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DINAME 2013 - Simulation of Dynamic Instabilities Induced by Sliding Contacts

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Abstract: When dealing with complex mechanical systems that include sliding contact, it is necessary to account for the coupling between the dynamic behavior of the system and the local behavior at the contact. A particular consequence of interaction between system dynamics and contact behavior is the occurring of vibrational instabilities of the mechanical system, induced by the frictional contact. The dynamics of bodies in sliding contact can thus become unstable, due to the modal coupling caused by the normal and frictional components of the contact forces. Friction induced instabilities are at the origin of several everyday issues such as squeaking of door hinges or brake squealing. In literature, a large number of works deal with this kind of instabilities and are mainly focused on applied problems such as brake squeal noise. This paper shows a more general numerical analysis focused on a simple system constituted by a deformable cylinder that rotates around a rigid cylindrical surface with friction. The parametrical complex eigenvalue analysis and the transient numerical simulations show how the friction forces can origin dynamic instabilities due to the coupling between two system modes, even for such a simple system (one deformable body). The simplicity of the system allows for a deeper analysis of contact instabilities. Results from the experimental analysis allow for validating the model and confirm the occurring of the simulated dynamic contact instabilities.

Keywords: *contact instabilities, friction induced vibrations, complex eigenvalue analysis, transient simulation*

NOMENCLATURE

t = time

M = mass matrix

K = stiffness matrix

C = damping matrix

G = constraint matrix at the contact

u = nodal displacement vector

\ddot{u} = nodal acceleration vector

t = time

Greek Symbols

α = Rayleigh's coefficient

β = Rayleigh's coefficient

λ = Lagrange multipliers

μ = friction coefficient

Subscripts

i iteration step

k slave node number

n normal to the contact

t tangential to the contact

c critical friction coefficient

INTRODUCTION

Mechanical systems embody several contact surfaces that are optimized to either minimize or maximize the friction at the interface, as a function of their application and the need to either transfer motion with low energy loss or to dissipate the kinetic energy of the system. Whenever relative motion between two system components occurs, through a contact interface, vibrations are induced by the relative sliding (Akay, 2002). The deformations due to the dynamic response of the system affect largely the local stress distribution; reciprocally the variation of the local contact status (sliding, sticking, detachment), the waves generated by the local ruptures, and the local contact characteristics (stiffness, damping, etc.) can largely modify the macroscopic dynamic response of the system (Di Bartolomeo et al., 2012).

In sliding contacts the 'friction induced vibrations' (Sheng, 2007) can result in either a low amplitude acoustic noise characterized by a large frequency spectrum, which is function of the surface roughness and the dynamics of the bodies in contact, or a high amplitude acoustic signal characterized by a harmonic spectrum. The second case is characteristic of dynamic system instabilities excited by the contact forces. Contact dynamic instabilities have been object of several analyses focused on specific applications as joint squeak or brake noises. In particular a large literature can be found on the subject of brake squeal (Kinkaid et al., 2003; Ouyang et al., 2005 and references therein), which is a high frequency sound emission and is characterized by an unstable vibration at a natural frequency of the brake system. Nevertheless this kind of harmonic sound emission, due to vibrations excited by the sliding contact, occurs often in commonly used mechanical systems (door hinges, chalk on a board, ...). Between the mechanisms retained responsible of such vibrations, the unstable coupling between two modes of the system is one of the most commonly adopted (Massi et al., 2007 and references therein).

In this work a more general approach is used to highlight this kind of dynamic instability, which is due to the contact forces that couple the tangential and normal modal deformations at the contact. A deformable disc, made of

polycarbonate, is in sliding contact with a rigid (steel) disc at its inner radius. The contact pressure between the inner radius of the polycarbonate disc and the coaxial steel disc is obtained by an expansion of the steel disc. Afterwards the sliding is obtained thanks to its rotation. In this way, any “system” effect, coming from a more complex mechanical system, is avoided and the unstable behavior can be directly related to the dynamics of the single deformable body.

First a 2D numerical model is developed to perform the prestressed modal analysis to recover the linear system dynamics; then, a transient nonlinear analysis is performed to simulate the contact instabilities and the results are compared with the system dynamics, highlighting that the contact forces excite unstable system modes.

An experimental set-up has been developed and experimental tests allow for validating the numerical results.

NUMERICAL MODELING

Experimental analysis of contact issues is quite challenging due to the impossibility of measuring local contact properties without modifying the contact itself. Numerical tribology is a powerful tool allowing for a deeper investigation of contact problems (Renouf et al., 2011). Moreover numerical tools allow for a multi-scale analysis by coupling the scale of the contact (from some nm to some μm) with the scale of the bodies in contact (order of some mm to some cm) and the scale of the mechanism (from some cm up to several meters, depending on the application). The numerical modeling can either couple different numerical techniques (Discrete Element Modeling, Finite Element Modeling, Multibody Modeling, ...) in a single code by a strong coupling of the governing equations, or just performing a weak coupling thanks to the solution of each model that becomes the boundary conditions for the model at other scales (Renouf et al., 2011).

