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Lateral Propeller Forces and their Effects on Shaft Bearings

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ABSTRACT

Out of damage cases reported to DNV, the most common machinery related damages experienced are those to propeller shaft- and engine bearings. Out of these damages, the vast majority of the damages occur in the aft most bearing. The main reason for the damages in this bearing is collapse of lubrication in the bearing caused by significant lateral forces and moments generated by the propeller. Historically, these propeller forces have not been accounted for in alignment and bearing design. In order to quantify the effects of the propeller forces on the shaft/bearing interaction, DNV has carried out a series of research projects where the lateral propeller forces have been measured directly on different vessel types in both steady state and transient operation. Analysis was carried out to investigate the effect of the propeller loads on the stern bearings of the vessel. The measurements have also been used as a benchmark to assess the predictive ability of current analytical methods. The results clearly illustrate the importance of being able to accurately predict lateral propeller forces in both steady state and transient conditions due to their significant effect on the bearing performance.

Keywords

Shaft Alignment, Propeller Forces

1 INTRODUCTION

Classical shaft alignment analysis has mainly focused on achieving optimum alignment of the shaft in static conditions. Dynamic effects affecting the alignment, such as propeller loads, have not been accounted for. The rationale for optimizing the shaft alignment in static conditions has been that the shaft alignment specification is carried out to support an alignment procedure that is conducted under static conditions. No provisions has been made for dynamic conditions because experience showed relatively few number of damages. In the beginning of the 1990's, the number of shaft bearing damages and in particular tail shaft bearing damages was found to increase significantly. The increase in bearing damages was attributed to increased hull deflections caused by more flexible hulls. In 2001, DNV initiated a joint industry research project with DSME and MAN B&W to investigate the effect of hull deflection on the alignment of a 320,000 DWT VLCC (Brodin et al 2002). The study proved the effects of hull deflections, but further identified the lateral propeller loads to have a significant effect on the tail shaft bearing in particular. A continuation of the study was carried out with DSME and Kristen Navigation to investigate the effect of propeller loads on the sterntube bearings of a 320,000 DWT VLCC (Vartdal 2006). The study proved the significance of the propeller load and a similar study was carried out for a container vessel in cooperation with HHI, HMM and Wartsila (Vartdal 2008). Similar DNV studies have also been carried out for twin screw LNG's. The studies have all shown that the lateral propeller forces can be very significant in a number of vessel operating conditions. This paper presents the magnitude of the propeller forces and the possible influence of these on the tail shaft bearing. This will highlight the importance of predicting these forces and the merits of the present state of the predictive tools.

2 MEASUREMENTS OF LATERAL PROPELLER FORCES

The lateral propeller forces and moments are generated as a result of the wake field as experienced by the propeller. This wake field, and hence the propeller loads are dependent on the operating condition of the vessel. The wake field for a vessel going straight ahead is virtually constant, whereas the wake field for a vessel which is turning will be changing with time. These are therefore termed steady state and transient operating conditions respectively.

The propeller forces were measured by attaching strain gauges on a cross section of the shaft which was positioned between the propeller and the tail shaft bearing. The strain gauge signals were monitored in vessel operation through telemetry.

The measurements also included proximity sensors located just aft of the tail shaft bearing, monitoring the movements of the shaft relative to the bearing.

A significant shortcoming of the measurements is that they do not distinguish between the contribution to the bending moment measured at the shaft from bending moment and transverse forces generated at the propeller. The significance of the propeller generated forces and moments are still clearly illustrated by the results.

The results presented on the bending moments measured at the shaft just in front of the propeller are non-dimensionalised by $\rho\omega^2D^4$ where ρ=density of water; ω=rotational speed of propeller and D is the diameter of the propeller. The notation is according to Figure 1.

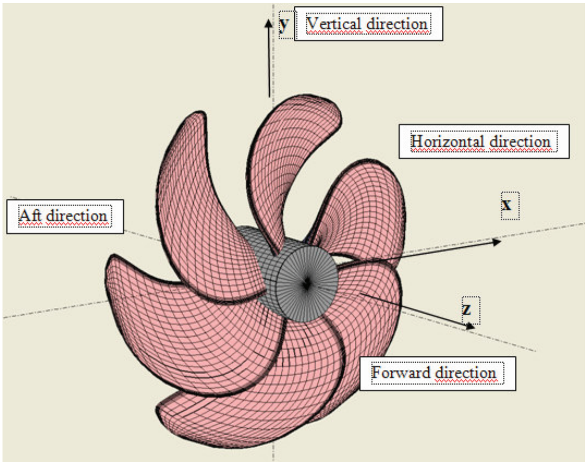

Figure 1 Notation

The propeller generated nominal bending moments measured at the shaft for a vessel operating at different engine powers in straight ahead condition is given in for a few vessel types in Figure 2.

