EXPERIMENTAL AND THEORETICAL INVESTIGATION OF DAMPING POTENTIAL OF SYSTEM-DEPENDENT TRIBOLOGICAL CONTACTS UNDERGOING TRANSIENT MOTION

Y. K. Younes and N. M. Ashour
Mechanical Design Department, Faculty of Engineering,
and Technology - Mataria, Helwan University, Cairo, Egypt.

ABSTRACT

The experimental and theoretical investigation reported in this paper presents design guides for evaluating the damping capacities of counterformal contact joints subject to an excitation energy. In a free vibration mode,the transient response of a system-dependent tribological contact is found to be affected by the dynamic characteristics of the support system and the frictional properties of the contact damper. The decay time has been taken as a measure for dissipation potential of the imparted initial energy. A support with high natural frequency and high structural damping is able to restore equilibrium position in minimal times irrespective of the frictional forces in the joint. For a low exciting initial energy,contact frictional damping dominates over the structure damping due to the relatively low magnitudes of velocity.

INTRODUCTION

In many mechanical systems,components subject to some frequent relative motion are usually designed to be in contact with each other with different conformity running dry or under boundary lubrication modes. The dynamic behavior of such frictional contacts represents a still poorly understood sub-area of frictional studies,[1]. This is,in large part,due to the fact that relative motions are very system-dependent and each joint or friction testing device may have its unique dynamic response. The considerable practical importance of contact dynamics led many researchers to undertake experimental and theoretical analyses dealing with friction induced vibrations. The squeal and chatter that they produce can cause excessive wear of components,severely damage the surfaces,lead to fatigue failure and may reach objectional noise levels,[2,3]. Apart from frictional problems related to continuous rolling and sliding speeds,transient motions have taken relatively little attention. The sudden transitions in friction that may take place in going from static to sliding conditions are seldom separated from the dynamic response itself. Such transitions occur routinely during start up, stick-slip oscillations and under reciprocating motions. Anand and Soom, [4] analytically investigated the dynamic effects on frictional contacts during acceleration from rest to a steady state velocity. Othman and Seireg,[5] recently experimentally studied the friction properties of Hertzian contacts under reciprocating sliding motion with different frequencies. Brockley and Ko,[6] analysed the running conditions for quasi-harmonic oscillations and showed that they occur when the sliding speed and the normal load are sufficiently high. Dahl,[7] proposed a solid friction damping model to determine the damping characteristics of the flexible support of a simple wire pendulum undergoing free vibrations. Recently, Menq, [8] has presented a model for conformal frictional joints undergoing vibrations on a microscale. In the present work,a theoretical and experimental investigation has been conducted to determine the damping capacities of frictional Hertzian contacts undergoing transient motions. The experimental model used for this purpose, Figures (1), (2), consists of a cylindrical rod attached to a support with adjustable stiffness. The rod is pressed against the rim of a disk with moderate radial loads. Since the contact area is small, the variations in surface roughness are minimum within the frictional path. Initial displacements are given to the rod (actuating energy) and the responses are acquired to determine the decay times under different design characteristics and variable operating conditions. For moderate contact pressures and relatively low sliding speeds and in the absence of flooded lubricant,boundary lubrication is the most probable mode of lubrication. A theoretical model has also been studied in order to describe the dynamics of the sliding rod and its transient response. The problem is formulated as a nonlinear single degree of freedom initial valued problem and the fourth order Runge-Kutta numerical technique has been used to get the solutions in the time domain.

SYSTEM MODELING

The dynamics of the elastically vibrating horizontal rod shown in Figure (1),can be evaluated by considering a single degree of freedom model having its governing equation for a damped motion written as follows:,

$$m \ddot{x} + c \dot{x} + k x = \pm \mu f(\dot{x}) N \quad (1)$$

or

$$\ddot{x} + 2 \zeta \omega_n \dot{x} + \omega_n^2 x = \pm \mu f(\dot{x}) r g \quad (2)$$

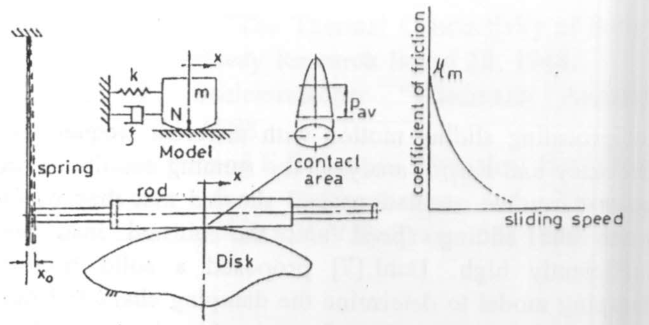

Figure 1. Mathematical model of the system.

