Determination of Maximum Loads for Drivetrain Components in Thrusters Using Flexible Multibody-System Models

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The usage of modern thrusters allows combining the functions 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 that is 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. This present paper concentrates on indirect drives where the driving torque is transferred by bevel gear stages and shafts from the motor to the propeller. Due to 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 is 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 multi-body 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.

Introduction

The different drivetrain and ship concepts, the complicated operational conditions, and the high demand on reliability lead to many different tasks and conditions that must be considered in the design process of thrusters. Therefore the occurring operational conditions are analyzed using simple, torsional oscillation models of the drivetrain as they exist today (Ref. 4). In addition to the typical concept where the fixed propeller is driven by a long shaft, water jet engines, thrusters and also special solutions like the Voith-Schneider drive are used. The thrusters are mounted underneath the ship hull and the thruster housing can rotate around the vertical axes so that they can be used as either pushing or pulling drive and also as ship rudder. The driving power can be directly supplied by an electrical motor, installed in the nacelle (ABB, Rolls-Royce).

Alternatively, the driving torque is transferred by gearboxes and long shafts from the driving unit in the ship hull to the propeller of the thruster (Schottel, Rolls-Royce, Wärtsilä). Thrusters driven by an electrical motor or combustion engine using a gearbox are able to operate with a constant driving speed if the provided thrust is adjustable by pitchable propeller blades. Due to their good maneuverability, thrusters are commonly used in ferries and tug boats. Thrusters are also often used if high demands are made upon the positioning accuracy required by ships for gas and oil production, as well as for scientific marine research.

In comparison to the typical driving concepts using a long shaft, changed design loads must be considered due to the combined function of driving and steering, as well as the paired arrangement on both sides of the ship (Fig. 1, left).

Also, the discontinuous operation and the area of application can influence the design process. Next to the occurring torques, bending moments around the vertical axes resulting from the steering movements of the thruster must also be taken into account for the different operational conditions. A relevant design load case results from the positioning of the thrusters on both sides of the rear. In high waves an emersion of propeller can occur so that during such immersion the

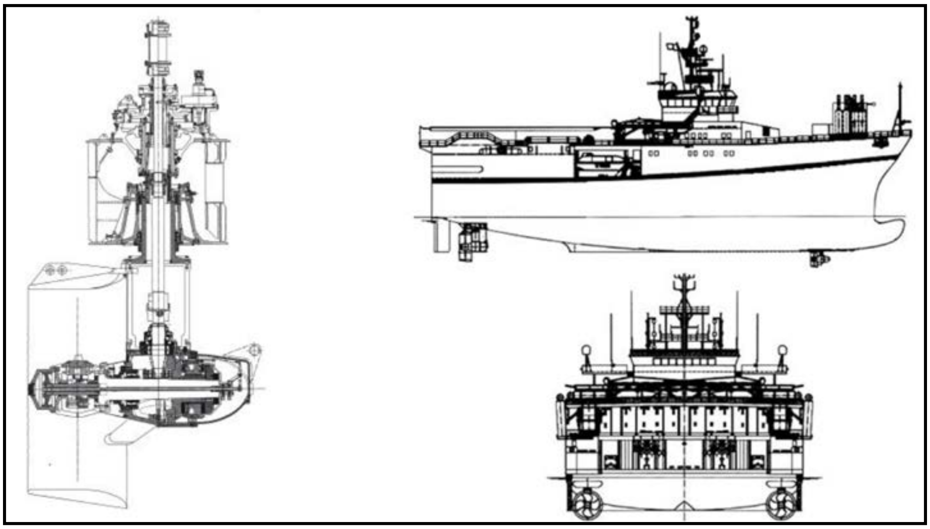

Figure 1 Positioning and design of the thruster (Ref. 16).

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blade tips are slamming on the water surface. This leads to short-time overloads that have to be transferred and supported by the drivetrain — without damage. The determination of occurring drivetrain components loads, and analysis of the dynamic behavior, can be achieved either by a complex measurement setup in the thruster nacelle or with the aid of detailed simulation models. The challenges of a measurement campaign are the difficult environmental conditions and missing accessibility to install sensors after the assembly of the thruster. Thus a measurement setup is time consuming and expensive. An availability of detailed measurement results for different drivetrain components will be an individual case and not applicable to design thrusters. The determination of the component loads using the simulation results of complex drivetrain models can already be performed during the product development process.

