A framework for holistic roll damping prediction

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ABSTRACT

In this paper a framework for holistic multi-tier roll damping prediction is presented. The approach provides a platform for best possible prediction given the different stages in the ship design process. Starting from the earliest design stage a semi-empirical model gives the foundation for a complete model that is applicable for all possible loading conditions and operational conditions. The components in the model are continuously updated with input from CFD calculations and model tests when available, and finally prior to delivery of the ship the model is assessed and tuned based on full scale trials. The approach is well suited to be used as roll damping input in operational guidance systems as well as to provide feedback to the design process in a systematic manner.

Keywords: *Roll damping, Roll decay, Ikeda's method, Full-scale, Model-scale, Holistic, Extrapolation, Operational guidance*

1. INTRODUCTION

Accurate roll damping modeling is crucial to assess and control vulnerability to critical roll responses both in the design stage and in the operation. Yet, the roll damping is rarely given sufficient attention (if any) in the design process when it comes to hydrodynamic optimization.

In a typical design process the vast majority of the hydrodynamic focus is put on predicting and minimizing the power requirement of the vessel. In most cases these efforts are concentrated to one single design point, reflecting the speed and loading condition that is stipulated in the new building contract. Semi-empirical methods are normally used for the first power predictions in the conceptual stage. This may involve established methods such as Holtrop Mennen (1982) or in house methods based on reference hulls. The second stage of the process normally involves hull line optimization using Computational Fluid Dynamics (CFD) and in the third stage the most promising hull shapes are evaluated using model tests. Typically one or two hull form alternatives and several propeller and rudder configurations are tested in the towing tank. Based on these tests full scale predictions are updated using well established transparent extrapolation procedures such as ITTC (1999). Prior to the delivery the vessel is taken out on sea trial where a speed trial is conducted. For practical reasons the speed trial is normally performed in ballast draught and evaluated for the contractual condition using procedures such as ITTC (2014) where weather effects and load case effects are eliminated. Throughout this process a power performance model is continuously updated and ultimately finalized after the sea trial, prior to the delivery of the ship. For design houses and ship yards the speed trial is a key event as contractual figures are assessed and feedback is given to the design process. A schematic picture of the different stages of the design process is given in figure 1.

If the roll damping has been given any attention in the design process this has likely been done in the model test stage by carrying out roll decay tests. At this stage the hull lines are more or less set and it is normally too late to make any drastic changes. For practical reasons the roll decay tests are likely carried out in the design condition only and the non-dimensional roll damping is evaluated from the decays and assumed to be valid for the full scale vessel, typically regardless of condition. However, the design condition does not necessarily have to be a realistic service condition and normally describes the vessels' performance in calm weather. For many ship types, ocean going vessels in particular, the loading condition and speed can be different for every voyage. Furthermore, the operation is certainly not limited to calm weather.

