DISCUSSION ON HYDROELASTIC CONTRIBUTION TO FATIGUE DAMAGE OF CONTAINERSHIPS

Quentin Derbanne\textsuperscript{a}, François-Xavier Sireta\textsuperscript{a}, Fabien Bigot\textsuperscript{a} and Guillaume de Hauteclouque\textsuperscript{a}

\textsuperscript{a} Bureau Veritas

ABSTRACT
Full scale monitoring campaigns and model tests have shown an important high frequency contribution to fatigue damage, coming from the global wave induced ship vibrations. This high frequency contribution is often assumed to be the hydroelastic contribution.

Numerical models are able to separate the quasi-static response from the total hydroelastic response. It is shown that the total linear hydroelastic contribution is the combination of three different hydroelastic effects: hydrostatic relaxation, low frequency dynamic amplification, and high frequency resonance.

Computations have been done for 6 container ships and linear long term fatigue damage is computed based on bending moment and torsion RAOs. The conclusion is that for vertical bending damage the hydrostatic restoring effect and the low frequency dynamic amplification effect are cancelling each other: the total hydroelastic contribution is equal to the high frequency contribution. However for torsion damage there is no hydrostatic restoring effect and an important part of the hydroelastic effect is located at low frequency: the total hydroelastic contribution to fatigue damage is larger than the pure high frequency contribution.

KEYWORDS
Fatigue damage; Springing; Containerships; High frequency; Quasi-static; Hydroelasticity; Bending moment; Torsion

1 INTRODUCTION
The increased size of the recent Ultra Large Container Ships and LNG ships reactualizes the hydroelastic wave induced type of ship structural responses. Usually the hydroelastic ship response is divided into springing and whipping, springing being defined as the steady ship vibrations induced by non-impulsive wave loading and whipping as a transient vibratory ship response induced by impulsive loading such as slamming.

In the past few years many measurement campaigns have been done on several containerships sailing all around the world. Ito & al.\textsuperscript{1} analyzed some measurement on a 278m containership sailing in 5m significant wave height. They showed that the contribution of high frequencies (HF) to the total fatigue damage is between 30% to 40%. Oka & al.\textsuperscript{2} showed that for a post panamax containership, after two and a half year of navigation between Japan and Europe, 46% of the fatigue damage is due to the HF content. Koo and al.\textsuperscript{3} have the same result of 28% HF contribution after 28 months of navigation of a 8000 TEU container vessel. Storhaug & al.\textsuperscript{4,5}
showed that according to model test results this HF contribution to fatigue damage can be higher: between 65% and 85%.

Using full scale measurement or model test data it is not possible to separate the quasi-static response of the ship from the hydroelastic response. The only option is to separate the HF response from the wave frequency response. But is the hydroelastic response only a HF response? The numerical simulations, however, are able to separate those two components, by taking into account or not the hydroelastic response of the hull. It is then possible to compute exactly the participation of hydroelasticity to fatigue damage and to compare with the HF participation.

2 HYDROELASTIC MODEL

The general methodology for hydroelastic seakeeping model is rather well known and the first developments can be attributed to Bishop & Price. In their work they used a Timoshenko beam model as a simplified model of the structure and the strip theory for seakeeping part. Since then several more or less sophisticated models were proposed: Wu & Price, Wu & Moan, Xia & Wang, Korobkin. Below we give a brief introduction in 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 and Malenica & al.

2.1 Linear rigid body quasi-static model

In this model, the rigid body motions are solved without considering any interaction with the elastic distortion. In a second step, the elastic response of the structure is considered in a quasi-static way (distortion is purely proportional to the applied loads, without any dynamic effect).

The seakeeping problem is solved for the 6 rigid body motions. In order to obtain a perfect equilibrium of the structural model we use two main ideas (Malenica & al.): Pressure is computed directly in structural points (instead of interpolation); Pressure components are transferred separately, and hydrodynamic coefficients (added mass, damping, hydrostatics & excitation) are computed by integration over the structural mesh. The seakeeping motions are computed by solving the following equation:

\[
-\omega^2 \left[ \begin{bmatrix} m_R \end{bmatrix} + \begin{bmatrix} A_R \end{bmatrix} \right] - i \omega \left[ \begin{bmatrix} B_R \end{bmatrix} + \begin{bmatrix} C_R \end{bmatrix} \right] \begin{bmatrix} \phi_R \end{bmatrix} = \begin{bmatrix} F_R \end{bmatrix}
\]

(1)

where \( m_R \), \( A_R \), \( B_R \), \( C_R \), \( \phi_R \) and \( F_R \) are respectively the mass matrix, the hydrodynamic added mass matrix, the hydrodynamic damping matrix, the hydrostatic stiffness matrix, the motion amplitudes and the excitation forces. Subscript \( R \) stands for Rigid body.

