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Some aspects of hydro-structure interfacing in seakeeping

Malenica Š.[(1)], *Stumpf E.*[(1)], *Delafosse V.*[(1)], *Chen X.B.*[(1)] *& Senjanović I.*[(2)]
[(1)]BUREAU VERITAS -DR, Paris La Defense, France
[(2)]Faculty of Mechanical Engineering and Naval Architecture, Zagreb, Croatia

ABSTRACT

An efficient interface between the hydrodynamic and structural model, is a critical element in hydro-structure analysis of floating bodies. Main difficulties are related to the fact that two types of meshes (hydrodynamic and structural) are not the same due to the different criteria in mesh generation. In this paper we discuss the methods which are able to treat efficiently all aspects of the hydro-structure interactions in the linear seakeeping. Both the classical rigid body case and the hydroelastic case are considered. The hydrodynamic model is made under the potential flow assumptions and Boundary Integral Equation (BIE) method based on the so-called source formulation, is used to solve the corresponding Boundary Value Problems (BVP). Only the frequency domain approach is discussed.

INTRODUCTION

In order to illustrate the main differences between the hydrodynamic and structural meshes, in Figures 1 and 2, we present the typical structural and hydrodynamic meshes for the container vessel. As we can see the meshes are significantly different. The hydrodynamic mesh contains only the panels below the mean waterline and is finer close to the waterline, while the structural mesh closely follows the different structural elements without "taking care" of hydrodynamics.

As far as the linear, frequency domain, seakeeping analysis is concerned there are two main issues in the hydro structure interactions:

- Pressure transfer from the rigid body hydrodynamic model to the structural model.
- Transfer of structural deformations from the structural model to the hydrodynamic model.

The first point is rather obvious, but the second one is slightly more unusual and concerns the hydroelastic interactions which will be discussed in more details later in the text. Briefly speaking, they are necessary in order to associate the hydrodynamic coefficients (excitation, added mass, ..) to the deformable modes of body motions.

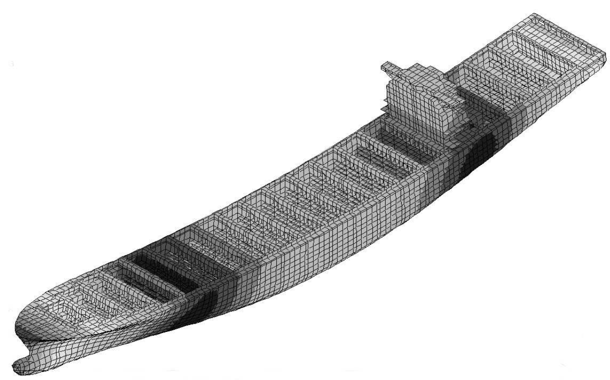

Figure 1: *Typical structural mesh.*

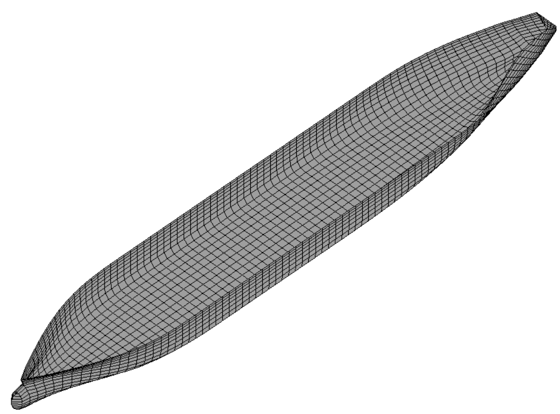

Figure 2: *Typical hydrodynamic mesh.*

RIGID BODY SEAKEEPING ANALYSIS

In the classical frequency domain linear rigid body seakeeping analysis, the problem is formulated under the potential flow assumptions and the total velocity potential is divided into the incident, diffracted and 6 radiated components:

$$\varphi = \varphi_I + \varphi_D - i\omega \sum_{j=1}^{N} \xi_j \varphi_{Rj} \tag{1}$$

where :

