HYDRO-ELASTIC ISSUES IN THE DESIGN OF ULTRA LARGE CONTAINER SHIPS – TULCS PROJECT

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ABSTRACT

The paper discusses the actual tools and methodologies used in the design of Ultra Large Container Ships (ULCS). The specificity of the ULCS, compared to the more classical ship designs, is that they are more likely to experience the hydro elastic type of structural response called springing and whipping. There are several reasons for that:

- increase of the flexibility due to their large dimensions (Lpp close to 400m) which leads to the lower structural natural frequencies
- very large operational speed (> 20knots)
- large bow flare (increased slamming loads)

Indeed all these facts increase the risk of the hydroelastic wave induced resonance in a moderate sea states (springing) and, at the same time, the risk of slamming is increased leading to higher probability of the important transient hydroelastic response (whipping).

These issues are receiving more and more attention in the recent past due to the novel ULCS projects which goes up to 18500TEU in capacity and close to 400m in length. The evaluation of the hydroelastic ship structural response, and its inclusion into the overall design procedure, is significantly more complex than the evaluation of the quasi static structural response, which is already non-trivial. The present paper discusses the actual status of the different tools and methods which are used in practice.

KEYWORDS

Hydro-structure interactions; hydroelasticity; springing; whipping; slamming; potential flow, boundary integral equation method; finite element method; CFD; model tests; full scale measurements

1. INTRODUCTION

The practical procedure for ship structural design involves the verification of two main structural failure modes:

- Yielding and buckling failure due to extreme event
- Fatigue initiated cracks in the structure

These two failure modes are fundamentally different and the methodologies for their assessment are also different even if some common points exist. The final goal of the extreme event analysis is to predict, for each structural member, the single most likely worst event during whole ship life while the goal of the fatigue analysis is to analyze the whole ship life by counting all the combinations of the stress ranges and number of cycles (S-N curves) for particular
structural detail.

For the classical ships (tankers, bulk-carriers, general cargo ships …), not exceeding certain size, the usual design practice passes through the direct application of the prescribed rules and procedures issued by different Classification Societies. In the case of extreme structural response, these procedures do not involve fully direct hydro-structure calculations and the final design load cases are given in the form of the equivalent simplified load cases which are constructed as a combination of the different extreme loading conditions for assumed operating conditions. Even if the procedure for determination of the extreme loading conditions relies partially on hydrodynamic calculations, the rule approach remains basically prescriptive approach with an important part of empiricism. On the side of the structural strength, other safety coefficients are introduced and the final calibration of the rule approach is done using the extensive feedback from experience which ensures the excellent safety record of the existing ships. Due to these calibration procedures it is not possible, in principle, to use the rule procedures for a novel designs which do not enter in the initial assumptions of the considered design (ship type, size…) and operations. As far as the fatigue life is concerned the rule approach uses the similar equivalent load case approach which allows for very rough verification of the fatigue life.

Within the so called direct calculation approach for the assessment of the ship structural reliability, the basic idea is very simple: the structural response of the ship should be directly calculated during whole her life using the fully coupled hydro-structural models and the identification of the extreme events and fatigue life will be determined directly.

Since the fully consistent non-linear hydro-structure calculations are not practically possible within the reasonable combination of CPU time and accuracy, one must consider some approximate solutions using the different levels of approximation at different steps of the overall methodology. One of the main purposes of the present paper is to discuss the actual state of the art of the different models.

2. HYDRO STRUCTURE INTERACTION MODELS

Before entering into the details of the different numerical models, let us first try to classify the different hydro-structural issues. In Table 1 the different issues are schematically separated with respect to the nature of the hydrodynamic loading and the nature of the structural response.

<table>
<thead>
<tr>
<th>TABLE 1</th>
<th>DIFFERENT HYDRO-STRUCTURAL ISSUES</th>
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<tr>
<td>QUASI STATIC</td>
<td>LINEAR</td>
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<td>DYNAMIC</td>
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As far as the hydrodynamic loading is concerned, the usual practice is to classify it into 3 different categories:

- linear hydrodynamic loading
- weakly nonlinear, non-impulsive loading
- impulsive hydrodynamic loading

Within the potential flow hydrodynamic models, which are of main concern here, the linear hydrodynamic loading means the classical linear diffraction radiation solution also called the seakeeping hydrodynamic analysis. The weakly non-linear loading means the non-impulsive part of the wave loading which is usually covered through the different variants of the so called Froude Krylov approximation which is combined with the large ship motions. The impulsive loading includes any type of the transitory loading such as slamming, green-water, underwater explosion … In this paper slamming loading is considered only.

