

POSSIBILITIES TO DETERMINE DESIGN LOADS FOR THRUSTERS

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Abstract. Drive trains of icebreakers in arctic conditions are subjected to high stress levels. Due to collisions between ice and propeller, overloads and torsional vibrations occur. The main objective of the “Arctic Thruster Ecosystem” (ArTEco) project is to increase the reliability of vessels in arctic conditions. In order to achieve this goal different load scenarios are analyzed with multibody-system (MBS) simulation tools complemented by a test rig facility located in Tuusula (Finland). It is operated by the technical research center of Finland (VTT) and Wärtsilä. On the test rig, various loads can be applied to different azimuth thrusters, which are equipped with diagnostic instruments. For further analysis a simulation model is assembled and verified by several time-based data sets and modal analysis of the test rig. Thereby, overloads and high dynamic loads that are unobtainable at the test rig become accessible in simulations. To reduce torsional vibrations, different damping systems, designed at the department of intelligent hydraulics and automation (IHA) Tampere and the VTT, can be analyzed with the MBS-model. Besides torsional vibrations, one objective is the displacement analysis of the bevel gears. Using simulation-based displacement data and the software BECAL [1], a precise contact pattern can be determined, which allows conclusions regarding the required safety factors.

1 INTRODUCTION

Shipping operation in the sensitive but harsh arctic environment combine the demand for extreme reliable and energy efficient machines with the necessary requirement to prevent pollution of the arctic ecosystem. The main objective of the ArTEco project is an improvement of arctic ship propulsion products, with the target to identify critical areas in ship propulsion design and improve products for this industry. The consortium comprises of skilled research organizations, system integrators, component manufacturers and technology providers. Significant progress in conceptual development is achieved by a new full size thruster test facility established in Finland in 2013. The test facility allows measurements in a way that is not possible when a unit is in operation. A 2 MW thruster can be run with full power at the test rig and the operational loads can be applied to the thruster by hydraulic cylinders. The test rig allows the measurement of bearing loads, gear and shaft displacement and shaft torque. Furthermore, oil temperatures and vibrations are also monitored.

One objective of the research at TU Dresden is to achieve a better understanding of loads

and resulting stresses for thrusters by assembling a simulation model of the Wärtsilä steerable thruster WST14 and the whole test rig, using the software SIMPACK [11]. The assembly of the comprehensive simulation model starts with a torsional vibration system which is stepwise enlarged with more degrees of freedom (DOF) and modally reduced finite element (FE) bodies to a flexible MBS-model with 6-DOF bodies.

The combination of detailed simulation models and full-scale measurements of the corresponding thrusters allows the determination of inaccessible component loads such as bearing and gearing forces. The simulation model of the test rig will be used to determine the forces that have to be introduced by the hydraulic cylinders to emulate the offshore load conditions at the test stand.

As shown in figure 1, the thruster housing is mounted on the structure of the test rig and the drivetrain is connected to an electrical motor and generator. Six hydraulic cylinders can be connected to the loading unit at the end of the propeller shaft, to apply thrust and side forces. The cardan shaft allows an inclination between the thruster and the generator sided gearbox and connects the propeller shaft with the gearbox input shaft. With an electric connection between generator and motor, it is possible to feed only the power losses to the system and use the generator's energy to run the motor.

