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GAMM 2015

Università del Salento

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S05: Nonlinear oscillations

The section covers all fields of vibrational problems in solid mechanics or mechatronics including nonlinear effects. Submissions may address, for example, systems with nonlinear material behavior, nonlinearities in joints, mathematical solution methods (analytical or numerical), control or description of nonlinear behavior like bifurcations or chaos, or experimental idendification of nonlinearities.

Rotordynamics of Two-Pole Turbo Generators with Refined Modelling of the Unbalanced Magnetic Pull

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Within this contribution the rotordynamic behaviour of two-pole turbo generators, which are typically used in power plants in combination with gas or steam turbines, is discussed. In view of the fact that a large part of the electric power is provided by these generators, understanding their dynamics is quite relevant. To this end a basic model originally proposed by Kellenberger [1] is reviewed and extended in this analysis.
For the rotordynamics of induction or synchronous machines one of the most important electromagnetic forces is the well known unbalanced magnetic pull (UMP). It results from the asymmetric magnetic field which is mainly caused by the excentricity of the rotor. In classic approaches the UMP acts radially outwards in direction of the smallest air-gap and against the shaft's restoring force [3]. Kellenberger [1] experimentally found that the direction and magnitude of this electromagnetic force may also depend on the rotor's orientation relative to the position of the smallest air-gap. It's influence becomes especially relevant, when large generators are regarded. This dependency has also been reported by Belmans et al. [2].
Kellenberger regarded this instance by introducing a small additional force which varies in direction twice along the circumference, considering the circumstance that the UMP changes due to the orientation of the magnetic poles. After nondimensional scaling the governing equations for the lateral movement of a classical Laval-Rotor (Jeffcott-Rotor) enhanced by a refined model of the electromagnetic forces read

$$
\begin{aligned}
Y'' + 2DY' + (1 - \lambda \cos(2\eta\tau))Y - \lambda \sin(2\eta\tau)Z &= \eta^2 \cos(\eta\tau - \Phi),\\
Z'' + 2DZ' - \lambda \sin(2\eta\tau)Y + (1 + \lambda \cos(2\eta\tau))Z &= \eta^2 \sin(\eta\tau - \Phi).
\end{aligned}
$$

Here Y and Z represent the specific horizontal and vertical position of the elastic center, D is the damping ratio, λ defines the ratio of additional electromagnetic force to the effective elastic force, η represents the ratio of angular rotation frequency to angular eigenfrequency and Φ is the orientation of the excentricity-axis of the centre of mass towards the rotor's pole axis.
For stationary motions Kellenberger found that the additional force can cause a resonance catastrophe even in presence of damping. In this contribution the analysis is extended by a stability analysis. It turns out that the parametric excitation caused by the additional force may influence the stability of stationary motions. Furthermore an excentric point of application for the electromagnetic forces is introduced and instationary motions are regarded. In this context the systems start-up behaviour is investigated and compared to that of the classic Laval-Rotor's as discussed in [5]. It is found, that the magnetic excentricity may alter the system's characteristics considerably and thus must be considered.

References

[1] W. Kellenberger. Der magnetische Zug in Turbogenerator-Rotoren als Ursache einer Instabilität des mechanischen Laufes. Archiv für Elektrotechnik. 50 (1966), 253–265.

[2] R. Belmans, A. Vandenput, W.Geysen. Calculation of the flux density and the unbalanced pull in two pole induction machines. Archiv für Elektrotechnik. 70 (1987), 151–161.

[3] G. Müller, K.Vogt, B.Ponick. Berechnung elektrischer Maschinen. John Wiley & Sons, 2012.

[4] U. Werner. Mathematical rotordynamic model for lateral vibration analysis of induction motors with dynamic eccentricities regarding start-up. Forschung im Ingenieurwesen 78.1-2 (2014), 27–43.

[5] R. Gasch. Biegeschwingungsverhalten rotierender Wellen beim Durchlaufen der kritischen Drehzahlen. Konstruktion 29.9 (1977), 355–365.

Stabilization of a rotating shaft by electromagnetic actuators

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It is known that imperfect (e.g. visco-elastic) supporting of a rotating shaft or presence of internal friction in the material it is made of may destabilize static equilibrium position during its operation [1], [2]. At a certain angular velocity, called the critical speed, the static equilibrium bifurcates into a new oscillatory state manifested by transverse vibrations of the rotating shaft. To prevent the system from such a situation, sometimes dangerous due to large amplitudes, passive and active methods are being incorporated. In this paper, an active approach employing electromagnetic actuators [3] is proposed and discussed. The electromagnetic resistant force developed by the actuators introduces additional external damping into the system. Moreover, the attractive forces depending on the gap between the shaft and electromagnets bring some additional stiffness. Both mechanical effects induced by electromagnetic phenomena, i.e. the enlarged damping and stiffness, change conditions in which the rotor loses its stability.

The paper presents fundamentals of dynamics of rotating shafts with internal friction and describes the principle of working of the electromagnetic actuators. The governing equation of motion of such a coupled mechano-electromagnetic structure is as following

$$\frac{\partial^2 \tilde{w}}{\partial t^2} + a^2 \left[1 + \beta \left(\frac{\partial}{\partial t} - \mathrm{i}\Omega\right)\right] \frac{\partial^4 \tilde{w}}{\partial x^4} + \tilde{F}_{Mw} \delta(x - x_a) = 0$$

where the complex parameters marked by tildes denote: $\tilde{w} = y + \mathrm{i}z$, $\tilde{F}_{Mw} = F_{My} + \mathrm{i}F_{Mz}$ and $\mathrm{i} = \sqrt{-1}$. In the above equations t is time, x is the axial coordinate, y and z – transversal displacements of the shaft, $a^2 = YJ/\rho A$ – parameter related to the material and geometrical data of the shaft (Y stands for Young's modulus, J – cross-sectional moment of inertia, ρ – volume density, A – area of the cross-section), β is the retardation time of the assumed Kelvin-Voigt model of the shaft material, Ω – angular velocity, F_{My} and F_{Mz} – respective components of the force exerted by the electromagnetic actuators, $\delta(.)$ – Dirac's delta function, x_a – coordinate of the actuators placement. The electromagnetic force F_{Mw} generated by a pair of each actuator in the y or z direction is