On the structural side the structural response can be classified into two main types:

- quasi static
- dynamic (often also called hydroelastic)
There is sometimes a certain misunderstanding observed in the literature regarding this separation mainly because both quasi static and dynamic structural responses are due to the dynamic loading. However, the fundamental difference in between the quasi static and dynamic structural response lies in the fact that the quasi static response does not account for the structural vibrations while the hydroelastic dynamic response does.

In the sections to follow, the different combinations of the hydrodynamic loading and structural responses are discussed more in details.

2.1 Linear quasi static hydro structure interactions

<table>
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<th>TABLE 2</th>
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<tr>
<td>LINEAR QUASI-STATIC HYDRO-STRUCTURE MODEL</td>
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Linear quasi static hydro-structure model represents the basis of any subsequent hydro-structure interaction methods. This is the simplest case of hydro-structure interactions but unfortunately the problem is still open for a general case which is mainly due to the difficulties related to the solution of the seakeeping problem with forward speed.

In order to make the procedure as efficient as possible the problem is usually formulated in frequency domain which is possible due to the assumptions of linearity of both the hydrodynamic loading and structural response. This leads to the definition of the RAO’s (Response Amplitude Operators) for different ship responses (motions, internal loads, stresses…) from which the maximum response for a given operating conditions (loading condition, speed, heading & scatter diagram) can be calculated using the spectral analysis.

The good point in the case of the quasi static structural responses is that the hydrodynamic and structural calculations can be performed separately. The usual procedure passes through the solution of the rigid body diffraction radiation problem using the Boundary Integral Equation (BIE) technique. Thanks to the linearity the problem is formulated in frequency domain and the total velocity potential and corresponding hydrodynamic pressure is decomposed into the incident, diffracted and 6 radiated components:

\[ \varphi = \varphi_I + \varphi_D - i\omega \sum_{j=1}^{6} \xi_j \varphi_{Rj} \quad , \quad p^{hd} = p_I + p_D - i\omega \sum_{j=1}^{6} \xi_j p_{Rj} \] (1)

The potential is calculated numerically using the Boundary Integral Equation technique in which the mean wetted body surface is discretized into a certain number of panels (see Figure 1) over which the singularity distribution is assumed in a certain form (constant or higher order). Without entering into much details of the BIE procedure, let us just mention that, in the general case, the final potential can be written in the form of the singularities distribution over the mean wetted body surface \( S_B \). There are different types of the singularity distributions and the simplest one is based on the so called pure source distribution:

\[ \varphi(x) = \iint_{S_B} \sigma(\xi) G(x;\xi) dS \] (2)

where \( \sigma(\xi) \) stands for the source strength and \( G(x;\xi) \) is the Green’s function.

Once calculated, the pressures are integrated over the mean wetted part of the body and the hydrodynamic coefficients are calculated so that the following rigid body motion equation can be written:

\[ \{-\omega^2 \begin{bmatrix} [M] + [A(\omega)] \end{bmatrix} - i\omega \begin{bmatrix} [B(\omega)] \end{bmatrix} + [C]\} \{\xi\} = \{F^{DI}(\omega)\} \] (3)

where:
\[
\begin{align*}
&M \quad \text{genuine mass matrix of the body} \\
&A(\omega) \quad \text{hydrodynamic added mass matrix} \\
&B(\omega) \quad \text{hydrodynamic damping matrix} \\
&C \quad \text{hydrostatic restoring matrix} \\
&\{F_{DI}(\omega)\} \quad \text{hydrodynamic excitation vector} \\
&\{\xi\} \quad \text{body motions vector}
\end{align*}
\]

The solution of the motion equation gives the body motions and the seakeeping problem is formally solved.

The next step consists in transferring the loading from hydrodynamic model to the structural finite element model. This is the critical step in the procedure and should be done with greatest care in order to build fully consistent loading cases which perfectly balance the rigid body inertia and the hydrodynamic pressure loads. As far as the rigid body inertia is concerned, the situation is simple and we should just make sure that the rigid body mass matrix is evaluated using the mass distribution from the FE model.

