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Scientific Program - Timetable

Sunday 22	Time	Monday 23	Tuesday 24	Wednesday 25	Thursday 26	Friday 27
	9:00		Contributed sessions (15 in parallel)	Plenary Lecture Moritz Diehl	Contributed sessions (15 in parallel)	Contributed sessions (14 in parallel)
	9:30					
	10:00	Registration		von Mises prize lecture		
	10:30					
	11:00		Coffee Break	Coffee Break	Coffee Break	Coffee Break
	11:30		Plenary Lecture Thomas Böhlke	General Assembly	Plenary Lecture Ferdinando Auricchio	Contributed sessions (11 in parallel)
	12:00					
	12:30		Lunch	Lunch	Lunch	
	13:00	Opening				
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	15:00	Plenary Lecture Giovanni Galdi	Plenary Lecture Nikolaus Adams		Plenary Lecture Stanislaw Stupkiewicz	
	15:30					
Registration pre-opening	16:00	Coffee Break	Coffee Break Poster session	Coffee Break	Coffee Break Poster session	
	16:30	Minisymposia & Young Reseachers' Minisymposia (10 in parallel)	Contributed sessions (14 in parallel)	Contributed sessions (15 in parallel)	Contributed sessions (15 in parallel)	
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S01: Multi-body dynamics

Multibody dynamics enables the simulation of a wide variety of systems, all characterized by having multiple parts in relative motion with one another. Applications span from biological to engineering systems, requiring diverse capabilities which range from real-time simulation to high fidelity modeling of complex multidisciplinary systems. Goal of this mini-symposium is to present a view on the latest developments in models and advanced numerical methods in multibody dynamics. Focus is on techniques that enable applications to complex real-life problems.

Impulse-based control of simple oscillators within the nonsmooth mechanics approach

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In this contribution, we present an impulse-based control scheme for a simple spring-mass oscillator subject to set-valued friction forces. In contrast to PD or PID control laws, the proposed control law prevents non-zero steady state errors and limit cycles. Motivated by Wouw and Leine [1], it is shown that impulse-based control laws can lead to satisfactory and in particular robust behavior under the account of uncertain system parameters, i.e., mass or static friction, if the oscillator starts sticking. An overview on frictional oscillators for which stick-slip phenomena occur and an introduction to standard control laws indicate the need of problem-specific extensions. In contrast to reference [1], the equations of motion are extended by a simple model of the actor dynamics allowing for a practical realization. By using the framework of nonsmooth mechanical systems and applying timestepping schemes, it is shown how to overcome the drawbacks of ordinary control techniques. Apart from a continuous state feedback, we apply an impulsive feedforward in the case of sticking which prevents permanent sticking and resolves unwanted behavior like steady state errors or limit cycles.

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Learning Robot Force/Position Control for Repetitive High Speed Applications with Unknown Non–Linear Contact Stiffness

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In many industrial applications manipulators are used to perform repetitive force controlled tasks. Typically such tasks are polishing, grinding, assembly as well as endurance testing of machine parts. The repetitive nature of such tasks allow for using iterative learning control (ILC) methods [1] or adaptive learning feed–forward control [2]. For such a controller it is necessary that the feed–back controlled system is stable, so that a learning feed–forward control, for example, minimizes a resulting error from one repetition to the next.

Considering a task, where a robot processes the same kind of workpiece in a recurring manner, the end–effector of the robot has to provide a predefined contact force while following a trajectory along the workpiece. To achieve this goal, a parallel force/position robot control, as suggested in [3], is best suited. Thereby, the force control manipulates a desired end–effector trajectory ${}_I\mathbf{r}_d$, of a position controlled system, in such a way, that the desired force ${}_I\mathbf{f}_d$ is reached. The force control law is described by the differential equation $\mathbf{K}_{A,p}\,{}_I\ddot{\mathbf{r}}_c + \mathbf{K}_{V,p}\,{}_I\dot{\mathbf{r}}_c = {}_I\mathbf{f}_d - {}_I\mathbf{f}$, where $\mathbf{K}_{A,p}$ and $\mathbf{K}_{V,p}$ denote positive definite controller parameters and the vector ${}_I\mathbf{f}$ represents the measured force in the inertial frame (I). Using the parallel composition ${}_I\mathbf{r}_r = {}_I\mathbf{r}_d + {}_I\mathbf{r}_c$, with the solution ${}_I\mathbf{r}_c$ of the above differential equation, the reference trajectory ${}_I\mathbf{r}_r$ is calculated. This reference trajectory serves as input of the position controlled system.

Consequently following a fast desired force trajectory leads to various difficulties. First of, all the exact contact position as well as the contact stiffness is not always known. Further, the stiffness may vary from one point on the surface to another. These problems can be addressed by a separate identification of the stiffness and measuring the contact point position of different kinds of workpieces. To avoid this time consuming work, the idea, inspired by ILC and learning feed–forward control, is to divide the task into an *on–line* task, where the robot is actually moving, and an *off–line* task, where the computation and trajectory correction is done. Different to classical ILC methods, the previous mentioned issues are overcome by controlling the task with stretched time by a factor t_{scale}, depending on the task. In this time stretched *on–line* task the robot is able to follow the desired position and force trajectory. By replacing the desired trajectory ${}_I\mathbf{r}_{d,j}$ in the next iteration step $(j+1)$ with the stored reference trajectory ${}_I\mathbf{r}_{r,j}$ from the actual step (j) the learning law ${}_I\mathbf{r}_{d,j+1} = {}_I\mathbf{r}_{d,j} + {}_I\mathbf{r}_{c,j}$, depending on the force error, is developed. To improve performance, the trajectory from the start to the contact point (and back from the last contact point to the start point) is calculated *off–line* using splines with continuous transition conditions at start and end. However, if a condition for the force error, for example $\Delta\mathbf{F}_{max} > \sup_{0\le\tau\le T} \|{}_I\mathbf{f}_d(\tau) - {}_I\mathbf{f}(\tau)\|$, where $\Delta\mathbf{F}_{max}$ is an upper error limit and T is the cycle time of one repetition, is satisfied, the time scale factor t_{scale} is reduced and a next iteration is started. Now the force controller only acts on small force errors, arising from contact damping and system dynamics. This iterations are done till $t_{scale} = 1$, so that the normal process time is reached. Thereby the desired trajectory ${}_I\mathbf{r}_{d,j}$ is trained for this group of workpieces and the learning process is stopped. For all subsequent workpieces, the process can be started with the original process time and force errors, arising from a slightly misaligned workpiece or other influences, are corrected again by the training process.

Summarizing, this paper proposes a learning robot force/position control for high speed force trajectory following. Experimental results, where a test object with non–linear stiffness is mounted at the end–effector of the industrial gantry robot and a repetitive disturbance acting at the contact point are presented.

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An Optimal Control Approach to the Simulation of Problems with Servo Constraints

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Mechanical systems with servo (or control) constraints appear in several crane models. A typical example is a robot for which we like to prescribe the trajectory of the end effector by a given number of input variables. Such models lead to DAEs of high index, often we deal with index-5 problems.

The simulation of such high-index DAEs is a challenging problem because of the involved stability issues known from DAEs. Due to the weak coupling of input and output, small changes in the desired trajectory may lead to high peaks in the input variables. Hence, index reduction techniques are inevitable for numerical simulations.

We consider the example of two mass points which are coupled by a spring. The task is then to control the movement of the second point whereas we may only actuate a force at the first [1]. This is a minimal example of a mechanical system with a servo constraint and gives a DAE of index 5.

In this talk, we relax the constraint of following the given trajectory exactly and consider an optimal control problem instead. For this, we consider the dynamics of the mechanical system and introduce a cost functional which penalizes deviations from the trajectory as well as the input and its derivatives. Of course, the DAE problems are not simply gone, as it becomes visible within variations of the penalization parameters. However, we can achieve good performances at lower costs and for trajectories that would not be allowed in the direct approach. We illustrate this fact in several numerical experiments.