Basics of Drivetrain Simulation

The analysis of drivetrains operating under high dynamic loads presupposes the assembly of a detailed simulation model that is able to represent the dynamic behavior 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. Despite the currently given possibilities of simulation software, the modeling process is very time-consuming; based on the present data to the drivetrain, a discrete simulation model must be assembled. A successive and modular assembly of fully parameterized simulation models allows a clear and reproducible modeling process, compared to the combination of all drivetrain components in one unstructured model.

The modular concept requires as a first step the decomposition of the drivetrain into its substructures. According to this approach a simulation model of a thruster consists of the following substructures: propeller; propeller shaft; coupling; motor; and an additional, subdivided gearbox. Further, the gearbox can be subdivided in different spur and helical gear stages, bevel gear stages, and planetary gear stages whereby in the analyzed thruster only bevel gear stages are used (Fig. 2). Each sub-structure consists of model components that can be subdivided into shafts, gear stages, bearings and supporting structures. The combination of single substructures leads to the final, complete simulation model of the thruster. Independent of the present requirements, an adjustment of the level of detail and the needed degrees of freedom for each sub-model can be performed. Compared to the work with one single simulation mode for the complete drivetrain, the usage of different sub-models enables an easy verification of the function and accuracy.

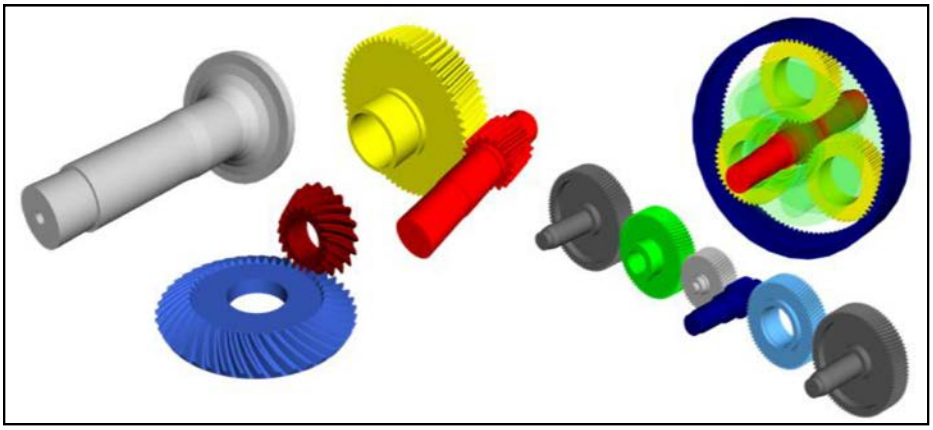

Figure 2 Different kinds of substructures.

Assembly of Thruster Simulation Model

The assembly and functions of a thruster are shown (Fig. 1, right). The nacelle, with the function of the gearbox and cover of the drivetrain, is mounted rotatable around the vertical axes in the ship's hull, and can be turned by an additional drive. The main driving machine in the ship hull transfers the required torque for a horizontally positioned driving aggregate by an elastic coupling and a bevel gear stage. For a vertical-mounted driving machine the torque is transferred directly by a coupling to the vertical driveline in the nacelle. The segmented drive line is supported by several bearings in the housing. The shaft segments, as well as the pinion of the bevel gear stage, are connected by geared couplings. The wheel of the bevel gear stage is mounted on a carrier. The carrier is directly connected to the propeller shaft, which is supported by a roller and a sliding bearing. The axial-mounted hub is used to support the four pitchable propeller blades, which can be positioned by a hydraulically acting linkage.

The dynamic behavior of the drivetrain is mainly characterized by the large motor- and propeller-side mass moment of inertia, as well as the high flexibility. Depending on the required thrust, propeller diameters up to 5 meters are installed. The occurring torque and bending moment during operation presuppose a stiff design of the propeller shaft. By contrast, thin shafts are used in the vertical driveline because the gear stage ratio lowers the torque and the resulting stress. The lower torsional and bending stiffness of these shafts must be addressed; also, the motor is connected by an elastic coupling with the drivetrain. Simplified, the complete system can be

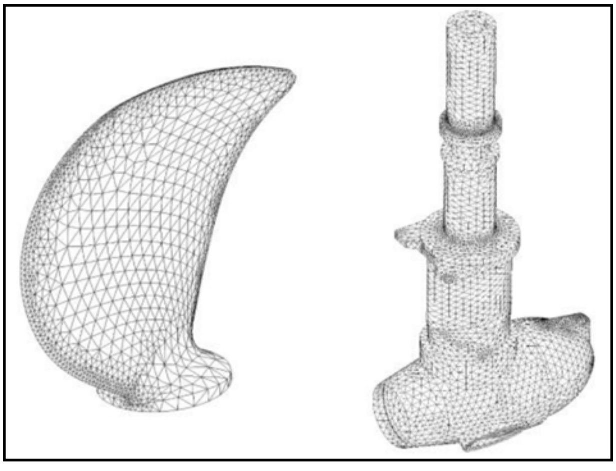

Figure 3 Finite-element model of the propeller blade (left) and the thruster housing (right).

described by two large masses of inertia, connected by a soft torsional stiffness.

