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Simulation Techniques for Multidisciplinary Problems in Vehicle System Dynamics

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SUMMARY

Simulation in vehicle system dynamics has its historical origin in the analysis of the purely mechanical behaviour using mechanical multibody system models. In multibody dynamics very efficient numerical methods for the evaluation and for the time integration of the equations of motion are available. These methods have been extended step-by-step to more complex engineering systems that may contain, e.g., flexible bodies and mechatronic or adaptronic devices. Multidisciplinary problems like the interaction of mechanical and hydraulic components or the interaction of vehicle dynamics and aerodynamics are handled conveniently by co-simulation techniques. The present paper summarizes some of these recent extensions of classical multibody dynamics such as multifield problems in the simulation of adaptronic devices, advanced models of contact mechanics and coupled problems including multibody dynamics, aerodynamics and structural mechanics.

Keywords: multidisciplinary problems, mechatronics, coupled systems, dynamical simulation, multibody dynamics.

1 INTRODUCTION

The increasing integration of mechanical, hydraulic and electronic components in engineering systems is accompanied by a close integration of the industrial design processes. There is a need for industrial simulation tools for such *mechatronic* systems, or more generally for multidisciplinary problems in vehicle system dynamics. These tools and simulation techniques are strongly influenced by the well established ideas, methods and software of multibody dynamics [1].

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The classical topic of interest in multibody dynamics are systems of rigid bodies being connected by joints and force elements like springs and dampers [2]. The equations of motion are given by

$$M(q)\,\ddot{q}(t) = f(t, q, \dot{q}, \lambda) - G^T(t, q)\,\lambda\,, \tag{1a}$$

$$0 = g(t, q) \tag{1b}$$

with q denoting the position coordinates of all bodies. $M(q)$ is the generalized mass matrix and f the vector of applied forces. Joints decrease the number of degrees of freedom in the system and may result in constraints (1b) that are coupled to the dynamical equations (1a) by constraint forces $-G^T\lambda$ with Lagrange multipliers λ and $G(t, q) := (\partial g/\partial q)(t, q)$. Very efficient numerical methods for the evaluation and for the time integration of (1) have been developed and implemented in industrial multibody simulation tools like ADAMS, SIMPACK or DADS, see [1, 3, 4].

Already in the early days of multibody dynamics these methods have been extended to more general mechanical systems that contain e.g. flexible bodies or force elements with internal dynamics [5]. Sophisticated modal reduction techniques are used to consider the elastic deformation of flexible bodies, see Section 2.1.

More complex multidisciplinary problems arise from the coupling of mechanical, hydraulic and electronic components in mechatronic devices. Formally, these systems are beyond the area of application of (flexible) multibody system simulation packages, but hydraulic and electric system components may be added straightforwardly as force elements with internal state variables as long as they are described by continuous-time differential equations. Recently, the methods of classical multibody dynamics have been extended to multifield problems in the simulation of adaptronic system components, see Section 2.2. Furthermore, advanced contact models have been developed that allow the efficient simulation of contact problems in a multibody system framework, see Section 2.3.

On the other hand discrete time components like digital controllers may slow down the simulation drastically since all numerical methods of classical multibody dynamics are tailored to equations of motion that are continuous in time. In the simulation of systems with discrete controllers the equations of motion (1) have to be extended to a mixed system with the continuous part

$$\begin{aligned} M(q)\,\ddot{q}(t) &= f(t, q, \dot{q}, \lambda, c, r_n) - G^T(t, q, c, r_n)\,\lambda\,, \\ \dot{c}(t) &= d(t, q, \dot{q}, \lambda, c, r_n)\,, \\ 0 &= g(t, q, c, r_n) \end{aligned} \tag{2a}$$

for $t \in [T_n, T_{n+1}]$ and discrete state changes at $t = T_{n+1}$:

$$r_{n+1} = k(T_{n+1}, q(T_{n+1}), \dot{q}(T_{n+1}), \lambda(T_{n+1}), c(T_{n+1}), r_n, r_{n-1}, \ldots, r_{n-l}) . \quad (2b)$$

The continuous state variables $c(t)$and the discrete state variables r_n stand for internal states of force elements as well as for the state variables of time continuous/time discrete mechatronic devices.

Mixed time continuous/time discrete systems are well known from other areas of technical simulation such as circuit simulation or chemical engineering. In the numerical solution of (2) classical time integration methods for continuous systems (1) are combined with updates of the discrete state variables r_n at the sampling points $t = T_n$ [6]. This approach has been implemented in the industrial simulation package SIMPACK [4, 7]. The solver for the continuous part has to be adapted to handle the frequent discontinuities efficiently [4, 8].

As a by-product this algorithm may be extended to a co-simulation interface for the simulation of multidisciplinary problems including multibody dynamics, see Section 3. A typical example is the consideration of aerodynamic effects in vehicle simulation by coupling a computational fluid dynamics tool with a multibody system tool, see Section 3.2.

Applied to the simulation of the dynamical interaction between vehicles and large elastic structures, co-simulation proved to be substantially more convenient and more efficient than classical techniques from flexible multibody dynamics [9]. In Section 3.3 simulation results are presented for two trains passing each other while crossing a bridge.

2 EXTENSIONS OF CLASSICAL MULTIBODY SIMULATION TOOLS

Multibody dynamics has its origin in the analysis of systems of rigid bodies. Even for complex multibody system models the simulation time on PC hardware is often close to real time. Therefore not only simulations but also parameter variations and the optimization of system parameters may be performed in reasonable computing time.

Special modeling techniques have been developed to achieve a similar numerical efficiency also for more complicated engineering systems including even multidisciplinary effects. In this section we study the (global) elastic deformation of bodies in a flexible multibody system (Section 2.1), the coupling of mechanical and electric fields in the simulation of adaptive structures (Section 2.2) and advanced models for the efficient consideration of local effects in the contact area between two bodies that come into contact (Section 2.3).

