HOMER - INTEGRATED HYDRO-STRUCTURE INTERACTIONS TOOL FOR NAVAL AND OFF-SHORE APPLICATIONS

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SUMMARY

The paper discusses the different hydro-structural issues in the design of ships and off-shore structures. The direct calculation tools and methodologies are presented within the BV software HOMER. This tool covers all the aspects of hydro structure interactions which are of concern in the design of any type of the floating bodies. Within the structural design of the floating bodies there are two main issues which should be considered, namely extreme structural response (yielding & buckling) and the response in fatigue. Even if the corresponding failure modes are fundamentally different, the overall methodology for their evaluation has many common points. Both issues require two main steps: deterministic calculations of hydro-structure interactions for given wave conditions and statistical post-processing in order to take into account the floating body lifetime operational profile. HOMER software covers both issues using the state of the art numerical methods and procedures.

1. INTRODUCTION

It is enough to take a look on Figure 1. to understand how difficult the numerical modelling should be for general seakeeping problem. Indeed, lot of different physical phenomena are involved (waves, ship speed, large ship oscillations, slamming, sprays, wind ...) and it is impossible to take all them into account at once.

![Figure 1: Ship sailing in waves.](image)

Even without considering the ship’s structural responses, the numerical modelling of the floating body hydrodynamic behaviour remains an open problem for a general case. This is true both for, the most commonly used, potential flow models and for the general CFD codes based on solving directly the Navier Stokes equations. The main problems of the modelling concern the correct representation of the waves generated by the interaction of the body with the sea waves, and the presence of the free surface which is not only unknown in advance but, at the same time, supports a highly non-linear boundary condition. The impossibility to solve the complete non-linear seakeeping problem at once, led to the different levels of simplification of the original non-linear boundary value problem.

On the structural side the situation is slightly less complicated because, most often, the linear structural problem is considered only. There exist nowadays very efficient numerical tools based on finite element method which allows for solving any type of linear structural problem either in quasi static or dynamic sense. This means that, once the correct hydrodynamic loading is transferred to the FE model, the evaluation of the structural response is rather straightforward.

The practical procedure for ship structural design involves the verification of two main structural failure modes:
- Yielding and buckling in extreme conditions
- Fatigue initiated cracks

These two failure modes are fundamentally different and the methodologies for their assessment are also different even if many 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 analyse the whole ship life by counting all the combinations of the stress ranges and number of cycles 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 built 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 structural strength side other safety factors 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, or off-shore type structures, 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 integrity, the basic idea is very simple: the structural response of the floating body should be calculated during whole its life using the 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. These models are discussed in this paper in the context of the Bureau Veritas HOMER software.

2. HYDRO-STRUCTURAL ISSUES

Before entering into the details of the different numerical models, let us first 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.

![Hydro-Structural Issues Table]

Table 1 Different hydro-structural issues (H – hydrodynamics, S – structure)

As far as the hydrodynamic loading is concerned, the usual practice is to classify it into 3 different categories:
- linear wave hydrodynamic loading
- weakly nonlinear non-impulsive wave loading
- local 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 combined with the large ship motions. The impulsive loading includes any type of the transitory local loading such as slamming, green-water, underwater explosion…

On the structural side the structural response can be classified into two main types:
- quasi static
- dynamic (often also called hydroelastic)

There is sometime 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

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.

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 the 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 relatively independently. On the hydrodynamic side, 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 usually formulated in frequency domain and the total hydrodynamic pressure is decomposed into the incident, diffracted and 6 radiated components:

\[ p^{\text{td}} = p_i + p_D - i\omega \sum_{j=1}^{6} \xi_j p_{R_j} \]  

(1)

The pressure 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 2) over which the singularity distribution is assumed in a certain form (constant or higher order).

![Typical Hydrodynamic Mesh in BIE Method]

Figure 2: Typical hydrodynamic mesh in BIE method.