![Figure 1: Hydrodynamic (top) and structural mesh (bottom) of the Container ship.](image)

Concerning the pressure part, the main problem comes from the fact that the hydrodynamic and structural FE meshes are usually quite different, because they were built according to very different philosophies (see Figure 1). This means that an efficient procedure for pressure transfer is necessary in order to consistently apply the hydrodynamic pressure onto the structural finite elements. If this step is not performed properly the final loading case will not be balanced and the structural response will be wrong especially close to the artificial supports, which have to be included in the FE structural model for free-free types of structures such as ships.

Most of the methods, which are used in practice, employ the different interpolation schemes in order to transfer the total hydrodynamic pressure (1) from hydrodynamic model (centroids of the hydro panels) to the structural model (centroids or nodes of the finite elements). Besides the problems of complex interpolation in 3 dimensions, it is important to note that the motion amplitudes, which are present in the definition of the total pressure, were calculated after the integration over the hydrodynamic mesh so that it is practically impossible to obtain the completely balanced structural model. This is because the FEM model has its own integration procedure which is usually very different from the one used in hydrodynamic model.

In order to obtain the perfect equilibrium of the structural model we introduce here two main ideas:

- Recalculation of the pressure at the structural points, instead of interpolation
- Separate transfer of the pressure components, and calculation of the hydrodynamic coefficients (added mass, damping, hydrostatics & excitation) by integration over the structural mesh

The first point is possible thanks to the very useful property of the Boundary Integral Equation technique which allows for the recalculation of the velocity potential at any point in the fluid as shown by Eqn. (2). This leaves us the choice to choose the characteristic points of the wetted structural finite elements and to perform the pressure integration on the FE mesh. This is important in order to make sure that the same body motions are used both for the rigid body inertia and for the re-composition of the total hydrodynamic pressure (1).

It is also important to mention that the integration of the pressures over the FEM mesh is
performed using the Gauss points at the finite elements. These Gauss points are used for integration only, and have nothing to do with the Gauss points used in the theory of the structural FEM program. The accuracy of the integration can be controlled by changing the number of Gauss points per element. The pressure evaluation at the Gauss points of the FEM mesh should be done carefully. Indeed, although the BEM and FEM mesh will model the same geometry, the meshing itself can be very different. Especially at curved parts of the geometry the Gauss points at the FEM mesh will not be exactly at the BEM mesh and part of the Gauss points might fall inside the BEM mesh. This causes problems for calculation of the pressure for bodies travelling with forward speed because in that case the hydrodynamic pressure also depends on the gradient of the potential which is discontinuous across the hydrodynamic mesh. Special preprocessing of the Gauss points is thus necessary.

Finally let us also discuss one important issue related to the application of the hydrostatic restoring action which is a bit specific and sometimes misunderstood in the literature. The confusion comes from the fact that the usual practice for hydrodynamic calculations is to solve the problem in the initial earth fixed reference frame while the structural calculations are usually done in the body fixed coordinate system. Due to the assumptions of linearity both methods are fully correct and the eventual differences are of higher order.

When calculating the linear hydrostatic restoring forces and moments the total contribution can be separated in two parts. The first one is the same for both earth fixed and body fixed coordinate systems and concerns the change of the hydrostatic pressure due to ship motions:

\[
p^\text{hs1} = -\rho g [\xi_3 + \xi_4(Y - Y_G) + \xi_5(X - X_G)]
\]

where \((X_G, Y_G)\) are the coordinates of the ship center of gravity.

The second one depends on the coordinate system in which the motion equation is written. In the case of the earth fixed coordinate system, which is used when formulating the hydrodynamic problem, this term is associated with the change of the normal vector:

\[
p^\text{hs2} = -\rho g \Omega \times n
\]

where \(\Omega\) denotes the rotational component of the motion vector \(\Omega = (\xi_4, \xi_5, \xi_6)\). Note that this term is pressure term and should be applied on the mean wetted body surface.

In the body fixed reference frame the normal vector does not change but the gravity changes, so that the term equivalent to (5) becomes:

\[
f^\text{hs2} = -mg \Omega \times k
\]

where \(k\) denotes the unit vector in Z direction. Note that this term is associated with the gravity forces and should be applied on each mass element of the finite element model.