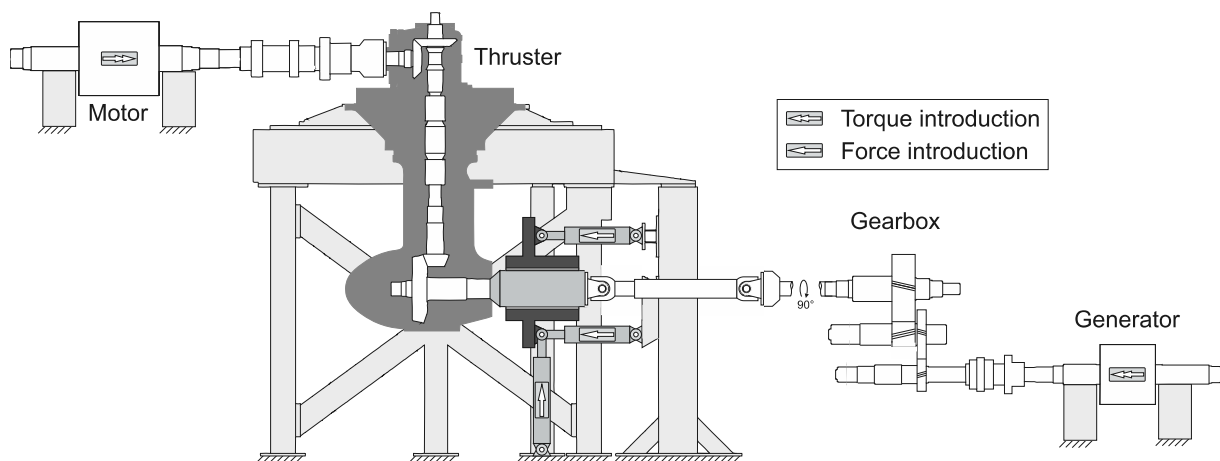


Figure 1: Schematic representation of the test rig

2 TEST RIG AND SIMULATION MODEL

The MBS-simulation allows to determine the dynamic behavior of the mechanical systems, consisting of flexible and rigid bodies. Each body has a mass distribution, allowing storing kinetic energy. With connecting spring damper elements, the energy exchanges and transfers from kinetic to potential energy. Therefore translator or rotatory spring damper elements, rigid connectors or various contacts are used. The mass and stiffness parameters of components given by their geometry, affect the natural frequencies of the system. They represent the frequencies with which the system will response on excitations. Besides the natural frequencies, the model represents the excitation due to the nonlinear characteristics of the gear contacts. The excitation of the gear contact by the variation of the number of teeth in contact and resulting changes in the stiffness are taken into account, according to ISO 6336 [6], [11]. In the MBS, the connection between shaft and housing is realized by spring-damper elements,

which can consider axial, radial and additionally the bearing moments as reactions. Depending on the purpose of the model, a constant stiffness or stiffness characteristics are used [10]. The cooperation partner SKF provides bearing stiffness matrices for each bearing and relevant load cases. The Euler-Bernoulli or Timoshenko beam theory are state of the art for fast calculation of shaft deformations and frequency responses [4]. The Timoshenko approach is used on all shafts of the test rig model, except the lower gearbox shafts. For a high level of accuracy, these shafts are modelled as FE-bodies, to include the flexibility of the bevel gear body. Since MBS-simulation tools compute with a higher resolution in the time domain, only a small number of DOF can be handled. To lower the DOF of FE-bodies, the component mode synthesis (CMS) is widely used in structural mechanics [9]. As shown in figure 2, m load introduction points (master nodes) are defined, which are connected to the nodes of the FE-model by n slave nodes. These multi-point constraints (MPC) are later used to introduce loads in the MBS. In addition to the static stiffness- and mass-matrix between the MPCs, a specified number l of natural frequencies is provided by the CMS. The usage of flexible MBS-models guarantees a more precise representation of the displacement and the excitable natural frequencies of the flexible body with a significantly lower computation time, compared to FE-simulation [2], [5].

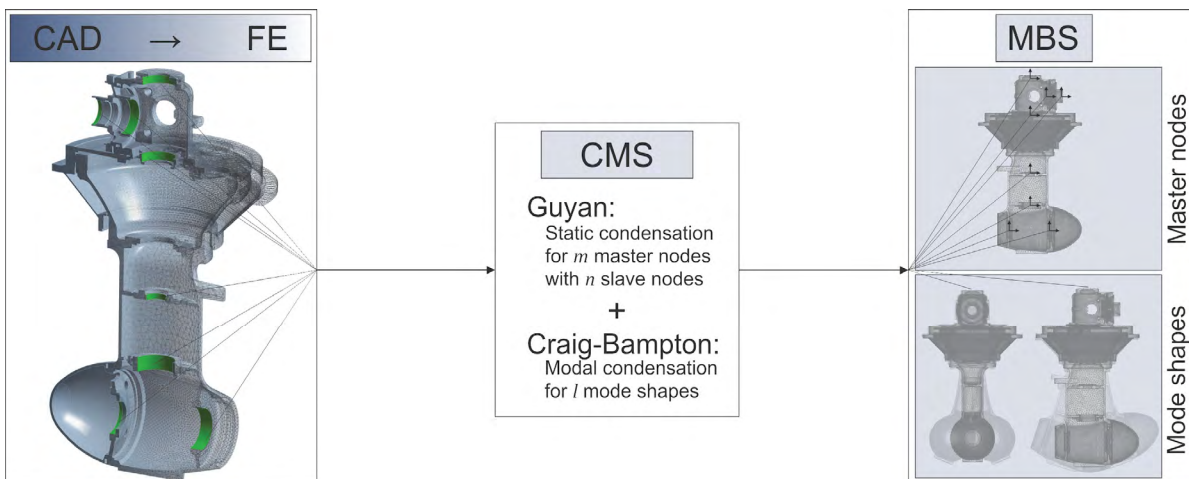


Figure 2: Workflow from CAD part to flexible MBS-body

3 MODEL VALIDATION

The test rig measurement data offer the opportunity to validate the simulation model. The validation is done by using the measurement results from modal analysis and time domain torque and displacement signals.

