IACS Unified Requirements for Polar Ships

Background Notes to

Longitudinal Strength

Prepared for:

IACS Ad-hoc Group on Polar Class Ships

Transport Canada

Prepared by:

Claude Daley

Memorial University

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Summary

The collision force formula for assessment of longitudinal strength is;

$$F_I = \text{MIN} \left| \begin{array}{l} Fv = .534\ K_I^{0.15}\ \sin^{0.2}(\varphi)\ (\Delta\ Kh)^{0.5}\ CF_L \quad [MN] \\ \\ 1.2\ CF_F \quad [MN] \end{array} \right. \qquad (s1)$$

where

$K_I = K_f/Kh$

K_f is a parameter that the defines the indentation stiffness and depends on the shape of the bow. Two cases are given:

a) for the case of a blunt bow form

$$K_f = (2\ C\ B^{1-e_b} /(1+e_b))^{0.9}\ (\tan(\varphi_{stem}\ ))^{-(1+e_b)(0.9)}$$

b) for the case of wedge bow form e_b =1 and the above simplifies to

$$K_f = (\ \tan \alpha_{stem} / \tan^2 \varphi_{stem}))^{0.9} \qquad \text{(wedge shaped bows } \alpha_{stem} < 80\text{deg)}$$

Kh = $\rho\ g\ Awp$
A_{wp} = ship waterplane area (m^2)
B = beam
CF_L = Class factor (representing ice strength and speed influences)
Δ = ship displacement (1000 t or Mkg)
ρ_w = water density (Mkg/m^3)
g = acceleration due to gravity

γ_{stem} = stem angle at the FP (measured down from the vertical)
α_{stem} = waterline angle at the FP

C = $1/ (2\ (L_B/ B)^{e_b})$
L_B = bow length (used in bow form equation $y = B/2\ (x/L_b)^{e_b}$)
e_b = bow shape exponent
CF_F = Flexural Failure Class Factor
Annex B describes the terms e_b and L_B in more detail.

The above formula is used to assess the shear forces and bending moments in a vessel. The resulting requirements are shown to be compatible with existing rules and experience. As well, it is shown that normally the open water wave bending requirements will govern design.

1. Introduction

This document describes how the longitudinal strength requirement in the IACS Unified Requirements for Polar Ships has been developed. The principles upon which the ice ramming force was developed are described. The ship and ice parameters are discussed. The background to the intended design load calculation methodology is described. The strength criteria are presented.

The Polar Rules base the ice loads on a specific collision scenario. The scenario for the plating and framing design is a glancing (or oblique) collision. The longitudinal strength requirement is based on a head-on ramming scenario. Figure 1 illustrates the two cases. The mechanics of the two cases are similar in many ways, though not identical. In the URs, the glancing collision loads [16] assume a 'Popov' type of collision, meaning that no ride-up onto the ice is considered. The head-on ramming loads are based on models which take account of the ride-up (beaching). Comparisons have been undertaken to ensure that the two approaches give compatible results using both models to represent impacts near the stem.

The longitudinal strength requirements for polar ships have been developed based on extensive study [1,2,3,4,5]. The proposed rule formulation is supported by field data, model tests, numerical models and two analytical solutions to the collision equations.

Generally wave loads will govern the longitudinal strength requirements. Only in certain cases will the ice loads dominate. These will tend to be in the cases of large vessels, with high ice class.

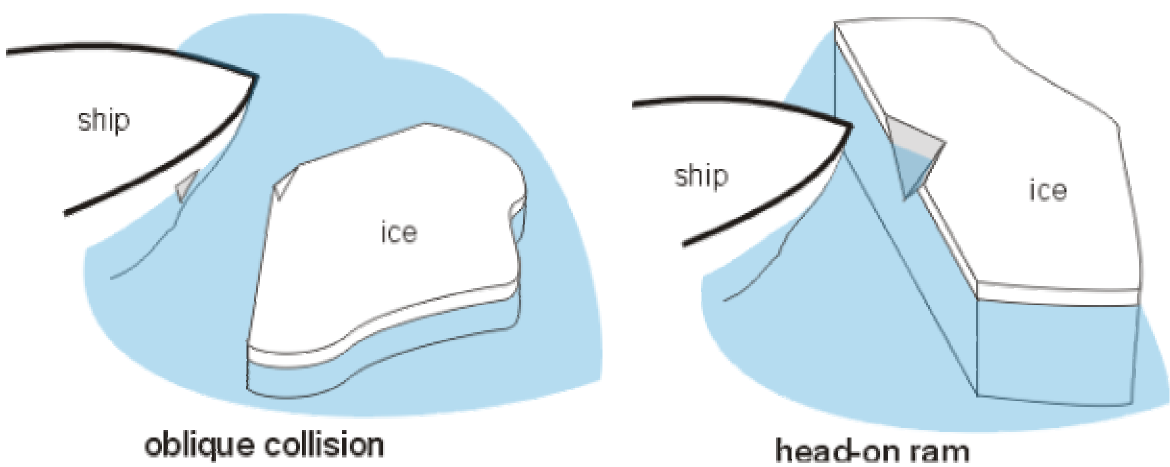

Figure 1. Oblique and ramming collision scenarios.

2. Head-on Collision Values

Ramming Force

As a ship collides head-on with the ice edge, all of its energy is initially kinetic. During the ram the kinetic energy is converted to potential energy and crushing energy. Fortunately, both the potential and crushing energy can be stated in terms of the maximum vertical force Fv. The maximum force occurs when the ship comes to rest at the end of the penetration. The potential energy depends on the vessel's hydrostatic properties (mainly the waterplane area). The ice crushing energy depends on the ice strength and the bow shape. **Annex A** presents an energy-based derivation of the head-on collision for two types of bow shape.

The proposed longitudinal strength requirement is based on head-on ramming. The proposed rule formula for vertical force is:

$$F_I = \text{MIN} \left| \begin{array}{l} .534\ \kappa^{0.15}\ \sin^{0.2}(\varphi)\ (D\ Kh)^{0.5}\ V \\ \\ 1.2\ \sigma_{if}\ h_{ice}^{2}\quad (MN) \end{array} \right. \qquad (1)$$

where

κ = Kice/Kh dimensionless ice strength

Kice is a parameter that defines the indentation stiffness of the ice as the stem of the ship penetrates the ice. The magnitude of Kice depends on both the ice strength and the shape of the bow. Two cases are given:

a) for the case of a blunt bow form

$Kice = p_1\ (2\ C\ B^{1-e_b}\ /(1+e_b))^{(1-ex)}\ (\tan(\varphi_{stem}\ ))^{-(1+e_b)(1-ex)}$

b) for the case of wedge bow form e_b =1 and the above simplifies to

$Kice = p_1\ (\ \tan\alpha_{stem}\ /\ \tan^2\varphi_{stem}))^{(1-ex)}$ (wedge shaped bows α_{stem} <80deg)

Kh = ρ g Awp
p_1 = ice pressure constant (MPa at 1 m^2)
ex = ice pressure exponent (we use -0.1)
φ = stem angle (measured up from horizontal)
e_b = bow shape exponent (see **Annex B**)
C = $1/\ (2\ (L_B/\ B)^{e_b})$
B = beam
L_B = bow length (used in bow form equation $y = B/2\ \ (x/L_B)^{e_b}$)
V = ship speed (m/s)
D = ship displacement (1000 t or Mkg)
ρ_w = water density (Mkg/m^3)
g = acceleration due to gravity
A_{wp} = ship waterplane area (m^2)

The calculation requires hull form parameters as well as ice class parameters. The bow form is assumed to be one of two forms, a simple wedge or a curved shape as sketched in Figure 2. The ice class parameters are given in Table 1.

Table 1 Ice class parameters and Longitudinal Strength Class factor.

Class	V [m/s]	p_1 [Mpa]	h_ice [m]	sig_f [MPa]	Longitudinal Strength Class Factor (CF_L)
1	5.70	6.00	7.0	1.40	7.46
2	4.40	4.20	6.0	1.30	5.46
3	3.50	3.20	5.0	1.20	4.17
4	2.75	2.45	4.0	1.10	3.15
5	2.25	2.00	3.0	1.00	2.50
6	2.25	1.40	2.8	0.70	2.37
7	1.75	1.25	2.5	0.65	1.81

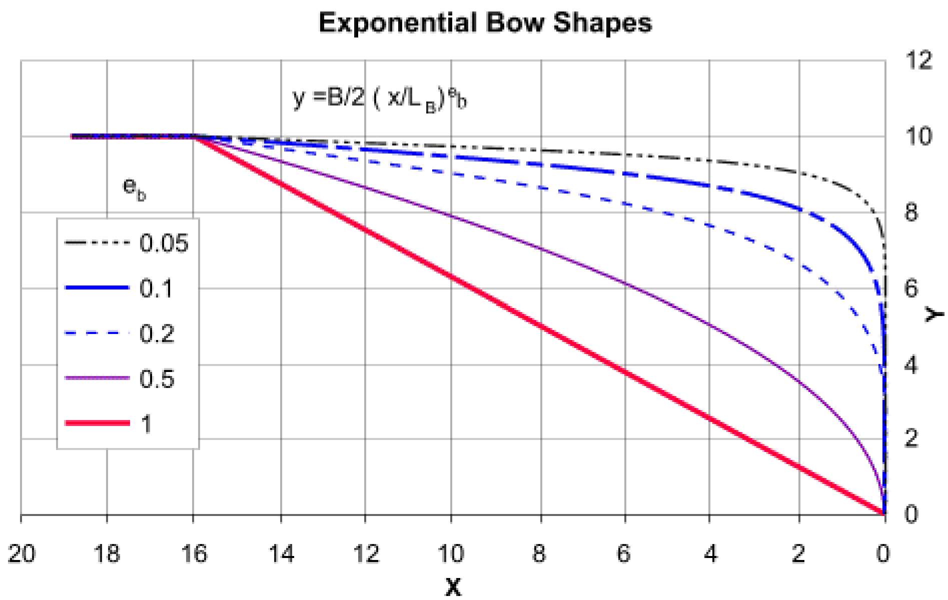

Figure 2. Bow shapes for various values of e_b. For this case B = 20, L_B = 16.