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 \( S^0 \). There
are different types of the singularity distributions and the simplest one is based on the so called pure source distribution:

$$\varphi(x) = \int_{S_0} \sigma(\xi) G(x; \xi) dS$$

(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:

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

(3)

where:

- $[M]$ genuine mass matrix of the body
- $[A(\omega)]$ hydrodynamic added mass matrix
- $[B(\omega)]$ hydrodynamic damping matrix
- $[C]$ hydrostatic restoring matrix
- $[F^{DI}(\omega)]$ hydrodynamic excitation vector
- $\{\xi\}$ body motions vector

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 the 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.

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. This means that an efficient procedure for pressure transfer is necessary 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 floating bodies.

Most of the methods, which are used in practice, employ the different interpolation schemes in order to transfer the total hydrodynamic pressure 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 load case. This is because the FEM model has its own pressure integration procedure which is might be very different from the one used in the hydrodynamic model.

In order to obtain the perfect equilibrium of the structural model two main ideas are introduced in HOMER:

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

The first point is possible thanks to the very important 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 for choosing the characteristic points of the wetted structural finite elements and for performing 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.

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 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:

$$p^{hs1} = -\rho g [\xi_3 + \xi_4 (Y - Y_C) + \xi_5 (X - X_C)]$$

(4)
where \((X_c, Y_c)\) 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:

\[
p^{hs2} = -hgZ\Omega \times n
\]  

(5)

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:

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

(6)

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 [9]. Once all the loads calculated, the final loading case of the FE model is composed of 3 parts:

\[-\omega^2m_\xi \xi_i, \quad -p^{hs1} + p^{hs2}, \quad -m_\xi g\Omega \times k\]

- Rigid body inertial loading
- Pressure loading
- Additional gravity term

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 (3) in which all the different coefficients were calculated using the information from the structural FE model directly.

2.1 (a) Boundary conditions

Let us spend some time on discussing the issue of the additional boundary conditions which are necessary in the FE analysis of the free-free structures such as floating bodies. Indeed, very often, it appears that there is some misunderstanding regarding the choice of these conditions. First of all, these additional boundary conditions are completely artificial and are necessary because of the drawback of the numerical FE models. Quasi-static analysis by the finite element method assumes that the model may not move as a rigid body (strain free). If this condition exists in a conventional finite element analysis, the stiffness matrix for the model becomes singular. Consequently, we cannot perform conventional finite element quasi static analysis on unconstrained structures. The practical procedure consists in adding the artificial supports, to the FE model, in order to constrain the 6 rigid body motions. The problem arose when the model is not fully balanced because these unbalanced loading will induce the non-zero reaction forces at the artificial supports. Depending on the choice of the support these reaction forces will be more or less important. For some reasons, the usual practice is to put these supports at the nodes far from the center of gravity as indicated in Figure 3 (top). It is quite clear that the reaction forces at the artificial supports will depend on the source of the unbalanced loading and we can never be sure that this choice of artificial supports will be the best one for the general case. In the case of the method presented here, the choice of the artificial supports become arbitrary because the loading is balanced before being applied to the FE model! In order to show how arbitrary the choice of the additional boundary conditions could be, in Figure 3, we show the stress distribution in the FE model obtained with two fundamentally different types of boundary conditions.

Figure 3: Different choice of the artificial supports. Top – 3 linear force supports (arrows indicate the constrained motion), Bottom – one clamped node at the foremost position of the bulb structure (double arrows indicate that both translations and rotations are constrained).

The first set of boundary conditions (Figure 3 - top) is the classical one, i.e. three additional supports each blocking the two translational motions, and the second set of boundary conditions (Figure 3 – bottom) is simply one
single node (foremost node at the bulb) which was clamped i.e. both 3 translation and 3 rotations are constrained. The results in terms of stresses are numerically identical. The apparent difference in the absolute displacements is due to the fact that the displacements are expressed relative to the boundary condition. This means that the difference in between two sets of displacements can be expressed in terms of 6 rigid body motions. Since the rigid body motions are strain free they do not contribute to the final stress/strain distribution. In this context, let us also mention the so called inertia relief method, which is often used in the commercial FE packages, and which allows for getting the reaction forces zero even if the loading is not balanced. The basic idea is to correct the rigid body accelerations in order to get the zero reaction forces. This is of course always possible. However this method is obviously wrong since it does not correct the balancing error at its source, but just spread the error over the whole ship through the additional inertia which normally should not exist. In some sense this method might be even more dangerous than the method of artificial supports because it does not give any idea about the amplitude of the unbalanced loads.