Additionally, all drivetrain components are supported in the nacelle, which is also an elastic system and only connected at the top to the ship hull. Under consideration of the acting forces, torque, and bending moments, a simulation model representing the torsional degrees of freedom can be only used for a simplified, rough analysis of the dynamic behavior. A comprehensive investigation of the entire system requires a detailed modeling of all relevant degrees of freedom in a simulation model.

All shafts of the drivetrain have to be modeled with the information to the torsional and bending stiffness, the mass and mass moment of inertia, as well as with the rotatory and translator degrees of freedom. The determination of the mass parameters can be performed using common three-dimensional CAD software or by means of simple analytical approaches. A higher effort is demanded to calculate the stiffness of the components. The torsional stiffness of the drivetrain is mainly characterized by the flexibility of the shafts. Especially thin shafts have to be considered with their elastic properties. Additionally, the bending stiffness of such shafts can have major influence on the dynamic behavior and occurring displacements. A simulation model required to represent shafts can be assembled by means of the method of discretization, by the implementation of beam models, or by using modally reduced elastic structures (Refs. 2, 5, 17 and 25).

The consideration of axial and radial degrees of freedom supposes the modeling of the bearings. Essentially, the modeling of the bearings is realized by a force element that 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 DLL's (Ref. 24).

To support the shafts in the thruster housing, the bearings are modeled as translatory spring-damper-elements, whereby the load-dependent bearing stiffness is implemented so that for all occurring load cases, this approach can be used. Also the properties of the elastically and geared couplings are described by spring-damper elements. For the motor side, coupling information of the stiffness characteristics is required and must be provided by the manufacturer. According to current knowledge, the stiffness of geared couplings can only be determined using analytical approaches (Refs. 1, 6 and 9). Information regarding radial stiffness and stiffness against inclination due to the comprehensive influence factors and uncertain calculation methods is not available. As a first approach, all possible degrees of freedom are locked by constraints or high stiffness, and the occurring influences on the dynamic behavior must be determined by sensitivity analysis. The model of the bevel gear stage between the vertical drive line and the propeller shaft must describe the transfer behavior for the torque, as well as for all force components, so that the dynamic properties of the complete drivetrain can be correctly represented. Next to the description of the non-linear characteristic of the stiffness resulting from the changing contact conditions, the backlash has to be considered during the calculation of the acting forces. For each step of the integration, the determination of the equilibrium between the acting forces, displacements, the inclination of the shafts and the inner gearing forces.must be ensured. The simulation software *Simpack* offers the toolboxes *Gearwheel* and *Gearpair* to model gearings and to describe the transfer behavior in detail.

An alternative modeling approach offers the mathematical description of the resulting forces in the gearing by means of user routines. Based on the calculation of the tooth normal force in the ideal pitch point, the complete tooth contact is simplified and described in one point. The tooth normal force consists of stiffness and damping-dependent parts. Information about the displacements and velocities in tangential, radial and axial directions resulting from the relative position of the gears can be determined by the joint states and the corresponding trigonometric relationships. The gearing stiffness can be considered as average contact stiffness according to DIN 3990 and variable gearing stiffness over the path of contact using Fourier coefficients.