2.1 *Flexible multibody systems*

In mechanical systems with lightweight components the elastic deformation of these bodies may not be neglected. It is supposed that the elastic deformation is small w. r. t. a *moving frame of reference* that follows the gross motion of such a *flexible* body. Then it is possible to describe the deformation w. r. t. this frame of reference by linear theory of elasticity.

Typically the elastic effects are considered only in the frequency range up to 50 … 100 Hz such that a modal approach is very attractive [10]. Then the displacement field may be expressed as

$$u = \Phi\, q_e \tag{3}$$

with the modes Φ and position coordinates $q_e(t)$ that represent the elastic deformation w. r. t. the moving frame of reference. Eigenmodes or more sophisticated mode shapes [11] are obtained from a preceding finite element analysis of the flexible body.

The equations of motion of *flexible* multibody systems, i.e. systems with at least one flexible body, are obtained similarly to (1) from the principles of classical mechanics [10]. They get the form

$$\begin{pmatrix} M_r & M_{re} \\ M_{er} & M_e \end{pmatrix} \begin{pmatrix} \ddot{q}_r \\ \ddot{q}_e \end{pmatrix} = f(t, q, \dot{q}, \lambda) - G^T(t, q)\, \lambda\,, \tag{4a}$$

$$0 = g(t, q) \tag{4b}$$

with a vector $q := (q_r, q_e)^T$ of position coordinates that describes the positions q_r of the rigid bodies and the moving frames of reference and the elastic coordinates q_e from (3). The elastic deformation and the motion of the frame of reference are coupled by the right hand side of (4a) and by the off-diagonal blocks M_{re}, M_{er} of the mass matrix.

Today the simulation of flexible multibody systems is a standard technique that is as efficient and robust as the simulation of classical rigid body systems [10].

2.2 *Multibody systems with active components*

Adaptive or *smart* structures are mechatronic devices, which allow modification of the properties and the response of elastic structures exposed to varying stimuli and environmental conditions. Particularly the compensation of the structural instability of lightweight vehicle components is one key field of application.

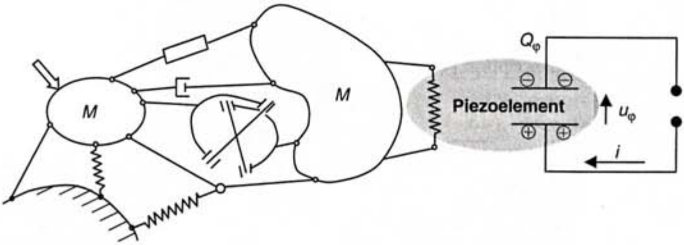

Fig. 1. Multibody systems with active components: Piezoelements.

The most promising way to achieve this purpose requires the integration of actuators and sensors as a constitutional part of a flexible component. Such build-in devices provide the base for the application of adequate signal processing and controlling techniques.

There is a wide range of supposable physical effects and corresponding material compositions. *Piezoceramics* proved to be an appropriate class of materials to reduce structural excitations actively. Therefore they may be used to enlarge the range of application of lightweight components.

Three main characteristics can be attributed to piezoelectric transducters: a) piezoelectric materials generate an electric field when subjected to mechanical strain (sensor effect); b) if an electric field is applied to them a deformation results (actuator effect); c) as a distinct feature piezoelectric actuators and sensors may be used as distributed devices.

These properties reveal that a multidisciplinary approach makes sense, which combines the description of the elastic displacement field and the electrostatic field.

The formulation is based on a constitutive equation, which describes the linearised relationship between the mechanical variables strain S and stress T and the electrical terms displacement D and field strength E by defining appropriate material constants [12]:

$$\begin{pmatrix} T \\ D \end{pmatrix} = \begin{pmatrix} c & -e^T \\ e & \varepsilon \end{pmatrix} \begin{pmatrix} S \\ E \end{pmatrix}. \tag{5}$$

The field equations are developed by means of Jourdain's principle of virtual power. Therefore the right hand sides of Eqs. (6) represent the integrals over all applied outer force and charge loads acting on volumes or boundaries. The variables v and a stand for the absolute velocity and acceleration respectively. The terms Q_φ and φ denote the applied charges and their electric potential. Furthermore (5) is used to eliminate the dependent variables T and D pointing out the coupling of mechanical and electric field by material description [13]:

$$\int \delta \boldsymbol{v}^T \varrho \boldsymbol{a} + \delta \dot{\boldsymbol{S}}^T \underbrace{(\boldsymbol{c}\boldsymbol{S} - \boldsymbol{e}^T \boldsymbol{E})}_{\boldsymbol{T}} \,\mathrm{d}V = \int \delta \boldsymbol{v}^T \boldsymbol{f}_V \,\mathrm{d}V + \oint \delta \boldsymbol{v}^T \boldsymbol{f}_B \,\mathrm{d}B\,, \quad \text{(6a)}$$

$$\int \delta \dot{\boldsymbol{E}}^T \underbrace{(\boldsymbol{e}\boldsymbol{S} + \varepsilon \boldsymbol{E})}_{\boldsymbol{D}} \,\mathrm{d}V = \oint \delta \dot{\varphi} Q_\varphi \,\mathrm{d}B\,. \quad \text{(6b)}$$

The resulting partial differential equation is discretised in space by a modal approach with a small number of eigenfunctions to impose high numerical efficiency. Therefore modes Φ have to be selected to describe the spatial distribution of the displacement field. The Green strain tensor S is now capable for calculation by means of the differential displacement-strain operator $\mathcal{L}$, see (7a).

