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Required simulation model detail to achieve reliable simulation results

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Abstract

The determination of the natural torsional frequencies of drivetrains is mandatory to predict the possible occurrence of resonances due to an excitation of mode shapes. The excitation can be caused by the imbalance or misalignment of shafts, by gear meshing frequencies, by loading characteristics or further excitation mechanisms. The measurement of the natural torsional frequencies on the first prototype can, in some cases, be too late with the result that opportunities are lost to realise major constructional changes. To avoid undesirable resonances during operation it is essential to use simulation models at an early stage of the product development process. The assembly of a simulation model pre-supposes the definition of system boundaries, the level of detail and the degrees of freedom.

These decisions result in the number of required parameters, the time and effort needed to model the system and the calculation time. However, in particular, stated expectations with regard to the targeted results and the accuracy of the model are of great importance in terms of the decisions to be made during the modelling process. The paper and presentation set out to demonstrate the influence of the level of model detail on the results to be achieved, as well as possibilities for using modelling approaches appropriate to the questions posed in different industrial applications (e.g. ships, cranes, bucket wheel excavators, rolling mills, wind turbines).

1 Introduction

The dynamic properties of drivetrains are frequently not only describable by torsional vibrations. The radial and axial vibrations occurring have to be considered in the design of the single drivetrain components as well. A precise knowledge of the dynamic properties and the resulting additional loads allows the detection of possible failures during the product development and can be avoided by constructive measures. The method of multibody system simulation offers extensive possibilities for the solution of these challenges. Unlike torsional vibration analysis, all six degrees of freedom can be considered when using the multibody system simulation. In addition to the rotation of the bodies around the rotational axis, the analysis of translational states in axial and radial direction as well as rotational states around the cross axes of the bodies are possible.

2 Basics of drivetrain simulation

The analysis of drivetrains operating under high dynamic loads pre-supposes the assembly of a detailed simulation which is able to represent the dynamic behaviour of the drivetrain in the frequency and time domain. Even if high performance computers are available, the level of detail of the simulation model has to correspond to the formulated question to ensure a feasible calculation effort. Based on the present data a discrete simulation model can be assembled. A successive and modular assembly of fully parameterised simulation models allows a clear and reproducible modelling process. The modular concept requires as a first step, the decomposition of the drivetrain into its substructures. Each substructure consists of model components which can be subdivided into shafts, gear stages, bearings and supporting structures. The combination of single substructures leads to the final, complete simulation model [8]. The degrees of freedom of each substructure can be adjusted to accommodate varying degrees of detail of the overall model. Submodeling enables easy verification of the appropriate level of model detail.

2.1 Modelling of drivetrain components

The dynamic behaviour of a drivetrain results from the gear ratio and the distribution of the mass, mass moment of inertia and stiffness. The determination of the mass parameters is possible with three-dimensional CAD models or by using simple analytical approaches. A great effort is required to accurately calculate the various stiffness parameters for all of the drivetrain components. The torsional stiffness of the drivetrain is mainly characterised by the stiffness of the shafts. Special consideration must be given to small diameter shafts whose elastic properties need to be taken into account. Additionally the bending stiffness of such shafts may have a considerable influence on dynamic behaviour as well as on the resulting displacements. The required simulation model can be assembled by using the method of discretisation, via the beam approach or by implementing modally reduced finite element models [1], [4], [9].

For models which include the axial and radial motion of the shafts the properties of bearings must be taken into account. Essentially, the modelling of the bearings is realised by a force element which introduces the reaction forces in the axial and radial directions as well as the reaction moments, if necessary. The bearing properties can be described by average bearing stiffness, characteristic curves or complex models imported as DLLs [13]. The required transmission can be realised by a gearbox consisting of a set of planetary, helical and/or bevel gear stages. The changing speeds and torques in a gearbox as well as the varying gearing stiffness resulting from the total overlap ratio have an important influence on the dynamic behaviour of the drivetrain and must be considered in the simulation model.

