UNDERSTANDING THE EFFECT OF ASSUMPTIONS ON SHELL PLATE THICKNESS FOR ARCTIC SHIPS

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ABSTRACT
The presence of ice requires specially constructed ships. The biggest challenge in this aspect is ice loads on ship's hull: processes of deformation and fracture of ice are not fully understood and ice geometry and mechanical characteristics can vary dramatically over a few meters. Regardless of which ice-strengthening requirements that are chosen as the basis for hull structural design, engineers make assumptions about the ship-ice interaction scenario, but ship-ice interaction scenarios outside rules requirements can be fatal for both operators and the environment. Concerning safe design, there is a need for better understanding the ice loads in conjunction with the models for assessing capacity of structural elements. This paper contributes to highlighting the current differences between guidelines of the International Association of Classification Societies for Polar Ships (IACS, 2011) and of the Russian Maritime Register of Shipping (RMRS, 2014) for ice loads and plate thickness requirements. The goal of this study is to summarize the assumptions that underlie design formulations and ice class factors in order to deepen our understanding of the rule formulae, their background and limitations.

1. INTRODUCTION
Forecasts and trends have indicated an increase in marine operations in polar waters. One of the ship-design challenges is proper strengthening of the hull, for which ice loads represent the dominant load. Details of structural requirements are based on a combination of experience, empirical data and structural analysis. Engineers make assumptions about the ice-structure interaction process, ice properties and its geometry, regardless of which ice-strengthening requirements are chosen as the basis for hull design. Concerning safety of ships, it is crucial to be aware of the assumptions that underlie formulae of each rule-set and to understand how those assumptions impact the design.

This paper contributes to highlighting the current differences between guidelines of the International Association of Classification Societies for Polar Ships (IACS, 2011) and of the Russian Maritime Register of Shipping (RMRS, 2014) for ice loads and plate thickness requirements. The goal of this study is to summarize basic assumptions that underlie design formulae and to discuss the effect of most important assumptions on shell plate requirements. This paper does not address the important Finish-Swedish Ice Class Rules (FSICR) and the discussion refers only to deterministic design formulations. Riska and Kämäräinen (2011) performed a comprehensive overview of the principles that underlie the FSICR. A probabilistic approach to design was presented by Kaldasaun and Kujala (2011) for ships that navigate in the Baltic Sea and by Ralph and Jordaan (2013) for Arctic ships; interested readers are referred to those papers for details.

2. DESCRIPTION OF ICE LOADS

In this section, the main equations are presented for ice-load calculations of IACS (Sec. I2.3) and RMRS (Vol. 1 Pt. 2 Sec. 3.10). The description is limited to ice loads in the bow region, where the highest loads are expected.

2.1 IACS ice loads

IACS' Unified Requirements for Polar Ships have been widely accepted by the industry as a design standard for vessels operating in polar waters. The design scenario is an oblique collision with a large ice floe having a 150° front angle (Figure 1). Ice loads on ship's hull are characterized by an average pressure (*p*) that is uniformly distributed over a rectangular load patch of height (*b*) and width (*w*) and depend on ice category, hull angles and ship displacement.

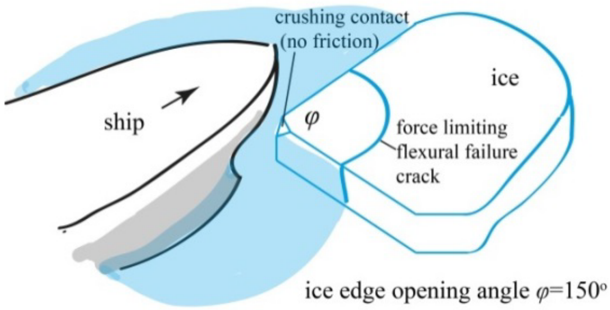


Figure 1. IACS' design scenario (Daley, 2000).

The waterline length of the bow is divided into several sub-regions. For each sub-region, the force, load patch aspect ratio, line load and pressure are calculated with respect to the mid-length position of each region. The width of the design load patch is determined by dividing the maximum force by the maximum line load. The height is determined by dividing the maximum line load by the maximum pressure. The average pressure within the design load patch is calculated by dividing the maximum force by the load-patch area. For a single sub-region in the bow area the ice loads read:

$$p = fa^{0.22} \cdot CF_C^{0.22} \cdot CF_D^{2} \cdot AR^{0.3} \cdot \Delta^{0.14}\text{, [MPa]} \tag{1a}$$

$$b = fa^{0.39} \cdot CF_C^{0.39} \cdot CF_D^{-1} \cdot AR^{-0.65} \cdot \Delta^{0.25}\text{, [m]} \tag{1b}$$

$$w = fa^{0.39} \cdot CF_C^{0.39} \cdot CF_D^{-1} \cdot AR^{0.35} \cdot \Delta^{0.25}\text{. [m]} \tag{1c}$$

Here CF_C is the crushing force class factor; CF_D is the load patch dimensions class factor; Δ is the ship displacement [kilotons]; $AR = 7.46 \cdot \sin(\beta) \geq 1.3$ is the load patch aspect ratio and *fa* is the bow shape coefficient, which is to be taken as:

$$fa = \min \begin{Bmatrix} (0.097 - 0.68 \cdot (x/L - 0.15)^2) \cdot \dfrac{\alpha}{\sqrt{\beta}} \\ \dfrac{1.2 \cdot CF_F}{\sin(\beta) \cdot CF_C \cdot \Delta^{0.64}} \\ 0.6 \end{Bmatrix}, \tag{2}$$

where CF_F is the flexural failure class factor; α is the waterline angle; β is the normal frame angle; L is the ship length and x is the distance from the forward perpendicular to the section under consideration.