2.2 (b) Top down analysis

It is important to mention the fact that, especially for fatigue assessment, 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 in 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 later. 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 [17] and one example is shown in Figure 4.

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 6). Once the RAO’s calculated a spectral analysis is used in order to calculate the characteristic stresses (mean, significant, maximum …) for particular sea state.

2.2 LINEAR HYDROELASTIC HYDRO-STRUCTURE INTERACTIONS

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

$h^i(x)$ 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 generalized radiation potentials with the following body boundary condition:

$$\frac{\partial \Phi_{BI}}{\partial n} = h^i n$$

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.

![Figure 4: Top down principles.](image1)

![Figure 5: Transfer of the modal displacements from structural (red) to hydrodynamic mesh (green).](image2)
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 [11]. Typical result of this interpolation is shown in Figure 5.

After solving the different boundary value problems for the potentials, the corresponding forces are calculated and the motion equation similar to (3) is solved and the modal amplitudes are obtained. The total stresses can now be calculated, at least theoretically, by summing the individual modal contributions and we can formally write:

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

where \(\Sigma(x, \omega)\) is the total stress, \(\sigma^i(x)\) is the spatial distribution of the 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 poor 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. However, we are interested in the total stress and we need to properly combine both types of responses. 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 [11] and one typical example of the final stress RAO decomposition is shown in Figure 6.

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 [17].

2.3 NON LINEAR 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 very 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:

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

where:

- \([m]\) modal mass matrix
- \([A^\omega]\) infinite frequency added mass matrix
- \([k]\) structural stiffness matrix
- \([C]\) hydrostatic restoring matrix
- \([b]\) structural damping matrix
- \([K(t - \tau)]\) impulse response functions matrix
- \([F(t)]\) linear excitation force vector
- \([Q(t)]\) nonlinear excitation force vector
- \([\dot{\xi}(t)]\) modal deformations vector
- \([\ddot{\xi}(t)]\) modal velocities vector
- \([\dddot{\xi}(t)]\) modal accelerations vector

It was shown in [3] that the impulse response functions can be calculated from the linear frequency dependent damping coefficients or added masses. Once the impulse response function matrix \([K(t)]\) calculated, the motion equation (10) is integrated in time and the nonlinear forces are added at each time step to the vector \([Q(t)]\).

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 structural 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 (a) Weakly nonlinear Froude Krylov loading

The simplest weakly nonlinear wave loading concerns the so called Froude Krylov approximation 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

\[\Sigma_{hs}(x, z) = \rho g z \langle \frac{\partial u}{\partial z} \rangle \]

Typical springing response and its decomposition into quasi static and dynamic parts.

Figure 6: Typical springing response and its decomposition into quasi static and dynamic parts.
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 7).

Additional simplification which can be added to the Froude Krylov approximation consist in the choice of the wave elevation which can be chosen in such a way that it includes or not the diffracted and radiated parts. Indeed, the simple choice of the incident wave elevation significantly simplifies the implementation procedure. However this approximation is reasonably justified only for specific operating conditions where the diffraction-radiation effects are not very important (slender ship in head waves …) and it should be used carefully for general cases.

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. One example of pressure transfer for weakly nonlinear model is shown in Figure 8.

Figure 7: Froude Krylov approximation.

Figure 8: Pressure transfer for weakly non linear model.

2.3 (b) Slamming and whipping
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 10, 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 [10] 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 [19]. GWM allows for the evaluation of the impact pressure along the arbitrary ship section and for arbitrary penetration velocity. 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 (10) 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 9.

