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SLAMMING IMPACT SIMULATION OF 2D WATER ENTRY FOR RIGID STRUCTURES

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Abstract. *Slamming impact is considered important factor for ship design engineers to predict the structural reliability under maximum pressure and hydrodynamic forces in a small duration of time. In this paper, we deal with rigid bodies by simulated 2D water entry problem based on Arbitrary Lagrangian Eulerian (ALE) built-in explicit finite element in Abaqus software. To validate the efficiency of models, various deadrise angles are applied from 4° to 80° for wedge. The prediction of maximum pressure and hydrodynamic forces are compared with the analytical formulations of the rigid body. Many factors are affecting on slamming phenomenon due to high deformation in the fluid. Therefore we have delineated the effect of the contact stiffness factor between the fluid and structure in penalty contact method, which influences on the results. Fluid domain mesh sizes convergence are applied with take in consideration the hourglass control effect, showed that the effect of the density of the mesh has high influence on the amplitudes stability for both pressure and force.*

1 INTRODUCTION

Slamming is the water impact loads that considered important in the structural design of all high-speed vessels and can cause both local and global effects in the ship structure, the global effect is often called whipping. Hydrodynamic may be important for global loads but also for local loads in the case of very high slamming pressure with very short duration [1].

The loads occur during the slamming of the body onto a water surface have relevance for the design of many vessels, unfortunately, the analysis of this scenario is challenging for several reasons. During impact, the free surface is characterized by a thin jet with velocities that are much larger that the body velocity. This means that free surface undergoes stretching and its topology becomes extremely complex [2].

The major interesting of the ship engineers is to find the analytical solution capable to provide the hydrodynamic load and the impact pressure distribution when the slamming event happens, since these pressures can be used during the design process [3]. The slamming pressure distribution and hydrodynamic forces can be used in the static structure analysis that locates the local impact induced stresses. This done when the local deadrise impact angle between the fluid and the structure is not very small at the impact position, and suppose that hydrodynamic calculations for a rigid body [1]. Several assumptions are adapted to calculate the pressure for the rigid body such as the airflow. Usually, these methods assume that the fluid is incompressible and irrotational. Von Karman was considered the first author attempted to calculate analytically the slamming phenomenon based on the mass conservation, but is not accounted the effect of the jet up-rise flow [4]. The method based on Wagner theory allowed taking in account the effect of the jet up-rise flow of the fluid, which assume the impact of the blunt body [5].

Kaushik et al [6] studied the water slamming of deformable sandwich panels using the commercial FE software LS-DYNA with the ALE formulation. All geometric nonlinearities are considered when studying deformations of the composite panel, assumed the fluid to be compressible. However, the authors have accounted inertia effects in the fluid and the solid, and examined delamination between the core and the face sheets. The pressure distribution on the wetted panel surface was found to be oscillatory. Peseux et al [7] have solved numerically two and three dimensional Wagner theory using finite element method for the rigid and deformable structures. A series of tests were performed experimentally for different deadrise angles and thickness, and analysis the distribution and evolution of hydrodynamic pressure. Stenius et al. [8] used numerical simulations of the hydroelastic problem related to panel-water impacts for high-speed craft, by applying different deadrise angles and velocities and different boundary conditions and panel properties. Siyauan & Mahfuz [9] used the finite element analysis to study the fluid structure interaction (FSI) for the sandwich structure. This performs by coupling between the finite element analysis (FEA) model with the computational fluid dynamic (CFD) model. The global model was used to construct the composite and the foam of the sandwich structure, then transporting the force and displacement for this model to the sub-model with refine mesh. The interlaminar and the shear stress distribution are determined at the core and faces. Yang and Qiu [10] has been computed slamming forces on two and three dimensional bodies entering the calm water with constant velocity, employing the Constrained Interpolation Profile (CIP) method. The nonlinear water entry problem is governed by the Navier-Stokes equations and solved by a CIP-based finite difference method on a fixed Cartesian grid. Wang and Soares [11] have investigated the water impact problem for three-dimensional buoys (hemisphere and cones) using ALE solver based on Eulerian formulation in LS-DYNA software. They concluded that the mesh density of the fluid is very influence factor on determine the hydrodynamic pressures. Furthermore, they noted that the influence of the constant impact velocity is larger than ones from the drop velocity when compared its total impact force and hydrodynamic pressure.

