POAC'13
Espoo, Finland

Proceedings of the 22nd International Conference on
Port and Ocean Engineering under Arctic Conditions
June 9-13, 2013
Espoo, Finland

Dimensioning loads for the cross-deck of icebreaking trimarans

Ville Valtonen [1]
[1]Aker Arctic Technology Inc.

ABSTRACT
Icebreaking trimaran is a new, promising concept, which is under development in Aker Arctic. As a part of this development project, the dimensioning loads for the cross-deck structure were evaluated. There is no previous experience of multihull ships operating in ice. Thus the dimensioning loads are derived from theoretical analyses and calculations based on measured full-scale ice loads.

The loads and structural response are analysed for an example trimaran, which operates in the Baltic Sea. Global loads on the cross-deck are evaluated in icebreaking, beaching on an ice ridge, compressive ice and maneuvering. For comparison purposes the open-water loads are also estimated. Based on these calculated loads, stresses in the cross-deck are calculated and the dimensioning scenarios are selected. The results suggest that the cross deck structure of an icebreaking trimaran similar to the example ship can be feasibly built. The methods presented can be used to dimension the cross-deck of an icebreaking trimaran for a first-year ice conditions.

INTRODUCTION
The aim of this study is to determine load cases for dimensioning of the cross-deck of an icebreaking trimaran, and to develop methods for calculation of these loads. For local dimensioning loads of the cross-deck, as well as for the hull and superstructure design, existing methods can be used. For dimensioning purposes, the main difference between icebreaking trimaran and existing ships and methods is the cross-deck.

All existing icebreakers are monohull vessels, and all existing trimarans are fast and slender open water vessels. For both of these, the dimensioning loads differ from those of the icebreaking trimaran. Thus the loads are based on theoretical analysis and statistical treatment of measured ice loads. Extended calculations and derivations of equations can be found on the Master's thesis of the author (Valtonen, 2012), on which this paper is based on.

EXAMPLE SHIP
For the analysis and comparison of loads, an example ship was used. The example ship was chosen to be similar to a concept ship under development in Aker Arctic, with simplified geometry. The example ship is a fairly light icebreaker operating in the Baltic Sea, with length of 75m, beam of 40m and draft of 5m. The total displacement is about 4500 tons, of which each side hull is about 275 tons. Hull shape of the example ship can be seen in figure 1 and main dimensions in figure 2.

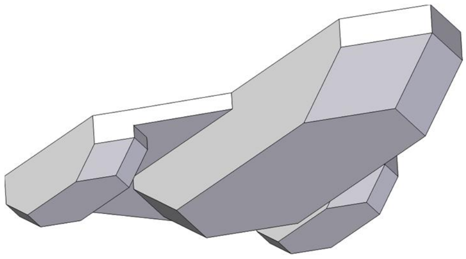

Figure 1: Example trimaran used in calculating and comparing the loads.

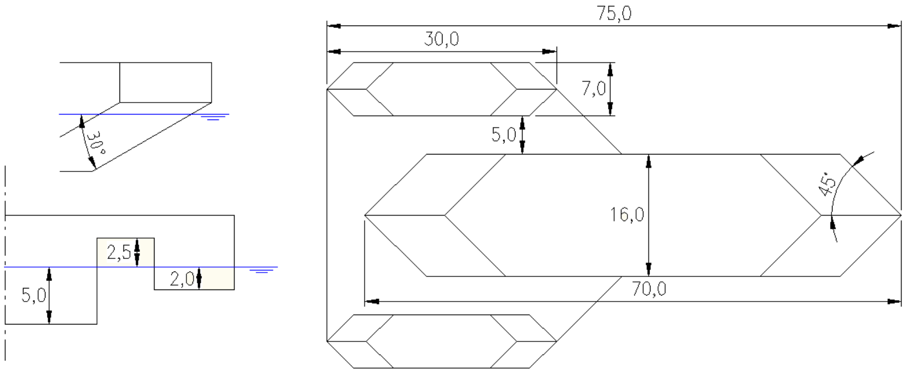

Figure 2: Dimensions of the example trimaran.