References

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An explicit approach for time-optimal trajectory planning for kinematically redundant robots

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Kinematic redundancy describes a manipulator's morphological property of featuring more joints $\mathbf{q} \in \mathbb{R}^n$ than necessary to assume any configuration in its task space $\mathbf{z}_\mathrm{E}^\top = \begin{pmatrix} \mathbf{r}_\mathrm{E}^\top & \boldsymbol{\varphi}_\mathrm{E}^\top \end{pmatrix} \in \mathbb{R}^m$ of given dimension, i.e. $m < n$. During the last years, the importance of kinematically redundant serial robots has risen due to striking advantages such as their improved flexibility and adaptiveness in structured workspaces. Their inherent capability of null space motion results in remarkable performance compared to conventional, non-redundant serial robots. Attempting to increase the cost-effectiveness of industrial processes, introducing time-optimal trajectories may yield economical advantages due to reduced motion cycle times.

While for non-redundant serial robots this problem has been addressed exhaustively, it has not been sufficiently covered for redundant manipulators. Well-known methods for obtaining minimum-time trajectories for non-redundant setups are not applicable to their full extent since for redundant robots not only the physical construction, but also the mathematical structure of the resulting equations differs greatly, e.g. the existence of a null space, i.e. joint velocities $\dot{\mathbf{q}}_0$ exist that have no influence on the current task space velocity of the end-effector such that $\dot{\mathbf{z}}_\mathrm{E} = \mathbf{J}(\mathbf{q})\,\dot{\mathbf{q}}_0 = \mathbf{0}$ where $\mathbf{J}$ denotes the geometric Jacobian. Additionally, no analytic inverse kinematics can be obtained since the direct kinematics equation system $\mathbf{z}_\mathrm{E} = \mathbf{f}(\mathbf{q})$ is under-determined.

In [1] a method for obtaining minimum-time trajectories along predefined, parameterized task space paths $\mathbf{z}_\mathrm{E}(s)$ of serial robots with one redundant degree of freedom is presented where s denote as scalar path coordinate. The robot joints are divided in a *non-redundant* set of joints $\mathbf{q}_\mathrm{nr} \in \mathbb{R}^m$ and a *redundant* part $\mathbf{q}_\mathrm{r} \in \mathbb{R}^{n-m}$. Time-optimal trajectories are obtained for the path parameter $s(t) \in [0,1]$ where $s(0) = 0$, $s(t_\mathrm{E}) = 1$, and the *redundant* joints $\mathbf{q}_\mathrm{r}(t)$ by means of an optimal control problem initiating null space motion in order to reduce the trajectory end time t_E. The resulting trajectories are only $\mathcal{C}^1$ continuous, which limits the method's range of possible applications.

The present contribution adapts the separation approach from [1] and is applied to serial robots with one redundant degree of freedom. Instead of finding time-optimal trajectories by means of an optimal control problem, time-dependent multi-interval B-spline curves of degree d for the path parameter $s = s(t)$ and the *redundant* joint $q_\mathrm{r} = q_\mathrm{r}(t)$ are assumed whose control points $\mathbf{c}_\mathrm{nr}$ and $\mathbf{c}_\mathrm{r}$ and the common end time t_E are optimization variables $\mathbf{x}$ of an optimization problem, i.e. $\mathbf{x}^\top = \begin{pmatrix} t_\mathrm{E} & \mathbf{c}_\mathrm{nr}^\top & \mathbf{c}_\mathrm{r}^\top \end{pmatrix}$. The *non-redundant* joint positions are then found using analytic inverse kinematics for the end-effector position computed by means of the parameterized path and the position of the *redundant* joint, i.e. $\mathbf{q}_\mathrm{nr} = \mathbf{q}_\mathrm{nr}(\mathbf{z}_\mathrm{E}(s), q_\mathrm{r})$. The optimization problem is subjected to the dynamics of the robot, and physical and technological constraints such as limitations in the joint velocities $\dot{\mathbf{q}}$, the joint accelerations $\ddot{\mathbf{q}}$, or the joint torques $\mathbf{Q}$. While changing the position of the control point influences the general shape of the B-spline curves, their property of local approximation yields advantageous local adaptiveness. A solution for finding the initial values for the optimization problem is to obtain them by means of a least-squares B-spline curve approximation of the resulting trajectory of a numerical first-order inverse kinematics approach for a generic task space trajectory along the predefined path.

With an appropriate selection of the *redundant* degree of freedom, the manipulator's kinematic redundancy can be explicitly exploited. The resulting joint trajectories can be expressed as piecewise polynomial functions that are continuously differentiable d times w.r.t. time t. Unlike most conventional methods trajectory data can be stored in a minimalistic way since only general parameters of the B-spline curves are required to generate the trajectory functions.

The presented approach was applied to the example of a planar three-link SCARA with straight lines as its given end-effector paths. Kinematic redundancy is introduced by only considering the Cartesian position of the end-effector in the plane but not its orientation as task space coordinates, i.e. $\mathbf{z}_\mathrm{E} = \mathbf{r}_\mathrm{E} \in \mathbb{R}^2$. As the motion is subject to joint torque constraints, continuous feed-forward torques are required to be computed yielding $d \geq 3$. Simulation results demonstrate time-optimality of the resulting trajectories.

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A numerical method for the servo constraint problem of underactuated mechanical systems

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Servo constraints are used in the inverse dynamics simulation of discrete mechanical systems, especially in trajectory tracking control problems [1]. The desired system outputs described in terms of system states are treated as servo constraints [2]. The governing equations take the form of differential algebraic equations (DAEs), in which servo constraints are algebraic equations. In fully actuated multibody systems, the control inputs are solved from the equations of motion by model inversion, as the input distribution matrix is nonsingular and invertible. For underactuated multibody systems the number of degrees of freedom is greater than the number of control inputs and the input distribution matrix can not be inverted anymore. In contrast to passive constraints, the realization of servo constraints with the use of control forces can range from orthogonal to tangential [3]. Therefore the determination of control inputs that force the underactuated system to realize the partly specified motion, is difficult. For example the (differentiation) index can exceed three for differentially flat underactuated systems. Thus index reduction techniques are necessarily applied to reduce the index, such as Blajer's projection approach [3], [4]. In the present work we apply index reduction by minimal extension [5] to differentially flat underactuated systems, such as overhead and rotary cranes. We show that the index can be reduced from five to three and even to one. We consider as well nonflat underactuated multibody systems where the stability of the internal dynamics is also of paramount importance and ensures the controllability of the system [6], [7], [8].

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Active damping control for an underactuated multibody system

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Light-weight robots and manipulators stand out due to their very good weight-to-load ratio and a low energy consumption. Unfortunately, the light-weight design yields a lower stiffness, which results in undesired elastic deformations, especially during high-speed working motion. One way to limit these unwanted oscillations can be implemented with modifications of the command signals, e.g. by input shaping or pre-computed feed-forward control [1]. However, a feedback control concept has the ability to provide a fast compensation of elastic deformations due to high-speed working motions, in case of unwanted environment contact or in the presence of other external disturbances.

In this contribution, an active damping control (ADC) for fast moving manipulators with significant structural elasticities is presented. This approach does not require additional actuators and it is suitable for a large variety of manipulators that exhibit structural vibrations, as long as they possess as many control inputs as rigid degrees of freedom. The ADC requires a robust and accurate position control for the actuated joints or bearings, which can be achieved by cascaded position, velocity, and current controllers together with a friction compensation. Then, the active damping control can be implemented by superimposing the position and velocity trajectories of the cascaded controllers with a motion that stabilizes the elastic deformation. Because of the underlying position control, this approach is applicable even in the presence of dry friction in the actuated joints or bearings.

