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DYNAMIC SIMULATION OF THE INTERACTION BETWEEN RACEWAY AND RIB CONTACT OF CYLINDRICAL ROLLER BEARINGS

TRACK OR CATEGORY

Rolling Element Bearings

AUTHORS AND INSTITUTIONS

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INTRODUCTION

The dynamic simulation of roller bearings gives a detailed insight into the behavior of the different contacts in the bearing. It is the tool of choice when targeting cage instability, slip or vibrations. Current models of the Institute of Machine Elements, Gears, and Transmissions (MEGT) of the University of Kaiserslautern include discretized contact calculations for raceway and rib contacts. The detailed pressure and asperity load ratio calculation is based on the contact solvers developed for the tribosimulation at the MEGT within the Collaborative Research Centre SFB926.

This contribution focuses on the modeling of mixed friction in raceway and rib contact. A model of a combined loaded cylindrical roller bearing is introduced and used for the investigation of the interaction between raceway and rib contact which is responsible, among other things, for the rolling element skew, the lubrication conditions in both contacts and additionally the overall losses in the bearing. The study is closed with a validation of friction torque, measured on a specialized test rig at different operation conditions.

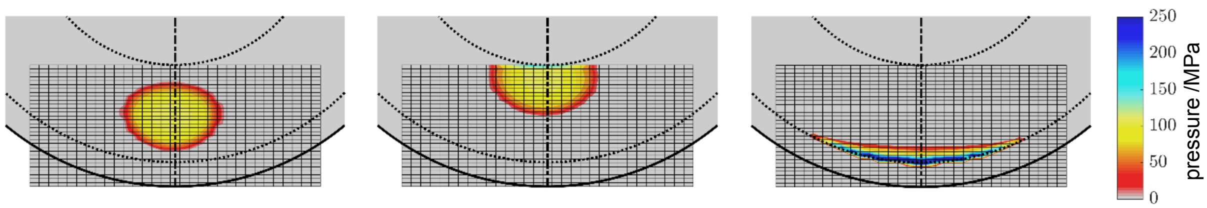
DYNAMIC SIMULATION OF ROLLING BEARINGS

The dynamic simulation offers great benefit over other methods when investigating the behavior of roller bearings. A recent review on rolling bearing modeling including dynamic simulation was given by GUPTA [1]. Quasi-static calculations are very fast and helpful for the determination of the bearing lifetime. ISO 15243 [2] however mentions six major failure mode groups, each with several different sub-groups. The bearing lifetime calculation according to ISO/TS 16281 [3] only accounts for one sub-group, the sub-surface material fatigue. For many other failure modes, the dynamic simulation is the method of choice to approach the prediction of bearing damage, e.g. cage instability or slip (e.g. [4,5]). Furthermore, the interactions between bearing elements and surrounding parts are very important when determining the actual loads and conditions for the calculations. Especially for large bearings or flexible housings, the elastic deformation of the bearings rings and raceways can change the load distribution and kinematics of the bearing significantly [6,7].

There are numerous different approaches and simulation models for various questions of the dynamic behavior of bearings. The MEGT has been developing such simulation models for over 10 years [8–11]. These models, based on commercial multibody simulation tools (MSC.ADAMS, Simpack), were extended with self-developed calculation routines for the numerical description of the contact. They feature detailed contact and friction calculations for the different contact locations. As the friction mainly determines the dynamic of a bearing, its calculation is one of the key aspects in the modeling process. This includes the description of the contact conditions (pressure, temperature and velocities), the lubricant properties and the surfaces topologies.

CONTACT MODELS

In general, the first step is the calculation of the contact situation based on the contact geometries and the body positions. The results (penetration, contact normal and contact point) are calculated for every single contact. As the contacts between rollers and raceways as well as rollers and rib usually cannot be described as Hertzian contacts, discretized calculation models are used to achieve an accurate calculation of the contact conditions. The raceway contact is modeled using the Advanced Slicing Technique (AST) developed by TEUTSCH [8] and improved by AUL [12], which divides the roller into several slices. The contact calculations are then carried out for every slice. In contrast to conventional slice models, like in ISO/TS 16281 [3], which calculate every slice individually, AST takes into account the influence of all slices based on the half-space theory. It can therefore predict edge-stresses, for example. For the roller end-rib contact a cell model is used to calculate the pressure distribution for arbitrary roller end and rib geometries. The pressure calculation is based on methods of the MEGT tribosimulation [13] and optimized for the usage in the dynamic bearing simulation. This optimization focuses on numerical efficiency, because the pressure distribution of each contact needs to be calculated more than 10^5 times for a second of simulated time. The optimized contact solver uses a conjugate gradient solver based on [14] and includes warm start capabilities to decrease the number of iterations needed [15]. As a result, computing times of less than one millisecond are achieved for a contact discretized with more than 1000 cells. Figure 1 shows exemplary results for the pressure distributions in the roller end-rib contact calculated with a dynamic simulation model of a cylindrical roller bearing. The three examples show the influence of small variations in roller end of rib geometry. An optimized contact geometry pairing is illustrated in Figure 1a. It results in a contact area in the middle of roller end and rib. A small change in the rib angle leads to a contact area on the edge of the rib and edge stresses (see Figure 1b). Both previous examples had a profiled roller end. A flat roller end with edge radius leads to significant higher contact pressures due to very small contact areas and non-conformal contacts (see Figure 1c).



a) Optimized contact pairing with profiled roller end *b) Poor contact pairing with profiled roller end* *c) Flat roller end with edge radius*

Figure 1: Pressure distributions in the roller end-rib contact calculated in the MBS model of a cylindrical roller bearing NJ2210 (radial load F_{radial} = 17,6 kN, axial load F_{axial} = 1,76 kN, shaft speed n = 500 min^{-1}), discretization: 32 × 32 cells

FRICTION CALCULATION

As the friction determines the actual kinematics respectively the dynamics of the bearing, the simulation models also include detailed friction models based on current research on elastohydrodynamic lubrication (EHL) [16]. This includes the calculation of solid and lubricant friction weighted with the mixed friction model from ZHOU and HOEPRICH [17]. For the correct modeling of the dynamic behavior, the friction calculation is divided into two different aspects: traction and losses. The latter include material hysteresis, rolling resistance as well as EHL rolling friction. The contact traction is calculated with dry friction and the lubricant shears stresses.

The mixed friction model requires the calculation of the ratio of asperity load and finally weights solid and lubricant friction accordingly. The model from ZHOU and HOEPRICH uses two parameters B and C for the determination of the asperity load share ϕ:

$$\phi = \frac{N_F}{N} = e^{-B \cdot \Lambda^C}$$

Where N is the contact load, N_F the asperity load and Λ the ratio of film thickness to composite surface roughness. The authors provide the values for the parameters for three different surfaces. The real contact surfaces, however, differ from these standard surfaces and can even change during operation time. Therefore, the tribosimulation model was extended to include elastic-plastic material behavior. This allows for the calculation of the asperity load share ratio for measured surface topologies [18]. The surface topologies were measured with a 3D confocal microscope at the Institute for Measurement and Sensor-Technology of the University of Kaiserslautern. The calculation process is shown in Figure 2 . It results in individual values for the mixed friction parameters for each contact.