where μ f($\dot{x}$) is the instantaneous, velocity-dependent, coefficient of friction at the contact point and r=N/mg is the ratio between the applied normal contact load N and the weight of the rod. Both internal damping and the viscous resistance of the air to the vibrating rod can be expressed by an equivalent damping ratio ζ. This ratio can be evaluated experimentally using the logarithmic decrement technique for the rod vibrating freely without contacting the disk. It has been shown in references [5,9,10] that in the case of stick-slip oscillations, the friction force is time-dependent during stick and velocity-dependent during slip and in the case of quasi-harmonic oscillations,the motion is governed by the velocity-dependent frictional force only. For dry contacts,most researchers adopt the exponential function to represent the friction-velocity relationship, which can be approximated and written in the following form, [9]:

$$\mu f(\dot{x}) = \mu_m e^{-\dot{x}} \quad (3)$$

Where $\dot{x}$ is the sliding velocity and μ_m is the maximum coefficient of friction, Figure -(1). For moderate loads and sliding speeds, with the contact surfaces being wet with a lubricant,the above approximate relation is assumed valid in the present investigation. It should be mentioned that complete agreement on the use of such a velocity-dependent function may not be confirmed. A literature survey reveals the fact that in a boundary lubricated joint,the solid surfaces are so close together that appreciable contact between opposing asperities is possible. The coefficient of friction in such a lubrication mode is determined predominantly by interaction between the solids and between the solids and the lubricant. The bulk flow properties of the lubricant play little or no part in the frictional behavior,[11].

The initial value problem described by the nonlinear equation (2) can be solved numerically to yield the transient responses in the time domain,[12]. In a free vibration mode,the initial energy imparted to the system can be in the form of a stored potential energy in the spring $\dot{x} = x_o (t=0)$ or an initial impulse $\dot{x} = \dot{x}_o (t=0)$. This energy can be expressed as:

$$\Delta E = k x_o^2/2 \quad \text{or} = m \dot{x}_o/2 \quad (4)$$

Either form of initial energies will be dissipated through the work done by the equivalent viscous damping forces described by the system damping ratio ζ and through the contact frictional forces. Along the transient time of sliding,the initial energy will equal:

$$\Delta E = \Delta E \text{ viscous} + \Delta E \text{ contact}$$

$$= \int_0^{td} (c\dot{x}) dx + \int_0^{td} \mu \, r m g dx$$

$$= m \left(\int_0^{td} 2 \xi \omega_n \dot{x} \, dt + \int_0^{td} \mu \, r g x dt \right) \quad (5)$$

where td is the decay time elapsed from t=0 to a complete stop of the vibrating system.
The dissipation ratio can be expressed as:

$$R = \Delta E \text{ viscous} / \Delta E \text{ contact} \quad (6)$$

An experimental evaluation of this ratio is difficult although a theoretical estimate can be achieved by performing the integrals expressed in equation (5). Both parts of the dissipated energy are velocity-depended. But, while the first part is directly proportional to $\dot{x}$, the second part decreases with an increase in the sliding velocity. Thus, control of the sharing ratio is difficult to achieve and for that reason,each joint design may have its unique dynamic response depending on the link stiffness and the tribological conditions at the contact.

TEST APPARATUS AND PROCEDURE

The purpose of the developed test apparatus,shown in Figure (2) is to carry out an experimental investigation on the transient dynamic behavior of a system-dependent sliding frictional contact. The sliding mass consists of a horizontal cylindrical rod 16 mm diameter and 610 mm long, put into tangential with the edge of a fixed steel disk,(crossed cylinders arrangement). The rod is free to vibrate along its axial direction against a vertical leaf spring fixed from one of its ends to the rigid frame. In addition to the rod weight,the normal contact load can be increased by a wire-pulley system. Selection of different spring stiffness can be made by adjusting the operating length of the steel leaf and by using different strip thicknesses. A BK type 4370,10 pc/ ms piezoelectric accelerometer is mounted on the rod end to acquire acceleration signals during the sliding motion. The output signals are amplified by a conditioning amplifier then fed to the BK 2034 FFT analyser for real time displays. Steel and aluminium rods have been tested ,running wet with XHP 20W-50 commercial engine oil. The system equivalent vibrating masses are 0.357 Kg. (for aluminium rod) and 0. 679 Kg. (for steel rod) which includes the mass of the accelerometer. The steel disk edge and rod surfaces were prepared mechanically by fine turning and grinding by a 400 grit abrasive paper yielding a surface roughness within the range of 0. 5 to 0.8 μm (centerline average). Before each test,the surfaces are cleaned with acetone and hand ground to restore surface quality. Initial axial displacements are given to the rod and the transient responses are acquired for each operating condition. Maximum coefficient of friction can be evaluated by rotating the disk until sliding of the rod is observed. The disk angular displacement provides a measure for the spring maximum deflection and force after being calibrated for its stiffness.