The design of the ADC is based on a flexible multibody model of the system. The equations of motion can be stated as

$$\begin{bmatrix} \boldsymbol{M}_{\mathrm{rr}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}}) & \boldsymbol{M}_{\mathrm{re}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}}) \\ \boldsymbol{M}_{\mathrm{er}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}}) & \boldsymbol{M}_{\mathrm{ee}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}}) \end{bmatrix} \begin{bmatrix} \ddot{\boldsymbol{q}}_{\mathrm{r}} \\ \ddot{\boldsymbol{q}}_{\mathrm{e}} \end{bmatrix} = \begin{bmatrix} \boldsymbol{f}_{\mathrm{r}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}},\dot{\boldsymbol{q}}_{\mathrm{r}},\dot{\boldsymbol{q}}_{\mathrm{e}}) \\ \boldsymbol{f}_{\mathrm{e}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}},\dot{\boldsymbol{q}}_{\mathrm{r}},\dot{\boldsymbol{q}}_{\mathrm{e}}) \end{bmatrix} + \begin{bmatrix} \boldsymbol{B}_{\mathrm{r}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}}) \\ \boldsymbol{B}_{\mathrm{e}}(\boldsymbol{q}_{\mathrm{r}},\boldsymbol{q}_{\mathrm{e}}) \end{bmatrix} \boldsymbol{u}\,, \tag{1}$$

in which $\boldsymbol{M}$ is the mass matrix, $\boldsymbol{q}$ is the vector of the generalized coordinates and $\boldsymbol{f}$ is the vector of generalized forces. The product of the input matrix $\boldsymbol{B}$ and the control inputs $\boldsymbol{u}$ account for the forces of the linear direct drives. The subscripts r and e distinguish the rigid and the elastic part, respectively. For perfect feedback controllers, the dynamics of the elastic part can be approximated by

$$\boldsymbol{M}_{\mathrm{ee}}(\boldsymbol{q}_{\mathrm{r,d}},\boldsymbol{q}_{\mathrm{e}})\ddot{\boldsymbol{q}}_{\mathrm{e}} = \boldsymbol{f}_{\mathrm{e}}(\boldsymbol{q}_{\mathrm{r,d}},\boldsymbol{q}_{\mathrm{e}},\dot{\boldsymbol{q}}_{\mathrm{r,d}},\dot{\boldsymbol{q}}_{\mathrm{e}}) - \boldsymbol{M}_{\mathrm{er}}(\boldsymbol{q}_{\mathrm{r,d}},\boldsymbol{q}_{\mathrm{e}})\ddot{\boldsymbol{q}}_{\mathrm{r,d}}\,. \tag{2}$$

The linearization of Eq. (2) with respect to the elastic coordinates and their derivatives about a perturbed working motion $\ddot{\boldsymbol{q}}_{\mathrm{r,d}} = \ddot{\bar{\boldsymbol{q}}}_{\mathrm{r,d}} + \Delta\ddot{\boldsymbol{q}}_{\mathrm{r,d}}$ yields the linear dynamics

$$\bar{\boldsymbol{M}}_{\mathrm{ee}}(\boldsymbol{q}_{\mathrm{r,d}})\Delta\ddot{\boldsymbol{q}}_{\mathrm{e}} + \bar{\boldsymbol{D}}_{\mathrm{ee}}(\boldsymbol{q}_{\mathrm{r,d}})\Delta\dot{\boldsymbol{q}}_{\mathrm{e}} + \bar{\boldsymbol{K}}_{\mathrm{ee}}(\boldsymbol{q}_{\mathrm{r,d}})\Delta\boldsymbol{q}_{\mathrm{e}} = \bar{\boldsymbol{H}}(\boldsymbol{q}_{\mathrm{r,d}})\Delta\ddot{\boldsymbol{q}}_{\mathrm{r,d}} + \bar{\boldsymbol{d}}(\boldsymbol{q}_{\mathrm{r,d}},\dot{\boldsymbol{q}}_{\mathrm{r,d}},\ddot{\bar{\boldsymbol{q}}}_{\mathrm{r,d}})\,, \tag{3}$$

in which $\bar{\boldsymbol{M}}_{\mathrm{ee}}$ is mass matrix, $\bar{\boldsymbol{D}}_{\mathrm{ee}}$ is the damping matrix, and $\bar{\boldsymbol{K}}_{\mathrm{ee}}$ is the stiffness matrix. The input matrix $\bar{\boldsymbol{H}}$ equals the linearized matrix $-\boldsymbol{M}_{\mathrm{er}}$. The linearized elastic dynamics is driven by the disturbance $\bar{\boldsymbol{d}}$ caused by the working motion. The goal of the ADC is the stabilization of the disturbed dynamics with the perturbation of the acceleration $\Delta\ddot{\boldsymbol{q}}_{\mathrm{r,d}}$, or more precisely, to increase the damping of the dominant mode of oscillation. Therefore, a modal decomposition of the model is performed and the dominant mode is isolated. The transformed equations of motion are evaluated at different operating points in order to provide a state-dependent distribution of the compensation term via a spline interpolation. For systems with multiple inputs, dominant directions of the input matrices at different working points are obtained by QR-decompositions. For the implementation of the feedback controller, it is necessary to obtain the elastic deformations of the system, e.g., with strain gauges. The output of the strain gauge y_{c} is processed by a band-pass filter in order to isolate the dominant mode of oscillation, differentiated, multiplied with a gain G and used as feedback yielding

$$m_{\mathrm{ee}}\ddot{q}_{\mathrm{dom}} + d_{\mathrm{ee}}\dot{q}_{\mathrm{dom}} + hG\dot{y}_{\mathrm{c,filt}} + k_{\mathrm{ee}}q_{\mathrm{dom}} = \bar{d}\,. \tag{4}$$

Due to the fact that the output is processed by a band-pass filter, the influence of the feedback to the other modes of oscillations is tolerable. In addition, there is no offset in the working motion.

The capabilities of the proposed ADC are demonstrated in simulation and in experiments with the help of a parallel manipulator with highly flexible links [3]. The experimental platform consists of two identical linear direct drives which are mounted on a granite plate. The positions of the sliders $s_i, i = 1, 2$ are used to describe the rigid-body motion of the system. Two revolute joints, each located on top of one slider, define the rotational axes of two links with different length. Another revolute joint connects the middle of the long link with the end of the short link, i.e. these links are arranged similar to the Greek letter λ. An additional mass, which is attached to the free end of the long link, defines the end-effector point. Hence, this parallel manipulator permits a non-redundant movement of the end effector in the horizontal plane. The currents of the direct drives, which are proportional to the forces that act on the sliders, are used as the control inputs u_i. The elastic deformations of the long link are measured with strain gauges. Due to the high stiffness of the short link, it is assumed to be rigid and no strain gauges are applied to it.

For rest-to-rest maneuvers, the desired trajectories of the slider positions $s_{\mathrm{d},i}$ and velocities $\dot{s}_{\mathrm{d},i}$ are obtained from the step response of two identical fourth-order low-pass filters, whose cutoff frequencies define the dynamics of the rest-to-rest maneuver. The desired trajectories are fed forward to two distinct P-PI-cascade controllers for the slider positions. The underlying control law, composed of a feed-forward and a feedback part, can be stated as

$$u_i(t) = u_{\mathrm{ff},i}(t) + K_{\mathrm{P,v}} e_{\mathrm{i}} + K_{\mathrm{I,v}} \int_0^t e_{\mathrm{i}} \, \mathrm{d}\tau \,, \tag{5}$$

in which e_{i} is the augmented velocity error according to

$$e_{\mathrm{i}} = \dot{s}_{\mathrm{d},i}(t) - \dot{s}_i(t) + K_{\mathrm{P,p}}(s_{\mathrm{d},i}(t) - s_i(t)) \,. \tag{6}$$

The controller parameters are set to $K_{\mathrm{P,p}} = 80\,\mathrm{s}^{-1}$, $K_{\mathrm{P,v}} = 20\,\mathrm{Cm}^{-1}$ and $K_{\mathrm{I,v}} = 400\,\mathrm{Am}^{-1}$. The feed-forward currents $u_{\mathrm{ff},i}$ in Eq. (5) are obtained from an online friction compensation. Based on this setup, the performance of the ADC is demonstrated on rest-to-rest maneuvers as well as on trajectory-tracking tasks. For trajectory-tracking tasks, it is possible to feed-forward precomputed trajectories of the motor currents as well.

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Guideway based damping control of vehicle suspensions

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Vehicle systems are usually modelled by multibody systems of different complexity depending on the engineering task. The most simple quarter car model describes already all the required basic dynamic phenomena. This model features five design parameters representing body mass, body spring, shock absorber or damper, respectively, wheel mass and tire spring. The stochastic process of the guideway unevenness can be approximated by white velocity noise and added to the vehicle's equations of motion, resulting in the state equations of the vehicle guideway system as shown, e.g., in [1].