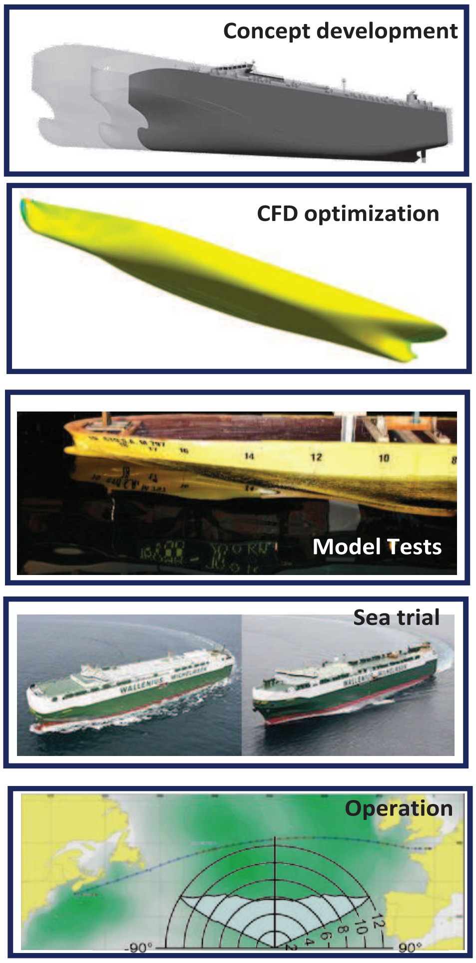

Figure 1: Illustrating the different stages in the design process

This paper presents a framework for a holistic multi-tier roll damping prediction approach where a roll damping model, that is applicable to all possible operational conditions, is developed and improved throughout the design and building process. The model is established in the earliest concept development stage and continuously improved all the way to sea trial and the delivery of the ship and is suitable to be used as input for operational guidance. The roll damping is threated on component basis and the extrapolation and tuning of these components is inspired by the well-established power prediction extrapolation procedures, such as ITTC (1999). The idea with this approach is, besides providing a platform for best possible prediction given the different stages in the design process and for the vessel in service, also to provide feedback to the design process in a systematic manner.

2. THE HOLLISTIC APPROACH

For the concept development stage the only feasible approach for estimating the roll damping is semi-empirical methods. Ikeda's method is the most established semi-empirical method and the damping is estimated as the sum of the following components:

hull lift	$\zeta_{Lift}(V)$,
bilge keel	$\zeta_{bk}(\omega_E, \varphi_a, V)$,
hull friction	$\zeta_{friction}(\omega_E, \varphi_a, V)$,
eddy making	$\zeta_{eddy}(\omega_E, \varphi_a, V)$,
wave damping	$\zeta_{wave}(\omega_E, V)$.

Besides the hull main parameters for the considered floating condition these components are also dependent on $\omega_E, \varphi_a\ and\ V$ which is the natural roll frequency, roll amplitude and forward speed. As load case specific components are considered the model is useful to identify operational conditions that may require particular attention and provides a good foundation for the hollistic roll damping model.

2.1 Updated Lift and Aerodynamic damping

Ikeda's original method (1978) as described in Himeno (1981) and ITTC (2011) gives physically relevant estimates but quantitatively not satisfying levels for unconventional designs such as modern volume carriers. However, the method can be significantly improved with small modifications of the hull lift component. Figure 2 shows a comparison for a modern Pure Car and Truck Carrier between model tests and Ikeda's bare hull damping where the hull lift coefficient has been estimated with non-viscous CFD and applied together with Yomuru's original expressions for the levers of the lift force and the effective angle of attack. As seen, satisfying agreement with model tests is obtained. As practically the same calculation model that is used for the power predictions can be used to obtain the lift coefficient of the hull the additional work to provide required input for this estimate is fairly limited.

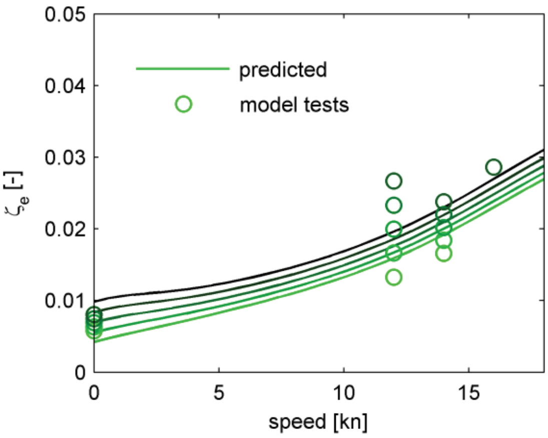

Figure 2: Predicted damping for the bare hull of a Pure Car and Truck Carrier in model scale using Ikeda's method with the lift damping component is estimated with hull lift coefficient from non-viscous CFD together with Yomuru's original expressions for the levers of the lift force and the effective angle of attack, compared with model tests for roll amplitudes of 2 to 10°.