For comparison of the loads, a finite element model of the example ship was constructed. In the model, main hull was considered to be rigidly supported and all loads were applied to the side hull. Simple beam theory cannot be used for comparisons, because the loads are applied in different locations and with different orientations. Additionally, the cross-deck structure is not slender enough to be considered as a simple beam.

In the model, a simple double bottom type structure was used for the cross-deck, and it was connected to web frames and bulkheads in both hulls, as shown in figure 3. Model consisted of linear 4-node shell elements of about 250mm size. The model was static, with linear elastic material. As the ship is symmetrical and loading of one side hull is independent of the other, only a half model was used, and the bow was cut away as shown in figure 4.

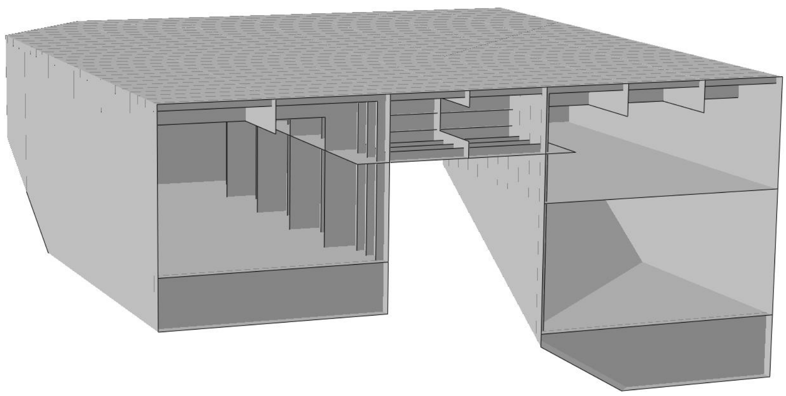

Figure 3: Structural arrangement of the example ship.

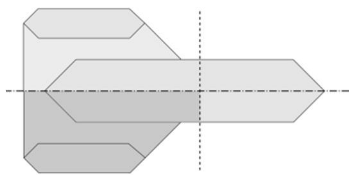

Figure 4: Because of symmetry, it was possible to use a half model in finite element analysis.

LOAD CASES

Dimensioning loads were calculated in different load cases, which include all significant loading scenarios in open water and first year ice. Open water loads were calculated in wave conditions of the Baltic Sea.

Ice loads were divided into two parts. Icebreaking loads include all loads from normal operation of icebreaker, for example breaking of level ice and ridges, collisions with ice floes and navigating in channels and brash ice. Additionally, special situations were considered, including beaching on a ridge, compressive ice and maneuvering. Ice loads were calculated in ice conditions of the Bay of Bothnia, which has generally the most difficult ice conditions in the Baltic Sea. All ice loads were modeled as line loads acting in the normal direction of plating on the waterline.

Open water loads

Open water loads were calculated mainly for comparison purposes with approximate methods. As there is not yet knowledge of the seakeeping properties of the trimaran, the assessment is not accurate enough for basic design purposes, but can be used for comparison against the ice loads and for concept level design. The loads considered included still water, wave and global slamming loads.

Wave loads were calculated with static-balance method and the load level was verified by comparing it to loads calculated by Germanischer Lloyd’s rules for high speed craft (2012) and Lloyd’s Registers rules for trimarans (2006).

In static-balance method, the trimaran was balanced transversely on a static wave with wavelength similar to trimarans beam. This scenario causes highest loads to the cross-deck (Blanchard & Chunhua, 2007). Wave height was chosen according to wave statistics and wave breaking criteria. Dynamic and inertia effects were taken into account by multiplying the loads with a suitable coefficient, which was chosen conservatively.

LR rules provide both rule formulas and methods for direct calculations. It was shown (Valtonen, 2012) that the rule formulas cannot be applied to the icebreaking trimaran due to the size of side hulls, large beam and large initial stability, which exceed the applicability limits of the rules. With some further research the rule formulas could be modified to apply also to wider trimarans, such as the icebreaking trimaran. The direct calculation procedure of LR rules for trimarans relies heavily on the use of either CFD or model tests, neither of which were possible within the resources of this study. Thus the simple inclination criterion for cross-deck dimensioning load was used. It does not provide full analysis of dimensioning loads, but can be used to approximate the magnitude of wave loads. Wave loads calculated by GL and LR methods agreed well with loads calculated by static balance method.