In this paper, first the dynamic of the mechanical system is investigated by a parametrical CEA (Complex Eigenvalue Analysis) to recover the system eigenvalues and the system modes (mechanism scale) that can result to be unstable; then a transient non-linear FE (Finite Element) analysis is developed to simulate the vibrational response of the system and to investigate the local contact behavior (contact scale) during the instability.

Linear model for CEA

The system is constituted of two coaxial cylinders (Fig. 1): the inner cylinder is made of steel and rotates with respect to the outer polycarbonate cylinder. The contact pressure at the sliding interface is assured by an expansion of the inner cylinder, obtained by providing a radial pressure on its inner surface. The commercial FE code ANSYS has been used to perform the parametrical CEA. The simple geometry of the system and the boundary conditions, which lie on the plane of the discs, allow for reducing the simulation on a simplified 2D model (Fig. 1). Nevertheless, a 3D model was also developed for a preliminary analysis to validate the in-plane dynamics of the 2D model and to verify that the obtained experimental results involve the in-plane dynamics of the system, which can be effectively simulated by the 2D model.

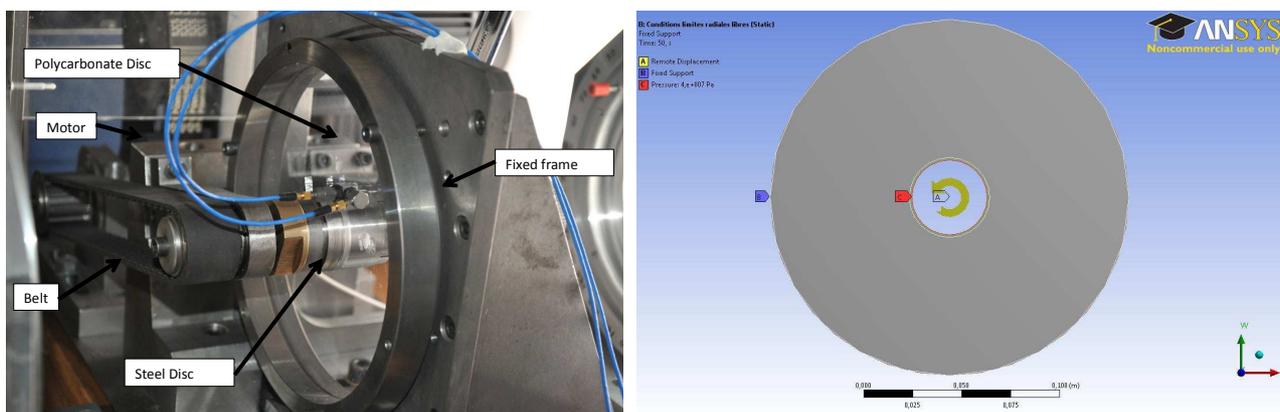


Figure 1 – Experimental set-up (left) and 2D simplified FE model (right).

First a prestressing static analysis is performed to apply the desired contact pressure (steel disc expansion) and the friction forces by introducing the relative rotation between the two discs; then, the prestressed CEA is performed. The analysis is repeated as a function of different parameters (friction coefficient, contact pressure, etc.); at each value of the system parameters the complex eigenvalues are extracted. The imaginary part of the eigenvalue gives the mode frequency while the real part is proportional to the modal damping; positive values of the real part of the eigenvalue (apparent negative damping) are associated to unstable modes. Figure 2 shows the mode shapes of the first 20 modes in frequency, ranging from 0 to 15 kHz. The modes are characterized by nodal circumferences and nodal diameters. When nodal diameters exist, two modes with the same deformed shapes and a phase of 90° in space are calculated, due to the axial-symmetry of the system. When the friction coefficient is nil the couple of mode are at the same frequency (same natural frequency and modal damping); when increasing the friction coefficient the asymmetry due to friction shifts the natural frequencies of the modes and some of them can couple toward an unstable behavior.