φ_I	-	incident potential
φ_D	-	diffraction potential
φ_{Rj}	-	radiation potential
ξ_j	-	rigid body motions

At the same time, the corresponding dynamic pressure is found from the linear Bernoulli equation, and the similar decomposition is adopted:

$$p = i\omega\varrho\varphi = p_I + p_D - i\omega \sum_{j=1}^{N} \xi_j p_{Rj} \tag{2}$$

In order to obtain the the total hydrodynamic pressure, the dynamic variation of the hydrostatic pressure should also be added to the above expression:

$$p^{hs} = -\varrho g[\xi_3 + \xi_4(Y - Y_G) - \xi_5(X - X_G)] \tag{3}$$

where the subscript "$_G$" denotes the position of the center of gravity, with respect to which the motion equation is written.

It is important to note that the motion equation is written in the so called earth fixed reference system, or in the system paralel to it if the body is animated with forward speed. For that reason the restoring matrix is not directly obtained by integration of the hydrostatic pressure (3) but also the change of the normal vector should be taken into account.

$$\boldsymbol{F}^{hs} = [\ \mathbf{C}\ ]\{\boldsymbol{\xi}\} = \iint_{S_b} [p^{hs}\boldsymbol{n} - \varrho g Z \boldsymbol{\Omega} \wedge \boldsymbol{n} dS] \tag{4}$$

where $\boldsymbol{\Omega}$ denotes the rotational component of the motion vector $\boldsymbol{\Omega} = (\xi_4, \xi_5, \xi_6)$. As it will be discussed later, this fact is very important in the context of the pressure transfer to the structural model.

After integrating the pressure over the wetted body surface, the corresponding forces are obtained and the rigid body motion equation, in frequency domain, is usually written in the following form:

$$\Big(-\omega^2([\ \mathbf{M}\ ] + [\ \mathbf{A}\ ]) - i\omega[\ \mathbf{B}\ ] + [\ \mathbf{C}\ ]\Big)\{\boldsymbol{\xi}\} = \{\boldsymbol{F}^{DI}\} \tag{5}$$

where:

$[\ \mathbf{M}\ ]$	-	genuine mass matrix
$[\ \mathbf{A}\ ]$	-	added mass matrix
$[\ \mathbf{B}\ ]$	-	damping matrix
$[\ \mathbf{C}\ ]$	-	hydrostatic restoring matrix
$\{\boldsymbol{F}^{DI}\}$	-	excitation force vector

For the sake of completness, let us also recall the final expressions for the excitation, added mass and damping:

$$F_i^{DI} = i\omega\varrho \iint_{S_b} (\varphi_I + \varphi_D) n_i dS \tag{6}$$

$$\omega^2 A_{ij} + i\omega B_{ij} = \varrho\omega^2 \iint_{S_b} \varphi_{Rj} n_i dS \tag{7}$$

As mentioned before, the Boundary Integral Equation method based on the source formulation is used to solve the Boundary Value Problems for different potentials. The general form of the BVP is:

$$\left.\begin{array}{ll} \Delta\varphi = 0 & \text{in the fluid} \\ -\nu\varphi + \dfrac{\partial\varphi}{\partial z} = 0 & z = 0 \\ \dfrac{\partial\varphi}{\partial n} = V_n & \text{on } S_b \\ \lim\big[\sqrt{\nu R}(\dfrac{\partial\varphi}{\partial R} - i\nu\varphi)\big] = 0 & R \to \infty \end{array}\right\} \tag{8}$$

where V_n denotes the body boundary condition which depends on the considered potential:

$$\frac{\partial\varphi_D}{\partial n} = -\frac{\partial\varphi_I}{\partial n} \quad , \quad \frac{\partial\varphi_{Rj}}{\partial n} = n_j \tag{9}$$

Within the source formulation of the BIE, the potental at any point in the fluid is expressed in the following form:

$$\varphi = \int\int_{S_b} \sigma G dS \tag{10}$$

where G stands for the Green function and the unknown source sterngth σ is found after solving the following integral equation:

$$\frac{1}{2}\sigma + \int\int_{S_b} \sigma\frac{\partial G}{\partial n} dS = V_n \quad , \quad on \quad S_b \tag{11}$$

This equation is solved numerically, after discretizing the wetted part of the body into a number of flat panels over which the constant source distribution is assumed.