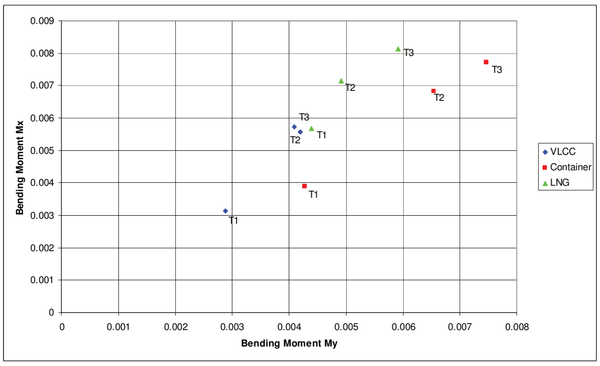

Figure 2 Ahead running propeller moments

T1,T2 and T3 denotes that the vessel is running at 50,75 and 90% of maximum continuous rated power respectively. All the propellers are right handed and the figure shows that the direction of the propeller loading is the same for all the vessel types, but of different magnitude. The propeller generated forces are seen to lift the propeller and push it towards starboard.

Figure 2 shows the nominal propeller forces. The amplitudes of the bending moment caused by blade order excitations are illustrated in Figure 3. The circles in the graph represent the variations of bending moments where the centre of the circle is the nominal bending moment, and the area represents the amplitude. This shows that the blade order amplitudes of bending moment variation are small to moderate. The effect of the propeller induced forces in this condition is also reflected by the proximity transducer measurements, monitoring the position of the shaft relative to the aft sterntube bearing. These measurements are shown for a limited time sequence in

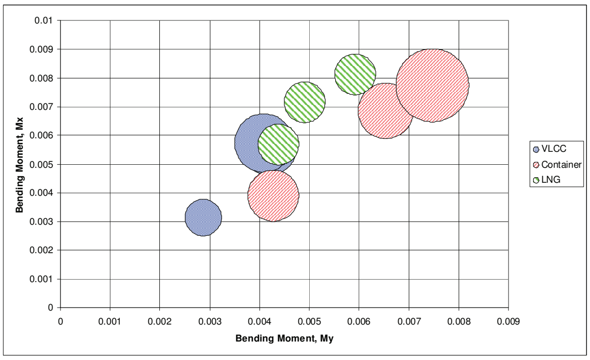

Figure 3 Bending moment amplitudes in ahead running

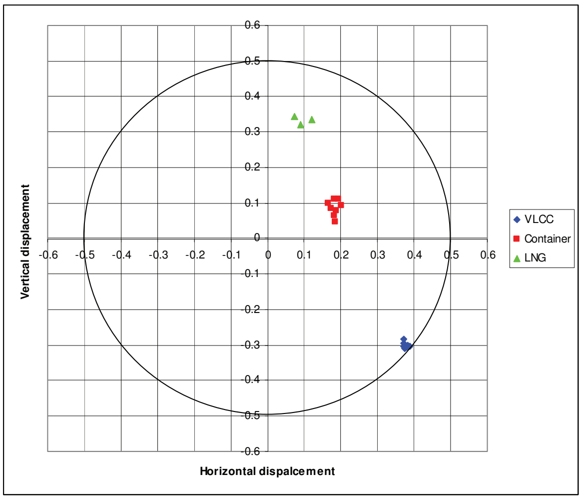

Figure 4 Proximity transducer measurements at tail shaft bearing

iFigure 4. The data are non dimensionalised by the bearing clearance and the figure illustrates the position of the shaft relative to the bearing clearance as illustrated by the circle. The position of the shaft is a function of the propeller weight and the propeller forces and also the alignment of the shaft which determines the pressure distribution in the tail shaft bearing. The figure indicates that for the container vessel and the LNG in normal straight ahead running conditions there is no contact in the aft part of the aft sterntube bearing. This means that the propeller generated forces are exceeding the effect of the propeller weight for these vessels. This was not the case for the VLCC. However, as indicated, such effects are vessel specific due to the dependence on the alignment so it may vary from vessel to vessel.

