DESIGN OF ICE BREAKING SHIPS

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Contents

Summary

The design of ice capable ships includes reaching an adequate performance, adequate hull and machinery strength and proper functioning of the ship in ice and in cold weather. Good ice performance requires hull shape that has a low ice resistance as well as allows different manoeuvres required. Good ice performance includes also a good propulsion thrust which can be achieved with propeller design and also designing the hull lines so that propeller-ice interaction is minimized. The adequate strength is achieved commonly by selecting a proper ice class and following the class rules. The designer must have some insight about ice loads in order to select the structural arrangement. This chapter describes the requirements for materials, equipment and general arrangement.

1. Designing an Ice Capable Ship

Understanding how ice is acting on a ship forms the basis of design of ships for ice. In this chapter some aspects of ship design for ice are covered, and mostly in a qualitative way. The reason for the qualitative approach is that no single and exact method for any aspect of ship design for ice exists. The designer is mostly forced to search for literature and then applies in various depths a multitude of methods found – and the final design is then a synthesis of results from different sources that the designer deems most appropriate. This judgement is at best if it is based on earlier experience; this makes ice design a difficult area as most valid experience is feedback from designs that have been realized and are operating in ice. The following outline about design aspects of ice capable ships should be considered as a general overview about design; the more exact numbers must be supplied by the detailed methods selected.

The design starting point is usually a functional specification (an example given in the box below) outlining the ice performance required. This specification is made often with interaction between the owner and a designer so that the different requirements are in balance. The balance of the different requirements ensures that no single requirement drives the design. Below is shown an extract of a detailed functional specification. Often the designer is given a free hand how the requirements are met but sometimes the owner has a clear idea of how the ship shall look like.

BALTIC ENVIRONMENTAL MULTIPURPOSE ICEBREAKER

The General Ice Performance Requirements

- **Average escort speed:** The average speed in all normal ice conditions in the operational area must be at least 8 – 12 knots;
- **Level ice ahead:** The ship speed must be at least 13 knots in 50 cm thick level ice with a flexural strength of 500 kPa and thin snow cover. The ship must be able to proceed with a 3 knots speed in 1.5 m thick level ice;
- **Level ice astern:** The ship must be able to go astern with 7 knots speed in 70 cm thick level ice (flexural strength 500 kPa, thin snow cover);
- **Manoeuvring capability:** The ship must be able to turn on spot (180^o) in 70 cm thick level ice (flexural strength 500 kPa, thin snow cover) in max. 2.5 minutes. The ship must be able to turn out immediately from an old channel with 5 m thick side ridges;
- **Old channels:** The ship must be able to maintain a high speed in old channels. Especially in a channel corresponding to the requirement of IA Super ships, she has to maintain at least 14 knots speed;
- **Ridge penetration:** The ship has to be able to penetrate with one ram (initial speed 13 knots) a ridge of 16 m thickness;
- **Channel widening:** The ship has to be able to make a 40 m wide channel in 50 cm thick ice (500 kPa, thin snow cover) at speed 4 knots;
- **Performance in compressive ice:** The ship must be able to maintain a 9 knots speed in compressive ice of thickness 50 cm.

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- **Temperatures:** Air temperature -35^o--$+30^o$ and sea water temperature -1^o--$+32^o$.

2. Historical Development of Ice Capable Ships

A short historical note on the development of icebreakers and ice going ships is presented by noting the major steps in the evolution. The first ice breaking ships appeared in mid 1840's in Hudson River in the US and in the Elbe River in Germany. First dedicated icebreakers appeared in 1860's and 1870's in the St. Petersburg and Hamburg harbours. Before the turn of the century several dedicated sea-going icebreakers were in service. The development of merchant ships for ice started towards the end of 19^{th} century. The year-round navigation in the Baltic started in 1877 with the introduction of the ship Express II sailing between the ports of Turku and Stockholm. The hull design of this ship and many similar ones followed that of the icebreakers; only the machinery power was larger in icebreakers.

Ships that were intended to sail independently in ice evolved in 1950's in the Soviet Union with the emergence of the Lena- and Amguema-series of ships (the latter is also called Kapitan Gotskij series). These ships had an icebreaking bow shape and a high strength for Arctic trade. Several series of Arctic ships has been built to Soviet and Russian owners (e.g. Norilsk and Norilsk Nikel-series) and to Finnish owners (Lunni-series) – the Canadian ships MV Arctic and MV Umiak 1 should be mentioned also. Since the early times the icebreakers and ice breaking ships have developed much based on several technological innovations, some of which are mentioned below.