Loading of the structural model

Due to the differences between the hydrodynamic and structural meshes, an efficient interfacing procedure is needed in order to properly transfer the hydrodynamic pressure onto the structural model. Most often, diferent interpolation scheme are used but they appear to be neither robust nor efficient for general 3D cases. What we propose here, is not to use the interpolation of the pressure but its **recalculation**. Indeed this becomes possible thanks to the particular characteristics of the source formulation which gives the continuous representation of the potential through the whole fluid domain $Z < 0$. In this way we avoid any interpolation problem and the pressure is smoothly redistributed over the structural mesh. At the same time, the communication between the hydrodynamic and structural codes is extremely simplified because it is enough for the structural code to give the coordinates of the points where the pressure is required and the hydrodynammic code just calculate the pressure using:

$$\varphi(\boldsymbol{x}_s) = \int\int_{S_b} \sigma(\boldsymbol{x}_h) G(\boldsymbol{x}_h; \boldsymbol{x}_s) dS \tag{12}$$

where $\boldsymbol{x}_s = (x_s, y_s, z_s)$ denotes the structural point and $\boldsymbol{x}_h = (x_h, y_h, z_h)$ the hydrodynamic point (panel center).

It is important to note that this method of pressure transfer, would not be possible in the case of the BIE method based on the so called mixed singularity distribution (sources + dipoles) which is discontinuous across the wetted body surface. For that reason, the mixed singularity distribution would lead to the dangerous peaks in the pressure distribution for the structural points which penetrates, even slightly, the hydrodynamic mesh.

It should be noted that the hydrodynamic pressure represents only one part of the total loading and the inertia loads, due to the body mass acceleration, should be added (substracted) to each finite element. However, the calculation of this part of loads is trivial.

The last but not least point which should be mentioned concerns the so called hydrostatic restoring which was briefly discussed above. Indeed, the structural response is calculated in the body fixed coordinate system while the hydrodynamic response is calculated in the earth fixed system. As we have seen, when solving the hydrodynamic problem, this is accounted for by the change of the normal vector (4). It can be shown (e.g. see Malenica(2003)) that, in the structural model, this is equivalent to the change of the gravity action for each finite element. In other words, the following loading should be added:

$$\boldsymbol{f}^g = -mg\boldsymbol{\Omega} \wedge \boldsymbol{k} \tag{13}$$

This terms acts as an additional inertia load and should be treated as such.

Let us also note that the structural calculations, that we are talking about here, still represents the so called quasi-static calculations even if the inertia terms are included. Indeed, the body motions remains the rigid motions, and the structural loading contains only the hydrodynamic pressure and rigid body inertia. In practice, the procedure in frequency domain, consists in solving two independent quasi static calculations separately for real and imaginary parts of the loading. In this way the final structural response is presented in the form of the so called RAO-s.

Finally, the floating body being a freely moving, the finite element structural codes usually require the additional boundary conditions in order to have a stable system and be able to carry out the calculation of the structural responses. This is usually done by adding the 3 supports at fore and aft parts of the body. The type of supports are chosen in order to have an isostatic system i.e. only 6 displacements are blocked.