Measurements were also carried out when the vessels were turning. The measurements showed a significant variation in the bending moments throughout the turns for all vessel types. The variation in propeller generated nominal bending moments during turning of the vessel is illustrated by plotting the magnitude and direction of the bending moment against the time elapsed from the initiation of the turn. The direction is defined such that 0º is along the –x direction and increasing angles counter clockwise as seen from the aft. The resultant bending moments during a starboard turn are illustrated in Figure 5.

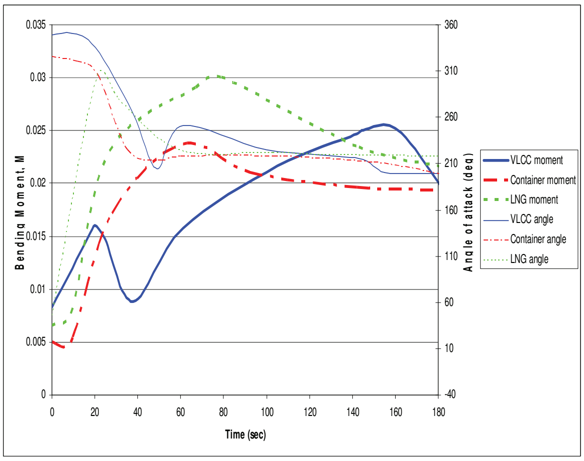

Figure 5 Propeller forces during starboard turns

The figure shows the significant variation in magnitude and direction of the propeller loads during a starboard turn. The vessels were all running at 90%MCR into the turn and the maneuver was carried out with 35° rudder angle. The angle turned after 3 minutes is different from vessel to vessel, but as an indication the VLCC has turned through approximately 95º, the container vessel through 105º and the LNG through 120º. The figure shows that for the LNG and the container vessel, the magnitude of the propeller load is increasing for the first part of the turn and then after reaching a maximum value, slowly decreasing. The corresponding direction is observed to be unstable before the maximum value is reached and then steady. For the VLCC the magnitude of the propeller force is varying throughout the measurement and reaching a peak value after approximately two and a half minutes whereas the direction is relatively stable after 1 minute of turning.

The magnitude of the propeller forces reach very high values for all vessel types, but are observed to be particularly large for the LNG vessel. This is most likely because the LNG vessel is a twin screw vessel. In this turn the propeller is the outboard propeller and this will significantly affect the wake field experience by the propeller.

The propeller forces are forcing the propeller down and towards port during the starboard turn. This is confirmed by the proximity transducer measurements as shown in Figure 6. This shows the movement of the shaft at certain intervals from the initiation of the turn. The initial position of the shaft should be similar to what was measured in straight ahead condition.

The propeller forces during a port turn are illustrated in Figure 7. For the port turn the magnitude of the propeller loads are seen to increase after an initial dip in the values and the peak value is reached at various times into the turn. After the peak value is reached the magnitude is rapidly decreasing for all vessel types. The peak load is very high for the VLCC in this maneuver and more moderate for the other two vessel types.

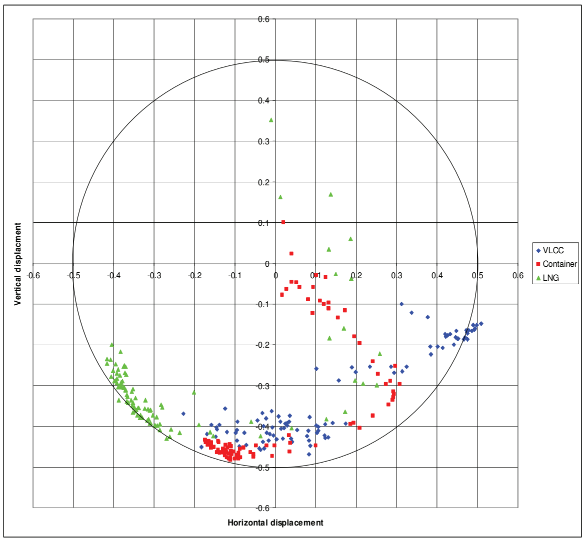

Figure 6 Proximity transducer measurements at tail shaft bearing during starboard turn