Slamming load was calculated with forced movement method. In slamming calculation, a V-bottom with deadrise angle of 30° was used instead of flat bottom, as that corresponds better to the real hull shape of the trimaran. Accurate treatment of slamming was not possible because of unknown seakeeping characteristics of the vessel and the static finite element model, which cannot accurately calculate the response to the highly dynamic slamming load.

Icebreaking loads

Icebreaking loads include various loads from normal operating, such as breaking of level ice and ridges and navigation in channels and brash ice. While there are theoretical models for loads in some of the individual situations, the overall effect of these loads is far too varied and complicated to be assessed reliably with a simple calculation. Thus the assessment was based on measured loads from IB Sisu (Kujala & Vuorio, 1986) and verified with loads measured on MT Kemira (Kujala, 1989). These measurements were conducted in the Bay of Bothnia.

The icebreaking load is applied to the whole bow area of side hull, as that produces maximum global load. Because the trimaran operates both ahead and astern in ice and breaks through ridge fields mostly astern, the icebreaking load is also applied to the stern of the side hull.

Both long-term measurements include daily load maxima from several winters, and can be treated statistically to estimate the maximum load during the operating life of the ship. Hänninen (2002) showed that the maximum ice loads follow Gumbel I-distribution, which is given by (Ochi, 1990)

$$F(q) = e^{-e^{c(q-u)}} \tag{1}$$

and the probability of exceedance is

$$F(q) = 1 - \frac{t_0}{t} \tag{2}$$

By combining (2) and (3), the most probable extreme load can be solved

$$q(t) = u - \frac{1}{c}\ln(-\ln(1-\frac{t_0}{t})) \tag{3}$$

where u and c are the parameters of the distribution, t_0 the length of one time interval used in measurements and t the return period. Probability of exceeding this load during the lifetime of the vessel is about 63% (Ochi, 1990), which is clearly not suitable for design purposes. Instead, it is desirable to define a smaller probability of exceedance r, say 1%. For large n and small r, the probability of exceedance can be presented in form

$$F(q) = (1-r)^{t_0/t} \approx 1 - \frac{rt_0}{t} + O(r^2) \approx 1 - \frac{rt_0}{t} \tag{4}$$

and the design extreme load becomes

$$q(t) = u - \frac{1}{c}\ln(-\ln(1-\frac{rt_0}{t})) \tag{5}$$

Based on the measurements from Terry Fox (Johnston et al., 2008), it is concluded that global ice load on ship bow is formed as a combination of local ice loads. Therefore, using the known scale dependency for ice load, the global line load in bow can be calculated from measured local line load q_0 with

$$q = \left(\frac{l}{l_0}\right)^{-0.6} q_0 \tag{6}$$

where l is the total load length of the area under consideration and l_0 is the load length used in the measurements. The same dependency can be used for a line load and a pressure load acting on a rectangular area with a constant height. Exponent of the scale dependency is chosen to be -0.6, based on measurements (Riska & Kämäräinen, 2011).

The differences in bow shape of the ships can be taken into account with the formula that has been used in IACS PC rules (Kujala et al., 2007)

$$F \propto \alpha\beta_n^{-0.5} \tag{7}$$

where α is the waterline angle and β_n is the normal frame angle. By combining equations (5), (6) and (7), the design load can be calculated by

$$q(t) = \left(\frac{l}{l_0}\right)^{-0.6} \left(\frac{\alpha}{\alpha_0}\left(\frac{\beta_n}{\beta_{n,0}}\right)^{-0.5}\right)\left(u - \frac{1}{c}\ln\left(-\ln\left(1-\frac{rt_0}{t}\right)\right)\right) \tag{8}$$

where x_0 refers to the measured ship and x refers to the new ship. The calculated design ice load for the bow of the side hull of the example ship was q=935kN/m, acting on a total load length of l=9.90m.