Even if spring-damper elements are used to support the components, the surrounding structure is assumed to have a stiff coupling to the global reference system. Thus the influences resulting from the flexible structure can be neglected. The implementation of flexible structures allows the representation of the flexibility of the supporting structure as well as the consideration of the stiffness of the drivetrain components, e.g. shafts and planetary carriers, with a higher degree of accuracy [2], [5], [6].

2.2 Analysis of natural frequencies, excitations and load cases

The resulting flexible multibody system model allows the determination of the natural frequencies and can consider various degrees of freedom. The resulting natural frequencies can be compared to the excitation frequencies in order to determine possible resonances. Relevant excitations are the first and second orders of the rotation frequencies of all drivetrain components, the meshing frequencies of the gear stages and further orders of excitation e.g. by rotors or propellers. The comparison of natural frequencies and excitations by means of a Campbell diagram reveals possible resonances. The analysis of the corresponding mode shapes allows further statements as to whether the excitation of a natural frequency can cause critical operation points or not.

The detailed flexible multibody system model also offers the possibility to determine the resulting displacements, deformations, velocities, accelerations, forces and torques under the dynamic loads. To model realistic loads an appropriate load model is required.

3 Application of the MBS-method to analyse drivetrains

The described methodology and system to assemble flexible multibody systems models for any mechanical drivetrains taken into account six degrees of freedom leads to complex models that allow analysis of the dynamic characteristics in the frequency and time domain. Dependent on the question at hand, the modelling approach and the level of detail must be adjusted. The following examples give an overview of the interaction between the required results and the creating and the assembling of simulation models to analyse different mechanical, electrical and control problems.

3.1 Analysis of the drivetrain of a ladle crane

The design of the drivetrain components for hoists in cranes requires in addition to the fulfilment of the function of lifting and positioning of loads, the consideration of high safety requirements. Especially in case of the operation of ladle cranes in the steel industry to transport open ladles, a failure of the drive can lead to material damage and personal injury. The required parameters to design a function and load-adapted emergency brake system can only be determined by a simulation software which can describe the dynamic behaviour of the entire system with sufficient accuracy. In standards a simplified approach is proposed, whereby the entire system is described by a two or three mass oscillator. For very simple drive systems, this approach is applicable. The real distribution of the mass moment of inertia and the stiffness, as well as the nonlinearity of coupling elements in complex systems is thus neglected [3], [14].

Figure 1 shows schematically an example of a 400 ton hoist of a ladle crane of the type used in the steel industry. The load of the ladle is moved by a traverse and two redundant cable drums, which are each connected via a drum gearbox to the power splitting gearbox and the two electric motors.

