COMPARISON OF DESIGN WAVE APPROACH AND SHORT TERM APPROACH WITH INCREASED WAVE HEIGHT IN THE EVALUATION OF WHIPPING INDUCED BENDING MOMENT

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ABSTRACT
The evaluation of extreme bending moment corresponding to a 25 years return period requires very long simulations on a large number of sea states. This long term analysis is easy to do with a linear model of the ship response, but is impractical when using a time consuming model including non linear and slamming loads. In that case some simplified methods need to be applied. These methods are often based on Equivalent Design Waves (EDW) which are calibrated on the extreme linear value.

The general practice is to define the EDW as a regular wave. A very simple method is to compute the non linear bending moment applying the pressure correction on the hull without recomputing the ship motions. A better method is to recompute in time domain the non linear ship response on this Design Wave. It is even possible to define a more realistic Design Wave, taking into account the frequency and directional content of the sea states used in the long term analysis: those waves are called Response Conditioned Wave and Directional Response Conditioned Waves.

The different methods are applied to an Ultra Large Container Ship (ULCS). Hydro-structure calculations are carried out on a severe design sea state, taking into account Froude-Krylov pressure correction, slamming forces and whipping response. Results of a very long computation are compared to the results of the Design Wave approaches.

Another method is proposed to compute very rare events. It is based on an artificial increase of the significant wave height of the sea state, and the assumption of the independence of the non linear effects to the significant wave height. Using this method it is possible, with a simulation of only a few hours, to predict a very rare short term event, corresponding to a very long return period. The results are compared to the Design Wave results and appear to be much more precise.

INTRODUCTION
The Equivalent Design Wave approach is widely used in the industry as an efficient engineering procedure. It is the basis of the hydrodynamic loads formulas in the different classification rules, such as the IACS Common Structural Rules.

The major benefit of the EDW approach is that it significantly reduces the computational cost. There are two major applications of design waves. The first one is the evaluation of the structural response, based on a few waves that maximize a given set of governing loads. The second one is the evaluation of non linear hydrodynamic loads. This paper focuses on the latter.

NOMENCLATURE
EDW Equivalent Design Wave
VBM Vertical Bending Moment
RCW Response Conditioned Wave
RAO Response Amplitude Operator
DSS Design Sea State

HYDROELASTIC MODEL
During last several years an efficient hydroelastic model based on state of the art numerical techniques has been developed in cooperation between TNO and Bureau Veritas. The model couples 3D hydrodynamics with 3DFEM structural dynamics. This model is able to compute the quasi static response of the ship as well as the dynamic response including springing and whipping effects.

The general methodology for hydroelastic seakeeping model is rather well known and the first developments can be attributed to Bishop & Price (1979). In their work they used the Timoshenko beam model as a simplified model of the structure, and the strip theory for the seakeeping part. Since
then several more or less sophisticated models were proposed: Wu & Price (1986), Wu & Moan (1996), Xia & Wang (1997), Korobkin (2005). Below we give a brief introduction to the basic principles of the model used in this study. The 3D BEM (Boundary element Method) model for the seakeeping is coupled to a 3D FEM model of the ship structure. A more detailed description of the applied 3D BEM model can be found in Newman (1994) and Malenica & al (2003).

**Linear hydroelastic Model**

In contrast to the well known rigid body seakeeping model, the hydroelastic model basically extends the motion representation with additional modes of motion/deformation chosen as a series of the dry structural natural modes. We write:

\[
\{H\}(x, y, z, t) = \sum_{i=1}^{N} \{\xi_i(t)\} \{h_i\}(w, y, z)
\]

where \( \{h_i\}(x, y, z) \) denotes the general motion/deformation mode which can be either rigid or elastic, and \( \{H\}(x, y, z, t) \) is the total motion at a point \((x, y, z)\). The above decomposition leads to additional radiation boundary value problems (BVP) for elastic modes, with the following change in the body boundary condition:

\[
\frac{\partial \varphi_{Rj}}{\partial n} = \{h_j\} \cdot \{n\}
\]

where \( \{n\} \) is the normal to the hull, and \( \varphi_{Rj} \) is the radiation potential corresponding to the mode \( j \).