The ice pressure is multiplied by a dimensionless pressure factor (PPF) to account for the peakedness character of pressure profiles. For example, $PPF = (1.8–s) \geq 1.2$ (s is the frame spacing in meters) for transversely framed plating.

2.2 RMRS ice loads

Russia is considered to be the most experienced nation with respect to ship operations in ice and low temperatures (Barents 2020, 2009). However, the scientific basis for RMRS' rules is often criticized outside of Russia (Riska, 2011). It is believed (IMO, 2014) that the model behind the ice loads has not been described in the publicly available literature. This section presents basic assumptions that underlie rule formulae.

The RMRS ice loads are characterized by maximum pressure (p) in the zone of hull-ice contact with maximum height (b) and maximum width (w). In the bow region, the pressure and corresponding load patch dimensions are determined as follows:

$$p = 2500 \cdot a_1 \cdot v_m \cdot \left(\frac{\Delta}{1000}\right)^{0.17}, \text{ [kPa]} \tag{3a}$$

$$b = C_1 \cdot u_m \cdot \left(\frac{\Delta}{1000}\right)^{0.33}, \text{ [m] and } \left(\frac{\Delta}{1000}\right)^{0.33} \leq 3.5 \tag{3b}$$

$$w = 11.3 \cdot \sqrt{b_i \cdot \sin(\beta_m)}\,,\ w > 3.5 \cdot \left(\frac{\Delta}{1000}\right)^{0.165}. \text{ [m]} \tag{3c}$$

Here Δ is the ship displacement [tons], a_1 and C_1 are factors depending on ice category; v_m and u_m are the maximum values of the shape factors v and u for the region; β_m is the frame angle in the section for which the u parameter reaches its maximum, and v and u are hull shape factors determined by the formulae:

$$v = (0.278 + 0.18\frac{x}{L})\sqrt[4]{\frac{\alpha^2}{\beta}} \qquad \frac{x}{L} \leq 0.25, \tag{4}$$

$$v = (0.343 - 0.08\frac{x}{L})\sqrt[4]{\frac{\alpha^2}{\beta}} \qquad \frac{x}{L} > 0.25, \tag{5}$$

$$u = k_B(0.635 + 0.61\frac{x}{L})\sqrt{\frac{\alpha}{\beta}} \qquad \frac{x}{L} \leq 0.25, \tag{6}$$

$$u = k_B(0.862 - 0.30\frac{x}{L})\sqrt{\frac{\alpha}{\beta}} \qquad \frac{x}{L} > 0.25. \tag{7}$$

Here, x, L and α are defined as in Section 2.1, $k_B = 1.0$ for $\beta \geq 7°$ and $k_B = 1.15–0.15 \cdot \beta/7$ for $\beta < 7°$. Equations 3a–3c are valid for ice going ships that have a standard hull form in accordance with requirements specified in RMRS' Sections 3.10.1.2.2 and 3.10.1.2.3.

3. SHELL PLATING REQUIREMENTS

The required minimum plate thickness is given by structural resistance and corrosion allowance. The structural resistance of the plating to the prescribed ice actions is assessed by means of rigid-plastic analysis. Analytical models adopted by the IACS and RMRS rules have been described in Daley et al. (2001) and Appolonov (2003), respectively. Principles that underlie the IACS and RMRS formulations are almost the same, and the two models produce very similar results. The models are based on a yield-line mechanism of a partially loaded rectangular plate, where the load patch has dimensions of width (*s*) and height (*b*).

For the transversely framed plating in the bow area, the shell thickness requirement is written as:

$$\text{in IACS: } t = 500 s \sqrt{\frac{PPF \cdot p}{\sigma}} \cdot \frac{1}{1+0.5\dfrac{s}{b}} + t_k \text{, [mm]} \tag{8}$$

$$\text{in RMRS: } t = 15.8 s \sqrt{\frac{p}{\sigma}} \cdot \frac{1}{1+0.5\dfrac{s}{b}} + t_k \text{, [mm]} \tag{9}$$

where p is the design ice pressure, b is the design load height, s is the frame spacing, σ is the upper yield stress [MPa], and t_k is the corrosion and abrasion allowance (see IACS I2.11 and RMRS Vol. 1, Pt. 2, Sec. 3.10.4.1). For a plate loaded over the entire length, b is the girder spacing.

The first term of Eq. 8 and 9 is a plate strip solution, and the second term $1/(1+0.5s/b)$ represents plate aspect ratio effects. Note a difference between Eq. 8 and Eq. 9, i.e., the leading coefficients of Eq. 8 and Eq. 9 are 500 and 15.8, respectively. This difference arises from a conversion of units because in the IACS recommendations, the pressure (p) is given in MPa whereas it is in kPa in the RMRS rules.

4. ASSUMPTIONS AFFECTING SHELL PLATE THICKNESS

Factors governing the shell plate thickness for ice-going ships can be divided in two groups, namely the assumptions that underlie models for calculating ice loads and the assumptions that underlie models for calculating structural resistance. The latter assumptions are typically better known and understood; see e.g., an evaluation of the IACS shell plate formulations by Nyseth and Holtsmak (2006). A discussion about the methodology and assumptions that underlie shell plate requirements has been presented in Hong and Amdahl (2007).