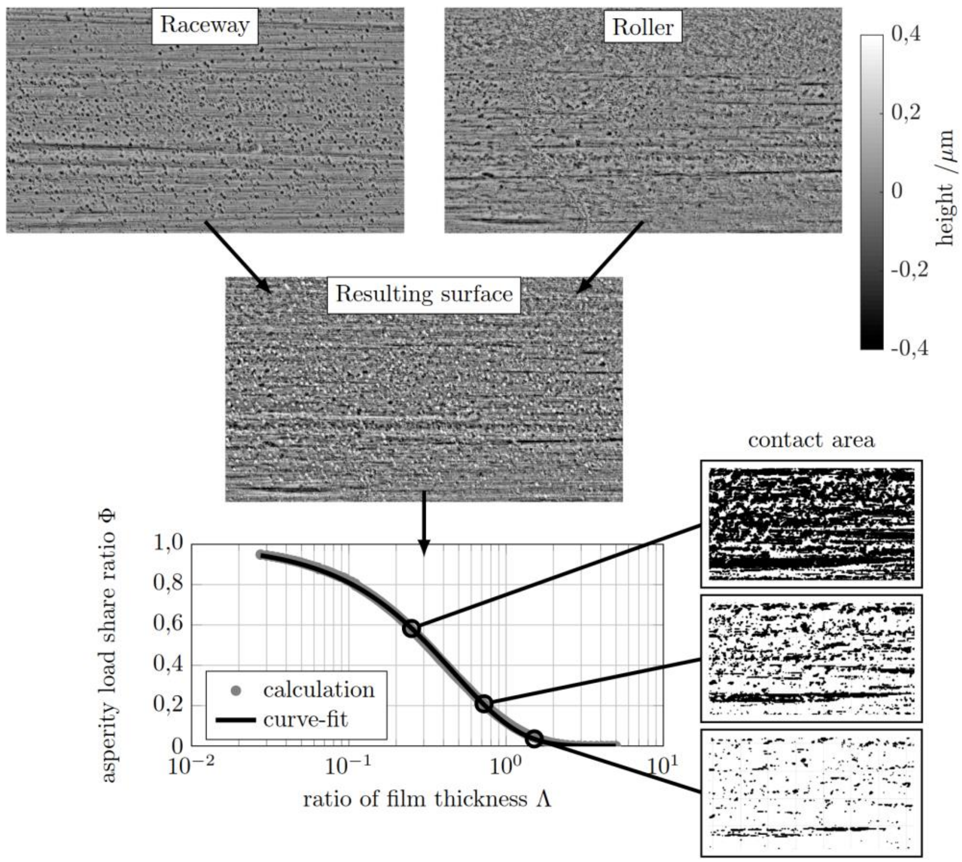


Figure 2: Determination of the mixed lubrication parameters according to ZHOU and HOEPRICH [17] based on measured surface topology (method from [18], example: inner race and roller from a CRB NJ216)

VALIDATION

The simulation models were validated using established static calculation models for the load distribution. For the bearing dynamic and friction, extensive experimental data was used. The friction torque of a rolling bearing is a suitable validation measure, as it includes the losses in every single contact in the bearing. Therefore, it allows the integral validation of contact and friction modeling. For that purpose, a self-made friction torque test rig was used [11].

Exemplary results are shown in Figure 3 for a combined radially-axially loaded cylindrical roller bearing (NJ 216). Radial load, temperature and lubricant were kept constant. Shaft speed and axial load were varied. For the radially loaded conditions, the friction torque increases with increasing shaft speed mainly because of higher rolling resistance. Since there is no axial load, only the raceway contacts significantly contribute to the bearing friction. This is why these conditions can be used for the validation of the raceway contact modeling. An additional axial load leads to higher friction torques as the roller end-rib contact also carries load and therefore produces friction losses. At higher speeds, the increase in friction is comparatively small, because the rib contact is in the full fluid film lubrication regime. With decreasing speed, the amount of asperity load increases, which leads to higher friction forces and increasing overall friction torque. This can be seen in the measured as well as simulated results at speeds below 1500 min^{-1}. The model predicts the friction torque increase very good over the range of shaft speeds, which indicates a very good model quality for the roller rib contact.

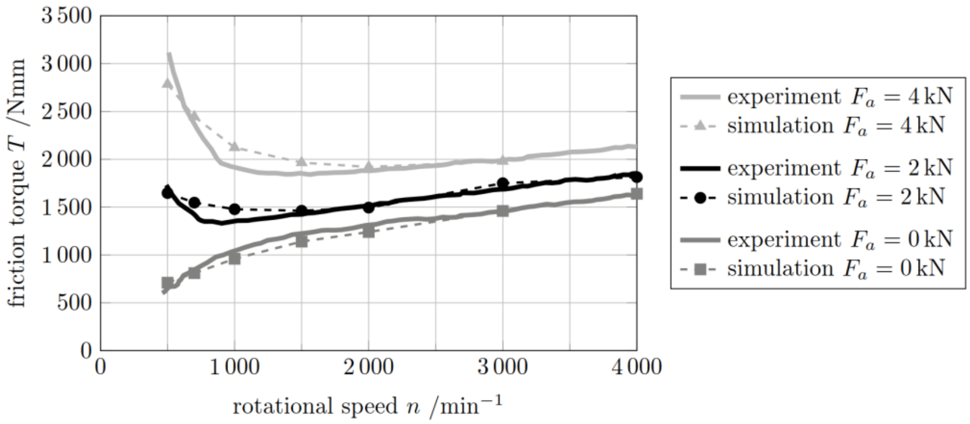


Figure 3: Comparison of simulated and measured friction torques for different speeds and axial loads (constant radial load F_{radial} = 20 kN and outer ring temperature T = 50°C, mineral oil ISO VG 100)

CONCLUTION

The dynamic simulation of rolling bearings is a very capable tool for the investigation of rolling bearing behavior and the prediction of possible failures. This contribution presents a modeling method based on commercial MBS tools with self-developed contact calculation routines. This concept allows for a very detailed modeling of the different contacts in the bearing. A new calculation approach is introduced to include precise pressure calculations for arbitrary contact geometries, which extends the capabilities of the existing models, e.g. to the simulation of combined loaded cylindrical roller bearings including roller end-rib contact. The consideration of mixed friction effects is achieved with an established mixed friction model. The required parameters are calculated based on real measured surface topologies. An exemplary comparison with experimental results shows a very good agreement between model and experiment in the full fluid film lubrication and mixed friction regime for both, raceway and rib contacts.

ACKNOWLEDGMENTS

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KEYWORDS
Rolling Bearings: Cylindrical Roller Bearings
Contacts: Contact Mechanics
EHL: EHL (General)

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Institute of Machine Elements, Gears, and Transmissions
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Jun. Prof. Dr.-Ing. B. Magyar

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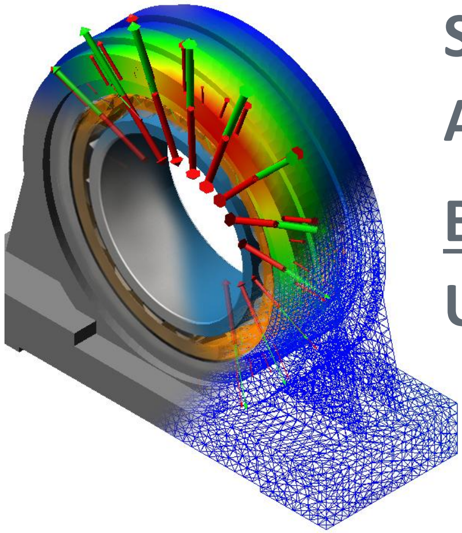
Dynamic Simulation of the Interaction between Raceway and Rib Contact of Cylindrical Roller Bearings