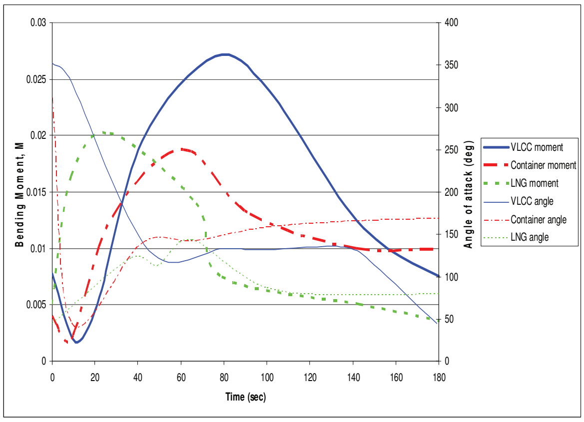

Figure 7 Propeller forces during port turn

The load direction is unstable in the beginning of the turn and is then seen to stabilize. The port turn is seen to force the propeller up and towards port for all the vessel types. This is confirmed by the proximity transducer measurements as shown in Figure 8. The positioning of the shaft in the bearing is observed to be quite different for the different vessel types.

3 EFFECT OF LATERAL PROPELLER FORCES ON TAIL SHAFT BEARING

A finite element analysis was carried out to investigate the effect of the lateral propeller forces on the pressure distribution in the tail shaft bearing. The analysis presented here is for the LNG vessel presented above, but the results are similar for the other vessel types. A model was created for the aft part of the shafting system of the vessel, and detailed contact analysis was carried out for the tail shaft bearings. The boundary conditions of the model are completely defined by the strain gauge measurements. The lubrication was not accounted for in the analysis. This will have a significant effect on the results, but the analysis presented here is intended to

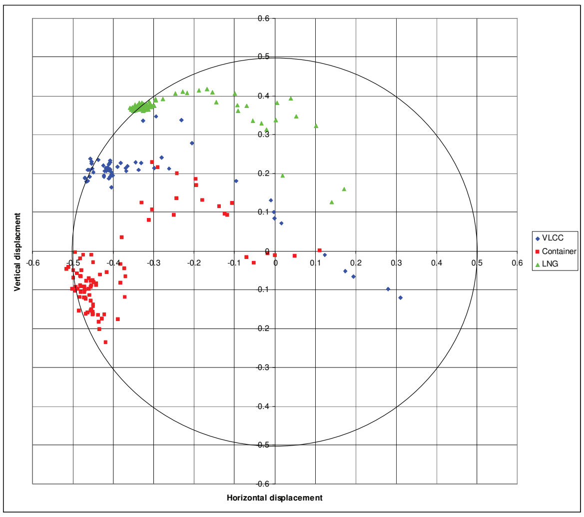

Figure 8 Proximity transducer measurements at tail shaft bearing during port turn

illustrate the challenges related to propeller loading rather than presenting the specific results. The finite element model in way of the sterntube bearings is shown in Figure 9.

Figure 9 Finite element model

In static condition where there is no influence of propeller load, the pressure distribution in the tail shaft bearing is shown in Figure 10. The analysis shows that contact exists between the shaft and the bearing in the very aft and very forward end of the bearing only due to the curvature of the shaft. The distribution of pressure between the aft and forward part is dependent on the shaft alignment selected. In this example the maximum pressure is in the aft part of the bearing. Figure 11 shows the pressure distribution in the tail shaft bearing during normal continuous ahead running condition. The propeller forces are observed to lift the propeller such that the contact disappears in the aft part of the bearing. The entire load in the bearing must therefore be carried by the forward part of the bearing.