The assumptions behind ice load formulae are difficult to find, especially those underlying the RMRS rules. In the first part of this section, the assumptions behind design formulae are summarized. The second part discusses the most influential assumptions and their effect on the plate thickness.

4.1 An overview of assumptions that underlie the IACS and RMRS ice loads

An understanding of how ice behaves during ship-ice collisions forms the basis for design. The assumptions behind Eq. 1a–1c and Eq. 3a–3c are summarized below. A detailed derivation of the IACS loads can be found in Daley (2000), whereas a complete derivation of the RMRS formulae is unfortunately unavailable in the public literature. In lieu of this, the overview is based on the following literature:

1) The description of a methodology for ice strengthening requirements for ice going vessels in Appolonov et al. (1996).
2) The solution by Kurdyumov and Kheisin (1974) and Kurdyumov et al. (1980) for an impact against a large ice floe with a rounded ice edge.
3) The solution by Kurdyumov and Kheisin (1976) for an impact between a spherically shaped indenter an ice wall.

4) The description of modifications proposed for the Kurdyumov and Kheisin hydrodynamic model (HDM) of ice crushing (Appolonov et al., 2002).

IACS considerations

Equations 1a–1c are only valid for vessels with icebreaking forms. The method assumes that a ship-ice collision has a short duration, such that the six-degrees-of-freedom (6 DOF) problem can be modeled by an equivalent 1-DOF model in which all motions between the ship and the ice are normal to the ship's side at the collision point. Frictional forces are disregarded. Ice *crushing* and *bending failure* modes are considered. The following assumptions are made to derive the oblique collision force due to ice crushing:

1) The ship is considered to be rigid, only the ice is deforming by crushing.
2) The maximum collision force is assumed to take place at the end of the interaction process. The force is determined by equating the kinetic energy of the ship to the crushing energy of the ice. The latter consideration is only valid for an infinite ice mass.
3) The crushing energy of the ice is determined by integrating the ice crushing force over the penetration depth. The crushing force is calculated by considering a uniform pressure distribution over the nominal contact area (A).
4) The average pressure (P) is found from a Sanderson-type pressure-area relationship $P=P_o \cdot A^{ex}$ (Sanderson, 1988), in which the exponent ex is always –0.1, and P_o varies between 1.25 to 6.0 MPa·m$^{0.2}$ depending on the ice class.

Furthermore, it is argued that the ice crushing force (and thus the average pressure) cannot exceed the force (average pressure) required to cause the ice to fail because of bending; hence, the values of p, b and w are restricted to be less than some fixed values. The ice pressure (Eq. 1a) and corresponding dimensions of the load patch (Eq. 1b and 1c) are determined by transforming the apparent (nominal) contact area into a 'design' load patch. This is done as follows.

1) The ice pressure is determined by dividing the collision force by the corresponding contact area, that is $A_{red} = w \times b$, where w and b are the width and height of the 'design' load patch, respectively.
2) To calculate A_{red}, a change in load patch shape from triangular to rectangular is assumed; see "Equivalent load patch" in Figure 2. A force and aspect ratio $AR = W_{nom}/H_{nom}$ are kept constant; W_{nom} and H_{nom} are the width and height of the triangular load patch, respectively. Furthermore, a reduction in size of the load patch is introduced, while maintaining the constant force and aspect ratio. This reduction accounts for a typical concentration of force that takes place as ice edges spall off. The load width for each sub-region is $w = W^{wex}$, where wex is always 0.7 and the height $b = w/AR$. The nominal- and the design load patches have the same aspect ratio but the design area is always smaller than the nominal area (see Figure 2).

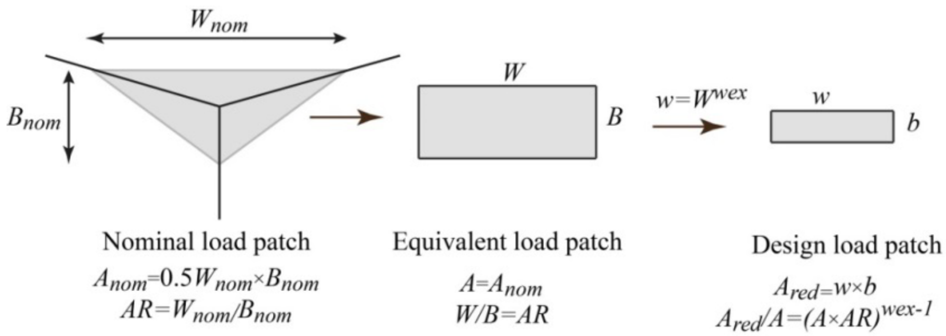


Figure 2. Nominal, equivalent and design load patches.