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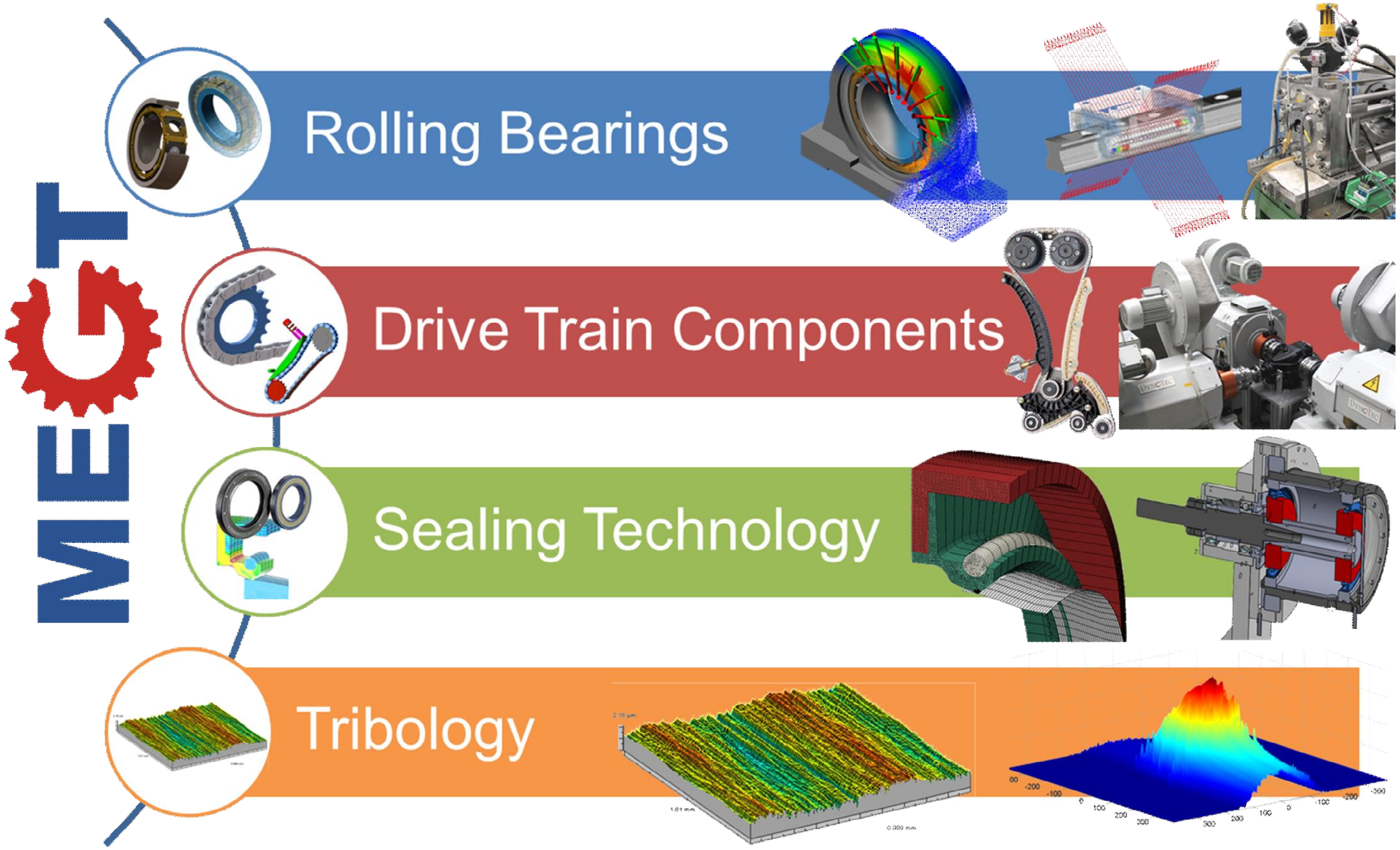

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- **Multi-Body-Simulation Model**
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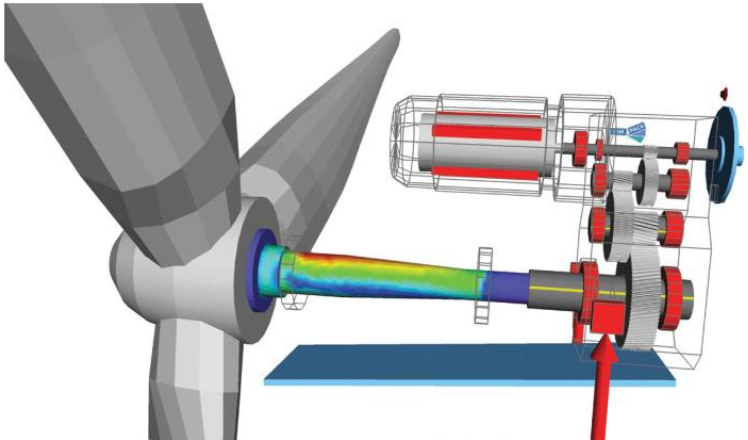
  - Contact Solver
  - Friction Calculation
- **Interaction Raceway-Rib Contact**
- **Validation**
- **Summary & Outlook**

Rolling Bearing

- Guidance and transmission of forces between moving parts
- Rolling elements between bearing rings

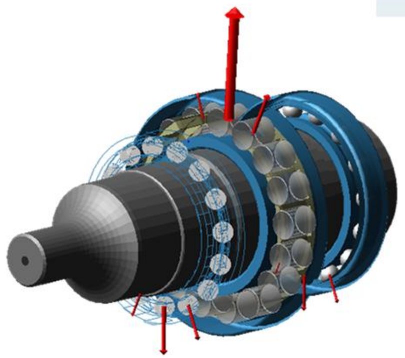
Dynamic Simulation

- Numerical method for the investigation of the dynamic behavior of systems



KOMAI TEKKO Inc: Wind turbine KWT300

Goals of Dynamic Sim. of Rolling Bearings

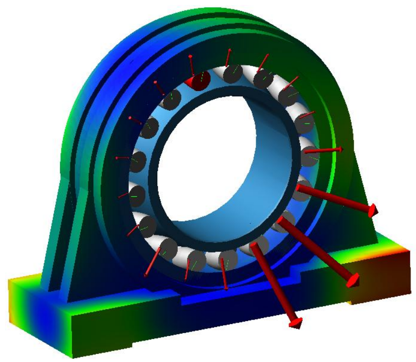


Failure modes

- Fatigue
  - Sub-surface
  - Surface
- Wear
- Corrosion
- Electrical erosion
- Plastic deformation
- Fracture and cracking

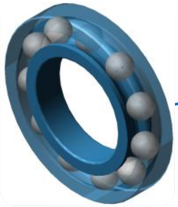
Calculation according to ISO/TS 16281

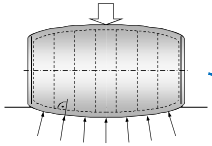
Prediction with dynamic simulation

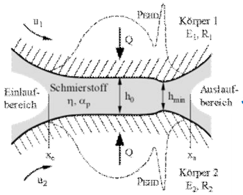


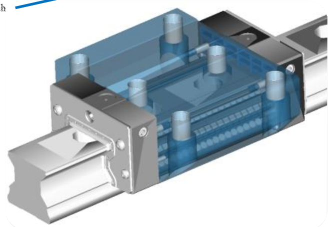
+ **Bearing loads from system**
+ **Interactions of bearings and surrounding parts**

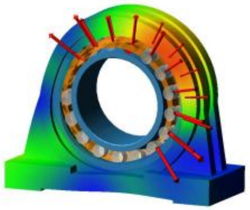
Milestones in Rolling Bearing MBS at MEGT











2000 **Start of the dynamic simulation of rolling bearings (ADAMS)**
- 2½D technique (dummy bodies)
- Ball bearings, Cylindrical roller bearings

2001 **Cage modeling**

2003 **CRB with Alternative Slicing Technique (incl. edge stress calculation)**

2004 **Extention of friction calculation: EHL, mixed lubrication**

2006 **Ball & roller rail guide**

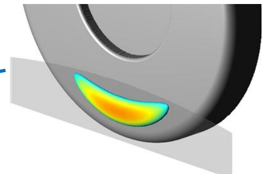
2009 **Major model updates**
- Optimization of calculation methods
- Multiprocessor capability
- 3D contact calculation