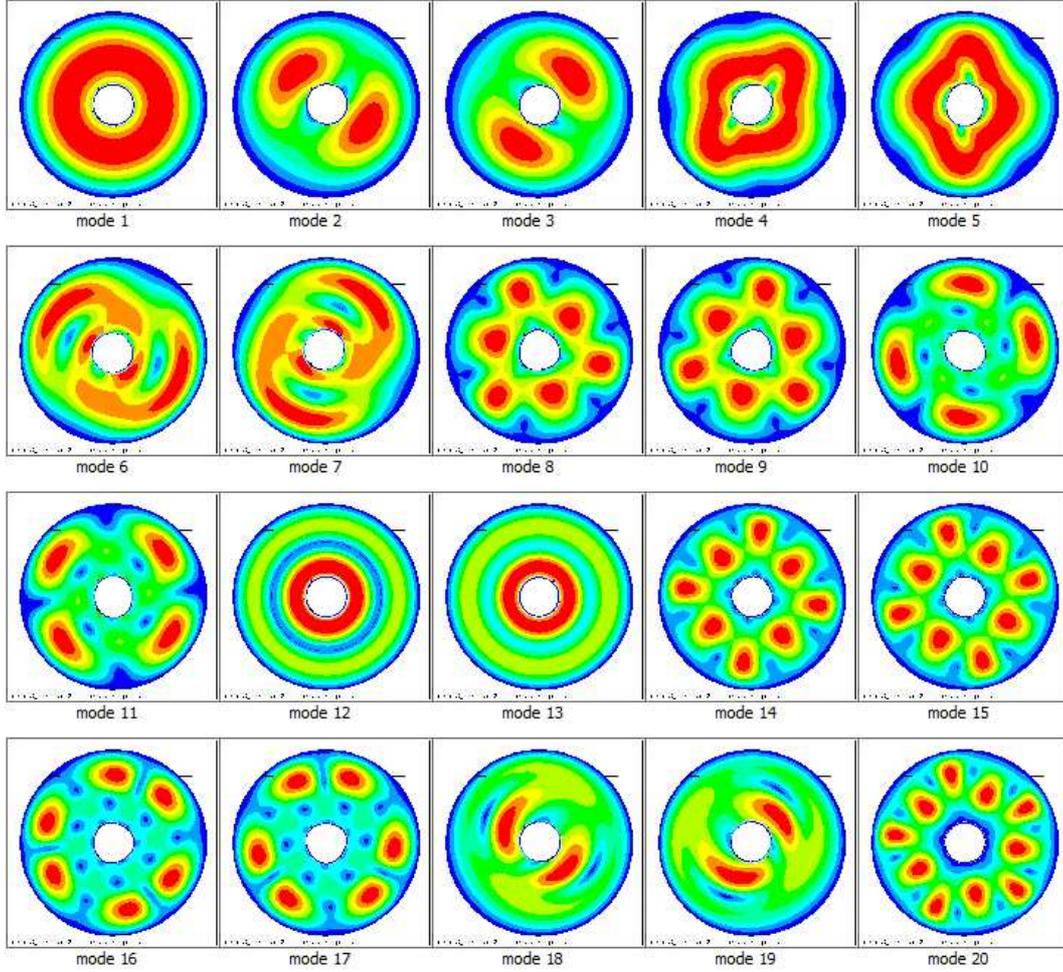


Figure 2 – System modes calculated by the CEA for the first 20 eigenvalues, between 0 and 15 kHz.

Non-linear model for transient analysis

Once the system parameters leading to the instability are identified, a transient FE analysis is performed with the explicit dynamic FE code PLASTD (Baillet and Sassi, 2005), developed to simulate the behavior of systems with frictional contact. The software is designed for large transformations and non-linear contact behavior and applies a forward Lagrange multiplier method for the contact between deformable bodies (Carpenter et al., 1991).

For the transient simulation, the formulation is discretized spatially by using the finite element method and discretized temporally by using an explicit Newmark scheme named the β_2 method. The contact algorithm uses slave nodes and target surfaces described by a four node quadrilateral element with 2x2 Gauss quadrature rule. The target segments are described by two nodes and approximated by bicubic splines (Baillet and Sassi, 2005). The forward Lagrange multiplier method is formulated by equations of motion at time ($t^i = i \Delta t$) with the displacement conditions imposed on the slave node at time t^{i+1} :

$$\begin{cases} M\ddot{u}^i + C\dot{u}^i + Ku^i + G^{i+1T} \lambda^i = F^i \\ G^{i+1} \{ X^i + u^{i+1} - u^i \} \leq 0 \end{cases} \quad (1)$$

where M and K are respectively symmetric and positively defined matrices of mass and stiffness of the system. C is the Rayleigh's proportional damping matrix:

$$C = (\alpha M + \beta K) \quad (2)$$

The Coulomb friction law is used; during each iteration the following two contact conditions for each slave node k are checked:

$$\begin{aligned}
 I) \lambda_n^k \leq 0 & \begin{cases} \lambda_n^k < 0 & (\text{contact}) \\ \lambda_n^k = 0 & (\text{separation}) \end{cases} \\
 II) \|\lambda_t^k\| \leq \mu \|\lambda_n^k\| & \begin{cases} \text{if } \|\lambda_t^k\| < \mu \|\lambda_n^k\| \rightarrow \dot{u}_t = 0 & (\text{stick}) \\ \text{if } \|\lambda_t^k\| = \mu \|\lambda_n^k\| \rightarrow \lambda_t^k \cdot v_t \leq 0 & (\text{slip}) \end{cases}
 \end{aligned} \tag{3}$$

where the normal and tangential contact force (λ) at each slave node k are compared. The first condition (3.I) means that the contact force is a compression force (without adhesion components), while the second condition (3.II) is associated with the use of a Coulomb friction law.