Road vehicle suspensions are generating vertical forces in the contact patches of the tires on the road providing horizontal contact forces for vehicle propulsion and guidance. The static contact forces are complemented by the dynamic loads due to suspension vibrations excited by the guideway unevenness. In addition, the shock absorber represents a non-restricted parameter for the free proposal of the suspension designer.

From the main tasks of a road vehicle suspension system follow as assessment criteria the driving comfort and the driving safety both mathematically defined by state variables. Due to the random excitation of the vehicle, both criteria are also random variables characterized by their standard deviations or variances, respectively. The covariance analysis is introduced and explicit algebraical equations for the criteria are found depending nonlinearly on the suspension parameters.

The assessment criteria comfort and safety are with respect to the shock absorber in conflict with each other requiring a multicriteria optimization resulting in a pareto-optimal set of parameters only [2, 3]. Therefore, in some of the latest passenger cars the damping parameter is adjustable by the driver with the options COMFORT, NORMAL, SPORT or SPORT+ [4, 5, 6]. Thus, the driver is left to solve the pareto-optimal problem. This means an additional task for the driver requiring at least some knowledge on the course of the guideway ahead.

In this paper it is shown how data from a navigation system can be used to adapt the shock absorber to either maximal comfort or maximal safety depending on the course of the guideway. In particular, maximal safety on a curvaceous road results in lower comfort. Thus, decreasing comfort means a warning to the driver to reduce the speed. The methods applied in this paper are originating from multibody dynamics and they are extended by the stochastic road unevenness and the guideway curvature accessed from a navigation system using GPS data.

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Modelling a pushbelt variator

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A pushbelt continuously variable transmission (CVT) consists of about 400 steel elements guided by two loop-sets and transmitting torque from a primary to a secondary pulley due to ring-tension and element-push forces. About 1500 degrees of freedom and about 3500 contacts in a spatial model of the system combined with nonlinear effects and coupled deformations of the single parts lead to a high complexity and not fully understood dynamic effects.

In the recent years, a joint research project between the *Institute of Applied Mechanics* of the Technische Universität München and *Bosch Transmission Technology* has dealt with the detailed modelling of CVTs ([1], [2] and [3]). The model is implemented in the general framework for nonsmooth multibody systems, MBSim [4]. Besides standard elements, i.e. rigid bodies and unilateral spring-dampers, specific elements for the pushbelt CVT have been derived and implemented including e.g. a coupled force law to represent the pulley-sheave deformation or different nonlinear beam models for the representation of the loop-sets.

This work summarizes the most important findings including a detailed validation of the results comparing stationary situations in local output curves, e.g. element axial forces, element push forces or spiral running, and in global output curves, e.g. thrust ratio or transmission torques. Furthermore different modelling techniques are compared concerning physical influence (i.e. forces and kinematics), but also numerical output (e.g. smooth vs. nonsmooth dynamics).

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Modal Analysis of Vehicle Power Trains

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Modal analysis - the determination of natural frequencies and mode shapes - provides essential insight into the dynamical behavior of mechanical systems. The mechanical systems under consideration consist of rigid and flexible bodies. These bodies are interconnected by nonlinear force elements. The time-dependent description of the system is based on the floating frame of reference approach. This approach leads to the standard system of second order differential-algebraic equations as used in the multi-body simulation software AVL-EXCITE [2]. The differential equation of this system is given as:

$$\boldsymbol{M}\boldsymbol{q}'' + \boldsymbol{D}\boldsymbol{q}' + \boldsymbol{K}\boldsymbol{q} = \boldsymbol{f}^{inertia}(\boldsymbol{y}, \boldsymbol{y}', \boldsymbol{y}'') + \boldsymbol{f}^{external}(\tilde{\boldsymbol{y}}, \tilde{\boldsymbol{y}}')$$

The vector of displacement variables of a single body consists of the global translation variables $\boldsymbol{x}$, the global rotational variables $\boldsymbol{\theta}$, and the local variables $\boldsymbol{q}$. They constitute the displacement vector $\boldsymbol{y} = (\boldsymbol{x}^T, \boldsymbol{\theta}^T, \boldsymbol{q}^T)^T$ for a single body. $\boldsymbol{M}$, $\boldsymbol{D}$ and $\boldsymbol{K}$ denote the mass, damping and stiffness matrix, resulting from spatial finite element discretization. Nonlinear inertia forces - like Coriolis and gyroscopic forces and torques - are collected in $\boldsymbol{f}^{inertia}$.

External forces and moments applied at a single body are covered by $\boldsymbol{f}^{external}$. This vector includes the forces and moments resulting from externally applied loads and torques, e.g. gravity or gas pressures. It also includes the forces and moments resulting from coupling between the bodies. Hence $\boldsymbol{f}^{external}$ does not only depend on coordinates of the body itself, but also on possibly all states of other bodies, which are summarized in the vector $\tilde{\boldsymbol{y}}$. Interaction between different bodies is modeled by force-elements (joints). These force-elements are preferred, as they often model radial or axial slider bearings, where clearance gaps, wear and lubrication are of importance. Beneath non-linear spring-damper force laws, more advanced hydrodynamic models like the Reynolds equations may be used for these force laws.

The differential equation is completed by algebraic equations called reference conditons, which seperate the global motion and elastic deformations in a unique manner [2]. The rotational motion of the global frame is parameterized by the four parameter family of quaternions [4]. The normalization condition of the quaternions gives another algebraic constraint on the single body level, in addition to the reference conditions.

In order to apply modal analysis, the highly nonlinear equations of motion are linearized at specified points in time. Inertia and connections forces yield a contribution to the mass, damping, and stiffness matrix of the multi-body system, like the force equilibrium approach in [3]. The linearization is dependent on the set of coordinates selected to describe the motion of the system, especially the coordinates used to describe global rotational motion of rigid bodies.

In this contribution different sets of large orientation parameters lead to different forms of the linearized equations of motion and to different natural frequencies and eigenmodes, similar to [5]. The resulting system of linearized ordinary differential equations has constant coefficients, as soon as a common reference frame can be chosen that renders the relative motion of the bodies independent of time. A standard eigenvalue analysis such as the characteristic exponent method [1] is applied to extract natural frequencies and eigenmodes.

The approach is evaluated by computing eigenvalues and eigenmodes of a vehicle power train. The vehicle power train consists of components like cranktrain/crankshaft, transmission with internal shafts as well as engine/transmission housing supported by mounts. Some of the bodies are flexible, some are rigid, some are rotating. The resulting eigenvalue problem is of medium size. The approach is validated against ANSYS and MSC ADAMS.

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Study on Real-Time Simulation of Elastic Multibody Systems with Application in Vehicle Dynamics

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Modern vehicle development largely uses simulation technology of various vehicle states to reduce cost-intensive tests of prototype vehicles and to perform component tests prior to application in cars. Real-time simulation models are used for virtual prototyping in Driver-in-the-Loop-environments, in Hardware-in-the-Loop tests of electronic control units and in tests of mechanical components. Also in modern model-based control concepts real-time modeling might be necessary [1]. These models have to be both accurate and calculable in real-time.

The majority of the current simulation models are built as rigid body models equipped with empirical components and measurement data. However, since one of the main goals of current vehicle development is weight reduction, deformation of vehicle components might not any more be negligible and may have significant influence on vehicle behavior. Flexible components such as suspension struts or the car body itself may be modeled with the finite element (FE) method, but implementing a full FE model of a car body in a vehicle dynamics simulation will largely exceed real-time capability. One method to reduce the simulation time is to use a reduced elastic body within the multibody simulation.

This contribution presents the implementation of a reduced flexible body in a real-time vehicle simulation environment. The research code Neweul-M^2 [2] has been used to build up the symbolic formulation of suspension kinematics as well as the movement of the car body itself. Tire behavior and tire-road contact have been realized with the popular MF-Tire model [3] combined with the OpenCRG road surface model [4]. Both models have been proven to be reliable and time-efficient for a real-time simulation environment.

Simulations have been performed with the real-time simulation model, including handling tests and performance on uneven road. The results show that the combination of a vehicle model with symbolic formulation and reduced flexible bodies provide an efficient combination to cover structural elasticity in real-time simulation. Further post-processing steps of the real-time simulation may include visualization of the flexible bodies and stress calculation with stress recovery methods for reduced elastic multibody systems.