2010 **Full complement CRB, spherical roller bearing**

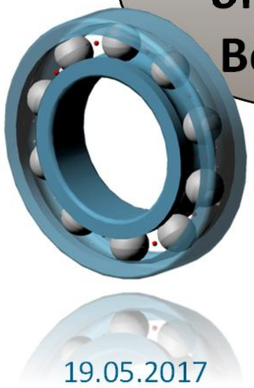
2011 **Transferring to SIMPACK**

2014 **Elastic structures**

2016 **Integration of contact solver**



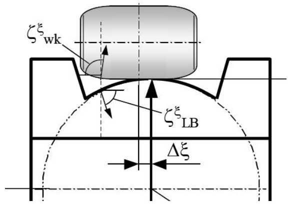
Main tasks



Dynamic Simulation of Rolling Bearings

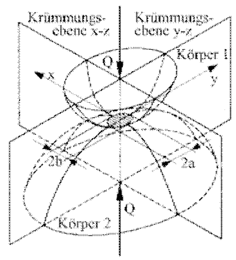
Contact

- Stability
- Accuracy
- Performance



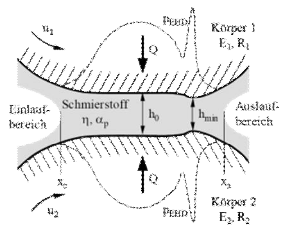
Normal force

- Load Distribution
- Contact Pressure
- Damping

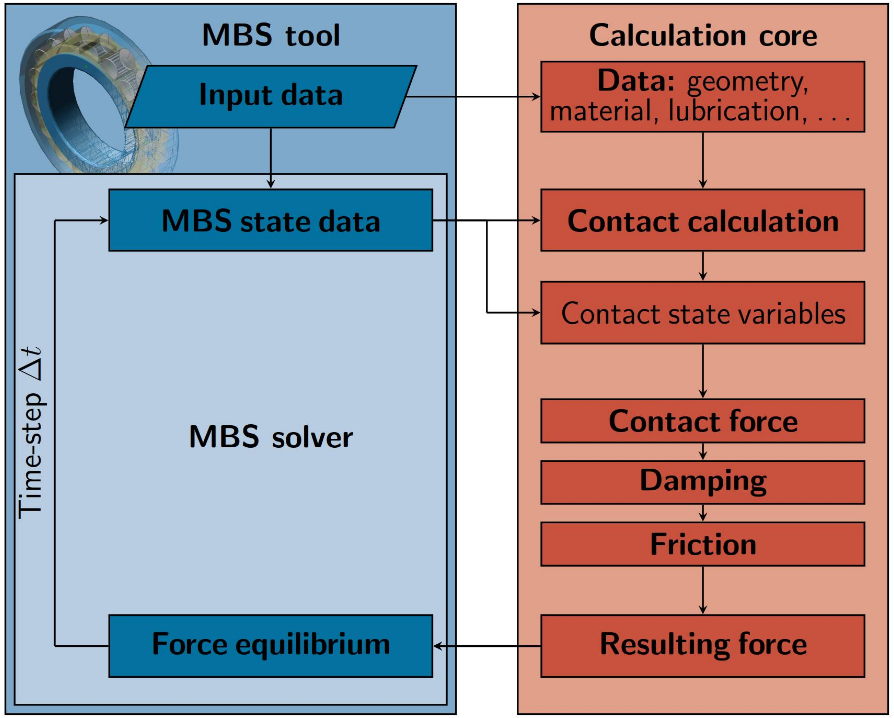


Friction

- EHL
- Lubricant
- Mixed Friction

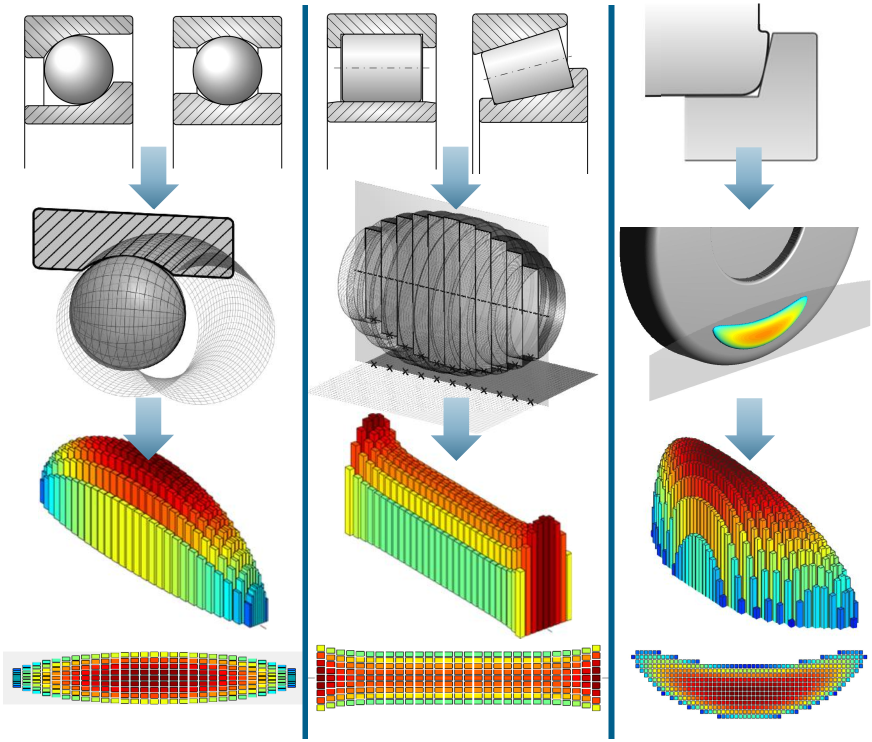


Model Structure – Overview



- **Model generation in MSC Adams with script language**
- **Calculation of all contact forces with self-developed routines (Fortran)**
- **Set up and solution of the equations of motion in the MBS solver**

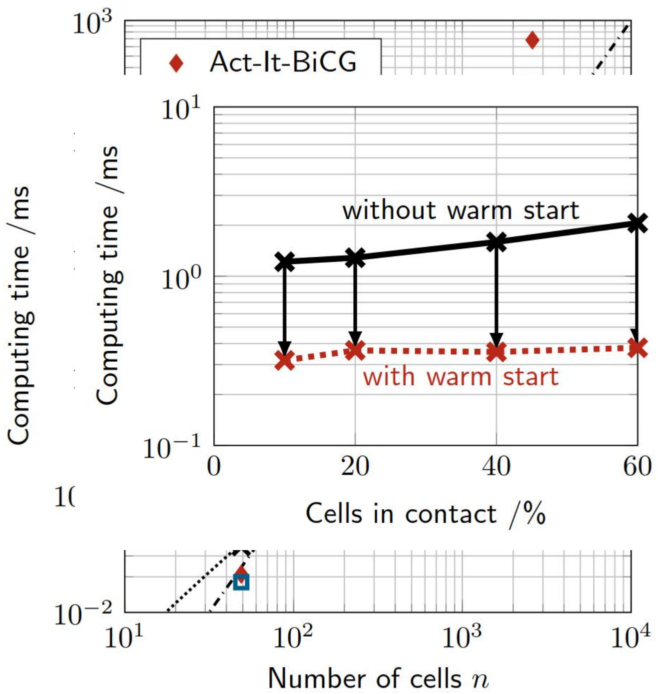
Contact models – Discretization



Division based on contact geometry
- Point contact (e.g. ball-race)
- Line contact (e.g. roller-race)
- Arbitrary contact (e.g. roller end-rib)

Calculation result:
- 2D-discretized pressure distribution
- Consistant input data for the friction calculation:
  - Local pressure
  - Local velocities
  - …

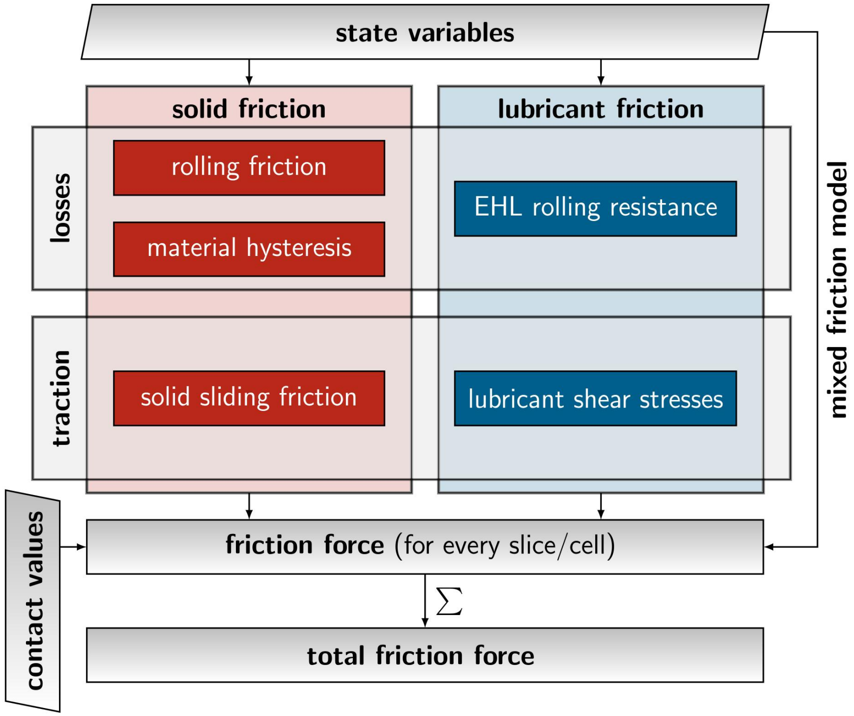
Contact models – Contact solver



- **Contact solver for the calculation of pressure distribution for arbitrary contact geometries**
- **Conflicting goals: accuracy ↔ computing time**
- **Goal**
  - **Computing time per contact ≤ 1ms**
- **Solution**
  - **State of the art solution methods**
  - **Usage of warm start capabilities**