(Note: (0,0) is at the FP.)

The rule equation combines the factors that are class related (p_1 and V) into a class factor for longitudinal strength. With ex set to -0.1, the formula becomes;

$$F_I = \text{MIN} \begin{cases} .534\, K_I^{0.15}\ \sin^{0.2}(\varphi)\, (\Delta\, Kh)^{0.5}\, CF_L \\ 1.2\, CF_F\ \ (\text{MN}) \end{cases} \qquad (2)$$

where

$K_I = K_f / Kh$ dimensionless ice strength

a) for the case of a blunt bow form

$K_f = (2\, C\, B^{1-e_b} /(1+e_b))^{(0.9)}\, (\tan(\varphi_{stem}))^{-(1+e_b)(0.9)}$

b) for the case of wedge bow form e_b =1 and the above simplifies to

$K_f = (\tan \alpha_{stem} / \tan^2 \varphi_{stem}))^{(0.9)}$ (wedge shaped bows α_{stem} <80deg)

$Kh = 0.01\, A_{wp}$ [MN/m]

CF_L = Longitudinal Strength Class factor (= $p_1^{(.15)}$ V)

CF_F = Flexural Failure Class Factor from Table 5.1

all other terms as per eqn (1)

3. Bending Moments

The bending moment is derived from the vertical force as follows:

$$M_{\max} = 0.1 \cdot L \cdot \sin^{-.2} \gamma \cdot F_I \tag{3}$$

The bending moment along the vessel is to be represented by the pattern described in Figure 3. The bending moment is shifted forward in comparison to the wave bending moment. (The open water values are typical of IACS member's rules - the DnV rules were consulted.)

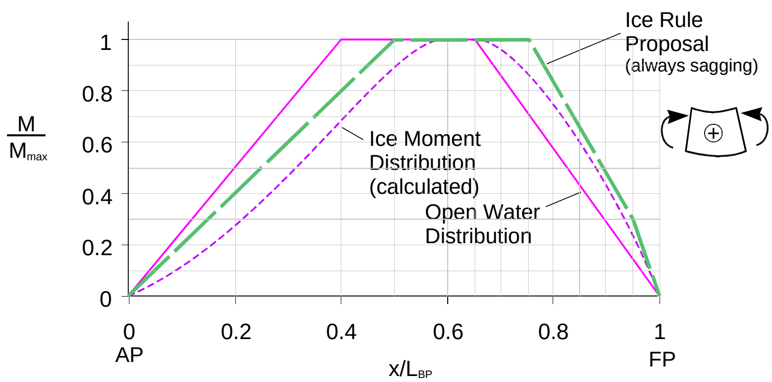

Figure 3. Ice Bending Moment Distribution Along Ship.

4. Shear Force

The shear force at the bow is the vertical ice force (equation 1). The distribution of shear along the ship is illustrated in Figure 4. It is quite different from the open water condition, stating from the maximum at the forward perpendicular. (The open water values are typical of IACS member's rules- the DnV rules were consulted.)

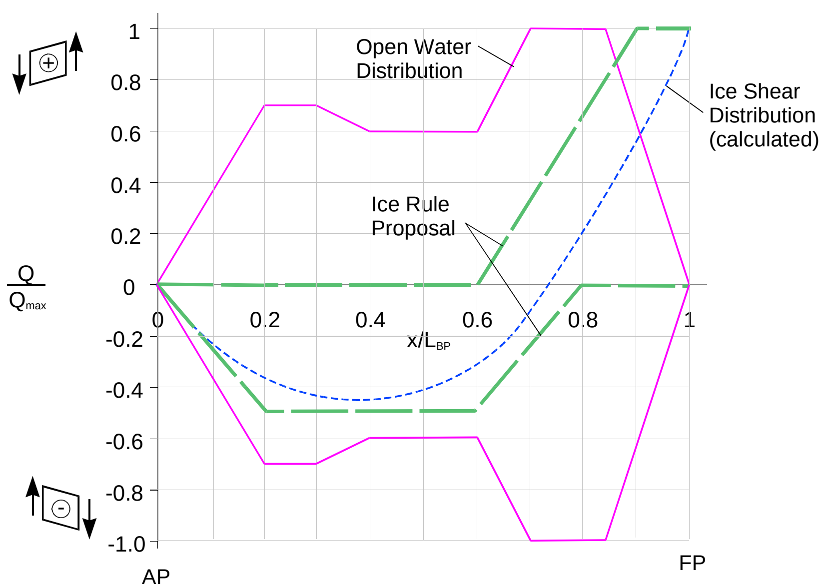

Figure 4. Ice Shear Force Distribution Along Ship.

5. *Longitudinal Strength Criteria*

The previous sections discuss the shear and bending moment values that will occur along the ship due to ice loads. Still water shear and bending moments will add to these. Together these will result in tensile, compressive and shear stresses throughout the hull. The permissible stresses under the combination of ice and still water bending moments and shear are given in Table 2.

The approach is modeled on standard IACS strength criteria. A distinction is made between allowable stress levels for icebreakers and for other ships, due to the more aggressive operations that the former may be called on to perform.

Table 2 Strength criteria for longitudinal strength

Failure Mode	Applied Stress	Permissible Stress when $\sigma_y/\sigma_u \le 0,7$	Permissible Stress when $\sigma_y/\sigma_u > 0,7$
Tension	σ_a	$\eta\ \sigma_y$	$\eta\ 0,41\ (\sigma_u+\sigma_y)$
Shear	τ_a	$\eta\ \sigma_y/\sqrt{3}$	$\eta\ 0,41\ (\sigma_u+\sigma_y)/\sqrt{3}$
Buckling	σ_a	σ_c for plating and for web plating of stiffeners $\sigma_c/1.1$ for stiffeners	
	τ_a	τ_c	

Where: σ_a = applied vertical bending stress [N/mm^2]
τ_a = applied vertical shear stress [N/mm^2
σ_y = specified minimum yield stress [N/mm$^{2]}$
σ_u = specified ultimate tensile strength [N/mm^2]
σ_c = critical buckling stress in compression, according to UR S11.5 [N/mm^2]
τ_c = critical buckling stress in shear, according to UR S11.5 [N/mm^2]
η = 0.6 for icebreakers
0.8 for other ship types

6. Comparison with Oblique Collision Forces

Oblique collision forces

Both the ramming scenario assumed in the longitudinal strength analysis and the oblique (or glancing) collision used in the plating and framing strength requirements are impact events, and could in principle be described using the same analytical model. However, the ramming case is much simpler (for example impact location is known, symmetry can be assumed) and thus it can be modeled more comprehensively. It is also easier to validate by physical testing, and as noted above many full- and model scale tests of ramming have been undertaken.

It is important for the credibility of both models that they provide similar solutions when an impact can be represented by either, i.e. for impacts at or near the stem.

Under the proposed URs, the ice load *Fn* for an oblique collision is based on a Popov-type collision, assuming a pressure-area based indentation of the ice edge;

$$F_n = (3 + 2 \cdot ex)^{\frac{2+2\cdot ex}{3+2\cdot ex}} \cdot Po^{\frac{1}{3+2\cdot ex}} \cdot \left(\frac{\tan(\phi/2)}{\sin(\beta') \cdot \cos^2(\beta')}\right)^{\frac{1+ex}{3+2\cdot ex}} \cdot \left(\frac{1}{2}\Delta_n \cdot V_n^2\right)^{\frac{2+2\cdot ex}{3+2\cdot ex}} \quad (4)$$

where:

Po : ice pressure (at 1 m^2) [Mpa] <class dependent>
ex : pressure-area exponent [no units, ex = -.1]
ϕ : ice edge opening angle [150 deg]
β' : normal frame angle (from vertical)
Δ_n : normalized mass (= Δ_{ship}/Co)
V_n : normalized velocity (= V_{ship} l) < V_{ship} is class dependent>
Co : mass reduction coefficient
l : x-direction cosine ($l = \sin(\alpha) \cos(\beta')$)

This can be expressed in simpler terms, with all hull angle terms collected into *fa* ;

$$F_n = fa \cdot Po^{.36} \cdot \Delta_{ship}^{.64} \cdot V_{ship}^{1.28} \quad (5)$$

where;

$$fa = lesser\_of \begin{cases} \left(0.097 - 0.068 \cdot \left(\frac{x}{L} - .15\right)^2\right) \frac{\alpha}{\sqrt{\beta'}} \\ \frac{1.2 \cdot \sigma_f \cdot h_{ice}^2}{\sin(\beta') \cdot Po^{.36} \cdot \Delta^{.64} \cdot V^{1.28}} \\ 0.6 \end{cases} \tag{6}$$

where:
x : distance from FP [m]
L: ship length
α : waterline angle

the force is limited by the flexural failure the ice force;

$$F_{n,\lim} = \frac{1}{\sin(\beta')} \cdot 1.2 \cdot \sigma_f \cdot h_{ice}^2 \tag{7}$$

where:

h_{ice} : ice thickness [m] <class dependent>
σ_f : ice flexural strength [Mpa] <class dependent>
β' : normal (true) frame angle

It can be seen that the ice crushing and bending mechanics are identical in the oblique and ramming models. The values of ice and ship parameters (velocity) underlying the class factors are also identical.

Comparisons between the ramming force (using equation (1)) and the oblique collision force (using equation (5)) at the stem were performed for a set of ships covering a range of sizes and hull forms. The values are plotted in Figures 5,6. There are two main differences between the ram and oblique collision. The ram equation assumes a flat (180°) ice edge, with beaching determining the maximum force. The oblique collision assumes a pointed (150°) ice edge with the maximum force occurring during the initial impact.

Generally the ramming and oblique collision analyses produce similar results, despite the differences in the formulations. This lends confidence to both solutions. For the conventional form the ramming forces are higher than the oblique collisions. This is to be expected due to the kinematics of the collision. In the oblique collision the conventional ship shape results in a lower effective mass. With the other shapes, the oblique collision force tends to be larger for the smallest vessels (5 kT). For the larger vessels, the two values are the same, because they are defined by flexural failure, which is the same for both cases. As long as the ram force is less than

2x the oblique force, there is no logical mismatch (for a ship with a conventional stem). A ram load would be divided on two sides of the stem, effectively giving two load patches.