In Söder et al (2015) it was demonstrated that aerodynamic damping not always is neglectable relative to hydrodynamic damping and therefore preferably shall be considered. The estimation of this component however requires input on the aerodynamic lift coefficient of the hull. This coefficient can either be estimated at an early stage from reference hulls or from CFD. Thus ζ_{lift} and ζ_{aero} can be replaced by $\zeta_{lift_{CFD}}$ and $\zeta_{aero_{CFD}}$. An updated roll damping model can thus be given by

$$\zeta = \zeta_{lift_{CFD}} + \zeta_{bk} + \zeta_{friction_S} + \zeta_{eddy} + \zeta_{wave} + \zeta_{aero_{CFD}}. \quad (1)$$

In Ikeda's model only the frictional component has a scale dependence and sub-index S here denotes full scale.

2.2 Extrapolation of model tests

Free roll decay model tests can be performed in the towing tank with the same model as used for the power predictions. Model tests at speed are typically performed with the same *Froude* number as the full scale vessel so the wave pattern shall be the same in the two scales. Currently there are no established scaling procedures for roll damping model tests. According to IMO (2006) scale models with bilge keels shall have a minimum length of 2m, the bilge keel height shall exceed 7mm and the scale factor shall not be larger than 1:75 to avoid viscous scale effects. As typical models at the established towing tanks often measures some 6 to 7m these requirements are normally fulfilled.

However, as the Reynolds numbers are different neglecting viscous scale effects is questionable, especially when the damping is low and the bilge keels are small. Worth noting here is also that model tests intended for power predictions are normally performed without bilge keels due to the uncertainties related to the viscous scale effects. An attempt is therefore made here on proposing an extrapolation procedure for model tests. The bare hull damping and the bilge keel component is threated separately and model test with and without bilge keels are required.

2.2.1 Bare hull extrapolation

To evaluate the bare hull damping a similar procedure as used in the ITTC (1999) power prediction extrapolation procedure is suggested. In those procedures the wave component, which is considered scale independent, is basically derived by deducting a semi-analytical expression for the viscous (and form) components. In a similar manner it is proposed to evaluate the wave damping component according to

$$\zeta_{wave_m} = \zeta_{bh_m} - \left(\zeta_{Lift_{CFD}} + \zeta_{friction_m} + \zeta_{eddy}\right) \quad (2)$$

where ζ_{bh_m} is the evaluated damping of the bare hull from the model tests and $\zeta_{friction_m}$ is the frictional component in model scale. In Ikeda's method the eddy component is not dependent on the Reynolds number which could be questioned. However, for simplicity the same assumption is made here.

Based on the result for the model tested load case a tuning function $k_{wave}(V)$ is used to tune the expression for the linear potential damping that was used in the earlier stage for best match with the evaluated wave damping for the tested case. The tuning function is obtained by minimizing the difference between the evaluated wave damping and the product of the tuning function and the linear potential damping $\zeta_{wave}(\omega_E, V)$ according to

$$\min_{k_{wave}(V)} \overline{\zeta_{wave_m} - k_{wave}(V) \cdot \zeta_{wave}(\omega_E, V)} \quad (3)$$

The full scale wave damping component can then be estimated as

$$\zeta_{wave_S} = k_{wave}(V) \cdot \zeta_{wave}(\omega_E, V). \quad (4)$$

The tuning function derived for the tested load case is thereafter held constant for other load cases.

2.2.2 Bilge keel extrapolation

To investigate how the bilge keels are subjected to viscous scale effects the boundary layer thickness at the bilges are studied for an actual hull shape. CFD calculations are performed in ANSYS with a 230m Pure Car and Truck Carrier in model scale 1:30 and full scale. The calculations are performed with a boundary layer mesh corresponding to y+ ~1 in model scale and y+~100 in full scale and with standard wall functions. Due to simplifications introduced with the wall functions in full scale in particular the results need to be considered with care. The boundary layers are shown in figure 3, as seen the differences in boundary layer thickness are remarkable. When considering that a typical bilge keel height of this kind of vessel is some 0.4 to 0.8m deep in full scale (or 1 to 3% of the breadth) it appears that scale effects needs to be considered even if IMO's guidance is met.