Figure 9: Subdivision of the bow part of the ship for slamming calculations.
The GWM provides the impact pressure along the ship section and this pressure is integrated over the overall FE model using the following relation:

\[ F^i = \sum_{j=1}^{N_s} \int_{L_j} p_j \mathbf{n}_j \mathbf{b}_j \, dl \]  

(11)

where \( F^i \) is the slamming load projected on the \( i \)-th mode, \( p_j \) is the slamming pressure on the section \( j \) and \( \mathbf{n}_j, \mathbf{b}_j \) are the mode shape \( j \), 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 significant increase of the overall CPU time for whipping calculations. One example of the typical whipping numerical simulations obtained using the present method is shown in Figure 10.

\[ \text{Figure 10: Typical numerical simulations of whipping.} \]

\[ \text{Figure 11: Initial partial FE model, modified FE model (middle) and hydrodynamic integration mesh (bottom).} \]

2.4 SOME PARTICULAR HYDRO-STRUCTURE PROBLEMS

In the previous sections we discussed the basics of the hydro-structure interactions in seakeeping. Without these basic models it is not possible to assess the structural integrity in a rational way. However, many practical problems in floating body design require additional improvements of the above described models. Here below we list some of them.

2.4 (a) Partial structural models

In ship design practice, and for some particular applications, the classification societies allow for the use of the so called partial structural models. The reason for that is very simple and is related to the fact that it is much easier to build the partial structural FE model (especially in the midship part) than the full FE model of the structure. The main problem, when using these models, is related to the influence of the boundary conditions. Indeed since only the part of the ship structure is modelled, it is not possible to have the influence of the rest of the structure in fully consistent way. That is why usually the structural response is analysed in the central part of the model only, where the influence of the boundary conditions is believed to be less important. Anyhow, in spite of the structural simplifications, the hydrodynamic loading should be fully balanced and should take into account the whole ship dynamics. The method which was chosen in HOMER consists in building the hybrid structural and hydrodynamic models by combining the existing partial FE model and the full hydrodynamic model. One typical example of this kind of calculations is shown in Figure 11.

In this way, the procedure for pressure transfer remains the same as that for full structural model. The loading of the parts which are not structurally modelled is applied at the corresponding centres of gravity and transferred to the FE model using the interpolation rigid beam elements. In this way the full balance of the hybrid FE model is ensured in the same way as for the full ship model.

2.4 (b) Mixed panel stick hydrodynamic model

Some floating structures, in particular for off-shore applications, are composed of many slender elements in addition to the main hull part. These slender elements are too small compared to the wave length and cannot be modelled using the diffraction-radiation approach based on the boundary integral equations method. In practice the Morison formula is used to take into account the loading on these parts of the structure. Special dedicated developments are necessary in order to take into account consistently the combined effects of the diffraction-radiation and Morison type of loadings. In practice the diffraction-radiation problem on hull part is solved first and the resulting velocity and acceleration are used to calculate the Morison forces. If we formulate the problem in frequency domain, due to the nonlinear
character of the Morison equation, the motion of the body has to be solved by iterations. Once the body motions evaluated the load transfer on the hull part is done according to the principles described before and the Morison loading is applied as concentrated forces on slender elements. In this way, the FE model is again perfectly balanced and reaction forces at the artificial supports are numerically zero. One example of calculations for semi-submersible platform are shown in Figures 12 and 13.

In the case of the floating body carrying liquid (LNG ships, tankers…) the influence of the liquid dynamics on body motions can be significant. Sloshing itself is extremely complex nonlinear hydrodynamic problem but its influence on body motions can reasonably be considered as a linear. That is why the same linear potential flow model can be used for sloshing too. That is exactly how this coupled problem is solved in HOMER [12]. In practice, the hydrodynamic problems for ship hull and internal liquid motion are solved separately and the coupled motion equation is built in a post-processing phase. Due to the same numerical method used for seakeeping and for sloshing same method for pressure transfer is used and the full balance of the FE model is guaranteed again. One example of the pressure distribution inside the tank is shown in Figure 14.

Figure 14: Pressure distribution in ship tanks.