With the above assumptions, Eq. 1a –1c can be rewritten as:

$$w = F_n^{\frac{wex}{2+2ex}} \cdot P_o^{\frac{-wex}{2+2ex}} \cdot \left(2\tan(\varphi/2)\sin(\beta)\right)^{\frac{wex}{2}}, \text{ [m]} \tag{10a}$$

$$b = w \cdot \left(2\tan(\varphi/2)\sin(\beta)\right)^{-1}, \text{ [m]} \tag{10b}$$

$$p = F_n / (w \cdot b), \text{ [MPa]} \tag{10c}$$

where

$$F_n = \min\begin{cases} \left(3+2ex\right)^{\frac{2+2ex}{3+2ex}} \cdot P_o^{\frac{1}{3+2ex}} \cdot \left(\dfrac{\tan(\varphi/2)}{\sin(\beta)\cdot\cos^2(\beta)}\right)^{\frac{1+ex}{3+2ex}} \cdot \left(0.5\Delta_n V_n^2\right)^{\frac{2+2ex}{3+2ex}} \\ \dfrac{1}{\sin(\beta)} \cdot 1.2 \cdot \sigma_f \cdot H^2 \end{cases}; \tag{11}$$

Δ_n and V_n are the normalized mass (or vessel displacement) and the normalized ship's speed to account for the orientation of the collision. Table 2 presents functional dependencies of the ice design pressure and the load height that are expressed via ice properties and ship characteristics.

It is noticed that the IACS approach assumes an ice shape, i.e., ice edge opening angle (φ) of 150° (Figure 1). The ship speed at the moment of impact (V), the ice thickness (H) and ice strength parameters (P_o and ice flexural strength, σ_f) are assumed to be class dependent. Each class factor (CF_C, CF_D and CF_F) is developed from these values (Daley, 2000). It is also noticed that IACS assigns characteristic values to the exponent *ex* in the Sanderson pressure-area relationship and the ice spalling parameter *wex* (i.e., *ex* is always –0.1 and *wex* is 0.7). Justification of the selected values is not provided in the open literature. Notably, the exponent *wex* is crucial with respect to the design pressures.

RMRS considerations

Although Appolonov et al. (1996) state that the exponents for vessel displacement in Eq. 3a–3c are like those of HDM in Kurdyumov and Kheisin (1976), a full interpretation of design ice loads and coefficients in Eq. 3a–3c could not be performed. According to the solution in Kurdyumov and Kheisin (1976), the exponent for the vessel displacement should be 0.11 and not 0.17 as in the RMRS rules. Moreover, the scenario considered by Kurdyumov and Kheisin (1976) (i.e., an impact between a rigid spherically-shaped indenter and an ice wall) is different from the RMRS design scenario (i.e., an oblique collision with a rounded ice edge). This inconsistency led us to the conclusion that the reference by Appolonov et al. (1996) to the work of Kurdyumov and Kheisin (1976) is incorrect. Instead, one should refer to the earlier work of Kurdyumov and Kheisin (1974), where ice loads are addressed for a ship's hull collision with a large floating ice floe.

This section presents the main assumptions made by Kurdyumov and Kheisin (1974) and Kurdyumov et al. (1980).

Similar to the IACS approach, the method assumes that the ship-ice collision is so quick that the 6 DOF problem can be modeled by an equivalent 1 DOF model. Ice crushing and bending failure modes are considered. Frictional forces between the ship side and the crushed ice are disregarded. To determine the ice crushing pressure the following factors are considered:

1) Analytical results should meet the following requirements: calculated impact durations correspond to those measured in the field, and shape of the force-time history curve should resemble that measured experimentally.
2) In contrast to the IACS approach, a non-uniform pressure distribution over the nominal contact area is considered (Figure 3). This pressure distribution is calculated by assuming a thin intermediate layer of crushed ice between the hull and the solid (undamaged) ice. Simplified Navier-Stokes equations are used to describe the behavior, i.e., the motions of the layer are treated as an incompressible, homogeneous viscous flow having viscosity μ.
3) The pressure (P) is directly proportional to the thickness of the intermediate layer (h), i.e., $P = k_p \cdot h^n$, where k_p is the proportionality constant and the exponent $n = 1.0$.
4) Body forces and the variation of pressure across the fluid film are neglected.
5) The pressure varies over the contact area, with a maximum at the center of the contact area. The pressure is a function of the ice crushing speed.
6) The ice crushing speed is a function of the penetration distance and is determined by equating the change in kinetic energy of the ship to the ice-crushing energy. The latter is determined by integrating the ice force over the penetration distance.
7) The ice edge may fail in bending or crushing. Minimum force required to break the ice is considered as a design force.
8) To account for ice-edge spalling effects, the height of the nominal contact area is reduced by a factor of 0.94.

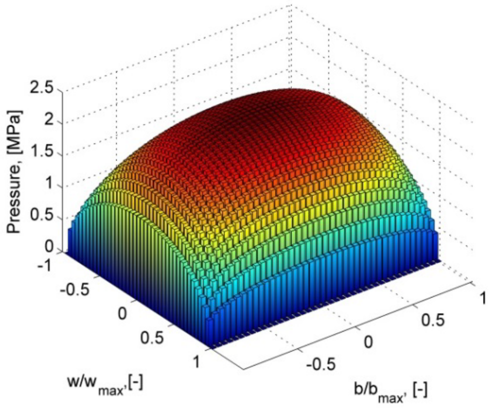


Figure 3. An example of spatial pressure distribution that was calculated with the Kurdyumov and Kheisin model.

