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M. Schulze, S. Dietz, A. Tuganov, J. Linn, H. Lang

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Fraunhofer-Institut für Techno- und
Wirtschaftsmathematik ITWM
Fraunhofer-Platz 1

67663 Kaiserslautern
Germany

Telefon: +49(0)631/31600-4674
Telefax: +49(0)631/31600-5674
E-Mail: presse@itwm.fraunhofer.de
Internet: www.itwm.fraunhofer.de

Vorwort

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Darüber hinaus bietet die Reihe ein Forum für die Berichterstattung über die zahlreichen Kooperationsprojekte des Instituts mit Partnern aus Industrie und Wirtschaft.

Berichterstattung heißt hier Dokumentation des Transfers aktueller Ergebnisse aus mathematischer Forschungs- und Entwicklungsarbeit in industrielle Anwendungen und Softwareprodukte – und umgekehrt, denn Probleme der Praxis generieren neue interessante mathematische Fragestellungen.

Prof. Dr. Dieter Prätzel-Wolters
Institutsleiter

Kaiserslautern, im Juni 2001

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Integration of nonlinear models of flexible body deformation in Multibody System Dynamics

Martin Schulze*, Stefan Dietz*, Andrey Tuganov†, Holger Lang#, Joachim Linn†

* SIMPACK AG
Friedrichshafener Straße 1,
82205 Gilching, Germany
[Martin.Schulze,
Stefan.Dietz]@simpack.de

\# Chair of Applied Dynamics
University of Erlangen-Nuremberg
Konrad-Zuse-Str. 3/5,
91052 Erlangen, Germany
Holger.Lang@ltd.uni-erlangen.de

† Fraunhofer Institut für Techno- und Wirtschaftsmathematik (ITWM)
Fraunhofer-Platz 1, 67663 Kaiserslautern, Germany
[Andrey.Tuganov,Joachim.Linn]@itwm.fraunhofer.de

ABSTRACT

A simple transformation of the Equation of Motion (EoM) allows us to directly integrate nonlinear structural models into the recursive Multibody System (MBS) formalism of SIMPACK. This contribution describes how the integration is performed for a discrete Cosserat rod model which has been developed at the ITWM. As a practical example, the run-up of a simplified three-bladed wind turbine is studied where the dynamic deformations of the three blades are calculated by the Cosserat rod model.

1. INTRODUCTION

Multibody System Dynamics has an ever-growing field of applications. With increasing performance of affordable computing hardware in MBS, like in other disciplines of simulation, there is a trend to increase the level of detail of the models. A growing number of models include system parts that are deformable and the deformation of these parts affects the system dynamics. Examples are the housings of gear drives, motors and generators, and movable parts like conrods, axles and blades. Classically, when deformable parts shall be included in an MBS model, a two-step approach is used: (a) their linear response to static and dynamic external excitation is analyzed by means of Finite Element (FE) methods, (b) a modal reduction technique is used to obtain an equation of motion that describes the part deformation with a minimum number of degrees of freedom (d.o.f.). The first step is usually performed with the help of FE software, the second step can be performed by MBS software like SIMPACK. The MBS software can then solve the EoM of the modally reduced deformable part along with the EoMs of other MBS parts in a fully integrated manner [9].

For applications where nonlinear material or deformation effects dominate the system behaviour this approach is not sufficient. Straight-forward extensions like nonlinear modal reduction require a-priori knowledge of the deformation state and therefore have to be taylored to specific applications [8] to obtain a computationally efficient implementation. For beams, SIMPACK provides with SIMBEAM a model which correctly describes the deformation up to second order [2]. A more general approach is to include a nonlinear EoM that describes the part deformation e.g. in a geometrically exact sense. Such an approach would usually be based on the positions and velocities of a number of nodes that describe the current geometry of the deformable part, e.g. by means of an FE model. The nonlinear EoMs can in principle be integrated into an MBS model in the same way as the EoMs of modally reduced system parts. Joint forces, external forces and constraints are directly applied to the nodes that describe the deformable part and thus there is a natural interface between the nonlinear EoMs and the EoMs of the other MBS parts.

For beams or beam-like structures one such nonlinear model is the Cosserat rod model that has been developed at the ITWM [7]. This model is based on a geometrically exact description of a beam that is exposed to bending, twisting, shearing and extension. Thus, it is a generic model and applicable to a wide range of industrial parts, e.g. all kinds of axles, chains and blades.

The following sections contain a brief summary of the derivation of the EoM of the Cosserat rod model and a description of how this model is integrated into SIMPACK's recursive MBS formalism. This is followed by a discussion of how the different SIMPACK solvers perform for MBS models where parts are described by the Cosserat rod model. Finally, benchmark examples including the rotor blade of a 5MW NREL wind turbine are studied in order to validate the method and – as a practical example – the deformations of the rotor blades of a simplified, three-bladed wind turbine model during run-up are calculated.