Beaching on an ice ridge

When the bow of the main hull beaches on a ridge, the forces on the cross-deck are not very large. Loads to the cross-deck caused by the sinking of the stern are much smaller than the loads caused by waves.

However, if one or both of the side hulls beaches on a ridge, the loads on the cross-deck may be very large. While not part of normal operations, this scenario can occur very easily when, for example, the main hull of the trimaran passes a ridge through an old channel or lead, while the side hulls hit the ridge and beaches on top of it.

Loads in this case were calculated by balancing the trimaran on top of a ridge and calculating the loads from static balance. Loads during the initial ram and the beaching event are included in icebreaking loads. Typical ridge in the Baltic Sea has a sail height of 0,5-2,0m (Kankaanpää, 1989) and the ship crushes some of the sail on beaching. Thus it was concluded that the bottom of the side hull can rise to 1,0m above water. Static balance in this condition was used to calculate the dimensioning load, as shown in figure 5.

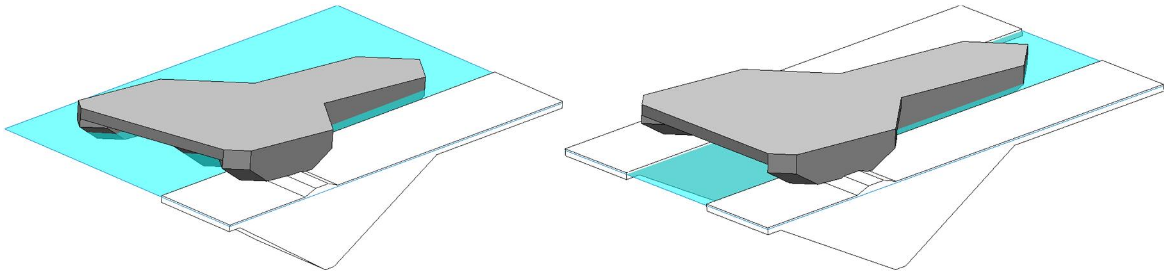

Figure 5: Trimaran with one and both side hulls beached on a ridge.

The force on the bottom of the side hull can be calculated from balance equations

$$\begin{aligned} \rho g \nabla + F &= mg \\ x_\nabla \rho g \nabla + xF &= x_m mg \\ y_\nabla \rho g \nabla + yF &= y_m mg \end{aligned} \tag{9}$$

where ρ is water density, g gravity, m mass of the ship and ∇ displacement. When there is a force F acting on the ship at point (x, y), the center of the buoyancy has to be at point

$$\begin{aligned} x_\nabla &= x_{\nabla 0} \frac{\nabla_0}{\nabla} + x_F \left(\frac{\nabla - \nabla_0}{\nabla} \right) \\ y_\nabla &= y_{\nabla 0} \frac{\nabla_0}{\nabla} + y_F \left(\frac{\nabla - \nabla_0}{\nabla} \right) \end{aligned} \tag{10}$$

and the force can be calculated from

$$F = (\nabla_0 - \nabla) \rho g \tag{11}$$

where ∇_0 corresponds to displacement when F=0.

The calculated force on the bottom of the side hull was 4985kN when one side hull was beached and 6442kN on each hull when both side hulls were beached. The reasonability of this dimensioning scenario was checked by calculating the load capacity of the ridge, as well as the necessary initial velocity for beaching. A rough estimation of load capacity for a 15m deep ridge gave a breakthrough load of about 10237kN. Thus the strength of ice does not limit the load, at least in case of sufficiently large ridge.

Beaching velocity was calculated with energy method, by assuming that the kinetic energy of the ship is absorbed by potential energy of raising the side hull, crushing of the ice and friction. Minimum velocity for beaching of both hulls was calculated to be 8,7kn, which is a realistic operating speed. These calculations support the conclusion that the beaching scenario is possible.