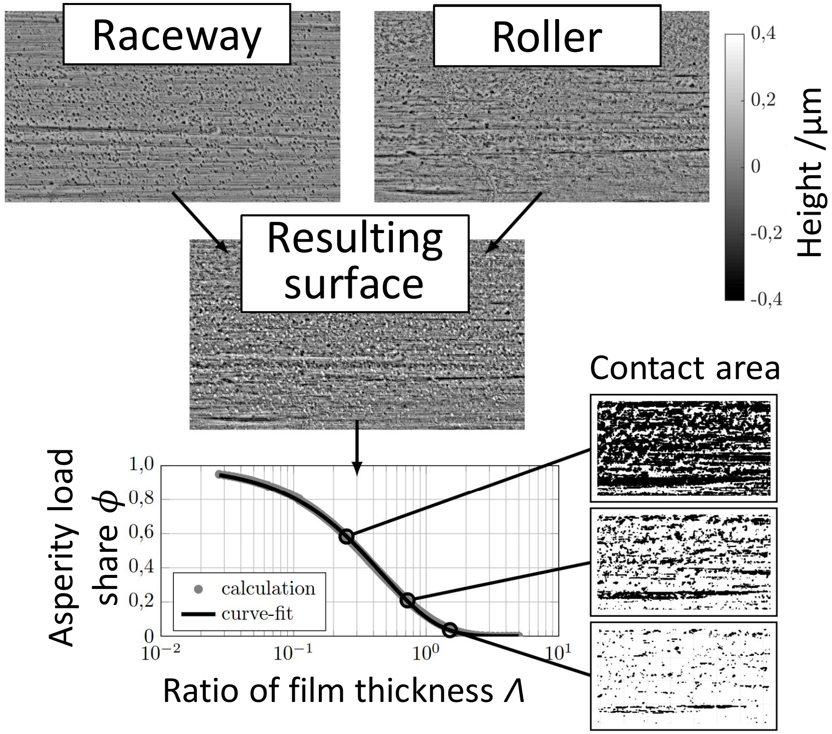
- **Result: 0,5 ms for approx. 1000 cells**

Friction Calculation – Overview



- **Traction**
  - Solid sliding friction
  - Lubricant shear stresses
- **Losses**
  - Rolling friction
  - Material hysteresis
  - EHL rolling resistance
- **Mixed friction model**

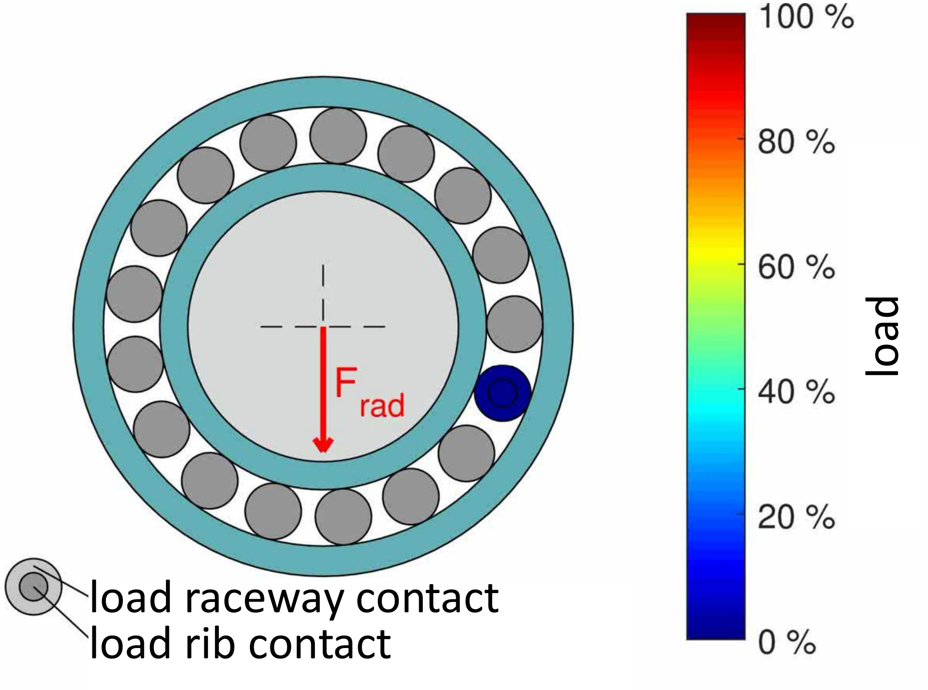
Friction Calculation – Mixed friction model



- **Measured surface topology**
- **Calculation of asperity load for different approaches of the surfaces**
- **Approximation of asperity load share with mixed friction model from Zhou/Hoeprich:**

$$\phi = \frac{N_{Asperity}}{N_{Contact}} = e^{-B \cdot \Lambda^{C}}$$

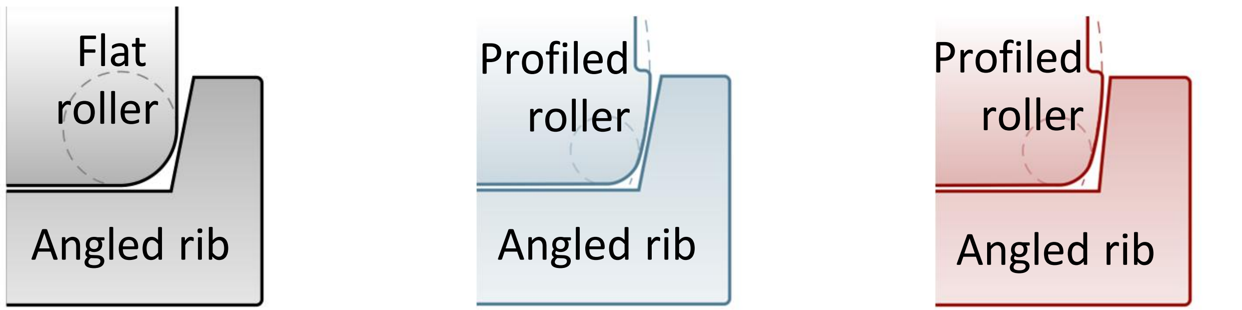
Simulation – Raceway and rib forces



- **Axial (rib) load only in combination with radial roller load**
- **No axial load on roller outside load zone due to tilting (bearing clearance)**

Simulation model data

- **Bearing type: CRB NJ216**
- **Roller and raceway geometry and topology from measurements**
- **Three different roller end and rib geometries:**