The hull shape of the early icebreakers in the 19^{th} century was characterized by a very small buttock line angle φ at the stem; values were usually smaller than 20^0 (definition of hull angles, see Fig. 1). The buttock lines and waterlines were rounded and the sides were inclined ($\beta > 0$). The rounded stem developed quite late (in the 1980's) as a sharp bow was long deemed favourable for ice breaking. The principle of hull lines design is and has been to make the flare angle ψ as small as possible.

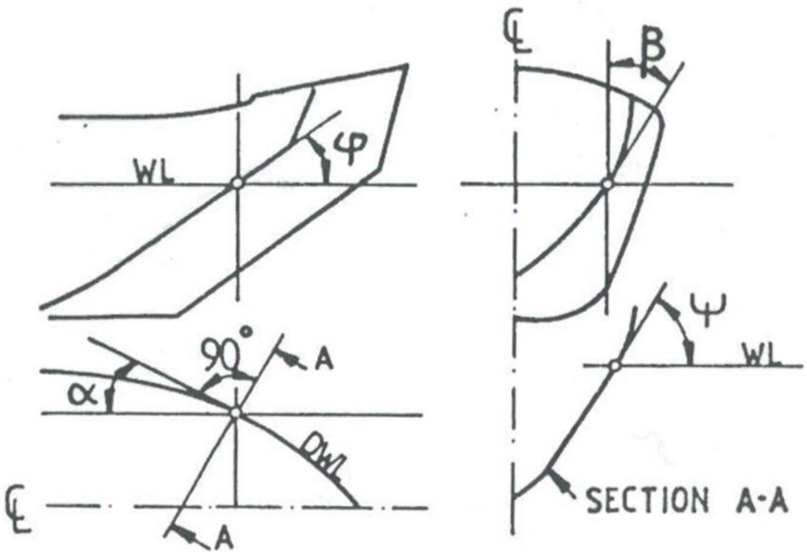


Fig. 1. Definition of the hull angles.

The general arrangement of icebreakers and also ice going ships has changed little during the years. The largest change in the arrangement took place in 1970's when the superstructure was changed into deck house i.e. no accommodation was placed in the hull, Fig. 2. The reason for the change was partly to increase the height of the bridge to improve the visibility and partly to avoid the noise and vibration caused by ice in the crew accommodation.

Fig. 2. Early icebreakers had accommodation in the hull like the first USCGC Mackinaw (left). After 1970's the deck house replaced the superstructure like in the Finnish icebreaker Sisu (right).

Machinery of icebreakers has experienced many changes since the early icebreakers with steam engines and fixed pitch propellers. The economy and torque capability of steam engines was improved much with the introduction of diesel-electric machinery (diesel main engine with generators and electrical propulsion motors). The first diesel-electric icebreaker was the Swedish IB Ymer in 1933. The diesel-electric machinery is more expensive than a direct diesel drive but the torque performance of a fixed pitch propeller with a direct drive is not good; the solution for this is the use of controllable pitch (CP) propellers. These became common in early 1980's in merchant ships. The bow propellers were introduced in icebreakers in the end of 19^{th} century (first European bow propeller icebreaker is the Finnish Sampo). The bow propeller improves the ice breaking capability by reducing the forces required to break ice and by reducing the friction. Only lately the bow propellers have been made superfluous by the introduction of so called Z-drives (azimuthing propulsion units); the first icebreaker with azimuthing propulsion was the Finnish multi-purpose icebreaker Fennica in 1993.

Strength of ship hull and machinery is still mostly designed based on experiences from earlier ships. When damages caused by ice have occurred, strengthening of the structures is indicated. These experiences have been collected into the rules of the classification societies and thus most of the strength design is even nowadays done following the classification society rules. The Baltic is the most active sea area for ice navigation and it is natural that the experiences from Baltic are followed worldwide. The experience from ship damages is reflected in the strength level used in the Finnish-Swedish Ice Class rules. Already these short notes from the historical development of ice design show how closely the design of ice capable ships is linked with the experience from earlier designs. As the collection of feedback is not a straightforward task by any means, those designers that can follow the performance of their design in ice operation have an advantage.

3. Performance in Ice

Ship performance in ice consists of ability to break ice and to manoeuvre in ice – these capabilities have been defined in the functional specification. The capability of breaking ice is measured in uniform ice conditions (level ice, brash ice) by the speed at which certain ice thickness can be broken. Ice ridges and multi-year ice floes are distinct ice features and the capability in these is measured by the ability to penetrate these. The speed that the ship makes in ice is determined by the ice resistance determined by ice properties, and the hull shape and main dimensions as well as the thrust provided by the propulsion. The manoeuvring performance is similarly determined by the transverse forces provided by the rudder(s)/azimuthing thrusters and the resisting forces mainly due to ice. It is thus clear that the performance in ice is influenced by the resisting forces and the propulsive forces and these can be improved (resisting forces minimized and propulsive forces maximized) by hull shape and propulsion design, respectively.

