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Experimental determination of hydrodynamic loads on the Wärtsilä pre-swirl stator EnergoFlow and validation of a prediction methodology for design loads.

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ABSTRACT

The market request for energy saving paved way to the development and application of a pre-swirl stator (PSS) commercially known as the Wärtsilä EnergoFlow. Amongst others, reliable determination of the stator fin loads is essential for a robust PSS design. Since the device is relatively young, this subject is a new field of expertise. Sea keeping tests with a model of a container ship were conducted to measure the forces on one stator fin for normal and extreme weather conditions. Two types of critical loads can be distinguished. One component originates from the wave-induced ship motions, the other is related to the re-entry after emergence above the water surface. A lifting line theory for the stator fins is implemented to transfer the ship motion and sailing speed into forces and moments. These simulated forces show in general an overestimation with a factor of 1 to 2, compared to experimental model test data. When the fins emerge out of the water and re-enter, peak forces occur. Because of the non-linear character of slamming effects, a load factor of 2.25 is defined, which is derived from the conducted experiments. The applied methodology is integrated into the so-called "Wärtsilä Load Calculation" tool. It can be concluded that the validated methodology provides reliable and conservative predictions of the forces acting on a PSS.

Keywords

Pre-swirl stator, EnergoFlow, Energy Saving Device, hydrodynamic design load verification, stator force model test

1 INTRODUCTION

Pre-swirl stators have been on the market for many years now, but not always have they lived up to expectations, both from hydrodynamic and structural point of view. To make a robust design the hydrodynamic loads on the stator fins need to be known during the full operation profile of the vessel. Because not many researches are available on this subject, Wärtsilä has investigated this topic in the GRIP and LeanShip project. This paper will show the experimental determination of hydrodynamic loads on the Wartsila PSS, originating amongst others from ship motion in waves and slamming. The validation of the Wärtsilä Load Calculation tool will be highlighted based on the conducted measurements.

2 PREDICTION METHODLOGY OF STATOR LOADS

2.1 Load types during mission profile

During the life cycle of a ship, the PSS is exposed to forces of different nature, and range from relatively low to extreme loads in various occurring frequencies and statistical probabilities. The ship motion due to waves induces an additional lift and thus a load on the fins of the PSS. These loads are dominant for the fatigue lifetime of the stator fins as stated by S. Paboeuf (2014). Slamming occurs when one of the fins which has emerged above the water surface at high sea state makes a re-entry (see figure1. The loads during a crash stop and when accelerating into a turn might be of significance. Thus, the PSS structural assessment needs to be based on low cycle occurring peak loads and high cycle fatigue.

Figure 1: EnergoFlow just before re-entering the water

2.2 Prediction method of loads originating from ship motion

Each vessel has a different response to the incoming waves and therefore Response Amplitude Operators (RAO) are used to determine vessel specific responses. RAOs describe the response of the vessel for a chosen range of wave frequencies and headings. These RAOs can be used to setup a transfer function which relates the response of the vessel per amplitude of excitation to the incoming regular wave. It is assumed that the vessel response is linearly dependent on the wave height. Equation (1) gives the motion transfer function.

$$\eta_k = \eta_{ka}\cos(\omega t + \theta) = RAO cos(\omega t + \theta) \qquad (1)$$

Where η_k [m or rad] is the motion for the k^{th} degree of freedom, η_{ka} [m/m or rad/m] is the response amplitude per unit wave amplitude for the k^{th} degree of freedom, ω [rad/s] is the angular frequency, t [s] is the time and θ [rad] is the phase angle ,which gives the phase relationship between the motion and the wave.

RAO's are calculated with the MARINTEK software VERES (Fathi, 2004). This software uses a linear, potential, 2D strip theory and the main vessel parameters are used as input for the calculation. The RAOs are used to estimate the fluid angles and velocities at the position where the PSS is located.

The coordinate system which is used is the following and is shown in Figure 2 below.