In this paper, a parametric study is carried out to provide the hydrodynamic load and the impact pressure distribution when the slamming event is happened with constant velocity. Indeed, the response of the structure will depend on the parameters of the numerical calculation inputs for describing the fluid-structure coupling.

2 MATHEMATICAL FORMULATION FOR ALE MODEL

ALE model consists of both the Lagrangian and Eulerian model, materials arbitrary coordinates are defined corresponding to the reference or global coordinate system. The relationship between these coordinates and the reference coordinate relative with the time are constructed the ALE model as described by equation (1) [12]

$$\frac{df(X_i,t)}{dt} = \frac{df(x_i,t)}{dt} + w_i \frac{df(x_i,t)}{dxi} \tag{1}$$

Where X_i, xi and $w_i = (v - u)$ are the Lagrangian coordinate, Eulerian coordinate and relative velocity respectively. From definition of relative velocity as a difference between the material velocity (v) and the mesh velocity (u) , formulations of the model are depended on the conservation equations as followed:

2.1 Conservation of the mass

For point of view of the conservation of the mass which means that the material density is constant relative with time and can be described as:

$$\frac{d\rho}{dt} = \rho \frac{dv_i}{dx_i} - w_i \frac{d\rho}{dx_i} \tag{2}$$

Where ρ is the density of the fluid.

2.2 Conservation of momentum

In conservation of the energy, the governing equation must be specified the appropriated initial boundary conditions to represent the ALE description corresponding to Navier-stock equations:

$$\rho \frac{dv_i}{dt} = \sigma_{ij,j} + \rho b_i - \rho w_i \frac{dv_i}{dx_j} \tag{3}$$

The stress tensor σ_{ij} is described as follows:

$$\sigma_{ij} = -p\delta_{ij} + \mu(v_{i,j} + v_{j,i}) \tag{4}$$

Where μ, is the dynamic viscosity and δ_{ij} is the Kronecker delta function and p is the pressure.

Define the initial boundary conditions for the fluid and the body domain as shown in figure (1), the equations can be solved.

$v_i = U_i^0 \;\; on \; \Gamma 1$ $\qquad$ $\sigma_{ij} n_j = 0 \;\; on \;\; \Gamma 2$

$\Gamma 1 \cap \Gamma 2 \; = \; 0$ $\qquad$ $\Gamma 1 \bigcup \Gamma 2 \; = \; 0$

$V_i(x_i, 0) \; = \; 0$ $\qquad$ on $t = 0$

Where

$\Gamma 1$ is the boundary of the body,

$\Gamma 2$ is the boundary of the fluid,

nj is the outer unit normal vector on the boundary of $\Gamma 2$,

δ_{ij} is the Kronecker's delta function.

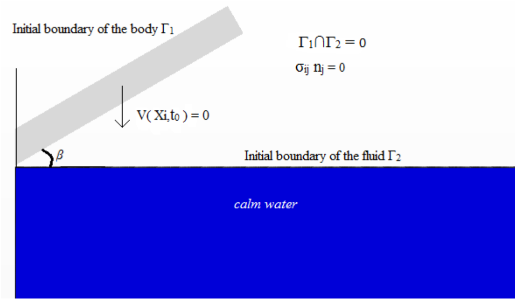


Figure 1: Initial boundary conditions for fluid and body domain

2.3 Energy equation

$$\rho \frac{dE}{dt} = \sigma_{ij} v_{i,j} + \rho b_i v_i - \rho w_j \frac{dE}{dx_j} \tag{5}$$

The velocity of reference coordinate is assumed zero in Eulerian equation, while the relative velocity are computed in advection phase to transfer from previous mesh to the new material mesh. For this reason, ALE equations are considered much more difficult numerically than the Lagrangian equations, where the relative velocity is zero.

ALE governing equations are implemented in two phases: the first is applied to the Lagrangian phase that changes of the velocity in the mesh moves with the material. Consequently an internal energy due to the internal and external forces is calculated. The equilibrium equations are:

$$\rho \frac{dv_i}{dt} = \sigma_{ij,j} + \rho b_i \tag{6}$$

$$\rho \frac{dE}{dt} = \sigma_{ij} v_{i,j} + \rho b_i v_i \tag{7}$$

Mass is conserved in the Lagrangian phase, since no material flows across the element boundaries. The advection phase is followed to determine a transport of mass, internal energy and momentum across cell boundaries. This may be thought of as remapping the displaced mesh at the Lagrangian phase back to its initial or arbitrary position.