It is possible to show that these two terms are completely equivalent [7]. Anyhow, the final loading of the FE model is composed of 3 parts

\[
\begin{align*}
-\omega^2 m_i \xi_i & \quad \text{ - Rigid body inertial loading (on each finite element)} \\
p^\text{hd} + p^\text{hs1} & \quad \text{ - Pressure loading (on wetted finite elements only)} \\
-m_i g \Omega \times k & \quad \text{ - Additional gravity term (on each finite element)}
\end{align*}
\]

In order to avoid the possible differences in between the pressure application in the different FEM packages, the pressure loading is applied on the structural model in the form of the nodal forces instead of the pressures. This means that the pressure integration over the finite elements is performed in a preprocessing stage using the prescribed finite elements shape functions together with the pressure values at the FE Gauss points.

It is clear that the above structural load cases will perfectly balance pressure and inertia components because this equilibrium is implicitly imposed by the solution of the motion equation.
in which all the different coefficients were calculated using the information from the structural FE model directly.

It is also important to mention the fact that, usually in practice, we are interested in the very local stress concentration at some particular structural details which means that the finite element model should be very refined around those details. This might lead to the prohibitive number of finite elements in the case when all these structural details are included into the global FE model. Practical way to solve this problem is based on the so called top-down procedure. This procedure consists in solving the global coarse mesh FE problem first, and in applying the coarse mesh displacements at the boundaries of the fine mesh after. In this way the fine mesh FE calculations are performed in the second step with the load cases defined by the prescribed displacements from the coarse mesh and by the local pressure and inertia of the fine mesh. Within the hydro-structure procedure presented here the implementation of the top-down procedure is rather straightforward [12].

Let us finally note that the above procedure should be performed for each operating condition (loading condition, wave frequency & heading) and for the real and imaginary part of the loading. The final results are the RAO’s of stresses in the particular structural details (Figure 2). Once the RAO’s calculated a spectral analysis is used in order to calculate the characteristic stresses (mean, significant, maximum …) in a particular sea state.

![Figure 2: Top down procedure and typical stress RAO.](image)

### 2.2 Linear hydroelastic hydro structure interactions

**TABLE 3**

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<th>LINEAR HYDROELASTIC HYDRO-STRUCTURE MODEL</th>
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<td>QUASI STATIC</td>
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<td>DYNAMIC</td>
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Depending on the ratio in between the structural natural frequencies and the wave excitation frequency the dynamic amplification of the structural response will be more or less important. In order to calculate the dynamic amplification factor it is necessary to use the full hydroelastic coupling procedure. There are different ways to do that and in this work the so called modal approach is used. Within this approach the total ship displacement is represented as a series of the different modal displacements:

\[ H(x, t) = \sum_{i=1}^{N} \xi_i(t) h^i(x) \]  

where:
- \( H(x, t) \) total displacement of one point on the body
- \( h^i(x) \) modal displacements (mode shape)
- \( \xi_i(t) \) modal amplitude
The rest of the procedure is very similar to rigid body analysis except that the number of degrees of freedom is increased from 6 to 6 plus a certain number of elastic modes. This modal approach implies the definition of supplementary radiation potentials with the following body boundary condition:

\[ \frac{\partial \varphi_{Rj}}{\partial n} = h^i n \]  

(8)

After solving the different boundary value problems for the potentials, the corresponding forces are calculated and the motion equation similar to (3) is written:

\[ \{-\omega^2([m] + [A]) - i\omega([B] + [b]) + ([k] + [C])\}{\xi} = \{F^D\} \]  

(9)

where \([m]\) is the modal structural mass, \([b]\) is the structural damping, \([k]\) is the structural stiffness, \([A]\) is the hydrodynamic added mass, \([B]\) is the hydrodynamic damping, \([C]\) is the hydrostatic restoring, \({\xi}\) are the modal amplitudes and \(\{F^D\}\) is the modal hydrodynamic excitation.

Contrary to the quasi static case where the hydrodynamic pressure need to be transferred from the hydrodynamic mesh to the structural FE mesh, in the present case the radiation boundary condition (8) implies the transfer of the structural modal displacements from the structural mesh to the hydrodynamic mesh. Since, within the FE method, it is not possible to recalculate directly the displacements at any required point, the non-trivial interpolation procedure is necessary [9]. Typical result of this interpolation is shown in Figure 3.

Figure 3: First natural structural mode and transfer of the modal displacements from structural to hydrodynamic mesh.

Once the modal amplitudes have been calculated the total stresses can be calculated, at least theoretically, by summing the individual modal contributions and we can formally write:

\[ \Sigma(x, \omega) = \sum_{i=1}^{N} \xi_i(\omega)\sigma_i(x) \]  

(10)

where \(\Sigma(x, \omega)\) is the total stress, \(\sigma_i(x)\) is the spatial distribution of modal stresses and \(\xi_i(\omega)\) are the modal amplitudes. Note that rigid body modes do not contribute to the stresses.