Only the contact non-linearities (sliding, sticking and detaching are allowed at the contact) are introduced into the model to simulate the time vibrational response of the system and to recover the local contact behavior. The friction law used at the interface is the classic Coulomb law with constant friction coefficient.

In the numerical models the inner disc is considered as a rigid surface, because of the high contrast between the compliance of the polycarbonate disc and the steel disc.

NUMERICAL RESULTS

Instability prediction by CEA

The CEA analysis allows for investigating the system eigenvalues as a function of the system parameters. Figure 3 shows the plot of the imaginary (frequency) and real part of the eigenvalues when increasing the friction coefficient. Because of the axial symmetry of the system (polycarbonate disc) two modes of the system are calculated at each eigenfrequency with a phase of 90° between the mode engenvalues (Fig. 2). The modes are numbered in Fig. 3 for increasing natural frequencies. When the friction coefficient is equal to 0, the real part of the eigenvalues is only due to the material damping contribution (black dotted line in Fig. 3). When increasing the friction coefficient, the real part of each couple of modes (at the same frequency) diverges from the starting value, due to the friction contribution (Massi et al., 2007). While one mode shifts towards the negative half plane, the other shifts toward positive values of the real part and becomes unstable. The critical friction coefficient (μ_c), for which the mode becomes unstable is reported on the right of each couple of modes. Lower critical friction coefficients are obtained at lower frequency, where the contribution of the material damping is lower.

Only some of the modes reach positive values of the real part (negative damping): the comparison between the deformed shape (reported in Fig. 3) of the modes 24-25, which stay stable, and the modes 26-27 and 28-29, which become unstable, highlights the role of the local deformation at the contact. Only modes with large local deformation at the contact zone can become unstable; in fact, the larger is the local deformation, the greater is the contribution of friction to the real part of the eigenvalue (due to the coupling between normal and tangential components).

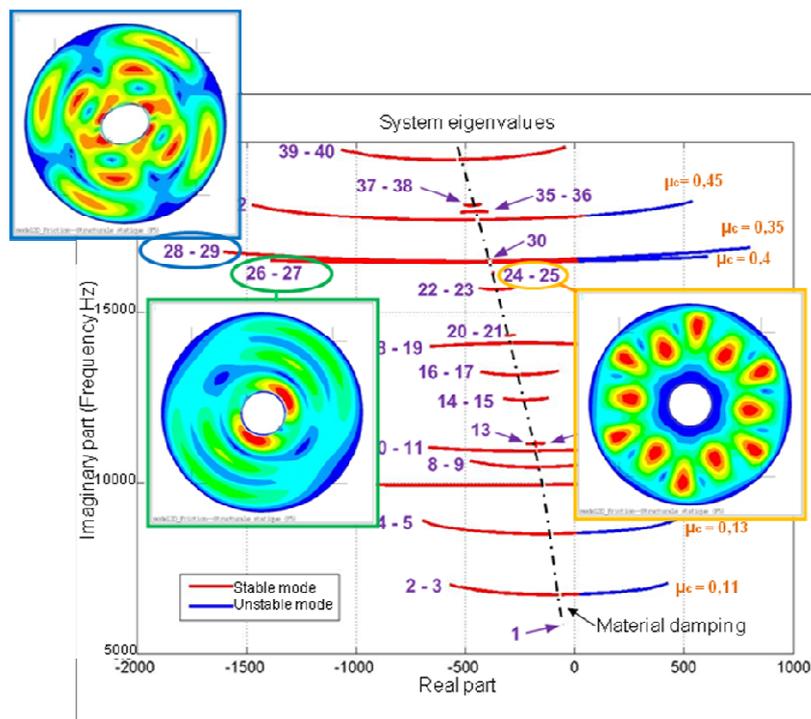


Figure 3 – System eigenvalues when increasing the friction coefficient. For each couple of modes the critical friction coefficient, μ_c (i.e. the value for which the mode becomes unstable), is reported on the right.