After solving the different BVP-s the resulting pressure is calculated using Bernoulli’s equation and integrated over the wet surface in order to obtain the corresponding forces, so that the following coupled dynamic equation can be written:

\[
-\omega_j^2 (\{m\} + \{A\}) - i\omega_j \{b\} + \{B\} + \{c\} + \{C\}\{\xi\} = \{F^{\infty}\}
\]

where:

- \( \{m\} \): modal genuine mass matrix
- \( \{A\} \): hydrodynamic added mass matrix
- \( \{b\} \): structural damping matrix
- \( \{B\} \): hydrodynamic damping matrix
- \( \{c\} \): modal structural stiffness matrix
- \( \{C\} \): hydrostatic stiffness matrix
- \( \{\xi\} \): modal amplitudes
- \( \{F^{\infty}\} \): modal excitation

The solution of the above equation gives the motion amplitudes and phase angles by which the problem is formally solved. Note that the motion equation includes both 6 rigid body modes and a certain number of elastic modes.

Several technical difficulties need to be solved before arriving to the above motion (Eq. 3). Certainly the most difficult one is the solution of the corresponding hydrodynamic BVP (Boundary value Problem). In this paper we do not enter into the detailed description of the methods used to solve the seakeeping problem at forward speed and we just mention that these difficulties remain the same, both for the rigid and elastic body. It is fair to say that the numerical methods which are used to solve the seakeeping problem nowadays are not fully ready yet for a general combination of speed, heading and frequency. However, most of the methods have approximate solutions to account for the forward velocity. The method used in this paper is the so called encounter frequency approximation which was reasonably well validated for the rigid body case.

The second technical difficulty concerns the application of the body boundary condition (Eq. 2) for the general mode of motion. The goal is to apply the modal deformation, computed on the structural mesh of the ship, to the hydrodynamic mesh used to solve the hydrodynamic problem. This interpolation problem can be solved through many ways. The procedure used here is explained in Tuiman & Malenica (2009).

The solution of the hydroelastic motion equation (3) automatically includes the linear springing response. Due to their huge dimensions and particular structural properties, the case of ULCS is quite particular because their first structural natural frequencies become quite low so that they can be excited in a linear sense. This means that the linearly excited springing might become dominant, which justifies the linear model for its assessment.

**Non Linear Time Domain Simulations**

In contrast to the linear springing calculations which can be performed in frequency domain, the non-linear springing and whipping simulations require time domain simulations. The procedure used in this paper is based on the method proposed in Cummins (1962) and elaborated in Ogilvie (1964). This method uses the frequency domain hydrodynamic solution and transfers it to the time domain using the inverse Fourier transform. In this way the following time domain motion equation is obtained:

\[
\left(\{m\} + \{A^-\}\right)\{\ddot{\xi}(t)\} + \{b\} \{\ddot{\xi}(t)\} + \int_0^t [\{K(t-\tau)\}]\{\dot{\xi}(\tau)\}d\tau + \{c\} + \{C\}\{\dot{\xi}(t)\} = \{F^{\infty}(t)\} + \{Q(t)\}
\]

where overdots denote the time derivatives and:

- \( \{A^-\} \): infinite frequency added mass matrix
- \( [K(t)] \): matrix of impulse response functions
It was shown by Ogilvie that the impulse response functions can be calculated from the frequency dependent damping coefficients:

\[ K_i(t) = \frac{2}{\pi} \int_{0}^{\infty} B_i(\omega) \cos(\omega t) \cdot d\omega \]  

(5)

After the impulse response functions \( K_i \) have been calculated, the motion equation (4) is integrated in time using the Runge Kutta 4th order scheme.

The main advantage of the time domain method lies in the fact that we can introduce non-linear components in the excitation forces \( Q(t) \). There are many non-linear effects which are missing in the linear model and the state of the art in numerical seakeeping, does not allow for the consistent inclusion of all of them. In this paper we include the so-called Froude-Krylov correction for the wave excitation and the non-linear slamming forces.