- The x-axis is positive in the forward direction
- The y-axis is positive to port side
- The z-axis is positive upwards

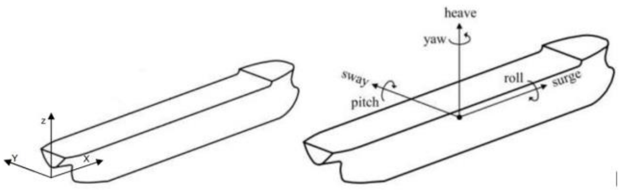

Figure 2: Coordinate system as used by BV (left) and the corresponding ship motions (right) (Paboeuf, Cassez, & Coache, 2014)

The fluid velocities at 6 degrees of freedom determine the incoming fluid angle at the PSS when the ship is horizontally aligned. The fluid angle is related to the main component to the lift and drag forces at the PSS. The fluid velocities are calculated using equation (2) and (3).

$$V_x = -V_0 + v_x\cos(\omega t + \varphi_x) \qquad (2)$$

$$V_z = v_z\cos(\omega t + \varphi_z) \qquad (3)$$

Where V_x [m/s] and V_z [m/s] are the relative fluid velocities at the PSS position, V_0 [m/s] is the ship speed, v_x [m/s] and v_z [m/s] are the relative fluid components due to waves and φ_x [rad] and φ_z [rad] are the phase angles.

The angle of attack for a PSS fin is calculated with equation (4) and (5). A representation of the angle of attack is shown in Fig 3 A symmetrical profile has zero lift at 0° angle of attack, but for a cambered profile, as shown in the figure, the zero-lift line shifts to a negative angle of attack. The benefit of this cambered profile is that it generates lift at 0° angle of attack.

$$\alpha = \alpha' - \varphi \qquad (4)$$

$$\alpha' = \operatorname{atan}\left(\frac{V_z}{V_0 - v_x}\right) \qquad (5)$$

Where α [rad] is the angle of attack, α' [rad] is the fluid angle and φ [rad] the pitch angle of the fin. The setting angle, the angular placement of the PSS and the trim angle of the vessel must be considered for a full description of the angle of attack.

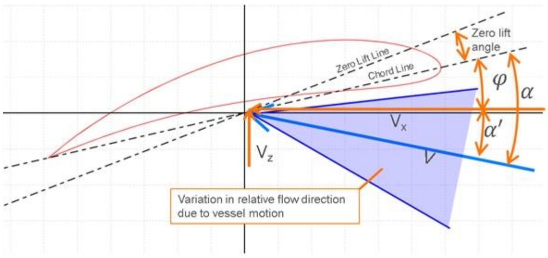

Figure 3: Angle of attack on a profile

The angle of attack is the target parameter because when it becomes larger, the force on the PSS increases. Equation (4) can be linearized resulting in equation [6] of which RAO can be made which is the target RAO. The combination of the angle of attack and the geometry of the PSS fins determine the lift and drag coefficients of the profile sections. These coefficients are used to calculate the forces at the fin using equation [7] and [8].

$$\alpha = \frac{V_z}{V_0} - \varphi \qquad (6)$$

$$L = \rho A V^2 C_L(\alpha) \qquad (7)$$

$$D = \rho A V^2 C_D(\alpha) \qquad (8)$$

Where L [N] and D [N] are the lift and drag forces respectively, ρ is the seawater density and is equal to 1025 kg/m^3, A [m^2] is the surface area of the fin, V [m/s] is the incoming velocity and C_L [-] and C_D [-] are the lift and drag coefficient respectively.

