Journal of Sound and Vibration 332 (2013) 5302–5321

Contents lists available at SciVerse ScienceDirect

Journal of Sound and Vibration

journal homepage: www.elsevier.com/locate/jsvi

Friction-induced vibration: Model development and comparison with large-scale experimental tests

T. Butlin *, J. Woodhouse

Cambridge University Engineering Department, Trumpington Street, Cambridge CB2 1PZ, England, United Kingdom

ARTICLE INFO

Article history:
Received 29 March 2012
Received in revised form
22 April 2013
Accepted 29 April 2013
Handling Editor: L.N. Virgin
Available online 29 May 2013

ABSTRACT

This paper presents a comparison between theoretical predictions and experimental results from a pin-on-disc test rig exploring friction-induced vibration. The model is based on a linear stability analysis of two systems coupled by sliding contact at a single point. Predictions are compared with a large volume of measured squeal initiations that have been post-processed to extract growth rates and frequencies at the onset of squeal. Initial tests reveal the importance of including both finite contact stiffness and a velocity-dependent dynamic model for friction, giving predictions that accounted for nearly all major clusters of squeal initiations from 0 to 5 kHz. However, a large number of initiations occurred at disc mode frequencies that were not predicted with the same parameters. These frequencies proved remarkably difficult to destabilise, requiring an implausibly high coefficient of friction. An attempt has been made to estimate the dynamic friction behaviour directly from the squeal initiation data, revealing complex-valued frequency-dependent parameters for a new model of linearised dynamic friction. These new parameters readily destabilised the disc modes and provided a consistent model that could account for virtually all initiations from 0 to 15 kHz. The results suggest that instability thresholds for a wide range of squeal-type behaviour can be predicted, but they highlight the central importance of a correct understanding and accurate description of dynamic friction at the sliding interface.

1. Introduction

Experimental validation of predictive models for friction-induced vibration has proved a very elusive goal over many years of research. One of the most regularly cited reasons is the difficulty in obtaining repeatable results from measurements (e.g. [1–3]). In addition, frictional instability (or 'squeal') can be caused by a wide range of mechanisms, such as linear instability, thermoelastic effects, or particular initial conditions that promote stick-slip motion. Matching the scope of a theoretical model to the actual mechanisms that cause observed instabilities is a challenging task.

The wide variety of physical effects that can lead to friction-induced vibration is highlighted by the many different models in the literature. These range from lumped parameter models that exemplify basic causes of squeal (sometimes described by terms such as 'mode-coupling' or 'negative damping': see Von Wagner [4] for a discussion of minimal models) to sophisticated finite-element models that include many physical effects, each of which potentially introduces new routes to instability (such as gyroscopic effects: e.g. Hochlenert et al. [5]; or thermoelastic instabilities: e.g. Afferante et al. [6]). A summary of these approaches can be found in the reviews by Kinkaid [7], Ouyang et al. [8] and Sheng [9].

* Corresponding author. Tel.: +44 1223 748522.
E-mail addresses: tb267@cam.ac.uk (T. Butlin), jw12@cam.ac.uk (J. Woodhouse).

0022-460X/$ - see front matter
http://dx.doi.org/10.1016/j.jsv.2013.04.045

There have been several examples of experimental studies for which good agreement with theoretical predictions has been obtained (e.g. [3]). Such papers represent encouraging progress, but close reading reveals that the level of agreement has not been achieved without a great deal of care in manufacturing and pre-conditioning for tests (such as running in of the frictional contact). This provides reassurance that some level of prediction may realistically be within grasp, but highlights the need for further study into the high sensitivity of friction-coupled systems.

In previous work [10–12], the sensitivity of theoretical predictions to parameter changes and model choice has been explored extensively. This work focussed on the linear instability route to 'squeal': the most widely considered in the literature and a logical starting point. Contact is considered to occur at a single point, but the theoretical model is otherwise very general. The numerical tests were based on transfer function measurements of a pin-on-disc test rig: details of these measurements, quantification of the inherent uncertainties and the effect of these on theoretical predictions are all described in Butlin and Woodhouse [13].