Ice Resistance

Ice resistance refers to the time average of all longitudinal forces due to ice acting on the ship. These ice forces are divided into categories of different origin;

- Breaking forces;
- Submergence forces; and
- Sliding forces.

In different ice conditions the relative importance of these components varies; in level ice the breaking component is usually the largest but in brash ice or in smaller ice floes the other two components become more important. The breaking force is related to the breaking of the ice i.e. to crushing, bending and turning the ice. Submergence is related to pushing ice down along the ship hull whereas the sliding forces include frictional forces. Usually the velocity dependency of the ice resistance is attributed to the last component. A sketch of ice resistance experienced by a ship is shown in Fig. 3.

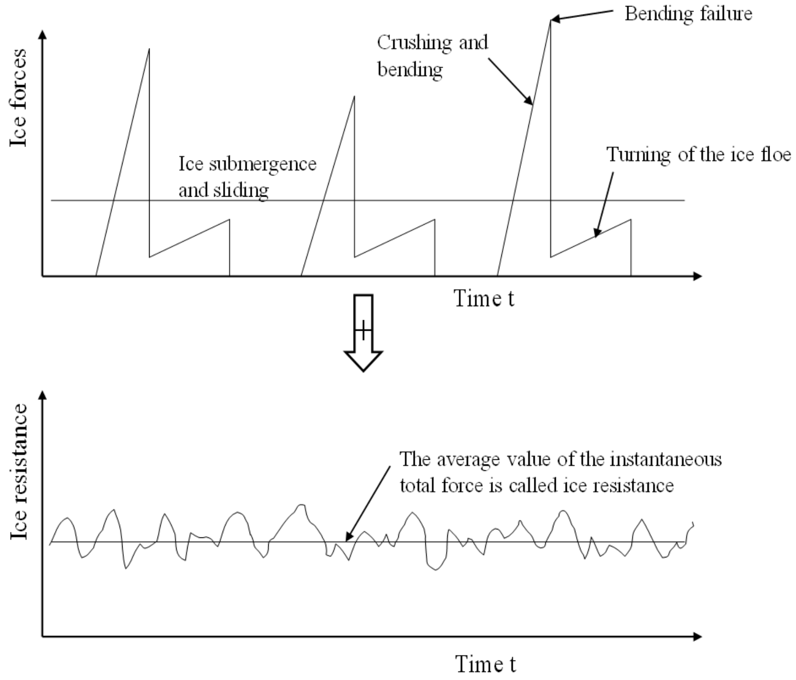


Fig. 3. The nature of ice resistance as an average longitudinal force.

Ice resistance in level ice is the basis of all other ice resistance formulations, this is investigated first. If a test is made in uniform level ice where the ship power is kept constant, the ship eventually reaches a constant speed. The total resistance in ice, R_{iTOT}, can be assumed to be equal to the propeller thrust (with so called thrust deduction deducted). If the power is decreased, a new, lower speed is reached and a new ice resistance point can be obtained. These schematic points are shown in Fig. 4. When decreasing the power further the point marked C is reached. Here any lower power brings the ship to a stop. If the test is carried out by starting a stopped ship and increasing the power, it is noted that the calculated thrust at the power when the ship starts to move is quite large (point A) and after start the ship accelerates to a speed beyond the point C. The points C to D can be extrapolated to zero speed – this gives the ice resistance at zero speed which is commonly identified with the breaking resistance.

