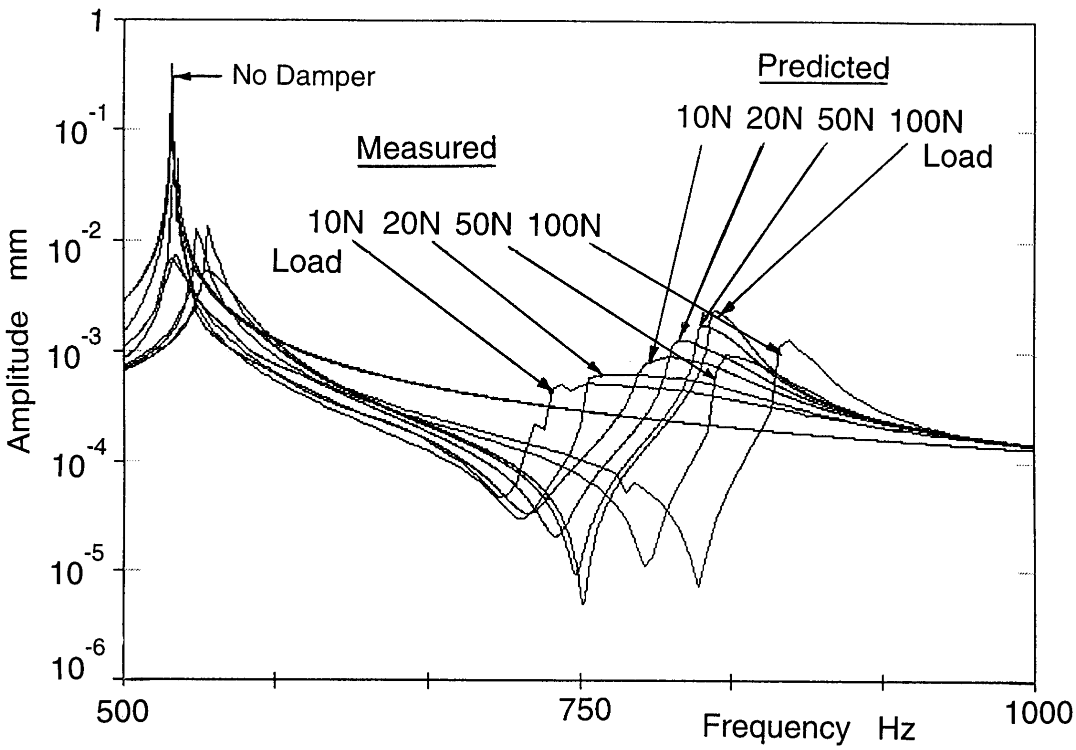

Figure 1 Response levels, measured and predicted, for a 2-cantilever beam rig with an under-platform damper, with 1N excitation force, and various damper loads.

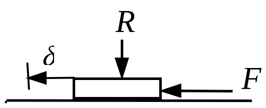

Figure 2 Friction force, F and normal reaction, R.