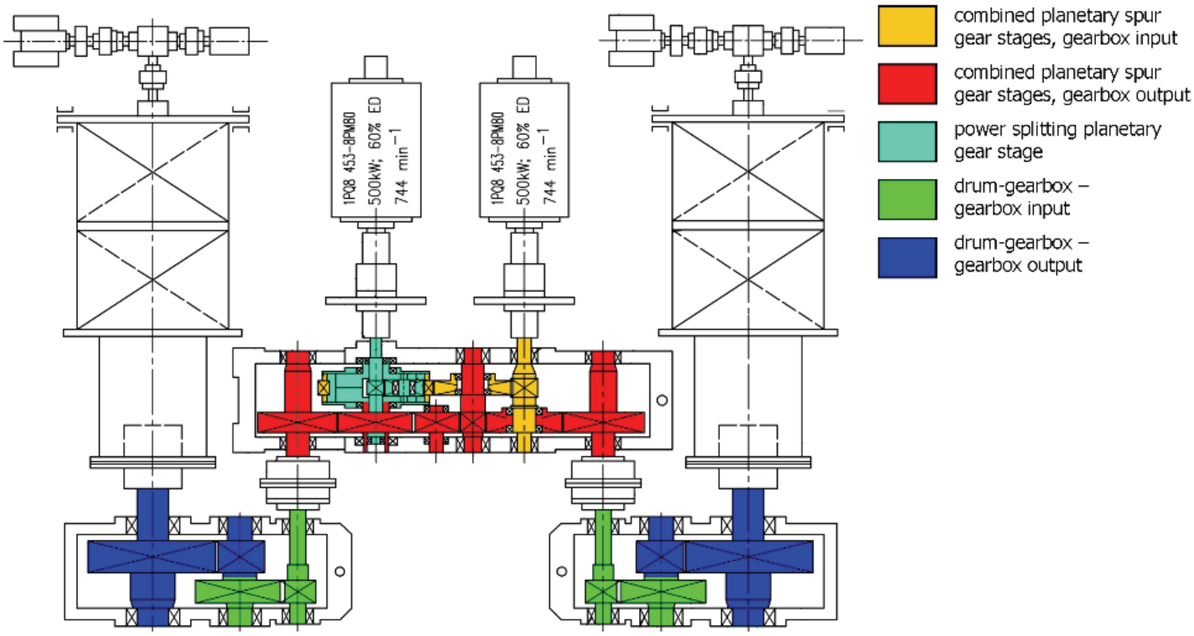

Figure 1: Drivetrain of a 400 ton hoist

Due to the complexity of the entire system a clear definition of system boundaries is not possible without a full understanding of the influences on system behaviour. Direct effects on the mode shapes of the drivetrain and the displacement of shafts under load result from the resilience of the gearbox housing. These effects can be taken into account by the implementation of modal reduced finite element structures. A similar procedure is possible to represent the flexibility of the trolley and the crane bridges. In addition to the support of the gearbox on the trolley frame, the extension of the system boundaries requires the modelling of all wheel-rail contacts between the trolley, the crane bridge and the rails on the crane runway. Figure 2 shows the complete model of the 400 ton ladle crane under consideration of the steel structures, which can be used to analyse the influence of the level of detail on the results in the frequency and time domain [12].