3. HOMER
The numerical software HOMER was developed in Bureau Veritas Research Department in order to cover all the above hydro-structural issues. HOMER represents the essential link in the hydro-structural chain ensuring the perfect hydrodynamic loading of the structural model. HOMER was made in such a way that it can be coupled with any structural or hydrodynamic software. This is possible thanks to the coupling philosophy which is based on clear separation of two parts of the problem:

- The hydrodynamic solver is used to solve the hydrodynamic boundary value problem only!
- The structural solver is used to solve the given loading case only!

All the rest is done by HOMER! This means, in particular, that the hydrodynamic coefficients (added masses, damping, excitation…) are calculated by HOMER after integrating the pressure over the finite element mesh, and the body motion equation is solved within HOMER.

For the time being HOMER is used with Hydrostar as the hydrodynamic solver and Nastran & Ansys as the structural solvers. The general coupling scheme of HOMER is shown in Figure 15. In addition to the hydrodynamic and structural solvers, HOMER is composed of 3 main modules:

- HMFREQ frequency domain solver
- HMTIME time domain solver
- HMSTPP post-processing tool
In addition, HOMER is open for coupling with different modules for evaluation of the local hydrodynamic effects such as slamming (module HMLSM in Figure 15).

Figure 15: HOMER computational scheme.

It is possible today to use HOMER for solving any of the hydro-structural issues arising in practice and that both for naval and off-shore applications. One of the latest examples for which HOMER was used is the cylindrical FPSO shown Figure 16. Both extreme and fatigue calculations were performed on this floater. This application was very particular in many aspects, one of them being the definition of the equivalent design waves for extreme structural response analysis. Indeed, due to the particular circular design of the structure, it was not possible to define the dominant loading parameters (such as bending moments, shear forces…) in advance. It was thus decided to perform the long term analysis of the structural response instead of the long term analysis of the loading parameters. For that purpose, several thousand stresses on the finite elements located at different parts of the structure were post processed in order to calculate the long term stress design values from which the design waves were deduced. This was possible thanks to the computational performances of HOMER regarding both the CPU time and the post-processing capabilities.

Figure 16: Recent use of HOMER for cylindrical FPSO.

Many other useful features, not discussed here in details, make HOMER unique in this field. Let us just mention few of them:
- Consistent inclusion of roll damping effects
- Intermittent wetting calculations
- Pure Morison loading on floating slender structures (wind mill support vessels…)
- Consistent inclusion of mooring loads
- Lifting operations
- …

4. DESIGN METHODOLOGIES

Let us spend some time on discussing one very essential aspect of hydro structure interactions i.e. how all these hydro-structure models should be used in practice. Indeed, the deterministic hydro-structure interaction models, which were discussed in this paper, represent one part of the direct calculation approach only. Another part is the overall methodology which has to be built around these models in order to properly represent the entire floating body life and be able to predict the structural failure modes (extreme and fatigue). In order for the design methodology to be consistent, the most important point is, probably, the choice of the operational profile which needs to be chosen in a reasonable way as possible. The situation is slightly less complicated for off-shore type of floaters but for the ship it is very difficult to define it consistently. 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 mean 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, sometimes 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 rely on the fact that the assumed operating conditions are too severe and there exist 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, 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 modelling 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 both to the modelling and to the 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 [5].

5. CONCLUSIONS

The purpose of the paper was to discuss the different hydro-structural issues in floating body design in the context of the so called direct calculation procedures. The numerical models are very similar for sailing ships and for stationary offshore vessels even if every floating body has its own particularities. The type of hydro-structure calculations critically depends on the floating body operational profile and not the same type of calculations has to be run for every floating body.

The theory presented here was implemented in the Bureau Veritas software HOMER. HOMER should not be looked at as neither hydrodynamic nor the structural software but rather as a loading and methodology software. It represents the essential link in between relatively independent hydrodynamic and structural calculations and it can be used both as the design tool and as the design verification tool.

Finally, let us 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 for the general cases. This point seems to be one of the most important challenges in the future. The role of the software such as HOMER is essential to raise this challenge.

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