2. NON-LINEAR COSSERAT ROD MODEL

2.1. Equations of motion

The current configuration of a Cosserat rod is completely determined by its centreline of mass centroids

$$\boldsymbol{x} : [0, L] \times [0, T] \to \Re^3, \quad (s,t) \mapsto \boldsymbol{x}(s,t) \tag{1}$$

and a quaternion rotation field $\boldsymbol{p}$ defining its orthonormal director frame field $\boldsymbol{R}(\boldsymbol{p})$

$$\begin{gathered} \boldsymbol{R} \circ \boldsymbol{p} : [0, L] \times [0, T] \xrightarrow{\boldsymbol{p}} S^3 = \partial B_1^H(0) \xrightarrow{R} SO(3), \\ (s,t) \mapsto \boldsymbol{R}(\boldsymbol{p}), \quad \boldsymbol{R} = \boldsymbol{R}(\boldsymbol{p}) = \left(\boldsymbol{d}^{(1)}(\boldsymbol{p}) \middle| \boldsymbol{d}^{(2)}(\boldsymbol{p}) \middle| \boldsymbol{d}^{(3)}(\boldsymbol{p})\right), \end{gathered} \tag{2}$$

by means of the Euler map

$$\boldsymbol{R}(\boldsymbol{p}) = (2p_0^2 - \|\boldsymbol{p}\|^2)\boldsymbol{E} + 2\Im(\boldsymbol{p}) \otimes \Im(\boldsymbol{p}) + 2p_0 \boldsymbol{\xi}(\Im(\boldsymbol{p})), \tag{3}$$

where $\boldsymbol{p} = p_0 + \Im(\boldsymbol{p})$ and $\boldsymbol{\xi}(.)$ is the alternating skew tensor ($\boldsymbol{\xi}(\boldsymbol{u})\boldsymbol{v} = \boldsymbol{u} \times \boldsymbol{v}$ for $\boldsymbol{u}, \boldsymbol{v} \in \Re^3$).

We define the material strain measures of the rod $\boldsymbol{V} = \boldsymbol{R}^T \partial_s \boldsymbol{x}$ and $\boldsymbol{\xi}(\boldsymbol{U}) = \boldsymbol{R}^T \partial_s \boldsymbol{R}$, where individual components V_1, V_2 measure transverse shear and V_3 the extensional deformation, U_1, U_2 measure the bending curvatures with respect to the director axes $\boldsymbol{d}^{(1)}$, $\boldsymbol{d}^{(2)}$, and U_3 the torsional twist around the cross section normal $\boldsymbol{d}^{(3)}$.

We introduce the potential energy of the elastic deformation of the Cosserat rod

$$W_{el} = \tfrac{1}{2} \int_0^L ds \left[\left((\boldsymbol{V} - \boldsymbol{V}_0) \,|\, (\boldsymbol{U} - \boldsymbol{U}_0) \right)^T \cdot \hat{\boldsymbol{C}} \cdot \left((\boldsymbol{V} - \boldsymbol{V}_0) \,|\, (\boldsymbol{U} - \boldsymbol{U}_0) \right) \right], \tag{4}$$

and the Rayleigh dissipation function

$$W_{visc} = \tfrac{1}{2} \int_0^L ds \left[\partial_t \left(\boldsymbol{V}^T \,|\, \boldsymbol{U}^T \right) \cdot \hat{\boldsymbol{D}} \cdot \partial_t \left(\boldsymbol{V} \,|\, \boldsymbol{U} \right) \right], \tag{5}$$

where the subscript $\cdot_0$ denotes the strain measures $\boldsymbol{V}$ and $\boldsymbol{U}$ of the undeformed rod. The elastic properties of the rod are determined by the effective stiffness parameters contained in the symmetric 6x6 matrix $\hat{\boldsymbol{C}}$. In the approximation range of small material strain, for homogeneous isotropic materials this matrix is diagonal and is given by:

$$\hat{\boldsymbol{C}} = \mathrm{diag}(GA\kappa_1, GA\kappa_2, EA, EI_1, EI_2, GI_3), \tag{6}$$

with stiffness parameters given by the elastic moduli E and G, geometric parameters of the cross section (area A and geometric moments I_k), and dimensionless shear correction factors κ_j which also depend on the cross section geometry. We assume a similar structure for the matrix $\hat{\boldsymbol{D}}$, which determines the viscous response:

$$\hat{\boldsymbol{D}} = \mathrm{diag}(\gamma_{S1}, \gamma_{S2}, \gamma_E, \gamma_{B1}, \gamma_{B2}, \gamma_T). \tag{7}$$

These elastic and damping potentials provide us with the following continuous EoM of the rod

$$\begin{cases} \rho A\ddot{\boldsymbol{x}} = \partial_S \boldsymbol{f} + \boldsymbol{f}_e \\ \rho\left(\boldsymbol{I}\dot{\boldsymbol{\omega}} + \boldsymbol{\omega}\times\boldsymbol{I}\boldsymbol{\omega}\right) = \partial_S \boldsymbol{l} + \partial_S \boldsymbol{x}\times\boldsymbol{f} + \boldsymbol{l}_e \end{cases}, \tag{8}$$

where $\boldsymbol{f} = \boldsymbol{R}\boldsymbol{F}$ and $\boldsymbol{l} = \boldsymbol{R}\boldsymbol{L}$ are the internal viscoelastic forces and moments with

$$\boldsymbol{F} = \hat{\boldsymbol{C}}_F \cdot \left(\boldsymbol{V} - \boldsymbol{V}_0\right) + \hat{\boldsymbol{D}}_F \cdot \partial_t \boldsymbol{V}\,, \qquad \boldsymbol{L} = \hat{\boldsymbol{C}}_L \cdot \left(\boldsymbol{U} - \boldsymbol{U}_0\right) + \hat{\boldsymbol{D}}_L \cdot \partial_t \boldsymbol{U}\,, \tag{9}$$

$\boldsymbol{f}_e$ and $\boldsymbol{l}_e$ are external viscoelastic forces and moments, $\boldsymbol{\omega}$ is the angular velocity of the director frame, ρ is the material mass density, and $\hat{\boldsymbol{I}}_R$ and $\boldsymbol{I} = \boldsymbol{R}\hat{\boldsymbol{I}}_R\boldsymbol{R}^T$ are the inertial tensors of the cross-section in the director frame and absolute coordinates, respectively.

