

A Novel Measurement of Dynamic Friction

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ABSTRACT

A tribometer, based on a pin-on-disc machine, uses a PZT drive to produce small sinusoidal fluctuations of sliding speed. The frequency and amplitude of these fluctuations can be controlled, and the dynamic response measured. Preliminary test results show that the dynamic friction variation is influenced by the contact materials, normal force, oscillation frequency and steady sliding speed. The variation of friction force amplitude and phase with frequency gives clues about the underlying state variables determining the friction. Modelling studies illustrate the expected behaviour for idealized friction laws governed by, for example, sliding speed, contact temperature, and “rate-state” laws.

INTRODUCTION

Brake squeal causes serious noise and instability problems in automobiles and it has drawn attention from engineers and researchers for many years. Many theories of brake squeal have been developed over the decades. Mills [1] first proposed the “falling friction curve” theory of brake squeal in 1938. This theory advocated that a necessary condition for a brake to squeal was a coefficient of friction which decreases with increasing sliding speed. This theory has received less attention in recent years [2] because it is only considered to be important for brake noise less than 100 Hz [3].

Spurr [4] developed his “sprag-slip” theory to explain brake squeal in 1961. Contrary to Mills’ theory of brake squeal, Spurr’s theory emphasised the role of the geometry of the contact (braking) system in the changing of coefficient of friction and leading to possible instability. Following Spurr’s idea, Jarvis and Mills [5], Earles and co-workers [6, 7, 8, 9, and 10], North [11], and Millner [12] developed various multi-degree-of-freedom models to predict the instability or squeal of different braking systems.

North [11] first considered brake squeal as a self-excited vibration induced by friction forces with constant coefficient of friction in 1972, and he modelled the friction force as a so-called “follower force”. Similar to North [11] in assuming constant coefficient of friction and self-

excited vibrations, Ouyang et al [13] developed a finite element model for a whole braking system considering the effects of the rotation of the disc rotor in 1999. He reached the conclusion that brake squeal is the consequence of unstable parametric resonance induced by the rotating dry friction between the disc and pads.

Rhee et al [14] proposed a mechanical impact model called “hammering” to explain noise generation in braking systems. This model does not rely on friction variation to explain the excitation of noise and vibration, but stick-slip can be treated as a secondary mechanism of hammering within this impact model. Rhee et al also remarked that hammering was in fact compatible with sprag-slip instabilities.

Duffour and Woodhouse [15, 16] studied the stability of frictional systems with a single point of sliding contact. They carried out analysis to investigate and classify the various possible routes to instability which can be formulated within linear theory. Stability criteria were derived in terms of transfer functions defined at the contact point, and the consequences of these criteria were investigated numerically by simulating the behaviour of generic systems.

Despite a long history of research, theories of brake squeal are still not completely convincing, and cannot be reliably used to predict brake squeal. One ingredient of any instability or squeal prediction is the assumed constitutive law for the frictional interaction. Duffour and Woodhouse [15, 16] stated that in the initial stage as the system starts to be unstable as sketched in Fig. 1, the dynamic coefficient of friction may be expressed as a Taylor approximation:

$$\mu(\omega) \approx \mu_0(\omega) + \varepsilon(\omega)v' \quad (1)$$

where $\mu(\omega)$ and μ_0 are the instantaneous and steady coefficients of friction, respectively; ω is the angular frequency; v' is the oscillation speed of a sliding pin; and $\varepsilon(\omega)$ denotes the slope of the straight line characterizing the linearised relation between the coefficient of friction and the pin oscillation speed. By allowing $\varepsilon(\omega)$ to be complex and to vary with frequency, it can express any linearised frictional law, whatever

internal state variables may be involved (e.g. contact temperature).

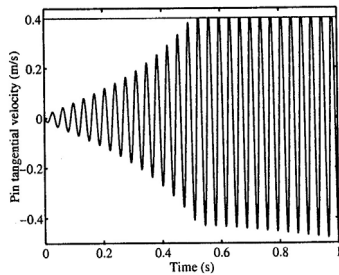


Figure 1 Pin tangential velocity from a simulation of the non-linear behavior of a pin-on-disc system when the underlying linear system is unstable. After an initial transient phase the system settles into a periodic regime of stick-slip oscillation [17].

Numerical simulation results by Duffour [17] and Butlin [18] show that $\varepsilon(\omega)$ has a very significant effect on the stability of a point-contact sliding system. There would be no hope of reliable prediction of squeal instability without knowing $\varepsilon(\omega)$, but it is a quantity which has never previously been measured. To this end a novel high-frequency dynamic tribometer has been developed. Several contact materials were tested at a range of frequencies, normal loads, and disc rotation speeds.

