IMPLEMENTATION OF A NOVEL SHIP SIDE STRUCTURE INTO A TANKER AND A ROPAX VESSEL FOR INCREASED CHRASHWORTHY

AUTHORS:

Sören Ehlers

Kristjan Tabri

Nicole Schillo

Janne Ranta

CORRESPONDENCE:

Helsinki University of Technology, Department of Mechanical Engineering, Ship laboratory, Tietotie 1C, 02015 TKK, Phone +358 9 451 3497, Fax +358 9 451 4173, soeren.ehlers@tkk.fi

ABSTRACT

The energy absorption of a novel ship side structure is numerically analysed. The structure is implemented into an ice going Tanker and a ROPAX vessel. The aim is to evaluate the benefits in crashworthiness of such an application over the existing conventional side structures. The collision simulations are performed with the explicit non-linear solver LS-DYNA up to the point of inner shell breaching. The obtained energy-penetration curves of the novel structure are then compared to energy-penetration curves of the conventional structure. The novel sandwich structure results in clear benefits in terms of increased energy absorption during collision; 30 per cent for the Tanker, and about 50 per cent for the ROPAX. The applied failure criterion and mesh size dependency are also discussed.

KEYWORDS:

Collision simulation, Tanker, ROPAX, novel crashworthy structures

INTRODUCTION

An average of 4.6 ship to ship collisions per month worldwide drive the development of novel ship side structures with a better crashworthiness than conventional structures. A crashworthy structure can absorb more energy until breaching of the inner hull, than a conventional structure. The tendency towards larger and less manoeuvrable ships is also noticeable, resulting in a higher risk of accidents [1]. Loss of life and severe environmental impacts due to oil leakage are possible collision consequences that need to be prevented, at least their severity. The amount of oil and oil products transported in the Baltic Sea has doubled since 1997 and is expected to increase to 160 million tonnes per year by 2010.

One approach to reduce the risks of ship collisions is to improve the crashworthiness of the ship side structures. TEKES (Finnish Funding Agency for Technology and Innovation) and several industrial partners were financing the national Törmäke-Project [ref. to webpage] that was carried out at the Ship laboratory of TKK (Helsinki University of Technology). One objective of the project was to develop a design procedure for new crashworthy ship side structures [2] for an Ice going Aframax Tanker and a ROPAX.

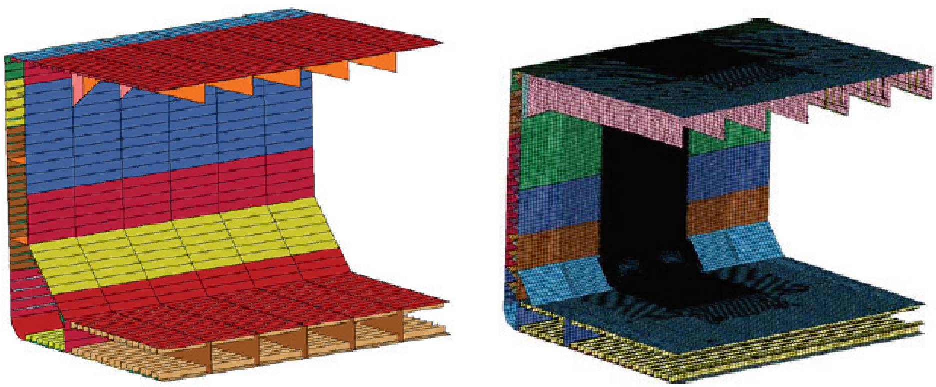
This paper presents the implementation of one novel ship side structure into a Tanker and ROPAX. The GL approval procedure for alternative structural arrangements is outlined and partly applied. Simplifications have been made to decrease the overall procedure time. As a result the obtained energy-penetration curves of the novel structure are compared to energy-penetration curves of the conventional structure. The applied failure criterion and mesh size dependency are also discussed.

THE TANKER AND THE ROPAX MODEL

The main particulars of the Tanker and ROPAX are presented in *Table* 1. The conventional models are created based on the main frame drawing of each ship, see Figure 1 and Figure 2. The novel ship side structure to be implemented was designed and optimized for its implementation into an ice going vessel [2]. To obtain maximum crashworthiness, the novel structure does not have any webframes (see Figure 4) allowing a wider spread of deformation and energy absorption. In both ship types the novel structure is only applied within the ship side structure. Novel structure was designed and implemented so that the change in the total mass of the ship was minimal. The Tanker and the ROPAX vessel with the novel structure are about 10% heavier than the conventional build ships.