Numerical example

In order to illustrate the efficiency of the proposed approach we chose the example of a rectangular barge with the dimensions: Length $L = 100m$, Breadth $B = 20m$, Depth $D = 10m$, Draught $T = 5m$.
Barge is exposed to the regular incident waves with different headings. Two types of the hydrodynamic meshes are considered: the first one is identical to the underwater structural part, and the second one is completely different. We consider head, beam and oblique waves respectively, and that for two cases mentioned above i.e. identical and different meshes of the hydrodynamic and structural models. The ABAQUS structural model with 3 additional supports is shown in Figure 3 and two hydrodynamic models in Figure 4. First we compare the cumulative results

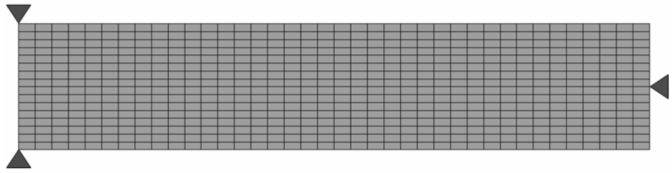

Figure 3: *Structural model of the barge and additional supports.*

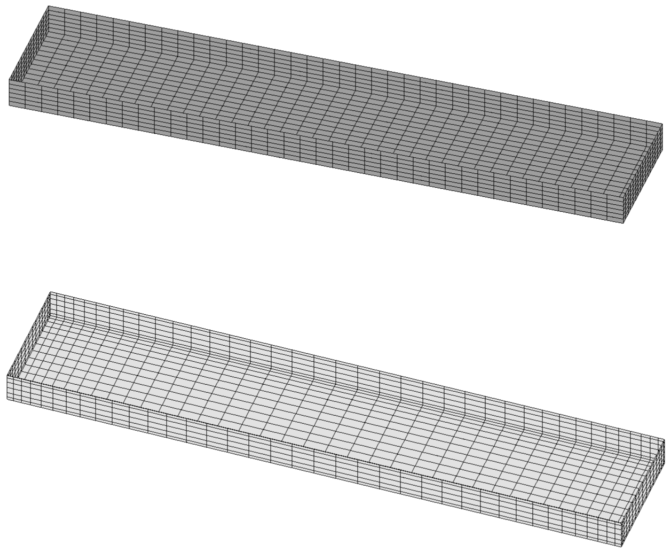

Figure 4: *Hydrodynamic models of the barge. Top - hydrodynamic mesh identical to the structural mesh, bottom - hydrodynamic mesh different from the structural mesh.*

obtained by HYDROSTAR (integration over the hydrodynamic mesh) and the sum of reaction forces after the ABAQUS run, for exclusive pressure loading i.e. without inertial loading. In Table 1 we can see that HYDROSTAR and ABAQUS give almost the same results, which proves the correctness of the above described method used for pressure transfer .

In additon to these results, in Figure 5, we present the internal sectional loads along the barge. The internal loads are defined as the difference between the pressure loads and the inertia loads at considered section. The results are presented for the wave frequencies $\omega = 0.4rad/s$ and $\omega = 0.8rad/s$, and the amplitude of the loads RAO's is presented. We can see again that the agreement between the results obtained by HYDROSTAR and ABAQUS is very good. However, we can still observe the small differences, which might be important depending on the frequency and quality of the meshes. The main reason for that is the fact that the body motions are calculated using the hydrodynamic coefficients obtained after integration over the hydrodynamic mesh, while the structural response is calculated by applying the total pressure (2), on the structural mesh. In order to minimize the differences, the structural codes usually performs some "tricks".

	Identical meshes		Different meshes	
Loads	A	H	A	H
$F_x[kN](\omega = 0.4)$	214.55	214.5	213.56	213.6
$F_z[kN](\omega = 0.4)$	0.19	0.18	1.48	1.48
$F_x[kN](\omega = 0.8)$	834.97	835.0	834.26	829.3
$F_z[kN](\omega = 0.8)$	572.85	572.9	573.27	573.3

	Identical meshes		Different meshes	
Loads	A	H	A	H
$F_y[kN](\omega = 0.4)$	661.18	661.2	661.64	661.6
$F_z[kN](\omega = 0.4)$	0.2	0.2	0.48	0.48
$F_y[kN](\omega = 0.8)$	1035.4	1035.6	1035.48	1036.0
$F_z[kN](\omega = 0.8)$	1800.7	1801.0	1800.7	1828.0