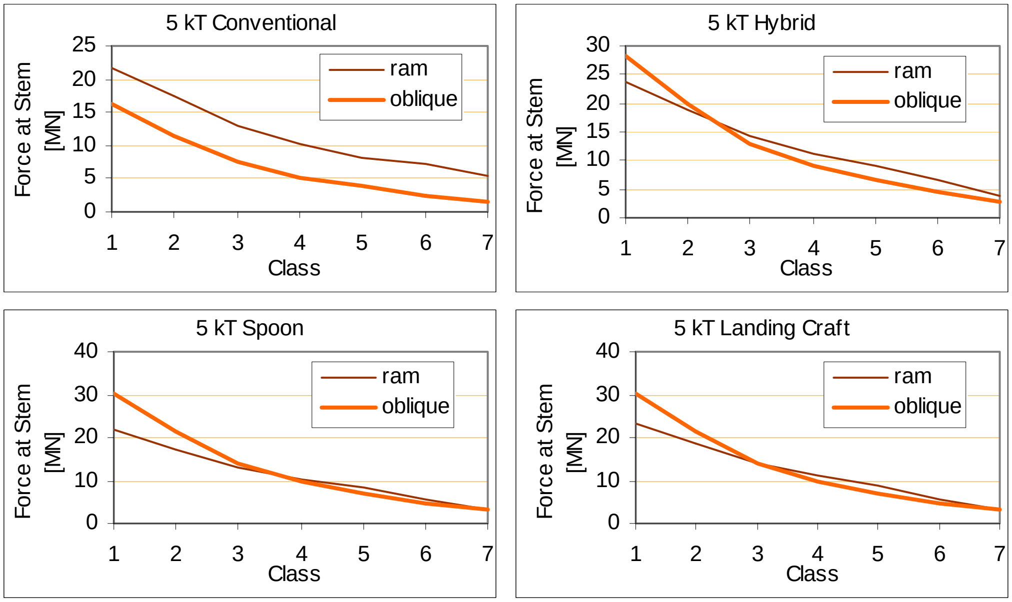

Figure 5. Force values for 5000 T ships.

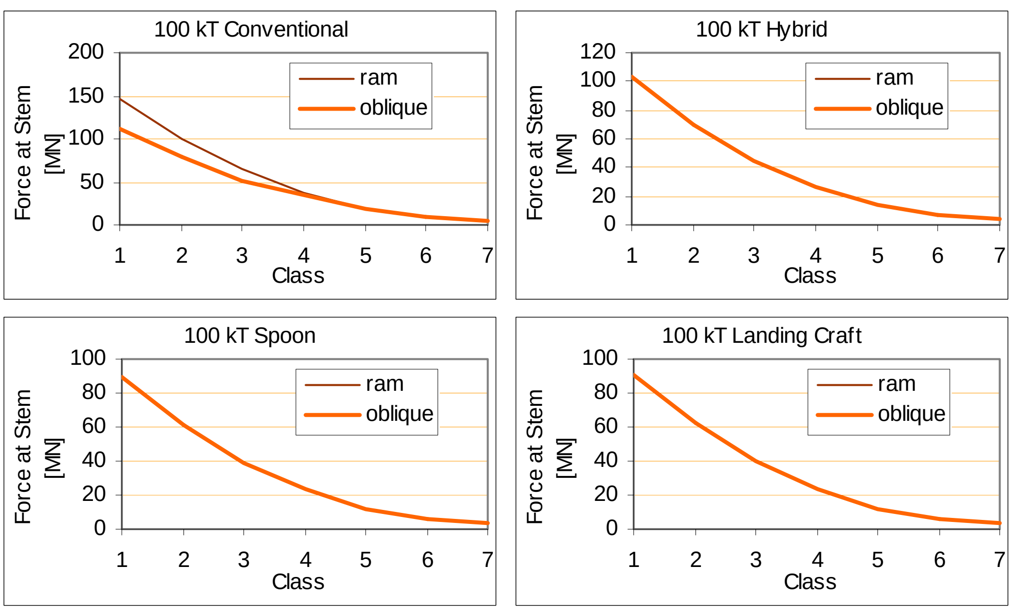

Figure 6. Force values for 100,000 T ships.

7. Comparison with open water requirements

Open water requirements

The wave bending moment and shear force values were calculated using the IACS Longitudinal Strength Standard (S11) [13], with still water values as recommended by DnV [14].

Two comparisons can be made. The wave bending moment (and shear force) can be compared to the ice bending moment (and shear force). As well, the required sections modulus, and shear area (using wave + still water, and ice + still water) can be compared to the as-built section modulus and shear areas. The sagging condition is the only one to be checked, as ice impact will only cause sag.

The maximum sagging wave bending moment *Mw* is;

$$Mw = -110 \cdot C \cdot L^2 \cdot B \cdot (Cb + 0.7) \cdot 10^{-6} \qquad \text{[MN-m]} \qquad (8)$$

where:

$C = 10.75 - \left[\frac{300 - L}{100}\right]^{1.5}$ for 90m <= L<= 300m

$= 10.75$ for 300m <= L<= 350m

$= 10.75 - \left[\frac{L - 350}{150}\right]^{1.5}$ for 350m <= L<= 500m

$= 0.044\,L + 3.75$ for 61m <= L<= 90m (ABS [15])

L: ship length [m]
B: moulded breadth [m]
Cb: block coefficient

The values along the vessel are those shown in Figure 3.

The still water sagging moment is estimated as [14];

$$Ms = -65 \cdot C \cdot L^2 \cdot B \cdot (Cb + 0.7) \cdot 10^{-6} \qquad \text{[MN-m]} \qquad (9)$$

This is 0.591 of the wave value. The distribution of still water bending along the hull is calculated according to the DnV rules [14]. (NOTE: the values used for still water bending are added to both ice and wave loads, and so precise values are unimportant in these comparisons)
The minimum section modulus (*S*) is to be;

$$S = \frac{Mw + Ms}{175} \qquad [\mathrm{m}^3] \qquad (10)$$

The values along the vessel are those shown in Figure 3.

The maximum shear force due to waves is ;

$$Fw = 30 \cdot C \cdot L \cdot B \cdot (Cb + 0.7) \cdot 10^{-5} \qquad [\mathrm{MN}] \qquad (11)$$

A still water value of *Fs = 0.591 Fw* is assumed. Further, a simple shear area (*SA*) criteria in keeping with the IACS approach is assumed;

$$SA = \frac{Fw + Fs}{110} \qquad [\mathrm{m}^2] \qquad (12)$$

These simple assumptions allow the comparison of ice, wave and as-built values for bending and shear.

Comparisons of rule requirements

The following plots compare the bending moments and shear forces for ice and waves. The ice values depend on the ice class.

Table 3. Vessel parameters examined in rule comparison.

Variable	value	description
L	100, 182, 247	Length of vessel [m]
B	14.3, 26.0, 35.3	Breadth of vessel [m]
T	5.7, 10.4, 14.1	Draft of vessel [m]
Cb	0.8	Block coefficient (summer wl)
Cwp	0.9	Waterplane Coefficient
Awp	1286, 4260, 7844	Waterplane Area in [m^2]
D	6.7, 40.3, 100.9	Displacement [kT]
Gama	30	stem angle (up fr. horiz.) [deg]
Bex	0.5	bow shape exponent
LB	20, 36.4, 49.4	bow length [m]
C_	0.5916	form parameter

The plots indicate that wave bending moments correspond to approximately ice class IPC3 for small ships, and above IPC1 for large ships (at midships). Shear values are harder to compare, because the pattern is so different. Peak wave shear values correspond roughly to IPC5 for small vessels, and IPC2 for large vessels.

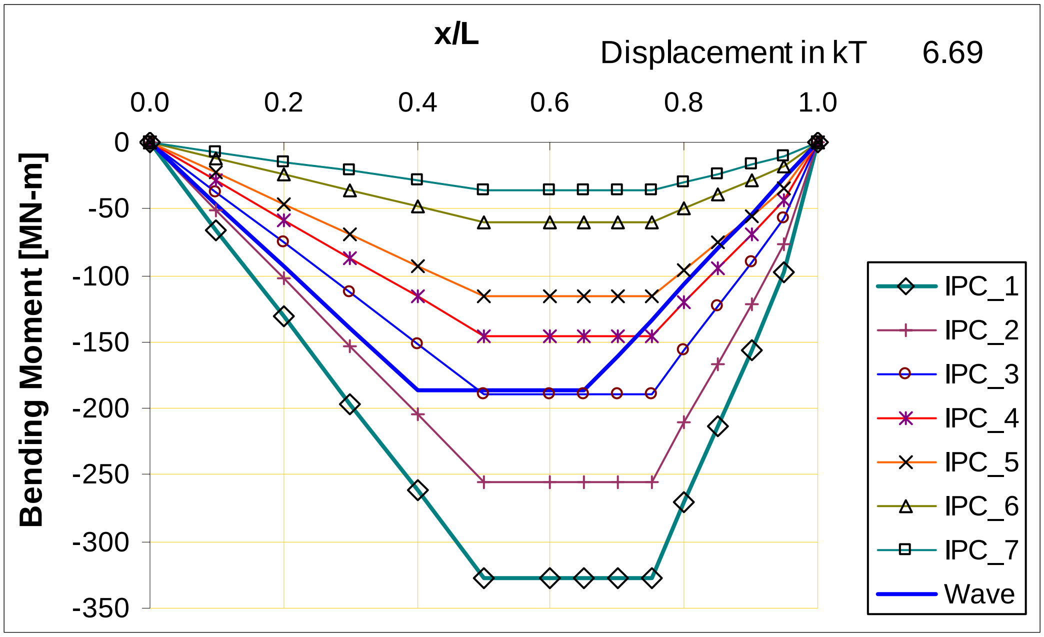

Figure 7. Wave vs. ice bending moments for 100 m vessel.