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A mechanical model of a passive dynamic walker with unilateral viscoelastic contact formulation

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Energy efficient locomotion is a demanding and important topic in lots of different research branches like for instance mechanical engineering, mathematics, computer science or medicine. The most simple model describing bipedal locomotion consists of a passive dynamic walker, a machine walking down an inclined plane with a slight slope with no external energy source other than gravity. In this contribution a mechanical multibody model of a passive dynamic walker with seven degrees of freedom is presented. It has been show in [1], that the energy efficiency of a dynamic walker can be improved by elastic couplings between its legs. This effect can also be observed in experiments. If the legs are coupled by a linear spring in a suitable fashion, the velocity of the walker increases significantly. Simulations are performed in order to confirm these results numerically using a detailed unilateral contact model. In order to describe the foot ground contact, a detailed contact model is needed, since the contact shows quite an interesting behavior. If the walker is standing on a plane the contact is in a sticking state. While walking one foot rolls over the ground until the edge of the second foot hits the ground. Since the relative velocity between the second foot and the ground usually doesn't vanish at the beginning of the impact, a silding motion occurs. This implies the necessity for a detailed contact model that is capable of describing sliding, sticking and rolling motions in a general unified way. The foot to ground contact of the walker is modelled by a bounded toroidal surface, describing the walkers foot and an inclined plane. The contact surface itself is discretized by a grid of point contacts, moving relative to the body. The kinetic contact model of this unilateral contact is capable of transmitting both sliding and sticking forces, where rolling is considered as a special form of sticking. Resistance against rolling and drilling friction are taken into account as well, in order to obtain the energy dissipation needed to attain a stable limit cycle solution. The contact formulation in normal and tangential direction is based on a viscoelastic description proposed in [2]. The ideally rigid nonholonomic constraint equations are approximated by an extension of the contact model given in [2] to viscoelastic nonholonomic motions. A theorem that proves the convergency of the solution of the viscoelastic constraint description to an ideally rigid constraint formulation is given [3]. Usually a general multibody system with nonholonomic constraint equations can be described by a differential algebraic equation in the following form

$$M(q_d)\ddot{q}_d = F(q_d, \dot{q}_d, t) - G^\mathsf{T}(q_d)\lambda_d, \tag{1}$$

$$0 = G(q_d)\dot{q}_d. \tag{2}$$

where M denotes the mass matrix, F the column matrix of gyroscopic and external forces, G the constraint matrix and λ the Lagrange multiplier. The corresponding viscoelastic description of the constraints is given by

$$M(q_v)\ddot{q}_v = F(q_v, \dot{q}_v, t) - G^\mathsf{T}(q_v)(\tfrac{c}{\varepsilon}z + \tfrac{d}{\sqrt{\varepsilon}}\dot{z}), \tag{3}$$

$$\dot{z} = G(q_v)\dot{q}_v. \tag{4}$$

with the internal variable z. The distance of the solutions can be estimated above according to:

$$\|q_d - q_v\| \le K\sqrt{\varepsilon}. \tag{5}$$

Thus the proposed contact model provides a suitable approximation of an ideally rigid nonoholonomic constraint equation, which can be used to simulate a three dimensional motion of a passive dynamic walker.

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Investigation of optimal bipedal walking gaits subject to different energy-based objective functions

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This contribution compares walking gaits of a bipedal robot model generated for different energy-based objective functions. There is a variety of optimality criteria for the analysis and generation of bipedal as well as quadrupedal gaits in the literature [1–3]. While there are some individual exceptions (e.g. maximum walking speed [3]) most of those criteria optimize either energy efficiency or stability. A combination of different criteria – based on energy and/or stability – is employed in inverse optimal control approaches trying to identify the objective criterion underlying gaits measured via motion capturing [4]. Furthermore, energy efficiency is frequently used as criterion for changes in the locomotion pattern, like transition from walking to running in bipedal locomotion [5] or transition between walking, trotting and galloping in quadruped locomotion [6].

Frequently encountered energy-based criteria are torques squared, the mechanical work or the absolute value of the mechanical work [2]. Those works are often normalized by weight and step length resulting in the cost of transport to compare different models. While some of those criteria are motivated by physical properties of the actuating motors or muscles, others, especially torques squared, are often used because of mathematical properties like convexity and global differentiability.

To investigate how different objective functions influence the generated gaits in bipedal walking, a simple planar rigid body model of a bipedal robot is simulated. The actual energy input for actuation by electric motors in all joints is derived and used as a benchmark for the energy-based criteria from the literature.

The hybrid zero dynamics (HZD) concept from [7] is used to derive a feedback controller for the robot model. The controlled model has one degree of underactuation which means it cannot transmit any torque to the ground (no flat foot). This allows for, rather than suppresses, the evolution of the model's natural dynamics. Sequential quadratic programming (SQP) is employed to generate optimal gaits for the different objective functions. The results are compared to the benchmark gait for actual energy efficiency.

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Coupling Elastic Bodies with an Enhanced Craig-Bampton-like Scheme

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Elastic multibody systems (EMBS) are widely used as description of mechanical systems that consist of separable components. These components are represented by elastic bodies that interact by the means of force and damping elements as well as boundary conditions. The floating frame of reference approach allows for large, nonlinear rigid body motion. Deformations of the elastic bodies are defined relative to the reference frame. The overall motion of the bodies consists of the superposition of the nonlinear rigid body motion and the linear elastic deformation, which delivers valid results for small deformations.

Linear elastic models are usually obtained from the linear Finite Element method. The spatial discretization leads to second order ordinary differential equations of large dimension for describing only the deformations with respect to the large reference motion,

$$\boldsymbol{M}\ddot{\boldsymbol{x}}(t) + \boldsymbol{K}\boldsymbol{x}(t) = \boldsymbol{f}(t), \tag{1}$$

with mass matrix $\boldsymbol{M}$, stiffness matrix $\boldsymbol{K}$ and force excitation $\boldsymbol{f}$. A reduction of the elastic quantities is required in order to reduce the huge numerical burden in the EMBS environment. The typical procedure relies on a projection of the nodal coordinates $\boldsymbol{x}$ onto a suited subspace $\mathcal{V} = \mathrm{colspan}(\boldsymbol{V})$ of small dimension. The Galerkin projection leads to a differential equation in terms of reduced generalized coordinates $\boldsymbol{q}_\mathrm{r}$, such that $\boldsymbol{x} \approx \boldsymbol{V}\boldsymbol{q}_\mathrm{r}$, structurally equivalent to Eq. (1).

The current standard reduction technique was introduced by Craig and Bampton [1], which bases on a splitting of the degrees of freedom into boundary and internal quantities. At boundary degrees of freedom, interactions with the environment are assumed to take place, e. g. force elements or constraints connecting the body with other bodies or the inertial system. The Craig-Bampton scheme combines static condensation [2] modes with fixed-interface eigenmodes, the solution of the eigenvalue problem with fixed interface coordinates. A major drawback of this scheme is that the eigenvalue problem, which is solved for the description of the internal deformation, does not consider any excitation of the system. However, if the boundary degrees of freedom move, the internal dynamics are indeed excited. There exist several methods, such as Krylov subspace methods and Balanced Truncation, that focus on the input-output behavior of systems, thus taking into account how the system is excited and where measurements are taken. In terms of approximation of the transfer behavior, these methods show approximation errors that are much smaller in comparison to plain modal truncation, [3]. This enables faster calculations. Also, automated variants and error estimators, respectively bounds, are available.

If the elastic body is connected to the environment, the input-output behavior also changes. In order to maintain the compatibility properties that are ensured with static condensation, a modified algorithm for the input-output based methods can be used. The fixed-interface eigenmodes in the Craig-Bampton scheme are replaced with ansatzfunctions from alternative model order reduction schemes, based on the excitation by inertial forces due to boundary movement. In this setup, arbitrary input-output based model reduction can be used and possibly time-varying boundary conditions can still be met.

The reduction scheme that is presented in this contribution is at least as reliable as and shows a faster error decay than the standard technique by Craig and Bampton.