To discretise our model we subdivide the arc length interval $[0, L]$ into N segments $[s_{n-1}, s_n]$ with the vertices $0 = s_0 < s_1 < \ldots < s_{N-1} < s_N = L$. At each vertex we define a position pair $\{\dot{\boldsymbol{x}}_n, \boldsymbol{p}_n\}$ and a velocity pair $\{\ddot{\boldsymbol{x}}_n, \boldsymbol{\omega}_n\}$. An alternative staggered discretisation grid with translational degrees of freedom defined at vertices, but rotary degrees of freedom associated with the segments was proposed in [7]. Please also refer to [7] for a detailed description of the discrete model and for the derivation of the discrete equations of motions. Here we will presume the discrete model having the following set of Newton-Euler equations for the vertex coordinates:

$$\begin{bmatrix} m_n\boldsymbol{E} & \\ & \boldsymbol{I}_n \end{bmatrix} \begin{pmatrix} \ddot{\boldsymbol{x}}_n \\ \dot{\boldsymbol{\omega}}_n \end{pmatrix} = \begin{pmatrix} \boldsymbol{f}_n \\ \boldsymbol{l}_n \end{pmatrix}, \tag{10}$$

where m_n and $\boldsymbol{I}_n$ are the lumped masses of the vertex, $\boldsymbol{E}$ is the 3x3 identity matrix, and $\boldsymbol{f}_n$, $\boldsymbol{l}_n$ are the sums of the elastic, inertia, damping and external forces and moments, respectively.

2.2. Adaption for mass-centre and shear-centre offsets

In order to capture the dynamic effects of a centreline of mass centroids that is not coincident with the elastic centreline, the Newton-Euler equation (10) for the vertex coordinates of the elastic centreline is replaced by its generalized counterpart [3] (the index *n* has been omitted in favour of a better readability):

$$\begin{bmatrix} m\boldsymbol{E} & -m\boldsymbol{R}\xi(\zeta) \\ m\xi(\zeta)\boldsymbol{R}^T & \boldsymbol{I} - m\xi(\zeta)\xi(\zeta) \end{bmatrix} \begin{pmatrix} \ddot{\boldsymbol{x}} \\ \dot{\boldsymbol{\omega}} \end{pmatrix} = \begin{pmatrix} \boldsymbol{f} - m\boldsymbol{R}\boldsymbol{\omega}\times(\boldsymbol{\omega}\times\boldsymbol{\zeta}) \\ \boldsymbol{l} - \boldsymbol{\omega}\times(\boldsymbol{I}\boldsymbol{\omega}) \end{pmatrix}, \tag{11}$$

where $\boldsymbol{E}$ is the 3x3 identity matrix and $\boldsymbol{\zeta} = (\zeta_1, \zeta_2, 0)^T$ is the offset of the mass centroid with respect to the elastic centreline given in coordinates of the local frame $\boldsymbol{R}$.

A further effect needs to be taken into account when arbitrary beam cross sections shall be modelled: depending on the cross section geometry, a force that acts perpendicular to the beam axis at some point of the elastic centreline may not only bend the beam in the direction of the force vector but also twist it. In order to prevent this twisting the force must act at the so-called shear centreline. In the approximation range of small material strain, an offset of the shear centreline with respect to the elastic centreline can be accounted for by inserting the nonzero shearing-twisting coupling elements $GA\varsigma_j$ in the material matrix $\hat{\boldsymbol{C}}$, where $\boldsymbol{\varsigma} = (\varsigma_1, \varsigma_2, 0)^T$ is the offset of the shear centreline with respect to the elastic centreline [5].

2.3. Implementation

The elastic and damping potentials (4) and (5) were implemented in Maple and automatically differentiated to produce the viscoelastic components of the right hand side of the equations of motion (10) and its Jacobi-matrix. The results were exported in C and encapsulated in a C++ dynamic library containing a C++ class for the rod element. A rod object can be created through the library's API by defining the following parameters: number of nodes of discretisation and their position and orientation, geometrical parameters such as length, cross-section area, mass and shear centre offsets, moments of inertia, Young's and shear moduli of the material, damping coefficients for bending, shearing, extension and torsion as in (7). The API allows the user to evaluate separately mass, elastic, viscous and inertial components of the rod's equations of motion for any given position/velocity state vector.

3. INTEGRATION INTO THE RECURSIVE MBS FORMALISM

3.1. Equation of motion of rigid and flexible bodies in a multi-body system

Based on the Newton-Euler equations (10) it is possible to integrate the flexible body description of section 2 into a SIMPACK MBS model. In SIMPACK, a recursive formalism is used in order to describe the system dynamics with a minimal number of d.o.f.: in the case of a rigid body these are the states of the joint which connects the rigid body the preceding body in a so-called kinematic chain. The joint states are the translational and rotational displacements of these two bodies with respect to each other. A flexible body has (in general) additional states that describe the current deformation state in a coordinate frame that is rigidly attached to the body. This coordinate frame is called the body reference frame (BRF).