Non-linear transient simulations

Once predicted the value of the parameters leading to the instability, the time simulation accounting for the contact nonlinearities is performed. Figure 4.a shows the acceleration of a node of the disc: first the inner cylinder is expanded to reproduce the dilatation of the steel cylinder in the experimental test bench (preload phase); then the inner cylinder rotates at constant rotational velocity (10 rpm), providing the friction at the inner surface of the polycarbonate disc. The initial vibrations occurring during the preloading phase (green dotted box) are due to the first contact between the two discs and are quickly damped by the material damping of the polycarbonate disc; then the unstable vibrations increase exponentially (at about 0.015 s), due to the apparent negative modal damping (positive real part of the eigenvalue) provided by the friction contribution; the vibrations rise up to a limit cycle due to the contact nonlinearities and characterizing phenomena such as squeal noise (Massi et al., 2007). Figure 4.b shows the power spectral density (PSD) of the signal during the preloading phase (green) when all the modes having a large radial component are excited; the PDS of the whole signal (red); the PSD of the unstable state (bleu) when the disc vibrates at the unstable eigenfrequency with an harmonic spectrum characteristic of vibrations due to the dynamic modal instability.

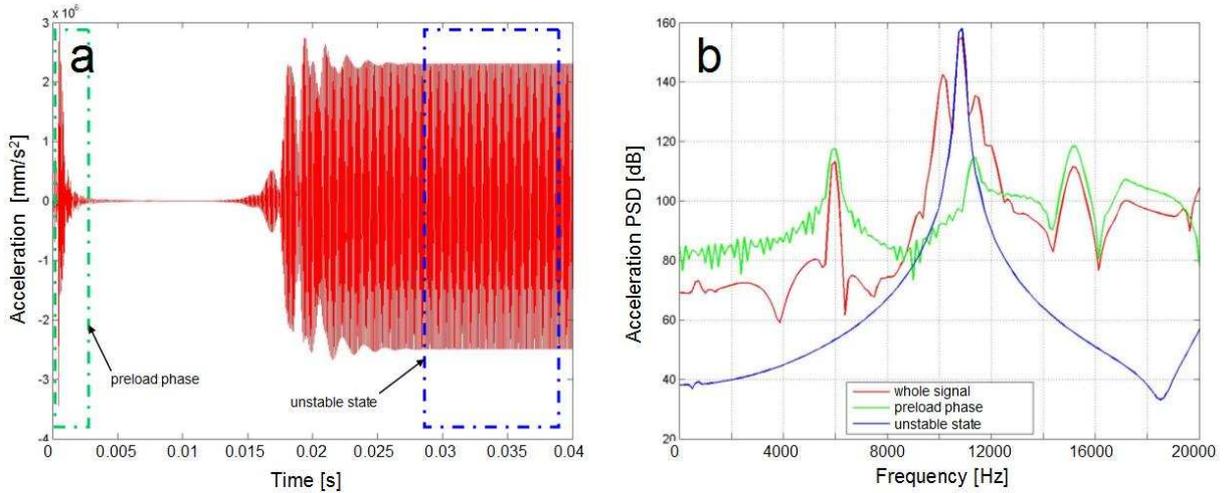


Figure 4 – a) Time signal of the node in-plane acceleration during the whole simulation with unstable parameters: preload phase (green box); unstable state (blue box). b) Power spectral density for the whole signal (red), the preload phase (green) and the unstable state (bleu).

The comparison between the linear CEA and the nonlinear transient simulation is shown in Fig. 5, where the deformed shape of the unstable mode at about 10 kHz, calculated by the CEA, and the velocity field of the system vibration at about 10.5 kHz, obtained by the transient nonlinear analysis during the simulated instability, are reported. Agreement in the deformed shape and the frequency is obtained, showing that the contact dynamic instability results in vibrations at a system eigenfrequency that is excited by the contact forces. The higher frequency obtained for the same mode with the nonlinear model is due to the larger stiffness of the inner disc, which is considered as rigid.

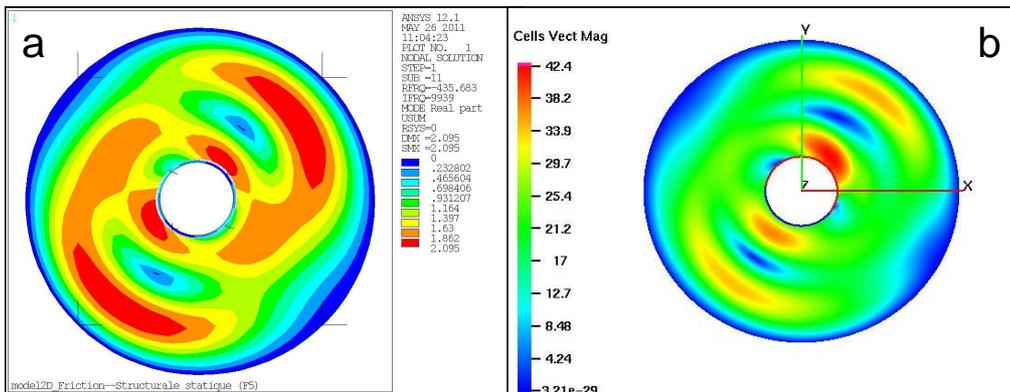


Figure 5 – Comparison between the predicted unstable mode shape by CEA (a) and the disc deformation (velocity field) during transient analysis (b).