**Froude-Krylov Approximation**

It is well known that the linear hydrodynamic model evaluates the pressure up to \( z = 0 \) only so that we can end up with negative pressure close to the waterline, as well as with zero pressure above the waterline and below the wave crest, which is unphysical. The so-called Froude-Krylov approximation is usually employed to correct this pressure distribution close to the waterline. The procedure used here is described in detail in Tuitman & Malenica (2009). We should keep in mind that several approximations are possible and, in spite of the quite complicated considerations, it is impossible to remain fully consistent. It is also important to note that several variants of the Froude-Krylov method are in use. The simplest one consists in taking only the incident wave into account in the definition of the relative wave elevation \( \eta_R \) (and thus neglecting the diffracted and radiated part). This choice highly simplifies the numerical implementation and this is the reason why it is most often employed. In that case we need to be extremely careful when interpreting the results. This Froude-Krylov approximation is likely to be valid only in the case of rather long waves where the diffraction and radiation parts play less important role in the overall contribution to relative wave elevation \( \eta_R \). However, in the case of ULCS which have quite a low block coefficient, i.e. rather slender form, the Froude-Krylov approximation can be reasonably justified for head or almost head wave conditions for which the springing and especially whipping are most likely to occur.

**Whipping**

The time domain calculation will automatically include the linear and non-linear springing. In order to calculate the slamming induced whipping response, the slamming forces should be calculated during the seakeeping calculation. This forces are added to the force vector \( \{Q(t)\} \) of Eq. (5).

The hydrodynamic modeling of slamming is extremely complex, and still no fully satisfactory slamming model exists. However, the 2D modeling of slamming is well mastered today and 2D models are usually employed to assess the slamming loads on ships. Within the potential flow approach, which is of concern here, several more or less complicated 2D slamming models exist, starting from simple von-Karman model and ending by the fully non-linear model. In between these two models there are several intermediate ones such as Generalized Wagner Model (GWM) and Modified Logvinovich Model (MLM) which are implemented in the presented method. We will not go into too much details of these models and we refer to Malenica & Korobkin (2007). The advantage of the MLM method when compared to GWM method lies in the lower requirements of CPU time. Most often, at least when rigid body impact is concerned, both methods give comparable results as shown in Fig. 1. The GWM method is used for the presented examples because this method has a wider range of validity and the number of slamming calculations remains limited. If many calculations, for example a complete scatter diagram, have to be performed the MLM method might become more attractive. The validity of the MLM calculation can be checked using the GWM results for the some cases.

![Fig. 1: Slamming forces on typical 2D section obtained by GWM and MLM method](image-url)

The use of 2D methods implicitly requires the employment of the so-called strip approach for 3D simulations. The usual procedure consists in cutting the 3D mesh into a certain number of 2D sections. For each particular section, independent calculations of slamming loading are performed and the overall slamming is obtained by summing up different contributions.

The implementation of this procedure is different for the most often used beam structural model and for 3DFEM model. More details can be found in Tuitman & Malenica (2009). Pressures obtained on the slamming sections are integrated on the modal deformations (including rigid and elastic modes). For each mode this gives the slamming force \( \{Q(t)\} \) to be added to Eq. (4).

The whipping response is automatically computed when solving Eq. (4) in time domain with the slamming forces taken into account in the right hand side.
EQUIVALENT DESIGN WAVES

An Equivalent Design Wave is a wave, or a group of waves, which is defined in order to maximize a given target parameter. This target parameter is often a load parameter (such as vertical bending moment, acceleration…), but it can be any type of ship response (local pressure, stress…). The linear ship response on the EDW is known, as well as its probability of occurrence, which is governed by a Rayleigh law. The non linear response is then computed, and associated to the same probability level.

Static Regular Design Wave

General practice is to choose a regular wave for the EDW, the main advantage being its simplicity. The EDW is defined by the following parameters: frequency $\omega$, heading $\beta$, amplitude $A$ and phase $\Phi$.