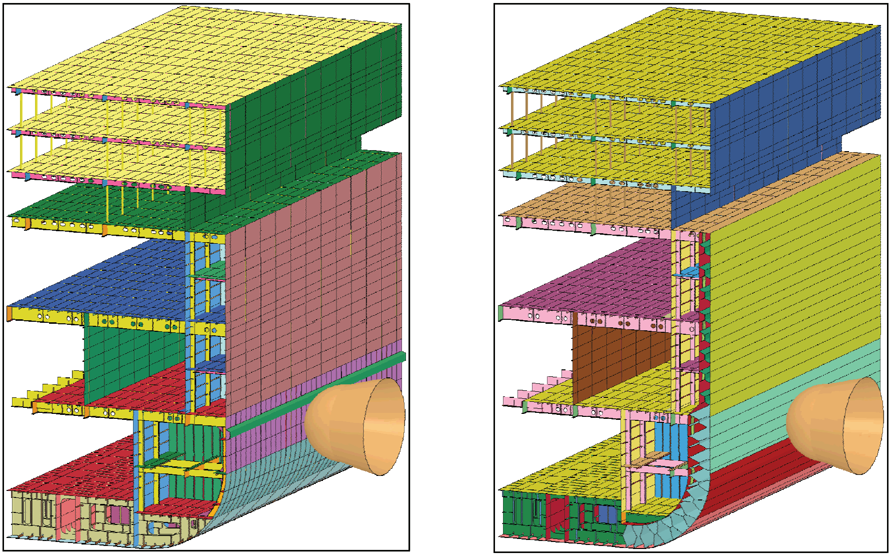
Table 1: Tanker and ROPAX main particulars

	Tanker	ROPAX
Length	233.4 m	188.3 m
Breath	44 m	28.7 m
Draught	15.3 m	6 m
Specialties:	- Ice strenghtened (1AS) - Double Acting	- Ice strenghtened (1AS)



a) conventional model b) novel structural model

Figure 1: Tanker models



a) conventional model b) novel structural model

Figure 2: ROPAX models

Table 2: FE model data

	Tanker model	ROPAX model
Length of the model:	27m	26.25m
New structure:	~750.000 Elements	~1.300.000
Conventional structure:	~1.200.000 Elements	~950.000

SIMULATIONS BASED ON THE GL-APPROVAL PROCEDURE FOR ALTERNATIVE ARRANGEMENTS

Germanischer Lloyd (GL) provides a guideline for the approval of new structures [3]. The approval procedure requires a comparison of the conventional structure with the novel structure. To be compared is the absorbed energy of the struck ship until the inner hull is penetrated. This is defined as the critical deformation energy which needs to be higher or equal for the novel structure than for the conventional. GL demands the evaluation of 16 different load cases: two shapes of the striking bow: one without bulb, one with bulb, four different vertical striking positions, striking at a webframe and striking between the webframes. The striking angle is set to 90°. The struck ship is at zero speed. The striking bow is assumed to be rigid. As the maximum absorbed energy of the ship side structure is of interest, the simulations can be performed quasi-statically.

The novel structure is only applied in the ship side and the surrounding structure remains unchanged. Therefore the simulations are only performed with a bulbous indenter without soft nose (Figure 3), as the crashing of the deck structure is not of interest.

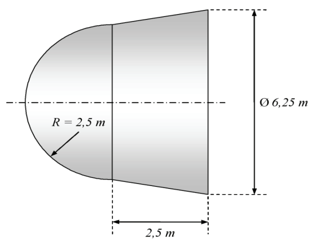


Figure 3: Bulbous indenter

In addition to the bow simplification, the striking positions are reduced. For the conventional structure striking on the webframe and between the webframes is simulated, for the novel structure having no webframes, striking between the bulkheads is simulated; both with three vertical striking positions for the tanker and two vertical striking positions for the ROPAX.

NUMERICAL MODELLING

LS-DYNA serves as a tool to simulate the force-penetration of a ship side structure under collisions with highly non linear deformations including rupture. The models are created with Ansys and converted to LS-DYNA input files. The post processing is made with LS-PREPOST.

The structures were modelled using four nodded quadrilateral Belytschko-Lin-Tsay shell elements with 5 integration points through the thickness. The initiation and propagation of fracture in the structure was modelled in LS-DYNA by deleting elements from the model. The *CONTACT_AUTOMATIC_SINGLE_SURFACE contact scheme is applied, and the reaction forces between structure and indenter are obtained by the *CONTAC_FORCE_TRANSDUCER_PENALTY card. A static friction coefficient of 0.3 and a dynamic friction coefficient of 0.1 were used. The calculations were displacement controlled. The rigid indenter was moved into the structure with a constant velocity of 10 m/s. Dynamics were neglected and therefore not considered in the simulations. Therefore all translational degrees of freedom are fixed on both ends of the model. Symmetry conditions are applied to the mid plane of the model.