The derivation of the recursive formalism is based on the EoM of a (rigid or flexible) body [9]:

$$\begin{bmatrix} \boldsymbol{M}_R & \boldsymbol{S}^T \\ \boldsymbol{S} & \boldsymbol{M}_E \end{bmatrix} \begin{pmatrix} \dot{\boldsymbol{v}}_R \\ \dot{\boldsymbol{v}}_E \end{pmatrix} = \begin{pmatrix} \boldsymbol{h}_R \\ \boldsymbol{h}_E \end{pmatrix} + \sum_i \begin{bmatrix} \boldsymbol{J}_{R,i}^T \\ \boldsymbol{J}_{E,i}^T \end{bmatrix} \begin{pmatrix} \boldsymbol{f}_i \\ \boldsymbol{l}_i \end{pmatrix}, \tag{12}$$

where $\boldsymbol{M}_R$, $\boldsymbol{M}_E$ and $\boldsymbol{S}$ are the mass matrices for the (rigid) body motion, for the elastic deformation, and for the coupling of motion and deformation, respectively, $\boldsymbol{v}_R$ and $\boldsymbol{v}_E$ are the velocities of the body motion and the flexible states, $\boldsymbol{h}_R$ and $\boldsymbol{h}_E$ are the sums of elastic, inertia and damping forces which arise from the deformation and which act on the body motion and deformation in the BRF, respectively, $\boldsymbol{f}_i$ and $\boldsymbol{l}_i$ are external forces and moments which arise from force elements, joints and constraints, and $\boldsymbol{J}_{R,i}$ and $\boldsymbol{J}_{E,i}$ are the Jacobian matrices of (12) with respect to the external forces and moments.

In order to obtain a recursive formalism the accelerations $\dot{\boldsymbol{v}}_R$ are expressed by the accelerations of the preceding body in the kinematic chain and the accelerations of the joint states that connect the body with its predecessor. The joint state accelerations are also expressed by the accelerations of the preceding body. For the details of this derivation refer to [9].

3.2. Integration of arbitrary flexible body descriptions into the recursive formalism

The EoM of a body in SIMPACK (12) naturally has the same analytic structure as the Newton-Euler equations (10). In fact, it is quite straight-forward to transform the Newton-Euler equations into the form (12) such that they can directly be integrated into the recursive formalism.

The basic assumption for this transformation is that the BRF is rigidly attached to some reference vertex of the structural model. The rigid body accelerations, $\dot{\boldsymbol{v}}_R$, are then equal to the accelerations of this reference vertex. The accelerations of the flexible states are equal to the accelerations of the other vertices relative to the reference vertex. These can be found by transforming the vertex positions and angular velocities which enter the Newton-Euler equations (10) as follows:

$$\begin{pmatrix} \boldsymbol{x}_n \\ \boldsymbol{\omega}_n \end{pmatrix} = \begin{pmatrix} \boldsymbol{x}_0 + \boldsymbol{R}_0 \hat{\boldsymbol{x}}_n \\ \hat{\boldsymbol{R}}_n^T \boldsymbol{\omega}_0 + \hat{\boldsymbol{\omega}}_n \end{pmatrix}, \tag{13}$$

where $\hat{\boldsymbol{x}}_n$ is the vertex position relative to the reference vertex position $\boldsymbol{x}_0$ in the reference frame $\boldsymbol{R}_0$, $\hat{\boldsymbol{\omega}}_n$ is the relative angular velocity of the vertex' local frames orientation $\hat{\boldsymbol{R}}_n(\hat{\boldsymbol{p}}) = \boldsymbol{R}_0^T(\boldsymbol{p}_0)\boldsymbol{R}_n(\boldsymbol{p}_n)$ with respect to the reference frame. By time derivation of (13) the following identity can be derived:

$$\begin{pmatrix} \begin{pmatrix}\ddot{\boldsymbol{x}}_0\\ \dot{\boldsymbol{\omega}}_0\end{pmatrix}\\ \vdots \\ \begin{pmatrix}\ddot{\boldsymbol{x}}_n \\ \dot{\boldsymbol{\omega}}_n\end{pmatrix} \\ \vdots\end{pmatrix} = \underbrace{\begin{bmatrix} \begin{bmatrix} \boldsymbol{R}_0 & \\ & \boldsymbol{E}\end{bmatrix} & & \\ \vdots & \ddots & \\ \begin{bmatrix} \boldsymbol{R}_0 & -\boldsymbol{R}_0\xi(\hat{\boldsymbol{x}}_n) \\ & \hat{\boldsymbol{R}}_n^T\end{bmatrix} & \begin{bmatrix} \boldsymbol{R}_0 & \\ & \boldsymbol{E} \end{bmatrix} & \\ \vdots & & \ddots \end{bmatrix}}_{F} \begin{pmatrix} \begin{pmatrix} \boldsymbol{R}_0^T\ddot{\boldsymbol{x}}_0 \\ \dot{\boldsymbol{\omega}}_0\end{pmatrix} \\ \vdots \\ \begin{pmatrix}\ddot{\hat{\boldsymbol{x}}}_n \\ \dot{\hat{\boldsymbol{\omega}}}_n\end{pmatrix}\\ \vdots\end{pmatrix} + \underbrace{\begin{pmatrix} \begin{pmatrix} \boldsymbol{0}\\ \boldsymbol{0}\end{pmatrix} \\ \vdots \\ \begin{pmatrix} \boldsymbol{R}_0\xi(\boldsymbol{\omega}_0)\left(2\dot{\hat{\boldsymbol{x}}}_n + \xi(\boldsymbol{\omega}_0)\hat{\boldsymbol{x}}_n\right) \\ \xi(\hat{\boldsymbol{\omega}}_n)\hat{\boldsymbol{R}}_n^T\boldsymbol{\omega}_0 \end{pmatrix}\\ \vdots\end{pmatrix}}_{G}$$