The hydrodynamic pressures and the rigid body accelerations are then applied to the structural model and the response is calculated using FEM. It is clear that the above structural loading will be in perfect equilibrium because this equilibrium is implicitly imposed by the solution of the motion equation in which all different coefficients were calculated by using directly the information from the structural FE model.

If \( \{ \phi \} \) represents the degrees of freedom of the FEM model, \( [k] \) the stiffness matrix and \( \{ f \} \) the external forces (hydrostatic and hydrodynamic pressure forces, gravity and inertial forces) then the quasi-static computation done by the FE solver can be written:

\[
[k] [\phi] = [f]
\]

(2)

2.2 Linear hydroelastic dynamic model

In this model, the elastic distortions are considered in a dynamic sense, and rigid body motions and elastic distortions are coupled through inertial and hydrodynamic terms.

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

$$\{H\}(x, y, z, t) = \sum_{i=1}^{N} \xi_i(t) \{h_i\}(x, y, z)$$  \hspace{1cm} (3)$$

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

$$\frac{\partial \varphi_{RE}}{\partial n} = \{h_j\} \cdot \{n\}$$  \hspace{1cm} (4)$$

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

$$\left\{-\omega^2 \begin{bmatrix} m_R & m_{RE} \\ m_{E} & m_E \end{bmatrix} + \begin{bmatrix} A_R & A_{RE} \\ A_{ER} & A_E \end{bmatrix}\right\} - i\omega \begin{bmatrix} B_R & B_{RE} \\ B_{ER} & B_E \end{bmatrix} - \begin{bmatrix} C_R & C_{RE} \\ C_{ER} & C_E \end{bmatrix} - \begin{bmatrix} 0 & 0 \\ 0 & k_E \end{bmatrix}\right\}\begin{bmatrix} \xi_R \\ \xi_E \end{bmatrix} = \left\{F_R \right\}$$  \hspace{1cm} (5)$$

The difference with Eq. (1) is that all matrices are of dimension N by N and are composed of the 6 by 6 rigid body matrix (with subscript \( R \)) and some other terms corresponding to the elastic modes (subscripts \( RE, ER \) and \( E \)). \( k_E \) is the modal structural stiffness matrix. The motion includes the 6 rigid body modes \( \xi_R \) and a certain number of elastic modes \( \xi_E \). The solution of the above equation gives the motion amplitudes and phase angles by which the problem is formally solved.

2.3 *Linear hydroelastic quasi-static model* 

Before we introduce this intermediate hydroelastic quasi-static model, let’s first have a closer look at the rigid body quasi-static model described in 2.1. The quasi-static structural problem described in Eq. (2) can be rewritten in a reduced modal form, using the same elastic modes as those considered in Eq. (3) and (5). The external forces are then expressed in terms of modal forces, corresponding to excitation terms, inertial, radiation and hydrostatic restoring terms corresponding to rigid body motions:

$$\{k_E\}\{\xi_E\} = \{F_E\} - \left\{-\omega^2 \begin{bmatrix} m_{ER} \\ 0 \end{bmatrix} + \begin{bmatrix} A_{ER} & 0 \\ 0 & A_E \end{bmatrix}\right\} - i\omega \begin{bmatrix} B_{ER} & 0 \\ 0 & B_E \end{bmatrix} - \begin{bmatrix} C_{ER} & 0 \\ 0 & C_E \end{bmatrix} - \begin{bmatrix} 0 & 0 \\ 0 & k_E \end{bmatrix}\right\}\begin{bmatrix} \xi_R \\ \xi_E \end{bmatrix}$$  \hspace{1cm} (6)$$

Eq. (1) and (6) can then be combined in a single equation.

$$\left\{-\omega^2 \begin{bmatrix} m_R & 0 \\ m_{ER} & 0 \end{bmatrix} + \begin{bmatrix} A_R & 0 \\ A_{ER} & 0 \end{bmatrix}\right\} - i\omega \begin{bmatrix} B_R & 0 \\ B_{ER} & 0 \end{bmatrix} - \begin{bmatrix} C_R & 0 \\ C_{ER} & 0 \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & k_E \end{bmatrix}\right\}\begin{bmatrix} \xi_R \\ \xi_E \end{bmatrix} = \left\{F_R \right\}$$  \hspace{1cm} (7)$$

This equation is formally very close to Eq. (5). The differences are:

- Elastic motions do not produce any dynamic force (inertial or damping)
- Restoring forces for the elastic modes are only due to the structural stiffness matrix (hydrostatic restoring matrix is ignored).