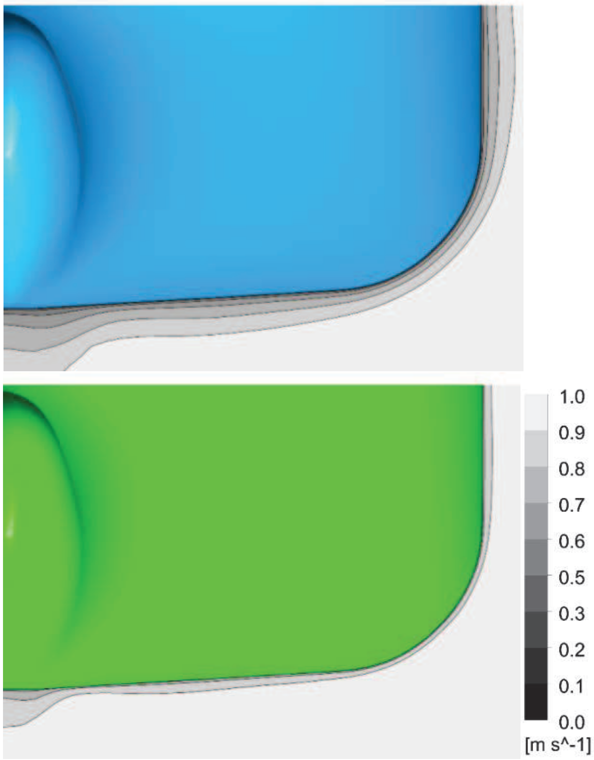

Figure 3: Comparing the boundary layer thickness in model scale at the top and full scale at the bottom for a 230m PCTC. Results are normalized and corresponds to Reynolds number that give the same Froude number, full scale speed 10kn.

In a greatly simplified manner it is investigated how the bilge keels could be affected by the different conditions by evaluating how the 2D drag of a 0.4m high flat plate perpendicular to a wall is dependent on the boundary layer thickness. Conditions are set to represent typical local Reynolds number Re_x of full scale and model scale bilge keels given a scale factor of 1:30. The velocity fields are shown in figure 4 where also a third case without boundary layer is added.

Figure 4: Comparing the boundary layer thickness over a wall where a 0.4m high flat plate is located perpendicular to the flow. Conditions are set to represent typical local Reynolds number in way of bilge keels in model scale (1:30), in full scale and without boundary layer (symmetry boundary condition).

For this specific case the calculations suggests that the drag coefficients of the plate in full scale is some 50% higher than in model scale. In addition, without any boundary layer (symmetry b.c.) the drag increases with additionally 70% relative full scale. In view of these results it is proposed to consider the scale effect of the bilge keel damping when extrapolating model tests. The following procedure is proposed.

The damping of the bilge keel component in model scale ζ_{bk_m} can be estimated as

$$\zeta_{bk_m} = \zeta_{tot_m} - \zeta_{bh_m} \tag{5}$$

where ζ_{tot_m} is the damping of the hull fitted with bilge keels in model scale. With a similar procedure as for the wave component a tuning function $k_{bk}(V)$ is estimated as

$$\zeta_{bk_m} = k_{bk}(V) \cdot \zeta_{bk}(\omega_E, \varphi_a, V) \tag{6}$$

The scale correction is estimated as the ratio between the mean dynamic pressure over the full scale bilge keel and the model scale keel according to

$$S_{bk} = \int_0^{\delta_S} \left(\frac{u(z)}{U_\infty}\right)^2 dz_S \Big/ \int_0^{\delta_m} \left(\frac{u(z)}{U_\infty}\right)^2 dz_m. \tag{7}$$

The velocity profile and the boundary layer thickness δ at the bilge keels can either be estimated using CFD or in a simplified manner based on Prandtl's $(1/7)^{th}$ power law together with the local Reynolds number Re_x at a longitudinal position x according to

$$\frac{u_z}{U_\infty} = \sqrt[7]{\frac{z}{\delta}} \tag{8}$$

and

$$\delta \approx 0.385\, x/\sqrt[5]{Re_x}. \tag{9}$$

The bilge keel damping in full scale can then be estimated according to

$$\zeta_{bk_S} = k_{bk}(V) \cdot \zeta_{bk}(\omega_E, \varphi_a, V) \cdot S_{bk} \tag{10}$$