By left-multiplying the equation by $\boldsymbol{F}^T\boldsymbol{M}$ and inserting (10) one obtains the EoM in the form (12). Therein, $\boldsymbol{F}^T\boldsymbol{M}\boldsymbol{F}$ is the mass matrix, $\dot{\boldsymbol{v}}_R = \left(\boldsymbol{R}_0^T\ddot{\boldsymbol{x}}_0, \dot{\boldsymbol{\omega}}_0\right)^T$ are the rigid body accelerations and $\dot{\boldsymbol{v}}_E = \left(\ldots, \ddot{\hat{\boldsymbol{x}}}_n, \dot{\hat{\boldsymbol{\omega}}}_n, \ldots\right)^T$ are the accelerations of the flexible states. The forces $\boldsymbol{h}_R$ and $\boldsymbol{h}_E$ can easily be identified. The external forces $\boldsymbol{f}_i$ and $\boldsymbol{l}_i$ are first added to the r.h.s. of the Newton-Euler equations (10) at the appropriate vertices. The corresponding Jacobian matrices $\boldsymbol{J}_{R,i}$ and $\boldsymbol{J}_{E,i}$ can then be identified in the transformed equation.

This transformation uniquely describes how the rigid body accelerations and the accelerations of the flexible states must be calculated during the evaluation of the right hand side (r.h.s.) of the MBS by any SIMPACK solver. This way, any flexible body that has a nodal interpretation of its flexible states can seamlessly be integrated into an MBS model. For flexible bodies that have different interpretations of their flexible states, analogous transformations can be performed.

3.3. Implementation

In order to demonstrate the integration of structural models into the recursive formalism in SIMPACK a prototypic library has been developed. This supplemental library has two APIs: one which is presented to the SIMPACK solver and one which is presented to the structural model that determines the behaviour of some flexible body in the MBS model. For communication with the SIMPACK solver functions exist to setup the flexible body, to define the connection of markers to vertices, to apply external forces and to perform kinematic measurements at these markers, and to evaluate the rigid body and flexible states' accelerations. The solver-side API has been designed in complete analogy to the existing solver-internal interface for a linear modal interpretation of the flexible body states. Thereby, the library calls can be used as drop-in replacements for existing routines. This way, the existing SIMPACK solvers can calculate the flexible body deformation with the new approach without further modifications to the code. The API for coupling a structural model to the library has been discussed in section 2.3.

4. INTERACTION WITH SIMPACK'S SOLVERS

4.1. Time integration

The EoM of the Cosserat rod (10) usually exhibits a stiff behaviour due to extension and shear deformation owed to the high frequencies of the stiff extension and shear modes. In practise, this out-rules the use of explicit time integration schemes because the minimal step-size for a stable solution would need to be unacceptably small. As for the implicit integrators, the DASSL-based solver SODASRT2 and the RADAU5 solver have shown to perform best. The SODASRT2 solver is usually faster than the RADAU5 solver. However, often a smaller tolerance needs to be set in order to obtain the same stability and the same accuracy for the solution as with the RADAU5 solver.

4.2. Static equilibrium

In order to find a static equilibrium of the MBS, SIMPACK ships with a Newton-Raphson based solver among others. This solver also works well when a Cosserat rod has been integrated in the MBS model.

4.3. Eigenvalue analysis

For an eigenvalue analysis SIMPACK performs a linearization of the r.h.s. for the complete MBS including all flexible states. This works without restrictions with respect to the structural models that are being used. Note that due to the quaternion formulation two extra eigenvalues per vertex with no physical meaning are calculated. These extra eigenvalues have zero eigenfrequency and can therefore be filtered out. However, also note that the physically correct eigenvalues of the reduced linear system are slightly different if the angular vertex velocities in the linearization state are not equal to zero. The linearization of the reduced system by eliminating the extra quaternion state has not yet been integrated into SIMPACK.

4.4. Performance considerations

In the case of multiple beams, beams with a larger number of vertices (>50), and stiff couplings between the bodies of the MBS the computational performance of all solvers is currently not sufficient. Especially it is not competitive when compared with dedicated finite element solvers. This is due to the fact that structural models usually have a large number of d.o.f. but expose a sparse structure for the system matrices. MBS systems in the recursive formulation, on the other hand, have a small number of d.o.f. but expose a dense structure for the system matrices. For obvious reasons the SIMPACK solvers are optimized for the latter problem class. Some approaches to address MBS with partially sparse system matrices already exist in SIMPACK: (a) the numerical Jacobian matrix is evaluated with a minimum number of r.h.s. calls if there are decoupled groups of d.o.f., (b) a Krylov method can be used for the corrector iteration of the implicit time integrator SODASRT2. These approaches do improve the performance to some extent. However, for large numbers of d.o.f. the measures are not sufficient.

4.5. Co-simulation

One measure to drastically improve the performance is co-simulation. This is usually possible when there are no stiff couplings between the bodies of the MBS and in a limited number of cases where stiff couplings do exist. The following simple co-simulation scheme can again be integrated seamlessly into the recursive formalism of the SIMPACK solvers: at the end of one recursion the accelerations of the flexible states are calculated. These would normally be copied into the state vector of one of the SIMPACK time integrator solvers. With co-simulation active, the accelerations are instead handed on to an additional solver which resides in the supplementary library. This additional solver then performs the time-integration of the flexible states for the duration of the next macro-time step of the SIMPACK solver. In its micro-time steps it directly uses the API to the structural models to update the contributions of the internal elastic, inertia and damping forces to the flexible states’ accelerations. The contributions of the coupling forces to the accelerations are kept constant over the macro-time step. At the beginning of the next macro-time step, the SIMPACK solver waits for the additional solvers to complete and then performs the kinematic evaluations that are needed to update the positions and velocities of the other bodies in the MBS and the coupling forces that are handed back to the co-simulated flexible bodies.