3 NUMERICAL INVESTIGATION OF THE SLAMMING PHENOMENON

For the slamming event with small period duration, large deformations of the fluid are happening, which caused the mesh distortion. Therefore, an automatic re-mesh method requires for the fluid domain, which consumes a high computational time. For this reason the suitable simulation of the interaction between the fluid and the structure is implemented. Arbitrary Lagrangian Eulerian (ALE) solver in ABAQUS software are used to allow the mesh to move and back to reference position during translating to another step. The material in the ALE region flowed with the mesh move. Hence, to prevent the element mesh destroy, a remap of the mesh are adapted using an advection phase to update fluid velocity and material variables for the new mesh.

ALE is a mixture of Lagrangian and Eulerian discretization. The region in the fluid that close to the impact area has a refine meshes size to govern the large deformation of the fluid in this position. While the outer region of this domain used another mesh size because the deformation in the outer domain of impact position is still moderate. Figure 2 illustrates the boundary conditions and the mesh part model for the water entry problem, which are tested to reduce the computational time as reason of high time consuming of coupling between the fluid and structure.

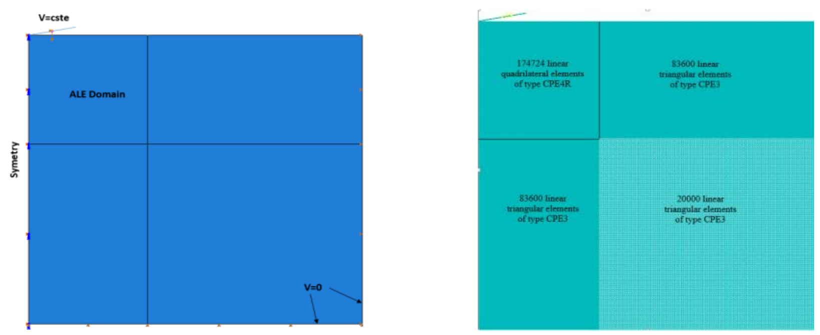


Figure 2: Boundary conditions and mesh regions for the slamming of two-dimensional wedge model.

For the slamming of the symmetric wedge, symmetrical boundary condition is applied about the symmetric plane with breadth of section 0.25 m and length 0.4 m. To prevent reflected waves effect from external boundaries, fluid domain length is specified as [13]:

$$L \geq \frac{c_0 T}{2} \tag{8}$$

Where L is the distance between the impact position and the external surface of the water domain, T is the time of the slamming event and c0 is the speed of the sound in the fluid. In Abaqus explicit, EOS can be defined by linear Us - Up formulation of the Mie-Gruneisen equation of state which can be exploited in water entry problem. Pressure can be determined as a function of the density and the internal energy as:

$$P = f(\rho, E_m) \tag{9}$$

$$P = PH\left(1 - \frac{\Gamma_0\ \eta}{2}\right) + \Gamma_0\ \rho_0 E_m \tag{10}$$

Where PH, Γ_0 , $\eta = 1 - \rho_0/\rho$ and ρ_0 is the Hugoniot pressure, material constant, nominal volumetric compressive strain and fluid reference density respectively.

$$PH = \frac{\rho_0 c_0 \eta}{(1 - s\eta)^2} \tag{11}$$

Where s is represented the linear relationship between shock velocity Us and particle velocity Up (Us= c0 + s Up).

Then the pressure of the fluid can be calculated as:

$$P = \frac{\rho_0 c_0 \eta}{(1 - s\eta)^2}\left(1 - \frac{\Gamma_0\ \eta}{2}\right) + \Gamma_0\ \rho_0 E_m \tag{12}$$

The water parameters are illustrated in table (1). For the modelling of the contact solid/fluid, the exponential soft contact with frictionless tangential behavior has been adopted. The mesh convergence is investigated in the region close to the impact position with element size 0.5mm for the fluid domain.

C_0 (m/sec)	μ (-)	s (-)	Γ_0 (-)	ρ(kg/m^3)
1420	0.001	0	0	1000

Table 1: Parameters of the water in the model.