A spectral analysis is performed to determine the wave environment of the vessel and to create a design wave. The wave environment is described by sea states. A sea state is described by a wave spectrum which has a significant wave height and a peak period as can be found in a wave scatter diagram. A JONSWAP spectrum in combination with the North Atlantic's wave scatter diagram is used to describe the wave environment. The North Atlantic is considered as the most severe area. The spectral density, described by the JONSWAP spectrum, is a function of the wave frequency and the response spectrum for a certain RAO can be calculated by multiplying it with the JONSWAP spectrum as shown in equation (9).

$$S_R(\omega) = |RAO|^2(\omega) S_\omega(\omega) \qquad (9)$$

Where $S_R(\omega)$ the response spectrum of the ship is, $RAO(\omega)$ is the transfer function of any first order quantity and $S_\omega(\omega)$ is the JONSWAP spectral density. A directional spreading function is used to include the angular distribution of the waves.

The wave scatter diagram (see Fig 3) is used to determine the probability of occurrence of a sea state. When a wave scatter diagram is applied for a long-term prediction, the operational speed profile needs to be considered and should be adapted for certain sea states. BV divided the scatter diagram in several parts where the ship's speed is adapted as function of the significant wave height.

The PSS loads are induced by the ship motion and are estimated by applying a design wave. Using the linearized fluid angle RAO, a wave is selected which maximizes the angle of attack on the PSS fins. The frequency domain result is transformed to the time domain which results in the design wave. This design wave is based on the highest occurring load in 25 years due to vessel motion.

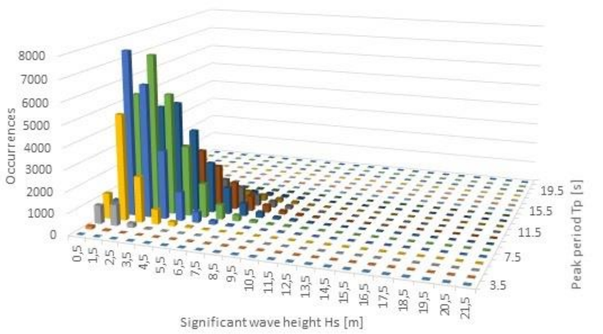

Figure 4: Wave scatter diagram, typical example per 100000 observations

A realistic operating profile must be defined to check what the maximum attainable vessel speed is in a specific sea state. If the operating profile is known, this profile is used in the calculation, but if the operating profile is not known, the operating profile as recommended by ABS is used, as shown in Table 1.

Table 1: The ship's operational speed profile (Shipping, 2016)

Significant Wave Height [m]	**Ship speed [knots] (V_d is design speed)**
> 0 – <= 6	100% V_d
> 6 – <= 9	75% V_d
> 9 – <= 12	50% V_d
> 12	25% V_d

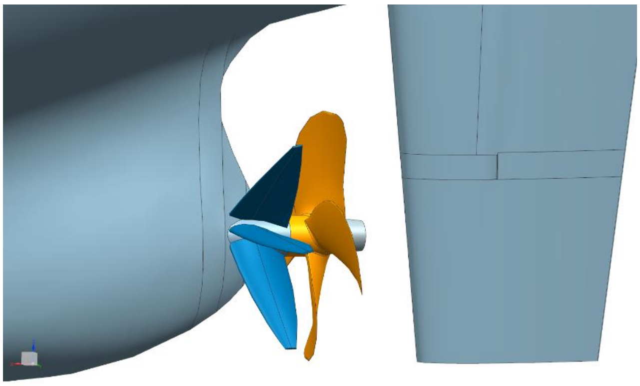

Figure 5: PSS shape used for the load calculation

Based on the parameters that are discussed in the previous paragraphs, a design wave can be defined based on the sea state which induces the largest fluid angle at the fins (Derbanne, Bigot, & Hauteclocque, 2012). The wave represents the most severe wave that the ship will experience in 25 years. The sea state spectrum with the largest fluid angle response is transformed to the time domain.

With the calculated wave and its corresponding maximum and minimum fluid angle in combination with the geometry of the stator fins the angle of attack can be determined. Subsequently, the hydrodynamic forces can be calculated by using, e.g, Prandtl's, lifting line theory (see Abbott & Doenhoff, 1959).