The rigid modeled hub is mounted axially on the propeller shaft and supports the four pitchable propeller blades. To consider the flexibility and resulting deformations of the blades under the high loads, the material and shape-dependent elasticity has to be considered. Due to the complex geometry, the method of discretization or beam approaches cannot be used so that on the basis of a detailed finite element model and the dynamic reduction the propeller blades are represented by modal reduced elastic structures (Fig. 3, left). The implementation of a flexible structure in *Simpack* is based on a meshed finite element model of the component geometry and the definition of the material properties. Additionally, the modeling of the connection points between the elastic structure and the rigid bodies of the multi-body system model is required. The connection points to the supporting spring-damper elements that can be modeled by means of multipoint constraints (MPC). However, the resulting FE model is assembled by many shell or solid elements and has therefore much more degrees of freedom as necessary to describe the rigid body motions in the *MBS* model. Because such complex models cannot be handled by a classic *MBS* solver, the level of detail of the finite element model must be reduced to the transfer behavior between the connection points. Additional information on the displacement of nodes, which are not used as connection points in the *MBS* model, is not available in the reduced model of the structure. The application of the reduction approach according to Craig-Bampton requires the definition of the connection points between the flexible structure and the rigid bodies. The mode shapes of the reduced model are used to determine the deformation under load (Refs. 3 and 7). The number of natural frequencies chosen for the modal reduction defines the valid frequency range and the accuracy of the model, which is also influenced by the choice of frequency response modes in the *Simpack* add-on module *Fembs* (Ref. 8).

A comparable proceeding is performed to represent the elastic properties of the thruster housing. The large mass of the propeller and the propeller shaft with the bevel gear wheel as well the propeller side loads have to be supported by the structure of the thruster housing, and have to be transferred to the large bearing in the ship hull. The expected deformations under the load will have an influence on the dynamic behavior of the drivetrain. Based on the geometry of the thruster housing a finite element model can be assembled (Fig. 3, right). The connection points for the support in the ship hull and the positions of the bearings for the drivetrain components are linked by constraints to a number of surface nodes in the area of the bearing seats. After the implementation of the reduced finite element model, the spring-damper elements representing the bearings will be defined between these connection nodes and the body marker of the MBS model. So all introduced loads are directly transferred as torque or supported by the bearings in the thruster housing and the ship hull.

Analysis of the Thruster Drivetrain in the Frequency Domain

To realize the described modularization the simulation model of the thruster consists of the sub-models motor; coupling; vertical driveline; bevel gear stage; propeller shaft; and 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 behavior of drivetrain components and the supporting structure (Fig. 4). Possible excitations are the rotation frequency of all drivetrain components in the first and second order; the gear meshing frequency of the bevel gear stage with the first order and higher harmonics; the rotation frequency of the propeller with the first order and higher harmonics corresponding to the number of installed blades, and disturbance of the flow due to the nacelle design. The named sources can excite torsional, bending, radial and axial mode shapes that have to be analyzed for each determined, critical operational speed. Especially the propeller-side excitations have an important influence on the dynamic behavior of the complete system, because the torque as well as the acting forces can lead to resonances with different harmonics of the rotation frequency of the propeller shaft.

In Figure 5 the Campbell diagram for the thruster with all natural frequencies and the first (1p), second (2p), third (3p), fourth (4p), and eighth (8p) order excitation of the propeller rotation speed, as well as the first (1p) and second (2p) order excitation of the gear meshing frequency — with up to 140 Hz shown exemplarily. The first natural frequency of the complete system at 10 Hz is characterized by a bending mode shape of the thruster housing against the support in the ship hull. The fourth order of the propeller rotation frequency could cause a resonance with this mode shape at an operational speed of 635 rpm (Fig. 4, left). In addition to the stiffness of the housing, there exists the stiffness of the bearing that supports the thruster in the ship hull and has influence on the mode shape. The first torsional mode shape of the drivetrain at 11 Hz is super-posed by a second bending mode shape of the housing. This natural frequency can also be excited by the fourth order of the propeller rotation frequency at an operational speed of 720 rpm (Fig. 4, right). The mentioned excitation frequency is caused by flow disturbance that occurs if a blade passes the thruster housing. The changing torque, bending moments and forces can lead to an excitation of both mode shapes.

Also the higher natural frequencies of the drivetrain are characterized by the super-positioning of housing and drivetrain mode shapes, and can be excited by higher harmonics of the propeller rota-

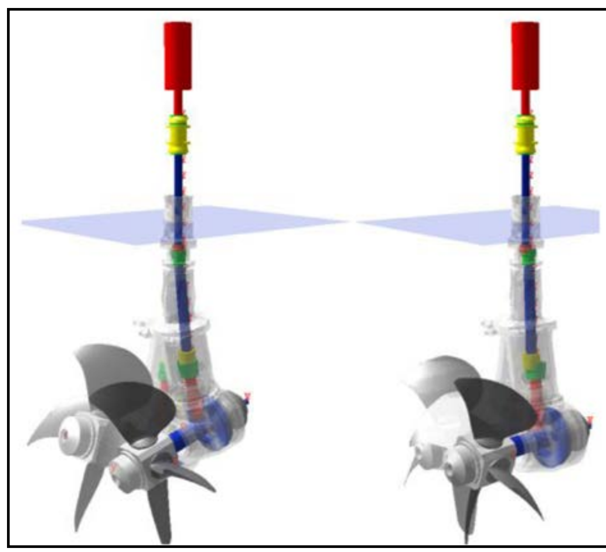

Figure 4 Mode shapes of the thruster (10 Hz, 11 Hz), (Ref. 16).