3.1 Frequency domain

The natural frequencies of the test rig are determined by acceleration sensors during a modal analysis. Besides the torsional frequencies, the longitudinal and transversal frequencies of the thruster housing have a major influence on the system behavior. Using information about inertia and stiffness of the drivetrain components, the torsional natural frequency can be calculated. They correspond well to the measurement results. The comparison of simulated

and measured natural frequencies of the thruster mode shapes in longitudinal, transversal and vertical direction show a good correlation. The natural frequencies of the torsional system, the 6-DOF-MBS-model and measurement results are compared in figure 3.

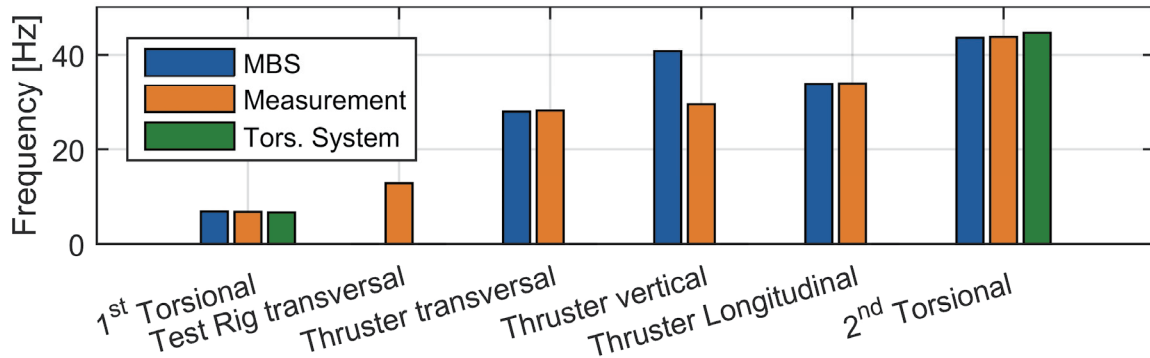


Figure 3: Comparison of natural frequencies of the test rig models

The analysis of the system behavior with a torsional system neglects further excitable frequencies. The test rig and thruster natural frequencies can only be seen in more comprehensive models. The thruster transversal, vertical and longitudinal natural frequencies are in good accordance to the measurement results, due to the considered structure of the flexible thruster housing including the horizontal test rig plate. The four stands of the test rig are not modelled, hence the test rig transversal frequency, which is of minor interest, is not included in the simulation. This rigid behavior, also in vertical direction, affects the thruster vertical mode shape, which does not match to the measurement results. The thruster longitudinal and transversal mode shapes are shown exemplarily in figure 2 (right). The modal analysis was done during stand still and without oil in the thruster housing. The damping and inertia effect of the oil is not included in the MBS-calculation.

3.2 Time domain

The complex control behavior of the motor and generator results in different torque characteristic compared to the simplified PI-control system of the MBS-model. In general, the system behaviors match, but resonance amplitudes do not correspond. Further investigations such as dampening adjustment are needed to achieve a better agreement between measurement and simulation.

To interpret the Hertzian contact stress, a precise determination of the relative bevel gear displacement is required. The use of the CMS reduced thruster housing and shafts is needed for this investigation. The lower gearbox gets loaded by thrust forces and bending moments, so that the stiffness characteristic is of high importance.

Besides the validation by measurement data, a comparison of the displacement results of the MBS-model is done using an analogous FE-model. In addition, the FE-model considers temperature distribution and bearing clearances, which are not included in a fast calculating MBS-model. On a similar level of detail of the models, this approach leads to a convincing accordance of displacement results from MBS and FE near the design torque. A slight deviation occurs at overloads and very low torque.