2.3 Prediction method of loads originating from slamming, turning circle and crash-stop

The so-called "off-design loads", for which the stator blades are exposed need to be considered. During events such as slamming, a turning circle and a crash stop, the hydrodynamic forces can be of significance for the structural design. A literature study was performed to investigate whether prediction methods for slamming loads on aft ships could be translated into slamming loads on stator fins. Amongst other publications; (Kapsenberg G. 2010) and (Shipping A.B. 2016) were used. Literature mainly deals with slamming loads on aft ships. It was concluded that slamming load prediction methods available for ship sterns cannot be used for stator fins, due to the different nature and strong interaction between the ship's stern and stator fins. Therefore, it was decided to define a slamming load factor. This factor is the ratio between the maximum load due to ship motion and the measured slamming loads from model tests. This factor will be used to multiply the forces as calculated by the WLC tool and thus estimate the slamming forces on the PSS.

No reliable engineering calculation methods could be found in literature for the crash stop. Therefore, the model test data will be used to explore the load levels of these events.

3 EXPERIMENTAL SETUP

Model tests were conducted at MARIN to investigate the hydrodynamic loads on the PSS by experiments and based (Dallinga R. 2017) on that validate and improve the Wärtsilä Load Calculation tool. A 1:33.7 model, designated no 8447 was used for the seakeeping tests. It was equipped with a raised forcastle, bilge keels and an active rudder. See fig 6

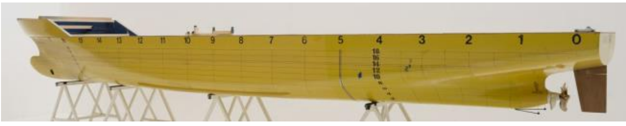

Figure 6: The 1:33.7 model

The model of the pre-swirl stator was printed in two parts in nylon material. The main part of the "hub" and the lower two blades were connected directly to the ship model. The upper blade was connected, through a ring-shaped connection to a 6-component force transducer which surrounded the conventional propeller shaft. Although the Wärtsilä EnergoFlow is normally designed with an outer ring connecting the stator blades, this ring was not applied to measure the forces on one stator only. Figure 7 illustrates the test devices.

Figure 7: Pre-swirl stator setup

The tests were performed in the Seakeeping and Maneuvering Basin at MARIN. The model was self-propelled during all tests.

The basic measurements consisted of the incident wave (at 3 positions) and the 6 motion components of the model. Additional measurements comprised the thrust and torque in the propeller shaft, just ahead of the hub, and the 6 components of the loads experienced by the isolated upper blade of the pre-swirl stator. The "stator loads" were limited to the forces acting on the highest blade element. The stator load sensor was developed in the EU funded project LEANSHIPS

3.1 Model Test results

A variety of tests was conducted from which the below listed are the most important ones.

- Measurements in waves; regular and irregular (BF7 - BF9 – extreme condition)
- "Manoeuvring"; acceleration, crash stop, turning circle
- (Transient) stator re-entry test

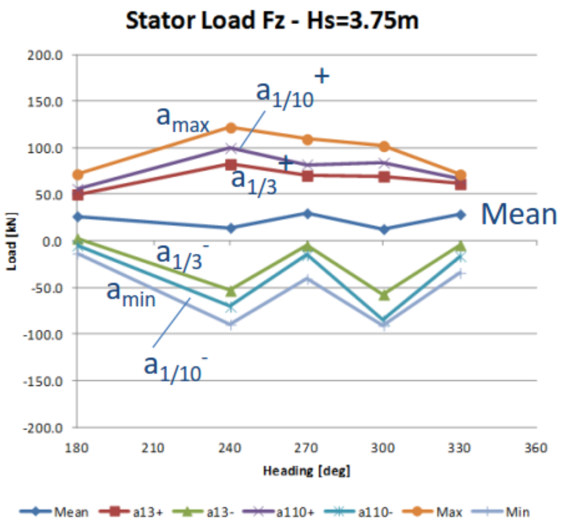

Figure 8: shows a summary of the measured stator loads in BF7 with a significant wave height of 3.75 m and a sailing speed of 20 knots. The stator load F_z is printed versus the heading. The highest load occurs in waves

from the bow and stern quarter. The maximum observed value is ~ 120 kN.