$$F_{Mw} = \mu_0 S N^2 \left[\left(\frac{i_{w2}}{2\left[\Delta - w(x_a)\right] + l/\mu} \right)^2 - \left(\frac{i_{w1}}{2\left[\Delta + w(x_a)\right] + l/\mu} \right)^2 \right]$$

where w denotes either y or z transverse direction, indices 1 and 2 correspond to the respective part of each actuator pair, μ_0 describes magnetic permeability of vacuum, μ – relative permeability of the magnetic core of the actuator, S – cross-section area of the core, N – number of wire coils wound around the core, l – core length, Δ – nominal gap (in the equilibrium position) between the shaft and the actuator core, i – dynamic current developed in core 1 or 2 of the actuator acting in the $w = y$ or $w = z$ direction, $w(x_a)$ – actual (dynamic) distance between the shaft and the actuator.

In the analysis, Galerkin's method has been used to discretize the system and find an approximating set of ordinary differential equations of motion. The carried out numerical simulations prove that the electromagnetic actuators may increase the critical rotation speed by even 50 percent. Additionally, highly nonlinear shape of the magnetization curve strongly contributes to near-critical behavior of the shaft and newly born self-excited vibration. It also determines the character of the bifurcating response of the entire system. The problem of efficiency of the applied method of stabilization as well as nonlinear effects brought about by the electromagnetic actuators are highlighted and addressed in the paper.

References

[1] F.M. Dimentberg. Flexural Vibrations of Rotating Shafts. London: Butterworth 1961.

[2] W. Kurnik. Stability and bifurcation analysis of a nonlinear transversally loaded rotating shaft. Nonlin. Dyn. 5 (1994) 39-52.

[3] T. Szmidt, P.M. Przybylowicz. Critical load and non-linear dynamics of Beck's column with electromagnetic actuators . Int. J. Nonlin. Mech. 67 (2014) 63-73.

Stability and bifurcation behaviour of a Laval-Rotor considering fluid forces in compliant liquid seals

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This contribution discusses the influence of fluid forces from compliant liquid seals on the steady state stability and bifurcation behaviour of a Laval-Rotor. In rotating machinery like pumps, annular seals are extensively used to reduce leakage flows between volumes of different pressure. Compliant seals offer the possibility to reduce the sealing gap and at the same time minimize the risk of the shaft rubbing against the outer sealing shell. Therefore, leakage flow is decreased and efficiency increased.
To analyse the fluid flow in sealings, full CFD-simulations or reduced models (like *Bulk-Flow*-models) are used. In order to incorporate the fluid forces resulting from the pressure field into rotor dynamic investigations, a non-linear semi-empirical model by Muszynska [1] is often applied. The turbulent flow through compliant foil seals in particular has been discussed in [2] and the advantages of compliant metal seals have recently been reviewed on the basis of experimental investigations in [3].

The model used in this work for the steady state stability and bifurcation analysis consists of a Laval-Rotor (Jeffcott-Rotor) where the disc runs in a liquid sealing. While the seal shell itself is assumed to be rigid, it is elastically connected to the environment. This elasticity is used to model the compliance of the sealing itself or of the surrounding structure. The Muszynska model is applied to account for the forces from turbulent, incompressible flow through the seal. The whole model is described by the equations

$$\boldsymbol{M}_R\ddot{\boldsymbol{r}}_R + \boldsymbol{B}_R\dot{\boldsymbol{r}}_R + \boldsymbol{K}_R\boldsymbol{r}_R = \boldsymbol{F}(\boldsymbol{r}_R,\boldsymbol{r}_S) \qquad \boldsymbol{M}_S\ddot{\boldsymbol{r}}_S + \boldsymbol{B}_S\dot{\boldsymbol{r}}_S + \boldsymbol{K}_S\boldsymbol{r}_S = -\boldsymbol{F}(\boldsymbol{r}_R,\boldsymbol{r}_S)$$

where

$$\boldsymbol{F}(\boldsymbol{r}_R,\boldsymbol{r}_S) = -\begin{pmatrix} m_f & 0 \\ 0 & m_f \end{pmatrix}(\ddot{\boldsymbol{r}}_R-\ddot{\boldsymbol{r}}_S) - \begin{pmatrix} \bar{D} & 2\bar{\tau}\Omega m_f \\ -2\bar{\tau}\Omega m_f & \bar{D} \end{pmatrix}(\dot{\boldsymbol{r}}_R-\dot{\boldsymbol{r}}_S) - \begin{pmatrix} \bar{K} - m_f\bar{\tau}^2\Omega^2 & \bar{\tau}\Omega\bar{D} \\ -\bar{\tau}\Omega\bar{D} & \bar{K} - m_f\bar{\tau}^2\Omega^2 \end{pmatrix}(\boldsymbol{r}_R-\boldsymbol{r}_S)$$

where $\boldsymbol{M}, \boldsymbol{D}$, $\boldsymbol{K}$ and $\boldsymbol{r}$ are the mass-, damping-, stiffness-matrices and the position vectors respectively. The subscripted R indicates rotor-related and the subscripted S seal-related variables and parameters. $\boldsymbol{F}(\boldsymbol{r}_R, \boldsymbol{r}_S)$ is the fluid force. Furthermore m_f is the coefficient of the fluid inertia, $\bar{D}$ is the coefficient of the fluid damping, $\bar{K}$ is the coefficient of the fluid radial stiffness, Ω is the rotating speed of the rotor and $\bar{\tau}$ is the fluid average circumferential velocity ratio.

The objectives of the analysis are to investigate the stability of the steady state and the related bifurcation behaviour. The dependencies of the critical eigenvalue on the parameters is investigated. In particular, the influence of the compliance of the sealing is discussed.
Further steps in this investigation comprise the discussion of anisotropic seal flexibilities, damping and unbalance.

References

[1] A. Muszynska. Improvements in Lightly Loaded Rotor/ Bearing and Rotor/ Seal Models. J. Vib. Acoust. 110 (1988), 129–136.