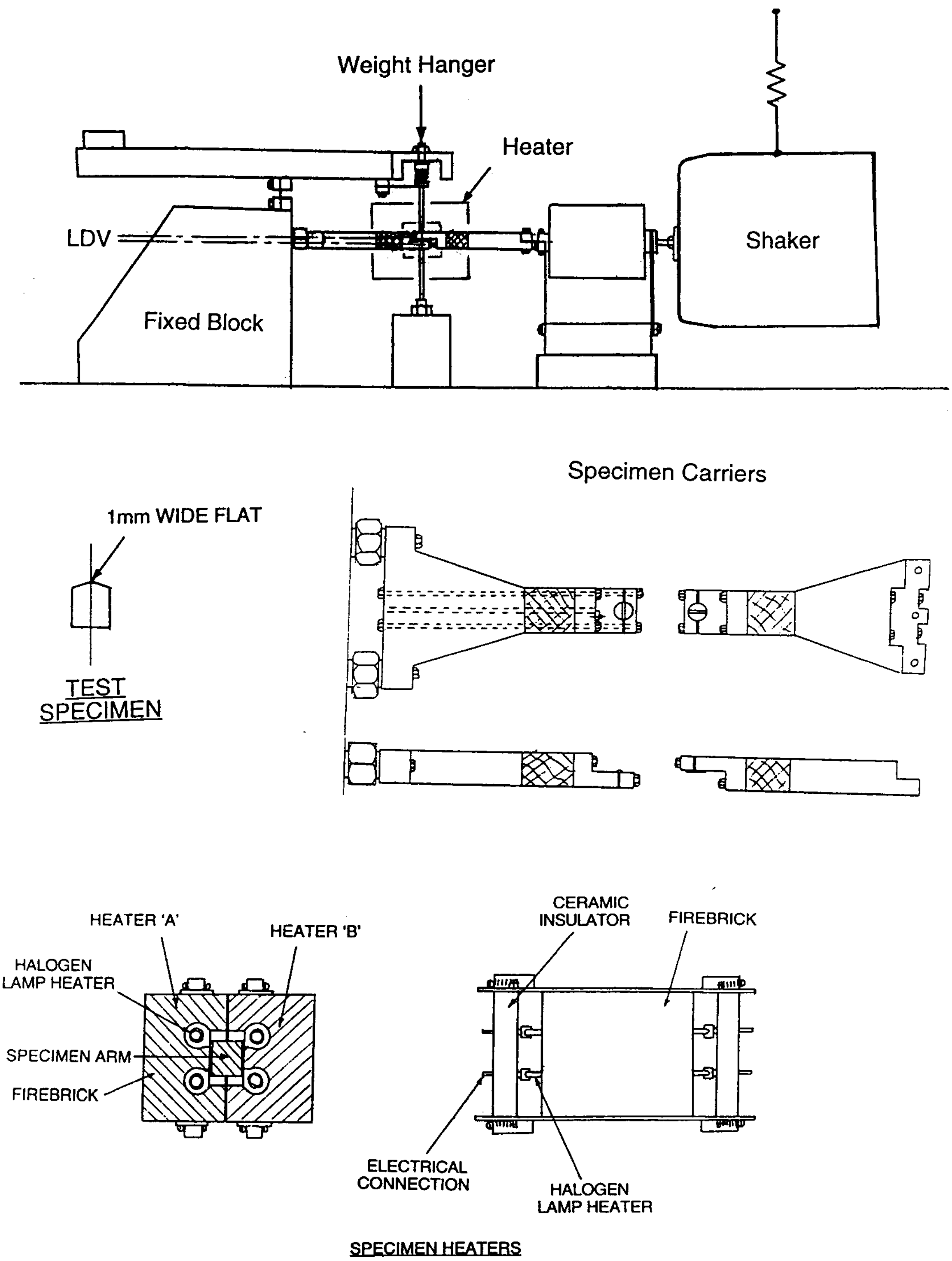

Figure 3 Friction damper test rig.

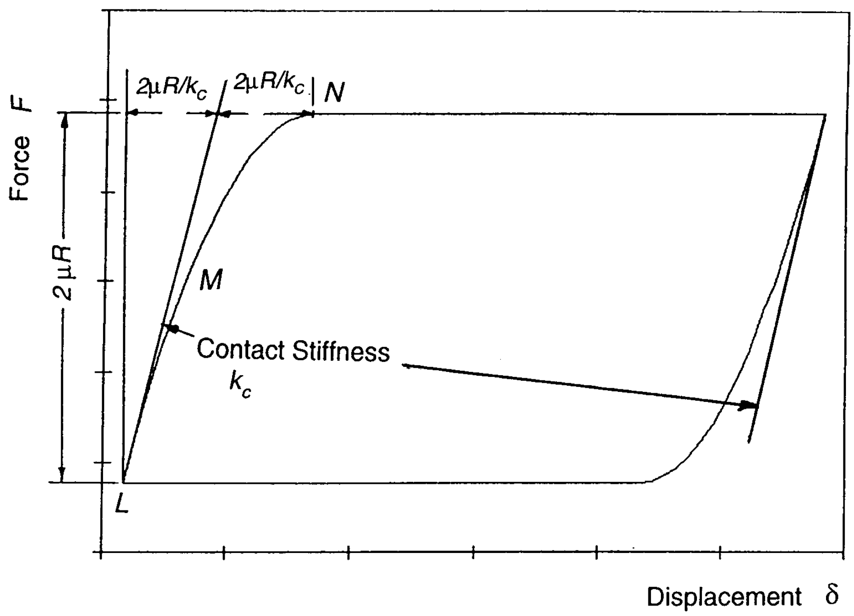

Figure 4 Hysteresis loop based on linear asperity theory.

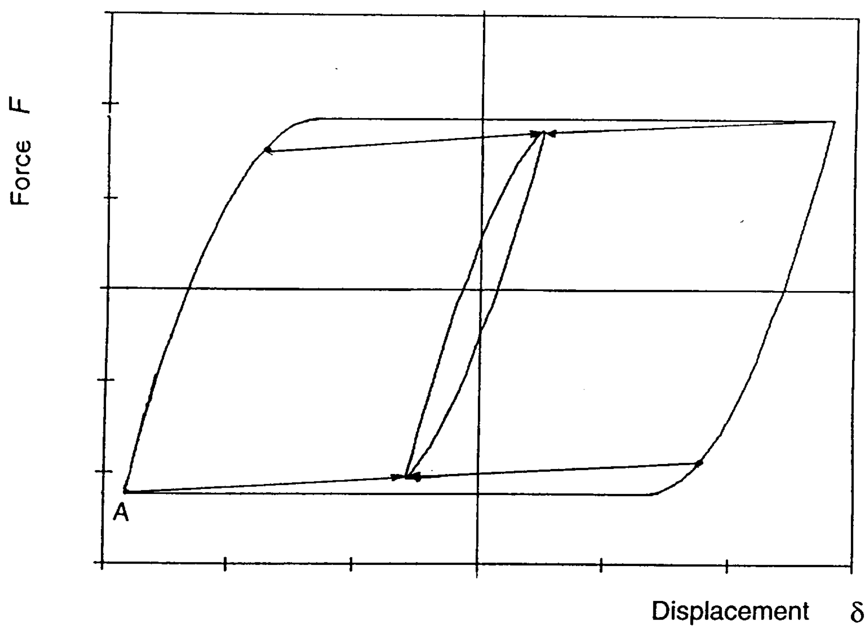

Figure 5 Derivation of small-amplitude hysteresis loop.