Taking into account Assumptions 1–8, Eq. 3a–3c can be rewritten as:

$$p = 0.662 \cdot V^{\frac{13}{24}} \cdot a_p \cdot (2R)^{-\frac{1}{12}} \cdot f_p(k) \cdot F_p\left(\frac{x}{L}\right) \cdot (\Delta)^{\frac{1}{6}}, \tag{12a}$$

$$b = 1.344 \cdot V^{\frac{7}{12}} \cdot (a_p)^{-\frac{2}{5}} \cdot (2R)^{-\frac{1}{6}} \cdot f_b(k) \cdot F_b\left(\frac{x}{L}\right) \cdot (\Delta)^{\frac{1}{3}}, \tag{12b}$$

$$w = 2 \cdot \sqrt{2R \cdot b \cdot \sin(\beta)}, \tag{12c}$$

where V is the ship's speed at the moment of impact; R is the radius of the ice floe, a_p is the ice crushing strength factor; F_p and F_b are the hull shape functions that depend on frame angles and waterline angles (in the RMRS rules, F_p and F_b are approximated by the hull factors v and u, respectively, see Eq. 4–7); $f_p(k)$ and $f_b(k)$ are the coefficients that limit the value of ice crushing force in accordance with Assumption no. 7. Both $f_p(k)$ and $f_b(k)$ are ≤ 1.0 and are expressed as a function of k:

$$f_p = 1.24\left(1-k^{\frac{7}{4}}\right)^{\frac{1}{6}} \cdot k^{\frac{1}{4}} \text{ and } f_b = \left(1-k^{\frac{7}{4}}\right)^{\frac{1}{3}}, \tag{13}$$

where k is a ratio between the current velocity of the ship and ship's speed at the moment of impact.

Equations 12a and 12b can be rewritten as functions of ice parameters and vessel characteristics. These functional dependencies are presented in Table 2 and can be compared with those of IACS.

Table 2. Functional dependencies of ice pressure and the load height.

Parameter	IACS	Kurdyumov and Kheisin (1974), Kurdyumov et al. (1980) and RMRS
Ice pressure (p)	$p = f_1(\varphi, ex, P_o, wex, V, \beta, \alpha) \cdot f_2(\beta, H, \sigma_f) \cdot \Delta^{0.14}$	$p = g_1(R, k_p^a, \mu^a, n, sp, V, \beta, \alpha) \cdot g_2(\beta, H, \sigma_f) \cdot \Delta^{0.17}$
Load height (b)	$b = f_3(\varphi, ex, P_o, wex, V, \beta, \alpha) \cdot f_4(\beta, H, \sigma_f) \cdot \Delta^{0.25}$	$b = g_3(R, k_p^a, \mu^a, n, sp, V, \beta, \alpha) \cdot g_4(\beta, H, \sigma_f) \cdot \Delta^{0.33}$

φ – ice-edge opening angle; P_o, ex – coefficient and the exponent in the Sanderson's pressure-area relationship; wex – ice spalling coefficient; V – ship's speed at the moment of impact; α, β – hull configuration parameters; H – ice thickness; σ_i – ice flexural strength; Δ – vessel displacement; R – radius of ice floe; k_p, n – proportionality constants in the pressure-intermediate-layer-thickness relationship; μ – viscosity of the crushed layer; sp – spalling parameter; [a] parameters k_p and μ are often combined into a single ice strength factor $a_p = (6\cdot\mu\cdot k_p^3)^{5/24}$ (units: $Pa^{5/6}\cdot(s/m^3)^{5/24}$).

Table 2 demonstrates similarities and differences between the IACS and HDM/RMRS formulations for the ice pressure and load height. Despite the fact that the IACS and RMRS rules are based on completely different sets of assumptions, the functional dependencies of the pressure and load height on the vessel displacement are remarkably similar. Both rules specify a particular design scenario as the design basis. Each rule set assumes that the ice pressure and the load height are a function of the following parameters:

1) *Ice geometry*, i.e., the floe angle and ice thickness in the IACS approach and the floe radius and ice thickness in the HDM.
2) *Ice mechanical characteristics*: in the IACS approach, these characteristics are the exponent and the constant in the Sanderson pressure-area relationship, the spalling parameter and ice flexural strength. In the HDM, these characteristics are the viscosity of the crushed layer, flexural strength and the proportionality constants between the ice pressure and thickness of the crushed layer.
3) *Parameters of the vessel*, i.e., speed, displacement and hull shape.

Both methods account for ice-edge spalling effects by reducing the size of the nominal contact area.

The main difference between two approaches is that IACS utilizes the Sanderson pressure-area relationship in which the average pressure decreases with increasing nominal contact area, whereas the RMRS approach assumes an intermediate crushed layer and uses Navier-Stokes equations to derive pressures. For a detailed discussion about physical plausibility of these assumptions and their effect on the ice pressure, the interested readers are referred to Kim and Amdahl (2015).

The IACS design pressure is *the average pressure over the design load patch area*; whereas the RMRS design pressure is *the maximum pressure at the center of the loaded area*. The data in Table 2, Eq 1a and 1b and Eq. 3a and 3b demonstrate that the pressure and the load height in the RMRS rules has a stronger dependency on the vessel displacement. That is $p_{RMRS}/p_{IACS} \propto \Delta^{1.2}$ and $b_{RMRS}/b_{IACS} \propto \Delta^{1.3}$.

In contrast to the IACS approach, the equations that relate the RMRS ice class factors (a_1 and C_1 in Eq. 3a and 3b) and the corresponding physical ice constants and the vessel characteristics are not available in the open literature. However, a quantitative comparison of the IACS and RMRS approaches can still be made.