[2] M. Salehi, H. Heshmat, J.F. Walton, S. Cruszen. The Application of Foil Seals to a Gas Turbine Engine. 35^{th} AIAA/ASME/SAE/ASEE joint Propulsion Conference and Exhibit 20-24 June, 1999, Los Angeles.

[3] L. San Andrés, A. Anderson, An All-Metal Compliant Seal Versus a Labyrinth Seal. J. Eng. Gas Turb. Power 137 (2015), 1–8.

Dynamic stability of composite rotating shafts with Brazier's nonlinearity

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Thinwalled rotating shafts reveal a considerable deformation of cross-section contour during bending. The ovalizing phenomenon is named Brasier's effect [1]. This results in a specific degressive geometric nonlinearity.

$$M/EJ = \kappa(1 - \gamma\kappa^2)$$

where: M - bending moment,κ - curvature, EJ - bending stiffness, and γ - constant depending on the geometric, strength parameters and the internal pressure q. It is assumed that the rotating shaft is loaded by uniformly distributed time-independent centrifugal forces. Another important problem of this paper is the description of the global damping of a laminated shaft. Despite the fact that in case of viscoelastic orthotropic plies the resulting constitutive equation is of higher order, the simple Voigt-Kelvin model is assumed. In our dynamics study the rotating symmetrically laminated hybrid circular cylindrical shell will be treated as a beam-like structure. It contains both the conventional (e.g.) graphite or glass) fibers oriented at $+\theta$ and $-\theta$ to the shaft axis and the thermoactive shape memory alloy fibers parallel to the shaft axis. The reduction is justified by a symmetric plies arrangement and negligible circumferrential stresses in the shaft. The shaft movement is described in immovable coordinate system y, z The shaft is assumed to be subjected to an axial stochastic force, which can be expresseed in terms of the Wiener process and the dynamics of shaft is described by the Itô differential equation

$$du = u_{,t}dt$$
$$du_{,t} = -\Big\{e\left[u_{,xx}(1-\gamma w_{,xx}^2)\right]_{,xx} + \beta e(u_{,xxxxt} + \omega v_{,xxxx})\Big\}dt - \varsigma u_{,xx}d\mathcal{W}$$
$$dv_{,t} = v_{,t}dt$$
$$dv_{,t} = -\Big\{e\left[v_{,xx}(1-\gamma w_{,xx}^2)\right]_{,xx} + \beta e(v_{,xxxxt} - \omega u_{,xxxx})\Big\}dt - \varsigma v_{,xx}d\mathcal{W}$$

where β - damping coefficient of shaft material, ς - intensity of wide-band Gaussian process. The shaft is assumed to be simply supported at its ends. The time dependent axial force destabilizes the rectilinear shape of the shaft. The direct Liapunov method is used to analyse the uniform stochastic stability of the equilibrium state. Using a measure of distance between solutions $\|.,.\|$ the trivial solution is uniformly stochastically stable if the following logic sentence is true

$$\bigwedge_{\epsilon\geq 0}\bigwedge_{\delta\geq 0}\bigvee_{r\geq 0}\|u(.,0),v(.,t)\| \leq r \Rightarrow P(\sup_{t\geq 0}\|u(.,t),v(.,t)\| \geq \epsilon) \leq \delta$$

Since the dynamic equations are strongly nonlinear, special attention is paid to a positive-definiteness of the appropriate energy-like Liapunov functional

$$V = \frac{1}{2}\int_0^\ell \Big\{u_{,t}^2 + \big(u_{,t} + \beta e u_{,xxxx}\big)^2 + v_{,t}^2 + \big(v_{,t} + \beta e v_{,xxxx}\big)^2 + 2e\big(u_{,xx}^2 + v_{,xx}^2\big)\big[1 - \frac{\gamma}{2}\big(u_{,xx}^2 + v_{,xx}^2\big)\big]\Big\}dx \leq V_*$$

where the functional V^* corresponds to the linearized problem $\gamma = 0$. Analysing the local positive-definiteness and the supermartingale property of functional V along the dynamics equations lead to sufficient stability conditions, expressed in terms of the rotation speed, the damping coefficients, the bending stiffness and the axial force intensity. The influence of SMA fiber activation on increase of stability region is analysed.

References

[1] E. Reissner. On finite bending of pressured tubes. ASME J. Appl. Mechs. 26 (1959), 366–392.

Synchronization effects in rotors partly filled with fluid

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In fluid-filled rotors self-excited vibrations occur induced by a surface wave of the fluid. A characteristic property is the instability over the full range of angular velocity above the natural frequency of the system. A possible explanation is the occurrence of synchronization effects between fluid and rotor.

The behaviour of rotors partly filled with fluid was mostly studied under the aspect of stability in steady-state conditions. For non-steady-state investigations, discrete models with reduced number of degrees of freedom and reasonable ability to model the system behaviour are desirable due to the complexity of fluid modelling.

This talk analyses several models and shows synchronization effects between fluid and rotor model.

On the self-balancing of the planetary rotor

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The phenomenon of self-balancing of rigid rotors is well known and investigated for rotors with fixed bearings [1], [3]. However, in some technical devices the rotor performs complex motions. An example of such system is a computed tomography scanner. Its anode rotates very fast in the housing of the X-ray tube. At the same time the X-ray tube itself rotates rather slowly around the patient's body. It is very important for CT scanner to keep the minimal possible level of vibrations in order to obtain good image quality. The objective of this paper is to investigate how and to which extent the self-balancing devices can be used for reducing vibrations in a planetary moving rotor.