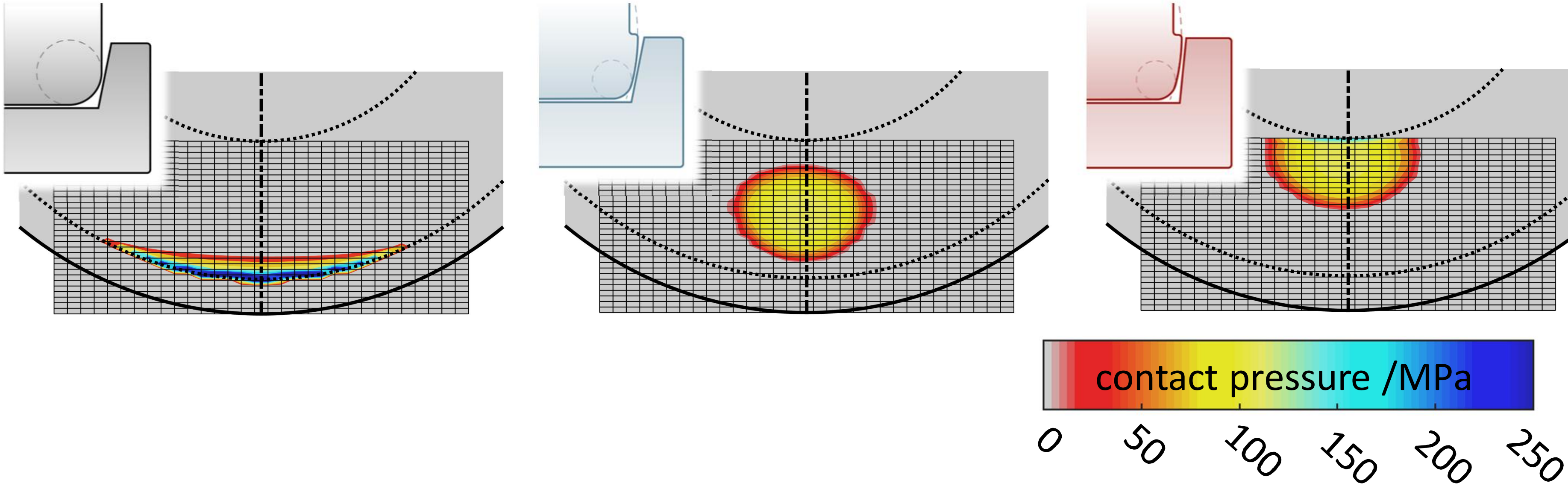


Optimized geometry pairing

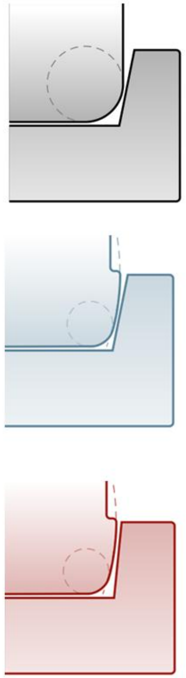
Poor geometry pairing

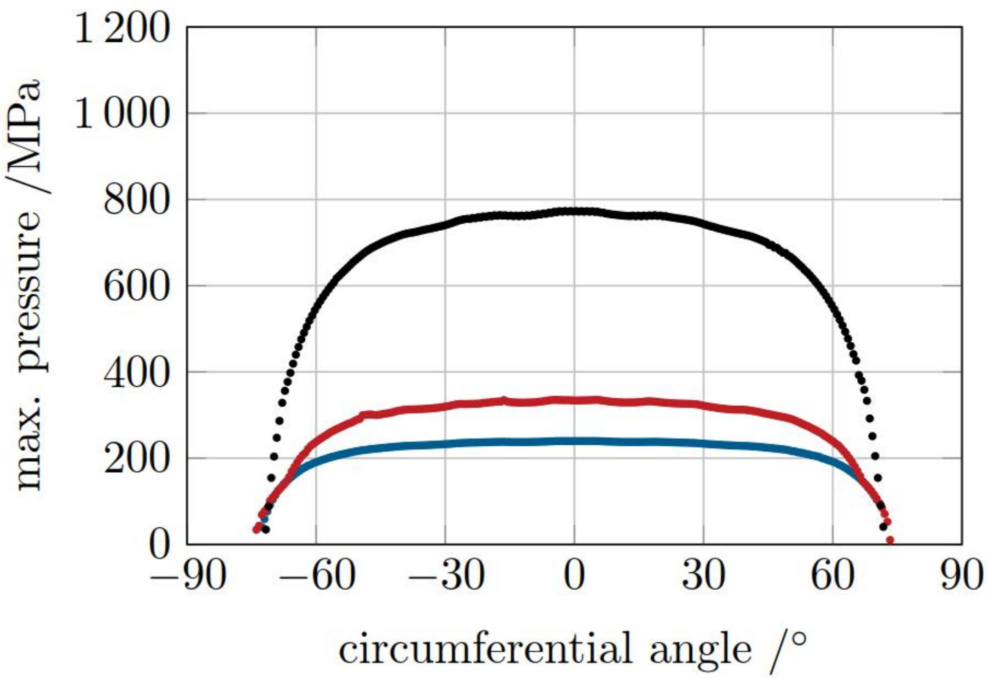
Simulation – Contact Pressure Roller End-Rib

- **Large influence of contact geometry pairing on contact pressure**



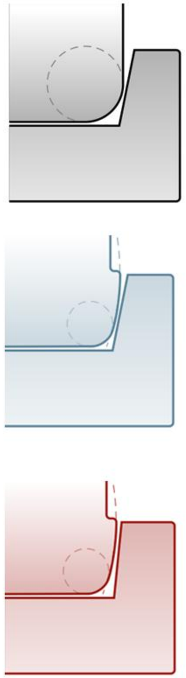
Simulation – Interaction Raceway-Rib

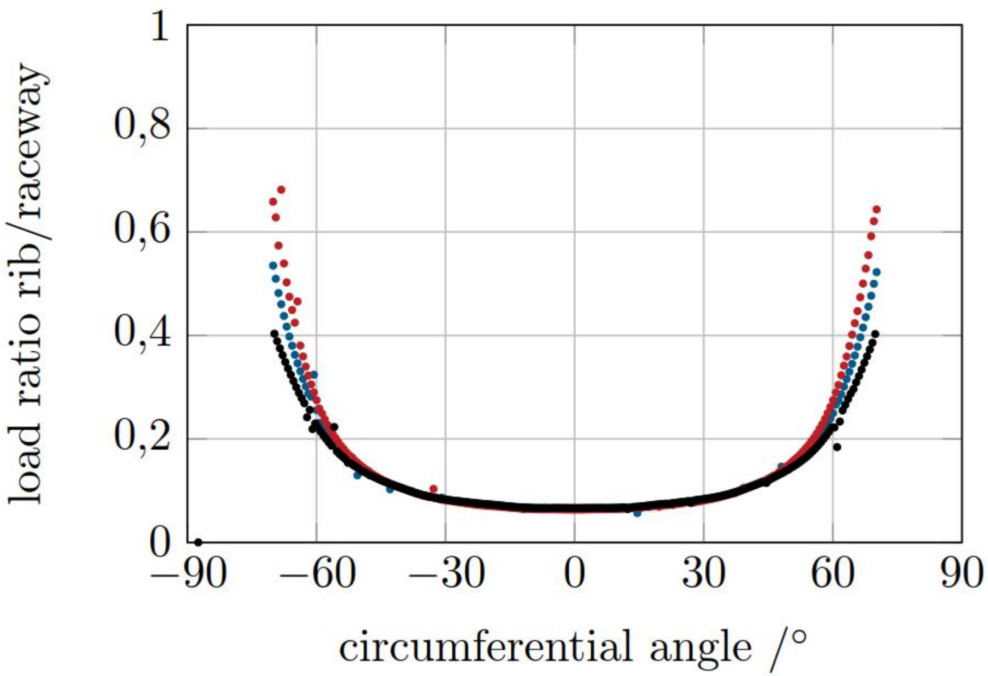




- **Load zone approx. 140°**
- **Small pressure changes over the larger part of load zone (approx. 100°)**