Table 1: *Total pressure loads on the barge, in head (top) and beam (waves) for different frequencies (A stands for ABAQUS and H for HYDROSTAR).*

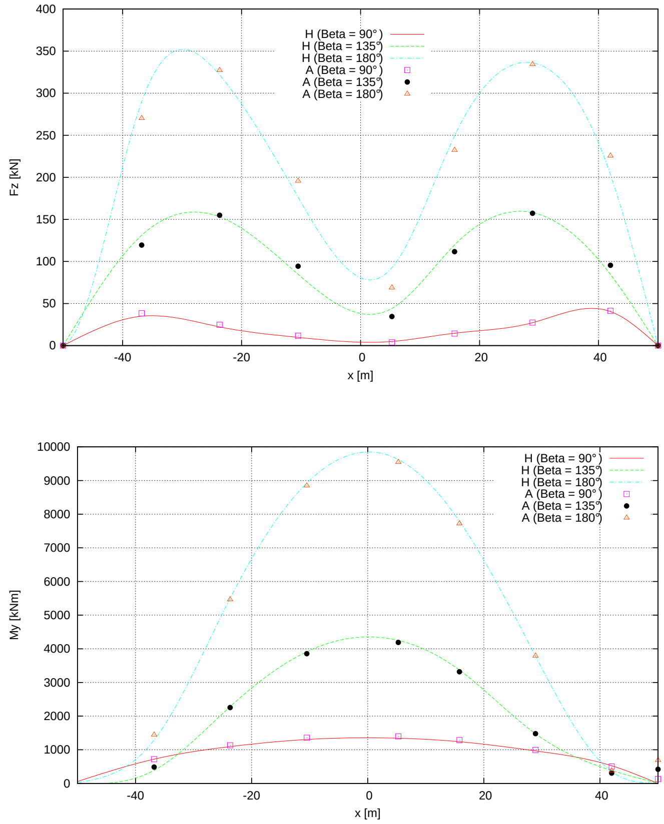

Figure 5: *Internal loads distribution.*

However these tricks remains rather arbitrary (change of the motions, redistribution of the mass, ...) and differs from code to code. In our opinion, the only way to obtain the "perfect" equilibrium, would be the separate transfer of the different pressure components. This implys that the motion equation should be solved only after runing the structural code and the hydrodynamic coefficients should be obtained by the integration over the structural mesh. This might appear slightly unusual for the structural engineer but seems to be the only way toward the perfect coupling. It is clear that the ship motions will be slightly modified as compared to the motions obtained using the hydrodynamic code, but these differences will be of higher order than the approximations implicitely adopted in the linear theory.

HYDROELASTIC SEAKEEPING ANALYSIS

The hydroelastic analysis of the floating body becomes necessary in the cases where the elastic structural natural periods become so high that can be excited by the waves contained in the usual sea spectra (T $\approx$ 5 to 25 seconds). This is the case of the so called springing phenomena whic appears on very large ships. However, this argument holds only if the springing is considered to be a purely linear phenomena which is not so obvious because the nonlinear aspects of the hydrodynamic loads (second and higher order) can also excite the flexible natural modes due to their high frequency content. Finally, it should be noted that the hydroelastic model is also necessary for the so called whipping problem which is caused by the impulsive loadings such as slamming. Anyway, regardless of the fact if the problem is linear or non-linear, the fully coupled hydroelastic problem needs to be solved, and here we present the procedure for construction of an efficient coupled model.

First we briefly recall some basics of the hydroelastic springing model in the context of the so-called modal approach. The main difference between the rigid body and elastic body seakeeping analysis lies in the representation of the body motions. In the case of rigid body the motions are represented by 3 translational and 3 rotational motions, while in the elastic case and in addition to the rigid body motions the dry elastic modes are also included.