5. EXAMPLES

5.1. Swinging rubber bar

For a thorough validation, the dynamic solution of the Cosserat rod model integrated into an MBS needs to be investigated. One benchmark example is a slender 1m long rubber beam with 5mm radius that is connected to the fixed inertial frame via a revolute joint and exposed to the gravitational field. No analytical solution for this problem is known. Therefore, the solution of SIMPACK is compared to the solution of Abaqus and the solution that has been obtained with a stand-alone implementation of the Cosserat rod model (where the vertex coordinates are given in absolute coordinates and appropriate boundary conditions are applied). For a thorough discussion of the results for the stand-alone Cosserat rod model and a comparison with Abaqus see [7]. The deviations of the coupled MBS model from the stand-alone model are found to be within tolerances of the time integration schemes. This proves the correctness of the coupling forces between the vertex accelerations and the joint state. Exemplary, the time evolution of the elastic beam centreline as it has been obtained with the SIMPACK model is shown in Figure 1.

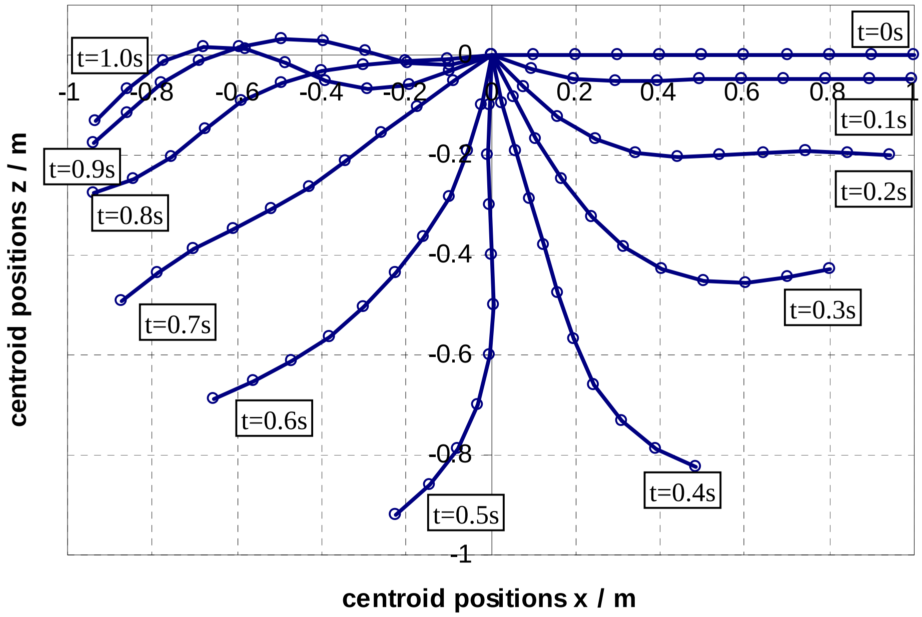

Figure 1. Time evolution of a swinging rubber beam.

5.2. Rotor blade

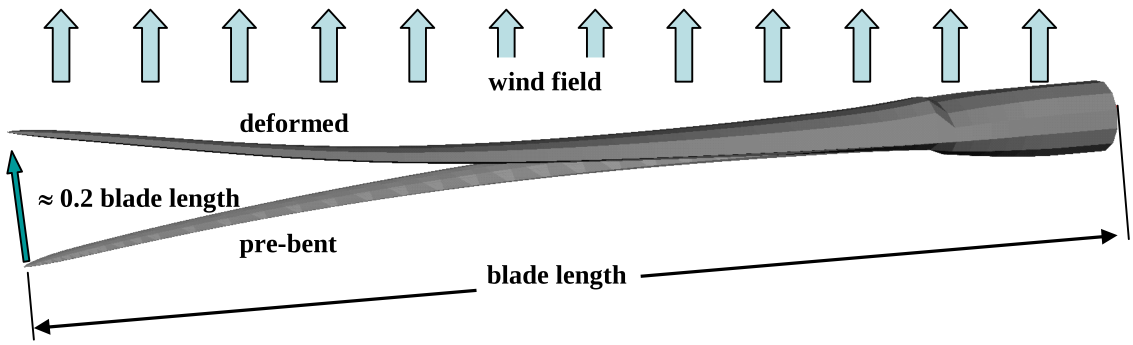

Figure 2. Wind turbine rotor blade being deformed by a constant wind field.

As a practical example the deformation of the rotor blade of a wind turbine is investigated: a 61.5 m blade of the NREL 5 MW reference wind turbine [6] is loaded with the force distribution of a wind field which is assumed to be constant. The setup is shown in figure 2. For reference, an Abaqus model of the blade has been created. Furthermore, a SIMBEAM/SIMPACK model with five linear modally treated beam segments exists for comparison. All models consist of 49 vertices and use the same material data.

One of the most important simulation results is the tilting of the blade at the vertices because in a fully coupled model this parameter will determine the aerodynamic forces that act on the blade, see [4]. Errors in the calculated tilting values of >0.05° can usually not be accepted in the design phase of a wind turbine rotor blade. Apart from the tilting the calculated frequency values and shapes of the lower eigenmodes need to match those of the real blade with high accuracy in order to avoid problems due to resonances in the completely assembled system by design. In this study, the Abaqus B31 model is taken as a reference since the B31 beam element is known to perform very well for all benchmark tests where the analytical results are known.