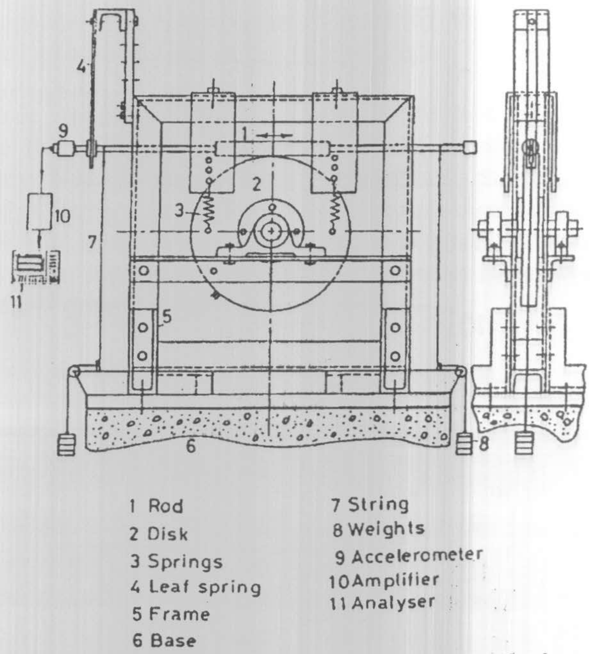

Figure 2. Layout of the test rig.

DISCUSSION OF RESULTS

The ability of a system-dependent tribological joint to restore equilibrium position,after being excited has been found to be affected by three main design factors. Namely :the natural frequency of the support system,its structural damping factor and ,finally, the frictional properties of the contact. Practically, apart from using additional damping elements,and for a specific system design,little can be made to control the first two factors. As illustrated in Figure (3), systems possessing a high natural frequency i.e. stiff structures restore their equilibrium positions faster than those with low natural frequencies. During the transient period,the imparted initial energy will be dissipated at a high rate through multiple strokes of relatively large amplitudes. Referring to Figure (4),in systems with negligible structural damping,the increase of the contact coefficient of friction to a value of 0. 1, enables the structure to dissipate the initial energy in one cycle. This condition cannot be achieved with a fully flooded film lubrication mode where coefficients of friction have much lower values. On the other hand,a dry joint may exhibit stick-slip phenomenon with probable high noise levels and high wear rates when subjected to transient loads.