Consider the system representing a rotating rotor on a rigid carrier. The rotor of mass M is fixed on the end of the carrier which length is R. The other end of the carrier is elastically suspended with two spring-dampers of a certain stiffness c and damping β. The carrier rotates around it's point of suspension with a constant velocity Ω. At the same time the rotor rotates around its symmetry axis with a given velocity ω. Its centre of mass has an offset γ relative to the rotation axis. Two pendulum balancers of mass m, moment of inertia J and length r are placed on the rotation axis of the rotor.

$$(M+2m)\ddot{x}+\beta\dot{x}+x=(M+2m)R\Omega^2\cos\Omega t+M\gamma\omega^2\cos\omega t+mr\sum_{i=1}^{2}(\dot{\varphi}_i{}^2\cos\varphi_i+\ddot{\varphi}_i\sin\varphi_i) \tag{1}$$

$$(M+2m)\ddot{y}+\beta\dot{y}+y=(M+2m)R\Omega^2\sin\Omega t+M\gamma\omega^2\sin\omega t+mr\sum_{i=1}^{2}(\dot{\varphi}_i{}^2\sin\varphi_i-\ddot{\varphi}_i\cos\varphi_i) \tag{2}$$

$$(J+mr^2)\ddot{\varphi}_i+\beta_\varphi(\dot{\varphi}-\omega)=mrR\Omega^2\sin(\omega t-\varphi_i)+mr(\ddot{x}\sin\varphi_i-\ddot{y}\cos\varphi_i);i=1,2 \tag{3}$$

The equations of motion of the whole system (1) - (3) can be split into two groups. The first two equations describe the radial vibrations of the carrier system. The last two equations describe the phases of the pendulums. Herewith it has been assumed that the damping both in the oscillation subsystem β and in the rotation degrees of freedom for pendulums β_φ is not small.

The approximate asymptotic solution has been obtained using averaging technique for the strongly damped systems [2]. There are four stationary solutions in this system. Only one of them provides the compensation of the unbalance. The stability investigation has shown that the compensating configuration is stable only in the overcritical domain of the rotation speeds of the rotor. The improved first order approximation has been calculated for the compensating configuration of the system in order to estimate the intensity of the remaining oscillations.

Hence the self-balancing of pendulum type is effective for planetary rotor in the overcritical speed domain. However the balancing is limited by the centrifugal forces causing small vibrations of pendulums. The expressions for the residual amplitudes of vibrations are obtained in dependency on the velocity of the planetary motion. Analytic results match very well with numeric simulations when velocity of planetary motion is sufficiently small.

References

[1] I.I. Blekhman. Synchronization of Dynamical Systems. Nauka, Moskow, (1971).

[2] A. Fidlin, J.J. Thomsen. Non-trivial effects of high-frequency excitation for strongly damped mechanical systems. Int. J. of Non-Linear Mechanics 43 (2008), 569–578.

[3] L. Sperling, B. Ryzhik, H. Duckstein. Single-Plane Auto-Balancing of Rigid Rotors. Technische Mechanik, 24 (2004) 1.

Control of Nonlinearly Coupled Oscillators Using a Method of Averaging

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Container cranes belong to the class of underactuated systems for which the design of control laws is a challenging task. We study the stabilization of certain modes of a crane system using the method of averaging [1]. Based on the local approximation to the system dynamics by a second order Taylor series expansion, the effects of nonlinear couplings are exploited to steer degrees of freedom, for which no linear actuation is available. The nonlinearity is assumed to be weak, which is shown to be a valid assumption in the proximity of certain operating points of the crane.

The Institute of Mechanics an Ocean Engineering runs a container crane test bench in the scale 1:6. It consists of a trolley and a load connected by four ropes which can be individually controlled. Furthermore, the experiment features measurements of all rope forces, the servo motion parameters for rope lengths and trolley position as well as a basic inertial measurement unit in the load itself. For verification of state estimation and control algorithms, a calibrated multi camera system is available.

For this crane system, there is no direct actuation of the rotation about the vertical axis of the load (in the literature, this motion is sometimes referred to as skew or yaw oscillation [2]). However, we enhance damping for this mode of load oscillation and provide information about potential excitation of this degree of freedom when other controllers – neglecting rotations – are applied. The synthesis model with a minimal number of degrees of freedom has been presented in [3]. This model includes two dimensional load swing as well as rotational load oscillations with inextensible ropes and individually variable rope lengths.

The quadratic coupling terms are analyzed using the method of averaging. Therefore, the solution of the linear system without quadratic coupling is used as a reference. Amplitude and phase of the linear solution vary slowly due to the nonlinear system dynamics. First, a pole placement strategy modifies the system to have specific resonant frequencies. Then, employing the variation of parameters for all degrees of freedom, amplitudes and phases are determined. Further, by means of averaging over one system period, the mean rates of change of the amplitudes and phases are determined.

From the mean rate of change, which is a function of the amplitudes and phases of the degrees of freedom involved, a design rule for control laws is deduced: A phase relation guarantees an average fastest decrease of the concerned amplitude. As a reference for the actuated degrees of freedom, a function of the states is derived which satisfies the phase condition. The state feedback control law is implemented as a variable structure reference tracking controller which is known to have good robustness properties [4]. Experimental results from the container crane test bench show the effectiveness of the approach.

References

[1] A.H. Nayfeh. Introduction to Perturbation Techniques (Vol. 3). John Wiley & Sons. (2011).

[2] U. Schaper, C. Dittrich, E. Arnold, K. Schneider, O. Sawodny. 2-DOF skew control of boom cranes including state estimation and reference trajectory generation. Control Engineering Practice, 33 (2014), 63–75.

[3] E. Kreuzer, C. Radisch. Active damping of rotational load oscillations at container cranes by individual rope length actuation. PAMM, 13(1) (2013), 461–462.

[4] B. Song, J.K. Hedrick. Dynamic Surface Control of Uncertain Nonlinear Systems. London: Springer. (2011).

On some aspects of the dynamic behavior of the softening Duffing oscillator under harmonic excitation

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The Duffing oscillator is a classic example of nonlinear dynamics and therefore taught as a standard in nonlinear vibration classes [1, 2]. The presentation deals with some problems arising, when a softening Duffing oscillator with very weak damping and harmonic excitation is under consideration and classical semi-analytical methods as Harmonic Balance, Perturbation Analysis or Multiple Scales are used. Those solutions are evaluated by several criteria and comparison with numerical analysis is performed. Additionally attractors in the parameter region of multiple stable solutions are considered.