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Back-Transformation into Physical Configuration Space after Model Order Reduction onto a General Subspace

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Model order reduction (MOR) techniques are state of the art and well established in control theory and likewise in structural mechanics. Methods that project onto a general subspace like KRYLOV-subspace-based reduction techniques are common practise, e.g. [1], [2]. In a general subspace $\mathcal{K}$ the coordinates of the reduced order model (ROM) have no physical meaning anymore, since they are not defined on the physical configuration space $\mathbb{R}$. Although being acceptable when applying a RITZ-approach, e.g. in Elastic Multi-Body-Dynamics (EMBS), any further utilization – where a physical interpretation is required – is not feasible, e.g. the coupling of single coordinates or the determination of mass and inertia properties.

To overcome this drawback, a novel *back-projection approach* was introduced by KOUTSOVASILIS [2], where the information is projected back onto the physical configuration space. By applying a linear and orthogonal projection onto a KRYLOV-subspace, the projection matrix $\mathbf{V}^{\mathcal{K}} \in \mathcal{K}^{N\times n}$ is gained. For a physical meaning, the system is partitioned into *master* and *slave* coordinates. The back-projection approach requires the inversion of the master-partition of the projection matrix $\mathbf{V}^{\mathcal{K}}_{mm}$, which finally yields to the back-projection matrix $\mathbf{V}^{\mathcal{K}\to\mathbb{R}}$. The reduced system matrices in physical configuration space, e.g. $\overline{\mathbf{M}}^{\mathbb{R}} \in \mathbb{R}^{n\times n}$, are derived.

$$\mathbf{x}^{\mathbb{R}}_m = \underbrace{\mathbf{V}^{\mathcal{K}}_{mm}}_{\mathbf{V}^{\mathbb{R}\to\mathcal{K}}} \bar{\mathbf{x}}^{\mathcal{K}}, \quad \mathbf{V}^{\mathcal{K}\to\mathbb{R}} = \left(\mathbf{V}^{\mathcal{K}}_{mm}\right)^{-1}, \quad \overline{\mathbf{M}}^{\mathbb{R}} = \left(\mathbf{V}^{\mathcal{K}\to\mathbb{R}}\right)^T \underbrace{\left(\mathbf{V}^{\mathcal{K}}\right)^T \mathbf{M}^{\mathbb{R}} \mathbf{V}^{\mathcal{K}}}_{\overline{\mathbf{M}}^{\mathcal{K}} \in \mathcal{K}^{n\times n}} \mathbf{V}^{\mathcal{K}\to\mathbb{R}} \quad \text{with} \quad \mathbf{M}^{\mathbb{R}} \in \mathbb{R}^{N\times N}$$

Due to the dimension of $\mathbf{V}^{\mathcal{K}}_{mm} \in \mathcal{K}^{n\times n}$ we call this approach *back-transformation* deviating from KOUTSOVASILIS [2]. By the back-transformation, the eigenvalues of the system are not affected. Unfortunately, the inversion works not sufficiently for an arbitrary reduced model, since the submatrix $\mathbf{V}^{\mathcal{K}}_{mm}$ may be ill-conditioned, see [2]. This is mainly caused due to the spatial distribution of interface coordinates, which is often unfavourable in mechanical models because of cluttered interfaces.

In the novel approach, the inversion is stabilized by introducing a *rank* criterion. If the product $\mathbf{V}^T_{mm}\mathbf{V}_{mm}$ has full rank n, the evaluation of the pseudo-inverse given by $\mathbf{V}^+_{mm} = \left(\mathbf{V}^T_{mm}\mathbf{V}_{mm}\right)^{-1}\mathbf{V}_{mm}$ is always possible. The rank increase is achieved by selecting *additional master* coordinates, by the help of sensor placement methods, e.g. the Effective Independence (EfI) [3] or the MoGeSeC-Algorithm [4].

The novel *back-transformation approach* is demonstrated at the example of a wheel-set axle FE-model and the complex model of a gear box housing. Due to the additional coordinates, the ROM dimension slightly increases, but with the benefit that the ROM is defined in physical configuration space. The quality of the recent ROM is verified by using correlation methods and the computational effort is evaluated.

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Topology Optimization of Flexible Bodies in Multibody Systems using the Floating Frame of Reference Approach

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Energy efficiency is a permanent issue in modern design of active multibody systems. In order to reduce the energy consumption, lightweight design techniques are applied to lower the moving masses and improve the mass to payload ratio. However, as a result the stiffness of the bodies decreases and nonnegligible structural deformations might appear during the working motion. For example in lightweight manipulators structural flexibility might lead to large unacceptable end-effector tracking errors. To decrease the structural deformation one might either use modern nonlinear control approaches or one might use optimization methods to design the members of the flexible system appropriately. In this research the second approach is taken, since it allows the use of well established basic control methods which are rather simple to implement. In order to obtain an optimized structural design, simulations of flexible multibody systems using the floating frame of reference approach are coupled with topology optimization for flexible members of the multibody system to improve their tracking behavior.

Topology optimization is a powerful tool for designing lightweight structures. This method tries to find the best distribution of material in a fixed design space. Therewith, this method allows for any formation of material inside the specified domain. The optimization problem is formulated using the solid isotropic material with penalization (SIMP) approach, see [2]. Such optimizations are often performed using finite element models. Thereby the objective functions are usually defined with respect to the compliance, displacements or stresses of the structure obtained from a set of static load cases. However, in order to improve system performance of dynamical systems the actual dynamic loads must be taken into account in the design of the system's components. First promising results of such an optimization philosophy are discussed in [1], [3], [4] for various types of active flexible multibody systems. However, many open questions remain, especially concerning, parameterizations, modeling and computational choices during such optimization procedures.

In this research the objective functions are computed from fully dynamical simulations of a flexible multibody system. Thus, all relevant loads on the flexible members of the system are captured. Consequently, all steps including the finite element modeling, the model reduction, the derivation of the equations of motion, establishing of feedback control and the transient simulation must be performed in each optimization loop. Thereby in each step a variety of possibilities exists. This talk will address some of these possibilities. Examples are the choice of the SIMP parameterizations, choice of global shape functions to approximate the elastic deformation field, computation of the gradients and type of objective functions.

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Nonexpansivity of the Newton's Cradle Impact Law

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In this paper, we present an impact law for Newton's Cradle with 3 balls which is kinematically and kinetically consistent *and* enjoys the maximal monotonicity property. Our aim is to divulge the structure of impact laws in order to be able to formulate maximal monotone impact laws for rigid multi-body systems that do not have the problems of existing impact laws such as kinematic, kinetic, and energetic inconsistency [1]. It is interesting to consider Newton's Cradle because its phenomena cannot be described by the classical Newton's or Poisson's impact laws.

Kinematic consistency means that the impenetrability of unilateral constraints requires the post-impact contact velocities γ_i^+ to be non-negative. In view of numerical integration, an impact-law should guarantee that arbitrary (also kinematically inconsistent) pre-impact contact velocities are mapped to kinematically consistent post-impact contact velocities.

The interest in the maximal monotonicity property stems from stability analysis and control of mechanical systems with unilateral constraints [3]. Since the maximal monotonicity property implies dissipativity it seems to be a physically reasonable property of an impact law.

We consider Newton's Cradle that consists of three balls of equal mass m that are aligned along the same axis. The velocities of the balls are given by $\mathbf{u} = (u_1 \;\; u_2 \;\; u_3)^{\mathrm{T}}$. The contact velocities are given by the relative velocities between the balls $\boldsymbol{\gamma} = (\gamma_1 \;\; \gamma_2) = (u_2 - u_1 \;\; u_3 - u_2)^{\mathrm{T}}$. The pre- and post-impact velocities are designated by $\mathbf{u}^-$ and $\mathbf{u}^+$ respectively. Analogously, $\boldsymbol{\gamma}^-$ and $\boldsymbol{\gamma}^+$ designate the pre- and post-impact relative velocities.

In this work, we analyze the classical outcomes of the 3-ball Newton's cradle and complement these with thought-experiments for kinematic inconsistent cases. Six different pre-impact configurations can be distinguished of which the outcomes are described by a *piece-wise linear* relation for the impact mapping S

$$\boldsymbol{\gamma}^+ = S(\boldsymbol{\gamma}^-) = \mathbf{Q}_i \boldsymbol{\gamma}^- \quad \text{for} \quad \mathbf{Q}_i \boldsymbol{\gamma}^- \geq 0 \quad i = 1, \ldots, 6 \tag{1}$$

that maps the pre-impact relative velocities to the post-impact relative velocities.