4.2 A comparison of IACS and RMRS rules: a case study

Strength levels are based on class factors which are developed from values for ice properties and ship's speeds. It is of interest to compare the IACS and RMRS rules by using these values.

Tables 3 and 4 present results of comparative analysis. They compare the ice pressure, the size of the load patch and the shell plate thickness in lowest Arctic categories (PC7, Arc 4) and the lowest non-Arctic (Ice 1). The tables show a link between ice loads, minimum strength and physical parameters such as the vessel speed, flexural ice strength and the ice thickness, for a transversely framed ship with displacement of 145 kilotons. The ice thickness and flexural strength in the IACS approach were derived from class factors. A relationship between the RMRS class factors and class-dependent physical parameters is not known, and a method was developed to determine that. It is presented in Figure 4.

Table 3. Class-dependent characteristics of ice loads for a 145-kiloton vessel, transversely framed with a spacing s = 0.8 m, hull angles α = 50°, β=61.8° (at x/L =0.21) and upper yield stress 355 MPa.

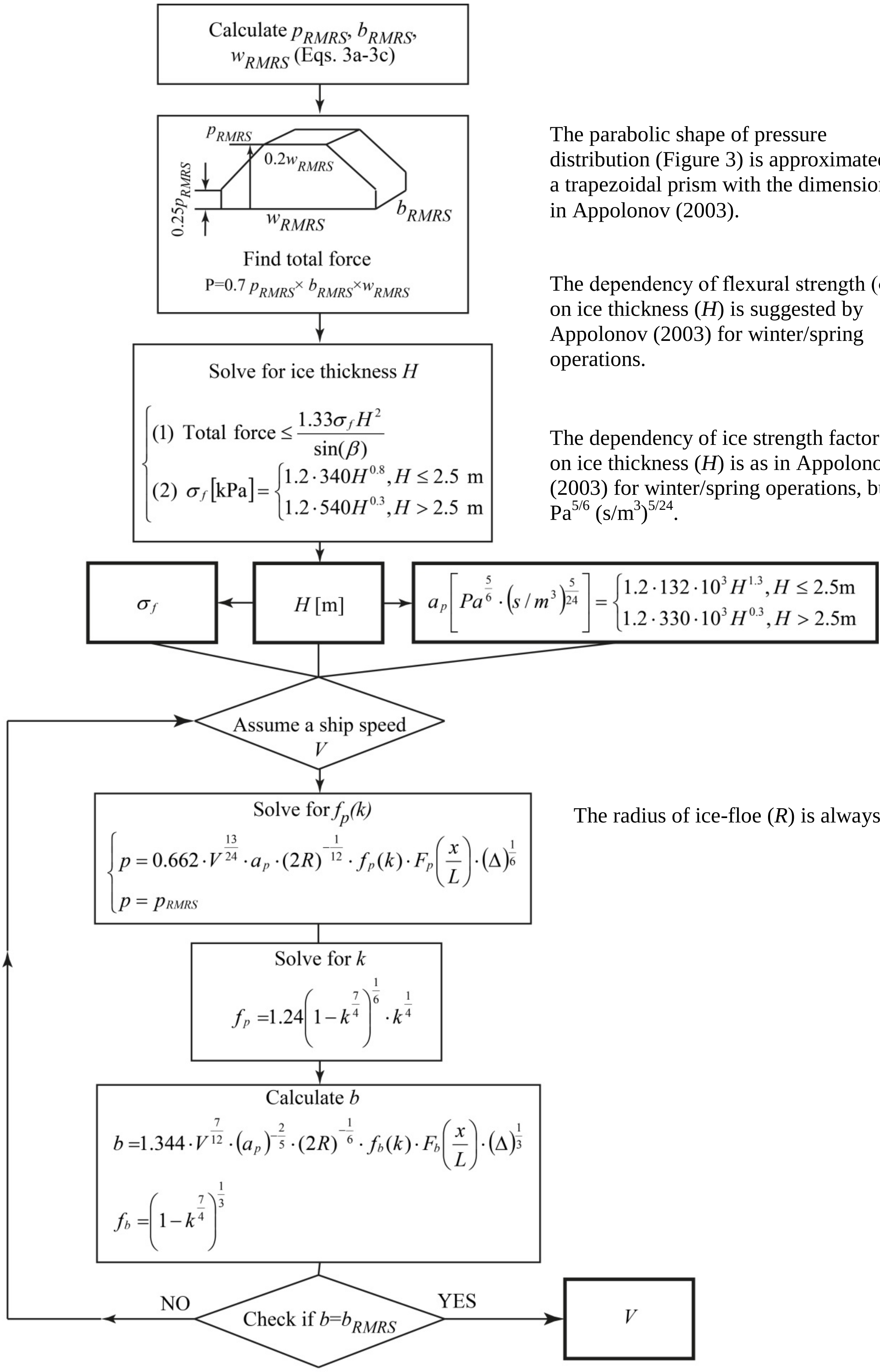
Rule	Ice Class[a]	Ice pressure, MPa	Load patch height, m	Load patch width, m	Aspect ratio	Total contact force, MN
IACS	PC7	3.2	0.52	3.39	6.57	5.53
RMRS	Ice1	1.6	0.92	10.2	11.1	10.7
RMRS	Arc4	3.6	1.2	11.5	9.76	34.2

[a] note that the IACS numbering system has higher ice class with a low number, while the opposite is in the RMRS rules.