The scalar electrical potential field φ is handled in analogous manner by the selection of appropriate mode shapes Φ_φ. The negative gradient operation then yields the electric field vector E. The discrete, only time dependent electric variable u_φ can be interpreted physically as the electric voltage applied to the electrodes of the piezoceramic devices [14]:

$$\boldsymbol{u} = \boldsymbol{\Phi} \boldsymbol{q}_e\,, \qquad \boldsymbol{S} = (\mathcal{L}\boldsymbol{\Phi})\, \boldsymbol{q}_e = \boldsymbol{B}\boldsymbol{q}_e\,, \quad \text{(7a)}$$

$$\varphi = \boldsymbol{\Phi}_\varphi \boldsymbol{u}_\varphi\,, \qquad \boldsymbol{E} = (-\nabla \boldsymbol{\Phi}_\varphi)\, \boldsymbol{u}_\varphi = \boldsymbol{B}_\varphi \boldsymbol{u}_\varphi\,. \quad \text{(7b)}$$

In discretised form Eqs. (6) are formulated using a 2×2 block matrix K. The electromechanical coupling matrix $K_{q\varphi}$ and the electric capacity matrix $K_{\varphi\varphi}$ arise out of the evaluation of the following only volume dependent integrals, a methodology which is already well known from the definition of the mechanical stiffness matrix K_{qq} [15]:

$$\boldsymbol{K}_{qq} = \int \boldsymbol{B}^T \boldsymbol{c} \boldsymbol{B} \,\mathrm{d}V\,, \quad \boldsymbol{K}_{\varphi\varphi} = \int \boldsymbol{B}_\varphi^T \varepsilon \boldsymbol{B}_\varphi \,\mathrm{d}V\,, \quad \boldsymbol{K}_{q\varphi} = \int \boldsymbol{B}^T \boldsymbol{e}^T \boldsymbol{B}_\varphi \,\mathrm{d}V\,.$$

The classical equations of motion have to be extended by adding the product of the applied voltage vector u_φ with the coupling matrix $K_{q\varphi}$ clearly demonstrating the use of the piezoceramic as a structural actuator:

$$\begin{pmatrix} \boldsymbol{M}_r & \boldsymbol{M}_{re} \\ \boldsymbol{M}_{er} & \boldsymbol{M}_e \end{pmatrix} \begin{pmatrix} \ddot{\boldsymbol{q}}_r \\ \ddot{\boldsymbol{q}}_e \end{pmatrix} = \begin{pmatrix} \boldsymbol{h}_r \\ \boldsymbol{h}_e \end{pmatrix} - \begin{pmatrix} 0 \\ \boldsymbol{K}_{q\varphi} \boldsymbol{u}_\varphi \end{pmatrix}, \quad \text{(8a)}$$

$$Q_\varphi = K_{q\varphi}^T q_c + K_{\varphi\varphi} u_\varphi \,. \tag{8b}$$

The sensor equation (8b) is needed to calculate the electric quantities, e.g. the electric charges Q_φ, if the piezoceramic components are used as sensors or are part of arbitrary electric circuits.

The verification example in Figs. 2 and 3 shows a slider crank mechanism moving with constant rotational velocity and exposed to a large axial force. Piezoceramic

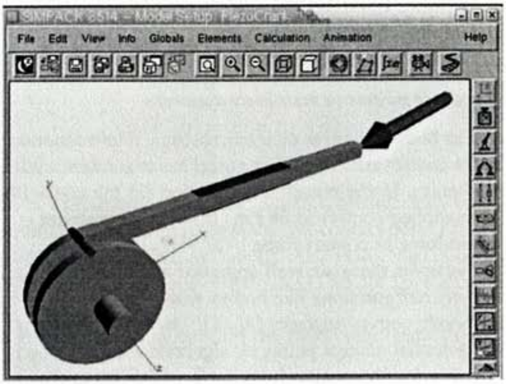

Fig. 2. Slider crank mechanism: Model setup.

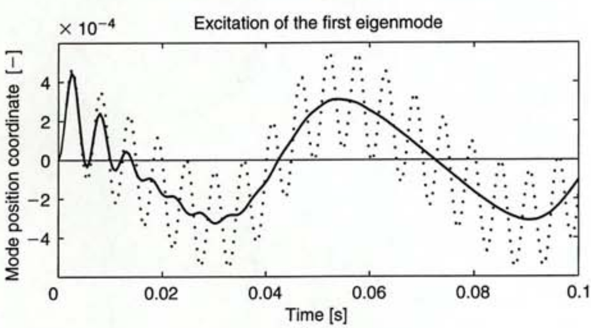

Fig. 3. Slider crank mechanism: Excitation of the first eigenmode, mechanism with piezoelements (solid line) and without piezoelements (dotted line).

actuators are mounted in central position on both sides of the slider. The piezoceramics are part of a serial vibration circuit, consisting of capacity, inductivity and resistance. The resonance frequency of the circuit was adjusted to the lowest natural frequency of the slider to achieve passive damping. The time plot compares the undamped behavior of the sliders first eigenmode to the damped one.

The comparison shows that piezoelectric devices are in principle well capable of reducing structural vibrations. For efficient use active control of the piezoelectric devices has to be set up. In [16] a railway carbody with piezoceramic patches has been considered. The positions of all patches and the control parameters are optimized to achieve desired damping characteristics.

2.3 *Advanced contact models in multibody dynamics*

The contact between bodies involves their macroscopic relative motion and the microscopic effects in the contact area. A contact model has to combine multibody dynamics with contact mechanics. In the equations of motion (1) the contact between bodies results either in an applied contact force $\boldsymbol{f}$ in (1a) or in a constraint (1b) with a corresponding constraint force as contact force.