4 SIMULATION RESULTS

Thrusters running in icy conditions are subject to heavy load peaks. Therefore the "Finnish Swedish travel agency" TRAFI gives load cases, defining the number of amplitudes, magnitude and length of the propeller-ice interaction ([3], [7], [12], [13]). In figure 4 the load case torque characteristics and the simulated propeller shaft torque, as the systems response, is shown.

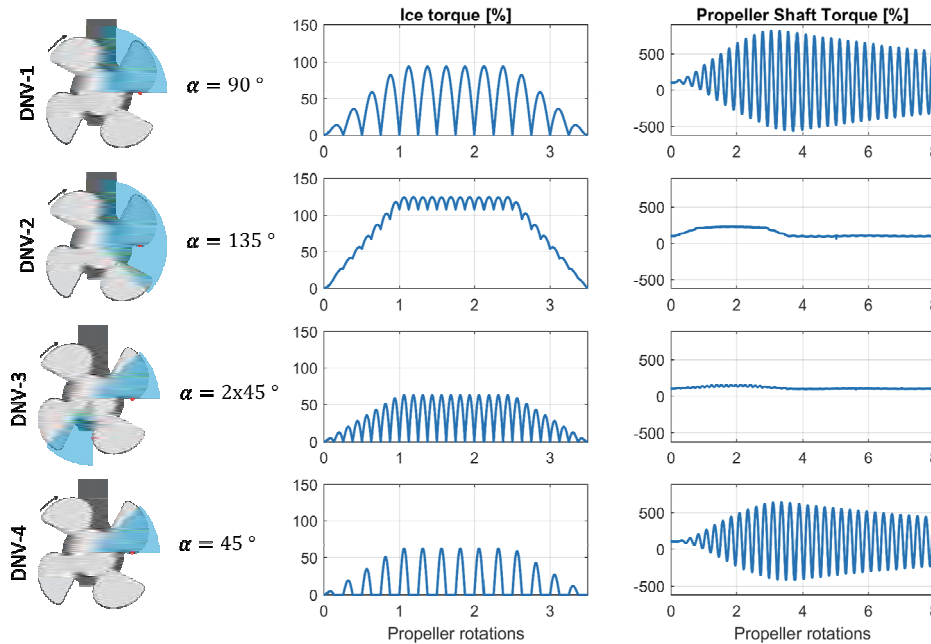


Figure 4: Ice load cases and system response

Thereby must be considered, that the test rig set up leads to torsional natural frequencies of the drive train, which are matching the blade passing frequency represented in the ice load cases. Because of that, high resonances can be seen in the ice load cases 1 and 4, since these load cases have major amplitudes at this frequency. The change of sign of the propeller shaft torque indicates a back flank contact in the lower gear stage.

To get a better understanding when back flank contact occurs, the half-sine ice load characteristic is modified for different amplitudes from 10 kNm to 70 kNm and varying contact length α from 10° to 90° . This results in a matrix of load cases. The minimum circumferential force in the lower gear stage is evaluable to identify critical load cases with back flank contact. In figure 4 the minimum force of the lower gear stage, normalized on the 100 % load case, is shown in relation to the ice torque amplitude and contact length. Besides the current WST14 design (middle), four more simulation results with smaller and bigger propeller inertia and similar loading are shown. The red marked areas represent the ice loads with back flank contact. With different propeller inertias, the 1st torsional natural frequency changes, so that resonances can be avoided and the high system responses are decreasing.

The ice loads, exciting with blade passing frequency, set the WST14 simulation model directly in resonance. This is shown by the high amplitudes seen in figure 4 for the load cases 1 and 4 and in figure 5 (middle).