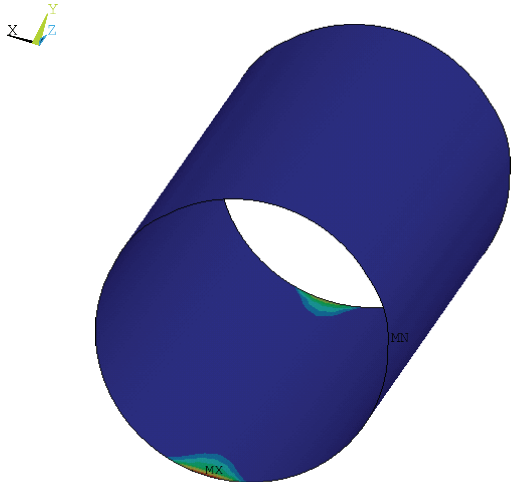

Figure 10 Contact pressures in tail shaft bearing in static condition

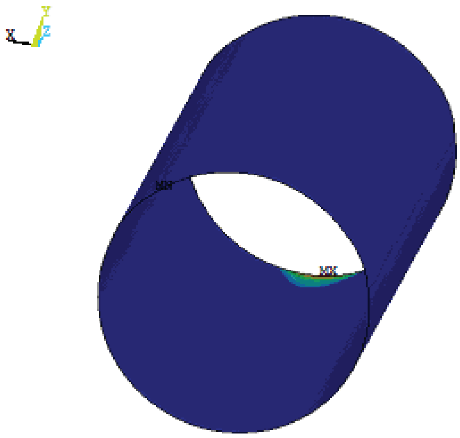

Figure 11 Contact pressures in tail shaft bearing for normal continuous running

The peak pressure in the bearing is increased by 20% compared to that in the static condition.

The pressure distribution in the tail shaft bearing for an extreme load during a starboard turn is shown in Figure 12. The view of the bearing has been altered in order to illustrate the contact pressure distribution. The figure shows that the propeller forces have forced the shaft towards the port side in the aft part of the bearing as was observed by the measurements. A corresponding shift in the pressures towards the starboard side is seen in the forward end of the bearing due to the shaft deflections caused by the propeller loads. The maximum pressure is seen in the aft end of the bearing and the maximum pressure in this condition is almost 4.5 times that of the maximum pressure in the static case.

The pressure distribution in the tail shaft bearing for an extreme load during a port turn is shown in **Figure 13**.

The figure shows that the shaft has been lifted and moved towards port by the propeller forces. The gap in the contact pressures in the aft part of the bearing is caused by the oil grove running axially along the bearing at both sides. The pressures are increased by the interference with the grove since the contact area is reduced. The pressure peak in the aft part of the bearing for this turn is almost 4 times that of the maximum pressure in the static case.

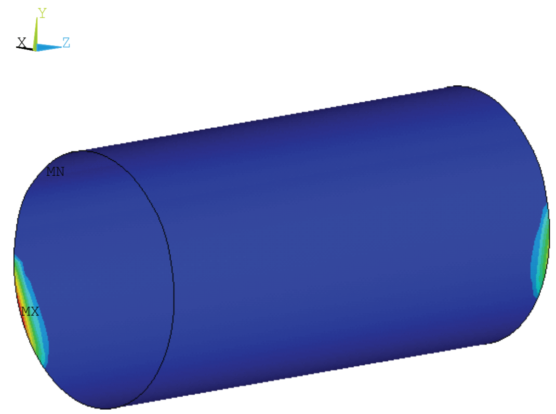

Figure 12 Contact pressures in aft sterntube bearing for starboard turn

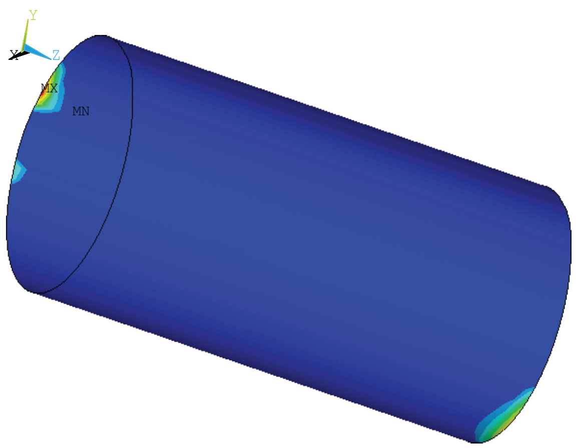

Figure 13 Contact pressures in aft sterntube bearing for port turn

The contact in the forward end of the bearing is seen to be located in the lower half of the bearing.

The contact pressures are critical if they are sufficient to break the lubrication and cause boundary lubrication or direct contact between the shaft and the bearing. The increase in friction will then cause heating. If the heat transfer in this area is not sufficient to compensate for this heat formation, the bearing material will be overheated. Such heating will most likely lead to bearing damage.

This study has shown that the bearing pressures experienced during turning far exceeds those experienced during normal continuous running. In classical shaft alignment analysis, the propeller forces have either not been accounted for or have been calculated in normal continuous running. Based on the results seen here it is evident that prediction of the propeller forces in turning conditions may be necessary to ensure safe operation of the tail shaft bearing in all operating conditions.