Loads in compressive ice

Loads in compressive ice are calculated based on a design formula developed by Daley and Kendrick (2008)

$$q = 1500\Psi k h_i^{1.25} l^{-0.7} \tag{12}$$

where k is coefficient for the effect of ice conditions, ψ is factor of safety, h_i is level ice thickness and l is load length. The design formula is developed for independently operating vessels, which do not have icebreaker support to reduce the loads, and thus it provides sound basis for load evaluation for the icebreaking trimaran. The equation is mainly based on measurements from the Baltic Sea (Daley & Kendrick, 2008), and thus the ice conditions coefficient can be taken as k=1,0. As loads are calculated to correspond to maximum expected load, and factor of safety is considered in allowable stresses, ψ=1. For the pressure-area relationship, slightly more conservative exponent of -0.6 is chosen instead of -0.7 used by Daley and Kendrick, because this was determined to correspond better to measurements (Valtonen, 2012).

The design formula (12) is based on loads measured from ships with vertical sides. It is known that even a slight inclination of the side decreases the ice loads in compressive situation significantly, as the ice fails in bending instead of crushing. For consideration of this effect, a formula from rules of Det Norske Veritas (2012) is used

$$q \propto \frac{\sin 10^\circ}{\sin \beta_n} \tag{13}$$

where $\beta_n > 10°$ is the frame normal angle. By combining all these and using conservative maximum of h_i=1,30m for ice thickness in the Bay of Bothnia (Kujala et al., 2007), design load can be calculated by

$$q = \begin{cases} 1500 h_i^{1.25} l^{-0.6} & , \beta_n \leq 10^\circ \\ \dfrac{260}{\sin \beta_n} h_i^{1.25} l^{-0.6} & , \beta_n > 10^\circ \end{cases} \tag{14}$$

For the example ship, a load of 273,6kN/m was distributed along length of 29.44m in compressive ice. This load was verified by comparing it against the measured loads from MT Kemira (Kujala, 1989) and loads obtained from damage records (Kujala, 1991). The design load was sufficiently larger than the loads measured from merchant ships to provide a reasonable basis for designing of an icebreaker.

Maneuvering loads

Maneuvering loads were calculated with energy method developed first by Popov and then by Daley (1999). In this method, effective kinetic energy of the ship is assumed to be equivalent with energy used for crushing of the ice

$$E_{kin} = W_{crushing} \tag{15}$$

Kinetic energy can be calculated from

$$E_{kin} = \tfrac{1}{2} m_e v_n^2 \tag{16}$$

where v_n is normal velocity and m_e is effective mass, which can be calculated with equations presented by Daley (1999). The geometry used in calculations is shown in figure 6. The trimaran collides sideways with a large ice floe, and the impact point is in middle of the side hull. In practice, most of the maneuvering loads are generated when operating inside ice field, and thus this approach gives a conservative upper limit for the loads.

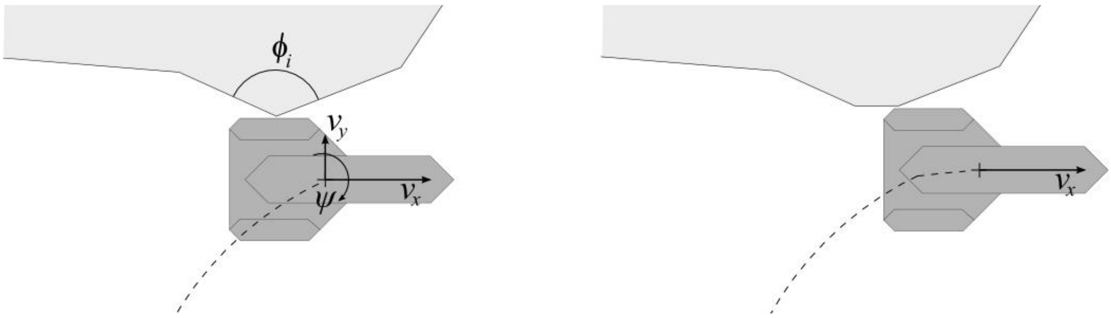

Figure 6: In maneuvering load scenario, ship turns sharply in open water and the side hull collides with a large ice floe.