A major result of the paper is that we can rigorously prove that the derived mapping (1) is non-expansive in the metric $\mathbf{G}^{-1}$, i.e.

$$\|\boldsymbol{\gamma}_1^+ - \boldsymbol{\gamma}_2^+\|_{\mathbf{G}^{-1}} \leq \|\boldsymbol{\gamma}_1^- - \boldsymbol{\gamma}_2^-\|_{\mathbf{G}^{-1}}, \tag{2}$$

where $\mathbf{G} = \mathbf{W}^{\mathrm{T}}\mathbf{M}^{-1}\mathbf{W}$ is the Delassus matrix. Therefore, it follows from [2] that the corresponding impact law is maximal monotone. Therefore, the impact mapping (1) that we propose guarantees kinematic, kinetic, and energetic consistency. The corresponding impact law has the maximal monotonicity property and it may allow to reveal further details about the general structure of set-valued impact laws.

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An annular Kirchhoff plate model tailored for rotating and non-rotating external loads

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The bending deformation of rotating annular plates and the associated vibration behaviour is important in engineering applications which range from automotive or railway brake systems to discs that form essential components in turbomachinery.

However the modeling of rotating discs often has to cope with the difficulty to describe non-rotating forces acting on the disc such as the normal and friction forces at a disc brake.

In the floating frame of reference approach commonly used in flexible multibody dynamics the deformation field is formulated in terms of material fixed Lagrangian coordinates. As a consequence it is easy to introduce external forces that rotate synchronically with the disc, since here the Lagrangian coordinate of the force attachment point is constant in time. On the contrary for non-rotating forces the Lagrangian coordinate of the force attachment point has to be evaluated at each considered point in time which requires a contact formulation.

Due to the rotational symmetry properties of an annular plate it is possible to specifically adopt the so-called Arbitrary Langrangian-Eulerian (ALE) representation [1] and formulate the bending deformation of a Kirchhoff plate in an elegant way. An additional sliding frame of reference is introduced that does not perform the rotating motion. Since the transformation between the floating and the sliding frame of reference can be given, both rotating and non-rotating external forces can conveniently introduced into the equations of motion.

The contribution will give an overview on the theoretical background of this approach that has been implemented in the object-oriented modeling language Modelica [2]. The discussion of some simulation examples will conclude the presentation.

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Performance assessment of a tracjectory-tracking approach for a manipulator with uncertainties using inverse fuzzy arithmetic

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Uncertainty analysis is concerned with the effects of model uncertainties on the prediction of the model response. Thereby, for instance, the reliability of a model and the conclusions drawn from the calculated results may be rated or be put into perspective with the associated real system. Furthermore, the behavior and the change of behavior of the system for the possible conditions may be investigated, providing a better understanding of the system.

On the one hand, this is possible in a direct approach by assigning uncertainties to the model – derived from assumptions, expert knowledge or identification procedures – and propagating those uncertain quantities through the model in order to obtain the range of responses for the feasible realizations of the model. On the other hand, an inverse approach can be applied, being capable of identifying uncertainties in a model in order to provide uncertainty bounds for a parameter estimation and to perform a model validation on that basis [1].

The uncertainties may stem from variability in the model parameters, i.e. from randomness. There are various methodologies for handling uncertainties of that type in the different analyses described above. Uncertainties due to a lack of knowledge or due to simplifications made during modelling are commonly termed epistemic uncertainties and are not characterized by randomness. Instead, the actually deterministic real system and the model differ due to the imperfect knowledge and the effects neglected during modeling. Therefore, fuzzy numbers are commonly used to model those epistemic uncertainties, since they comply with the possibilistic nature of this class of uncertainties. Hence, fuzzy arithmetical methods have to be employed for the analysis of systems with epistemic uncertainties [2].

In this contribution, the fuzzy arithmetical approach is applied to an inverse problem in multibody dynamics in order to assess the system performance considering intrinsic uncertainties. A major issue during the design phase is indeed characterized by the imperfect knowledge of the model parameters and the actual operational configuration. For instance, the controller design is performed for a nominal model that is assumed to resemble most of the system dynamics.

In particular, a trajectory tracking problem of a manipulator with highly flexible links will be addressed that possesses unstable internal dynamics. A feed-forward control based on stable inversion is applied which provides an exact tracking of the end-effector motion for the nominal system without disturbances. As usual, in order to account for the inherent uncertainties and disturbances, the system is supplemented by an additional feedback control. Since the feed-forward control provides a trajectory planning with desired drive positions and velocities, a collocated feedback can be employed for the drives [3]. Hence, stability of the control scheme can be achieved despite the non-minimun phase characteristic of the system.

The effect of the model uncertainties are then to be identified, as the control scheme only affects the drive positions and thus does not unconditionally ensure that the end effector tracks the desired trajectory with satisfactory accuracy, especially in the case of significant elastic deformation. Hence, the performance of the system shall be assessed by investigating the robustness of the tracking approach with respect to inexact matching of initial conditions and uncertain model parameters. Therefore, a concept for estimating the robustness of a system with respect to epistemic uncertainties within a model is presented using inverse fuzzy arithmetic.

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Variational integrators for thermo-viscoelastic discrete systems

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Variational integrators are modern time-integration schemes based on a discretization of the underlying variational principle. They thus skip the direct formulation and time discretization of differential equations. In this paper, Hamilton's principle is approximated by an action sum, whose vanishing variation results in discrete Euler-Lagrange equations or, equivalently, in discrete evolution equations for the position and momentum. Variational integrators are, by design, structure preserving (symplecticity and momentum) and show excellent long-time behavior in total energy.

In this work, heat transfer is accounted for after the dissipationless type II model of Green and Naghdi. We will show that, in this way, thermal effects enter the variational principle both via the free energy, and Fourier's law of heat conduction accounting for thermal dissipation. This formulation requires the notion of thermacy, a quantity also called "thermal displacement" whose time derivative corresponds to the temperature. It results in a natural definition of the entropy as "thermal momentum" and entropy flux as "thermal force". The viscoelastic effects are also introduced via the free energy by an internal variable formulation. In order to include the corresponding viscous and thermal virtual work (mechanical and thermal virtual dissipation), Hamilton's principle is extended by D'Alembert terms, which account for the time evolution equation of the internal variable and Fourier's law.

From this variational formulation, variational integrators using different orders of approximation of the state variables as well as of the quadrature of the action integral are constructed and compared. A thermo-viscoelastic double pendulum comprised of two discrete masses connected by thermoelastic springs and dash pots (Poynting-elements), and subject to heat conduction both between the two springs and between each of the springs and the environment, serves as a discrete model problem.

Variational integrators of higher order for flexible multibody systems

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Variational integrators are structure-preserving time stepping schemes for dynamical systems which are derived from a discrete variational principle. This is in contrast to the usual discretization of Euler-Lagrange equations derived from a continuous variational principle. Therefore, a variational integrator always takes the form of discrete Euler-Lagrange equations or the equivalent position-momentum equations.

In this presentation, we consider the motion of a flexible rope subject to different boundary conditions as model problem for a distributed parameter system. One considered boundary condition is to consider the rope as flexible connection between two mass points, which represent a physical model of a tethered satellite system. Furthermore, we simulate the motion of the rope subject to time-dependet Dirichlet boundary conditions, as a fixed or moved suspension point, for instance. The introduction of non-holonomic constraints by the Lagrange multiplier technique makes this possible.

The corresponding variational integrators are derived from a space-time discretization of Hamilton's principle. The space discretization is based on one-dimensional linear Lagrange polynomials and a two-point Gaussian quadrature, whereas the time discretization is based on higher-order polynomials and higher-order quadrature rules. In this way, we obtain higher-order accurate variational integrators.

As numerical examples, we consider the flexible rope system subject to different combinations of Dirichlet and Neumann boundary conditions, and investigate the convergence behaviour and computional costs of the variational integrator in comparison to standard time stepping schemes. Further, we also compare different polynomial bases with equidistant or other special interpolation points in the time approximation.

