Modeling of Elastic Cages in the Rolling Bearing Multi-Body Tool CABA3D

Dmitry Vlasenko(✉) and Bodo Hahn(✉)

Schaeffler Technologies AG & Co. KG, Industriestr. 1-3,
91074 Herzogenaurach, Germany
{vlasedit,hahnbdo}@schaeffler.com

Abstract. The paper is concerned with the dynamical simulation of rolling bearings with elastic cages in the simulation software CABA3D (Computer Aided Bearing Analyzer 3D). The modeling process (import of finite element models, model reduction, model verification) is considered in detail. Two different contact simulation methods (slice contact and node-to-surface contact approaches) are shown and compared.

1 Introduction

Simulation of rolling bearing dynamics is extremely helpful for the understanding of processes acting in bearings. The information about kinematics, load distribution and friction forces of bearing elements (rings, cages, rolling elements, etc.) provided by simulation software is needed for the cost-effective optimization of bearing design as well as prediction and prevention of bearing failures.

Nowadays CABA3D (Computer Aided Bearing Analyzer 3D) developed by Schaeffler Technologies is one of the most powerful industrial software tools for the simulation of dynamic processes in rolling bearings. CABA3D, in contrast to commercial, general-purpose multi-body software, has elasto-hydrodynamic friction and contact models specially designed for rolling bearings [2].

Lightweight cages are widely used in distinct types of rolling bearings. The advantages of the low cage weight include the reduction of inertial and centrifugal forces while maximizing the lubricant effectiveness. Often, lightweight cages cannot be simulated as rigid bodies since the cages' elasticity has considerable influence on the dynamic behavior. Investigation of many effects like deformation of cage pockets, cage-instability, the response on impacts or stresses and fatigue is only possible if the deformation of the cages is considered.

2 Model Order Reduction Techniques in CABA3D

Finite element (FE)-models of elastic cages usually have several thousand degrees of freedom and extremely high eigenfrequencies. This entails the need for reduction of the models before the model dynamics is simulated.

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Since the deformations of cages are small compared to their rigid body motion in rotational and translational direction, the most suited reduction methods in this case are methods based on the floating frame formulation [7, 8, 11]. According to the method, the total motion of an elastic body is divided into two parts: rigid body motion represented by the motion of the body reference frame and deformations with respect to this frame. In this method, the position d^i of the i-th node in the global inertial frame over time t is expressed as

$$\mathbf{d}^i(t) = \mathbf{x}(t) + \mathbf{A}(t) \cdot \left(\bar{\mathbf{r}}_0^i + \bar{\mathbf{u}}^i(t)\right) \tag{1}$$

where $\mathbf{x}$ denotes a position for the origin of the body reference frame in the global coordinate system, $\mathbf{A}$ is a body rotation matrix, $\bar{\mathbf{r}}_0$ is the undeformed local position of the node in the body frame, and $\bar{\mathbf{u}}^i$ is the local nodal displacement.

The model reduction techniques assume that the deformations vector $\bar{\mathbf{u}} = \left(\bar{\mathbf{u}}^1 \quad \ldots \quad \bar{\mathbf{u}}^N\right)$ can be approximated by linear combinations of several generalized deformation modes (Eq. 2).

$$\bar{\mathbf{u}} = \mathbf{S} \cdot \mathbf{q} \tag{2}$$

S is the modal matrix, consisting on deformation modes, and **q** is the vector of modal coordinates. It is assumed that the count of modes is much less than the count of nodes in the FE-model, e.g. the size of **q** is much less than the size of $\bar{\mathbf{u}}$.