3.1 Effect of the hourglass control

The mesh of Lagrangian domain deforms with the material, while the Eulerian mesh is fixed in the space, add the material flows from one cell to another. Lagrangian is easier to handle (particularly the definition of boundary conditions), but with large mesh deformation and highly distorted leading to inaccurate calculations, or even failure.

Hourglassing is essentially a spurious deformation mode of a Finite Element Mesh, resulting from the excitation of zero-energy degrees of freedom. It typically manifests as a patchwork of zig-zag or hourglass like element shapes (Figure 3), where individual elements are severely deformed, while the overall mesh section is non-deformed.

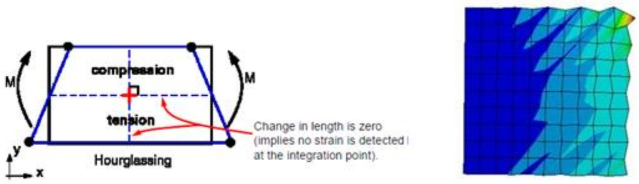


Figure 3: Hourglass behaviour for a single first-order reduced-integration element with no strain.

In this section the hourglass effect is investigated, built in artificial stiffness of the elements to prevent this kind of deformation. There are five element controls (combine, Enhanced, relax-stiffness, stiffness and viscous) that are added as artificial stiffness for the elements. By verify the total artificial energy that used to control hourglass is small (<1) relative to the total internal energy, as shown in table (2).

contact	Hourglass control	Artificial energy (ALLAE)	Internal energy (ALLIE)	Relative difference <1
Kinematic method	Combine	19,288	46,878	**0,588**
	Enhanced	0,006	14,288	0,999
	Relax-stiffness	0,006	14,356	0,999
	stiffness	0,007	14,277	0,999
	viscous	43,809	67,109	**0,347**
Penalty method (scale factor = 0.05)	Combine	37,138	56,518	**0,342**
	Enhanced	0,017	15,571	0,998
	Relax-stiffness	0,017	15,571	0,998
	stiffness	0,004	15,795	0,999
	viscous	44,559	59,940	**0,256**

Table 2: Comparison of the energy histories of hourglass element controls.

By comparison, values of the energy histories (artificial and internal energy) for two types of the contact methods with different hourglass controls are illustrated in table (2). Total hydrodynamic forces and the hydrodynamic pressure along the length of the body contact surface are also compared for their amplitude values and oscillation as shown in figure 4. From these results, we can conclude that the suitable contact coupling is penalty contact method with two best hourglass controls (combine and viscos). As a result, they have minimum difference of energy and minimum vibration for both hydrodynamic pressure and hydrodynamic force. The simulation results have a good agreement with analytical approaches.

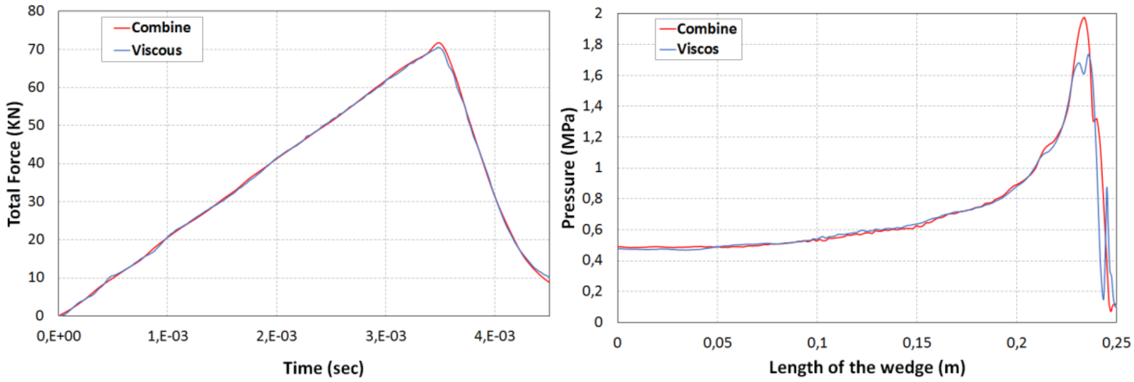


Figure 4: Total Force and Pressure distribution due to the contact pressure on the wedge surface with penalty contact method with deadrise angle 10°, impact velocity 8 m/s.