The convergence of this series is in general very slow and it is not practically possible to include very large number of modes. As far as the dynamic amplification effects are concerned this is not important because the main dynamic contribution to the stresses comes from the first few lowest structural modes and the rest is quasi static. In order to make the stress calculation procedure efficient, it is thus necessary to calculate the dynamic and quasi static contributions separately. The way how to perform this separation was presented in [9] and one typical example...
of the final stress RAO decomposition is shown in Figure 2. Finally let us also note that the, previously discussed, top-down procedure for evaluation of the local stresses needs also to be adapted, in order to include the dynamic amplification effects [12].

2.3 Non-linear quasi static and hydroelastic hydro-structure interactions

In order to include the nonlinear effects into the dynamic model, the usual practice is to work in time domain, even if some particular problems might also be solved in frequency domain using the higher order hydrodynamic theories. However, these higher order theories are not practical in the present context of calculation of the structural stresses. There are different ways of performing the time domain simulations but probably the most practical one is based on using the linear frequency domain results and transferring them into time domain following the approach proposed by Cummins [3]. It is important to note that this approach is valid for both the quasi static ship response as well as for the hydroelastic one. Indeed, the only difference is the number of degrees of freedom which is increased in the hydroelastic model.

Within this approach the motion equation is written in the following form:

\[
([\mathbf{m}] + [\mathbf{A}^\infty])\ddot{\mathbf{x}}(t) + ([\mathbf{k}] + [\mathbf{C}])\mathbf{x}(t) - [\mathbf{b}]\dot{\mathbf{x}}(t) + \int_0^t [\mathbf{K}(t - \tau)]\dot{\mathbf{x}}(\tau) d\tau = \{\mathbf{F}(t)\} + \{\mathbf{Q}(t)\}
\]

where:
- \([\mathbf{m}]\) modal mass matrix
- \([\mathbf{A}^\infty]\) infinite frequency modal added mass matrix
- \([\mathbf{k}]\) structural stiffness matrix
- \([\mathbf{C}]\) hydrostatic restoring matrix
- \([\mathbf{b}]\) structural damping matrix
- \([\mathbf{K}(t - \tau)]\) impulse response (memory) functions matrix
- \([\mathbf{F}(t)]\) linear excitation force vector
- \([\mathbf{Q}(t)]\) nonlinear excitation force vector
- \([\mathbf{x}(t)]\) body motions/deformations vector

It was shown in [3], that the impulse response functions can be calculated from the linear frequency dependent damping coefficients using the following relation:

\[
K(t) = \frac{2}{\pi} \int_0^\infty B(\omega) \sin \omega t \ d\omega
\]

Once the impulse response function matrix \([\mathbf{K}(t)]\) calculated, the motion equation (11) is integrated in time and the nonlinear forces are added at each time step to the vector \([\mathbf{Q}(t)]\). There are various types of non-linearities which have to be included in the hydrodynamic model and it is not practically possible to account for all them.

Within the nonlinear effects which are believed to be particularly important in ship design, there are two main types: the first one is the Froude Krylov loading combined with large ship motions, and the second one is the strongly non-linear impulsive loading such as slamming. Note that, on the structural side, the response remains linear.

2.3.1 Weakly nonlinear hydrodynamic loading

| TABLE 4 |
| WEAKLY NON-LINEAR HYDROELASTIC HYDRO-STRUCTURE MODEL |

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<th>LINEAR</th>
<th>QUASI STATIC</th>
<th>DYNAMIC</th>
<th>IMPELLUS NON LINEAR</th>
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<tbody>
<tr>
<td>WEAVE NON LINEAR</td>
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The simplest weakly nonlinear wave loading concerns the so called Froude Krylov model which we briefly discuss here below. This model is relevant in practice both for the local fatigue loading of the side shell structural details close to the waterline, and for the modifications of the nonlinear
internal loads distribution (bending moments, shear forces…).

According to the linear theory, the hydrodynamic model "stops" at the waterline \((z = 0)\) so that locally (close to the waterline), negative hydrodynamic pressures might occur. There exist different variants of the Froude Krylov model and the simplest one is rather intuitive and consists in adding the hydrostatic part of pressure below the wave crest (in linear sense) and by putting zero total pressure above the wave trough. The problem basically reduces to the evaluation of the (linear) wetted part of the ship at each time step.