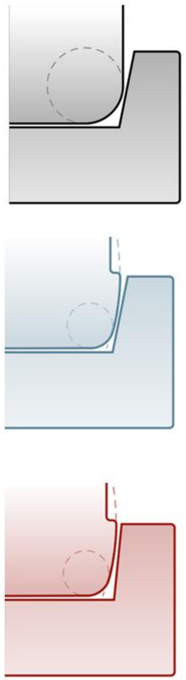
Simulation – Interaction Raceway-Rib

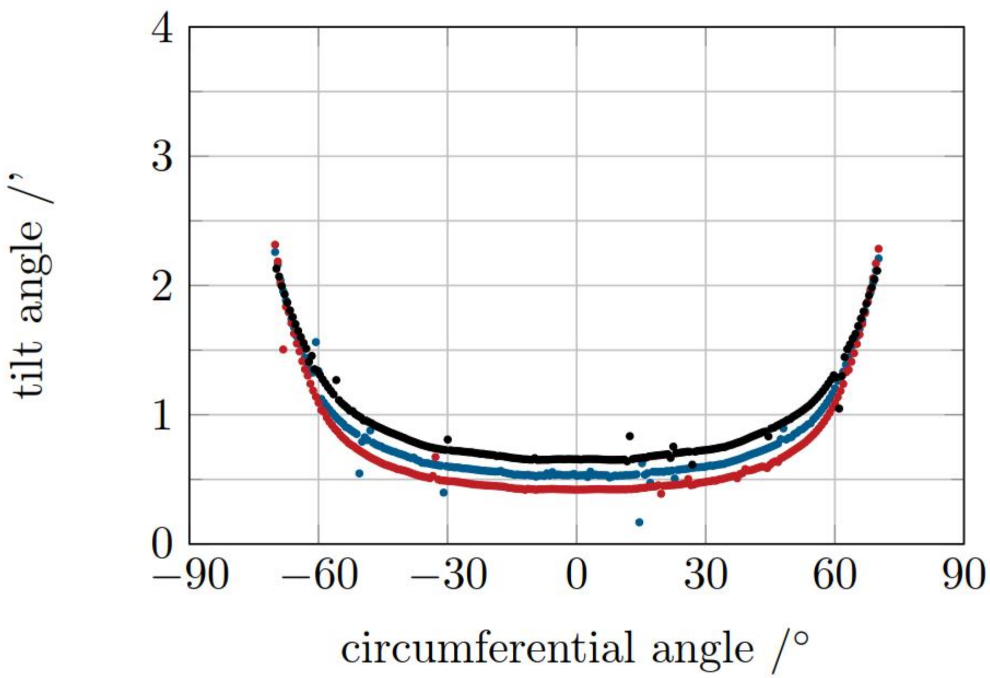




- **Entry and leaving of load zone:**
  - **Geometry influence**
  - **Large rib/raceway load ratio**

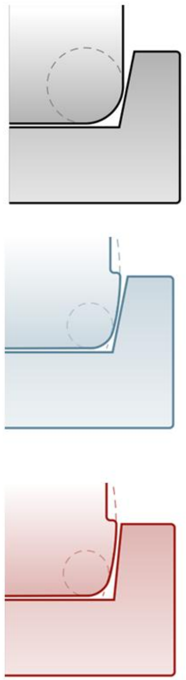
Simulation – Interaction Raceway-Rib

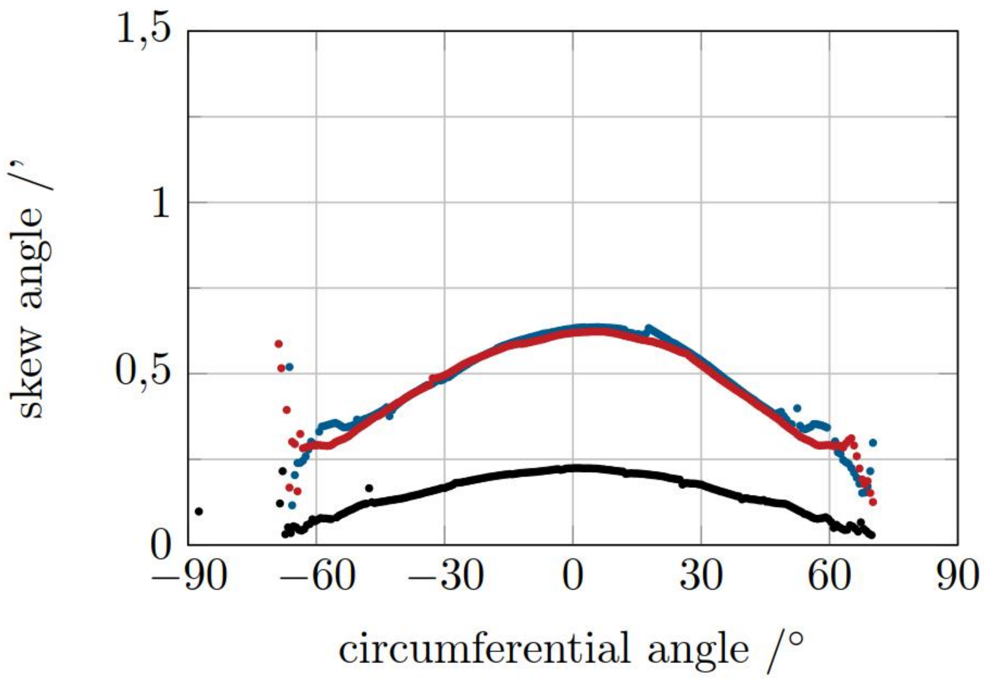




- **Entry and leaving of load zone:**
  - **Large tilt angle**
- **Small rib geometry influence (dominated by raceway geometry)**

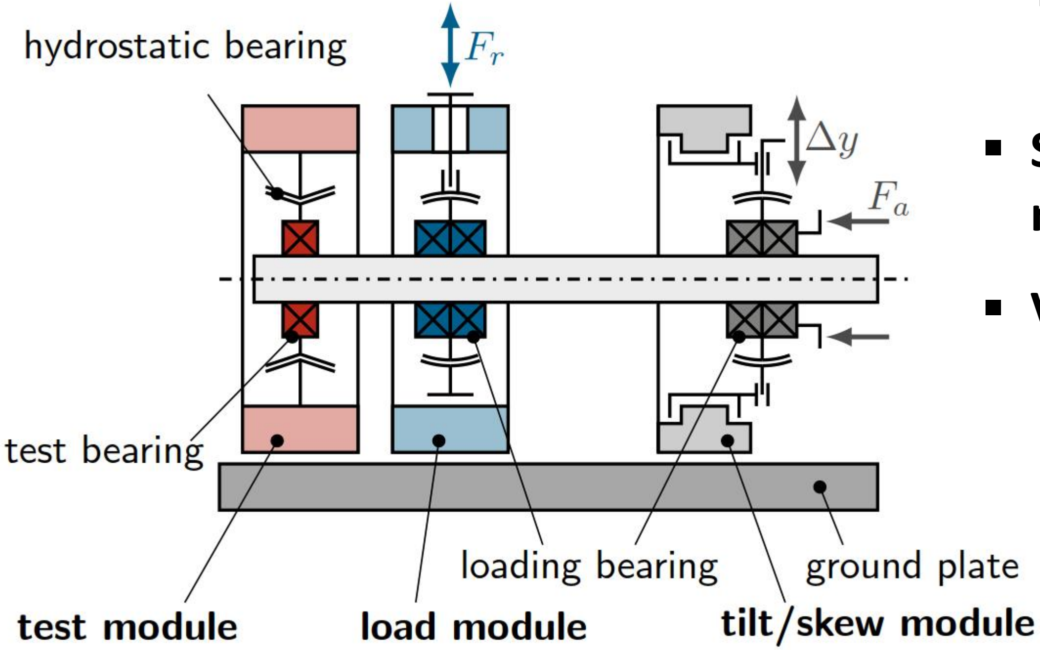
Simulation – Interaction Raceway-Rib



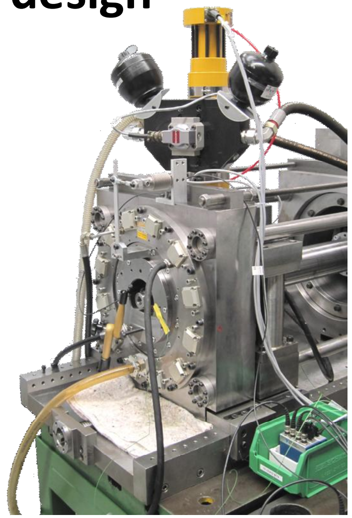


- **Skewing dominated by roller end-rib pairing**
- **Wide contact area of flat roller leads to small skew angles**