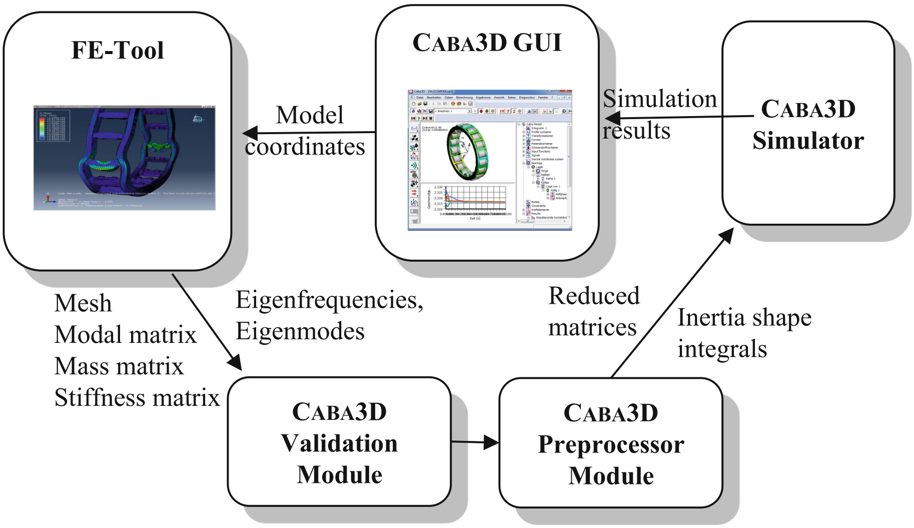

Fig. 1. Simulation scheme of elastic cages

The advantages of reduced models are low computational cost and small simulation results file size compared to FE simulations. At the same time, reduced models can also create an accurate simulation of macro- and micro-elasticity of cages (e.g. ovalization of cages, deformation contacting faces, etc.) [10].

The standard methods of modal reduction (e.g. Craig-Bampton [3]), are already implemented in FEA tools (e.g. ABAQUS, ANSYS) and can be used for the simulation of elastic cages in CABA3D.

The simulation scheme of elastic cages is shown in Fig. 1:

1. On the first stage modal matrix, mass matrix and stiffness matrix, together with the eigenfrequencies and eigenmodes, are imported from the FE-tool to the CABA3D Validation Module where the quality of model reduction is validated using different methods: Normalized Relative Eigenfrequency Difference (NRED), Modal Assurance Criterion (MAC), comparison of frequency response, etc. [5, 7].
2. If the quality of the reduction is sufficient, the next modeling step is started. The model parameters (modal matrix, mass matrix, stiffness matrix, mesh data) are imported to the CABA3D Preprocessor Module. The module generates the reduced mass, stiffness and damping matrices together with the time-constant inertia shape integrals of the reduced model [7, 11].
3. During the third step, the dynamics of a bearing model including the elastic cage is simulated over time. The simulation results (coordinates, velocities, forces, pressures, etc.) are calculated.
4. The simulation results are analyzed in CABA3D GUI. Using motion data, application engineers generate 2D and 3D diagrams showing time changes of needed parameters (forces, deformations, etc.) or make the animation of bearing motion.
5. The time history of modal coordinates can also be transferred back to the FE-tool to calculate stresses in the material of the cage.

3 Contact Modeling of Elastic Elements in CABA3D

Cages in rolling bearings have multiple contacts with rolling elements, rings and other cages. Obviously, the cage deformation has an influence on the location of contact and on the geometry of contact areas. Depending on the desired accuracy and performance of the simulation, different strategies can be used.

3.1 Slice Contact Model

If the local deformations of contact areas are neglectable we can use the slice model contact method that has already been successfully applied in rigid bearing dynamics [2, 4, 9]. In this approach, the contact faces are virtually treated as discrete slices (circular or polygonal) and analytical surfaces, as it is shown in Fig. 2. To find the intersection points between slices and surfaces the analytical root finding algorithm is used [9]. The contact forces and torques acting on contacting bodies are calculated as a sum of forces and torques acting on each slice. The method is accurate and numerically efficient in the case of simple geometries (e.g. cylinder, torus).

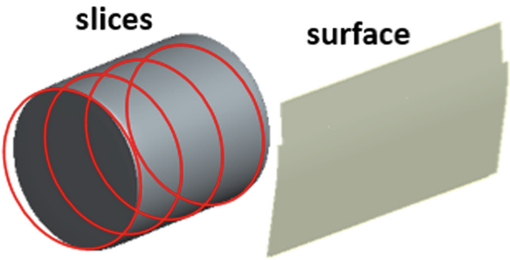

Fig. 2. Slice contact model

To account for the influence of elastic deformations of contact areas, the distributed coupling algorithm [1] can be used. In this algorithm, a contact surface is connected to a group of coupling FE-nodes, as it is shown in Fig. 3.