![Figure 4: Linear (left) and nonlinear (right) hydrodynamic pressure distribution.](image)

On hydro-structure interaction side the situation is significantly more complex when compared to the linear case. In addition to the obvious technical difficulties (large motions, identification of wetted FE elements…), the radiation component of the loading appear to be particularly difficult to take into account. The direct application of the Cummins method implies the evaluation of the impulse response functions for every structural point, but the calculation effort to do this is huge. The approach which is chosen here is based on the use of the frequency domain radiation pressures only. This approach implies performing the hydro-structure interaction calculations as a post-processing of the seakeeping calculations because the Fourier transform on the seakeeping velocity must be done to obtain the amplitudes and phases of the different frequency components. Even if these manipulations will results in small numerical inaccuracies, they appear to be practically negligible for the balancing of the structural FE model.

### 2.3.2 Slamming & Whipping

**TABLE 5**

| **IMPELLSIVE NON-LINEAR HYDROELASTIC HYDRO-STRUCTURE MODEL** |
|---|---|---|
| **S** | **H** | **IMPULSIVE NON LINEAR** |
| QUASI STATIC | X | X | X |
| DYNAMIC | X | X | X |

Slamming represents very important source of ship structural loading both from local and global points of view. Very high localized pressures appear during the slamming event, and at the same time the corresponding overall forces are very high. This means that not only the local ship structure will be affected by slamming, but whole ship will “feel” the slamming loading through the so called whipping phenomena. Whipping is defined as the transitory global ship vibrations due to slamming and one example of the typical whipping response is shown in Figure 5, where it can be clearly seen that the quasi static loads are significantly increased by the high frequency whipping vibrations.

The hydrodynamic modeling of slamming is extremely complex and still no fully satisfactory slamming model exists. However, the 2D modeling of slamming appears to be well mastered today and the 2D models are usually employed in practice. 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 nonlinear model. In between these two models there are several intermediate ones among which is the Generalized Wagner Model (GWM) first introduced in [14]. GWM allows for evaluation of the impact pressure along the arbitrary ship section and for arbitrary penetration velocity.

![Figure 5: Typical full scale measurements of whipping.](image)

Integration of the 2D GWM slamming model into the overall hydroelastic model is not trivial and strong coupling is required at each time step. Indeed the overall relative velocity of each section is determined from the global dynamic model (11) and is used as an input for GWM through an iterative procedure. Due to the 2D assumption of the GWM, the practical procedure to include 3D effects passes through the so called strip approach where the part of the ship (usually ship bow and stern) is cut into several strips each of them being considered separately from slamming point of view, as shown in Figure 6.

![Figure 6: Subdivision of the bow part of the ship for slamming calculations.](image)

The GWM provides the impact pressure along the ship section and this pressure is integrated over the overall FE model using the following relation:

$$F^i = \sum_{j=1}^{N_s} \int_{L_j} p_j \mathbf{h}_i^j \mathbf{n}_j b_j \, dl_j$$  \hspace{1cm} (13)

where $F^i$ is the slamming load projected on the $i$-th mode, $p_j$ is the slamming pressure on the section $j$ and $\mathbf{h}_i^j, \mathbf{n}_j, b_j$ are the mode shape $i$, normal vector and width of the section $j$ respectively.

Due to the separate slamming calculations for each strip, the non-trivial interpolation procedures in space and time are necessary in order to properly transfer the impact pressure onto global FE model of the ship.

Compared to the well-known original Wagner model, which represents the reference for the impact problems, GWM allows accounting for the exact body geometry. The price to pay is that the Boundary Value Problem for velocity potential at each penetration depth should be solved numerically. This leads to a very significant increase of the overall CPU time for whipping calculations. In order to reduce the CPU time it is possible to use the property of the GWM solution which allows for the separation of the different slamming pressure components in the following form:

$$p(t) = P_v(\zeta, \eta)\dot{\zeta}^2(t) + P_w(\zeta, \eta)\dot{\zeta}(t)$$  \hspace{1cm} (14)
where $\zeta(t)$ is the relative penetration depth and $\eta$ is the transversal coordinate on the ship section.

This fact allows for pre-calculating the sectional slamming characteristics and to use the look-up tables for interpolation of slamming pressures and forces for the instantaneous penetration depth. In this way a huge amount of the CPU time can be saved.

One example of the typical whipping numerical simulations obtained, using the present method, is shown in Figure 7.