A large-scale series of squeal tests was described in detail in Butlin and Woodhouse [14]: the key aims were to test repeatability, experimentally explore sensitivity to perturbations and enable direct comparison with theoretical predictions. The analysis presented in [14] was entirely data-driven and no attempt was made to interpret the results in the light of theoretical models.

This paper presents a comparison between these results and predictions from the theoretical model (developed by Duffour and Woodhouse [15,16]). The full set of measured results is undeniably complicated. Earlier papers have presented and discussed particular subsets (e.g. [11]), but the purpose of the present paper is to address the question: 'is it possible to account for *all* the observed instabilities within the framework of the linearised theory that will be presented in Section 2?'. Before tackling this question, an overview of the theoretical model is described and extended to accommodate a more general description of linearised friction laws.

2. Model outline

2.1. System dynamics modelling

The model previously used for theoretical predictions is described in detail by Duffour and Woodhouse [15,16] and Butlin and Woodhouse [10]: it is updated here to allow for a more general linearised friction model. The system to be analysed is sketched in Fig. 1. A 'disc' is driven at steady velocity V_0 and a 'brake' is pushed against it with a dynamically varying normal force N, composed of a steady equilibrium pre-load N_0, plus a small fluctuating component N' such that $N = N_0 + N' e^{i\omega t}$. Similarly, the force tangential to the sliding direction due to friction, F, can be expressed as a steady equilibrium force F_0 plus a small fluctuating component F' such that $F = F_0 + F' e^{i\omega t}$. The normal and tangential displacements (associated with small departures from the state of steady sliding) of the disc at the point of contact (marked with a dot) are denoted x_1 and y_1 respectively, and x_2 and y_2 for the brake. The contact stiffnesses in the tangential and normal directions are illustrated as lumped together with the 'brake' dynamics, and the contact region could be defined either to exclude these springs (region I) or to encompass them (region II). The definition of x_2 and y_2 follows from the choice of contact region. The sign convention defined in Fig. 1 has been chosen to be consistent with Wang and Woodhouse [17] and because it is more intuitive than the previously used convention from Butlin and Woodhouse [10]. For steady running with V_0 positive, F and N are both positive.

The linear dynamics of the disc and brake can be described in terms of transfer functions:

$$\begin{bmatrix} x_1 \\ y_1 \end{bmatrix} = \begin{bmatrix} G_{11}(\omega) & G_{12}(\omega) \\ G_{21}(\omega) & G_{22}(\omega) \end{bmatrix} \begin{bmatrix} N' \\ F' \end{bmatrix}, \quad \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} = \begin{bmatrix} H_{11}(\omega) & H_{12}(\omega) \\ H_{21}(\omega) & H_{22}(\omega) \end{bmatrix} \begin{bmatrix} N' \\ F' \end{bmatrix} \tag{1}$$

where $G_{ij}(\omega)$ represent the disc's response and $H_{ij}(\omega)$ represent the equivalent responses for the brake. These transfer functions can be determined using standard vibration measurement techniques. The convention of the vibration literature is followed by using transfer functions defined as the Fourier transform of an impulse response, rather than the Laplace

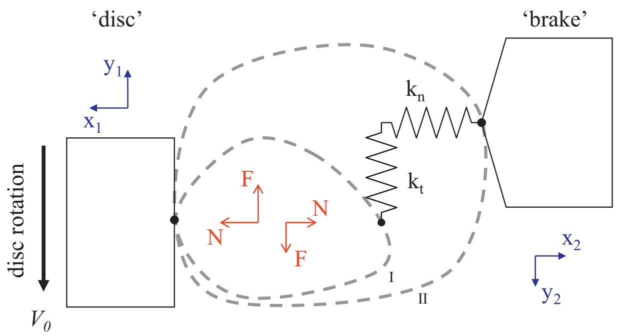

Fig. 1. Two linear subsystems coupled by a single point sliding contact with definition of variables. Two regions are marked with a dashed line and labelled 'I' and 'II', representing two options for defining the contact.