References

[1] K. Klotter, Technische Schwingungslehre, Band 1, Einfache Schwinger, Teil B, Nichtlineare Schwingungen (Springer Verlag, 1980)

[2] I. Kovacic and M. J. Brennan (eds.), The Duffing Equation, Nonlinear Oscillators and their Behaviour (John Wiley and Sons Ltd, 2011)

1 : 3-Resonance in a Hopf-Hopf bifurcation

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In this talk the bifurcations of stationary and periodic solutions of the reduced system close to the simultaneous occurence of two Hopf bifurcations with 1 : 3-resonance are investigated. The unfolded Normal Form equations on the Center Manifold are given by

$$\dot{z}_1 = (\lambda + i\omega + A_1|z_1|^2 + A_2|z_2|^2)z_1 + A_3\overline{z}_1^2 z_2, \tag{1}$$

$$\dot{z}_2 = (\mu + 3i\omega + i\delta + A_4|z_1|^2 + A_5|z_2|^2)z_2 + A_6 z_1^3, \tag{2}$$

where λ, μ and δ denote the unfolding parameters and the coefficients $A_j \in \mathbb{C}$ of the cubic terms are obtained from the nonlinearities of the underlying model. In the absence of the resonance the last terms in both equations could be eliminated and after introducing polar coordinates $z_j = r_j \exp(i\varphi_j)$ the amplitude and phase equations would decouple. Due to the resonance the resonance angle $\psi = 3\varphi_1 - \varphi_2$ cannot be eliminated and the reduced dynamics takes place in a 3-dimensional space.

In the non-resonant case there exist two single-mode periodic solutions as primary branches and a mixed-mode quasiperiodic solution, which bifurcates from the periodic solutions in a pitchfork bifurcation. For the resonant system the term $A_6 z_1^3$ precludes the existence of a pure mode-1 solution, instead a Duffing-like szenario for the fast oscillation is observed. The pure mode-2 solution is still possible, but its loss of stability is governed by a nonlinear Mathieu equation.

Numerical investigation of the bifurcation equations shows, that also a Shilnikov szenario occurs, which leads to very chaotic behaviour.

On the chaotic behavior of the non-ideal vibrating systems

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The paper discusses the chaotic behavior of the non-ideal vibrating systems composed from structures to which an electrical motor is connected. The structural response of such systems may act like energy sink under certain conditions so that a part of the energy supplied by the source is spend to vibrate the structure rather than increasing the drive speed, according to the Sommerfeld effect [1]. According to this effect, the energy source is influenced by the response of the system and consequently, the system exhibits jumps at critical values of the energy source. The system mimics a disappearance of the energy in the resonance regions which can affect the stability of the system, sending it to chaos [2]-[4]. The chaotic behavior of non-ideal vibrating systems is characterized by an infinite number of unstable periodic orbits which becomes unstable in the least two directions in the vecinity of a transition point. The transition chaos-hyperchaos and the formation of hyperchaotic attractor are described in this paper for a structure equipped with an electromechanical vibration absorber. The growing of the higher-dimensional attractor is explained by bursting along new unstable directions. The appearance of the first unstable orbit with more than one unstable direction represents the beginning of the riddling of the basin of attraction and bubbling of the chaotic attractor. This phenomenon is a typical way by which higher-dimensional attractors grow by bursting along the new unstable direction.

References

[1] A.Sommerfeld. Beitrage Zum Dynamischen Ausban Der Festigkeitslehe. Physikal Zeitschr 3 (1902), 266-286.

[2] J.L.P.Felix, J.M.Balthazar. Comments on a nonlinear and nonideal electromechanical damping vibration absorber. Sommerfeld effect and energy transfer. Nonlinear Dynamics, 55 (2009), 1-11.

[3] L.Munteanu, V.Chiroiu, T.Sireteanu. On the response of small buildings to vibrations. Nonlinear Dynamics, 73(3) (2013) 1527-1543.

[4] L.Munteanu, C.Brişan, V.Chiroiu, D.Dumitriu, R.Ioan. Chaos-hyperchaos transition in a class of models governed by Sommerfeld effect. Nonlinear Dynamics, 78(1) (2014).

Modeling and vibration analysis of rotors with hydrodynamic MF-film bearings

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The dynamic behaviour of rigid rotors in usual passive hydrodynamical bearings is quite well recognized with the basic phenomenon of flutter-type instability under increasing rotation speed [1]. It is known that modal parameters of such two-degree of freedom system, i.e. eigen-frequencies and damping coefficients strongly depend on the rotation speed and one of the two dampings decreases and decays at the critical speed what leads to bifurcation of a rotation-dependent equilibrium and a near-critical self-excited transverse vibration with sub- or postcritical amplitude evolution [1]. On the other hand, a thin oil film provides viscoelastic rotor support that can reduce transverse vibrations caused by external excitations like unbalance and bearing housing movement.

In this paper the slide bearing serves as a semi-active viscoelastic support due to application of a magnetic or magnetorheologic fluids (MF or MRF) [2]. Both types of adaptive lubricant show changes of dynamic viscosity when exposed to magnetic field. The shear MRF effect is applied and the shear rate is proportional to the angular speed (ω) of the rotor, even with the basic for MF/MRF Bingham fluid model, the effective lubricant viscosity depends on both - magnetic field (H) and rotation speed. Thus, under increasing angular speed the rotor behaves like a smart structure, providing quickly a necessary uplift force [3], increasing flutter critical speed and reducing viscous resistance due to drop of the viscosity. The MF/MRF properties are adopted from the literature [4, 5]. The dynamic analyses of stability, bifurcation and resonant behaviour are made for a rigid rotor supported in one rolling and one slide bearing, thus performing transverse vibrations under both nominally constant load (Q) and additionally harmonic forcing, with the giroscopic effect taken into account. The equation of the rotor transverse motion can be expressed as follows:

$$\frac{d\mathbf{u}}{dt} = f\left(\mathbf{u}, \omega, Q, H, t; p\right)$$

where the notations not defined above are: $\mathbf{u}$ - 4-dimensional vector of displacements and velocities around nontrivial equilibrium within the bearing clearance, $f\left(\mathbf{u}, \omega, Q, H, t; p\right)$ - vector function derived from the rotor angular momentum law, including nominal transverse load, fluid-film forces resulting from the Reynolds equation for the fluid flow and possible harmonic excitation in case of resonant behaviour and p - 7-dimensional quantity representing all the mass and geometric parameters of the rotor.

The results are presented in form of a 3D dimensionless rotor stability map, modal parameters as functions of rotor speed and magnetic field, resonant amplitude behaviour under unbalanced and kinematic excitations affected by the magnetic field.