MATERIAL PROPERTIES

The material data used in the simulations was obtained by Peschmann [4] in the following power law of the form

$$\sigma = \mathrm{K} \cdot \mathrm{e}^{\mathrm{n}} \qquad \text{(Eq.1)}$$

with a strength coefficient *K=730* and a strain hardening index *n=0.20*. The experimental yield stress was found to be 284 MPa. The Young's Modulus equals 206 MPa, Poisson ratio equals 0.3 and the steel density is 7850 kg/m³.

EVALUATION OF THRU THICKNESS STRAIN AND MESH SENSITIVITY ACCORDING TO GL

Thinning strain, or thru thickness strain, in shell elements is evaluated on the basis of the incompressibility condition. The thru thickness strain criterion in LS-DYNA erases an element if the through thickness strain reaches its critical limit. To evaluate the critical thru thickness strain at the moment of fracture, an empirical criterion is presented by [3] and [4]:

$$\varepsilon_f(l_e) = \varepsilon_g + \varepsilon_e \cdot \frac{t}{l_e} \qquad \text{(Eq.2)}$$

where ε_g is the uniform strain and ε_e is the necking strain, t is the plate thickness and l_e is the individual element length. It is commonly recommended that the ratio l_e/t is not less than 5 for shell element. The values of uniform and necking strain achieved from thickness measurements related to the calculated stress states given in [3] are 0.056 for the uniform strain and 0.54 for the necking strain in the case of shell elements. Implementation of those values is done for material 123 of LS-DYNA. The pre-calculated critical strain values are input for the material definition. The critical strain values are a function of the actual shell thickness and the prescribed element size in the contact region of 50x50 mm. Therefore the number of needed material definitions for an individual finite element model is equal to the number of structural components with different thicknesses.

TIME STEP CONTROL: MASS AND STIFFNESS SCALING

LS-DYNA solves the governing equations using explicit time integration. Explicit methods lead to numerically unstable solution if the time step is too large. In a stable solution the stress wave is not allowed to cross the whole element during the time step. For shell elements the reasonable time step *Δt* can be calculated by means of the equation below [5].

$$\Delta t = \frac{L}{c} \qquad \text{where } \ c = \sqrt{\frac{\mathrm{E}}{\rho\left(1 - \nu^2\right)}} \tag{Eq.3}$$

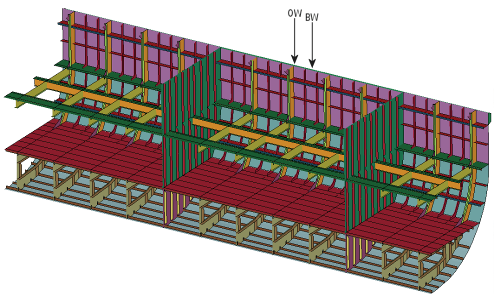
Where L is the maximum element edge length, c is speed of sound stress wave in shell element, E is Young's modulus, ρ is density of material and ν is Poisson coefficient. Substituting the expression for c into the expression for Δt shows that the time step size becomes larger when the density of the material becomes larger. At the same time the kinetic energy appears to be more significant during analyses. Hence, the usage of a larger density value is reasonable when the deformation process of structures is not disturbed much due to increased kinetic energy. Usually disturbance is not significant in large scale analyses where contact and failure regions are local and loading is relatively slow. In these calculations local effects are considered and therefore material density is set ten times larger than the actual value. Resulting in a larger time step and a stable result achieved in less time. After each calculation the time histories for the deformation and the kinetic energy of the struck ship were compared to assure the insignificant proportion of the kinetic energy.

In addition to mass scaling, stiffness scaling was used to degrease the simulation time. The basic idea is to control the minimum time step size, and keep it as a function of the mesh size within the contact area, and not as a function of a few smaller elements within the complete model. For a very small element below the mesh size of the contact area, for example due to certain geometry, the elastic modulus will be modified to keep the time step above the value defined by this smallest element. Surely changing the stiffness of an element has to be handled with care. However the total amount of smaller elements than those within the contact area is less than 0.5‰, and comparative simulations showed no difference. The minimum time step was therefore chosen to be 4e-6 s. The simulation time however was cut down to ~20% of the initial time.