Error estimation approach for controlling the comunication step size for semi-implicit co-simulation methods

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This contribution presents an approach for controlling the macro-step size in connection with co-simulation methods [1, 3]. The investigated step-size controller is tailored for semi-implicit co-simulation techniques. Concretely, we consider predictor/corrector co-simulation approaches [2]. By comparing variables from the predictor and the corrector step, an error estimator for the local error can be constructed. Making use of the estimated local error, a step-size controller for the macro-step size can be implemented. Different numerical examples are presented, which show on the one hand the applicability of the method and on the other hand the benefit of a variable macro-grid with respect to simulation time.

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A Reproducible Excitation Mechanism for Analysing Electric Guitars

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The history of musical instruments dates back nearly as long as the humanity itself. During the last centuries, the development of the quality as well as of the play of musical instruments was pushed empirically and experimentally by instrument makers and players. This development can hardly be surpassed. Therefore, at first sight, the scientific study of instruments does not seem to allow any further significant improvement. It is, however, very important to gain insight into these complex systems and to achieve an understanding of the generated sound and of what makes it agreeable for the listener.

A very popular instrument is the electric guitar, whose sound is commonly assumed to originate mainly from the plugged electronics. This assumption, however, is not compatible to the large variety of commercially available models of electric guitars, featuring a significant spread in material and construction, and also in quality and price. In fact, there is reason to believe that there is a not to be underestimated influence of the used materials, which is also motivated by investigations of acoustic guitars. However, the sound of acoustic guitars is mainly formed in there hollow body [1], strongly influenced by the structural behavior of the body and the neck. The body of an electrical guitar, instead, is made of solid wood, and a sound hole does not exist.

The goal of this investigation is to gain insight into the sound generation of electric guitars, the interaction between string excitation and structural response as well as the influence of material and geometry parameters. This contribution presents an experimental set-up for measuring the structural behavior of the guitar body using Laser-Doppler-Vibrometers, to be compared to the generated sound. To ensure a well-defined and reproducible excitation of the strings, a sophisticated mechanism is constructed, making use of the methods of multibody dynamics and optimization. With this mechanism, it is possible to determine the influence of different parameters on the induced vibrations and the generated sound, which can be assessed by specific criteria [2]. Exemplarily, the influence of the adjustable settings of the electronics and the pickups are presented.

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Transient amplification of maximum vibration amplitudes

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Many low damped structures as turbine blades or drill strings are exposed to high dynamical loads causing high vibration amplitudes. These applications comprise sub-critical eigenfrequencies. Hereby, the lower eigenfrequencies have to be passed before reaching the operating point. Most investigations of vibration amplitudes caused by a resonance passage deal with the computation of one degree of freedom systems. Thereby, it has been shown that the stationary vibration response provides the highest possible amplitude. Further it can be stated that the maximum vibration response of the resonance passage decreases with an increasing sweep velocity, see [1].

To describe linear systems with isolated modes it is straight forward to investigate only one-degree-of-freedom systems. Since the system can be decoupled it is sufficient to regard the separated modes individually. Subsequently the mode shape can be described by the multiplication of the amplification function of the mode and the belonging eigenvector. There are only some recent works that deal with resonance passages of vicinal modes, e. g. [2] and [3]. But no revised statements are provided concerning the behavior of the maximum amplitudes. In this paper a three dimensional system comprising nearby modes is studied to investigate the amplitude of a resonance passage. To calculate the transient vibration response an analytical approach is used. It will be shown that the maximum amplitude of the stationary vibration response is not the least upper bound for the maximum amplitude of the resonance passage and that the maximum amplitude may rise while the sweep velocity increases. Hence, regarding a multi degree of freedom system the maximum amplitude of the resonance passage can exaggerate the maximum amplitude of the stationary vibration response. This is statement extends the known statements about transient vibrations.

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Prox formulation of the cavitation problem in elastohydrodynamic lubrication contact

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This contribution presents an alternative method, how to find the cavitation region in elastohydrodynamic (EHD) lubrication using an augmented lagragian approach. A theoretical framework for the use of the prox formulation instead of a Linear Complementary Problem (LCP) is given and the application for multibody systems with EHD contacts is shown. The prox formulation is solved by the Newton-Raphson method. As an example, the number of iteration for the solution using the prox formulation is compared with the number obtained by the pivot based Murty algorithm during time integration of a rotor with unbalance in an elastic bearing.

In EHD lubrication theory, the pressure distribution in the fluid film between two contacting rigid or elastic bodies is usually calculated by a numerical solution of the Reynolds equation. In zones with negative pressure, the fluid is said to cavitate and therefore, the negative pressure is set to the cavitation pressure. In order to ensure mass conservation at the boundary between fluid zone and cavitation zone, the pressure and the pressure gradient have to vanish, which is known as the Reynolds boundary condition. The so called cavitation problem can be formulated as a LCP. The aim is to find a fluid region with positive pressure p and vanishing flux q and a cavitation region with a zero cavitation pressure and a negative flux:

$$\begin{aligned} b - A \cdot p &= q \\ 0 \le p \perp q &\ge 0 \end{aligned} \tag{1}$$

A widely spread cavitation algorithm for this problem is the pivot based Murty algorithm, which was applied first by Goenka 1985 in a blockwise form [1].

Here, a prox formulation of the cavitation problem is presented as an competitive alternative. The prox formulation was first stated in the beginnings of the 1990s by Alart and Curnier for frictional contact problems [2]. In [3] the prox formulation and the well known LCP are compared for simulation of multibody systems with contact. Hence, for the treatment of EHD contacts in multibody systems, a prox formulation of the cavitation problem can be a useful alternative.

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Rigid body motion in a medium: data preparation for execution of experiments

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Proposed work presents next stage of the study of the problem of the plane-parallel motion of a rigid body interacting with a resistant medium through the frontal plane part of its external surface. Under constructing of the force acting of medium, we use the information on the properties of medium streamline flow around in quasi-stationarity conditions (for instance, on the homogeneous circular cylinder input into the water). The medium motion is not studied, and we consider such problem in which the characteristic time of the body motion with respect to its center of masses is comparable with the characteristic time of motion of the center of masses itself.

If in [1], we represent the asymptotical stability conditions of the rectilinear translational deceleration (drag), and in [2, 3], we obtained the new multi-parametric family of phase patterns in the space of quasi-velocities, then in this work, we prepare the qualitative material for the preparation of further natural experiments on the motion of the hollow circular cylinders in a medium.

Let give the brief summary to the previous stages of studying. Also, by the reason of complexity of nonlinear analysis, the initial stage of such a study is the neglecting of the dependence of the medium interaction force moment on the angular velocity and use of such dependence on the angle of attack only [1]).

From the practical view point it is important the problem of studying of stability of so-called unperturbed (rectilinear translational) motion under which the velocities of body points are perpendicular to the plate (cavitator).

The whole spectrum of results found under the simplest assumption on the absence of the medium damping action on a rigid body allowed the author to make the conclusion that it is impossible to find those conditions under which there exist the solutions corresponding to the angular body oscillations of a finite amplitude.

The experiment in the motion of homogeneous circular cylinders in the water (see [2]) justified that in modelling the medium action on the rigid body, it is also necessary to take account of an dependence of the medium interaction force moment on the angular velocity of the body. Herewith, there arise the additional members that brings a dissipation to the system.

In studying the class of body motions with the finite angles of attack, the principal problem is finding those conditions under which there exist the finite amplitude oscillations in a neighborhood of the unperturbed motion. Therefore, there arises the necessity of a complete nonlinear study.

In earlier author's works, one has succeeded to use the instability of the rectilinear translational body motion for the methodological purposes (see [3]), i. e., in determination of unknown parameters of the medium action on the body in quasi-stationarity conditions.

The account of the medium damping action on the rigid body leads to an affirmative answer to the principal question of the nonlinear analysis: under the body motion in a medium with finite angles of attack, in principle, there can arise stable auto-oscillations which can be explained by the account of an additional dependence of the medium action on the body angular velocity that brings an additional dissipation to the system.

Furthermore, under the applying of methodology of studying of the dissipative dynamical systems of certain type, we obtain the new multi-parametric family of phase patterns on the two-dimesional cylinder; this family consists of the infinite set of topologically non-equivalent phase patterns changing its topological types under the variation of the system parameters by the degenerate way (see also [1, 2, 3]).

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