In multibody dynamics there are well approved and numerically efficient contact models for standard configurations like bodies with primitive surface geometry [17] or bodies with smooth, convex surfaces [4, 18]. The latter approach is based on the determination of potential contact points by algebraic auxiliary conditions (1b). It is in general not suitable for non-convex body surfaces that may result in discontinuous and non-unique movements of the contact point. Moreover, the definition of contact forces by unilateral constraints and impact hypotheses, unilateral spring-damper force elements or Hertzian contact cannot provide the desired modeling quality of real collision processes.

In this section two alternative contact models are introduced. Based on the classical rigid body contact model a quasi-elastic model has been developed for the wheel-rail contact. A more general approach represents all body surfaces by polygonal surfaces and uses contact elements for the computation of the contact forces.

Rigid and *quasi-elastic contact model* The most simple contact model is the rigid contact between two bodies. Here the undeformed surfaces of both bodies are supposed to be in contact without penetrating each other. To formulate these two conditions analytically the surface of the first body is parametrized by surface coordinates $s \in S \subset \mathbb{R}^2$. For all points $P(s)$ on the surface of the first body the *distance function* $\Delta(s; \boldsymbol{q})$ is defined as the distance of $P(s)$ to the surface of the second body, measured in the direction of the outer normal in $P(s)$. The distance function Δ depends on the relative position and orientation of both bodies and is determined by the position coordinates $\boldsymbol{q}$ of the multibody system.

The two bodies do not penetrate each other iff

$$\Delta(s; \boldsymbol{q}) \geqslant 0, \ (s \in S).$$

The two bodies are in rigid contact iff the contact condition

$$0 = \min_{s \in S} \Delta(s; \boldsymbol{q}) =: \gamma(\boldsymbol{q}) \tag{9}$$

is satisfied. This contact condition $\gamma(\boldsymbol{q}) = 0$ is part of the constraints (1b).

If both bodies are in contact then there is at least one *contact point* $P(s^*)$ that is characterized by

$$\Delta(s^*; \boldsymbol{q}) = \min_{s \in S} \Delta(s; \boldsymbol{q}) = 0 \,.$$

For convex bodies with smooth surfaces $\boldsymbol{s}^*$ depends continuously on $\boldsymbol{q}$ and (9) defines a smooth constraint (1b). In general, however, the contact point $P(s^*)$ does not always depend continuously on $\boldsymbol{q}$, but it may "jump". As a result $\gamma(\boldsymbol{q})$ is only continuous but not differentiable and the equations of motion (1) do not have a continuous solution $(\boldsymbol{q}, \boldsymbol{q})$, see [19].

In the *quasi-elastic* contact model [19] the rigid body contact condition (9) is formally regularized by

$$0 = -\nu \ln \left(\int_S \exp(-\frac{1}{\nu} \Delta(s; \boldsymbol{q})) \, \mathrm{d}s \; / \; \int_S \mathrm{d}s \right) =: \gamma_\nu(\boldsymbol{q}) \tag{10}$$

with a small parameter $\nu > 0$. Similar to a full elastic contact model the quasi-elastic contact condition (10) considers function values $\Delta(s; \boldsymbol{q})$ for all points $P(s)$ in the contact area between the two bodies. The resulting constraint $\gamma_\nu(\boldsymbol{q}) = 0$ is smooth for bodies with smooth surfaces. For $\nu \to 0$ the regularized function $\gamma_\nu(\boldsymbol{q})$ converges pointwise to $\gamma(\boldsymbol{q})$.

From the practical point of view condition (10) is too complicated and the evaluation of γ_ν is very time consuming. Nevertheless it is used successfully for modeling the wheel-rail contact. In this special application the symmetry of wheel and rail may be exploited to simplify (10) substantially [20, 21]. This quasi-elastic contact model is part of the key features of SIMPACK Wheel/Rail.

Polygonal contact model The quasi-elastic model is one of the highly efficient, accurate and validated methods that have been developed for a special contact problem. For the representation of general contact mechanics we propose a *polygonal* contact model that is more flexible. It is based on the representation of body surfaces by polygonal surfaces which can be exported conveniently from CAD and VR software.

Instead of the analytical relations of classical contact mechanics the polygonal contact model uses *collision detection* algorithms similar to VR applications. These algorithms are applied to all relevant pairs of polygonal body surfaces in the MBS model. Based on the relative position and orientation of both bodies the collision detection algorithm checks if the surfaces intersect. Furthermore all intersection lines are obtained (if there are any).

In principle this problem could be solved by testing all possible pairings of polygons of the surfaces for intersection, but this fairly simple method has complexity $O(n^2)$ with n denoting the number of polygons. Even for quite small surfaces of some hundred polygons the resulting calculation effort is not acceptable. Collision detection based on hierarchical *bounding volumes* [22] operates considerably more efficiently without loss of accuracy.

In a pre-processing step before the first collision analysis a binary tree of bounding volumes is generated for every surface. These bounding volumes consist of identically oriented cuboids. The root element is defined such that it contains the whole surface. The children of the lower levels are obtained by continued subdivision of their parents, see Figure 4. In the end the cuboids of the leaf elements enclose single polygons of the surface.

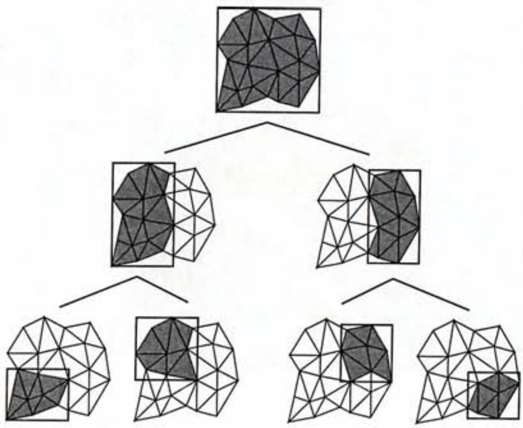

Fig. 4. 2D example for a bounding volume hierarchy.