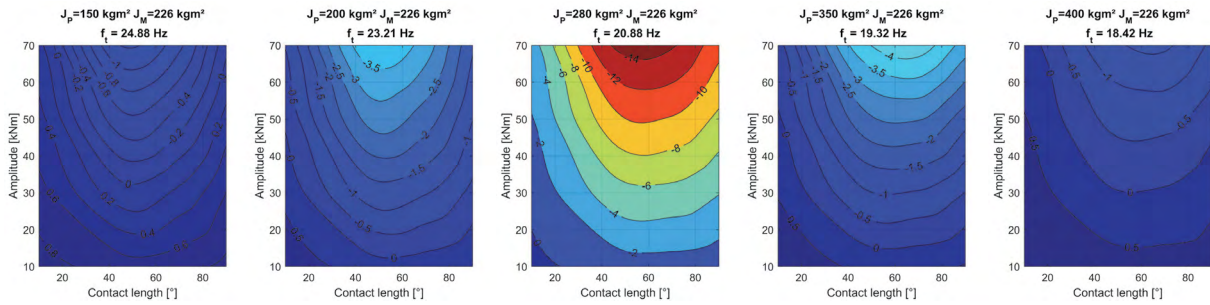


Figure 5: Load variation matrix for different propeller inertias

Besides torsional vibration the effect of bending moments on the propeller affect the gear contact. For the given thruster no information about the propeller are given, so that bending moments resulting from oblique inflow cannot be determined. With a general investigation to bending moments acting on the propeller in the range of the nominal torque, this topic is being investigated. Therefore the model is loaded with torque and different combinations of vertical and horizontal moments on the propeller. In figure 6 the definition of bending moment and resulting Hertzian contact stresses are given. The displacement of the gears is determined with the SIMPACK simulation model. Following the displacements are used to compute the contact stress with the software BECAL. The contact stress is nominated to the 100 % load case. The changes are relatively small compared to resulting stress changes due to torsional overloads.

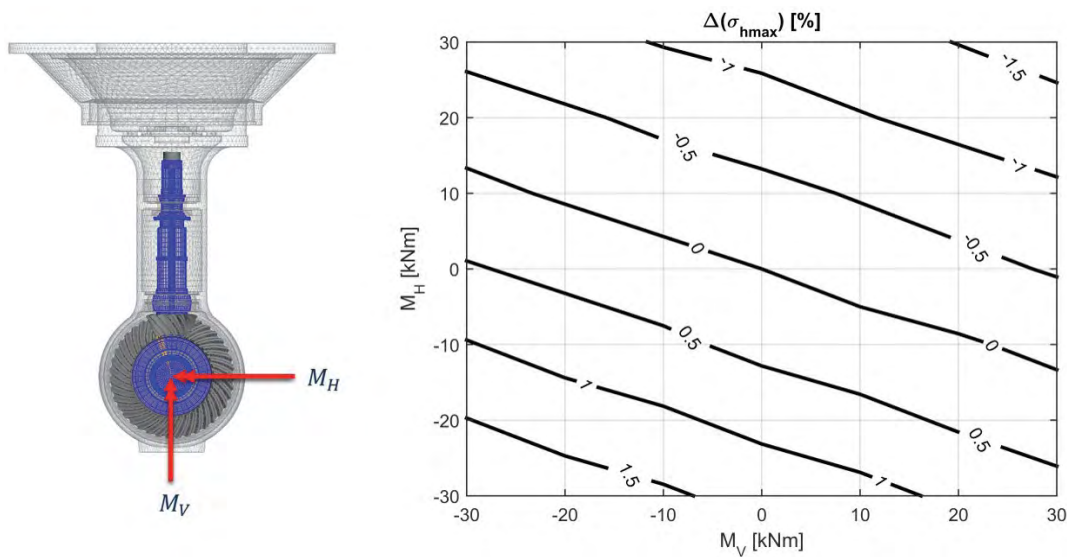


Figure 6: Hertzian contact depending on propeller bending

The Hertzian contact stress is more influenced by the horizontal moment than by the vertical moment. The contact pattern is shifted more to the heel or toe, resulting in different curvature of the flank and different lever to transmit the torque. This effect is seen within a range of +/- 1.5 % of the nominal contact stress. The small deviation of the pressure is the result of the relatively high lead crowning.

5 CONCLUSION

The test rig established by the VTT and Wärtsilä provides the opportunity to analyze the systems behavior, which cannot be done during operation of a vessel. Comprehensive overload tests were performed using different bevel gears. The resulting knowledge gives all partners a better understanding of the complex behavior of azimuth thrusters.

The flexible 6-DOF-MBS-model allows the determination of complex dynamic system behavior under consideration of flexible structures and their excitable mode shapes. The lower bevel gear stage and the flexible housing cross connect the torsional degree of freedom with the transversal thruster housing mode shape. This results in a more flexible torsional drivetrain, then determined with a simple torsional model. As a result, the determined torsional frequency drops and can be shifted in the working point, causing resonances.

The determination of relative gear displacement with SIMPACK shows good accordance to the comprehensive FE-model. A higher accuracy for overload situations is achieved with FE-simulation, when the contact pattern moves to the toe or the heel. Without the knowledge of the actual gear geometry and FE-mesh strategies, the SIMPACK model gives reliable relative displacement results and takes less computational time.

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