4 CFD MODEL OF NOMINAL FLOW AROUND THE TWIN SCREW LNG VESSEL

In order to estimate the bending moments imposed on the propulsion shaft by the rotating propeller, the flow quantities in which the propeller is working and interacting with must be properly determined.

During an arbitrary turn, a given ship goes through a continuum of speeds, rotation rates about its own axis, shaft rpm's and drift angles. In order to roughly estimate the flow during such conditions in a fast and simple manner, some simplifications must be made. In this paper, the following assumptions have been made:

- The propeller is assumed to be properly submerged for its inflow to be independent of free surface and gravity effects.
- Cavitation is assumed not to influence the results, hence the flow is purely one phase.
- The transient effects emerging from the constant change in acceleration of the ship itself is negligible.
- The boundary layer on the hull can be to some extent represented by a two-equation turbulence model (except from in the aft third part of the ship).
- The two propellers do not interact with eachother.

Given these simplifications one can easily see that the model only describes one infinitesimal moment in time during the turn, i.e. it is a quasi steady model.

These assumptions are in this paper going to be used to construct a model describing one of the conditions that give the extreme bending moments on the propeller shaft of the twin screw LNG.

The motion of the ship at the time the propeller loads were most critical was extracted from logs containing ship speed, position of the ship, yaw rate, rudder angle, heading and rpm as a function of time.

The first simulation consists of a quasi steady flow simulation of the bare hull by using a rotating frame of reference with a trimmed hexahedral mesh. The discretized finite volume domain consists of a box of length 4L, depth of L and width of 4L where L is the length of the ship. Moreover, the cell size on the ship surface is kept at 1.5 percent of the breadth of the ship but refined around and between the skegs in order to capture the wake. 15 prism layers is grown perpendicular to the ship, where the thickness of the layers are chosen to give the mesh a y+ value of 25-80 and give a smooth transition to the external mesh. This resulted in a total of approximately 5.5 million finite volume cells.

The surface was fixed at a position 6% above the ballast draft and defined as a symmetry plane. Menter's k-omega SST turbulence model was used to capture the turbulent effects of the flow. The equations of motion were solved for to second order in both time and space. The quantity monitored for convergence was the velocity profile of the wake behind both skegs. Courant numbers were kept in the range of 0-5 after some initial iterations with about 3 times larger time-steps.

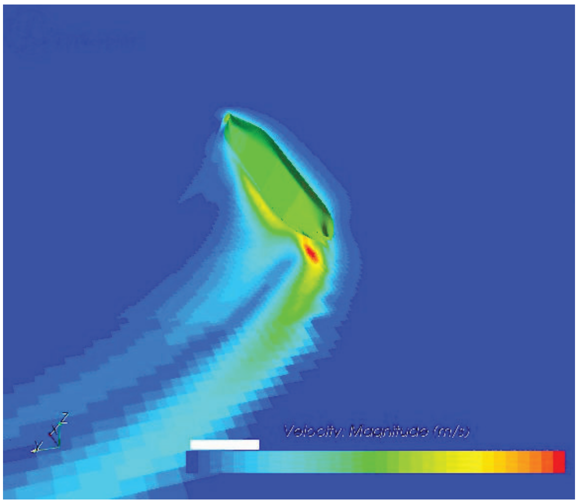

Figure 14: Velocity profile on the fixed surface

Now, all flow quantities in the domain are extracted and the vectorial quantities are transformed to a coordinate system translating and rotating with the hull.

The mesh is further cut at approximately 1/3L fore of the aft perpendicular and a distance L from the ship at port- and starboard-side and below the ship. The boundary conditions for this new domain are imposed from the previous simulation with velocity fields on four sides and a pressure field on the final side bounding the domain. The centrifugal and Coriolis force is inserted manually into the domain by momentum sources in all cell centers, and a steady state simulation was set up with this cut domain to verify that the velocity fields in the two regions were consistent.

5 ANALYSIS OF THE PROPELLER GENERATED SHAFT BENDING MOMENTS

One propeller with radius R is inserted into a cylinder of thickness 1.5 R and a radius of 1.5 R. This cylinder is further placed behind the starboard skeg such that the propeller is located about 1/4 R aft of its true position due to practical meshing constraints. The rudder was placed at the correct location relative to the propeller. The domain was again meshed with refinements 2 propeller diameters fore and aft of the propeller. The cylinder surrounding the propeller contained about 4.5 million cells, while the external mesh contained about 2.2 million.