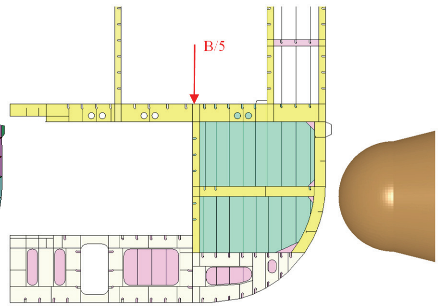
RESULTS OF THE SIMULATION

The resulting energy-penetration curves for the Tanker and ROPAX vessel are given in Figure 5 and Figure 6. The energy-penetration curves are end at the point of inner hull tearing. Figure 4 shows a graphical presentation of the impact locations. The GL approval procedure is developed for double side structures where the inner hull tearing is the critical failure. In ROPAX ships the comparison of two structures is not that straight forward, due to the conventional single hull. The critical failure is defined as tearing of the B/5 bulkhead; see Figure 5. Thus, to compare the novel and the conventional structure the calculations are carried until tearing of the B/5 bulkhead. The deformation energy at that point is used for comparison. In Figure 5 and Figure 6 the circles present the onset of tearing of the external plating and the curves end at the onset of tearing of the representative inner plating. The simulations reveal that there is a clear difference between the conventional and the novel structure. On the low penetration values the conventional structure is stiffer and absorbs more energy. As this stiffness is due to rather stiff web frames, the stresses concentrate on smaller area and the tearing of the external plating occurs early. In the novel structure the deformations are divided into larger area and the tearing occurs later. When the penetrations increase, the response of the conventional structure becomes softer compared to the novel structure. In case of the ROPAX vessel the B/5 bulkhead in both structural versions is breached almost at the same penetration depth, but due to the softer response of conventional structure, its energy levels at breaching are much lower.

The novel structure absorbs 30% more energy implemented into the Tanker and 50% more energy implemented into the ROPAX.



a) horizontal positions between the web frames (BW) and on the web frame (OW)



b) the vertical impact location at 4.8 m from the bottom

Figure 4: Locations of the impact points along the ROPAX ship

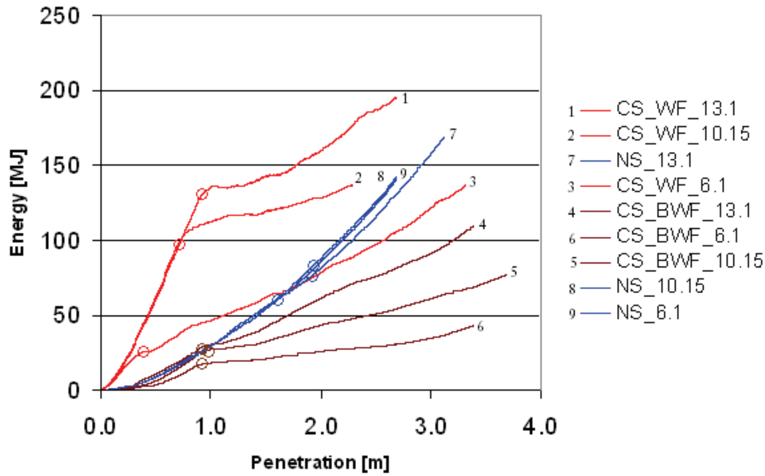


Figure 5: Energy-Penetration curves for the Tanker (CS-conventional structure, NS-novel structure, WF-webframe, BWF-between webframe, the number indicates the vertical striking position measured from the keel)

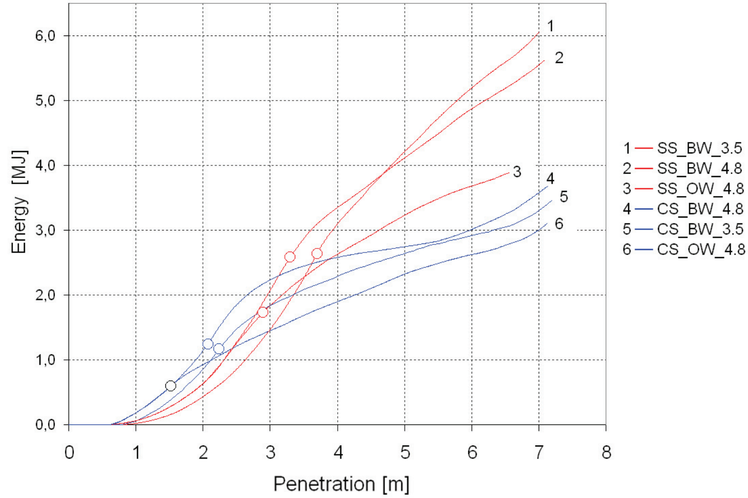


Figure 6: Energy-Penetration curves for the ROPAX (CS-conventional structure, SS-novel structure, OW-webframe, BW-between webframe, the number indicates the vertical striking position measured from the keel)

DISCUSSION

Energy curves, especially the points of failure, presented for different structures aim to be used for comparative purposes and not as the absolute values. Several references [6,7] show that the determination of the onset of tearing in such a large FE calculations is a sophisticated task and is dependent on material properties, mesh density, failure mechanism etc. Here, the different structures and their collision areas are modelled strictly following the same principles. Thus, there are no significant differences between the behaviour of different structures and the effects of above mentioned complications are minimized.

The paper presents the potential of novel structures as, with slight increase in ship mass, the energy absorption capacity can be significantly increased. With further development and optimization of new structures the weight of the ship could still be reduced to be equal or even less than that of conventional ships.

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