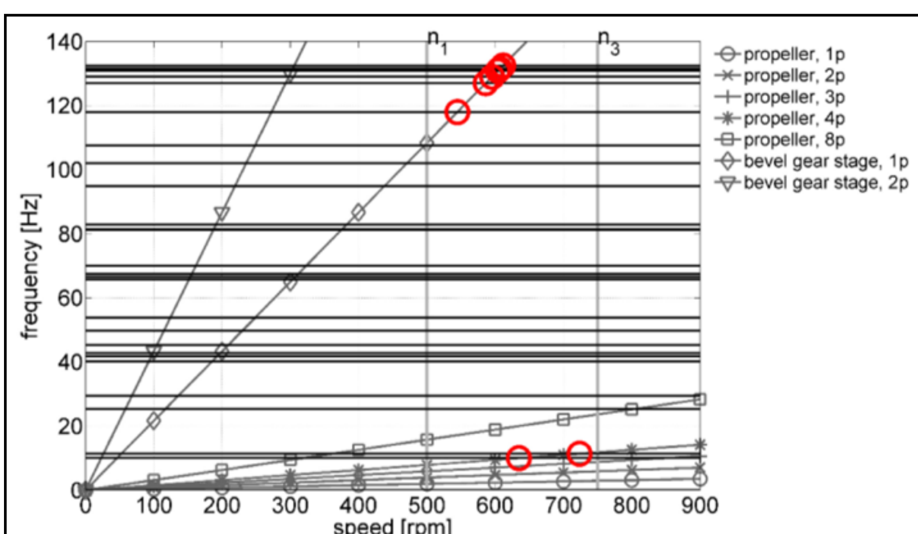

Figure 5 Campbell diagram for the flexible multi-body system model of the thruster.

tion frequency. For the analyzed drivetrain, only the fourth and eighth orders of the propeller rotation frequency are relevant. For future investigation, the interesting frequency/speed range is limited by the lower border of the operational speed and gear meshing frequency of the bevel gear stage at approximately 110 Hz. In the analysis of the higher frequency range, all possible intersections between the gear meshing frequencies and the determined mode shapes have to be taken into account. Due to the exciting gearing force components in tangential, radial, and axial direction, an excitation of torsional, axial and bending mode shapes, as well as mode shapes against the shaft support, are possible. In addition to the theoretical investigations using the Campbell diagram, a comprehensive evaluation of the excitability can only be performed by the simulation of a slow run-up and a detailed analysis of the simulated velocities, accelerations and torques.

Analysis of Thruster Drivetrain in Time Domain

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 analyzed. This requires an enlargement of the mechanical model to characterize the acting motor and propeller side loads in detail.

The description of the electric motor can be realized by modeling the different control loops in *Matlab/Simulink*. Important for a realistic motor model is the knowledge of all motor parameters. These parameters must be provided by the manufacturer of the electric motor. Regarding the presented thruster, only some rough information on the motor was available, so that by means of speed-torque characteristics a simplified model is used to describe motor behavior. The modeling of the propeller-side loads requires a comprehensive discussion on the proper modeling approach. To analyze a simple torsional vibration model, the information to the occurring torque is sufficient and already allows a first evaluation of the dynamic behavior and testing of the model. The simplified consideration of the torque neglects the important influences of the thrust forces and bending moments that are also applied at the propeller. These load components cause bending of the thruster housing, displacement of the propeller shaft, and also have an impact on the contact conditions in the bevel gear stage. Next to the analysis of operational states under full load for different flow angles, especially extreme load cases like the immersion of the propeller at maximum input power, can be seen as critical to the reliable operation of the thruster, and should be investigated with the model (Ref. 13). Until now, no comprehensive measurement results for such thrusters are available, so that the occurring loads during the immersion and emersion of the propeller were analyzed only with scaled models in water tanks. When the propeller approaches the water surface, the surrounding water is already mixed with air, so that the thrust and the acting torque decreases. The motor speed increases due to the lower-resisting torque. A further immersion of the propeller causes at first a water-free movement of the upper blade. At the moment of the blade immersion the resisting force suddenly increases, which leads to a short-time increase of the torque in the drivetrain (Refs. 10–12). To determine the occurring component loads during such load cases, a very detailed propeller force model is necessary. The introduction of the water-resisting forces is carried out by modeling a discrete load distribution over the blade length for the tangential and axial force component, separable for each blade. The calculation of the acting forces for each load introduction point occurs — independent of the rotation angle, the distance from the water surface, and measurement-based assumptions for the force progression by means of a *Matlab/Simulink* model. The introduction of the resulting forces on the propeller blades in the *MBS* model allows a first analysis of the occurring loads for shafts, bearings and gearings. Figure 6 shows the torque of the propeller shaft and the motor speed over the simulation time as a comparison between measured and simulation results.