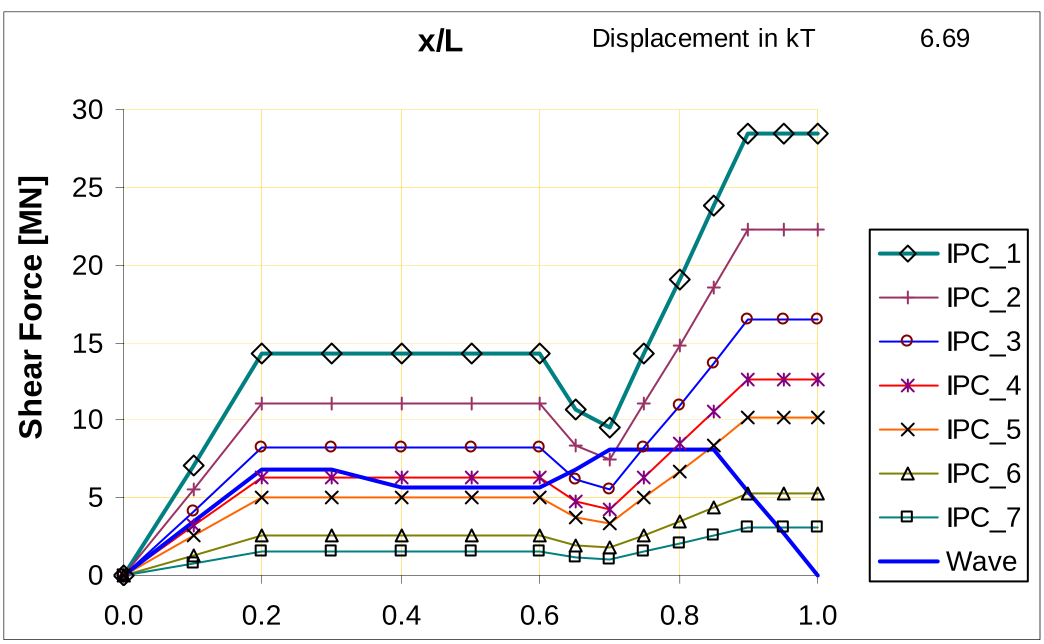

Figure 8. Wave vs. ice shear force for 100 m vessel.

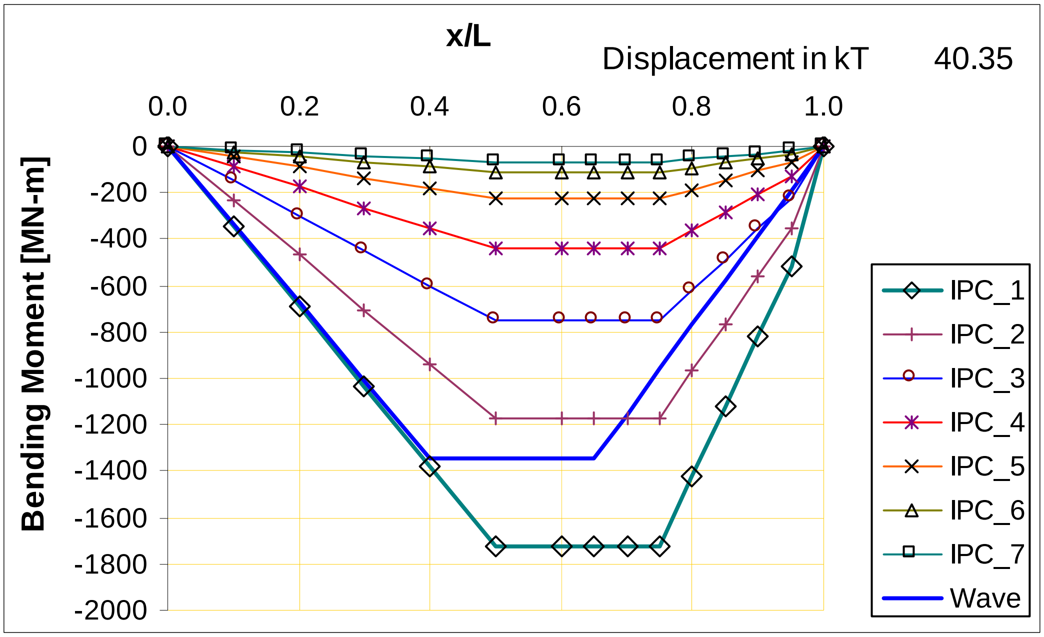

Figure 9. Wave vs. ice bending moments for 182 m vessel.

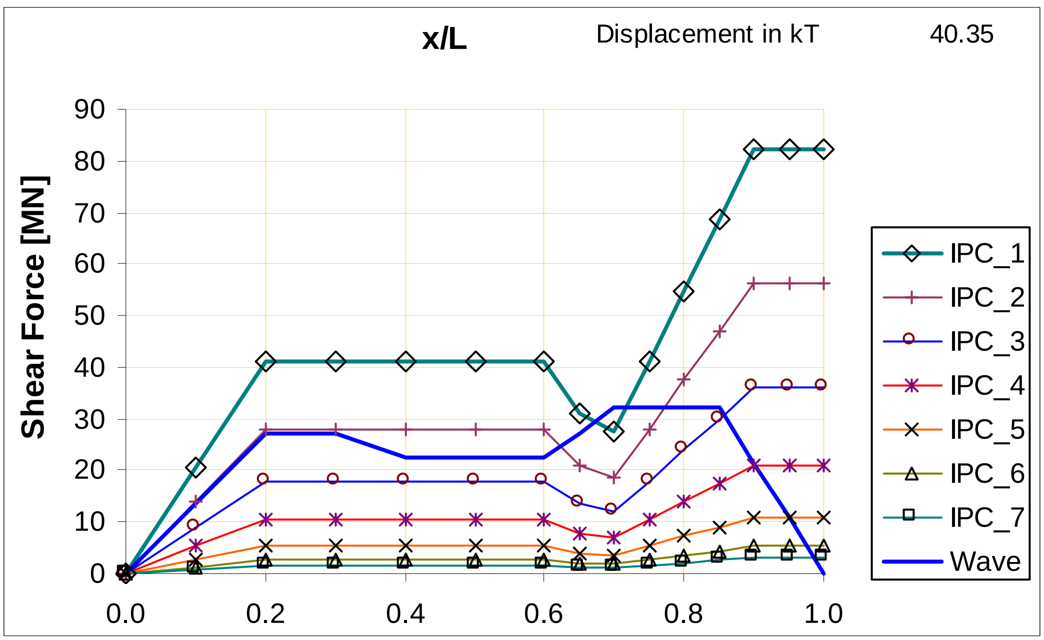

Figure 10. Wave vs. ice shear force for 182 m vessel.

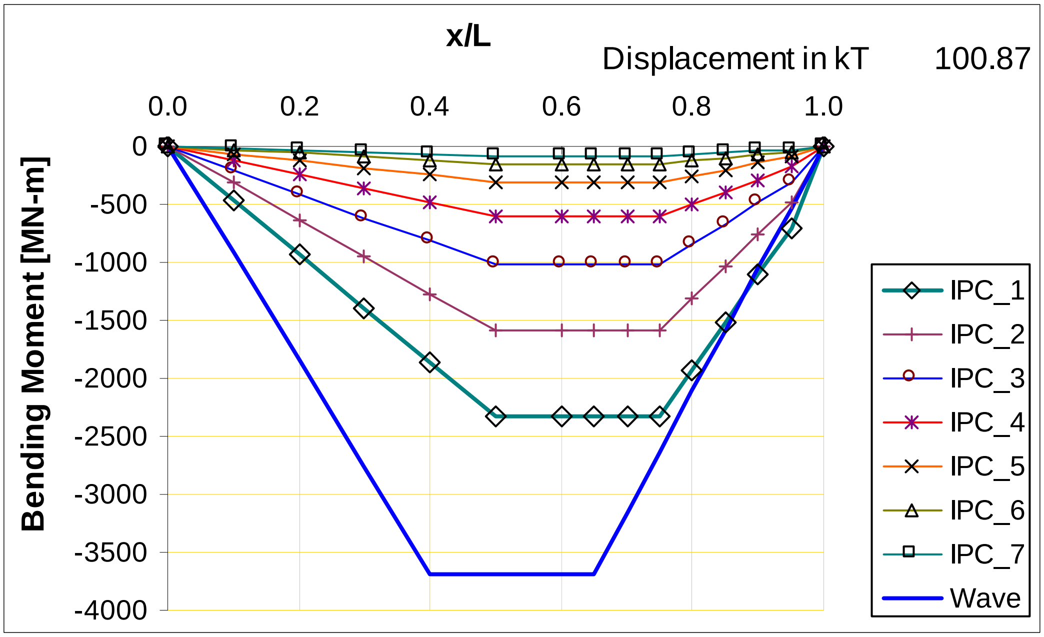

Figure 11. Wave vs. ice bending moments for 247 m vessel.

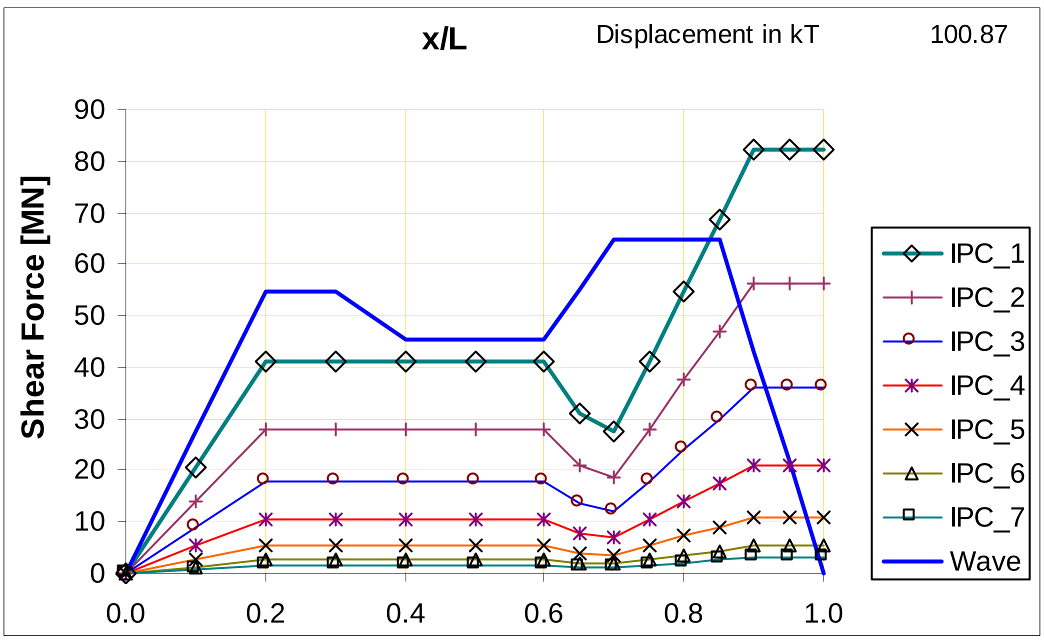

Figure 12. Wave vs. ice shear force for 247 m vessel.