Force needed to crush the ice is

$$F(\zeta) = p(\zeta)A(\zeta) = \sigma_{i,c} A(\zeta)^{-0.6} A(\zeta) = \sigma_{i,c} A(\zeta)^{0.4} \tag{17}$$

where $\sigma_{i,c}$ is crushing strength of ice and ζ is indentation depth. For the geometry shown in figure 6, the contact area is

$$A(\zeta) = 2h_i \zeta \tan(\phi_i/2) \tag{18}$$

Thus the crushing force is

$$F(\zeta) = \sigma_{i,c} \left(2h_i \zeta \tan(\phi_i/2)\right)^{0.4} \tag{19}$$

and corresponding energy is

$$W_{crushing} = \int_0^{\zeta} F(\zeta) d\zeta = \tfrac{1}{1.4} \sigma_{i,c} \left(2h_i \zeta \tan(\phi_i/2)\right)^{0.4} \zeta^{1.4} \tag{20}$$

By combining equations (15), (16) and (20), maximum indentation can be solved

$$\zeta = \left(\frac{1.4 \tfrac{1}{2} m_e v_n^2}{\sigma_{i,c} \left(2h_i \tan(\phi_i/2)\right)^{0.4}} \right)^{\frac{1}{1.4}} \tag{21}$$

and the maximum force can be then calculated by combining equations (19) and (21)

$$F(\zeta) = \sigma_{i,c}^{\frac{5}{7}} \left(\tfrac{7}{5} h_i m_e v_n^2 \tan(\phi_i/2) \right)^{\frac{2}{7}} \tag{22}$$

For the example ship, it was assumed that the ship turns with maximum speed and minimum turning radius, which were estimated to be v=10kn and R=L=75m. Crushing strength of ice depends on the loading direction and scale. For this calculation, value of $\sigma_{i,c}$ =1500kPa on nominal contact area of 1m^2, based on measurements of first-year ice crushing against ships side in the Baltic Sea (Daley & Kendrick, 2008), was used. The maximum indentation into the ice was ζ=3,091m and the maximum force F=6911kN.

RESULTS

The load cases were compared with the finite element model shown in figures 7 and 8. Calculated maximum stresses in the cross-deck structure are shown in table 1.

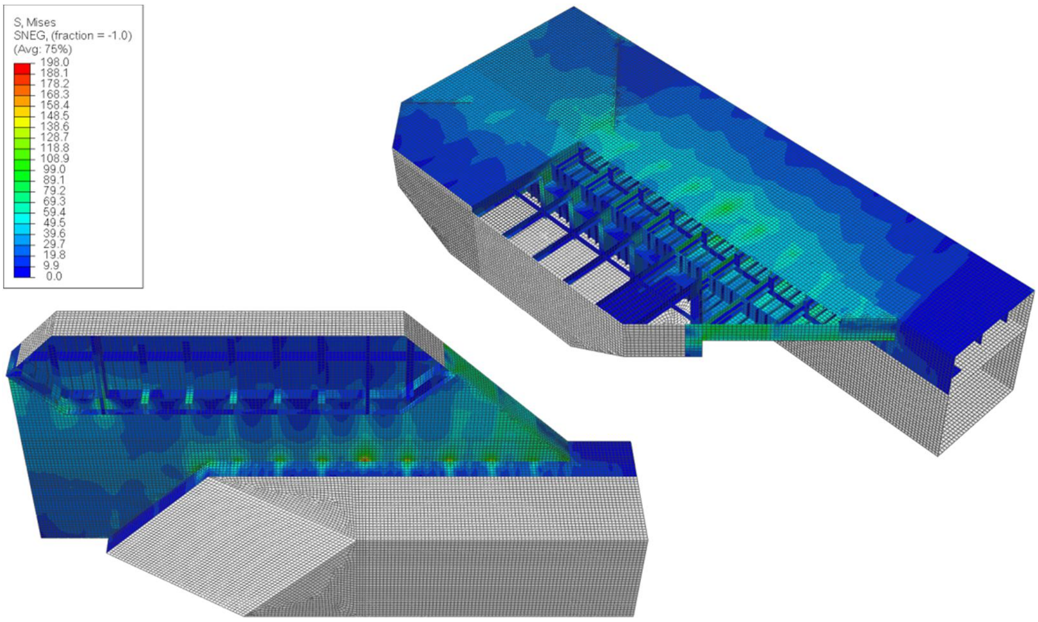

Figure 7: Distribution of von Mises stress in load case 5, icebreaking ahead.