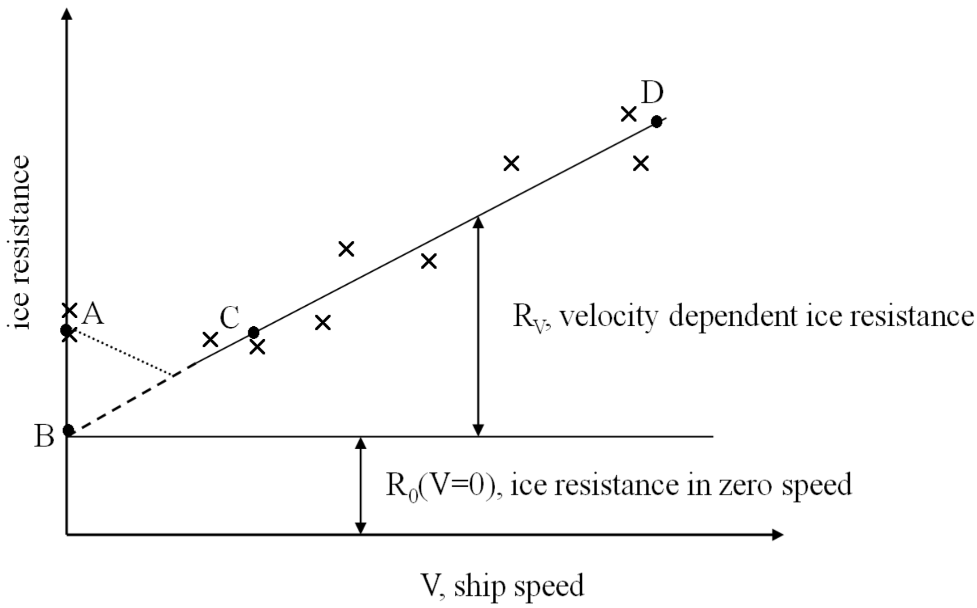


Fig. 4. Measured ice resistance points.

The total resistance in ice is assumed to be the sum of the pure ice resistance R_i and open water resistance R_{ow}

$$R_{iTOT} = R_i + R_{ow},$$

even if this assumption is inaccurate. The total resistance in ice and the open water resistance can be determined experimentally in model tests and then the pure ice resistance can be determined by subtraction. The ice resistance is further divided into components mentioned above, thus the ice resistance is

$$R_i = R_B + R_S + R_F,$$

where the components are the breaking, submergence and friction component, respectively. Most methods used to calculate the ice resistance are based on regression on full scale and model scale data. The regression assumes the ice resistance to be linear with ship speed and to consist of these three components, see for example Lindqvist (1989) or Riska et al. (1998). Thus the calculation methods for ice resistance are at best semi-empirical, and these methods should be used cautiously, especially outside the range of validity. The calculation methods to determine the ice resistance should be used only in the conceptual design phase as these methods cannot account for the details of the hull shape. When the design proceeds, ice model tests should be carried out to finalize the hull shape.

The ice resistance in broken ice (brash ice) can be determined similarly as the ice resistance in level ice. The only exception is that the breaking component is different; it exists and is attributed to cohesive forces present in broken ice. Brash ice resistance formulations are presented for example in Riska et al. (1998).

Ice resistance in ridges is dealt with similar methods as in brash ice. The major difference is, however, that as brash ice resistance depends on brash ice thickness H, the ridge resistance depends similarly on the ridge thickness, which is different *at each location along the hull*. Thus the resistance from a ship length segment Δx at the location x (in some suitable fixed coordinate system) is (see Riska et al. 1998)

$$\Delta R_R = R_R(H_R(x)) \cdot \Delta x$$

where $R_R(H_R(x))$ is the ridge resistance in ridge thickness H_R *per unit ship length*. Here the ship speed is not mentioned as the ridge resistance is commonly treated as speed independent – and the speed dependency is allocated to the open water resistance. The total resistance in a ridge is thus

$$R_{R,TOT} = R_{R,B}(H_R(x_{bow})) + \int_{L_{PAR}} R_R(H_R(x)) \cdot dx + R_{ow}(v)$$

where the $R_{R,B}$ is the ridge breaking resistance acting at the ship bow. The speed dependency in open water resistance is emphasized. The length of the ship parallel midbody is denoted as L_{PAR}.

As the ship is moving, location of the ridge relative to the ship is changing ($x = x(t)$) and as the ridge is not of uniform thickness, ridge resistance is changing constantly with time. It is consequently more suitable to speak of the energy required to penetrate certain size of ridge. This energy depends on ridge cross sectional area A and ship dimensions, for large tanker the energy has been determined to be about $C \cdot A$ where C is about 1 kJ/m^2 (Riska et al. 2006). A way to determine the ridge resistance and the penetration energy in model tests is shown in Fig. 5.