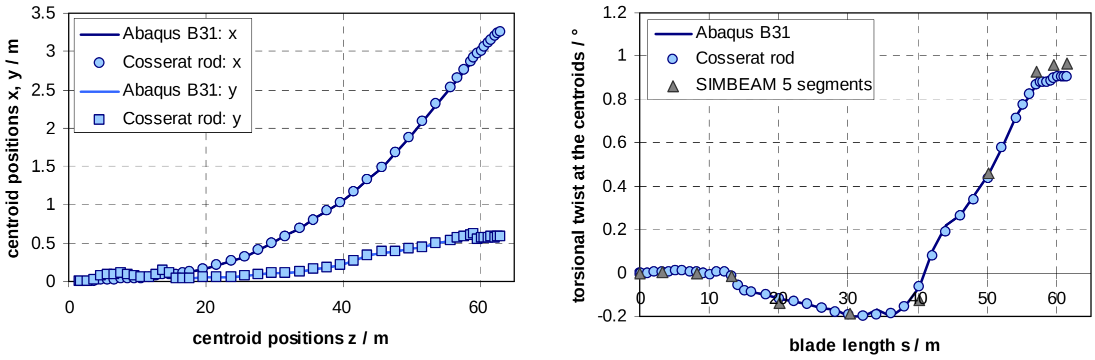

Figure 3. Static deformation of the NREL 5MW wind turbine rotor blade in the constant force distribution of a wind field: comparison of the Abaqus B31, the Cosserat rod and the segmented SIMBEAM models.

In figure 3 static deformations are compared which are obtained with the three models for the given load case. The deflections of all models match up to a deviation of ≤0.5%. In the case of the important torsional twist the deviation of the Cosserat rod model from the Abaqus B31 model is ≤0.025° for all centroids along the blade length. The segmented SIMBEAM/SIMPACK model on the other hand has a deviation of up to 0.6° which is not acceptable for fully coupled simulations of a wind turbine. Note that the segmented SIMBEAM/SIMPACK model can be improved in order to meet the desired accuracy by choosing a larger number of segments while keeping the total number of vertices constant. The Cosserat rod model, however, provides almost the same accuracy as the Abaqus B31 model without additional modelling effort.

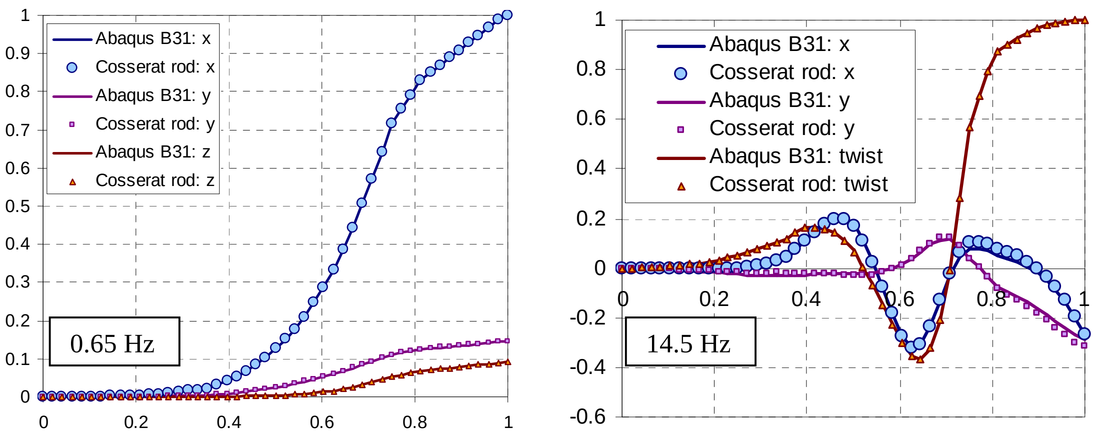

Figure 4. Comparison of the first and the twelfth eigenmodes of the NREL 5MW wind turbine rotor blade obtained with the Abaqus B31 and with the Cosserat rod models, respectively (normalized units).

Figure 4 shows a comparison of the first and the twelfth eigenshapes of the blade as they are obtained with the Abaqus B31 and with the Cosserat rod models, respectively. As can be seen, the results for the first eigenmode (first bending mode in x-direction) perfectly match. In fact, the twelfth eigenmode (third twisting mode) is the first one with notable deviations. The reason why the results of the two models are such close is inherent: the linearization of the (finite-difference) Cosserat rod model is mathematically almost equivalent to the linearization of the (finite-element) Abaqus B31 model such that the differences that are owed to the numerical solution procedures are predominant.