3.2 Effect of the penalty stiffness factor (PFAC)

The penalty stiffness factor (PFAC) is a factor for scaling the contact stiffness of the two pairs of the contact surfaces, which applies to nodes of the water-wedge interface to estimate the coupling force. The penalty coupling method is working as a spring system in each of both ends that attached to the structure and fluid nodes. The coupling force calculates relative with the penetration rate.

$$F = Kd \tag{13}$$

Where K and d are representing the spring system stiffness and penetration respectively.

The spring stiffness is depended on the scaling factor, bulk modulus of the fluid and the mesh size of the fluid as descripted in equation (14):

$$K = p_f \frac{k\ A}{V} \tag{14}$$

Where p_f , A and V are penalty scale factor, the average area of the structure element and the volume of the fluid element, which are in the coupling state.

By applying a range of penalty scale factor (1, 0.2, 0.1 and 0.05), a comparison study on slamming force and pressure distributions is performed. In this model, the default value 0.05 is applied as optimum value according to amplitude and profile of total force and pressure distribution as shown in figure 5. Notes this factor has small influence on results compared with mesh size density effect. When used a high value of scale factor, coupling stiffness has been increased between nodes of fluid-structure interface. Consequently, it causes a high oscillation in pressure and the force without penetration. In the other hand, low value of scale factor leads to decreasing stiffness. In this situation, a non-physical penetration appears and leakage will be done. The free water surface elevation and pressure histories for 10° deadrise angles is illustrated in figure 6.

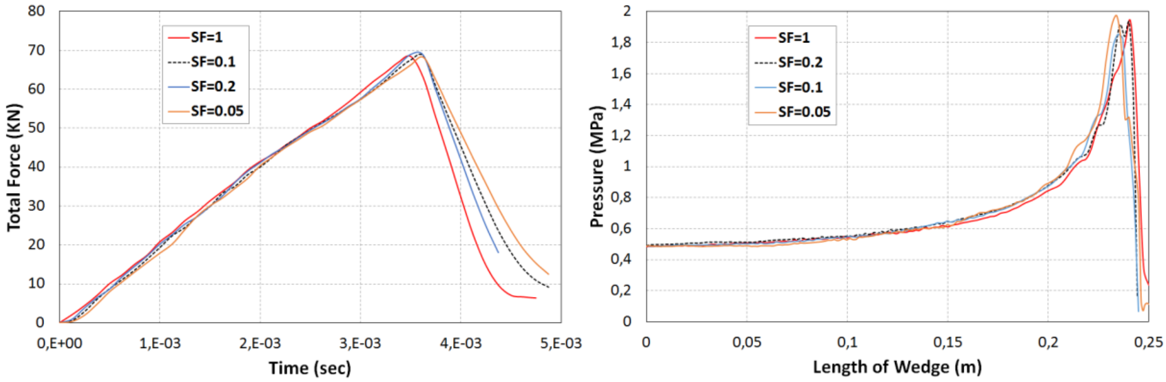


Figure 5: Total Force and Pressure on the wedge surface with different stiffness scale factors of penalty contact method with deadrise angle 10°, impact velocity 8 m/s.