We write for general body motion:

$$\begin{aligned} \boldsymbol{H}(x,y,z) &= \sum_{i=1}^{N} \xi_i \boldsymbol{h}^i(x,y,z) \qquad (14) \\ &= \sum_{i=1}^{N} \xi_i [h_x^i(x,y,z)\boldsymbol{i} + h_y^i(x,y,z)\boldsymbol{j} + h_z^i(x,y,z)\boldsymbol{k}] \end{aligned}$$

where $\boldsymbol{h}^i(x,y,z)$ denotes the general modal vector function.

This representation genaralize the rigid body model, since the rigid body motions can be representad by the following modal functions:

$$\begin{array}{lcl} \boldsymbol{h}_1 = \boldsymbol{i} & , & \boldsymbol{h}_4 = \boldsymbol{i} \wedge (\boldsymbol{R} - \boldsymbol{R}_G) \\ \boldsymbol{h}_2 = \boldsymbol{j} & , & \boldsymbol{h}_5 = \boldsymbol{j} \wedge (\boldsymbol{R} - \boldsymbol{R}_G) \\ \boldsymbol{h}_3 = \boldsymbol{k} & , & \boldsymbol{h}_6 = \boldsymbol{k} \wedge (\boldsymbol{R} - \boldsymbol{R}_G) \end{array} \qquad (15)$$

The overall procedure remains similar, so that the final motion equation is the same (5), except the number of unknown is increased for number of flexible modes.

The difficulty in coupling of the hydrodynamic and structural models is related to the transfer of the modal displacements from structural to hydrodynamic model. Indeed, the body boundary condition for the radiation potential being:

$$\frac{\partial \varphi_{Rj}}{\partial n} = \boldsymbol{h}^j \boldsymbol{n} \qquad (16)$$

we need to transfer the modal displacement vector $\boldsymbol{h}^j$ from the structural onto the hydrodynamic mesh.

It is important to note, that in the common case where the beam model is used to represent the structural response, this task do not represents important difficulties because the structural deflection is represented by 1D function. For example, within the beam model approximation, the modal function for vertical bending is written in the form:

$$\boldsymbol{h}^i = -\frac{\partial w}{\partial x}(Z - Z_N)\boldsymbol{i} + 0\boldsymbol{j} + w\boldsymbol{k} \tag{17}$$

where Z is the vertical coordinate of the point on the body surface and Z_N is the vertical position of the neutral axis.

However, the simplified beam model seems to not be representative for structural response of the ships with complicated structure (container ship, large passanger ships, catamarans, ...) and the use of the general 3D FEM models can not be avoided. In that case the interpolation procedure between the structural and hydrodynamic meshes needs to be performed. Here we present the method able to do that in the most general cases. The general algorithme is briefly described below.

For each hydrodynamic point (panel center) the following steps are performed:

1. Looking for the 3 closest structural points. In the case that one of the 3 points is within small enough distance (ϵ) from the considered hydrodynamic point, we retain that structural point for interpolation.
2. The list of the structural finite elements containing at least one of the above defined points is created.
3. The hydro point is projected onto the surfaces created by the retained structural elements. If the projection falls inside the element, the corresponding distance is calculated. The element with smallest distance from the hydro point is retained for interpolation.
4. In the case that the hydro point do not project on any element from the list, the projection on the sides of the considered elements is performed. If the projection falls onto the side, the corresponding distance is calculated. The side with the smallest distance from the hydro point is retained for interpolation.
5. In the case that there is neither point at ϵ distance, nor element nor the element side on which point project, the structural point closest to the hydro point is retained for interpolation.
6. The interpolation using the shape functions of the retained finite element, of the structural displacements is performed on the projection of the hydro point and the calculated displacements are associated to the hydro point.

This procedure was verified on several ship types and showed to be extremely efficient. In Figure 6 we present one axample for a rather complex ship type. As we can see the ship deformations are correctly transfered from the structural to hydrodynamic mesh.