Comparison with as-built values

Construction practices and practical constraints tend to result in scantlings that may be quite different from (above) the rule minimums as determined from overall strength requirements. In the bow, for example, local wave (or ice) loads will generally dominate . The impact of the proposed requirements can thus only be determined by comparing with as-built scantlings.

The first comparison is with an oil tanker 135.9 meters long, with a nominal Russian Register L1 ice class (pre-1999 rules. Note: the new 1999 Russian classes use a LU designation. The old L1 is very approximately equivalent to the new LU4, and generally considered similar to the F/S Baltic 1A class). This will translate to approximately IPC7. The vessel properties are given in Table 4. The bow shape is plotted in Figure 13. The wave and ice required section modulus values are compared with the as-built values in Figure 14. The wave and ice required shear area values are compared with the as-built values in Figure 15.

The comparisons show that the ice requirements for the lower classes are not higher than the as-built values and would not require any changes in the vessel.

The second comparison is with a bulk carrier 128.9 meters long, with a nominal F/S Baltic 1AS ice class. This will translate to approximately IPC6. The vessel properties are given in Table 5. Properties for vessel 2 - "Bulk Carrier". The bow shape is plotted in Figure 16.The wave and ice required section modulus values are compared with the as-built values in Figure 17. The wave and ice required shear area values are compared with the as-built values in Figure 18.

The comparisons show that the ice requirements for the lower classes are not higher than the as-built values and would not require any changes in the vessel.

Table 4. Properties for vessel 1 - "Oil Tanker"

Variable	Value	description
L	135.9	Length of vessel [m]
B	22.5	Breadth of vessel [m]
T	8.7	Draft of vessel [m]
Cb	0.777	Block coefficient (summer wl)
Cwp	0.9	Waterplane Coefficient
Awp	2753	Waterplane Area in m^2
D	21.19	Displacement in kT
Gama	30	stem angle (up fr. horiz.)
e_b	0.5	bow shape exponent
L_B	20	bow length in m
C_	0.4714	form parameter

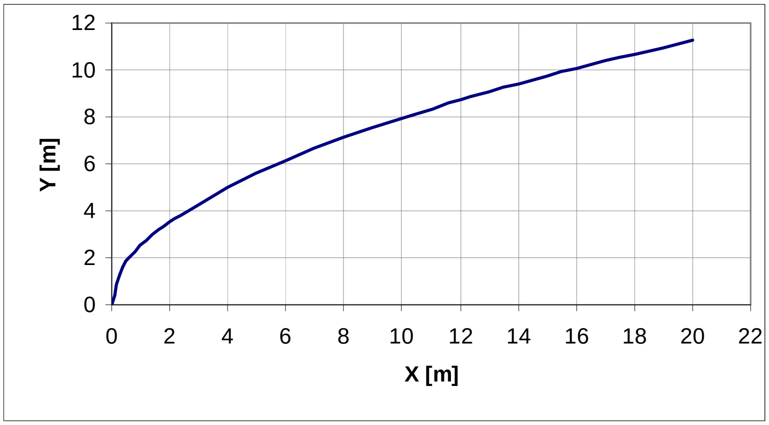

Figure 13 Bow waterline shape for Ship #1 (136m Oil Tanker).

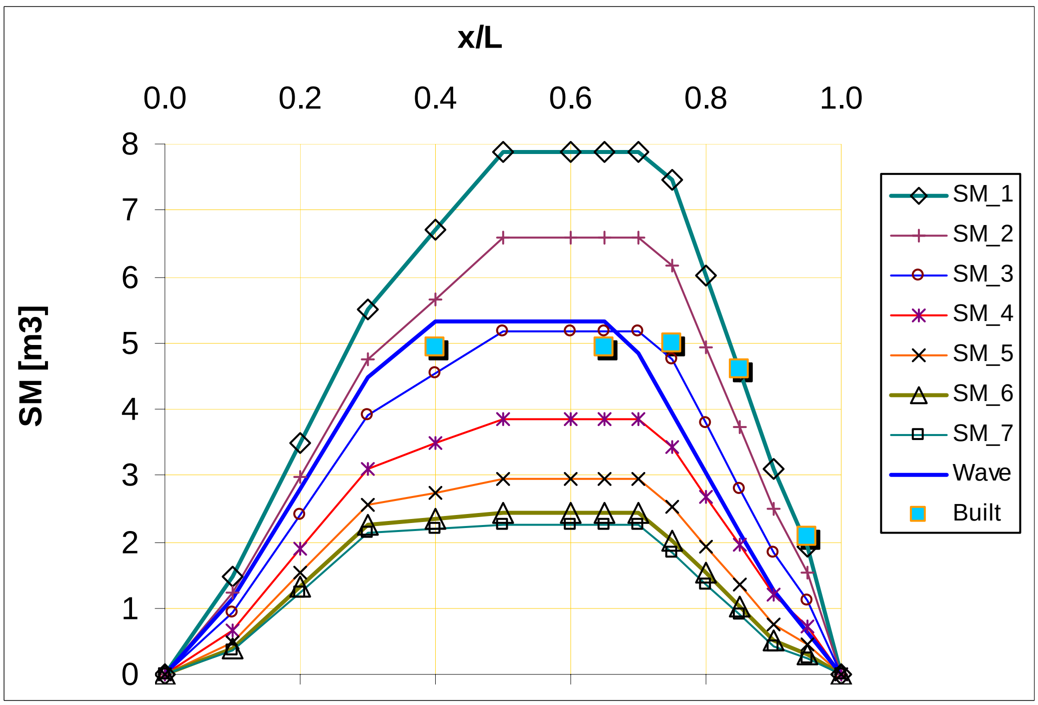

Figure 14. Wave and Ice Section modulus requirements compared to the As-built values for Ship #1 (136m Oil Tanker).

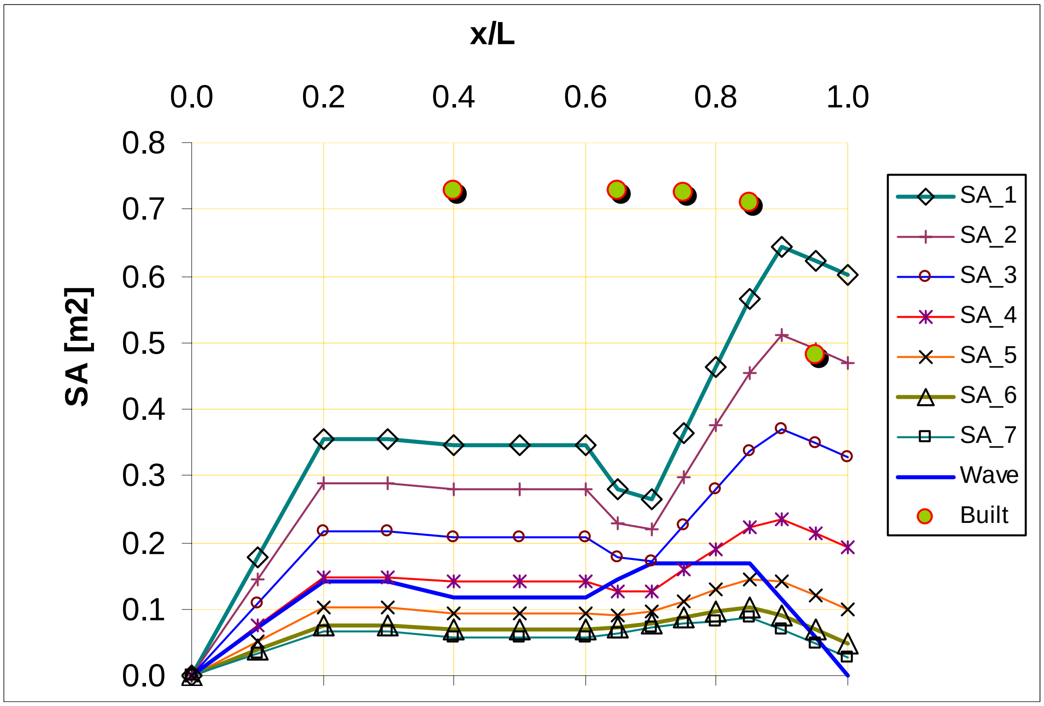

Figure 15. Wave and Ice Shear Area requirements compared to the As-built values for Ship #1 (136m Oil Tanker).

Table 5. Properties for vessel 2 - "Bulk Carrier"

Var	val	desc.
L	128.8	Length of vessel in m
B	21.6	Breadth of vessel in m
T	8.0	Draft of vessel in m
Cb	0.575	Block coefficient (summer wl)
Cwp	0.9	Waterplane Coefficient
Awp	2504	Waterplane Area in m^2
D	13.08	Displacement in kT
Gama	40	stem angle (up fr. horiz.)
Alfa	40	entrance angle
e_b	1	bow shape exponent
L_B	11.9	bow length in m
C_	0.27546	form parameter

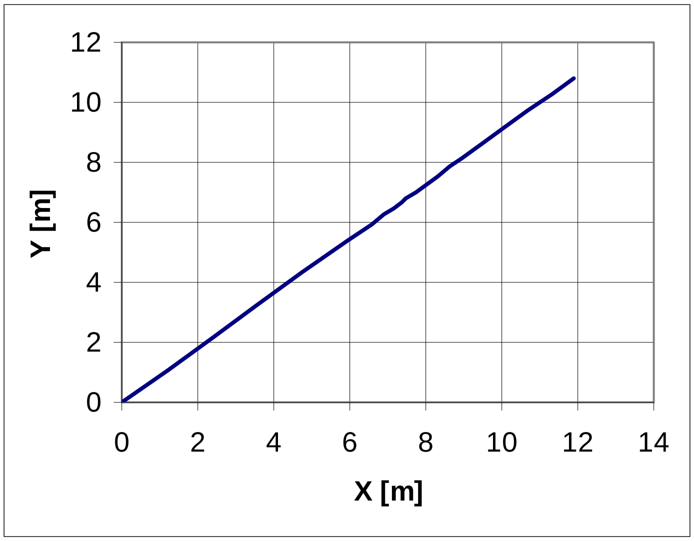

Figure 16 Bow waterline shape for Ship #2 (129m Bulk Carrier).