EXPERIMENTAL DESIGN AND TESTING

PRELIMINARY TEST RIG

Before the design of the high-frequency dynamic tribometer (HFDT), feasibility tests were carried out using a preliminary test rig. The conceptual design of this preliminary rig is shown schematically in Fig. 2. A dynamometer with a screwed-in pin made of the desired material was driven by a vibration actuator that generated oscillation of the pin. To allow this vibratory movement, the dynamometer assembly is mounted on a platform supported on leaf springs that permits motion in one direction but is stiff in other directions. The pin was pressed against a moving disc that was driven by a variable speed motor. The disc can be made of any desired material. To allow the pin to be brought into and out of contact with the disc, a second leaf-spring assembly was used, perpendicular to the first one. This mechanism also allows the normal force to be controlled, via a screw system acting through a coil spring. The stiffness of the rig could be adjusted by changing the length and thickness of the leaf springs.

The setup of the preliminary test rig is photographed in Fig. 3. The rotating steel disc and the dynamometer made of aluminium alloy holding a small polycarbonate pin can be seen in the picture. While the disc rotates, a prescribed tangential sinusoidal oscillation was applied to the pin-dynamometer subsystem using a moving-coil

vibrator, also visible in Fig. 3. Three of the four adjustable leaf springs associated with the vibrator are clearly visible, while those adjustable leaf springs associated with the normal force are not visible in Fig. 3. The coil spring through which the normal load is applied can be seen on the right-hand edge, behind the grey wires in Fig. 3.

For initial tests, a steel disc was used that was 15.2 cm in diameter and 1.8 cm thick, which could rotate at speeds up to 250 rpm driven by the motor (A.C. Motor Type 2128 HB, Hoover Limited). The shaft axis is horizontal, and the disc rotates in a vertical plane. The vibrator was computer-controlled using a National Instruments 6024E I/O card. The acceleration of the pin was measured using the accelerometer (Model DJR A/25/E 460) as shown in Fig. 3. The normal and tangential contact forces were measured using the dynamometer, which can also be seen in the middle of Fig. 3: details of the design of the dynamometer can be seen in [19].

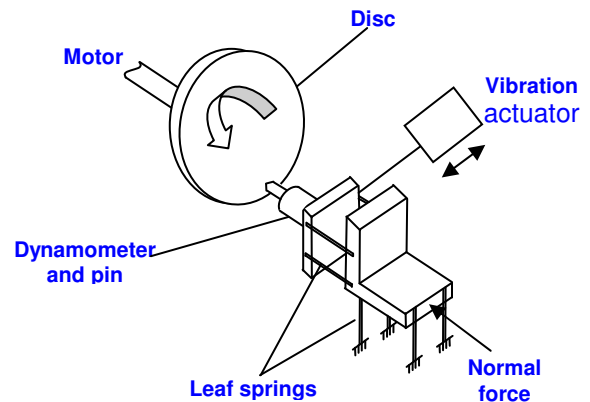


Figure 2 Conceptual design of the test rig

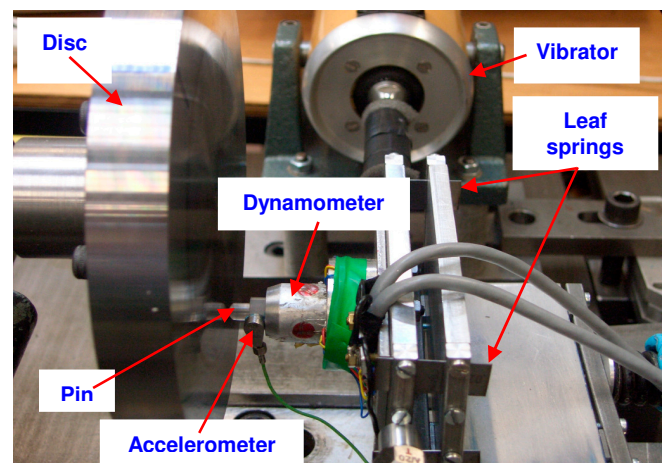


Figure 3 Setup of the preliminary test rig

CALCULATION OF $\varepsilon(\omega)$

Since the coefficient of friction may vary with sliding velocity and frequency, the instantaneous frictional force

may also vary with sliding velocity and frequency. The frictional force $F(\omega)$ may be written in terms of a constant term and a fluctuating component as:

$$F(\omega) = F_0 + F'(\omega) \quad (2)$$

Assuming that the normal force is a constant, i.e. $N(\omega) = N_0$,

$$\mu(\omega) = \frac{F_0 + F'}{N_0} = \mu_0 + \frac{F'}{N_0} \quad (3)$$

Comparing with (1), $\varepsilon(\omega)$ may be written as:

$$\varepsilon(\omega) = \frac{F'(\omega) / N_0}{v'(\omega)} \quad (4)$$

The value of $\varepsilon(\omega)$ can thus be deduced from the complex amplitudes of FFTs of the measured force and velocity waveforms, evaluated at the excitation frequency. Averaging procedures may be used to improve signal-to-noise ratio.