The choice of the heading is not always straightforward. If the design wave is used to compute the non linearity of the response on a given sea state, then the heading should be the heading of the sea state. If the design wave is used to compute the non linearity of the long term response, then the heading should be the heading of the most contributive sea state to the long term response. Special care should be given when the sea states are short crested, as explained by de Hauteclocque et al. (2012). In that case the main heading of the sea state may not correspond to an important ship response, and we may have to choose the heading as the one giving the maximum RAO response.

The frequency is the one maximizing the RAO of the linear ship response at the chosen heading.

The amplitude and phase are defined such as the linear ship response reaches the targeted response $X$ at $t=0$.

$$A = \frac{X}{\text{RAO}(\omega, \beta)}$$

$$\phi = -\text{phase}(\text{RAO}(\omega, \beta))$$

The acceleration is then modified such as the difference is balanced by inertial forces. It has to be underlined here that the resulting accelerations are no more in accordance with the ship motions! The non linear ship response, such as hull girder forces or local stress can then be computed. This non linear response is supposed to have the same probability as the targeted linear response. It has to be noted that this method is unable to take into account the whipping response. Hence it is only used to take into account Froude-Krylov non linearity.

Regular Design Wave

This method is using the same regular design wave as the one defined above. The difference is that the ship response is computed using a time-domain non linear seakeeping code solving Eq. (4), which may include slamming forces and whipping response. The simulation has to be done on at least five wave periods, so that the periodic ship response is reached. The maximum (minimum) non linear response on the wave period is supposed to have the same probability as the targeted linear response. This maximum (minimum) may occur at a different time step than the linear maximum (minimum).

Fig. 3 shows the ship response on this regular design wave. We can observe that the non linear ship motion, resulting from the non linear time domain simulation (Froude-Krylov forces only or Froude-Krylov and slamming forces), may be significantly different from the linear ones. Thus the hull girder loads and local stress response may be very different from those computed using the Static Regular Design Wave.
Response Conditioned Wave

The idea behind the Response Conditioned Wave is to include the wave spectrum and the ship response in the definition of the design wave. The first application was used by Tromans et al (1991) to construct the most likely extreme storm wave of a given wave spectrum. Using the ideas from Friis-Hansen and Nielsen (1995), Taylor et al. (1995) and Adegeest et al. (1998), Dietz (2004) developed the most likely response wave (MLRW). Using the amplitude and phase angle information from the transfer function, this method uses linear theory to condition the incident wave profile to cause a predefined specific response at a certain time instant. The linear ship response on this design wave is the Most Likely Extreme Response, corresponding to the average of all possible responses reaching the target value at \( t=0 \).

The RCW is thus an irregular wave train, containing several components (amplitude-frequency-phase). The RCW is defined by the following parameters: spectral characteristics \((T_p, \gamma)\), heading \(\beta\), amplitude \(A_i\) and phase \(\Phi_i\).

The choice of the heading is the same as for the regular design wave: either the heading of the short term sea state, either the heading of the most contributive sea state to the long term extreme, with special care to be given to short crested sea states.

The spectral characteristics \((T_p, \gamma)\) are those of the short term sea state, or those of the most contributive sea state to the long term extreme.

The amplitude and phase of all the components are computed from the linear response RAO and from the sea state spectrum \(S(\omega)\) having the spectral characteristics \(T_p, \gamma\) (the significant wave height can be set to 1m as the amplitude of the RCW is independent from the chosen wave height).

\[
A_i = S(\omega) \| RAO(\omega, \beta) \| \frac{X}{m_0} \Delta \omega \\
\phi_i = -\text{phase}(RAO(\omega, \beta)) \\
m_0 = \int S(\omega) \| RAO(\omega, \beta) \|^2 d\omega
\]  

(7)

The advantage of such an EDW is that it contains much more information than a regular EDW. It contains both RAOs and wave data, which should lead to a more realistic wave. An example of RCW is shown in Fig. 5.