Table 4. Class-dependent physical parameters for a 145-kiloton vessel, transversely framed with a spacing s = 0.8 m, hull angles α = 50°, β = 61.8° (at x/L =0.21) and upper yield stress 355 MPa.

Rule	Ice Class	Plate thickness, mm	Derived ship speed (safe speed[d]), m/s	Equivalent ice thickness, m	Flexural strength, MPa
IACS	PC7[a]	23	1.5 (1.54)	2.5	0.65
RMRS	Ice1[b]	19	0.56 (2.57)	2.8	0.89
RMRS	Arc4[c]	30	0.96 (4.12)	4.7	1.03

[a] for independent summer/autumn operations in thin first-year ice up to 0.7 m thick (IMO, 2014) which may include old ice inclusions; [b] for independent navigation in open pack ice up to 0.4 m in the non-arctic seas; [c] for independent navigation in open first-year ice up to 0.6 m thick in winter/spring, and up to 0.8 m thick in summer/autumn; [d] permissible safe speed which a ship under the ice conditions (a–c) may reach when navigating in ice.



The parabolic shape of pressure distribution (Figure 3) is approximated by a trapezoidal prism with the dimensions as in Appolonov (2003).

The dependency of flexural strength (σ_f) on ice thickness (H) is suggested by Appolonov (2003) for winter/spring operations.

The dependency of ice strength factor (a_p) on ice thickness (H) is as in Appolonov (2003) for winter/spring operations, but in $Pa^{5/6}\ (s/m^3)^{5/24}$.

The radius of ice-floe (R) is always 25 m.

Figure 4. A workflow diagram that shows how to determine physical parameters such as flexural strength, ice thickness and ship's speed before the impact from the known ice class factors.

Table 3 demonstrates that the ice pressure of PC7 is twice that of Ice1. This is expected because the corresponding RMRS ice class is usually set to Arc4 for which the design pressure is close to that of PC7. The total ice force for PC7 is much smaller than that for Ice1 and Arc4. This was not fully expected as it is normally believed to be the opposite for Ice1 and PC7, and that the ice-load levels are the same for Arctic categories PC7 and Arc4. The variation in load-patch area is very large, and it is the main contributor to the differences in force. For local design of plating, the plate dimensions will limit force values. In this context, the total forces are not comparable.

Data in Table 4 should be treated as theoretical design points and can only be used for comparison purposes. It is noticed that the IACS and RMRS formulae are based on a somewhat different design point scenario, and the ice thickness is not the actual ice thicknesses but rather an *assumed* value which takes into account thicknesses of level ice, ridges and snow cover. It is realized that the RMRS ice thickness for Arc4 is almost twice of that of IACS (PC7). With bigger flexural strength, the total force is approximately 6 times larger in RMRS than that of IACS. The IACS approach compensates this difference by a substantial reduction in contact area, so that the ice pressure, governing the design of plates and stiffeners, becomes comparable to the RMRS pressure. Still, there is a substantial difference between plate thicknesses in Arc4 and PC7. This is because the load patch in IACS is relatively narrow compare to the frame spacing ($b/s < 1$), whereas $b/s > 1$ in RMRS. When $b/s < 1$, the load height gives a noticeable discrepancy between the plate thicknesses of IACS and RMRS, but when b/s > 1 this difference is less important.

The results indicate that the IACS and RMRS approaches are not consistent in terms of load patches, but ice pressures are similar. The derived ice thickness and ship speed do not reflect actual operating conditions such as operations under icebreaker escort, safe speeds and different ice regimes (including ice concentration, ice type and floe sizes). These speeds can be significantly higher than those listed in Table 4.

4.3 Effect of assumptions on plate thickness: a case study

The RMRS rules were found to be robust with respect to the input parameters such as of V, R, β, a_p and sp, as long as a linear dependency between the pressure and thickness of the crushed layer is considered. The sensitivity analysis of ice crushing loads in Kim and Amdahl (2015) showed that the exponent in the Sanderson pressure-area relationship (*ex*), the spalling parameter (*wex*) and the ice crushing factor $a_p = (6\mu k^3)^{5/24}$ are the most important assumptions. Small variations in *wex* and a_p have substantial effect on the ice crushing pressure. a_p is a part of class factors and does not appear in the RMRS formulae exponents, but the same cannot be said about *wex* in the IACS model. *wex* is always 0.7 (Daley, 2000). A value of 1.0 would indicate no spalling. The design load patch area will be the same as the nominal contact area, and the required plate thickness will also change. In this section, the effect of *wex*-assumption on the plate thickness is investigated. Figure 5 demonstrates some of the issues involved in selecting the *wex* parameter.

A vessel with characteristics as listed in Figure 5 was considered. The normalized ship speed was taken as in Daley (2000), i.e., $V_n = V{\cdot}\sin(\alpha)\cos(\beta)$, where the V is the forward velocity of the vessel at the moment of impact. V depends on the ice class and was derived from the crushing class factor $CF_C = P_o^{0.36}{\cdot}V^{1.28}$. The normalized mass of the vessel (or normalized vessel displacement) was determined by equating the IACS solution for the design force and Eq. 11. In Eq. 8 the corrosion and abrasion allowance was disregarded because it is independent of the ice parameters. To illustrate the *wex*-effect, a transversely framed plating within one section in the bow region (at x/L=0.21) was considered.