Further possibilities for the description of the propeller-side loads are given by the computational fluid dynamics (CFD). The method is used to analyze the ship hull and interactions between ship hull and thruster during the design process. Because of the detailed, computational-intensive models, a CFD simulation can only be done for single revolutions of the propeller and defined environmental conditions. A combination of CFD and *MBS* simulation is due to long simulation time by using the currently available computer performance for the dynamic simulation not applicable. But the propeller forces and torques can be pre-calculated for defined quasi-static load cases and introduced in the *MBS* model using force elements (Ref. 15). To improve the

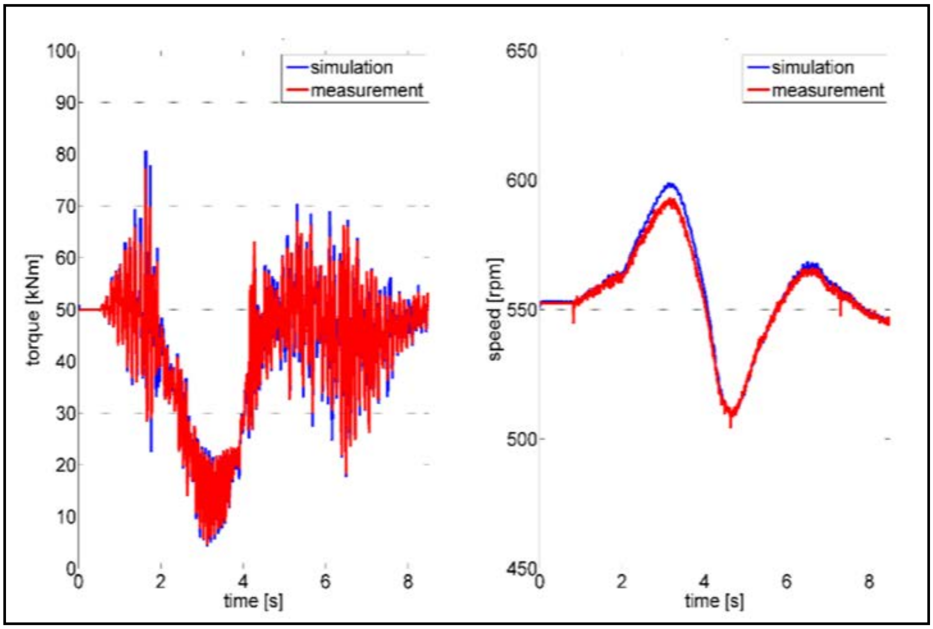

Figure 6 Time series for the immersion and emersion of the thruster.

propeller force models and to validate the mechanical simulation model, the measurement of the different real occurring operational loads by means of an extensive measurement setup over an extended period is mandatory.

Conclusion

The described methods can be used to model the mechanical components, as well as the acting propeller loads, so that basic observations of the dynamic behavior of the thruster drivetrain can be expressed. The comparatively simple constructive design of the drivetrain is characterized by the high flexibility of the driveline and the thruster housing. This leads in combination with the large propeller and motor inertias, as well as the acting forces, to high dynamic states in the drivetrain. If a sudden increase or decrease of the propeller-side torque occurs, first the twist of the drivetrain must be resolved before the backlash in the gearings or couplings can affect the dynamic drivetrain behavior. The occurrence of back flank contact in the bevel gear stage is possible if the propeller load changes with an amplitude and for the duration, so that a twist-free drivetrain exists. The motor-side coupling can reduce overloads at the motor. Damage to the drivetrain can only be avoided by an overload protection at the propeller shaft. The challenge is the design of an overload protection that can limit torque reliably and is small enough for installation in the available space in the thruster housing (Ref. 14).

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