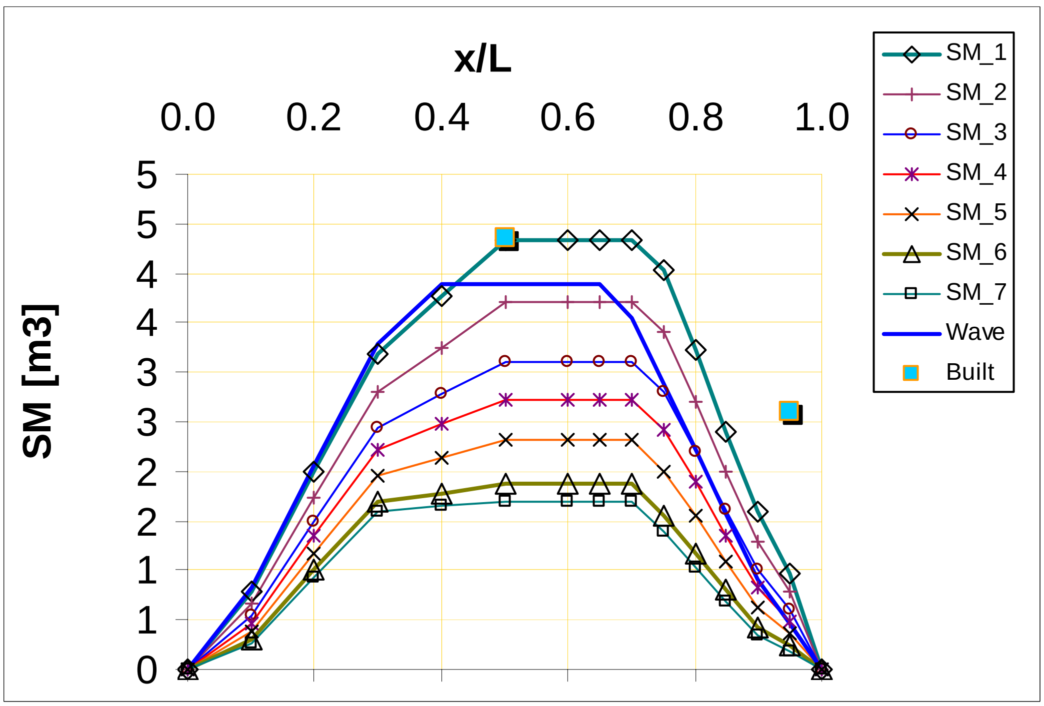

Figure 17. Wave and Ice Section modulus requirements compared to the As-built values for Ship #2 (129m Bulk Carrier).

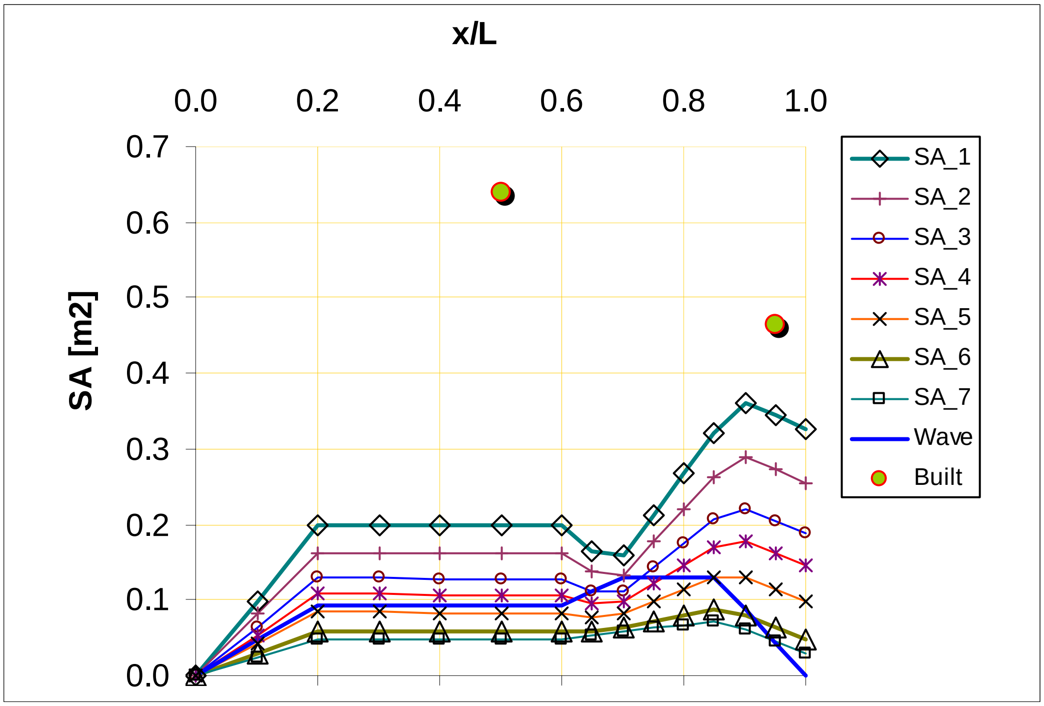

Figure 18. Wave and Ice Shear Area requirements compared to the As-built values for Ship #2 (129m Bulk Carrier).

8. Conclusion

The longitudinal strength requirements are based on the scenario of a ship ramming head-on into ice. The rules will ensure a minimum safe level of hull girder strength. The methodology employed can easily be applied to non-standard design conditions, if desired. The analysis has shown that the open water rules and normal construction practice will normally result in greater hull girder strength than is required for ice. Nevertheless, the requirements are needed to ensure that all ice class vessels will have an adequate level of hull girder strength.

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Annex A - Derivation of Ramming Force by Energy Method

To start we assume that initial kinetic energy is equal to the sum of ice indentation (crushing) energy and pitch/heave potential;

$$KE = PE + IE \tag{A1}$$

The kinetic energy is;

$$KE = 1/2\ M\ V^2 \tag{A2}$$

The potential energy, assuming linearity in heave and pitch is;

$$PE = 1/2\ Fv^2/Kb \tag{A3}$$

where Kb is the effective vertical stiffness at the bow;

$$Kb = \frac{\rho \cdot g \cdot Awp}{1 + \left(\frac{L}{2\lambda}\right)^2} \tag{A4}$$

where

λ is the radius of gyration of the waterplane (i.e. $Iwp = \lambda^2\ Awp$)

It is assumed that, for most ships;

$$Kb = \rho\ g\ Awp\ /5 \tag{A5}$$

This gives;

$$PE = 5/2\ Fv^2/\ (\rho\ g\ Awp) \tag{A6}$$

The ice indentation energy is found by integrating the force over the penetration depth, which can be done with normal or vertical force/distances (results are the same). The basic equation is;

$$IE = \int_0^{\zeta} Fv\ d\zeta \tag{A7}$$

where ζ is the vertical penetration of the ice edge (see Figure A1).

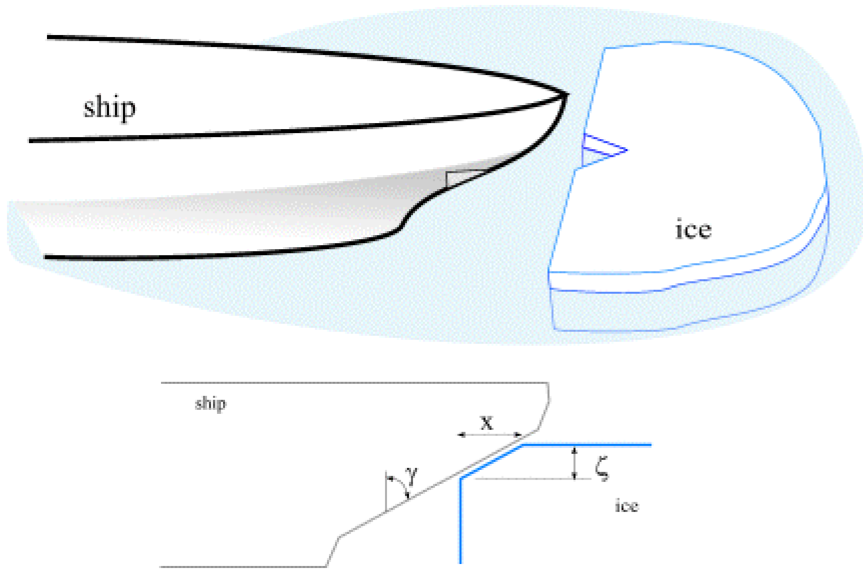


Figure A1. Ice indentation geometry for ramming collision.

The vertical force is a function of the vertical projected area, as follows;

$$Fv = Pav\ Av$$

$$= p_1\ Av^{ex}\ Av$$

$$= p_1\ Av^{1+ex} \qquad \text{(A8)}$$

The vertical projected area is a function of horizontal indentation x, and the bow shape. There are two bow shape cases that will be considered; 1) a simple wedged shaped bow and 2) a curved waterline bow described by a 'power' equation.