Then the collision detection starts with the consideration of the bounding volumes in their actual relative position and orientation, going ahead from the roots to the leafs. Cuboids of child elements are only analysed if their parents intersect, and the quite costly intersection test of polygons [23] is performed for intersecting leaf elements only. Depending on the contact state this method may speed up collision detection by several orders of magnitude [22].

If the collision detection algorithm indicates that two bodies of the multibody system are in contact then the contact forces between both bodies have to be determined. Finite element discretizations can be used for the very accurate computation of contact forces, but their numerical effort exceeds by far the range of classical multibody system simulations [24]. In practical applications there is an urgent need for more efficient methods which may be less accurate. *Contact element* methods are characterized by regularly arranged elastic contact elements in the contact area.

Postulating rigid bodies covered by elastic layers leads to the simplest *Mattress model* which can be imagined as unilateral springs attached to the bodies' surfaces [25]. The *Influence function method* [26] approximates the bodies by elastic half-spaces. It represents the next step concerning modeling quality as well as numerical effort. Here adjacent contact elements influence each other resulting in the consideration of shearing stresses.

These contact element methods can also be applied to the tangential contact problem such that the friction forces may be obtained [25]. As soon as the stresses of the contact elements are known, they can easily be summed up to obtain the resulting forces and torques that are part of the term $\boldsymbol{f}(\dots)$ in the equations of motion (1).

3 CO-SIMULATION FOR MULTIDISCIPLINARY PROBLEMS

Today efficient and well approved monodisciplinary simulation tools are readily available in all practically relevant fields of application. But multidisciplinary effects involve different subjects of physics and are often beyond the range of application of these classical tools. In many cases they may be handled conveniently by the coupling of two or more already existing spezialized simulation tools. In Section 3.1 some basic ideas and problems of these *co-simulation* (or *simulator coupling*) techniques are summarized. Two practical applications are considered in more detail in Sections 3.2 and 3.3.

3.1 *Basic principle*

Most multidisciplinary problems have a clear modular structure that may be adopted in the simulation using for each subsystem its own simulation tool for model setup and time integration [27]. Well established standard software tools are used for the individual subsystems. In this manner the subsystems are integrated by *different* time integration methods such that each of these methods can be tailored to the solution behaviour of the corresponding subsystem (*co*-simulation).

The communication between subsystems is restricted to discrete synchronization points T_n. In each subsystem all necessary information from other subsystems has to be provided by interpolation or-if data for interpolation are not yet available – by extrapolation from $t \leqslant T_n$ to the actual *macro step* $T_n \to T_{n+1}$.

From the viewpoint of a multibody system tool the coupling variables to other simulation tools may be considered as a special type of discrete variables r_n in (2): In the multibody system tool the values of the coupling variables r_n are kept constant during the whole macro step $T_n \to T_{n+1}$. For all other simulation tools in the co-simulation environment the update formula (2b) involves the time integration from the synchronization point $t = T_n$ to the synchronization point $t = T_{n+1}$ to get r_{n+1}.

Co-simulation techniques are convenient but they may suffer from numerical instability. Furthermore, interpolation and extrapolation introduce additional discretization errors. Careful choice of the *macro stepsize* $H := T_{n+1} - T_n$ is needed. In typical standard applications stability and accuracy may be guaranteed if H is in the range between 0.1 ms and 10 ms.

For certain classes of coupled problems the instability phenomenon has been analysed in great detail. Several modifications of the co-simulation techniques help to improve their stability, accuracy and robustness also for larger macro stepsizes [27, 28].

3.2 *The interaction of vehicle dynamics and aerodynamics*

Traditionally, aerodynamic effects in vehicle dynamics have been considered by aerodynamic coefficients. But a much more detailed analysis of this multidisciplinary coupled system gets possible if a multibody system code is coupled with a solver from computational fluid dynamics (CFD). Such strategy is capable to describe virtually every *unsteady* aerodynamic phenomenon and to take into account the reciprocal interaction between mechanical and aerodynamical system.

A new application field for this coupled approach is the behaviour of ground vehicles under unsteady aerodynamic loads, for example due to the interaction with other vehicles. Such problems can not be handled by the conventional approach based on aerodynamic coefficients. The typical case of two high speed trains passing by each other is presented below.

The flow around high speed trains in absence of cross wind can be assumed to be inviscid and irrotational, leading to a linear aerodynamic model. Such a flow model is called *potential* flow and is widely used in aircraft aerodynamics. Its discretized numerical formulations, the *panel methods* [29, 30], lead to small computational effort and other benefits compared to nonlinear aerodynamic models.

A potential flow can be described by the Laplace equation by introducing a scalar field function Φ:

$$\nabla^2 \Phi = 0 \tag{11}$$

whereby potential function and velocity field are directly connected:

$$\boldsymbol{u} = \nabla \Phi \,. \tag{12}$$

The boundary condition for the flow only requires that the normal component of the relative velocity on the vehicles walls $\partial\Omega_V$ vanishes, i.e. that the normal component of the absolute velocity u is equal to the velocity of the wall v:

$$\nabla \Phi \cdot \boldsymbol{n} = \boldsymbol{v}(\dot{q}) \cdot \boldsymbol{n} \qquad \text{on } \partial\Omega_V \tag{13}$$

which shows that the potential Φ must depend on the velocity of the vehicles $\dot{q}$.

Using Green's formula Eq. (11) can be rearranged to obtain an expression for the potential Φ as integral on the vehicles walls $\partial\Omega_V$ of a *source* distribution σ divided by the norm of the position vector r. A *doublet* distribution, which arises in the general formulation, is not necessary for the case of ground vehicles because no special conditions, such as the Kutta-condition, have to be satisfied.