5.3. Wind turbine

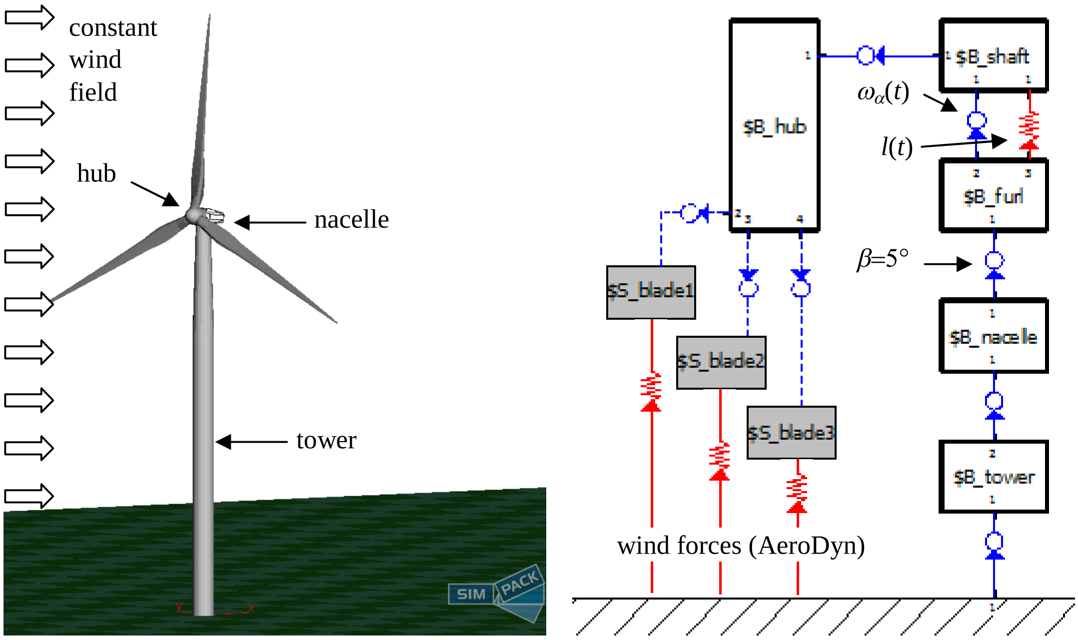

Figure 5. Simplified model of a three-bladed wind turbine.

As a last example a simplified, three-bladed wind turbine is investigated. The wind turbine consists of a tower, a nacelle for the gear box and the generator, a hub and three blades. The hub is connected to the nacelle via a shaft which is furled at a constant angle of 5° against the horizontal plane. In this study, only the three blades are modelled as Cosserat rods (with co-simulation active), the other parts are assumed to be rigid. Aerodynamic forces along the blades are calculated with the NWTC AeroDyn Software [1] for a spatially constant wind field (20m/s) at an update rate of 50Hz. A constant outer torque l is acting at the shaft to simulate a load. The aerodynamic forces and the outer torque are ramped in the time intervals [0s, 10s] and [0s, 20s], respectively, in order to reach the steady state within few rotations. An initial clockwise rotation velocity of the turbine of $\omega_\alpha = 0.5\pi\ \mathrm{s}^{-1}$ is assumed. Figure 5 contains a screenshot and a block diagram of the model in the SIMPACK pre-processor.

A dynamic simulation for the time interval [0s, 40s] is performed in order to study the deformation of the rotor blades. In the debug build of SIMPACK, a full run takes ~3h on a modern PC. Exemplary the deformation at the three blade tips perpendicular to the movement plane and the rotation velocity of the turbine are shown in figure 6 for steady state conditions. As can be seen, gravity and the shadow effect of the tower cause oscillations of the blade deformations with the rotation frequency and of the angular shaft frequency with three times the rotation frequency. The total deformations are >10% of the blade length. For such deformations, a nonlinear model like the Cosserat rod is required in order to achieve accurate results for second-order effects such as the twisting of the blades and consequently for the aerodynamic forces and finally for the efficiency of the wind turbine. A comparable accuracy may be achieved with a segmented linear-modal model for the rotor blade deformation when all modes are included in the simulation. However, the calculation time on a typical PC hardware for the complete dynamic simulation with such a model is about 10 times longer than the co-simulation with the nonlinear Cosserat rod model.

6. CONCLUSION

By transforming the EoM, nonlinear structural models can directly be integrated into the recursive MBS formalism of SIMPACK. Such a model is the discrete Cosserat rod model which can be used to model a wide range of beams and beam-like structures in industrial applications. The performance of this model is comparable with the nonlinear FE Abaqus B31 beam element.

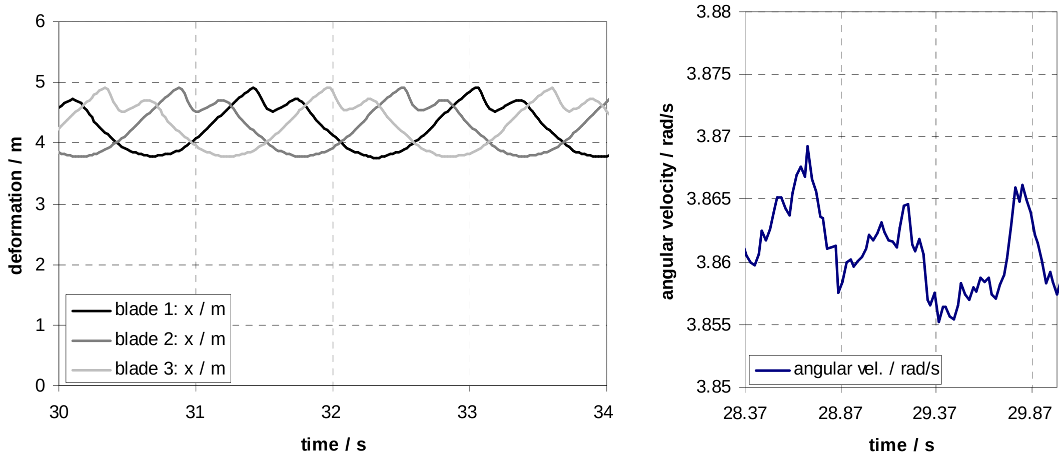

Figure 6. Steady state of the simplified wind turbine. The left diagram shows the blade deformation perpendicular to the movement plane, the right diagram the main shaft velocity for one rotation.

When integrated into SIMPACK's MBS formalism, the existing SIMPACK solvers can directly be used to determine the static and dynamic behaviour of the model. This way, e.g. the nonlinear deformation of the rotor blade of a wind turbine can be calculated up to a high precision. The computational performance of the method depends on the stiffness of the beam structures and the couplings. Co-simulation allows to perform integrated simulations of a wind turbine that out-beat the performance of the existing methods in SIMPACK both w.r.t. to the accuracy of the results and the computational performance.

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