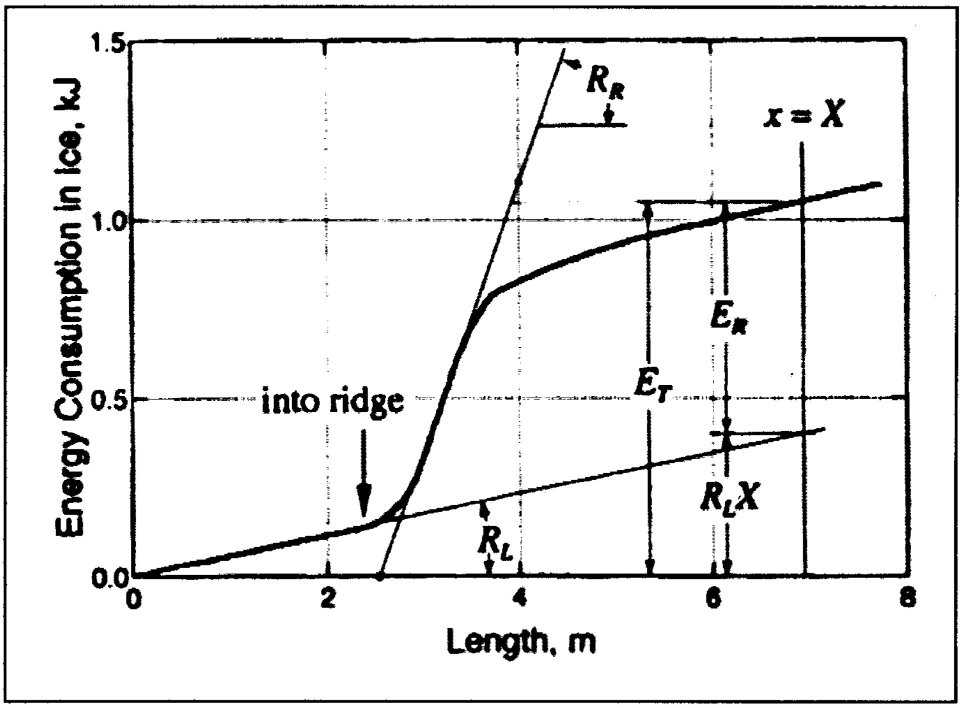


Fig. 5. Energy consumed in penetrating ice ridges, E_R, based on ice model tests (Izumiyama & Uto 1995).

Performance in ice

Measures by which the ship performance in ice is described can be seen from the functional specification described above. These measures can include:

- Speed(s) achieved in certain level ice thickness (for example 3 knots in 1.5 m thick ice with a snow cover of 20 cm);
- Penetration of certain size ridges with a stated impact speed (for example a ridge of maximum thickness 8 m penetrated with an initial speed of 10 knots); and
- Ship turn of 180^0 in less than certain time in certain ice thickness.

Ship design proceeds so that at early design phase some analytical methods are applied to determine the level ice performance, ridge penetration and brash ice performance. Some of these methods are described in Riska et al. (1998) and Juva & Riska (2002) but it is difficult to find a comprehensive presentation of methods to be used. The methods presented in references are at best rudimentary and as there are no analytical methods that can be applied in determining the manoeuvring performance, ice model tests are necessary at the end of conceptual design phase to verify the design.

Performance in level ice is described with ice thickness versus ship speed (h_i – v) plots. In these plots the speed the ship can reach in specified ice thicknesses at full power is drawn, see Fig. 6. In an early design phase the thickness-speed plot can be determined as follows. First the ice resistance curves (ice resistance versus ship speed at different level ice thicknesses) are obtained by some semi-empirical ice resistance formulation or from ice model tests. Next the net thrust concept is used; this is the propeller thrust available to overcome the ice resistance i.e.

$$T_{NET} = T(1-t) - R_{ow},$$

where the superposition principle of pure ice resistance and open water resistance R_{ow} is assumed valid, T is the thrust of propeller(s) and t the thrust deduction coefficient. As the propeller thrust, thrust deduction nor open water resistance are available in early design, the expression for net thrust must further be simplified. This is done by using an expression for bollard pull T_B and a quadratic factor for the speed dependency as follows

$$T_{NET} = T_B \cdot \left(1 - \frac{1}{3}\frac{v}{v_{ow}} - \frac{2}{3}\left(\frac{v}{v_{ow}}\right)^2\right) = K \cdot (P_D \cdot D_P)^{2/3} \cdot \left(1 - \frac{1}{3}\frac{v}{v_{ow}} - \frac{2}{3}\left(\frac{v}{v_{ow}}\right)^2\right)$$

where v_{ow} is the open water speed of the ship, P_D propulsion power, D_P propeller diameter and K an empirical factor for bollard pull (for more information of this see Juva & Riska 2002). The values of 0.78 for single screw and 0.98 for double screw ships can be used.