Since $\partial\Omega_V$ depends on the vehicles position, Φ depends on q as well:

$$\Phi(\boldsymbol{r}, \boldsymbol{q}, \dot{\boldsymbol{q}}) = \frac{1}{4\pi} \int_{\partial\Omega_V} \sigma \, \frac{1}{|\boldsymbol{r}|} \, \mathrm{d}s \,. \tag{14}$$

The source distribution σ on $\partial\Omega_V$ is unknown and has to be determined using the boundary condition (13). When σ has been computed, Φ and u can be derived using (14) and (12).

The Bernoulli Equation can now be applied to obtain the pressure field:

$$\frac{\partial \Phi}{\partial t} + \frac{|\boldsymbol{u}|^2}{2} + \frac{p}{\rho} = \frac{p_\infty}{\rho} = \text{const} \quad \Rightarrow \quad p(\boldsymbol{r}, \boldsymbol{q}, \dot{\boldsymbol{q}}, t) \,. \tag{15}$$

It is finally possible to compute the resulting flow force L_{flow} and torque M_{flow} related to the origin O:

$$\boldsymbol{L}_{\text{flow}}(\boldsymbol{q}, \dot{\boldsymbol{q}}, t) = - \int_{\partial\Omega_V} p \cdot \boldsymbol{n} \, \mathrm{d}s \,, \tag{16a}$$

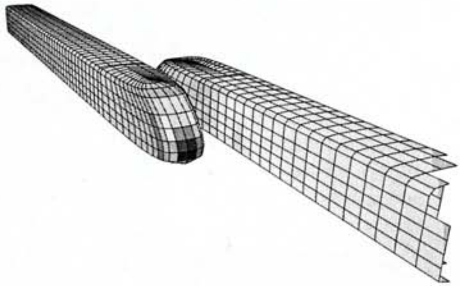

Fig. 5. Passing manoeuvre on open track. Model setup in co-simulation CFD–MBS, shading according to pressure distribution.

$$M_{\text{flow}}(q, \dot{q}, t) = -\int_{\partial\Omega_V} r \times p \cdot n \, \mathrm{d}s \tag{16b}$$

which couple the flow equations (11) and (13) with the multibody system equations (1).

The used panel method adopts a discretization of the surface integral in (14). The finite surface elements are called *panels* and on each of them the source distribution σ_i is supposed to be constant. The boundary condition (13) leads to an algebraic linear system whose unknown vector is the discrete source distribution σ_i and whose dimension is thus the number of panels. Eq. (15) has also to be discretized: pressure distribution and forces (16) can be finally obtained on a discrete time axis.

In order to minimize the computational effort the number of "aerodynamic" time steps must be minimized. The panel method allows very large time steps compared to the multibody system part. Furthermore, the flow and driving dynamics are quite weakly coupled. For these reasons a co-simulation technique has been implemented. In each macro step $T_n \to T_{n+1}$ Eq. (15) is discretized once using the macro step-size H as timestep. The flow field is thus resolved only at the syncronization points T_n and kept frozen between them. In this way sufficiently accurate results are obtained as can be seen from additional numerical tests with halved macro stepsize. The multibody system part of the coupled problem is solved by a standard integrator for multibody dynamics with stepsize and order control.

The simulation of a wide range of typical driving manoeuvres (passing on open track and at tunnel entrance, tunnel run-in and run-out, etc.) has been performed, see Figure 5 for a characteristic example. One of the most interesting results is that small disturbances leading to partial unloading of wheels can considerably amplify the dynamical response of the vehicles to the impulsive aerodynamical loads. This effect

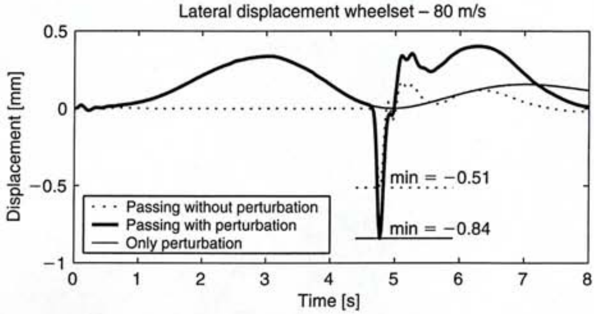

Fig. 6. Lateral displacement of the first wheelset for a symmetrical passing manoeuvre on open track.

is strongly influenced by various factors, e.g. the phase gap between disturbance and excitation. Figure 6 shows such a typical situation: a small, low frequency perturbation, which could be caused by cross wind or track irregularities in a real environment, lets the displacement of the wheelset (thick line) reach much larger values than in the ideal case (dashed line).

Performed co-simulations also pointed out that, even if the unsteady aerodynamic loads can exert very large influence on the driving dynamics, the effects of the induced vehicle motion on the surrounding flow, i.e. of the multibody variables on the aerodynamic variables, is of some influence only when the fundamental frequency of the transient loads approaches the lowest natural frequencies of the car motion. In the case of symmetrical passing manoeuvres this condition is satisfied at very small driving velocities, which are of no technical interest because at low speed the transient loads have small amplitude. Furthermore the vehicle motion causes modifications in the aerodynamic forces which induce in turn no significant alteration of the mechanical behavior, e.g. of the lateral displacement of the wheelsets. The influence can be thus neglected for all studied cases.

3.3 *Dynamical interaction between rail vehicle and elastic guideway for trains crossing a bridge*

High speed trains or heavy trucks crossing a bridge induce loads on the elastic structure and may damage the bridge. Computer based simulations are used in the design and optimization of improved vehicle suspensions to reduce the risk of damage (the "road-friendly truck") [31]. Furthermore, the simulation results show the influence of vehicle–guideway interaction on vehicle dynamics.