What we propose here is an intermediate model, where the elastic distortions are considered in a quasi-static way (no inertial or damping effects), but the coupling between rigid body modes and elastic modes is considered in the hydrostatic stiffness matrix. The elastic and rigid modes are the solution of the following equation:
This model is a pure theoretical model that may not correspond to any real ship configuration. However it is used in this paper in order to separate the hydrostatic coupling effect from the hydrodynamic coupling effect and to emphasize that hydroelasticity is not only a dynamic effect.

3 COMPUTATIONS

3.1 Ships characteristics

6 container ships are chosen for this study. The following table shows their main characteristics, including their first wet natural frequency in bending and torsion.

<table>
<thead>
<tr>
<th>TEU</th>
<th>Loa (m)</th>
<th>Wet natural frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Torsion</td>
</tr>
<tr>
<td>≈ 5000</td>
<td>≈ 280</td>
<td>0.564</td>
</tr>
<tr>
<td>≈ 8000</td>
<td>≈ 330</td>
<td>0.390</td>
</tr>
<tr>
<td>≈ 9000</td>
<td>≈ 350</td>
<td>0.347</td>
</tr>
<tr>
<td>≈ 11000</td>
<td>≈ 360</td>
<td>0.321</td>
</tr>
<tr>
<td>≈ 13000</td>
<td>≈ 370</td>
<td>0.332</td>
</tr>
<tr>
<td>≈ 16000</td>
<td>≈ 390</td>
<td>0.292</td>
</tr>
</tbody>
</table>

The first 12 natural modes are used in the linear hydroelastic model. The mode shapes of the first three lowest modes are shown in Figure 1. These are a pure torsion mode, a combined horizontal bending and torsion mode, and a pure vertical bending mode. The natural frequencies, including hydrodynamic added mass, are given in Table 1.

A linear structural damping of 2% of the critical damping is used for all the modes. This structural damping is very difficult to evaluate, and is absolutely not linear. However the state of the art is to approximate this effect by a linear damping of about 0.5% to 3%. The effect of this damping will be discussed at the end of the paper.

The roll viscous damping has been modeled by a linear damping of 5%. The damping forces are spread along the bilge, and used in the computation of sectional loads.

Figure 1 : Mode shapes
3.2 **Hull girder loads RAOs**

The three hydro-structure models described in the first section are used with all the 6 ships, and the ship responses are computed for a speed of 15 knots and for every heading between 0 and 350° with a 10° step. Vertical bending moment and torsion moment RAOs are computed at several sections along the ship (between 20 and 28 sections, depending on the number of transverse bulkheads).

For each load the following RAOs are computed and compared:
- Rigid body quasi-static (RB-QS), using the linear rigid body quasi-static model
- Hydroelastic quasi-static (HE-QS), using the hydroelastic quasi-static model
- Hydroelastic dynamic (HE-Dyn), using the hydroelastic dynamic model
- Low frequent (LF), which is obtained from the hydroelastic dynamic RAO by applying a low pass filter with a cut frequency equal to 80% of the first natural wet mode (torsion mode).

Figure 2 shows a typical example of non dimensional vertical bending moment (VBM) RAO in head sea and torsion moment RAO in quartering sea (120°). These RAOs give a very good example of the different effect of hydroelasticity:

![Figure 2: VBM RAO at 0.5L (180°) and torsion RAO at 0.25L (120°)](image)

The first effect is the **hydrostatic relaxation**: due to the hydrostatic restoring forces, the QS VBM is lower than the RB VBM. By increasing the total stiffness, the hydrostatic forces are decreasing the hull girder loads. This effect is negligible in torsion, because the hydrostatic stiffness is very low compared to the structural stiffness, but is significant for vertical bending.

The second effect is the **dynamic amplification** at frequencies much lower than the resonance frequency. This tends to increase both VBM and torsion, even if the frequencies where the RB loads are maximum are far away from the resonance frequency. It is interesting to see that at the frequencies that maximize the RB VBM, the dynamic amplification effect and the hydrostatic relaxation effect are cancelling each other.