Case 1 - Wedge Bow

This derivation is as follows. The breadth at x is;

$$y(x) = tan(\alpha)\ \ x \qquad \text{(A9)}$$

The vertical area is thus;

$$Av(x) = 2\ \ \ 1/2\ \ y(x)\ x = tan(\alpha)\ \ x^2 \qquad \text{(A10)}$$

Hence the vertical force is;

$$Fv = p_1\ (tan(\alpha)\ \ x^2)^{1+ex} \qquad \text{(A11)}$$

Using the relationship;

$$x = \zeta \ tan(\gamma) \tag{A12}$$

we can write;

$$Fv = p_1 \ (\zeta^2 \ tan(\alpha) \ \ tan^2(\gamma) \ )^{(1+ex)} \tag{A13}$$

Using;

$$K_{ice} = p_1 \ (\ tan(\alpha) \ \ tan^2(\gamma) \ )^{(1+ex)} \tag{A14}$$

We can write;

$$Fv = K_{ice} \ \ \zeta^{(2+2ex)} \tag{A15}$$

From this we can do two things. We can invert this to express ζ as a function of *Fv* (which we will need later);

$$\zeta = (Fv/K_{ice})^{1/(2+2ex)} \tag{A16}$$

and we can integrate (A16) to get the indentation energy.

$$IE = K_{ice} \ \zeta^{(3+2ex)} \ /(3+2ex) \tag{A17}$$

Using our expression for ζ, we get;

$$IE = Kice^{-1/(2+2ex)} \ \ Fv^{(3+2ex)/(2+2ex)} \ /(3+2ex) \tag{A18}$$

Note that in the simple linear case with *ex*= -.5, the above equation reduces to
$IE = Fv^2/(2 \ K_{ice})$, as it should.

We can now write an equation that has *Fv* as the only unknown;

$$\frac{1}{2} M \cdot V^2 = \frac{5}{2} \frac{Fv^2}{\rho \cdot g \cdot Awp} + \frac{Fv^{\frac{(3+2ex)}{(2+2ex)}}}{(3+2ex) \cdot Kice^{\frac{1}{(2+2ex)}}} \tag{A19}$$

Unfortunately there is no general analytical solution for Fv in (A19), except for certain cases (e.g. the linear case in which ex = -.5, as solved below). We will use an empirical equation for Fv. First, However we will derive a similar solution for curved bows.

Case 2 - Curved Bow (Ramp or Spoon Form)

This derivation is as follows. The breadth at *x* is;

$$y(x) = B/2\ (x/L_B)^{e_b} \tag{A20}$$

which can be re-written as ;

$$y(x) = C\ B^{1-e_b}\ x^{e_b} \tag{A21}$$

Were $C=1/(\ 2\ (L_B/B)^{e_b})$

The vertical area is thus;

$$Av(x) = 2\int_0^x y(x)\,dx = 2\ C/(1+e_b)\ B^{1-e_b}\ x^{1+e_b} \tag{A22}$$

Hence the vertical force is;

$$Fv = p_1\ (2\ C/(1+e_b)\ B^{1-e_b}\ x^{1+e_b})^{1+ex} \tag{A23}$$

Using the relationship;

$$x = \zeta\ tan(\gamma) \tag{A24}$$

we can write;

$$Fv = p_1\ (2\ C\ B^{1-e_b}/(1+e_b))^{1+ex}\ (\zeta\ tan(\gamma))^{(1+e_b)(1+ex)} \tag{A25}$$

Using;

$$K_{ice} = p_1\ (2\ C\ B^{1-e_b}/(1+e_b))^{1+ex}\ (tan(\gamma))^{(1+e_b)(1+ex)} \tag{A26}$$

We can write;

$$Fv = K_{ice}\ \zeta^{(1+e_b)(1+ex)} \tag{A27}$$

From this we can do two things. We can invert this to express ζ as a function of *Fv* (which we will need later);

$$\zeta = (Fv/K_{ice})^{1/((1+e_b)(1+ex))} \tag{A28}$$

and we can integrate to get the indentation energy.

$$IE = Kice /((1+ e_b)(1+ex)+1) \ (\zeta^{(1+ e_b)(1+ex)+1}) \qquad \text{(A29)}$$

Using our expression for ζ, we get;

$$IE = Kice^{-1/((1+ e_b)(1+ex))} \ Fv^{1+1/((1+ e_b)(1+ex))} /((1+ e_b)(1+ex)+1) \qquad \text{(A30)}$$

Note that in the simple linear case with e_b = 1, ex= -.5, the above equation reduces to $IE = Fv^2/(2\ K_{ice})$, as it should.

We can now write an equation that has Fv as the only unknown;

$$\frac{1}{2} M \cdot V^2 = \frac{5}{2} \frac{Fv^2}{\rho \cdot g \cdot Awp} + \frac{Fv^{1+\frac{1}{(1+e_b)(1+ex)}}}{((1+e_b)(1+ex)+1) \cdot Kice^{\frac{1}{(1+e_b)(1+ex)}}} \qquad \text{(A31)}$$

Unfortunately, once again, there is no general analytical solution for Fv, except for certain cases. Equation (A31) can easily be solved numerically, but we only get specific values.

In the linear case the above equation simplifies to;

$$\frac{1}{2} M \cdot V^2 = \frac{5}{2} \frac{Fv^2}{\rho \cdot g \cdot Awp} + \frac{Fv^2}{2 \cdot Kice} \qquad \text{(A32)}$$

which is solved to give;

$$Fv = \sqrt{\frac{1}{5 + \frac{1}{\kappa}}} \cdot \sqrt{M} \cdot \sqrt{Kh} \cdot V \qquad \text{(A33)}$$

where;

kh = ρ g Awp

κ = Kice/Kh (in general)

= p_1 Const tan(φ)/(ρ g Awp) (in the linear case)

The above linear solution is quite similar to the analytical solution produced by Riska, with the main difference being the exact form of the κ term. κ can be thought of in several ways. It is a non-dimensional ice strength. It is also a ratio of stiffness values, i.e. the ratio of ice indentation stiffness to bow translation stiffness (heave/pitch stiffness). The balance between ride-up and crushing will be determined by this stiffness ratio, thus governing the nature of the impact and the maximum force.

To express Fv as a function of the ship and ice parameters we need to use an empirical equation. The equation is of the form derived by Riska [2], and modified in [4]. The only change is that κ is defined differently (with e_b).

$$Fv = .534\ \kappa^{0.15}\ \sin^{0.2}(\varphi)\ (\Delta\ Kh)^{0.5}\ V \qquad (A34)$$

where

κ = Kice/Kh dimensionless ice strength
Kice is a parameter that the defines the indentation stiffness of the ice as the stem of the ship penetrates the ice. The magnitude of Kice depends on both the ice strength and the shape of the bow. Two cases are given:

a) for the case of a blunt bow form

$$Kice = p_1\ (2\ C\ B^{1-e_b} / (1+e_b))^{(1+ex)}\ (\tan(\varphi_{stem}))^{-(1+e_b)(1+ex)}$$

b) for the case of wedge bow form e_b =1 and the above simplifies to

$$Kice = p_1\ (\tan\alpha_{stem} / \tan^2\varphi_{stem}))^{(1+ex)}$$ (wedge shaped bows α_{stem} <80deg)

Kh	=	ρ g Awp
p_1	=	ice pressure constant (Mpa at 1 m^2)
ex	=	ice pressure exponent (we use -0.1)
φ	=	stem angle (measured up from horizontal)
e_b	=	bow shape exponent (see Annex 2)
C	=	$1/ (2\ (L_B/ B)^{e_b})$
B	=	beam
L_B	=	bow length (used in bow form equation $y = B/2\ (x/L_B)^{e_b}$)
V	=	ship speed (m/s)
Δ	=	ship displacement (1000 t or Mkg)
ρ_w	=	water density (Mkg/m^3)
g	=	acceleration due to gravity
A_{wp}	=	ship waterplane area (m^2)

Equation (A34) is intended to be a closed form solution to equation (A31) (which can only be solved numerically). For verification, equation (A34) was solved for 168 cases, of which 126 were specific idealized hull forms of various sizes and classes (see Table), and 42 were for randomly generated hull forms covering a wide range of sizes and forms (see Figure A2). The results were compared to the numerical solution of equation (A31). The agreement, as shown in Figure A3, is excellent, over the full range of parameters.

Table A1. Regular Grid for the first 126 Cases (6 x 3 x 7).

Displacement [Kt]	5.52	18.63	51.13	78.24	108.67	204.60	
Bow form	Spoon e_b = .55	80/20 e_b = 1	Oden e_b =.05				
Velocity [m/s]	5.70	4.60	3.50	2.75	2.25	2.00	1.50

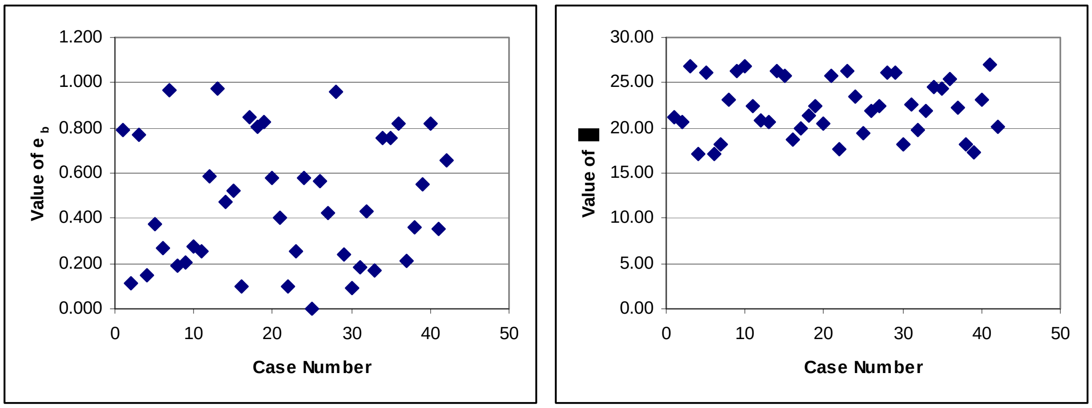
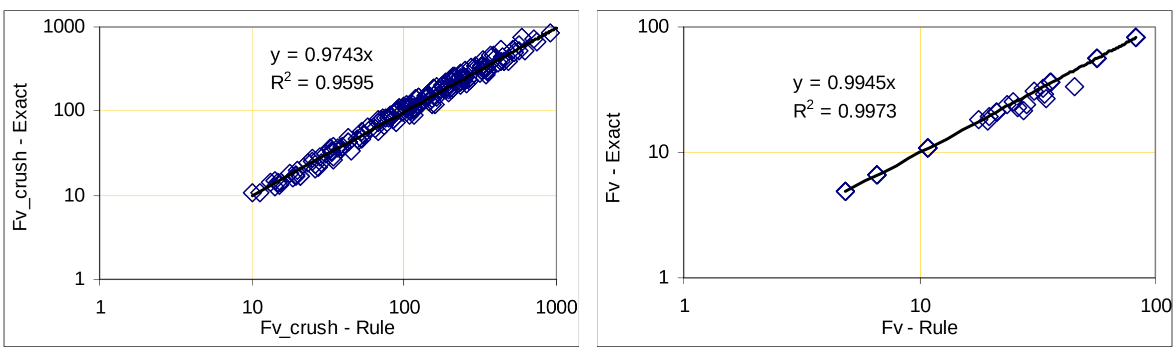


Figure A2. Random values of stem angle and e_b used for the second 42 cases.