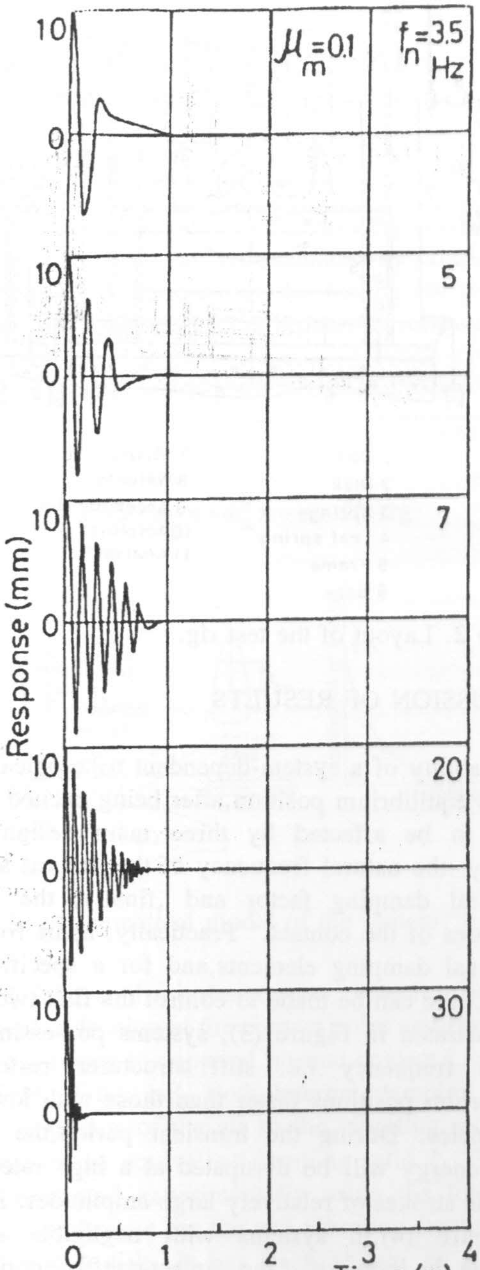

Figure 3. Transient responses of system having different natural frequencies (m=0.5 kg., ζ=0).

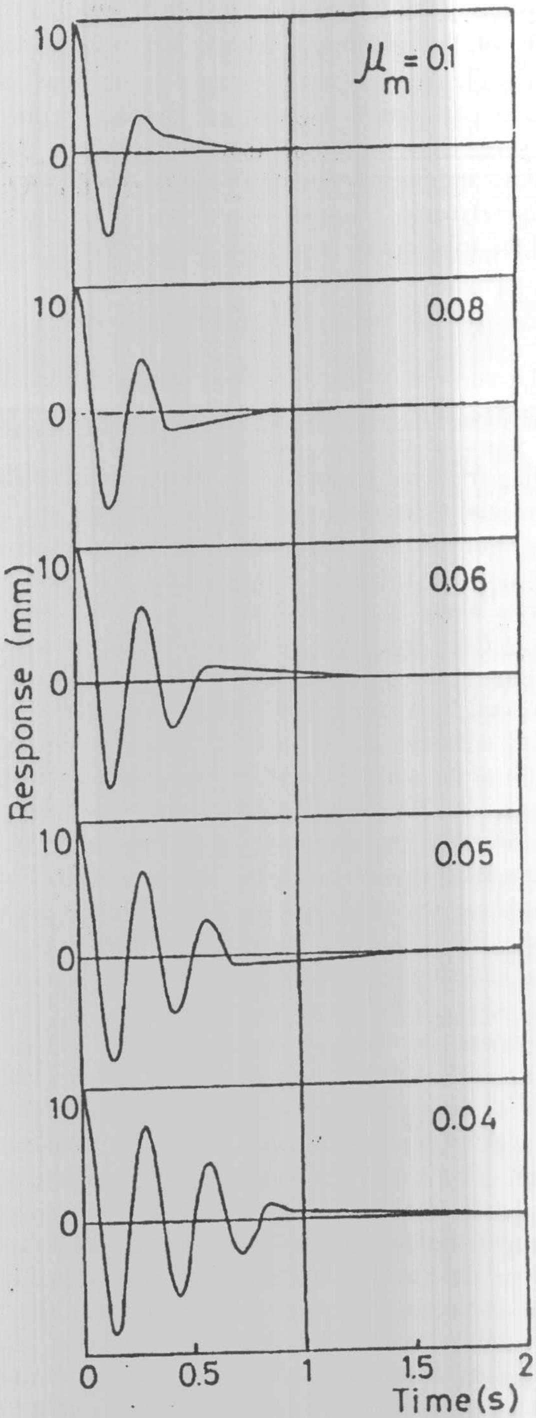

Figure 4. Transient responses of a low natural frequency system sliding with different coefficient of friction (f_n=3.5, m=0.5 kg., ζ=0).

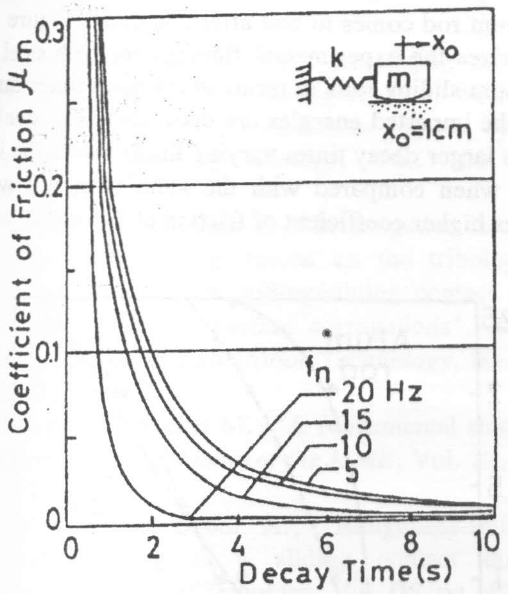

Figure 5. Variation of the decay time with the coefficient of friction at contact for systems having different natural frequencies.