The third effect is the high frequency **resonance**. At the natural frequency of the first bending mode or torsion/horizontal bending mode, the ship response becomes very important.

3.3 **Simplified model**

In order to better understand the different effects of hydroelasticity, a simplified model is proposed. The ship distortion is modeled by a single degree of freedom, with a structural stiffness $k$. In a rigid body computation, the ship distortion is then directly proportional to the external wave forces, and inversely proportional to the stiffness $k$.

$$\xi_{RB-QS} = \frac{F}{k} \quad (9)$$

In a quasi-static approach including the hydrostatic stiffness $C$, the distortion is inversely proportional to the total stiffness $k+C$. Hence the ratio between the rigid body distortion and the quasi-static distortion is $k/(k+C)$:
\[ \xi_{HE-QS} = \frac{F}{k+C} = \xi_{RB-QS} \frac{k}{k+C} \quad (10) \]

In a dynamic approach, the ship distortion is amplified by the resonance, depending on the natural frequency \( \omega_0 \) and the damping ratio \( \eta \):

\[ \xi_{HE-D_{1dof}} = \xi_{HE-QS} \frac{1}{1 - \frac{\omega^2}{\omega_0^2} + 2i\eta \frac{\omega}{\omega_0}} = \xi_{RB-QS} \frac{k/(k+C)}{1 - \frac{\omega^2}{\omega_0^2} + 2i\eta \frac{\omega}{\omega_0}} \quad (11) \]

Figure 3 shows the amplification factor of this simplified model, for different values of damping ratio. The amplification at very low frequency is less than 1: this is the relaxation effect of the hydrostatic restoring forces. The amplification at the resonance frequency is highly dependant on the damping ratio. However at lower frequency, the amplification is still significant, and rather independent from the damping ratio.

![1 dof amplification factor](image)

Figure 3: Amplification factor of a 1 dof model

According to this simplified model, we can say that:

- The difference between the rigid body approach and the hydroelastic quasi-static approach is governed by the ratio between the hydrostatic stiffness and the structural stiffness;
- The difference between the low frequency approach and the hydroelastic quasi-static approach is governed by the ratio \( \omega/\omega_0 \), and is independent from damping;
- The difference between the dynamic approach and the low frequency approach is governed by damping.

### 3.4 Damage computation

Pseudo stress RAOs are computed from the bending moment or torsion RAOs assuming that the stress is proportional to one of those loads.

\[ \sigma_{VBM}(\omega) = \alpha \cdot VBM(\omega) \]
\[ \sigma_T(\omega) = \beta \cdot T(\omega) \quad (12) \]

\( \sigma_{VBM} \) represents the stress in a vertical bending moment governed detail. \( \sigma_T \) represents the stress in a torsion moment governed detail. The value of \( \alpha \) and \( \beta \) is not important since we will only look at relative differences between the computed damages.

Fatigue damage is computed using a spectral approach. The sea states are described by a Bretschneider spectrum and a directional spreading in \( \cos^2 \). The short term distribution of stress cycles is assumed to be described by a Rayleigh distribution. The fatigue damage is then computed using the Miner sum and a single slope S-N curve with \( m=3 \). Using a Rayleigh
distribution is making the assumption that the stress spectrum is narrow-banded, which is not true. However Boutillier & al.\textsuperscript{14} showed that for such kind of springing response, the difference of fatigue damage between a Rayleigh distribution and a Rainflow count is only in the order of 10%.

The long term fatigue damage is computed by summing all the short term fatigue damages, using the IACS scatter diagram which describes the probability of each sea state in North Atlantic. An equi probability has been taken for all the directions, and the speed is considered to be 15 knots in all the sea states.

Four different fatigue damages are computed, corresponding to the 4 different types of RAOs described in 3.2: Rigid body quasi-static damage ($D_{RB-QS}$), Hydroelastic quasi-static damage ($D_{HE-QS}$), Hydroelastic dynamic damage ($D_{HE-Dyn}$) and Low frequency damage ($D_{LF}$). The impacts of the three hydroelastic effects on fatigue damage are separated using the following increase factors:

$$1 + \alpha_{HS} = \frac{D_{HE-QS}}{D_{RB-QS}} \quad ; \quad 1 + \alpha_{LF} = \frac{D_{LF}}{D_{HE-QS}} \quad ; \quad 1 + \alpha_{HF} = \frac{D_{HE-Dyn}}{D_{LF}}$$

where $\alpha_{HS}$ is the increase due to \textit{hydrostatic relaxation}, $\alpha_{LF}$ is the increase due to low frequency \textit{dynamic amplification}, and $\alpha_{HF}$ is the increase due to \textit{resonance}.