References

[1] W. Kurnik. Hopf Bifurcation in the Dynamics of a Rotor Bearing System. W. Schiehlen (Ed.), Nonlinear Dynamics in Engineering Systems. Springer Verlag. (1990), 181-188.

[2] N. Tipei. Theory of lubrication with ferrofluid: application to short bearings. Transactions of the ASME. (1982), 104, 510-515.

[3] A. Perek. Stability and Resonant Behaviour of a Rotor in Slide Bearings with Bingham Fluid As a Lubricant. Machine Dynamics Research. (2012), Vol. 36, No 2, 107-118.

[4] S. Odenbach, S. Thurm. Magnetoviscous effects in ferrofluids. Springer-Verlag, Berlin, Heidelberg. (2002), 594, 185-201.

[5] R. Patzwald. Magnetische Fluessigkeiten als Schmierstoff in hydrodynamischen Gleitlagern. Disertation. (2001).

A DAE formulation for geared rotor dynamics including frictional contact between the teeth

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A DAE approach is presented for geared rotor dynamics simulations with rigid helical evolvent gears. It includes the normal contact force between the teeth as well as tangential components.

Given the evolvent tooth flank geometry of gear 1 and gear 2 ($\boldsymbol{x}^{(1)}$ and $\boldsymbol{x}^{(2)}$) [1], the contact line $\boldsymbol{x}_c$ and the velocity difference in the contact $\dot{\boldsymbol{x}}^{(2)}(\boldsymbol{x}_c) - \dot{\boldsymbol{x}}^{(1)}(\boldsymbol{x}_c)$ are found. The requirement of no penetration of the teeth yields a non-holonomic constraint and the contact normal force [2]. The tangential force is obtained by applying Coulomb's friction model [3], even though in experiments all situations between dry friction and hydrodynamic lubrication are observed [4, 5, 6]. The friction force is the integral tangential force acting on all flanks. In addition, a friction caused torque acts on the center of rotation of the gears.

The approach is used to investigate the dynamics of two rotors which are connected by gears. Both of them have three translational DoFs. The driving rotor has a given angular speed, while the driven rotates unrestrainedly and is connected to a rotational damper.

Two different solutions of the problem are compared. The first solution is produced by directly integrating the index-2 DAE using a modified Runge-Kutta algorithm. Because of the periodicity of the geometry, it is obvious that the stationary solution is periodic. For this reason, a harmonic ansatz is applied to calculate the variables' amplitudes as a second solution.

The two solutions are compared and the evolution of the amplitudes with respect to the angular speed is studied.

References

[1] K. Zirpke. Zahnräder. VEB-Fachbuchverlag (1978).

[2] G. Jehle, A. Fidlin. Friction induced vibrations in shift gearboxes. Z. Angew. Math. Mech. 94 (2014), 911–916.

[3] K.L. Johnson. Contact mechanics. Cambridge university press (1987).

[4] K.F. Martin. A Review of Friction Predictions in Gear Teeth. Wear 49 (1978), 201–238.

[5] G.H. Benedict, B.W. Kelley. Instantaneous Coefficients of Gear Tooth Friction. ASLE transactions 4(1) (1961), 59–70.

[6] S. He, S. Cho, R. Singh. Prediction of Dynamic Friction Forces in Spur Gears using Alternate Sliding Friction Formulations. J. Sound Vib. 309 (2008), 843–851.

Adaptive Fuzzy Sliding Mode Controller and Observer for a Dive Cell

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Water particles move on orbitals under the influence of surface gravity waves. These orbitals are usually visualized by optical methods, such as the PIV method. We develop a dive cell which is supposed to reach and stay on an isobar.

Since for that case the average density of the dive cell equals the density of the surrounding water, the dive cell describes the same trajectories as the water particles. The dive cell is composed of a pressure sensor, a micro controller and an actuator to change the volume of the dive cell. Furthermore, an acceleration sensor and a mounted SD card provide the possibility to analyze the kinematics of the dive cell, hence of the water particle trajectories. In order to reach and stay on a certain isobar with the dive cell an adaptive fuzzy sliding mode controller including an observer is developed and compared to a plain vanilla sliding mode controller.

In a first step the equations of motion are derived under consideration of quadratic viscous damping as well as the actuator dynamics (single integrator dynamics):

$$\ddot{z} = \frac{1}{m}\left[\rho_w g(V_0 - V_w) - mg - k\dot{z}|\dot{z}|\right] \quad , \tag{1}$$

$$V_w = b \cdot u \quad , \tag{2}$$

with the variable z as the dive depth, the mass of the dive cell m, water density ρ_w, gravitational acceleration g and a viscous damping coefficient k. The value of V_w is the volume of water in the piston of the dive cell, while V_0 represents the volume corresponding to zero buoyancy which in turn should reasonably correspond to the middle piston position. Due to the nonlinear damping and the uncertain parameters, such as the hydrodynamic added mass, a sliding mode approach is chosen for the design of controller and observer, which has proven to be a well-established methodology for robust nonlinear controller synthesis in the presence of modeling imprecisions and external disturbances. After the derivation of the plain vanilla sliding mode controller, we enhance the controller by an adaptive term, which is computed via an adaptive fuzzy algorithm according to [1]. Since the controller requires access to all of the system states while the noisy pressure measurement provides the position state only, we further augment the system by a sliding mode observer. The adaptive fuzzy sliding mode controller shows superior performance in comparison to the plain vanilla one which is demonstrated by means of numerical simulations.

References

[1] Wallace M. Bessa, Max S. Dutra, Edwin Kreuzer. Depth control of remotely operated underwater vehicles using an adaptive fuzzy sliding mode controller. Robotics and Autonomous Systems. 56 (2008), 670–677.