2.3 Full scale assessment

The roll damping model for the full scale vessel is now given by

$$\zeta = \left(\zeta_{Lift_{CFD}} + \zeta_{bk_S} + \zeta_{friction\,S} + \zeta_{eddy} + \zeta_{waves_S} + \zeta_{aero_{CFD}}\right) k_{corr}. \tag{11}$$

where k_{corr} is an overall tuning coefficient or correction factor.

To assess the model and establish k_{corr} full scale trials needs to be performed. In Söder et al. (2012) full scale roll-decay tests were performed by inducing roll motion using controlled rudder impulses. This approach is suitable to use here and a sample roll decay test is illustrated in figure 5.

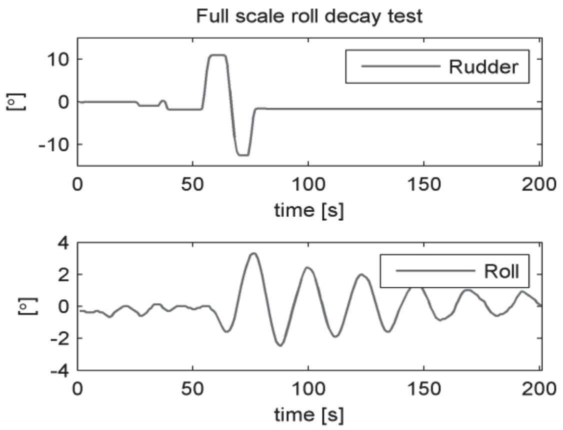

Figure 5: Time series of rudder angle and roll angle during a full scale roll decay test onboard a Pure Car and Truck Carrier.

The tests could preferably be carried out prior to delivery during the ordinary sea trail, for instance during the speed tests after completion of each speed measurement during the speed runs. This is particularly suitable as double runs typically are carried out with and against the wind direction so uncertainties related to the wind damping can be minimized.

3. EVALUATION

In figure 6 the roll damping for a Pure Car and Truck Carrier, as given by the complete model is illustrated together with model test and full scale results. In this case the model tests and full scale tests were carried out at virtually the same load case. The weather condition during the full scale trials was calm so the aerodynamic damping was negligible. The results from the complete model are given without overall correction factor as well as with correction factor. As seen there is a fairly large gap between these two curves which requires further attention. Scale effects not properly accounted for or biases in the test setup are likely causes which need to be investigated thoroughly.

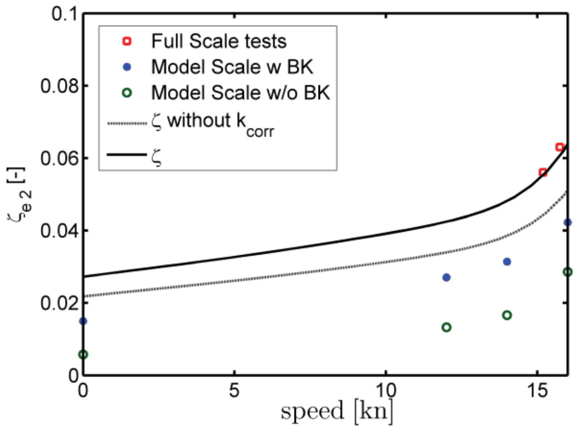

Figure 6: Damping as evaluated from model tests, full scale tests and the complete model. All three methodologies with virtually the same load case. The linear equivalent damping at 2° is given for all cases.

To demonstrate application of the holistic model it is used to estimate the damping for two "off design" conditions for the same vessel, a partial loading condition and a scantling condition. The linear equivalent damping for 2, 4 and 6° are given in figure 7 and as seen the difference in damping is large for these two cases.