The hydroelastic contribution to fatigue damage is defined as the part of damage due to hydroelasticity:

$$\% \text{Elastic} = \frac{D_{HE-Dyn} - D_{RB-QS}}{D_{HE-Dyn}} = 1 - \frac{1}{1 + \alpha_{HS}} \frac{1}{1 + \alpha_{LF}} \frac{1}{1 + \alpha_{HF}}$$

The high frequency contribution to fatigue damage is different, as it is only the part due to high frequency resonance effect:

$$\% HF = \frac{D_{HE-Dyn} - D_{LF}}{D_{HE-Dyn}} = 1 - \frac{1}{1 + \alpha_{HF}}$$

4 \quad RESULTS AND DISCUSSION

4.1 Hydrostatic restoring effect

The rigid body damage and the quasi-static damage are computed in vertical bending and in torsion at all the sections along the ship. Their comparison gives the hydrostatic restoring effect as defined in Eq. (13). Figure 4 shows this effect along the ship length for all the 6 ships. The effect on torsion damage is negligible, whereas it varies from -8% to -18% on midship VBM damage.

According to Eq. (10), the hydrostatic effect is mainly depending on the ratio $C/k$. For the torsion mode and the horizontal bending mode, this ratio is below 0.5% for all ships. For the
vertical bending mode, this ratio is between 3% and 7%. According to Eq. (10), the increase due to hydrostatic relaxation should be:

$$\alpha_{HS} = \frac{D_{HE-QS}}{D_{RB-QS}} - 1 = \left(\frac{k}{k + C}\right)^3 - 1 = \frac{1}{(1 + C/k)^3} - 1$$  \hspace{1cm} (16)

For each ship, the average of the hydrostatic relaxation for all the bulkheads between 0.2Lpp and 0.8Lpp is plotted against the ratio C/k, either for vertical bending (for VBM damage), or for horizontal bending (for torsion damage). Figure 5 shows that hydrostatic relaxation effect on fatigue damage is strongly related to the ratio C/k and that the simplified model predicts this effect quite well. It is interesting to notice that this effect is a pure geometrical effect. It is independent from speed, and independent from the damping ratio. This is the same effect that reduces the calm water bending moment when the ship is considered flexible.

![Figure 5](image)

**Figure 5**: Hydrostatic effect on fatigue damage

### 4.2 Low frequency dynamic amplification

Figure 6 shows the damage increase due to dynamic amplification effect. This effect tends to increase with the ship size. It varies from 8% to 24% for midship VBM damage, from 8% to 32% for torsion damage at 0.25 Lpp and from 7% to 38% for torsion damage at 0.75 Lpp.

![Figure 6](image)

**Figure 6**: Low frequency amplification effect on fatigue damage

As shown in Figure 3, the increase due to low frequency dynamic amplification is mainly depending on the ratio between the encounter frequency and the natural frequency. The maximum fatigue damage occurs for sea states having their peak energy content corresponding to the peak of the stress RAO.

- For vertical bending moment, this peak is located in head sea, at a wave length corresponding to 90% of the ship length;
\[ \lambda = 0.90 \cos(\mu) \cdot L_{pp} \quad ; \quad \mu = 180^\circ \] (17)

- For torsion moment, this peak is located in quartering sea (120°), when the projection of the wave length on the ship direction is equal to 85% of the ship length.

\[ \lambda = 0.85 \cos(\mu) \cdot L_{pp} \quad ; \quad \mu = 120^\circ \] (18)

This is illustrated in Figure 7, showing the VBM and Torsion RAOs of the 6 ships in a non dimensional way.