Effect of friction coefficient

When increasing the friction coefficient, the interaction between normal and tangential deformation increases because of the larger tangential (frictional) contact force; consequently the coupling between the system modes increases. Figure 6 shows the system mode at about 10 kHz (mode 6) calculated for 3 different values of the friction

coefficient, respectively equal to 0.01, 0.3 and 1. Figure 7-a shows the locus plot of the modes 6 and 7 when increasing the friction coefficient from 0 to 1. Figure 7-b shows the respective modal damping of the modes 6 and 7 as a function of the friction coefficient; for friction coefficient equal to 0.01 the real part is negative and the damping positive, e.i. the mode is stable; for friction coefficient larger than 0.15 (critical friction coefficient) the mode is unstable.

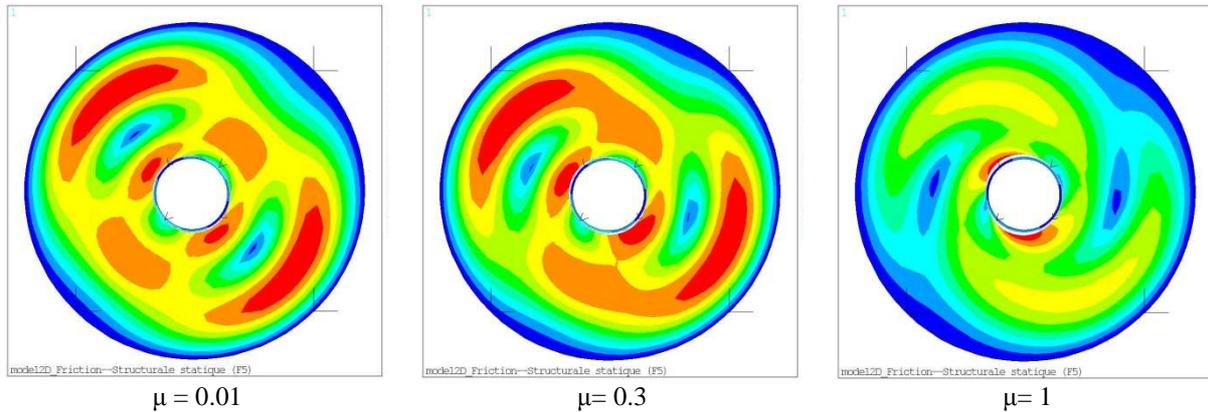


Figure 6 – Deformed shape of the mode 6, at about 10 kHz, for 3 different values of the friction coefficient; from left to right friction coefficient equal to 0.01, 0.3, 1.

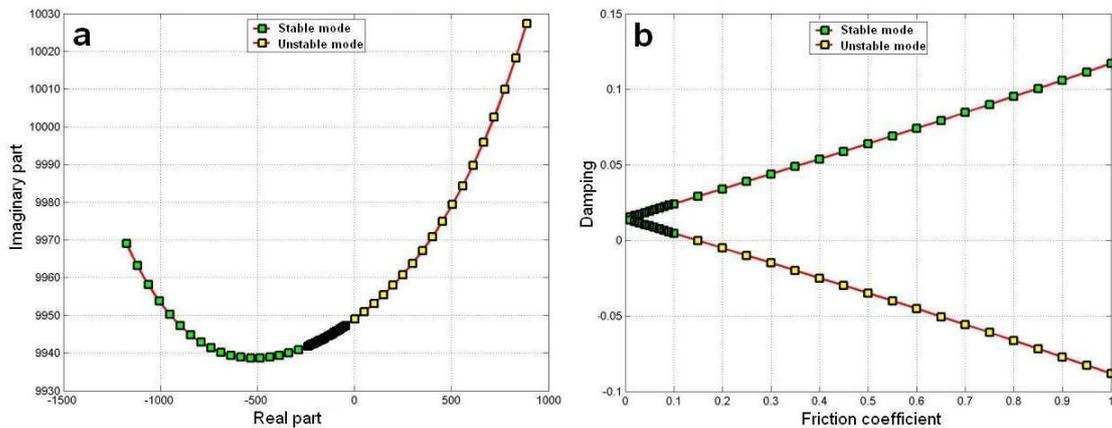


Figure 7 – a) Locus plot for the modes 6 and 7 when increasing the friction coefficient from 0 to 1; b) Modal damping as a function of the friction coefficient for the modes 6 and 7. The critical friction coefficient is 0.15.