The RCW produces an asymmetric response. Thus two different waves are needed to maximize respectively hogging and sagging bending moment, as shown in Fig. 6 and Fig. 7.
APPLICATION TO EXTREME WHIPPING INDUCED BENDING MOMENT

Non linear whipping simulations have been done on a 380m long Ultra Large Container Ship. The natural wet frequency of the first vertical bending mode is only 2.3 rad/s. And because of the large bow flare, the ship is very sensitive to slamming.

Linear computations

The vertical bending moment amidship has been computed with a linear seakeeping software, for a speed of 5 knots and for every headings. The extreme 25 years value has been derived with a long-term analysis based on the IACS scatter diagram. The contributions from all the sea states (Fig. 8) shows that the sea state having the highest contribution is: $H_s = 15.5\text{m} – T_p = 17.6\text{s}$.

This sea state is referred as the Design Sea State, because it is the most probable sea state which will give the 25 years vertical bending moment response. However the return period of the extreme 25 years bending moment on this sea state is 2h20min, which means that even on this high sea state the extreme 25 years bending moment is a rare event.

Non linear computations on the Design Sea State

Non linear seakeeping simulations are done in time domain on the Design Sea State. The sea state is defined by a list of 200 components with a random phase and an amplitude defined according to the sea spectrum. 177 sequences of 18 minutes are done in order to have a 53 hours long simulation. Three type of time domain simulations are done on all the 177 sequences. A first linear simulation is done for comparison purpose. A second simulation is done including the Froude-Krylov non linear forces but excluding the slamming forces. A third simulation is done with the slamming forces and hence the whipping response. Those irregular responses are then processed with an up-crossing analysis in order to derive the hogging and sagging bending moment extremes distributions. The up-crossing analysis of the non linear ship response is based on the up-crossing cycles of the linear response as shown in Fig. 9, which means that only the highest maxima and the lowest minima in the linear low frequency cycle are taken into account.

Hence we have the same number of extremes in the linear and non linear distributions. The linear extremes are following...
a Rayleigh distribution, while the non linear extremes may follow a very different and unknown distribution, as shown in Fig. 10. VBM values shown in Fig. 10 and others have been divided by the linear long term value: a value of 1.0 means the linear long term value.

Fig. 10: VBM extreme distribution on the Design Sea State

In order to have extreme values on this sea state (corresponding to a return period greater than 1h), a very long simulation is needed to get a good convergence on the results (about 20 times the return period). Here a 53h simulation has been done. The extremes corresponding to a return period lower than 2h40min (exceedence rate greater than 0.38/h) may be considered as converged. This is just enough to get results for the extreme 25 years bending moment, which correspond on this sea state to a 2h20min return period. The highest 20 extremes (tail of the distributions in Fig. 10) are not converged and should not be considered in the further analysis.

For each return period, or each linear extreme value (related to the return period by the Rayleigh law), we can define the following correction coefficients by comparing linear and non linear extremes.

- The Froude-Krylov correction is defined as the increase of VBM due to the non linear Froude-Krylov forces:
  \[ \text{Froude-Krylov correction} = \frac{VBM_{FK}}{VBM_{Lin}} - 1 \]  
  (8)
- The whipping correction is defined as the increase of VBM due to whipping, as compared to the Froude-Krylov non linear VBM:
  \[ \text{Whipping correction} = \frac{VBM_{whip}}{VBM_{FK}} - 1 \]  
  (9)
- The total correction is defined as the total increase of VBM.
  \[ \text{Total correction} = \frac{VBM_{whip}}{VBM_{Lin}} - 1 \]  
  (10)

These correction factors have been computed for all the linear extremes and shown in Fig. 11. We will only consider the part of the curve in bold, which corresponds to the 20% highest extremes, without the highest 20 extremes for which the correction factor may not be converged. The reason to remove the 80% lowest extremes is that we are focusing only on the highest and rare events.

Looking at the correction corresponding to the linear long term extreme value (VBM=1 in Fig. 11), the Froude-Krylov correction is -13% in hogging and +32% in sagging, and the total correction is +18% in hogging and +75% in sagging.