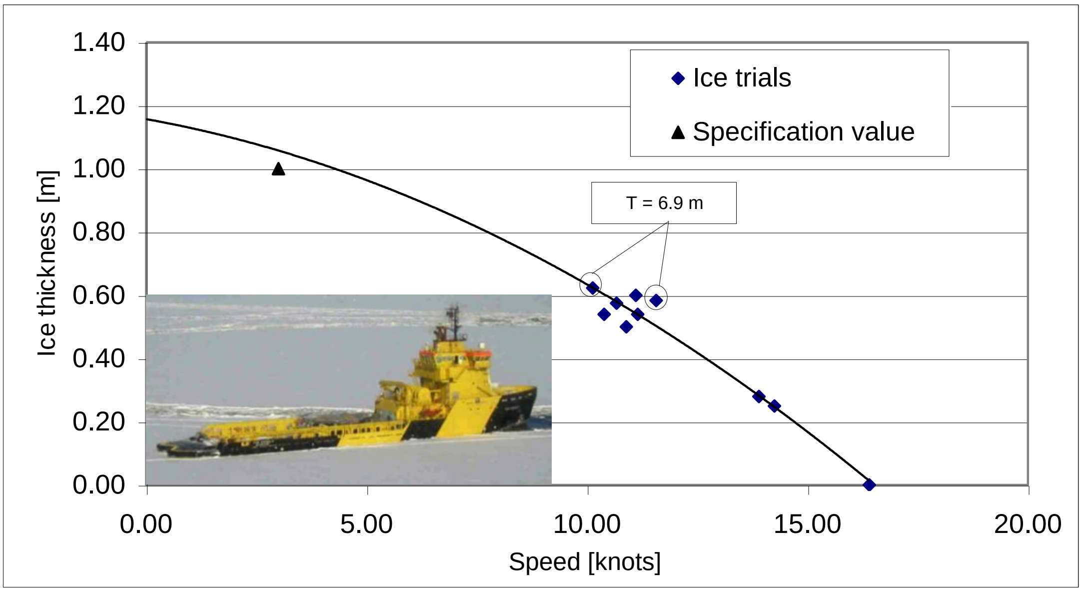


Fig. 6. Performance plot for the icebreaker Tor Viking (Riska et al. 2001).

The points where the ice resistance curve at each ice thickness (h_1, h_2 and h_3 in the graph below) intersect with the net thrust curve give the points on the h_i – v plot. In Fig. 7 a word of caution is mentioned viz. the net thrust concept assumes no propeller – ice interaction and depending on the propulsion layout, hull lines and ice thickness, this interaction can be severe. Thus designers often make a margin for this interaction. The resistance-net thrust plot and the resulting h_i-v plot are shown for the USCGC Mackinaw in Fig. 8. The ship performance in old navigation channels (brash ice) is determined similarly as in level ice, only resistance formulation used is different, see Riska et al. (1998).

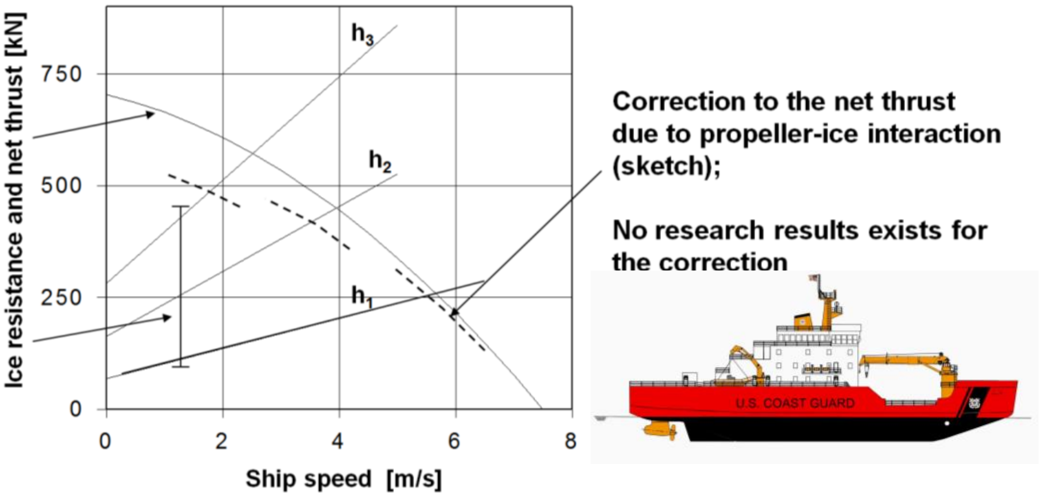


Fig. 7. Ice resistance curves for different level ice thicknesses and the net thrust curve for USCGC Mackinaw.

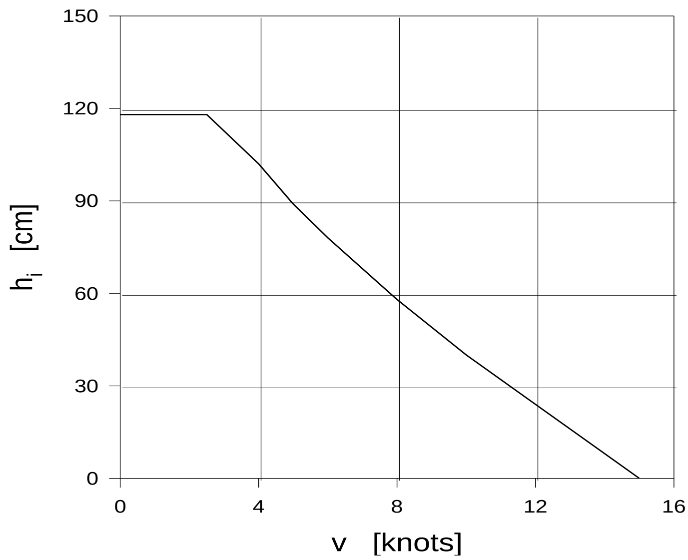


Fig. 8. Resulting h_i-v curve from Fig. 7 for USCGC Mackinaw.