Figure 5 demonstrates that for each ice class, the required plate thickness increases with decreasing *wex*. With *wex* between 0.5 and 1.0 a variety of plate thicknesses can be obtained;

the thicknesses are in the range from 17 to 28 mm for PC7. This effect is more pronounced for higher ice classes. In this context, it is important to obtain an accurate value of *wex* for design. This means that the assumption of a constant *wex* = 0.7 should be supported by relevant experimental data (or experience). For example, measurements taken on vertical fixed structures reveal that ice can fail in ductile, ductile-brittle or brittle mode depending on the interaction speed and ice confinement (Sodhi, 2001 and Bjerkås, 2004). The ice load can be distributed over the full nominal contact area or only a fraction of it. Physically, it may therefore be reasonable to assume that the value of *wex* will depend on the indentation speed and the ice thickness that, in turn, may vary during collision and be different for the various ice classes. Scant experimental data on ship-shaped structures prevent further discussions of the physical plausibility of the chosen value of *wex*. We recommend additional studies to improve the basis for this parameter.

<u>Vessel characteristics</u>

Displacement (Δ) (tons)	145 000
Length (L)	280 m
Hull angles at x/L=0.21	α=50°
	β=61.8°
Upper yield stress (MPa)	355
Frame spacing	0.8 m

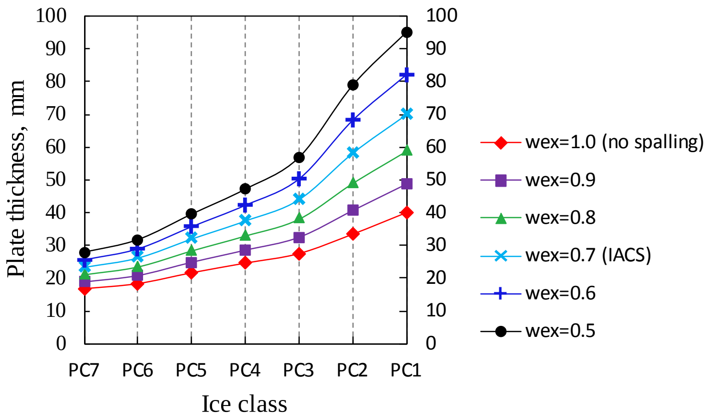


Figure 5. Effect of spalling parameter (*wex*) on the plate thickness.

Regarding the HDM/RMRS approach, further studies are needed to elucidate whether it is possible to improve the model by including a different relationship between the pressure and thickness of the crushed layer.

5. CONCLUDING REMARKS

The local design of plates and stiffeners in Arctic vessels is based on a combination of experience, empirical data and structural analysis. It is crucial to be aware of the assumptions that underlie formulations in each rule-set and to understand how those assumptions influence the design. This study has summarized and compared the basic assumptions behind ice load formulations and shell plating requirements of IACS and RMRS.

1. The principles behind the IACS and RMRS capacity formulae for shell plating are almost the same and are based on the yield-line mechanism for a partially loaded rectangular plate.
2. To assess ice loads, IACS uses the Daley model for the case of a glancing impact on the bow, and RMRS uses the Kurdyumov and Kheisin HDM of a ship’s hull collision with a floating ice floe.
3. The advantage of the IACS approach is that it is rather easy to understand and the detailed derivation of ice loads is available in the literature. There is an explicit relation between ice loads and physical parameters.
4. A full understanding of the assumptions behind the ice loads formulae of RMRS remains challenging because many important parameters such as ice geometry, strength and vessel speed are hidden behind the ice-class factors.
5. The main difference between two approaches is that IACS utilizes Sanderson’s pressure-area relationship, in which the average pressure decreases with increasing

nominal contact area whereas the RMRS approach assumes an intermediate crushed layer and uses Navier-Stokes equations to determine ice-pressure distribution over the nominal contact area.

6. The IACS design ice-pressure is the average pressure over the design load patch area whereas the RMRS design pressure is the maximum pressure at the center of the loaded area.
7. IACS' and RMRS' dependencies of the design pressure and the design load height on the vessel displacement are remarkably similar. Both codes assume the ice pressure and the load height as a function of ice geometry, ice mechanical characteristics and parameters of the vessel. They account for the ice-edge spalling effects by reducing the size of the nominal contact area.
8. The drawback of ice-load calculations with the IACS or RMRS method is that several assumptions (i.e., characteristics of ice spalling and ice strength factors) are introduced. They are not well known individually, but will substantially affect the plate thickness values.
9. In the IACS model, the ice-spalling parameter is always constant and equal to 0.7; however, the smaller the ice-spalling parameter, the smaller the load patch area, and the required plate thickness increases. By varying this parameter between 0.5 and 1.0 a variety of plate thicknesses may be obtained. The thicknesses are in the range from 17 mm to 28 mm for PC 7. This effect is more pronounced for higher ice classes and further studies are recommended to refine this parameter.
10. Further studies of the HDM/RMRS model are needed to elucidate whether it is possible to improve model predictions by including a different relationship between the pressure and thickness of the crushed layer.

This paper highlights the current similarities/differences between guidelines of the International Association of Classification Societies and the Russian Maritime Register of Shipping for ice loads and plate thickness requirements. The presented information can help engineers to better understand the various rule-formulae, their background and limitations when designing novel ice-going ships for which there is limited empirical data. It can be also used to provide operational guidance.

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