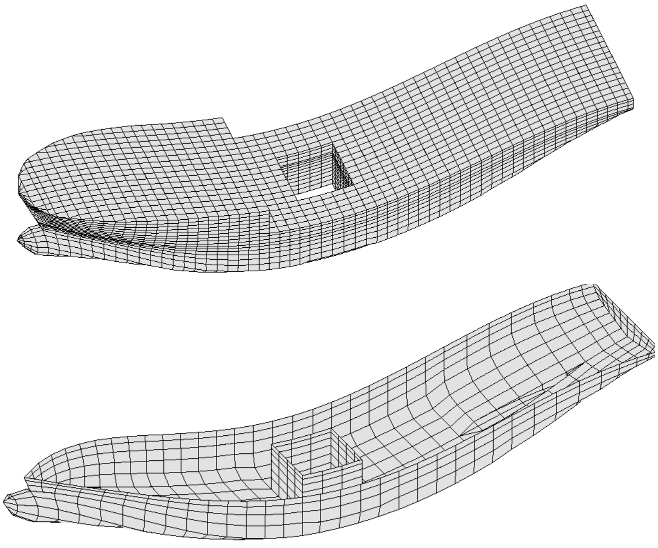

Figure 6: *First bending mode on the structural mesh (above) and its projection on hydrodynamic mesh (below).*

CONCLUSIONS AND FUTURE WORK

We briefly discussed here two important aspects of the hydro-structure interactions in the context of the linear seakeeping of floating bodies. The main problems are related to the communication between the hydrodynamic and structural models i.e. the interpolation of the different quantities from one mesh to another. There is a sometimes tendency to make the hydro and structure meshes identical in which case the hydro structure coupling can be slightly simplified. However, we do not think that making the identical meshes for the structural and hydrodynamic model, is a good solution because it is too restrictive from the users point of view. In the present paper, very efficient solutions are proposed in the context of the general hydrodynamic 3D panel codes (e.g. HYDROSTAR) and the general structural 3D FEM codes (e.g. ABAQUS).

The further work will consist in considering the following important points:

- Improvement of the equlibrium by considering separate transfer of pressure for diffraction and radiation parts, and consequent calculation of the hydrodynamic coefficients by integration over the structural mesh.
- Improvement of the method for the floating body animated with forward speed. Indeed, in this case the proposed method might fail in the cases where the structural points fall inside the hydrodynamic model. This is due to the fact that the calculation of the pressure in the case with forward speed involves not only velocity potential but also its gradient (fluid velocity). In the simpliest case the pressure may be written in the form:

$$p = i\omega_e \varrho \varphi + \varrho U \frac{\partial \varphi}{\partial x} \tag{18}$$

The problem lies in the discontinuity of the gradient of the velocity potential across the body boundary in the source based BIE method. The solution consists in putting the structural points from the inside of the body boundary onto it. This may be achieved using the similar procedure which was used for transfer of modal displacements in the case of hydroelastic analysis.

- Consideration of the nonlinear cases. In this case the time domain analysis is required and the pressure needs to be integrated up to the exact free surface and not up to $z = 0$ (see Fig. 7) It is easy to imagine that the situation is much

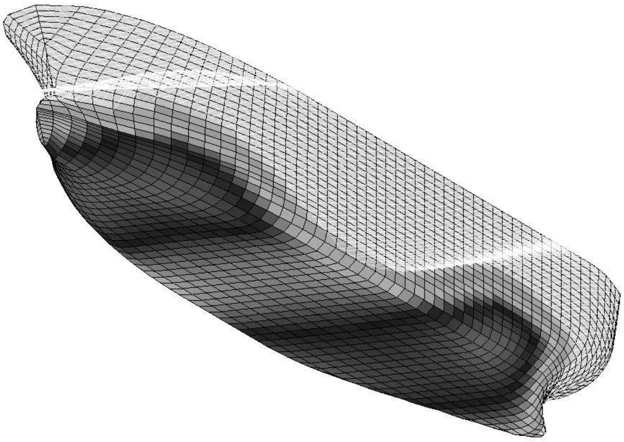

Figure 7: *Nonlinear pressure distribution.*

more complicated. However, the combination of the ideas presented here may be used to produce an efficient model.

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