The dynamical interaction of vehicles and elastic bridges or, more generally, the interaction of multibody systems and large elastic structures is a multidisciplinary problem that involves both multibody dynamics and structural mechanics.

In principle, the coupled problem could be handled as flexible multibody system but a huge number of eigenmodes would be necessary to resolve local effects in the elastic structure. On the other hand the reference configuration of bridges is inertially fixed in contrast to the moving frame of reference in flexible multibody dynamics. For both reasons the co-simulation approach is an interesting alternative that is in the present application not only more convenient but also more efficient than the methods from flexible multibody dynamics [9].

The vehicle is modeled in the industrial multibody system tool SIMPACK resulting in equations of motion of the form (1). The industrial finite element package NAS-TRAN is used for the modeling of the bridge:

$$\boldsymbol{M}_b\ddot{\boldsymbol{w}} + \boldsymbol{D}_b\dot{\boldsymbol{w}} + \boldsymbol{K}_b\boldsymbol{w} = \boldsymbol{p}_b(\boldsymbol{F}(t, \boldsymbol{q}, \dot{\boldsymbol{q}}, \boldsymbol{\lambda}))\,. \tag{17}$$

Here M_b, D_b, K_b denote mass, damping and stiffness matrix. The load vector pb is determined by the forces F (t, q, q, λ) that depend on the state (q, q, λ) of the multibody model and represent tyre forces or wheel-rail contact forces.

In the framework of co-simulation the multibody model could be coupled directly to this finite element model (17) with $n_w \approx 15{,}000$ degrees of freedom. But modal reduction with the $n_b \approx 100$ eigenmodes φ_j in the frequency range between 0 Hz and 40 Hz gives nearly identical simulation results and reduces the numerical effort drastically [9]. After modal reduction the equations of motion may be decoupled into the system

$$m_j\ddot{w}_j + d_j\dot{w}_j + k_jw_j = \boldsymbol{\varphi}_j^T\boldsymbol{p}_b(\boldsymbol{F}(t, \boldsymbol{q}, \dot{\boldsymbol{q}}, \boldsymbol{\lambda}))\,,\ \ (j = 1, \ldots, n_b) \tag{18}$$

that may be solved very efficiently in each macro step using semi-analytical methods [32].

In the beginning of each macro step the force terms $F(t, q, q, \lambda)$ are transferred to the elastic structure. In Stage 1 these loads are used in the time integration of (18). The resulting elastic deformation w of the bridge is added to the track irregularities in the multibody system tool. In Stage 2 of the macro step the equations of motion (1) are integrated for $t \in [T_n, T_{n+1}]$ using standard methods from multibody dynamics. This co-simulation algorithm is stable and sufficiently accurate if the macro stepsize H is in the range of 1.0 ms.

As a typical result Figure 7 shows the vertical displacement of the bridge at the fixed position $x_b = 28.0$ m. In this manoeuvre two trains pass each other on the bridge. The first train reaches x_b after 2.63 s and the second train after 7.06 s. At both time instances the elastic deformation reaches a local maximum.

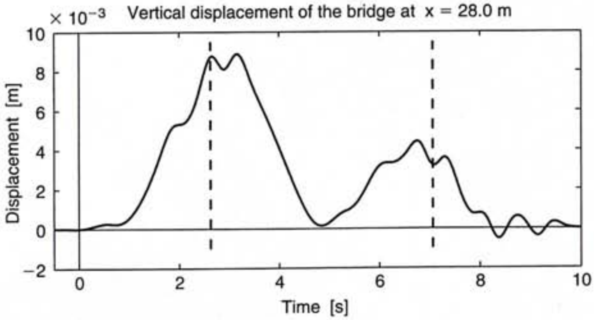

Fig. 7. Vertical displacement of a bridge with two trains passing each other.

In the simulation a Pentium III PC was used with sophisticated nonlinear multibody system models for both trains ($n_q > 100$). The coupled simulation was performed in 580 s cpu-time compared with a cpu-time of 295 s for the pure multibody simulation of the two vehicles [9]. The co-simulation approach allows an efficient dynamical simulation of this complicated multidisciplinary engineering problem.

SUMMARY

Multibody system simulation tools provide a powerful basis for the simulation of multidisciplinary problems in the field of vehicle system dynamics. Main strategies are the extension of existing multibody tools and the coupling with other simulation tools via co-simulation interfaces.

Recent extensions of classical multibody system simulation tools are modal reduction techniques for distributed physical phenomena in multifield problems, advanced contact models and solvers for the efficient time integration of mixed continuous/discrete systems. Multibody system tools are not longer restricted to the classical field of (flexible) multibody dynamics but may be used as well in the simulation of complex mechatronic devices.

Special techniques have been developed and implemented to improve the efficiency and the numerical stability in the co-simulation of multidisciplinary effects in coupled mechanical systems. Combining the panel method with a multibody system model the influence of aerodynamics on vehicle dynamics was studied. Co-simulation was also used successfully in the analysis of the dynamical interaction between high speed trains and elastic guideways. These complex practical applications illustrate that multidisciplinary problems involving multibody dynamics, aerodynamics and structural mechanics may be solved efficiently by the coupling of simulation techniques from these different fields of technical simulation.

The stepwise extension of well established specialized simulation tools and their coupling by co-simulation will definitely be one of the future trends in the development of simulation methods and simulation software.