![Bending moment](image)

Figure 7: Typical numerical simulations of whipping.

### 3. DESIGN METHODOLOGIES

As mentioned in the introduction the basic idea of the direct approach should be very simple: the structural response of the ship will be directly calculated during whole her life and the identification of the extreme events and fatigue life will be determined directly. There are two main difficulties which need to be resolved before being able to proceed in this way:

- **Modeling difficulties (numerical or experimental)**
- **Choice of the representative operating conditions**

Both issues are equally important. The difficulties related to the pure hydro-structure modeling issues were discussed in the previous sections and here below we briefly discuss and comment the difficulties related to the choice of the representative operating conditions.

In order for the design methodology to be consistent, the most important point is, probably, the choice of the ship operational profile which needs to be done in as reasonable way as possible. Indeed, we do not know in advance in which real conditions ship will operate but the ship owner usually requires the possibility to operate worldwide which means that the methodology should take into account the fact that the ship will encounter the most extreme wave events existing in all the oceans worldwide. At the same time we should keep in mind that the operating conditions do not means the sea state definition only and the sea state should always be associated to the ship loading conditions, wave heading and ship’s speed. How the ship master will operate the ship in particular sea conditions is big question. Not every ship master will chose the same decisions and all possible decisions should be taken into account in order to cover the most critical cases. The feedback from the ship operations clearly shows that the sister ships do not experience the same fatigue life consumptions on the same sailing route, when operated by different people. How this should be taken into account?

The usual practice is to fix in advance some combinations of speeds and headings which are chosen in rather empirical way which unfortunately appears to be “too empirical” and not very logical in some cases. At the same time the most common choice of the worst sea states is the so called North Atlantic scatter diagram also recommended by IACS. The problem is that, if we apply directly these operating conditions within the direct calculation approach (based either on numerical simulations or on model tests) the extreme design parameters (bending moments, shear forces, torsional moments, accelerations…), which are obtained at the end of this procedure, usually significantly exceed the prescribed IACS values at a given probability level. We should mention however that this fact is not necessarily very critical in practice because we count with the fact that the assumed operating conditions are too severe and there are other safety factors which are introduced when considering the ship structural resistance. The proof that the existing procedures are not “too bad” is the excellent safety record of the existing ship fleet. However, we should be very careful when applying the same procedures for novel designs which exceed the limits of the classical ship designs. One of such ships is the Ultra Large Container Ship concept.
which significantly exceeds the initial assumptions about the ship size and structural flexibility. In particular, and as mentioned in the introduction, the structural natural frequencies of those ships are very low which, combined with relatively high speed, can give rise to the important hydroelastic contributions to the structural response both from extreme and fatigue points of view. Knowing all the difficulties which we have to properly account for the quasi static part of the response already, we can easily imagine how difficult will be to take consistently into account the hydroelastic effects. Additional important difficulty is related to the numerical modeling and that not only from the accuracy point of view but also because of the sometimes very extreme CPU time requirements. Indeed, even if we accept the above discussed numerical models to be good enough, the associated computational effort is huge and it is not practically possible to perform very long time numerical simulations for arbitrary operating conditions. The same is true for the model tests.

For the time being, due to all the mentioned difficulties related to both the modeling and operational issues, the usual direct (or better to say “quasi direct”) calculation procedures passes through the definition of the equivalent design waves, equivalent design wave episodes, equivalent design sea states and in some cases (whipping) through the long term direct calculations combined with some additional assumptions. A state of the art overall methodology for direct calculation approach is presented in [2] and interesting discussions on hydroelastic contribution are given in [4].

The real question we should ask ourselves is: how much are we far from the reality, both in our initial assumptions and in our models? Unfortunately, the actual state of the art does not allow for giving fully clear answer to this question. Significant work on the subject is undertaken in many research projects worldwide and one of those projects is the European project TULCS which we briefly present below.

4. TULCS Project

In 2009 a dedicated European project was initiated with the intention to bring more light onto the above discussed issues. The project name is TULCS (Tools for Ultra Large Container Ships) and the Consortium is composed of 13 partners: Bureau Veritas, Marin, Compagnie Maritime d’Affrètement – Compagnie Générale Maritime, Canal de Experiencias Hidrodinamicas, Ecole Centrale Marseille, Technical University Delft, University of Zagreb, Technical University of Denmark, University of East Anglia, SIREHNA, WIKKI, HYDROCEAN, Brze Vise Bolje. In addition, the world biggest shipyard Hyundai Heavy Industry is also participating to the project as the associated partner.