(a) (b)

Figure A3. Fit between Exact Energy based solution (A31) and empirical equation (A34) for the crushing force (a) and the rule force (with flex limit) (b), for 168 cases.

Refer to the Excel workbook long_str3.xlw for the actual calculations.

Annex B - Description of Bow Shape Terms

The ice impact force during a head-on collision depends on the bow shape as well as ice strength, ship size and velocity. For the purpose of determining the load, the bow shape generally need to be idealized.. The ice interaction equation require that the contact area be expressed as algebraic functions of the indentation. To do this the bow form must be expressed as a single equation. It is not necessary to have the equation be valid for the whole bow. It is only necessary that the equation be valid for the stem region. Small errors in the shape will be insignificant in the force calculation. Consequently, two options for defining the bow form have been developed (i.e., the forces have been solved for two types of bow shape equations). As a result the Kice and K_f terms each have two possibilities, one for any simple wedge bow and one for any rounded (spoon) form.

Wedge Shaped Bow

Many conventional bows are shaped like a simple inclined wedge at the waterline. The idealization of this type of bow is very easy. The bow is assumed to be equivalent (for ramming load) to a simple inclined wedge with a waterline angle α equal to the waterline angle of the real bow at the FP. The only two terms that are needed in this case are α (waterline angle) and φ (stem angle). Figure B1 illustrates this case.

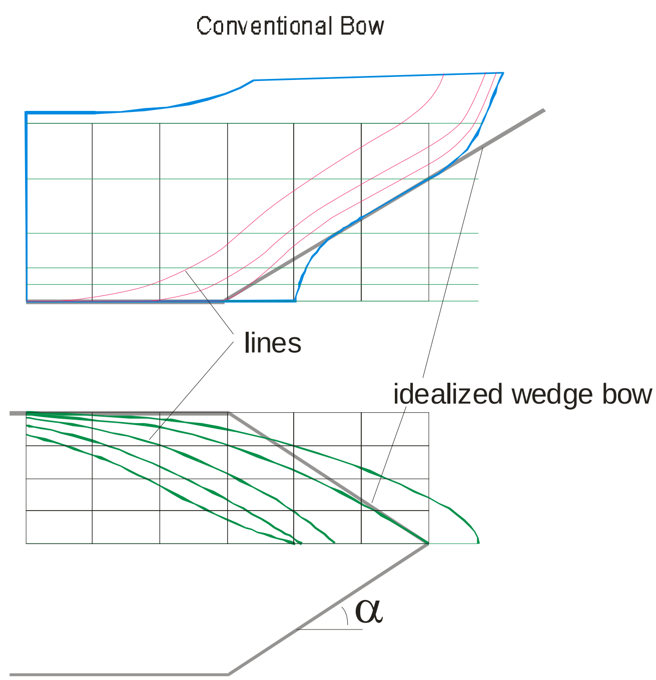


Figure B1. Lines and simple wedge idealization for a conventional bow form.

Spoon Shaped Bow

There are many bows that are spoon or ramp shaped, with a stem waterline angle α =90°. In this case the simple wedge idealization produces values that are too large. Instead, a simple function of the form;

$$y = B/2\ (x/L_B)^{e_b} \qquad \text{(B2)}$$

can be used to represent the bow shape. The e_b term can range $0 < e_b \leq 1$. When $e_b = 1$, the bow is a simple wedge. As eb $\rightarrow 0$, the bow becomes like a landing craft. Figure B2 illustrates the idealization for spoon form. Figure B3 (with stem to the left) shows how the shape changes with e_b. Table B1 gives a set of x and y coordinates of a hypothetical bow that is quite flat.The values in italics are found by fitting an equation of the form of (B2) to the hull x/y data. The fitted curve is plotted in Figure B4.

There are various ways of fitting a curve to data. The simplest way to fit Eqn (B2) to a bow form is to select two points on the bow and find e_b and L_B . If the two points are;

point 1 = $(x_1,y_1) = (x_1, c_1\ B/2)$,
point 2 = $(x_2,y_2) = (x_2,c_2\ B/2)$

the shape parameters are;

$$e_b = \ln(c_2/c_1)/\ln(x_2/x_1) \qquad \text{(B3)}$$

$$L_B = x_2\ c_2^{(-1/e_b)} \qquad \text{(B4)}$$

It should be kept in mind that the indentation will be confined to a few meters or less. The waterline shape approximation need only fit the first few meters.

Recalling that the bow local loads require the determination of hull angles at several stations in the bow, it is reasonable to select two points at the same locations as used in the bow calculations, normally .05L and .1 L aft of the stem. These would be points 1 and 2, from which e_b and L_B could be calculated. It should again be noted that L_B needs to be large enough to cover the area of indentation, but does not normally need to extend out to the maximum beam of the ship.

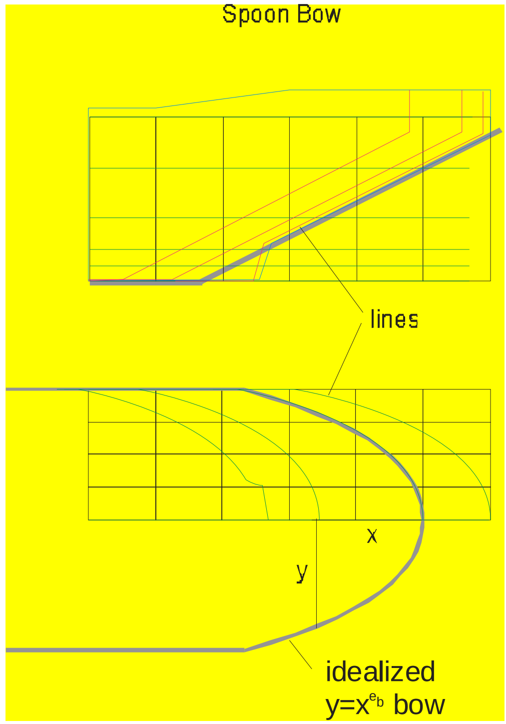


Figure B2. Lines and simple idealization for a spoon bow form.

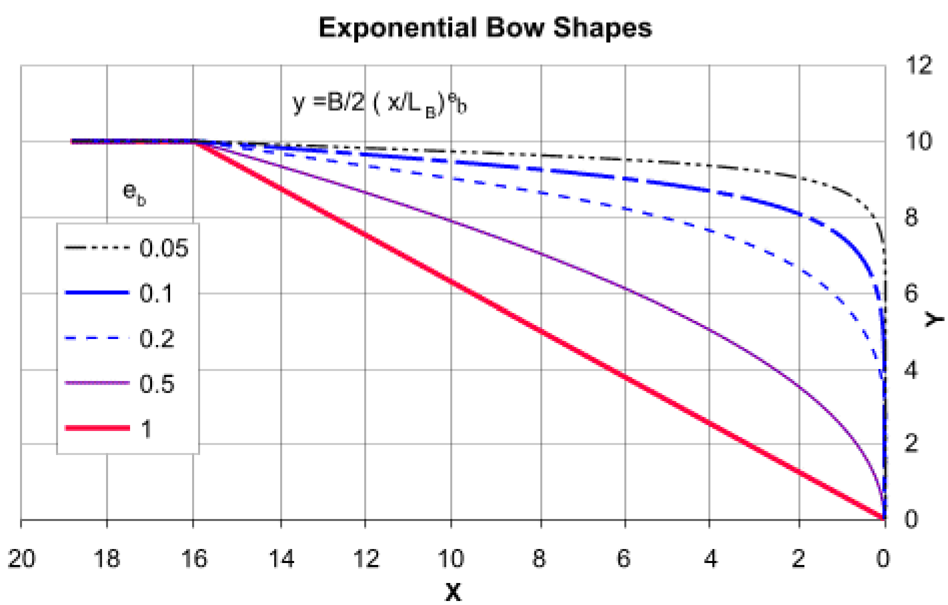


Figure B3. Bow shapes for various values of e_b. For this case B = 20, L_B = 16

Table B1. Example of a set of hull form coordinates and a fitted equation.

x	y_hull	*y_fit*
0.01	0.1	*1.91*
0.25	4	*3.87*
1.56	5.86	*5.79*
3.24	7	*6.80*
6	8	*7.78*
8	8.1	*8.29*
	e_b	*0.22*
	LB	*6.8*
	B	*8*

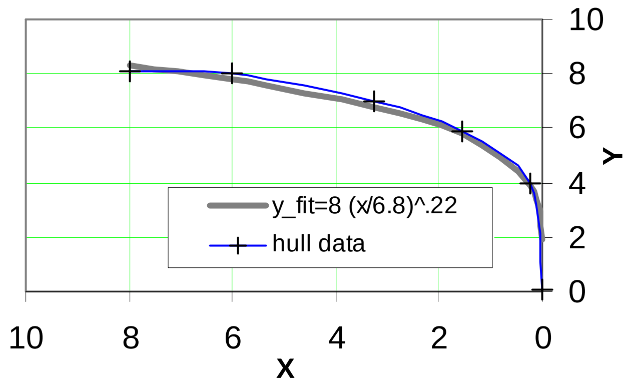


Figure B4. Example of a set of hull form coordinates and a fitted equation.