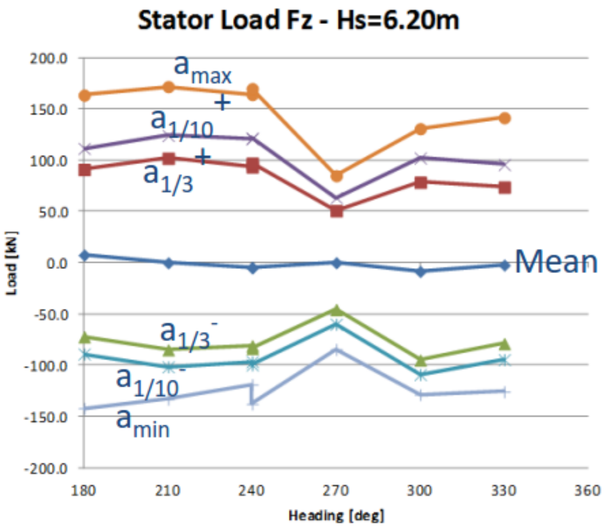

Figure 9: Measured stator loads versus heading at BF9.

A turning circle test was conducted with a freely accelerating model. Figure 10 shows the various data of the Port Side turn test. It was observed that the accelerating ship yields highest load far into the turn. The highest value in late stage in blade-freq. resonance is ~300 kN.

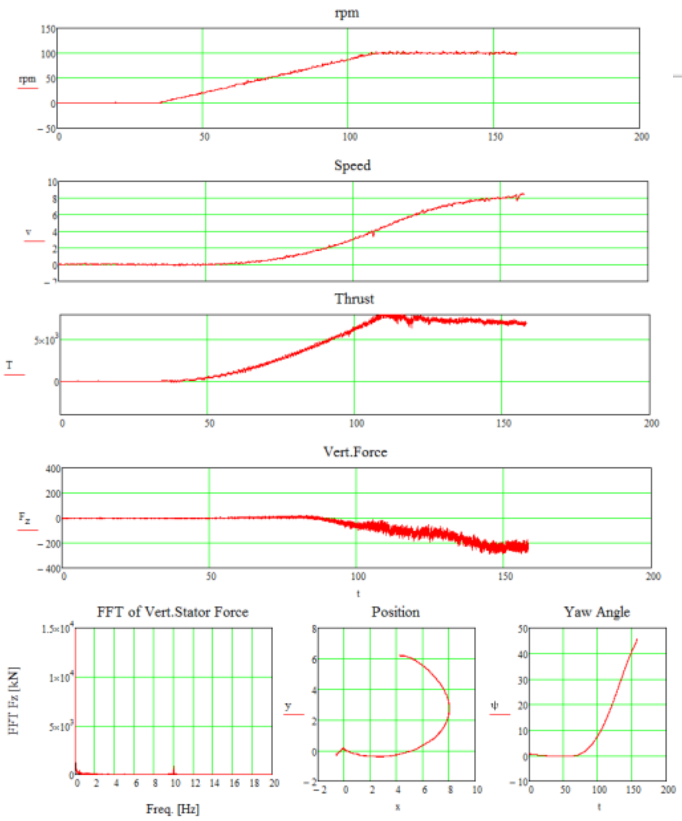

Figure 10: Time histories of turning circle test.