Turning performance in ice is measured by the diameter of the turning circle (divided by the ship length). Turning diameters for two icebreakers are shown in Fig. 9. The requirement for escort icebreakers and other ships that have to manoeuvre well in ice is that the turning circle diameter should be less than 5*L*. The turning ability measured by the turning circle diameter is not the only measure for manoeuvring capability of an icebreaker. Important is to perform certain manoeuvres in shortest time possible. This manoeuvring performance often includes different escort manoeuvres that icebreakers commonly do, see Fig. 10. A good manoeuvring capability can be achieved by a proper hull form design, having a large rate of turn of the rudders/azimuthing thrusters and providing a large transverse force. It is thus clear that azimuthing thrusters provide a manoeuvring ability that cannot be surpassed with other turning means.

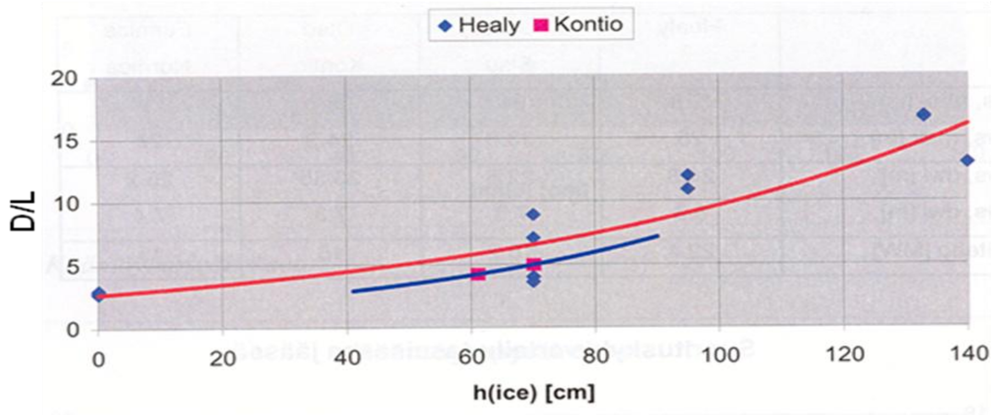


Fig. 9. Turning circle diameter *D* of two icebreakers divided by the ship length *L* in different level ice thicknesses (Hänninen & Riska 2001)

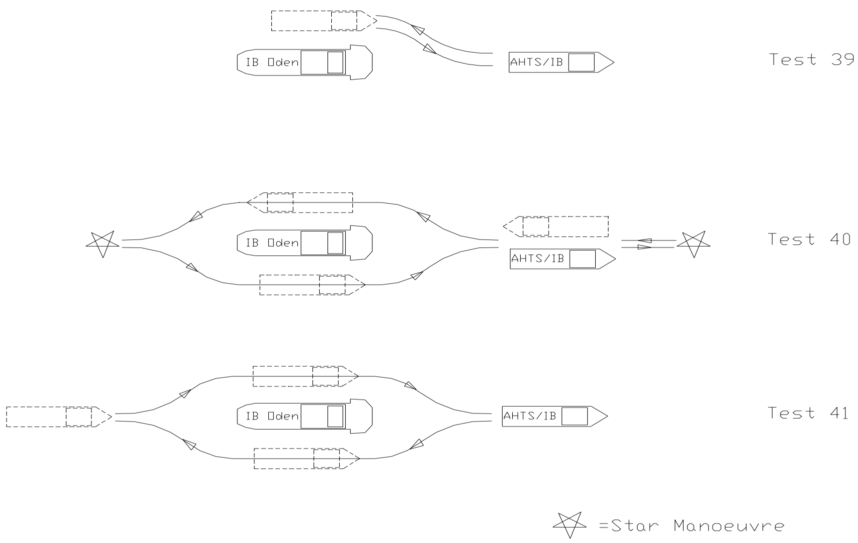


Fig. 10. Three different manoeuvres performed when the escorted ship is stuck in ice. These manoeuvres were performed in the full scale trials of the icebreaker Tor Viking and times for each operation were 13’47’’, 17’20’’ and 17’40’’ for tests 39, 40 and 41, respectively (Riska et al. 2001).