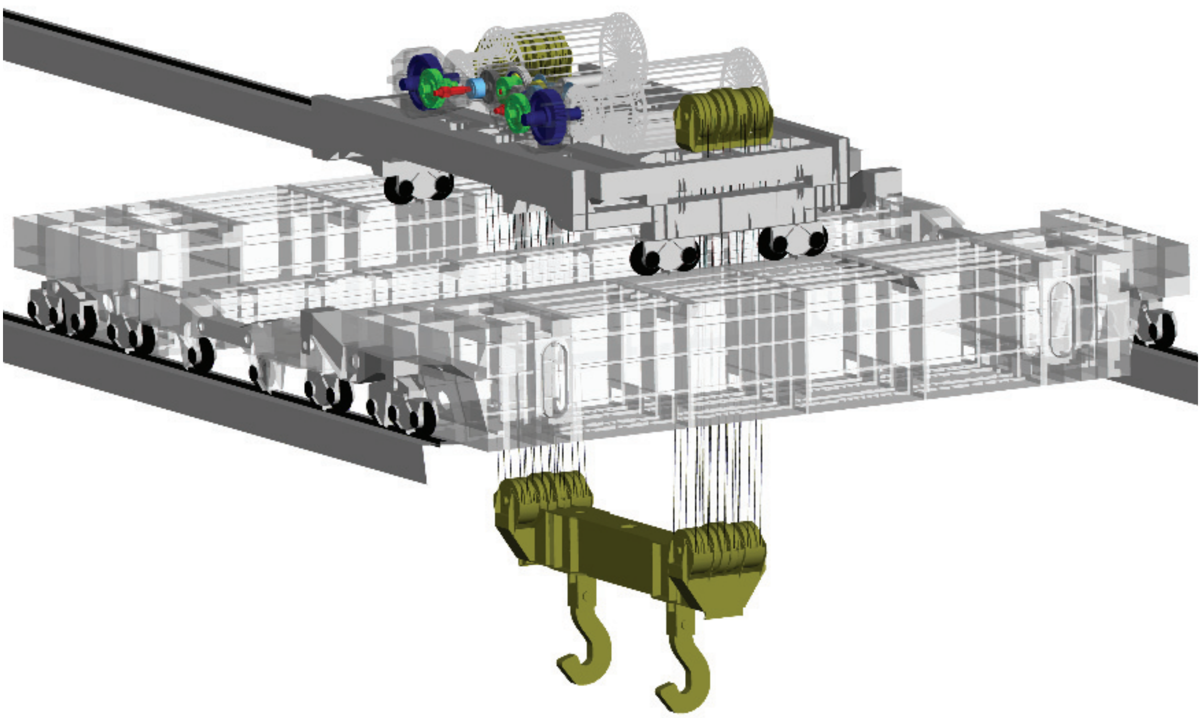

Figure 2: Complete simulation model of the ladle crane

Based on the comparison of the maximum tooth forces for the load cases "emergency stop", without incurring damage and after a drum-side shaft failure, the influence of modelling depth can be evaluated (Figure 3). The comparison between the torsional oscillator and the rigid simulation model with six degrees of freedom for all components shows the opposite trend of the influences of the level of detail on the maximum occurring forces for the load cases considered. After the implementation of elastic structures for the gearboxes, the trolley and the crane bridges, the maximum loads are 10 % lower.

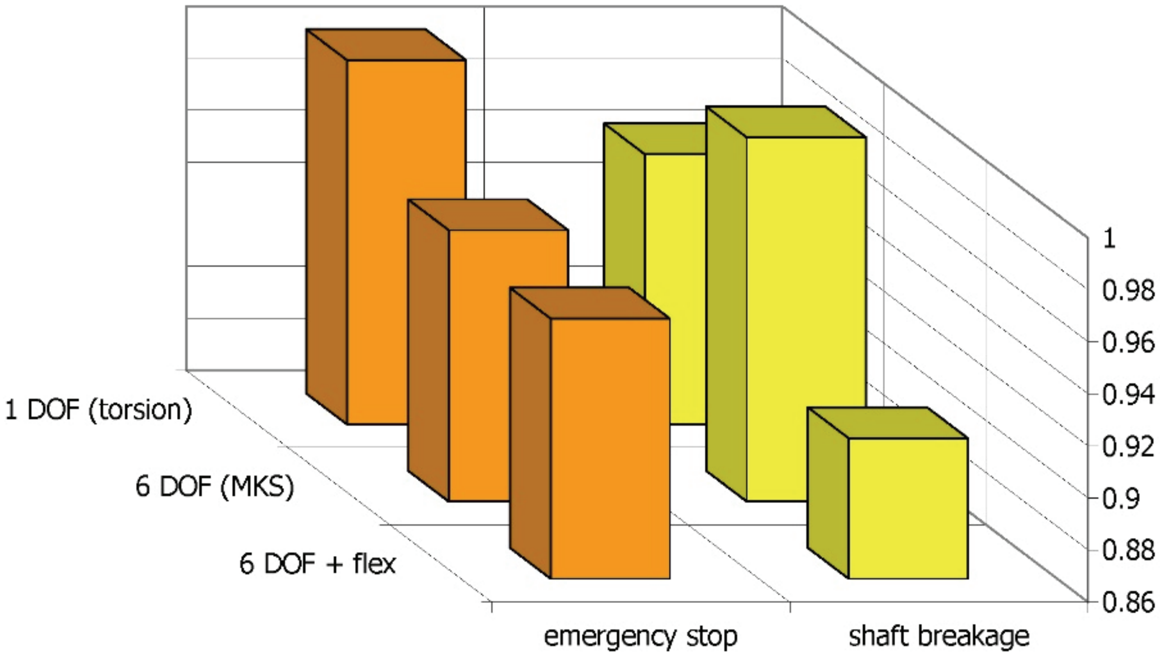

Figure 3: Influence of modelling depth on resulting maximum loads in the gearbox

The relatively small deviations of the resulting loads do not justify the high effort that is required to create the complex flexible multibody system simulation model. Due to the complex mode shapes of the drivetrain components and the load-dependent misalignment of the gears, the exact representation of the dynamic properties of the hoist requires the consideration of all six degrees of freedom for all drivetrain components and gearbox housing stiffness.