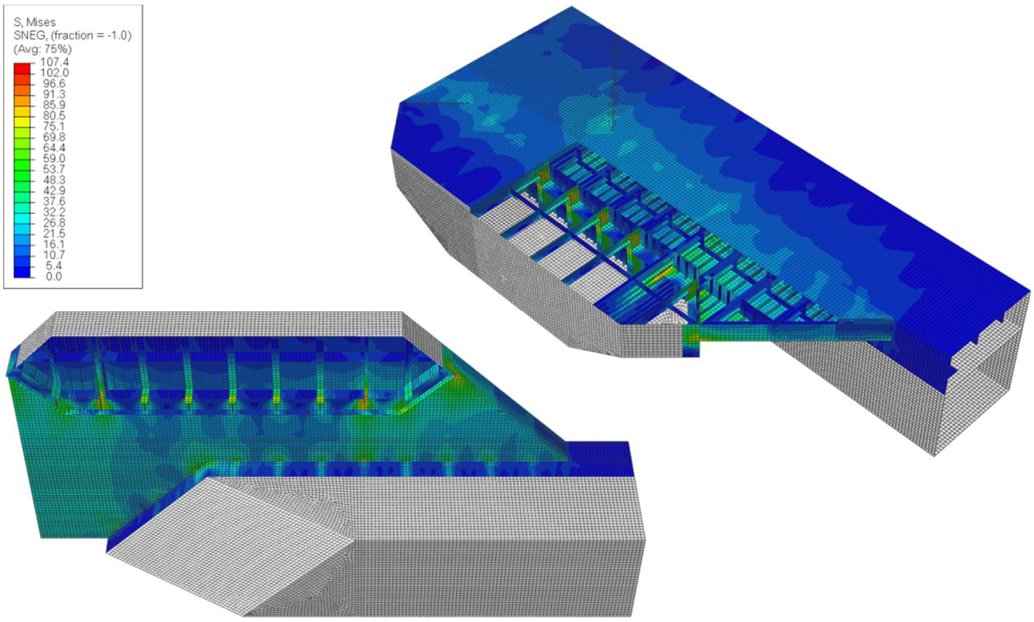

Figure 8: Distribution of von Mises stress in load case 11, maneuvering.

Table 1: Maximum von Mises stresses in cross-deck structure in all load cases.

Load case	σ_{vM} (MPa)
1. Still water	21
2. Wave load, static balance, hogging	166
3. Wave load, static balance, sagging	303
4. Slamming	(82)
5. Icebreaking, ahead	198
6. Icebreaking, astern	267
7. Compression, whole side	78
8. Compression, fwd. half	78
9. Compression, aft half	64
10. Beaching	94
11. Maneuvering	107

DISCUSSION AND CONCLUSIONS

For the example ship operating in Baltic Sea conditions, and trimarans with similar main dimensions on other comparable sea areas, wave loads and ice loads are of the same magnitude, and both have to be considered in design.

For concept design, the approximate method used in this paper is sufficient for estimation of the open water loads. For basic design of the vessel, a more accurate assessment of wave loads and slamming with either model tests or CFD is necessary. Wave load in sagging condition produced largest stresses for the example ship. In design, both sagging and hogging load cases should be considered. Slamming load was well below wave loads, but it's highly dynamic nature may cause vibrations or whipping, and further research should be made.

Of ice loads, icebreaking loads produced largest stresses and are clearly the primary dimensioning loads in ice. Similarly to the wave loads, icebreaking loads cause highest stresses in the joint of the main hull and the cross-deck. Beaching loads were significantly smaller and are not necessary to be considered.