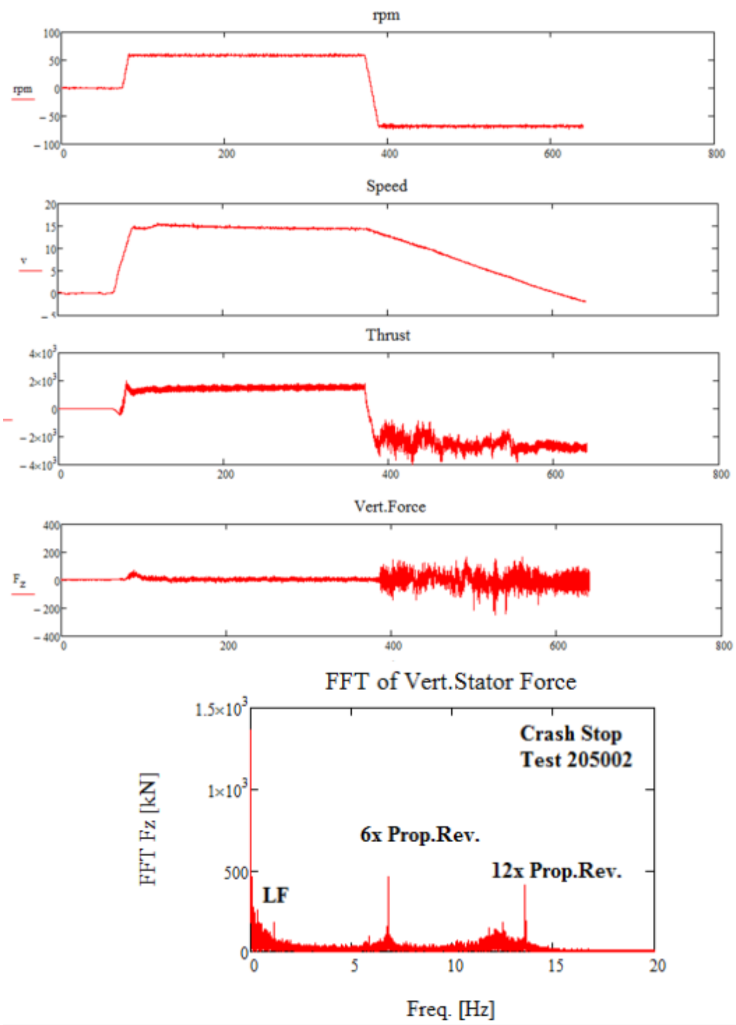

Figure 11: A crash stop was conducted starting at 15 knots. The time histories can be seen in Figure 11 below. The response is a combination of 1st and 2nd blade harmonics, and low frequency phenomena Highest measured loads are ~250 kN.

Re-entry tests were conducted both transient and with a drop test. The recordings of the transient test can be seen in Figure 12 below. The test shown holds for 10 m significant wave from 210 deg, at a low speed of 5 knots. The graph clearly shows the stator fin leaving the water surface (exit) with a load of 200 kN. After some time, the re-entry takes place with a max load of ~270 kN. Similar runs were conducted were max forces at re-entry were measured of ~370 kN (not shown in graph).

The drop tests showed similar phenomena with considerably lower max load values (approx. 130 kN). The ultimate loads observed are summarized in table 2

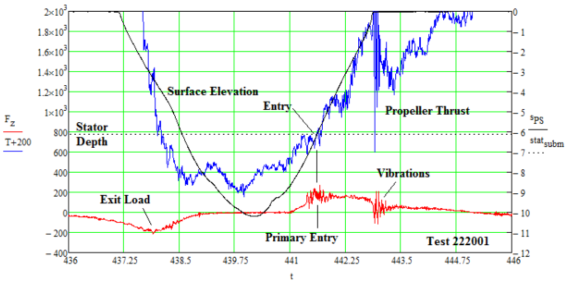

Figure 12: Time histories of transient re-entry loads in high waves

Table 2: Ultimate Loads

Condition	**Observed max load**	**Relevance for strength assessment**
Irregular waves, 20 knots at BF7, waves from 120 deg	120 kN	fatigue
Irregular waves, 15 knots at BF9, waves from forward direct.	170 kN	fatigue
Crash stop at 15 kn	250 kN	Off-design, peak load
Transient re-entry test 5 kn 10 m significant wave height	370 kN	Off-design, peak load
Turning circle during acceleration, yaw angle up to 40 deg.	300 kN	Off-design, peak load

The off-design loads (250-350 kN) are of significance, and exceed the observed forces during transient conditions, even compared in extreme sailing modes (120-170 kN).