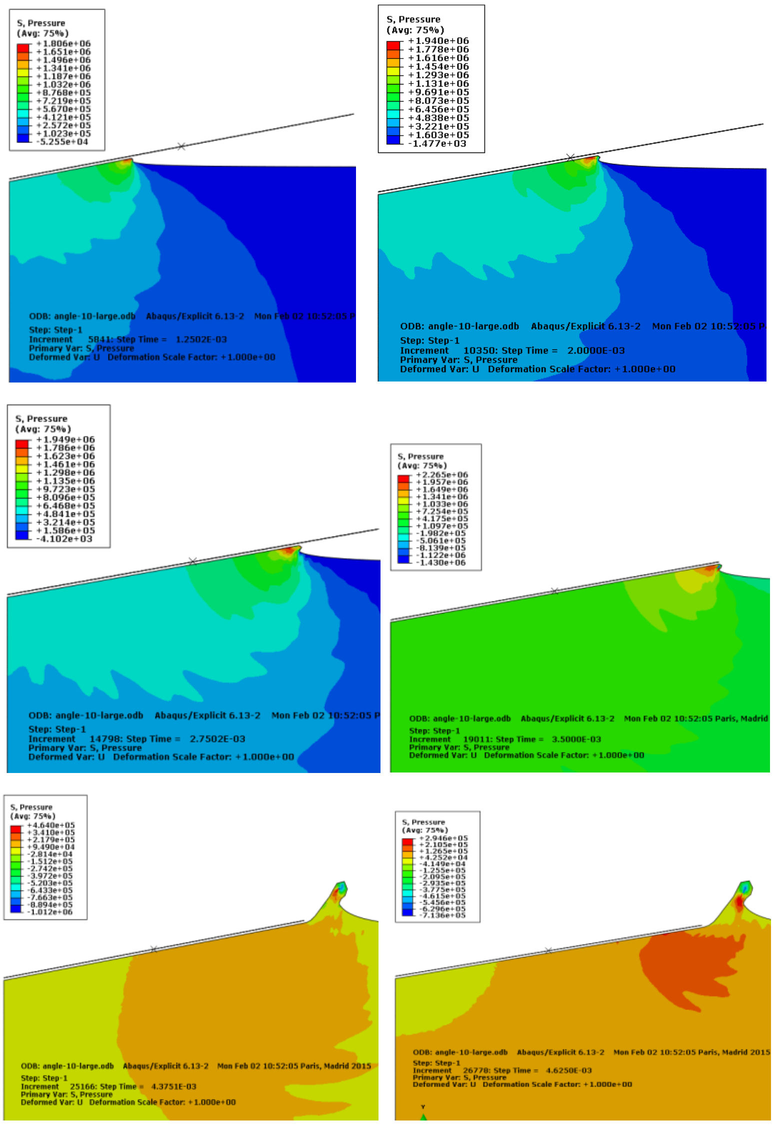


Figure 6: Free water surface and pressure histories for deadrise angle 10°, impact velocity 8 m/s.

3.3 Deadrise angles effect on the hydrodynamic pressure

Slamming phenomena happens in short time duration. For this situation, a high magnitude peak pressure occurs, which is considered very important for the ships design in the naval applications. This phenomenon can be cause global and local damages in the structure due the interaction between the structure and the fluid. The pressure is travelling along the panel width that largely dependent on the velocity impact and the angle between the structure and the fluid. Hydrodynamic effects are important when the deadrise angle between the body and the water surface is small [3].

The kinematic boundary of the fluid-structure is changing when the structure responds to the dynamic pressure, this kinematic coupling between the fluid and the structure is a type of hydroelastic effects. For the rigid body, the pressure decreases relatively with increasing of the deadrise angle under constant velocity. Indeed, that the wedge cuts more easily through the water, which is shown in figures 7 and 8. In these figures, a good agreement is observed with the analytical methods [14], obviously that the coefficient pressure is decreasing according to increase the deadrise angle. In equation (15), the non-dimensional pressure corresponding to the deadrise angles has been calculated as:

$$Cp_{max} = \frac{p_{max}}{0.5\rho\, v^2} \tag{15}$$

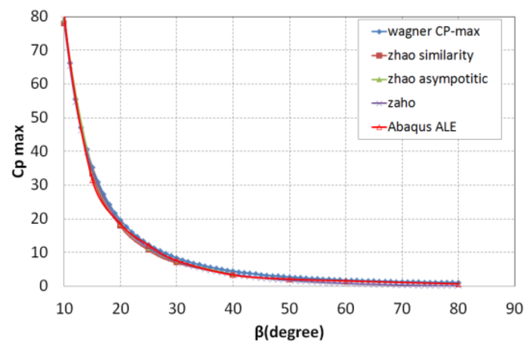
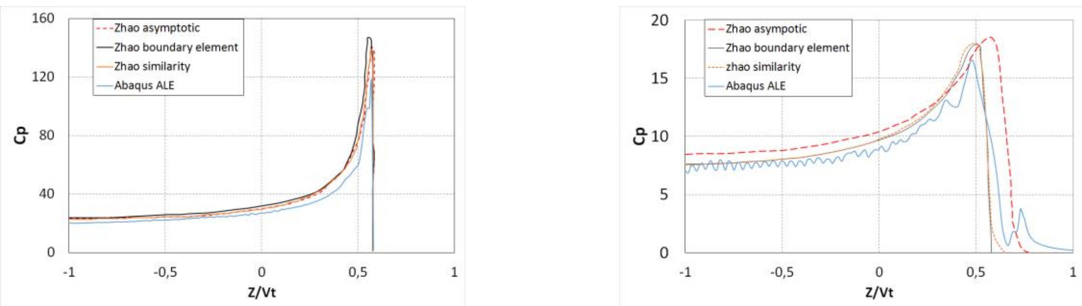