In compressive ice and maneuvering, the stress level was much lower than in open water and in icebreaking. However, as the load is mainly horizontal instead of vertical, the stress distribution is different, and highest stresses occur in the joint of side hull and cross-deck. Stress levels caused by maneuvering were slightly higher than those caused by compression. Thus maneuvering is also a dimensioning load. As the loads are so close to each other, both should be still checked and the higher one used in dimensioning of the cross-deck and side hull joint.

The example ship demonstrates that the cross-deck of an icebreaking trimaran of similar size and operating conditions can be feasibly built. Necessary material thicknesses and overall structural arrangements are reasonable, and should not pose any major problems in design. It was also seen that one must consider the web framing in cross-deck area carefully, as the web frames have an important role in distributing the stresses from cross-deck to hull.

For trimarans differing significantly from the example ship in either main dimensions or operating environment, the methods outlined here can be used to calculate the loads, providing that underlying assumptions are considered as necessary. All load cases should be checked, as the changes in main dimensions or operating conditions could change the relative load levels. For loads in multiyear ice conditions, further research should be made.

Currently available tools are sufficient for designing of an icebreaking trimaran. Open water loads can be estimated for concept design with static-balance method or with rule formulas, when proper care is given to use the formulas only inside their limits of applicability. For more accurate assessment, CFD or model tests can be used. For ice loads, the methods outlined in this paper can be used.

ACKNOWLEDGEMENTS

This study was supported by Aker Arctic Technology Inc. Guidance and expert advice from Erkki Ranki (Aker Arctic) and Professor Pentti Kujala (Aalto University) is much appreciated.

REFERENCES

Blanchard, T. and Chunhua, G., 2007. Rules for the classification of trimarans. Journal of Naval Engineering, vol. 44, issue 1, 2007.

Daley, C. and Kendrick, A., 2008. Direct design of large ice class ships with emphasis on the midbody ice belt. Proceedings of OMAE 2008.

Daley, C., 1999. Energy based ice collision forces. Proceedings of POAC ’99, vol. 2. Expanded edition of the conference paper with full derivations.

Det Norske Veritas, 2012. Rules for Classification of Ships, Part 5, Chapter 1, Section 4, Ships for navigation in ice – Vessels for Arctic and ice breaking service.

Germanischer Lloyd, 2012. Rules for Classification and Construction, Chapter I-3-1, High speed craft.

Hänninen, S., 2002. The Use of Statistical Methods in Determination of Design Ice Load on Ship Hull Frame in the Baltic Sea. Master’s thesis, Helsinki University of Technology.

Johnston, M., Ritch, R. and Gagnon, R., 2008. Comparison of impact forces measured by different instrumentation systems on the CCGS Terry Fox during the bergy bit trials. Cold Regions Science and Technology, vol. 52, p. 67-82.

Kankaanpää, P., 1989. Structure of first year pressure ridges in the Baltic Sea. Proceedings of the POAC ’89, vol. 1 p. 87-102.

Kujala, P. and Vuorio, S. 1986. Results and statistical analysis of ice load measurements on board icebreaker Sisu in winters 1979 to 1985. Winter Navigation Research Board, Research Report no. 43.

Kujala, P., 1989. Results of long-term ice load measurements on board chemical tanker Kemira in the Baltic Sea during winters 1985 to 1988. Winter Navigation Research Board, Research Report no. 47.

Kujala, P., 1991. Damage statistics of ice-strengthened ships in the Baltic Sea 1984-1987. Winter Navigation Research Board, Research Report no.50.

Kujala, P., Suominen, M. and Jalonen, R., 2007. Increasing Safety of Icebound Shipping, Final Scientific Report of SAFEICE project. Helsinki University of Technology.

Lloyd’s Register, 2006. Rules for the Classification of Trimarans.

Ochi, M.K., 1990. Applied probability & stochastic processes in engineering and physical sciences. Wiley.

Riska, K. and Kämäräinen, J., 2011. A review of ice loading and the evolution of the Finnish-Swedish ice class rules. Proceedings of the SNAME Annual Meeting & Expo 2011.

Valtonen, V., 2012. Determining dimensioning loads for the cross-deck of icebreaking trimarans. Master’s thesis, Aalto University.