The Vibrational Behavior of Coupled Bladed Disks with Variable Rotational Speed

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Low pressure steam turbine blades are subjected to high static and dynamic loads during operation. These loads strongly depend on the turbine's rotational speed, leading to entirely new load conditions. To avoid high dynamic stresses due to the forced vibrations, a coupling of the blades, such as shrouds or snubber coupling, is applied to reinforce the structure. In this work the influence of the rotational speed on the vibration behavior of shrouded blades is investigated. Two fundamental phenomena are considered: the stress stiffening and the spin softening effect. Both effects are caused by centrifugal forces and affect the structural mechanical properties, i.e. the stiffness matrix $\mathbf{K}$, of the rotating system. Since the rotational speed Ω appears quadratically, it is possible to derive the stiffness matrix as a second order matrix polynomial in Ω^2 [3].
In the case of shrouded blades, contact forces between neighboring blades must be taken into account. The contact status and the pressure distribution in particular is strongly influenced by the rotational speed, respectively, centrifugal forces, caused by the untwisting and radial deformation of the blades. For the calculation, a three dimensional structural mechanical model including a spatial contact model is considered. The solution of the nonlinear equations of motion is based on the well known Multiharmonic Balance Method [2]. Here, the nonlinear forces are computed in the time domain and transferred in the frequency domain by the use of the Fast Fourier Transformation (FFT), also known as the Alternating Frequency Time method (AFT) [1].

References

[1] T.M. Cameron, J.H. Griffin. An Alternating Frequency/Time Domain Method for Calculating the Steady-State Response of Nonlinear Dynamic Systems. Journal of Applied Mechanics (1989), Vol. 56, pp. 149-154.

[2] C. Siewert, L. Panning-von Scheidt, J. Wallaschek, C. Richter. Multiharmonic forced response analysis of a turbine blading coupled by nonlinear contact forces. Journal of Engineering for Gas Turbines and Power 132.8 (2010): 082501.

[3] A. Sternchüss, E. Balmès. On the reduction of quasi-cyclic disk models with variable rotation speeds. Proceedings ISMA (2006).

Multi-mode model of a piezomagnetoelastic energy harvester under random excitation

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The transformation of ambient vibrational energy into electric energy through the use of piezoelectric energy harvesting devices has been the subject of numerous investigations [1]. A commonly studied energy harvesting device which performs especially well under broadband excitation is the piezomagnetoelastic energy harvester investigated by Erturk *et al.* [2] which is usually discretized for the fundamental vibration mode resulting in a single-mode model.

This contribution presents the study of a multi-mode model of the piezomagnetoelastic energy harvester under random excitation. The probabilty density function (PDF) is computed as the solution of the corresponding Fokker-Planck equation using a Galerkin type method [3, 4]. Based on the PDF, the resulting voltage variance is computed as a measurement for the expected power output as demonstrated in [5]. The results of the multi-mode model are then compared with the results of the single-mode model.

References

[1] R.L. Harne, K.W. Wang. A review of the recent research on vibration energy harvesting via bistable systems. Smart Mater. Struct. **22** (2013).

[2] A. Erturk, J.Hoffmann, D.J.Inman. A piezomagnetoelastic structure for broadband vibration energy harvesting. Appl. Phys. Lett. **94** (2009).

[3] U. von Wagner, W.V. Wedig. On the calculation of stationary solutions of multi-dimensional Fokker-Planck equations by orthogonal functions. Nonlinear Dyn. **21** (2000), 289–306.

[4] W. Martens. On the solution of the Fokker-Planck-Equation for multi-dimensional nonlinear mechanical systems (Shaker Verlag, Aachen, 2014).

[5] W. Martens, U. von Wagner, G. Litak. Stationary response of nonlinear magneto-piezoelectric energy harvester systems under stochastic excitation. Eur. Phys. J. S. T. **222** (2013), 1665–1673.

Optimal impedance load of a bistable energy harvester

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Within this contribution we investigate a bistable energy harvester regarding its optimal impedance load. A bistable energy harvester exhibits different type of oscillations: Intra-well (about a stable equilibrium), cross-well (between the wells) and inter-well (about the unstable equilibrium). The occurring oscillation type depends for instance on the excitation parameters or the initial conditions. It already has been observed ([1]) that the optimal impedance, which allows to maximize the power output, varies for each occurring oscillation type. In our investigations we complement these findings with analytical and numerical calculations. For our analysis we examine the non-dimensionalized coupled equations of a bistable energy harvester

$$\begin{aligned} x''(t) + \mu\, x'(t) - \delta\, v(t) - \alpha\, x(t) + \beta\, x(t)^3 &= y(t) \\ \phi\, v(t) + \eta\, x'(t) + v'(t) &= 0 \end{aligned}$$

whereby

$$\begin{aligned} & y(t) = \hat{f} \sin(\Omega\, t) \\ & \mu\ , \delta\ , \beta\ , \phi\ , \eta\ > 0. \end{aligned}$$

As indicated we will limit our analysis to harmonic excitations.

The results show that for intra-well and inter-well oscillations more than one optimal impedance load can be found, at which the power output exhibits a maximum. These impedance loads can strongly deviate compared with the ones of the linearized system. The optimal impedance loads also show strong sensitivity to changes in the excitation amplitude $\hat{f}$.

References

[1] S. C. Stanton, B. A.M. Owens, B. P. Mann. Harmonic balance analysis of the bistable piezoelectric inertial generator. Journal of Sound and Vibration 331 (2012), 3617–3627.

On vibrations in non linear, forced, friction-excited systems

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In development of friction brakes the trend is towards new experimental methods which include external excitation of brakes systems during dynamometer testing [1]. The underlying idea is to be able to identify in a more efficient and reliable manner parameter regions in which limit cycle oscillations can occur. Ideally, testing could be conducted in parameter regions where the steady sliding state of the non linear, non forced, friction-excited systems is still stable. In this context, this contribution discusses vibrations in non linear, forced, friction-excited systems of the form

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{D}\dot{\mathbf{q}} + (\mathbf{K} + \mathbf{N})\,\mathbf{q} + \mathbf{f}_{\mathrm{NL}}\,(\dot{\mathbf{q}}, \mathbf{q}) = \mathbf{f}_{\mathrm{ext}}\,(t)\,,$$

where $\mathbf{q}$ is the vector of generalized coordinates. $\mathbf{M}$, $\mathbf{D}$, $\mathbf{K}$ and $\mathbf{N}$ are the mass matrix, the damping matrix, the stiffness matrix and the circulatory matrix, respectively. $\mathbf{f}_{\mathrm{NL}}\,(\dot{\mathbf{q}}, \mathbf{q})$ and $\mathbf{f}_{\mathrm{ext}}\,(t)$ are the vectors of non linear restoring forces and external excitation. A minimal model (see [2] for more details) is employed to numerically reproduce the test procedures, i.e. stepped sinusoidal excitation of the system in the frequency range of interest. In pre-flutter studies, i.e. studies in parameter regions where the steady sliding state is stable, frequency response functions for different excitation directions and amplitudes are calculated. Based on these results we try to challenge and derive criteria which allow for the estimation of stability borders. Energy terms are used as supporting measures. In post-flutter studies, i.e. studies in parameter regions where limit cycle solutions exist, transient and steady-state solutions are determined. Regions of entrainment are estimated.