Fig. 11: Froude-Krylov and total correction on the Design Sea State

Non linear computations on EDWs

To avoid such a long simulation, methods based on EDW are usually used. The Design Waves are calibrated to produce the linear response corresponding to a given return period. The non linear response computed on this Design Wave is supposed to correspond to the same return period. The three types of EDW defined above are used to compute the non linear VBM response due to Froude-Krylov loads. The Regular Design Wave and the RCW have been used to compute the non linear VBM including Froude-Krylov forces, slamming forces and whipping response.

Several EDW of different amplitudes have been used to compute the non linear response corresponding to several values of linear VBM. For each of them, the Froude-Krylov and total correction factors are computed and compared to the ones given by the 53h simulation (Fig. 12 and Fig. 13).

The static regular design wave is giving very poor results. For the extreme long term value it predicts a -1.5% and a +51% correction in hogging and sagging respectively. This is mainly explained by the fact that the position of the ship on the
design wave is very different from the real position, and that the load case is not correctly balanced (see Fig. 3).

However the regular wave prediction in hogging is -22% instead of -13%! The total correction is better predicted by the RCW. Surprisingly, results from the regular wave are however still quite good but a bit over predicted.

**Non linear computations on Design Sea State with increased wave height**

The drawback of the design wave approach is that these waves are an approximation of the real waves the ship will encounter in an irregular sea states. The drawback of doing a time domain simulation on the Design Sea State is that a very long simulation is needed in order to exceed the linear long term value at least 20 times, in order to get a converged non linear extreme value. A simple idea to shorten the duration of the simulation is to increase the wave height of the sea state, thus exceeding the target linear extreme more often.

If $H_s$ and $H_{s_2}$ are the significant wave height of the Design Sea State and the increased sea state respectively, the exceedence rates $n_1$ and $n_2$ of a given VBM linear response is computed according to the Rayleigh law:

$$n_1 = N \exp \left( \frac{X^2}{2m_{0,1}} \right) \quad \text{with} \quad m_{0,1} = m_{0,1} \left( \frac{H_{s_2}}{H_s} \right)^2$$  

$$n_2 = N \exp \left( \frac{X^2}{2m_{0,2}} \right)$$  

(11)

Where $N$ is the total number of VBM cycles per hour in the two sea states, $m_{0,1}$ and $m_{0,2}$ are the spectral moments of order 0 of the response spectrum of sea states 1 and 2 respectively. Hence the exceedence rate in the increased sea state can be computed directly from the $H_s$ ratio:

$$n_2 = N \left( \frac{m_{0,2}}{m_{0,1}} \right)$$  

(12)

For instance, the extreme long term linear VBM which return period is 2h20min in the Design Sea State ($H_s=15.5m$), has a return period of only 18min on a 18.8m $H_s$ (21% increase). Hence a 6h simulation is required to get a converged value for the long term response. On a 22.5m $H_s$ (45% increase), the return period is only 5min, and 1h40min simulation is enough to get a converged result.

We now make the assumption that the correction factors are independent from the wave height of the sea state. That is to say that the correction factor computed on higher sea state for a given linear extreme value is still valid on the Design Sea State for the same linear extreme value. In other words, a very rare event on the DSS and a rare event on the higher sea state, corresponding to the same linear value, have exactly the same non linear corrections. There is no theoretical reason to do this assumption. However it should be noted that this assumption is already implicitly done when using the EDW method. With a Regular EDW, the wave frequency and heading are independent from the DSS. The wave amplitude is adjusted to...
reach the given linear response. Hence the same EDW will be used to get the non linear response corresponding to the same linear response whatever the DSS is. When a RCW is used, its characteristics depend on the DSS heading and period, but are independent from the significant wave height.

In order to verify this assumption, 5h time domain simulations have been done on 13 sea states having the same peak period and increasing wave height, given in Table 2. The DSS used above is sea state 10.