The velocity fields calculated by the nonlinear transient analysis, for the 3 values of the friction coefficient, are reported in Fig. 8:

- For friction coefficient equal to 0.01 (left), the system is stable and the velocity field is due to the radial deformation due to the system preload. The local contact forces stay constant in time and the acceleration calculated at the nodes of the system show no vibrations after the preloading phase.
- For friction coefficient equal to 0.3 (center), the system is unstable and a harmonic vibration at the unstable mode frequency is calculated (Fig. 4). The local contact forces oscillate at the unstable frequency, due to the system (mode) deformation at the contact surface; the oscillation of the contact forces at the mode frequency introduces energy in the unstable mode itself, allowing for the auto-excited vibrations.
- For friction coefficient equal to 1 (right), the stress concentration in a localized zone of the contact (as predicted as well by the CEA, Fig. 6-right) brings to a zone of sticking between the two solids; the passage between stick and slip along the contact surface during the rotation of the inner disc excites the system dynamics; the PSD of the acceleration calculated at the system nodes shows a complex dynamics with peaks corresponding to all the system modes. In this case the dynamic modal instability is replaced by a different unstable state characterized by stick-slip phenomena.

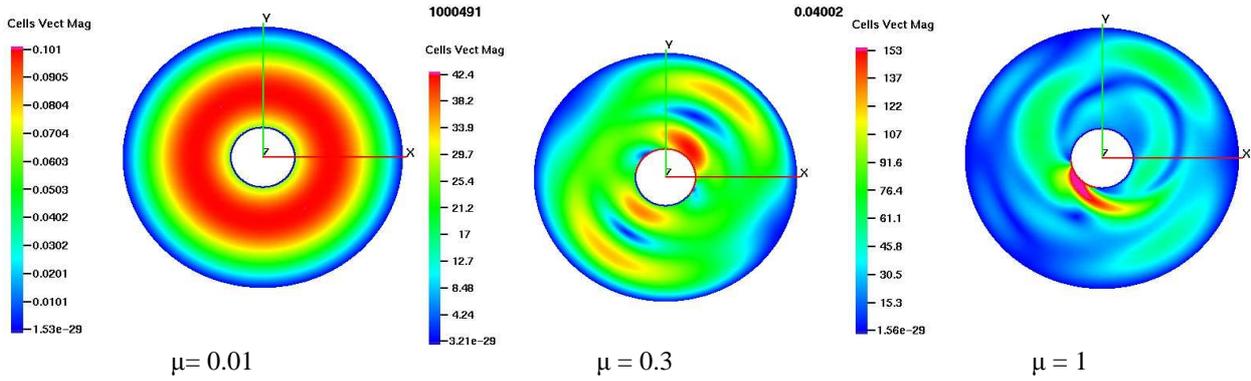


Figure 8 – Velocity fields calculated during the transient nonlinear analysis, for 3 different values of the friction coefficient; from left to right friction coefficient equal to 0.01, 0.3, 1.

Effect of damping

When dealing with numerical simulations including sliding contact, a key parameter to account for is the material damping (Di Bartolomeo et al., 2012; Cantone and Massi, 2011). A convergence analysis performed on the numerical models (ANR????) showed that with a too low material damping it is not possible to obtain the convergence as a function of the mesh and the integration time step. In fact, the energy introduced into the system (unstable mode) by the friction forces is not dissipated, increasing the amount of stored vibrational energy and, consequently, modifying its dynamic behavior. On the contrary, a too large material damping dissipates all the vibrational energy that is introduced into the system by friction, forcing the system to an artificial stable behavior.

For this reason an experimental modal analysis has been performed on the polycarbonate disc to calculate the experimental modal damping in the range of frequency between 0 and 15 kHz (ANR???????) and to recover the corresponding values of the Rayleigh's coefficients: $\alpha=40 \text{ s}^{-1}$; $\beta=4.5e^{-7} \text{ s}$.

EXPERIMENTAL VALIDATION

The simplified geometry of the system allowed developing an experimental set-up (Fig. 1) able to maintain as constant as possible contact and dynamic boundary conditions, in order to reproduce the dynamic instability and to measure the vibration of the system and the local behavior at the contact (ANR?????). To verify experimentally the numerical simulations, tests have been reproduced experimentally with the same parameters used in the transient simulation, i.e. 10 rpm of rotational velocity and 40 MPa of nominal contact pressure.

The dynamic instability characterized by harmonic vibration at about 10.5 kHz has been reproduced experimentally. Figure 9 show the acceleration measured by an accelerometer (along the vertical in-plane direction) during tests (Fig. 9-a) and the same acceleration calculated by the transient numerical analysis with friction coefficient equal to 0.3 (Fig. 9-b). Figure 10 shows the PSD of the experimental acceleration signal, to be compared with the numerical results (Fig. 4). The same harmonic vibration at 10.5 kHz is recovered, i.e. the same natural frequency is excited numerically and experimentally, with comparable amplitude of vibration (between 100 and 300 m/s^2).