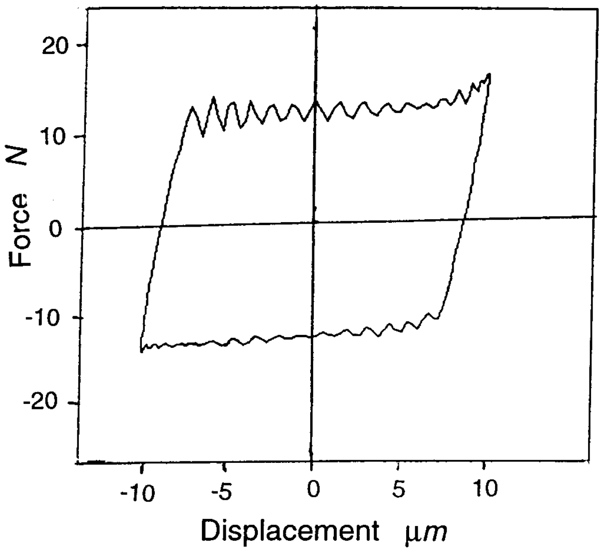

Figure 6   A hysteresis loop (measured at 1000°C) distorted by extraneous dynamic interference.

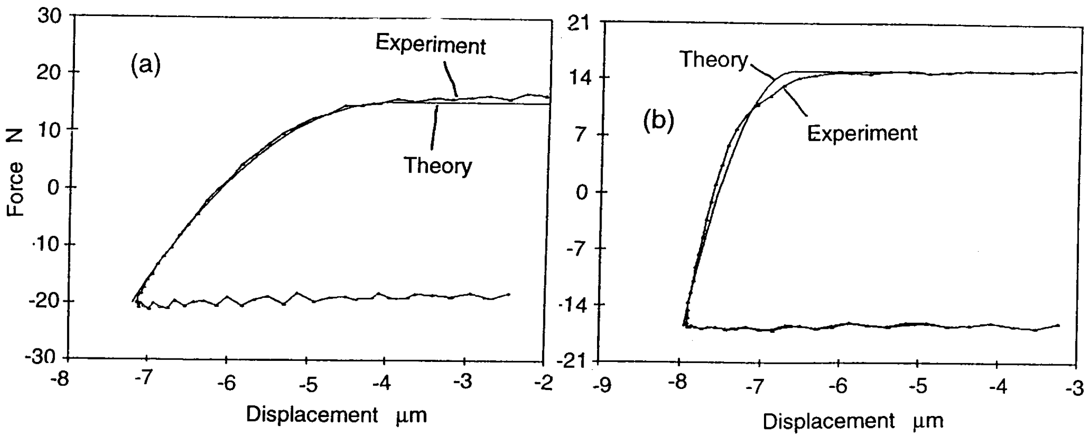

Figure 7   Partial hysteresis loops illustrating good (a), and poor (b), conformity with linear asperity theory.

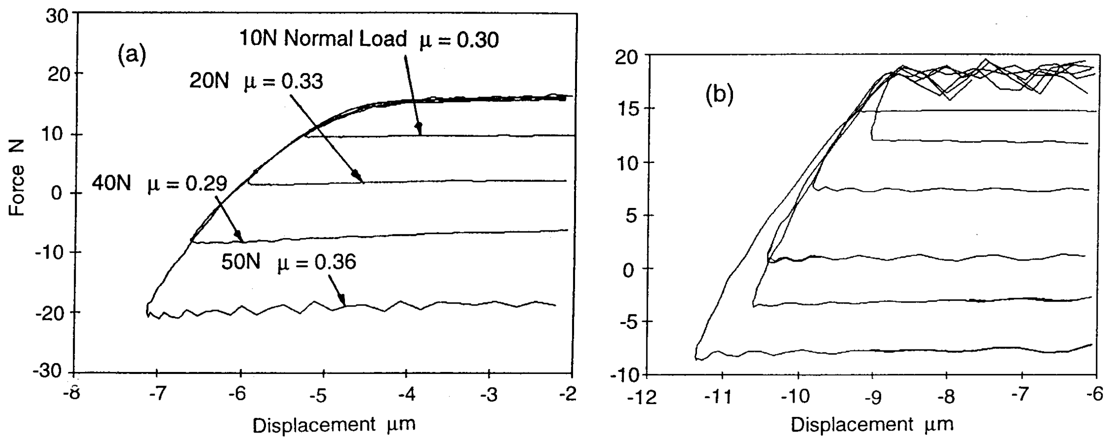

Figure 8 Sets of partial hysteresis loops, offset to illustrate close (a), and variable (b), agreement of micro-slip curves.