4 VALIDATION WÄRTILÄ LOAD CALCULATION TOOL

For the validation of the Wärtsilä Load Calculation (WLC) tool a variety of data is used. First, the calculated RAO's of hull 8447 were compared with data from the model tests. The correlation is reasonably well [Mortel 2017]. Secondly, the lift and drag forces as calculated with the lifting line method from Prandtl was verified with validation data, as described in [Voermans, 2017]. The available stator load data generated by CFD from the studies performed in the GRIP project [Prins et al, 2016] were used as validation. A proper correlation was found given the applied methodology.

Finally, the above described model tests were used to validate the WLC. From the extended data set, two transient conditions will be shown in this section. Table 3 shows the measured versus the simulated load, for hull 8447.

Table 3: Measured load versus simulated stator load hull 8447, without slamming

Condition	**Model test max load**	**Calculated max load**	**correlation**
Irregular waves, 20 knots at BF7, significant wave 3.75 m	120 kN	241	2.0
Irregular waves, 15 knots at BF9, significant wave 6.2 m	170 kN	189	1.1

In the model tests a Beaufourt 9 sea state condition is tested with a significant wave height of 6.2 m and a ship speed of 15 knots. In this sea state, no slamming occurs, and the measured loads are approx. 170 kN. The same condition is simulated in the WLC tool which shows a maximum load of 189 kN. For the similar condition but with a Beaufourt 7 sea state, a load on the upper fin of 120 kN was measured while a load of 241 kN was predicted. It is a conservative calculation with a correlation factor of 2. Other conditions showed similar correlations in between 1 and 2, which is regarded as a good result when considering the applied simplifications for the complex hydrodynamic effects.

Table 4: Measured load versus simulated stator load hull 8447, with slamming

Condition	**Model test max load**	**Calculated max load**	**Slamming factor**
Transient re-entry test 5 kn, 10 m significant wave height	370 kN	166	2.25

Hydrodynamic loads for slamming were clearly recorded during the transient re-entry test at a sailing speed of 5 kn and 10 m significant wave height. The corresponding calculated load during ship motion is 166 kN where 370 kN was measured. With the determination of a slamming factor of 2.25 a reasonable and reliable estimation can be made of the hydrodynamic forces occurring during slamming events.

The slamming forces are the highest off-design loads from the ones observed.

It can be concluded that both the transient hydrodynamic stator loads assessment and off-design loads are calculated reasonably accurate with the WLC tool, and thus the tool is validated.

The predicted loads are input for fatigue and ultimate strength assessment where the necessary safety factors are applied to meet the specified requirements e.g. from Class Regulations [DNV-GL. (2015)].

It must be noted that fatigue loads are often determining for the strength assessment.

Further research will be done to improve the overall quality of the load predictions. This will be done by conducting more experiments with other ship types and refinement of the theoretical methods applied.

5 CONCLUSIONS

- Two types of loads can be distinguished. The first "linear" component is related to the wave induced ship motions, the second "non-linear" component is related to the re-entry after emergence above the water surface.
- The developed 6 component force transducer performed accurately, and stator loads could be recorded accurately.
- The predicted linear load component resulting from ship motion correlates reasonably well with the experimental data. Calculations are on the conservative side. The correlation factor lies in between 1 and 2, which is considered as a good result when taking he applied simplifications for the complex hydrodynamic effects into account
- Observed off-design loads during slamming events, turning circle and crash-stop are of significance. A slamming factor of 2.25 is derived from experiments. This enables an estimation for off-design loads
- The Wärtsilä Load Calculation tool is validated and can thus be used to generate load data for strength assessment of the Wärtsilä EnergoFlow.

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