Figure 7 : Non dimensional VBM RAO (180° at 0.5L) and torsion RAO (120° at 0.25L)

The increase in fatigue damage due to low frequency amplification is then mainly governed by the amplification at the frequency of the maximum of the rigid body RAO. According to Eq. (11), and neglecting the imaginary damping term, the damage increase due to low frequency amplification should be:

\[ \alpha_{LF} = D_{LF} / D_{HE-QS} - 1 = \left( 1 - \frac{\omega_e^2}{\omega_0^2} \right)^{-1} ; \quad \omega_e = \omega - \omega^2 \frac{U}{g} \cos(\mu) ; \quad \omega = \sqrt{\frac{2 \pi g}{\lambda}} \] (19)

Figure 8 shows for each ship the average of the low frequency dynamic amplification for all the bulkheads between 0.2Lpp and 0.8Lpp, plotted against the ratio \( \omega_e / \omega_0 \), either for vertical bending (for VBM damage), or for horizontal bending (for torsion damage). It shows that low frequency amplification effect on fatigue damage is strongly related to the ratio \( \omega_e / \omega_0 \) and that the simplified model predicts this effect quite well. This effect is speed dependent, as it depends on the encounter frequency. It is however independent from the damping ratio.

Figure 8 : low frequency amplification effect on fatigue damage
4.3 High frequency resonance amplification

Figure 9 shows the damage increase due to high frequency resonance effect. This effect varies from 22% to 31% for midship VBM damage, from 1% to 19% for torsion damage at 0.5 Lpp and from 26% for torsion damage at 0.75 Lpp.

It is difficult to model this effect with the simplified model proposed in 3.3. The increase of fatigue due to HF is depending on the height of the resonance peak at the natural frequency, which depends on the damping ratio, and on the excitation forces. The computations have been done with two values of damping (2% and 1%). Figure 10 shows for each ship the average of the high frequency resonance effect for all the bulkheads between 0.2Lpp and 0.8Lpp, plotted against the wave frequency corresponding to resonance, either for vertical bending in head sea (for VBM damage), or for horizontal bending in quartering sea (for torsion damage). It shows that the influence of high frequency depends a lot on the amount of damping, and that damage increase is higher when the resonance frequency is corresponding to a low wave frequency (for which there is energy in the sea spectrum). It is not possible however to derive a clear relation as it was done in 4.1 and 4.2.

![Hydroelastic HF increase of VBM fatigue damage](image)

![Hydroelastic HF increase of Torsion fatigue damage](image)

Figure 9 : Resonance effect on fatigue damage (2% structural damping)

![Increase in fatigue damage due to high frequency resonance](image)

Figure 10 : Resonance effect on fatigue damage

4.4 Difference between high frequency effect and hydroelastic effect

The hydroelastic contribution to fatigue damage, defined in Eq. (14), and the high frequency contribution defined in Eq. (15) are now compared in Figure 11.

For VBM damage these two contributions are more or less the same. It is explained by the fact that the hydrostatic relaxation effect and the low frequency dynamic amplification effect are counteracting and are cancelling each other. Hence the total hydroelastic contribution is only governed by the high frequency resonance contribution. Thus taking the high frequency part of the VBM, measured at model scale or at full scale, is a good approximation of the total hydroelastic effect.
However for torsion damage, there is no hydrostatic relaxation effect and the total hydroelastic effect is the sum of the low frequency amplification effect and the high frequency resonance effect. Figure 12 shows the ratio between HF contribution and elastic contribution to fatigue damage. For torsion damage, the low frequency effect is more important than the high frequency effect for smaller ships. The two effects are about the same for the longest ships. Hence considering only the high frequency effect from full scale measurement largely underestimate the total hydroelastic effect on fatigue damage.

CONCLUSIONS

The effects of hydroelasticity on fatigue damage have been discussed, based on hydroelastic computations done for 6 container ships. Three effects can be highlighted. A pure quasi-static effect due to the hydrostatic restoring forces tends to reduce the internal loads. A dynamic amplification effect at frequencies lower than the resonance frequencies tends to increase significantly the quasi-static loads. Around the natural frequency, a very high springing peak produces an important resonance.

The conclusion is that for vertical bending damage the hydrostatic restoring effect and the low frequency dynamic amplification effect are cancelling each other: hence the total hydroelastic contribution is equal to the high frequency contribution. The state of the art to interpret full scale measurement of vertical bending moment is based on that lucky equivalence.
However for torsion damage there is no hydrostatic restoring effect and an important part of the hydroelastic effect is located at low frequency: the hydroelastic contribution to the total fatigue damage is larger than the pure high frequency contribution that can be measured from full scale monitoring. Even in the case when the HF contribution is negligible (due to a high damping ratio for instance) the damage increase due to hydroelasticity remains significant. This means that all the evaluations of the influence of hydroelastic effects in torsion coming from full scale measurement or model tests are a lower limit of the real hydroelastic effect.

In this study only linear springing is considered. More generally, non linear springing and whipping response will probably increase the high frequency part of fatigue damage.

REFERENCES