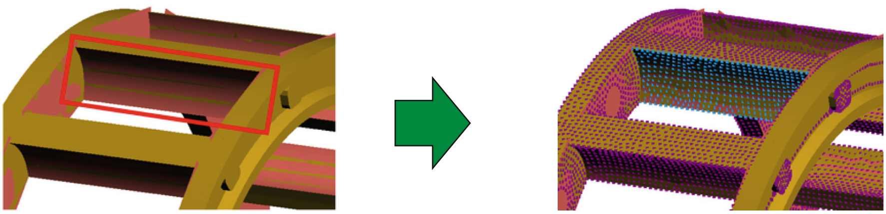

Fig. 3. Coupling of a contact area with FE-Nodes

The position and the velocity of the area coordinate system with respect to the body frame are calculated from the displacements of coupling nodes. The contact forces and moments acting on the surface are distributed at each coupling node.

3.2 Node-to-Surface Contact

The node-to-surface contact model is used if local deformations are important. In this case, every FE-node on the surface of the cage is considered and checked whether it has contact with other bearing elements, as it is shown in Fig. 4.

The distance from FE-nodes to the rigid surface of the contact partner is calculated analytically [6]. In comparison to the slice contact model, this approach is more accurate and can be implemented for all surface types. The disadvantage of the method is a higher computational cost because of the big number of nodes.

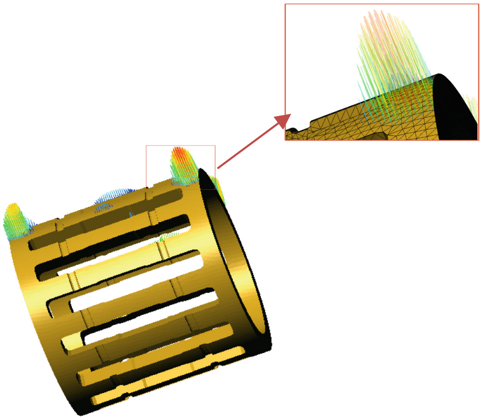

Fig. 4. Pressure distribution in the node-face contact

4 Simulation Results

4.1 Validation

One important step before the models are used in real applications is validation. In the following, two examples are presented.

The first one shows a comparison of the deformation and stress calculation using CABA3D with the results of a finite element analysis. A simple static load case is used for this comparison. Figure 5 shows very good correlation for both stresses and deformation. Even stress concentrations are calculated accurately which is important for a potential strength assessment in a postprocessing step.

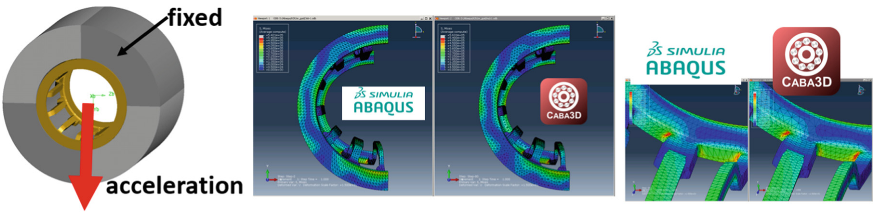

Fig. 5. Comparison of stresses and deformations

The second example (Fig. 6) shows the contact pressure distribution between the cage and the outer ring due to constant acceleration. It can be seen clearly that the slice model cannot represent such a complex shape. Some contact points are not even located on the cage. However, if we look at the node-to-surface model the pressure distribution is almost identical to the finite element analysis.