The results reveal that non linear, forced, friction-excited systems are rich in phenomena. The examined criteria for the estimation of stability borders work reliably in vicinity of the Hopf point. A further enhancement of the methods is therefore necessary to ensure proper interpretation of test results also in larger distance from the bifurcation point which is a priori unknown.

References

[1] P. Stegmann, S. Kruse, K. Augsburg. Comparison of Different Excitation Concepts for the Characterization of Brake Systems Robustness regarding Squeal. In: Proceedings of Eurobrake 2014, Lille, France, 2014.

[2] S. Kruse, M. Tiedemann, B. Zeumer, P. Reuss, N. Hoffmann, H. Hetzler. The influence of joints on friction induced vibrations in brake squeal. J. Sound Vibr., in press.

A finite element model for a soft robot equipped with a flexible limb

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In many industrial and biomedical fields, robots and automata are used to reduce dangerous or repeatable tasks people do not wish to perform, to overcome human limitations in strength and speed, or to operate in hostile environments where humans are unable to work [1, 2]. Recently, a new kind of robot, known as a soft robot, has been the object of extensive research. This research has been motivated by the potential benefits of soft robots in applications to healthcare, cooperative human assistance, service robots and biomechanically compatible interactions [2, 3]. With the absence of a skeleton-like structure and the use of soft materials, complex locomotion or movements of the soft robot are possible. However new control strategies and mathematical models are required to advance the field of soft robots. The development of accurate yet feasible models is a particularly challenging problem.

In the presented research, a dynamic model for a soft robot that features in a new locomotion scheme proposed by Zhou et al. [4] is developed and analyzed. The soft robot studied consists of a single flexible limb attached to a rigid mass. By carefully controlling the intrinsic curvature of the limb and exploiting friction forces, locomotion, in principle, is possible. Here, we develop a rich model for the robot by using a finite element based model for the soft limb. The feasibility of the proposed locomotion scheme is then assessed and conclusions generated for expanding the locomotion scheme to more complex soft robot designs.

References

[1] M. Hägele, K. Wegener. Service-Roboter-Visionen. Hanser Verlag, (2004).

[2] C. Majidi. Soft robotics: A perspective - current trends and prospects for the future. Soft Robotics SoRo, 1 (2013), 5 –11.

[3] S. Kim, C. Laschi, B. Trimmer. Soft robotics: a bioinspired evolution in robotics. Trends in Biotechnology, 31 (2013), 287 –294.

[4] X. Zhou, C. Majidi, O.M. O'Reilly. Flexing into Motion: A Locomotion Mechanism for Soft Robots. Flexing into Motion: A Locomotion Mechanism for Soft Robots, Submitted for publication (2014)

On the influence of vibrations on macroscopic frictional contacts

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Ultrasonic vibrations in the micrometer amplitude range have been proven to be capable to influence frictional contacts with respect to decreasing friction coefficients in experiments on small test set-ups solely loaded by their small weight [1, 2]. In this presentation experimental results on applying this effect to large objects are presented. To get large vibration amplitudes and velocities detailed pre-investigation on the excitation of the structures in contact have been performed. These structures are investigated in a roller rig designed for these purposes.

References

[1] V. L. Popov, J. Starcevic, and A. E. Filippov. Influence of Ultrasonic In-Plane Oscillations on Static and Sliding Friction and Intrinsic Length Scale of Dry Friction Processes. Tribology Letters 39 (2009), 25–30.

[2] V. L. Popov, J. Starcevic, and E. Teidelt. Influence of In-Plane and Out-Of-Plane Ultrasonic Oscillations on Sliding Friction. Energy and Environmental Aspects of Tribology. (2010), 128–136.

On the Effect of Contact Compliance on Vibrational Smoothing of Dry Friction

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High-frequency vibrations may be utilized in order to smooth the characteristics of dry friction at low sliding velocities and, consequently, quench undesired friction induced phenomena such as stick-slip motion [1]. Many studies have been published so far, most of them using classical Coulomb friction models and yielding conpact results. Unfortunately, the agreement with related experimental results is insufficient. As the Coulomb model overestimates the smoothing effect, improved modelling seems to be necessary [2].
In order to overcome one of the main disadvantages of the Coulomb friction model, contact compliance is considered here. Based on the friction model suggested by Dahl, the effect of longitudinal and transverse high-frequency vibrations on a 1-DoF friction oscillator is investigated.
Due to some shortcomings of Dahl's friction model, further improvement of the model is discussed. The *LuGre*-model can be regarded as an extension of Dahl's model capturing velocity dependencies. The model proposed by Dupont et al. additionally accounts for stiction regimes and is therefore the most evolved friction model used within this contribution [3].
It is shown, that the qualitative agreement between measurements and simulations is highly improved when accounting for contact compliance. Using Dahl's model, compact analytical results can be derived from approximation, while the extended models have to be evaluated numerically. Quantitative results can only be obtained after experimental determination of model parameters.

References

[1] Thomsen, J. J.: *Using fast vibrations to quench friction-induced oscillations.* Journal of Sound and Vibration 228 (1999), 1079–1102.

[2] Storck, H. et al.: *The effect of friction reduction in presence of ultrasonic vibrations and its relevance to travelling wave ultrasonic motors.* Ultrasonics 40 (2002), 379–383.

[3] Dupont, P. et al.: *Single state elastoplastic friction models.* IEEE Transactions on Automatic Control 47 (2002), 787–792.