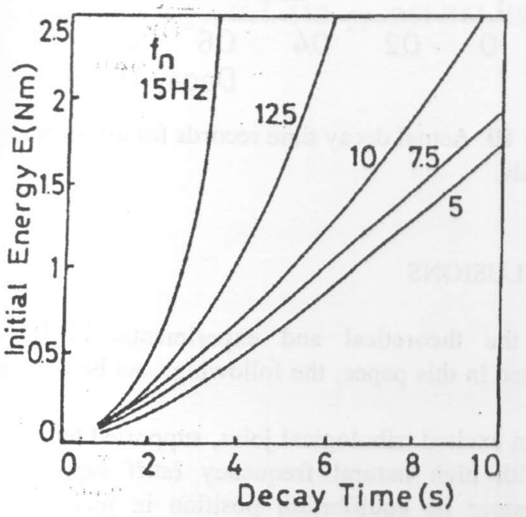

Figure 6. Variation of the decay time with the initial energy imparted to the system (μm=0.1, ζ=0).

Figure(5) summarizes these observations, where a system with ζ= 0 has been given an initial displacement of 10 mm and let to vibrate freely under the sole action of the contact frictional forces. Decay times are shown to be less than two seconds for a large range of natural frequencies and for a μm value equal to 0.1, although initial energies are not equal for all cases,(ΔE=0.5 Kx_0^2). It is interesting to mention here that this class of problems can be referred to as initial energy problems, beside being initial valued problems. When the energy dissipation time is concerned, Figure (6), the decay time is not affected by the initial values. Figure (7) shows the variation of the energy dissipation ratio between the system damping and the contact friction and the decay time for different initial energy levels imparted to the system. For systems having a high structural damping and at high energy levels,the joint frictional characteristics have little to do in restoring equilibrium positions and the role of the mechanism link design dominates in achieving minimal decay times.

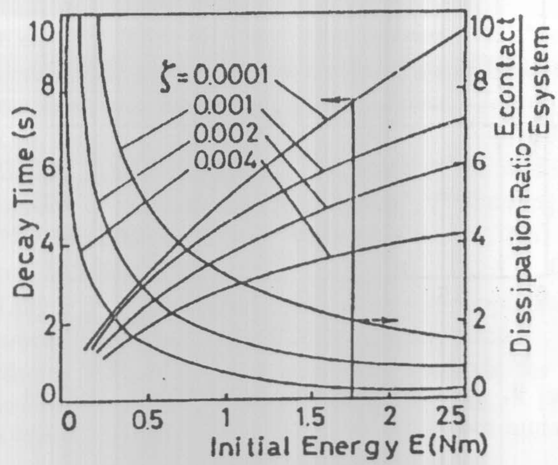

Figure 7. Variation of the dissipation ratio and decay time for different damping ratios ζ (f_n=10 Hz).

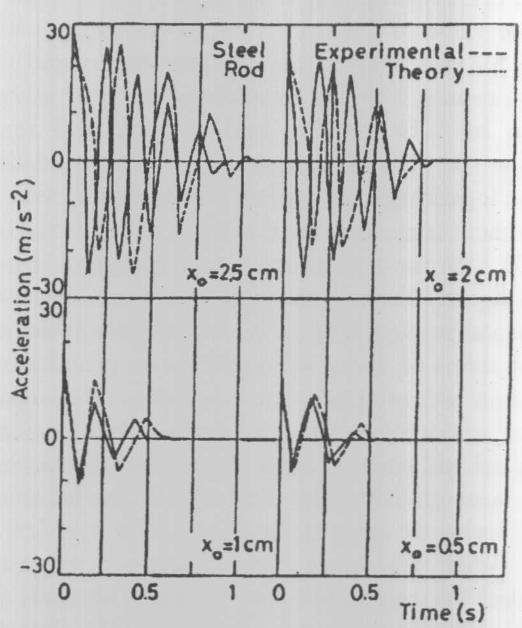

Figure 8. Actual transient acceleration records for steel rod.