Figure 7: Non-dimensional coefficient pressure Cp_{max} according the deadrise angles



(a) 7.5°

(b) 20°

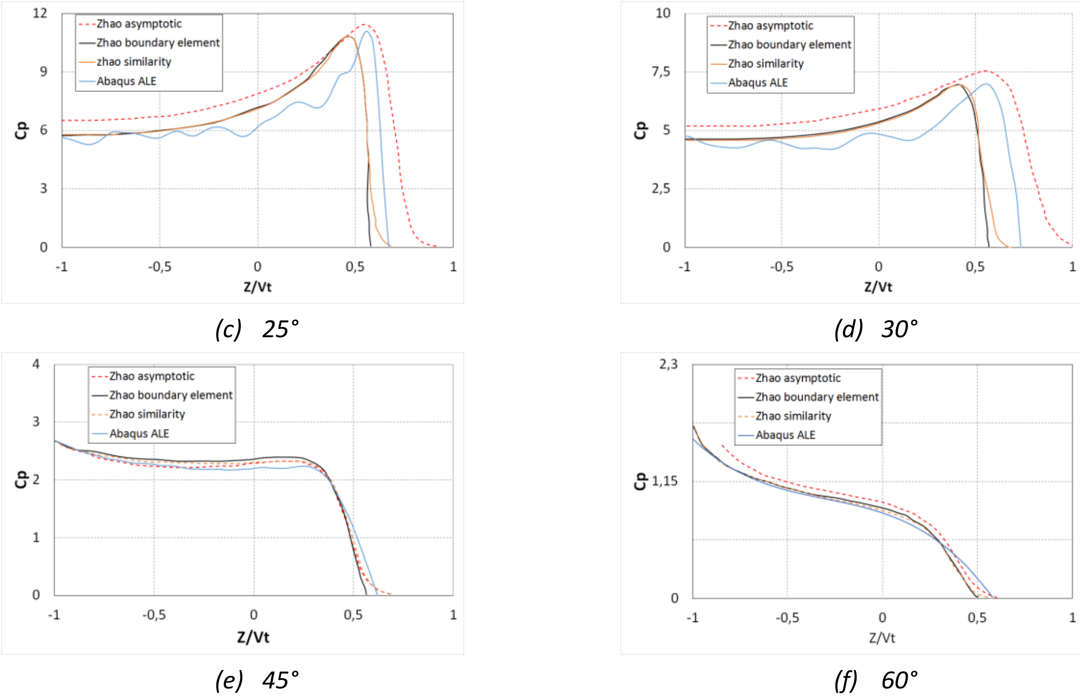


Figure 8: Coefficient pressure distribution for arrange of deadrise angles with constant velocity 13 m/s.

4 CONCLUSIONS

In this study we are simulated the water entry for 2D rigid structure by using ALE (Arbitrary Lagrangian-Eulerian) solver, which can be represented the solid deformation as fluid with water properties. ALE combines the properties of Lagrangian and pure Eulerian using the adaptive mesh technique, and can be remap the mesh with each step increment to prevent the distortions of the mesh due to high deformations. This technique is depending on the coupling contact, therefore, penalty contact method have been used. By defining the appropriate parameters for this method such as stiffness scaling factor and mesh density, it can be satisfied the stability and prevented the penetration between the rigid body and the water. When compared these simulations with previous methods, a good agreement can be observed between the measured impact pressures for deadrise angles large than 30° and the analytical methods. Noted that high pressure peak close to move to the keel of the wedge, and it was dropped with increase the deadrise angle. For the deadrise angles less than 10°, some difference are happen compared with the analytical methods, and pressure with sharply peak occurs close to the jet flow region. The pressure peak for a wedge body is stay unchanged approximately along the wedge-water interface with time histories and it is reducing after the flow separation happens.

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