PRELIMINARY MEASUREMENTS

Problems with resonance or squeal were encountered when this preliminary test rig was operated outside a certain range of the operational parameters. To find out these limitations, the natural frequencies and mode shapes of the pin-on-disc system were obtained by using modal testing techniques as described in [19]. It was found that the normal contact force has a significant effect on the resonant frequencies of the system. The friction-induced vibration of the contact system was also affected by the disc rotation speed and the contact materials. However, by keeping these parameters within a restricted range, measurements of $\varepsilon(\omega)$ could be obtained.

Three pins (polycarbonate, nylon, and PTFE) and three discs (steel, aluminium, and glass) were used in the preliminary tests. The maximum disc rotational speed was limited to 60 rpm and the maximum normal contact force was limited to 50 N to avoid serious vibration or squeal of the pin-on-disc system. Repeated measurements of ε were carried out at each frequency to check the repeatability of the measurements.

A typical $\varepsilon(\omega)$ measurement result for a polycarbonate pin on a glass disc is shown in Fig. 4. It can be seen that the modulus of $\varepsilon(\omega)$ below 600 Hz was larger than it was above 600 Hz, which may due to the resonances below 600 Hz of the preliminary test rig. As a result, the values of $\varepsilon(\omega)$ below 600 Hz were less reliable than those over 600 Hz. Beyond 600 Hz, the modulus of $\varepsilon(\omega)$ for the polycarbonate pin increased with frequency;

however, this was not observed from the measurement results for the nylon and PTFE pins. Figure 5 demonstrates the measured $\varepsilon(\omega)$ for a PTFE pin with a glass disc.

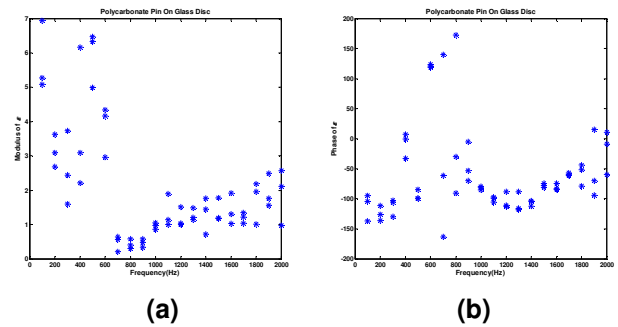


Figure 4 (a) Modulus and (b) Phase of $\varepsilon(\omega)$ of polycarbonate pin on glass disc at steady sliding speed 0.1 m/s, $v'/v_0 = 5\%$ at 2 kHz, normal force 15 N

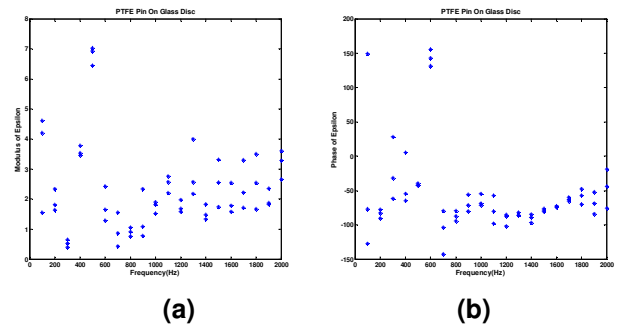


Figure 5 (a) Modulus and (b) Phase of $\varepsilon(\omega)$ of PTFE pin on glass disc at steady sliding speed 0.155 m/s, $v'/v_0 = 2\%$ at 2 kHz, normal force 15 N

DEVELOPMENT OF THE NOVEL HIGH-FREQUENCY DYNAMIC TRIBOMETER

Guidelines for the Design of the High-Frequency Dynamic Tribometer

Based on the investigation of the preliminary test rig, the following guidelines for design of the novel High-Frequency Dynamic Tribometer (HFDT) were developed:

- A smoother drive was needed for the disc, both in terms of speed constancy and the alignment of the disc;
- A stiffer pin-dynamometer subsystem was needed to avoid self-excitation at frequencies below 2 kHz;
- A stiffer actuator was required to drive the pin subsystem sinusoidally and the pin oscillating speed needed to be controlled and measured accurately;