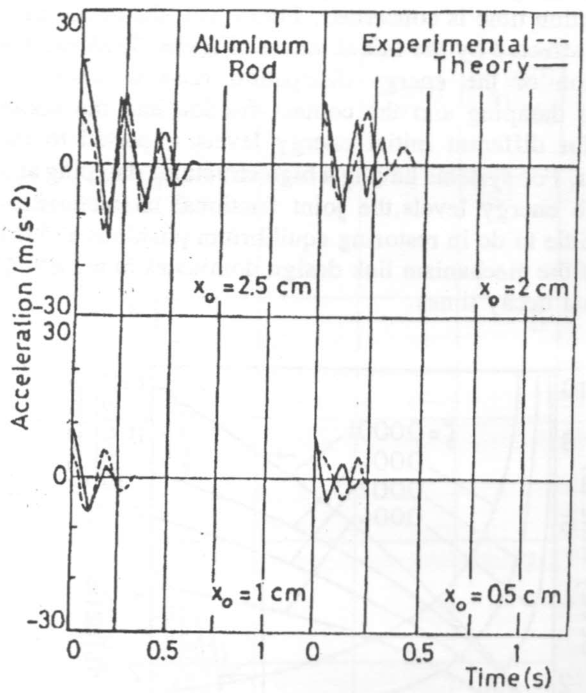

Figure 9. Actual transient acceleration records for aluminium rod.

Figures (8) and (9) show typical experimental results obtained for lubricated sliding steel and aluminum rods, and the analytical results obtained by solving numerically equation (2). A search has been carried out to obtain the maximum value of the coefficient of friction μm which provides the closest agreement between theory and actual repetitive results. For the considered cases μm values are found to be 0.19 for the steel rod and 0.3 for the aluminium rod. The damping ratio ζ has been calculated using the logarithmic decrement for the system vibrating freely without contacting the fixed disk and is found equal to 0. 009 with the natural frequency being equal to 5.3 Hz. The figures illustrate the good agreement between the experimental responses and the corresponding theoretical results in terms of decay times and the magnitudes of the acceleration values. Complete coincidence between the two plots could not be obtained for all cases, especially for the steel rod when given a large initial displacement. This is, most probably, due to the nonlinear behavior of the leaf spring at large deflections. After the first two cycles, (when the spring is vibrating with smaller amplitudes) the correlation between plots becomes more acceptable. The same goes for the case of the aluminium rod,but in this case the damping, due to the high frictional properties at the contact, brakes the system in fewer cycles. For small initial displacements th aluminium rod comes to rest after one cycle. Figure (1(summarizes the experimental findings for both steel an aluminium sliding rods in terms of transient times durin which the imparted energies are dissipated. The steel ro exhibits larger decay times varying linearly with the inpu energy when compared with the aluminium rod whic provides higher coefficient of friction at the contact area

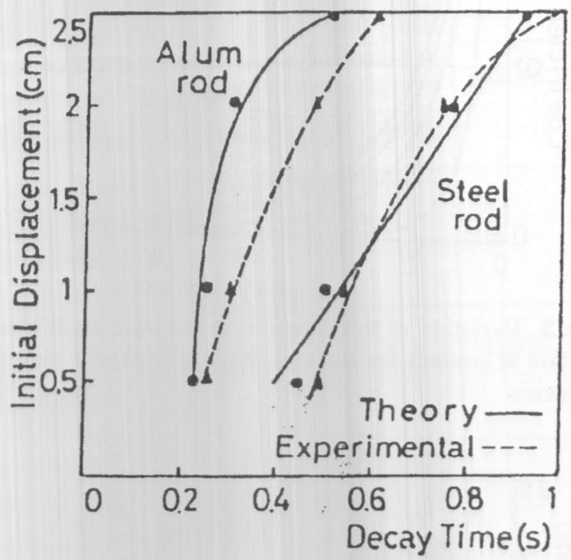

Figure 10. Actual decay time records for different slidin materials.

CONCLUSIONS

From the theoretical and experimental investigatio presented in this paper, the followings can be concluded

1. An excited tribological joint, supported by a systen with high natural frequency (stiff support), wil restore its equilibrium position in minimal time when compared with an elastic support.
2. The contact frictional properties play small role i damping the system motion when the suppor possesses a high structural damping.
3. For low levels of excitation energy,the frictiona contact dissipation potential dominates over that o the system damping.
4. The exponential form of the coefficient of frictio variation with the sliding speed could be used i modeling lubricated Hertzian contacts at moderat loads.

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