The solver parameters were unchanged except from a drastic decrease in time-step (approximately 75 per blade passing) and the turbulence model was changed to the "Linear Pressure Strain Reynolds Stress model" implemented in Star CCM+. The bending moments were used to monitor the convergence of the system, and the results are shown in Figure 2.

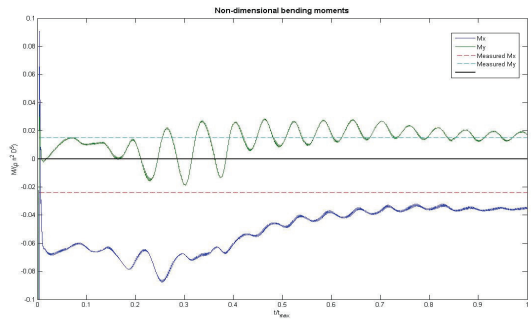

Figure 16: Measured and simulated bending moments (non-dimensional)

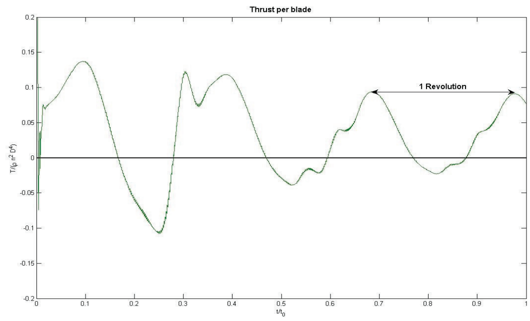

Figure 15: Thrust per blade (non-dimensional)

The thrust per blade is also monitored, and the result is plotted in Figure 3.

From Figure 3, one can see that the load per blade still changes slightly between the two last revolutions, meaning the solution has not converged completely. However, by considering Figure 2 this seems to only affect the magnitude of the dynamics of one of the moments, and not the mean value.

From Figure 2, one can clearly see that the model is unable to accurately predict the moment about the horizontal axis. One reason may be that the horizontal velocity from right- to left hand side in Figure 4 is wrongly estimated in the model, given that the mesh on the propeller truly captures the inflow correctly. Since the propeller was inserted ¼ R aft of its true location, it may experience a higher velocity in the transversal direction due to the increased distance to the boundary layer around the hull which will obviously deflect the water. The maximum thrust load on the blades is reached when the blade is pointing downwards and meeting the flow in Figure 4. This max peak may be reduced due to cavitation thus reducing the moments about the horizontal axis. Moreover, free surface effects could also influence the results. The condition itself may also induce errors since transient effects from the constant change of accelerations of the hull are neglected.

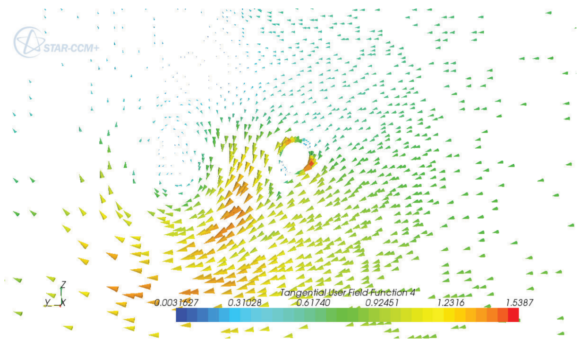

Figure 17:Normalized Wake field 1/4R fore of the propeller

Another source of error may be that the Coriolis term inserted in the model is implemented such that the body forces emerging from the fictious rotation of the entire system are all the time referring to the velocities from the previous iteration. When each time-step is completed, the mesh on the propeller moves, and due to the no slip boundary condition the Coriolis force will start with a wrong direction. Thus the number of inner iterations per time-step should possibly have been increased to avoid such problems.

However, even with all the deficiencies above, the results are giving the right trends and a fair estimate of the bending moments on the propeller shaft.

5 CONCLUSIONS

This study has shown the significance of the lateral propeller forces generated in vessel turning on the performance of the tail shaft bearing of the vessel. The CFD analysis carried out has also shown that it should be possible to predict these forces with reasonable accuracy. Damage experience has shown that a large proportion of tail shaft bearing damages occur during turning of the vessel. In order to avoid such damages occurring in the future the transient forces during vessel turning may have to be accounted for in the shaft alignment analysis. However, this type of damage is not relevant for all types of vessels and the relevance of such calculations should therefore be considered on a case by case basis.

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