3.2 Analysis of a thruster drivetrain

The usage of modern thrusters allows the combining of the drive and the ship rudder in one unit, which are separated in conventional ship propulsion systems. The horizontally oriented propeller is supported in a vertically rotatable nacelle, mounted underneath the ship's hull. The propeller can directly or indirectly be driven by an electric motor or combustion engine. Direct drive requires the installation of a low speed electric motor in the nacelle, in indirect drives the driving torque is transferred by bevel gear stages and shafts from the motor to the propeller. Due to their closed and inaccessible construction, high reliability has to be achieved. Especially for the design of the highly-loaded bevel gear stages accurate information of the occurring loads are required. The available experience of the operation of thrusters shows, that primarily rarely occurring special load cases must be considered in the design process. Such operational conditions can only be determined by expensive long-term measurements. By means of a detailed multibody system simulation model of the thruster, it is already possible to develop a basic knowledge of the dynamic properties of the drivetrain and to determine design loads for drivetrain components [10].

To fully realise the described modularisation, the simulation model of the thruster consists of the submodels motor, coupling, the vertical drive line, the bevel gear stage, the propeller shaft and the propeller. All components are assembled in a complete model of the thruster and supported in the modally reduced finite-element model of the thruster housing. The release of all degrees of freedom and consideration of all supporting and connecting spring-damper elements allows, by comparison of natural frequencies and excitation frequencies, the determination of critical operational speeds and the analysis of the dynamic behaviour of drivetrain components and the supporting structure (Figure 4).

Besides the analysis of possible excitations of natural frequencies in the range of the operational speed by means of the detailed simulation model the occurring loads for all drivetrain components and different operational conditions can be analysed. This requires an enlargement of the mechanical model to characterise the acting motor and propeller side loads in detail.

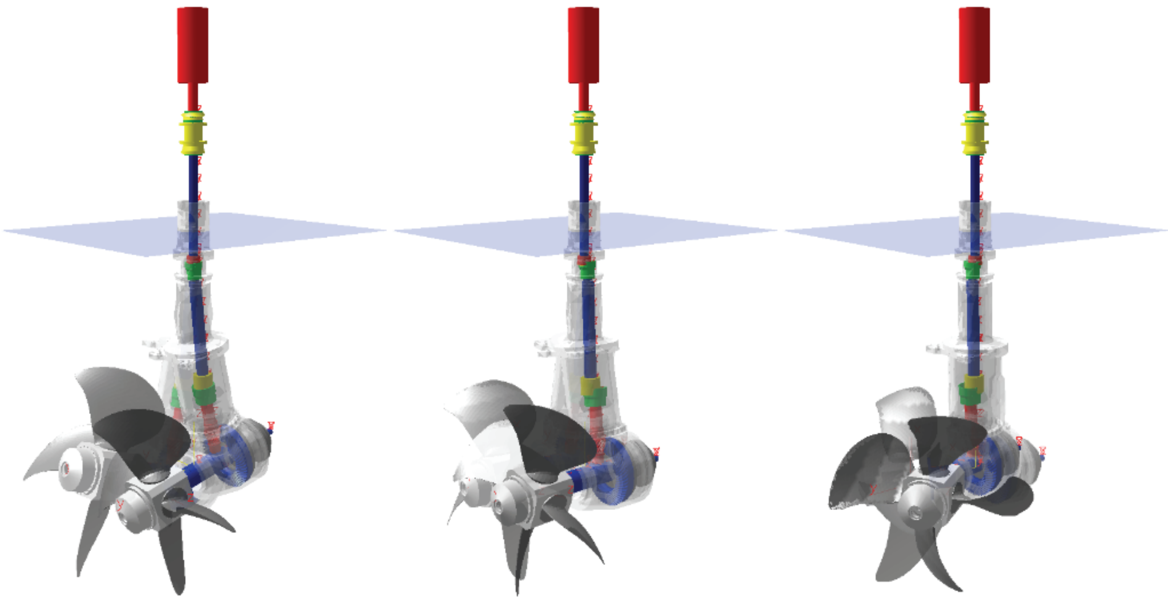

Figure 4: Mode shapes of the thruster (10 Hz, 11 Hz, 25 Hz)

The description of the electric motor can be realised by modelling the different control loops in MATLAB/SIMULINK. To determine the occurring component loads during different operational conditions, a very detailed propeller force model is necessary [7]. The introduction of the water-resisting forces is carried out by modelling a discrete load distribution over the blade length for the tangential and axial force components, separable for each blade. The introduction of the resulting forces on the propeller blades in the multibody system model allows a first analysis of the occurring loads for shafts, bearings and gearings [11]. Figure 5 shows the torque of the propeller shaft and the motor speed over the simulation time as a comparison between measured and simulation results.