The project combines the numerical methods, model tests and the full scale measurements within a dedicated methodology for analysis and exploitation of the results. On the numerical side the accent is put on the potential flow based numerical solvers even if the CFD methods are included in order to evaluate their practical applicability to these problems. Each critical part of the overall technical problem is considered within the dedicated Work Package (WP). The final project decomposition is as follows:

WP2 - Identification of the real operating conditions
WP3 - Global quasi static wave loading & responses
WP4 - Global hydro elastic loading & responses
WP5 - Local hydrodynamic loading and responses - slamming
WP6 - Model tests
WP7 - Full scale measurements
WP8 - Analysis and the development of the rational design methodology

The example ship is the 9200 TEU ULCS Rigoletto owned by CMA-CGM and built by HHI. It was very important to have the detailed design of the same ship for all the developments (numerical, experimental and full scale) which allowed for consistent analysis and exploitation of the different results. Within the WP2 special attention was given to the identification of the real operating conditions (CMA-CGM) and of the design requirements (HHI). A dedicated questionary
was sent to more than 40 ship masters and useful information about the ship operational profile were collected (CMA-CGM). In WP3 different developments of the numerical seakeeping methods for the case with forward speed (the problem still open) were undertaken (MARIN, BV and TUD). Practical tools are developed and coupled with the 3D FE structural models. In WP4 an efficient procedure for hydroelastic coupling with simplified non-uniform beam model and full 3D FE model was developed and coupled with the seakeeping tools (BV & UNIZAG). Different, potential flow based, slamming models (2D and 3D) were developed in WP5 (UEA, MARIN, ECM) and, at the same time, the CFD calculations were performed by SPH and VOF numerical methods (Hydrocean & Wikki). The methodologies for their integration into the overall hydroelastic seakeeping tool in time domain were developed and their implementation is ongoing. In WP6 two types of model tests were performed: ECM did the simplified hydroelastic model tests on hyper elastic body while Cehipar performed large number of experiments on elastic and rigid model of Rigoletto for various operating conditions. Extensive full scale instrumentation (motion inertial unit, accelerometers, local and global strain gauges, wave radar system for sea state measurements...) was installed onboard Rigoletto ship and useful data are collected (MARIN & Sirehna). The analysis and comparisons of the different sorts of results are ongoing in WP8 and the overall methodology for design is discussed (BV, DTU, UNIZAG, BVB). Let us also mention that, within the TULCS project, an International Workshop on Springing & Whipping [10] was organized and was open to the Institutions outside of the project. The workshop was very well attended, by all the actors from the shipbuilding process (Class societies, shipyards, ship operators, Research Institutes, Universities...), and useful information were collected and used for better identification of the project developments and project objectives. The second Workshop is in preparation this year.

5. CONCLUSIONS

The main purpose of the paper was to discuss the different hydro-structural issues in ship design in the context of the so called direct calculation procedures. Special accent was put on springing and whipping hydroelastic effects.

It is fair to say that the modeling of springing and whipping is still a challenge and that there is no fully satisfactory numerical tool able to deal with these issues fully consistently. At the same time the other available tools such as model tests and full scale measurements have their own drawbacks (high cost, limited number of the covered cases, representatives of the beam model...) so that no definite opinion can be made on this subject for the moment. Anyhow, it seems to be clear that more attention should be given to these issues in the near future because there is clear evidence (numerical, model tests & full scale) that these hydroelastic types of structural responses exist and that their effects can be quite important both for fatigue and extreme response issues. In addition, the analysis of some recent accidents indicates that the whipping and springing are likely to be the, at least partial, reason for the structural failure.

It is important to note that, even without the inclusion of the hydroelastic effects, the classical quasi static hydro-structure interactions are also not perfectly mastered today and their inclusion into the design procedures still appears to not be fully satisfactory. This is true both for the imperfections of the deterministic hydro-structure calculation models and (even more) for the overall methodology for their inclusion (representative sea states, operational conditions, probability levels...).

Finally, let us also mention one very important point which might become very important in the near future and which concerns the compatibility of the rule based approach and the direct calculation approach. Indeed, it would be reasonable to expect that the two approaches should give the same answer (yes or no) in terms of the acceptance of the particular ship structural design. However, due to the quite different background of the two approaches, it seems to be very hard to ensure the compatibility of the two approaches in the general case.
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REFERENCES

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