The occurrence of the dynamic instability is really sensitive to both the system dynamics and the contact conditions (Kinkaid et al., 2003; Ouyang et al., 2005; Massi et al., 2007); due to the impossibility of obtaining exact symmetry in the experimental tests, the unstable vibrations do not reach a steady state condition and a periodicity on the increasing of the unstable vibrations can be observed experimentally (Fig. 9-a); this period coincides with the period of rotation of the inner disc that changes periodically the contact conditions and the dynamics of the system. New tests with better symmetry of the system are under development.

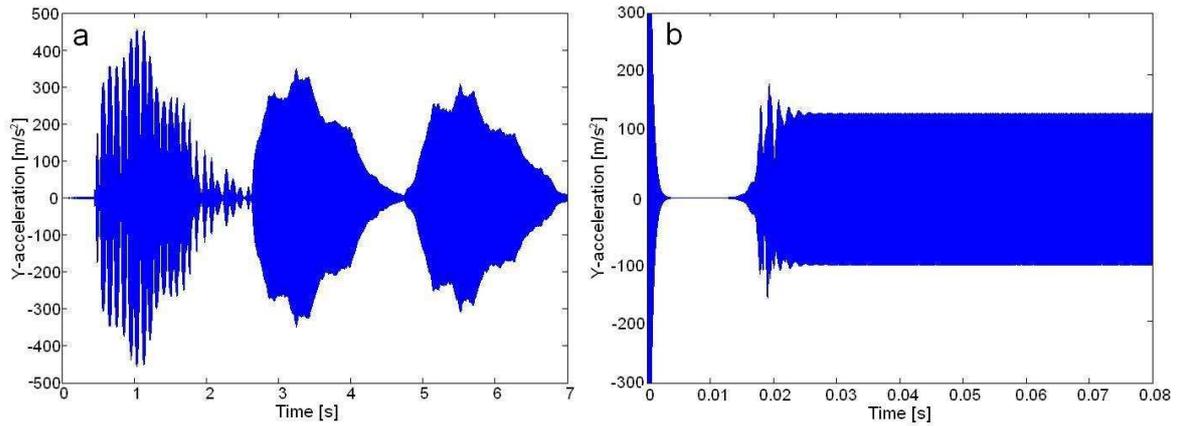


Figure 9 – Vertical in-plane acceleration measured experimentally (a) and calculated by the transient nonlinear simulation (b).

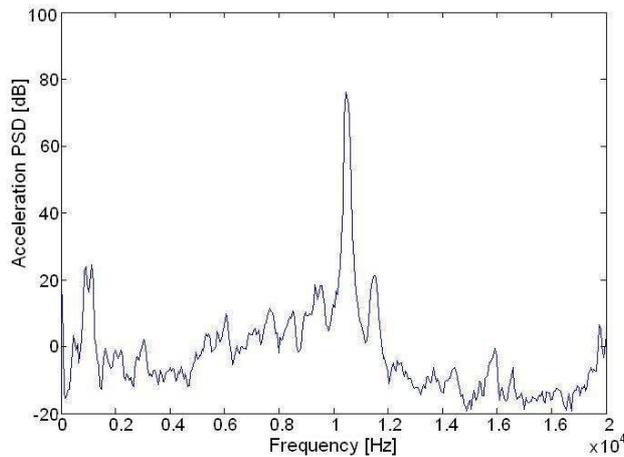


Figure 10 – Power spectral density of the acceleration measured experimentally, showing a harmonic vibration at 10.5 kHz.

RESULTS

This paper shows a numerical analysis of dynamic instabilities induced by frictional contact. The objective of this work is to generalize the recent findings in applicative issues, like brake squeal noise, to investigate the physical phenomenon. At this aim a simple system consisting in a deformable body sliding on a rigid surface is developed and investigated. Numerical results show the occurring of unstable vibrations excited by the contact forces. The comparison between CEA and transient simulation allows for asserting that the vibrations are due to an unstable eigenfrequency of the system, which reaches a positive value of the real part (apparent negative damping) due to the contribution of the friction forces. The parametrical analysis shows that the system, increasing the friction coefficient, switches from a stable state (without vibrations) to a dynamic instability, characterized by harmonic vibrations, up to a stick-slip behavior exciting the whole system dynamics. A key role in the occurrence of the instability and in the convergence of the simulations has been found to be the material damping; only realistic material damping, recovered experimentally, assures the convergence of the transient simulation allowing for an energy equilibrium between the energy introduced into the system by the friction and the energy dissipated into the material. Comparison with experimental tests has been performed to validate the numerical results.

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