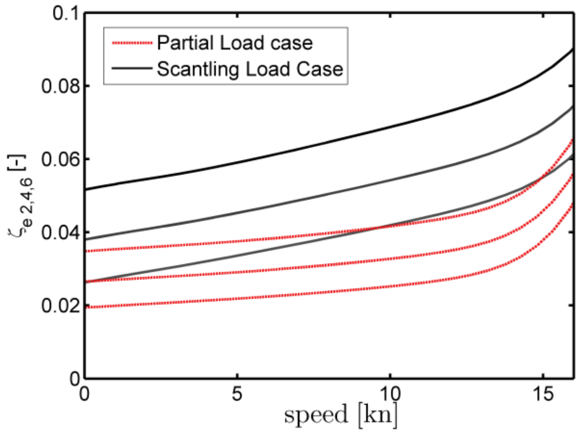

Figure 7: Damping as evaluated from the holistic model for a partial load case and a scantling case for a Pure Car and Truck Carrier. The linear equivalent damping at 2,4 and 6° are given, counted upwards.

4. DISCUSSION

Roll damping can be estimated using semi-empirical methods, computational fluid dynamics (CFD) calculations, model tests or full scale tests. None of these methods may alone be sufficient to capture the full roll damping behavior of a given ship in any given condition. However, they can all provide a valuable contribution in the different stages of the design process and in service.

The roll damping is rarely given sufficient attention (if any) in the design process when it comes to hydrodynamic optimization. Yet, with insufficient damping a new design may need to be operated with restrictions (more conservative routing) or loaded with restrictions (less cargo or more ballast water) to get an adequate dynamic behavior in certain conditions. Therefore, assessing the dynamic behavior of the vessel in different service conditions at an early stage is crucial when optimizing the design to identify if any operational conditions require particular attention.

For operational guidance systems providing in-situ ship-specific decision support, such as Ovegård et al (2012), a proper consideration of damping in the actual condition is crucial to provide relevant guidance and thus improve the safety level and avoid unnecessary deviations. With irrelevant information in onboard decision support systems guidance will be too rough which will lead to reduced safety level or unnecessarily conservative operation.

Scale effects related to roll damping requires more attention. The CFD calculations in this paper indicate that the scale effects, especially related to the bilge keels can be significant. Further work is required and the here presented holistic approach is a way forward for addressing the problem.

5. CONCLUSIONS

In this paper a framework for holistic multi-tier roll damping prediction has been presented. The approach provides a platform for best possible roll damping prediction given the different stages in the design process and for operation.

Starting from the earliest design stage Ikeda's semi-empirical model complemented with an aerodynamic component gives the foundation for a complete model that is applicable for all possible loading conditions and operational conditions. As the hull lines evolves the model can be updated with input from CFD calculations providing the hull specific lift coefficient and a more precise lift damping component. In the next stage of the design process updated input is provided from model tests. The bare hull damping and the bilge keel damping is threated separately and model test with and without bilge keels are required to establish these components.

To evaluate the bare hull damping semi-analytical expressions for the viscous components and lift components are deducted from the total damping and the remaining part is considered to be the *Froude number* dependent potential damping. A tuning function is used to match the evaluated potential damping for the tested case with the model for linear potential damping that typically is calculated using strip theory. The method incorporates a simplified scaling procedure for the bilge keel component reflecting the different viscous effects and in the model scale relative to full scale. The scaling procedure is based on the differences in dynamic pressure over the bilge keels due to the different boundary layer and results demonstrate that these effects can be considerable. Finally prior to delivery of the ship the model is assessed and tuned based on full scale trials. In this stage the final model that can be used as input for operational guidance is assessed and feedback to the design process can be given in a systematic manner.

Further work is needed on assessing tuning functions that are robust for different load cases for the potential damping. Model tests in different load cases and speeds are needed together with linear potential calculations for the corresponding conditions. Assessment of the full scale correlation factor also requires further attention. The accuracy of full scale trials need to be investigated and guidelines for successful tests established.

6. ACKNOWLEDGEMENTS

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