Table 2 : Sea state characteristics

<table>
<thead>
<tr>
<th>N°</th>
<th>Hs (m)</th>
<th>N°</th>
<th>Hs (m)</th>
<th>N°</th>
<th>Hs (m)</th>
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<tr>
<td>1</td>
<td>3.00</td>
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<td>6.25</td>
<td>10</td>
<td>15.50</td>
<td></td>
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</tbody>
</table>

The correction factors are then computed for each sea state, and plotted together in Fig. 14 (as before, the corrections factors are only plotted for the highest 20% extremes, excluding the 20 highest extremes, which means that about 240 VBM extremes are used for each sea state). Each line is representing the correction factors computed for one sea state.

Fig. 14: Froude-Krylov and Total correction using sea states of different wave height

It can be seen that the correction factors computed from the different sea states are all in line. It means that for a given linear VBM response, the correction factor is independent from the sea state. Despite the 5h duration of the simulations, the correction factors are not completely converged, which explains the little scatter of the results. In order to improve the robustness of the method, it is proposed to fit a third order polynomial approximation on the results.

\[ Froude-Krylov_{\text{correction}} = P_{FK}(X_{Lin}) \]
\[ Total_{\text{correction}} = P_{Tot}(X_{Lin}) \]

This polynomial correction can then be used to compute the non linear extreme corresponding to any return period on any sea state (with the same heading and period), by applying the correction to the linear extreme having the same return period T.

\[ X_{FK} = X_{Lin}(1 + P_{FK}(X_{Lin})) \]
\[ X_{Tot} = X_{Lin}(1 + P_{Tot}(X_{Lin})) \]

\[ X_{Lin} = \sqrt{2m_o \ln(N.T)} \]

Table 3 gives the correction factors computed with the highest sea states, corresponding to the linear long term extreme bending moment, and the correction factors computed with the polynomial approximation. The results are very close to those of the DSS (Hs = 15.5m), and the accuracy is considerably enhanced compared to the EDW method (Table 1). Fig. 15 shows all the results on the same plot.

Table 3 : Froude-Krylov and total correction corresponding to the linear extreme VBM, using increased wave height

<table>
<thead>
<tr>
<th></th>
<th>Hog</th>
<th>Sag</th>
<th>Hog</th>
<th>Sag</th>
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</thead>
<tbody>
<tr>
<td>Hs = 15.5</td>
<td>-13%</td>
<td>+32%</td>
<td>+18%</td>
<td>+75%</td>
</tr>
<tr>
<td>Hs = 18.8</td>
<td>-16%</td>
<td>+33%</td>
<td>+13%</td>
<td>+75%</td>
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<td>+33%</td>
<td>+12%</td>
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<td>+31%</td>
<td>+12%</td>
<td>+74%</td>
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<tr>
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<td>-17%</td>
<td>+33%</td>
<td>+13%</td>
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</table>

Fig. 15: Froude-Krylov and Total correction using EDWs, sea states of different wave height and polynomial approximation
CONCLUSIONS

A 53h long simulation hydro elastic simulation on a high Design Sea State has been done, including non linear Froude-Krylov and slamming loads. Results in Vertical Bending Moment have been compared with faster methods using Equivalent Design Waves.

The Static Regular Design Wave method gives very poor results for the effect of non linear Froude-Krylov loads, and is unable to compute whipping response. Despite its simplicity of use, this method should be avoided.

Methods based on a dynamic response on a Regular Design Wave or a Response Conditioned Wave give better results, even for the extreme whipping response. However errors in the order of 10% can still occur. One important observation from this study is that RCW method is not giving significantly better results than the Regular Wave method. This method should however be preferred, because the definition of the Design Wave has more physical sense, and the computational time is nearly the same as the Regular Wave.

Much more accurate results are obtained when the simulations are done on irregular sea state with an increased wave height. Increasing the significant wave height decreases the return period of a given linear extreme, and hence decreases the needed duration of the simulation. Long duration simulations on many sea states of different wave height showed that the difference between the non linear and the linear response doesn’t depend on the sea state significant wave height, but just on the value of the linear response. Hence it is possible to characterize this non linear correction by a polynomial, and use this polynomial to compute the non linear ship response corresponding to any return period on any sea state with the same period and heading.

REFERENCES


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