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REFERENCES

1. W.O. Schiehlen, editor. *Multibody Systems Handbook*. Springer–Verlag, Berlin Heidelberg New York, 1990.
2. R.E. Roberson and R. Schwertassek. *Dynamics of Multibody Systems*. Springer–Verlag, Berlin Heidelberg New York, 1988.
3. E. Eich-Soellner and C. Führer. *Numerical Methods in Multibody Dynamics*. Teubner–Verlag, Stuttgart, 1998.
4. W. Rulka. *Effiziente Simulation der Dynamik mechatronischer Systeme für industrielle Anwendungen*. PhD thesis, Vienna University of Technology, Department of Mechanical Engineering, 1998.
5. W. Kortüm, W.O. Schiehlen, and M. Arnold. Software tools: From multibody system analysis to vehicle system dynamics. In H. Aref and J.W. Phillips, editors, *Mechanics for a New Millennium*, pages 225–238, Dordrecht, 2001. Kluwer Academic Publishers.
6. P.I. Barton, R.J. Allgor, W.F. Feehery, and S. Galán. Dynamic optimization in a discontinuous world. *Ind. Eng. Chem. Res.*, 37:966–981, 1998.
7. M. Arnold and G. Hippmann. Handling of discontinuities in SIMPACK (in German). Workshop of the VDI/VDE–GMA Special interest group 1.30 on "Modelling and Simulation Techniques for Automatisation" at the University of Stuttgart, November 2001.
8. A. Fuchs. Effiziente Korrektoriteration für implizite Zeitintegrationsverfahren in der Mehrkörper-dynamik. Master Thesis, Munich University of Technology, Department of Mathematics, 2002.
9. S. Dietz, G. Hippmann, and G. Schupp. Interaction of vehicles and flexible tracks by co-simulation of multibody vehicle systems and finite element track models. 17th Symposium Dynamics of Vehicles on Roads and Tracks IAVSD 2001, August 20–24, Copenhagen, 2001.

10. R. Schwertassek and O. Wallrapp. *Dynamik flexibler Mehrkörpersysteme*. Vieweg, 1999.
11. S. Dietz. *Vibration and Fatigue Analysis of Vehicle Systems using Component Modes*. Fortschritt-Berichte VDI Reihe 12, Nr. 401. VDI–Verlag, Düsseldorf, 1999.
12. J. Tich" and G. Gautschi. *Piezoelektrische Meßtechnik*. Springer–Verlag, Berlin, 1980.
13. M. Rose and D. Sachau. Multibody simulation of mechanism with distributed actuators on lightweight components. In *Proceedings of SPIE's 8th Annual International Symposium on Smart Structures and Materials*, Newport Beach, CA, 2001.
14. R.L. Clark, W.R. Saunders, and G.P. Bibbs. *Adaptive Structures – Dynamics and Control*. John Wiley & Sons, New York, 1998.
15. A. Preumont. *Vibration Control of Active Structures*. Kluwer Academic Publishers, Dordrecht, 2002.
16. A. Heckmann and O. Vaculín. Multibody simulation of actively controlled carbody flexibility. In *Proc. of the International Conference on Noise & Vibration Engineering ISMA2002, Leuven*, 2002.
17. DADS contact modeling guide. LMS CADSI Leuven, Belgium, 1997.
18. F. Pfeiffer and Ch. Glocker. *Multibody Dynamics with Unilateral Contacts*. Wiley & Sons, New York, 1996.
19. M. Arnold and K. Frischmuth. Solving problems with unilateral constraints by DAE methods. *Mathematics and Computers in Simulation*, 47:47–67, 1998.
20. H. Netter. *Rad–Schiene–Systeme in differential-algebraischer Darstellung*. Fortschritt-Berichte VDI Reihe 12, Nr. 352. VDI–Verlag, Düseldorf, 1998.
21. M. Arnold and H. Netter. Approximation of contact geometry in the dynamical simulation of wheel-rail systems. *Mathematical and Computer Modelling of Dynamical Systems*, 4:162–184, 1998.
22. G. Zachmann. *Virtual reality in assembly simulation – Collision detection, simulation algorithms and interaction techniques*. PhD thesis, Darmstadt University of Technology, Department of Computer Science, 2000.
23. T. Möller. A fast triangle-triangle intersection test. *Journal of graphics tools*, 2:25–30, 1997.
24. P. Eberhard. *Kontaktuntersuchung durch hybride Mehrkörpersystem/Finite Elemente Simulationen*. Shaker Verlag, Aachen, 2000.
25. K.L. Johnson. *Contact Mechanics*. Cambridge University Press, 1985.
26. J.J. Kalker. *Three-Dimensional Elastic Bodies in Rolling Contact*. Kluwer Academic Publishers, Dordrecht Boston London, 1990.
27. R. Kübler and W. Schiehlen. Two methods of simulator coupling. *Mathematical and Computer Modelling of Dynamical Systems*, 6:93–113, 2000.
28. M. Arnold and M. Günther. Preconditioned dynamic iteration for coupled differential-algebraic systems. *BIT Numerical Mathematics*, 41:1–25, 2001.
29. L.L. Erickson. Panel methods – an introduction. Technical Report 2995, NASA, 1990.
30. T.W. Chiu. Modern panel method in vehicle aerodynamics: Formula-one cars and trains. *Flow at Large Reynold Numbers. Advances in Fluid Mechanics*, 11:35–74, 1997.
31. M. Valášek and W. Kortüm. Nonlinear control for semi-active road-friendly truck suspension. In *Proc. of the Intern. Symp. on Advanced Vehicle Control, AVEC'98*, pages 275–280, 1998.
32. S. Dietz, M. Arnold, W. Duffek, G. Schupp, and G. Hippmann. Co-Simulation of MBS and FEM for the simulation of vehicles crossing a bridge, (in German). In M. Arnold, editor, *Working materials of the Workshop "Fahrzeug–Fahrweg–Wechselwirkung", March 5, 2001, Oberpfaffenhofen*, pages 20–28, DLR German Aerospace Center, Technical Report IB 532-01-04, March 2001.