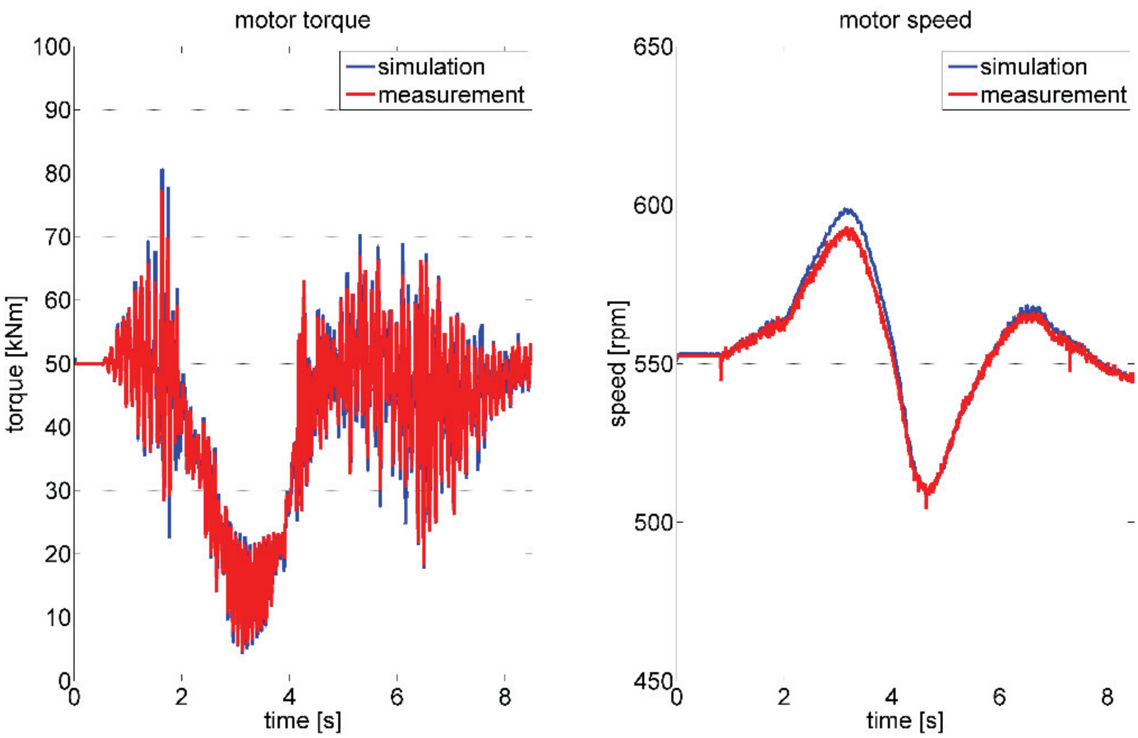

Figure 5: Time series for the immersion and emersion of the thruster

4 Conclusion

The illustration, the assessment and the understanding of the dynamic properties of complex drive systems require the use of simulation models that can represent all relevant system properties with sufficient accuracy. Therefore the used modelling approach has to adapt to the design of the drivetrain to simplify unessential details and to consider the stiffness and inertia of important components. The resulting simulation model of the complete drivetrain may combine different levels of detail in the modelling approach. This requires the proof that the connection between a highly detailed part of the model and simplified models does not influence the results. Hence each assembly of a model is based on a simple torsional vibration model which is successively extended. The results must be analysed after each change of the model. The knowledge of the correct system boundaries often requires the existence of a very detailed simulation model which is simplified in successive steps. A comparable procedure can be necessary to determine the optimum solver settings. Due to the many different influencing factors and dependencies, a much higher effort is required to assemble a reliable, reproducible and understandable multibody system simulation model, compared to a torsional vibration model.

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