Validation

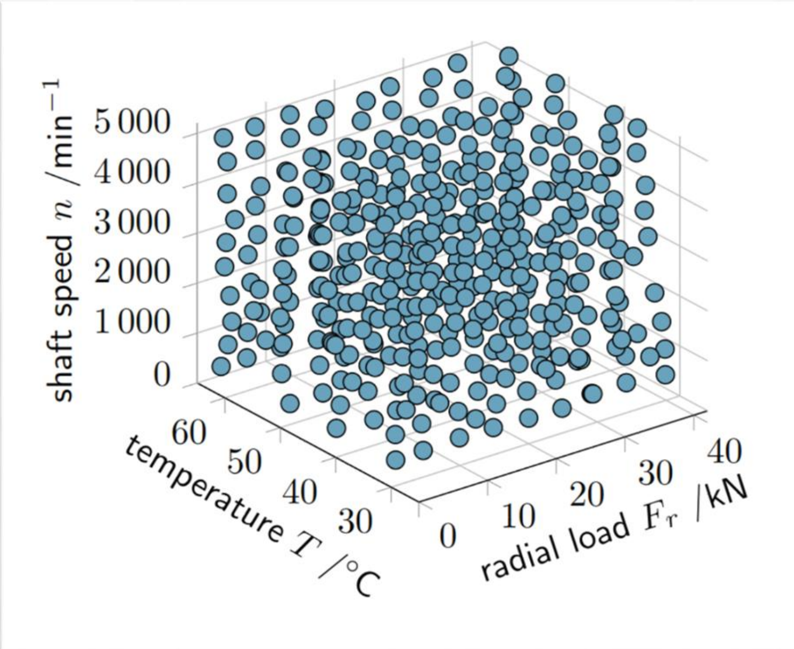


- **Test bearing**
  - **CRB NJ216C3**
- **Statistical experimental design resp. evaluation**
- **Validation measures**
  - **Friction torque**
  - **(Cage slip)**
  - **(Roller slip)**



Validation – Experimental Design

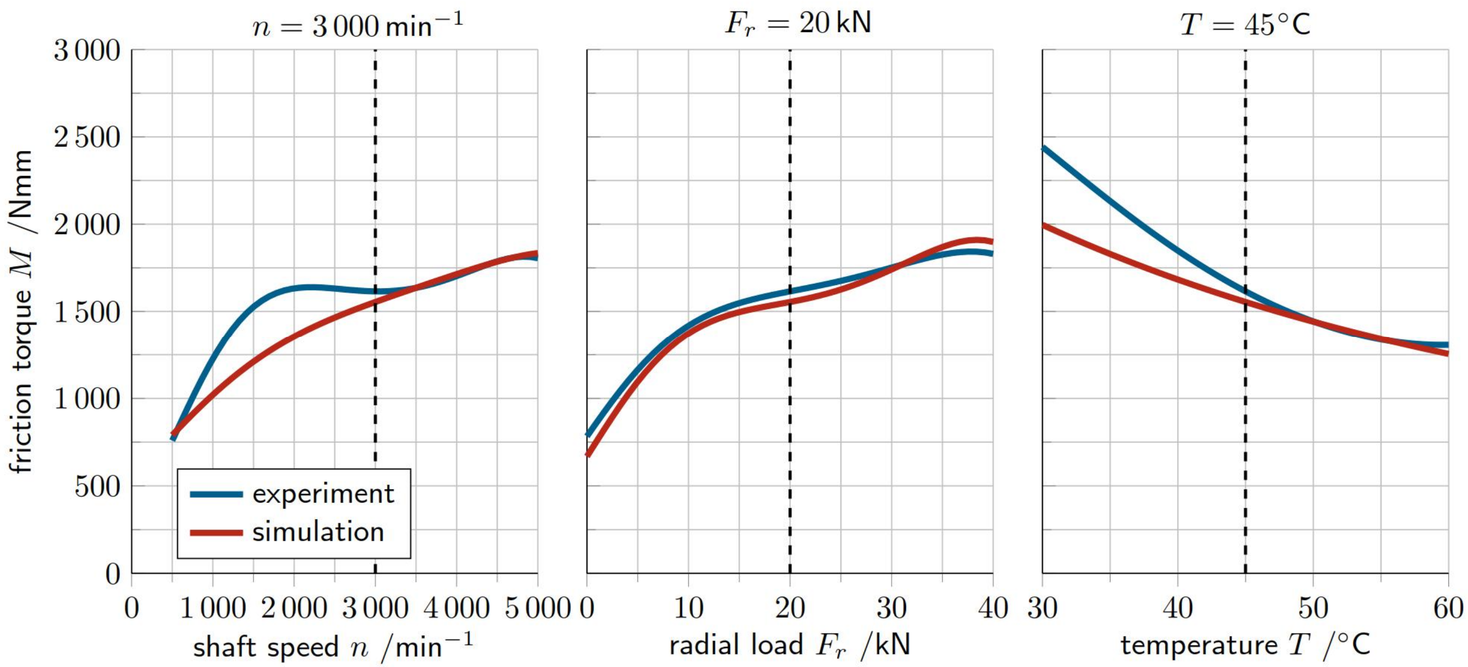
NJ216 C3
Radial load



factor levels
9
10
4

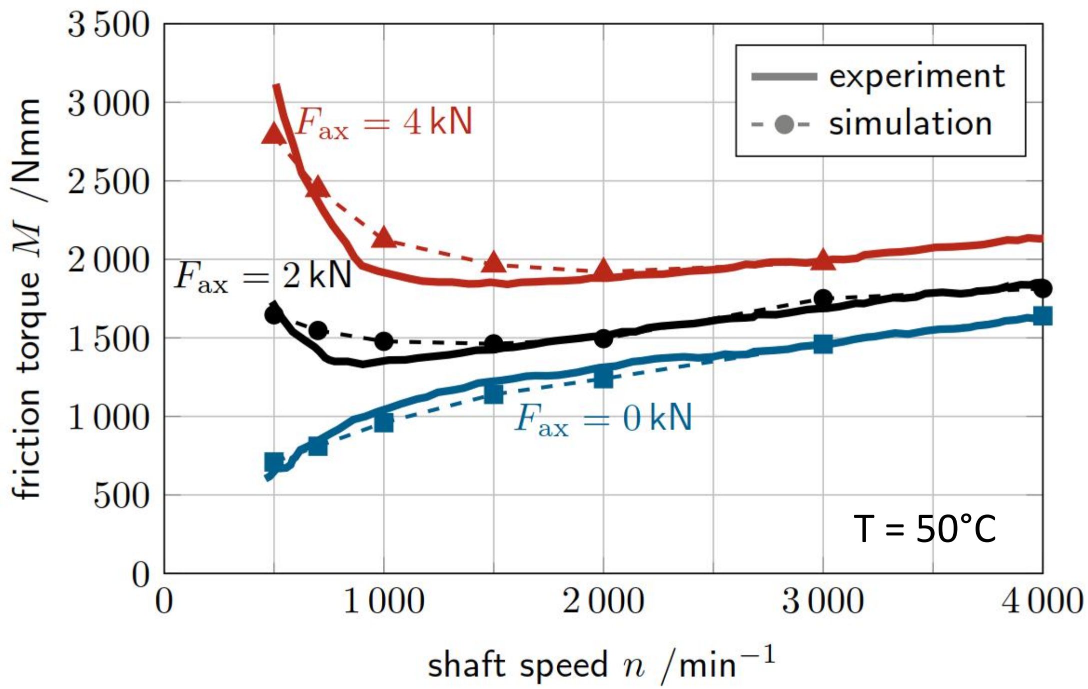
Validation – Friction Torque

NJ216 C3
Radial load



Validation – Friction Torque

NJ216 C3
Combined load



Summary

- Detailed MBS models for dynamic bearing behavior
- Integration of contact solver for arbitrary contact geometries
- Advanced friction models
- Extensive validation with experimental data

Outlook

- Integration of drag and churning losses
- Extention from contact solver to EHL solver
- Coupling with thermal model of bearing and environment

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