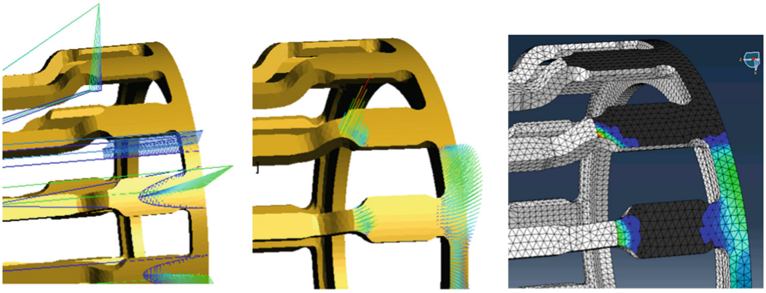

Fig. 6. Contact pressure: slice model, node-to-surface model, FE analysis

4.2 Application Example

Figure 7 shows a common application example of the elastic cage model: A two-stroke crank drive that contains four bearings. Especially the cages at the crank pin and at the piston pin experience high loads due to acceleration. These two cages are modeled with the described elastic cage model. All contacts between rollers and cages are modeled with the slice model due to simple cage pocket shapes. The contacts between cages and their outer guidance surface use the node-to-surface model.

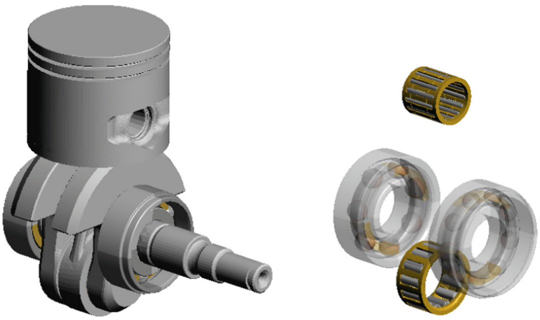

Fig. 7. Crank drive model with four bearings

Figure 8 depicts one example that shows that the modeling of the cage can have a significant influence on the simulation results. The diagram shows the forces between the cage and the guidance surface at the conrod. An impact occurs due to the contact alteration of the cage between the top and bottom dead center of the piston. When using a rigid cage model, the forces would be highly overestimated during the impact phase. However, there is good agreement between the two models during the time span with constant acceleration.

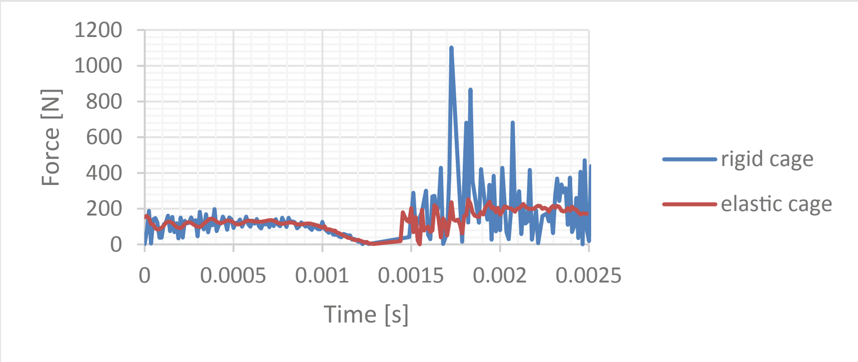

Fig. 8. Contact force: cage to conrod

5 Conclusions

In this article, the modeling of elastic cages in the rolling bearing multi-body tool CABA3D is described. It is shown that the Craig-Bampton reduction method is well suited for the simulation of cage dynamics. The advantages of the approach are the minor computational costs, small simulation results file size and accurate simulation of elasticity effects.

Depending on the desired accuracy and performance of the simulation, different strategies for the contact calculation can be used. The first alternative is the slice-to-face contact model, where contact areas are virtually divided into slices. The cage deformations influence the position and orientation contact surfaces, since they are connected to FE-nodes using distributed couplings. The local deformations of contact areas are neglected.

In contrary, if local deformations of contacting faces are important, the node-to-surface contact model can be used. In comparison to the slice model, this approach is more accurate but is more numerically expensive because of the high number of nodes.

Reduction strategies and contact models are validated with high-level FE methods. The example of the crank drive shows that the developed elastic cage model can be used for practical applications. It enables the development and improvement of new cage designs.

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