Ship performance in ridges is not measured by ridge resistance or any speed reached in ridges as commonly the largest ridges cause a resistance, even defined as an average value as in Fig. 5, that is so large that the delivered thrust cannot overcome it. Ridges are penetrated by consuming the kinetic energy of the vessel; thus the correct parameter for ridge capability is the energy required to penetrate ice ridges. This depends on ship displacement, ship main dimensions and on bow shape, see Riska et al. (1998).

Hull Shape Design

The hull shape design of ice breaking ships aims at:

- Minimizing the ice resistance by selecting optimal beam and bow shape;
- Ensuring good operational (manoeuvring) characteristics;
- Enabling the ship to go astern as much and as well as the operational description requires; and
- Ensuring a proper undisturbed operation of the propeller(s) by minimizing the amount of ice impacting on the propeller(s).

The most important parameters for ice resistance are the beam *B* and the stem angle ϕ. Large beam causes more resistance and thus narrower ships with a large *L/B* ratio (especially if there is a draught restriction) is the result. For an icebreaker small beam is not, however, good as the escorted ships should get as wide channel as possible. Typical largest icebreaker beams at present are about 26 m. A smaller stem angle induces larger bending force while keeping the horizontal force component smaller, thus ice breaking ships have quite small stem angles, 20^o to 25^o is common. Nowadays also the stem is rounded as this decreases the crushing at the stem, Fig. 11.

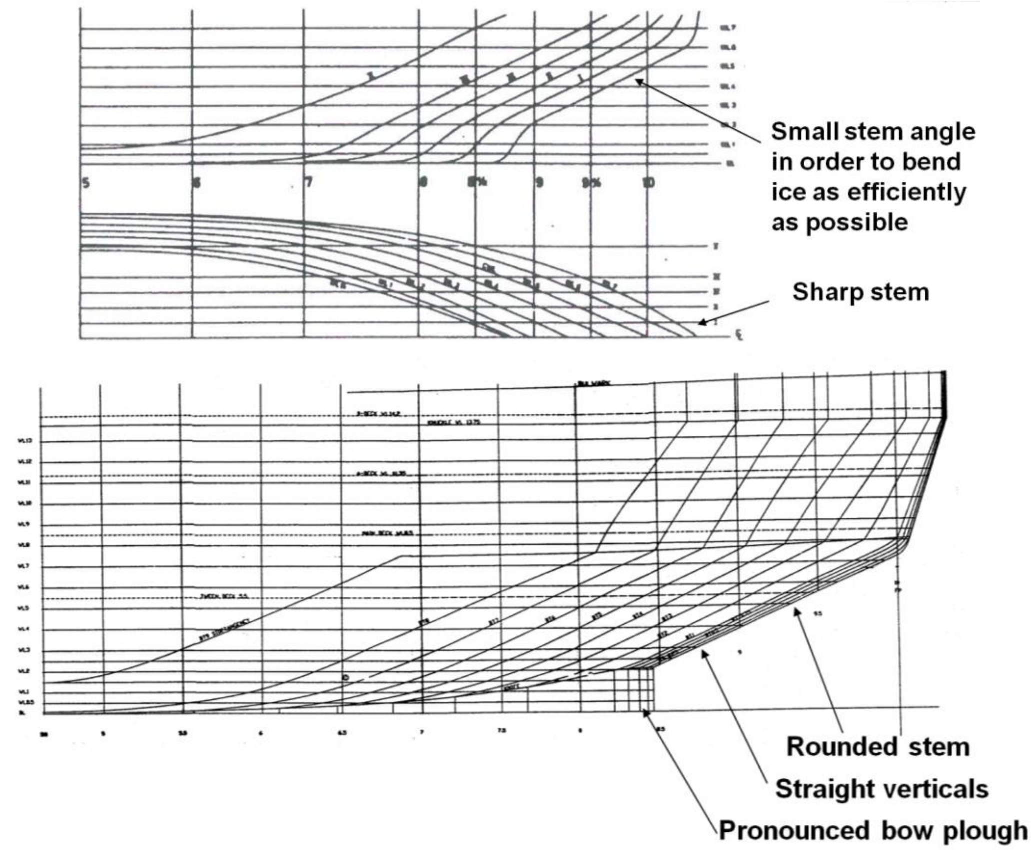


Fig. 11. Classical ice breaking bow shape (top) and modern icebreaker (bottom).

Manoeuvring characteristics are improved if the transverse force is large (large rudders). This has led to using azimuthing thrusters (more of these below). From the hull shape perspective, the stern shoulder area is crucial for good manoeuvring characteristics. If the stern shoulders break ice in bending, the ship turns better as the resisting force for turning is this way minimized.

The performance astern is important if the ship has to navigate independently. When encountering ridges, the ships often get stopped and in order to be able to proceed, the ship must be able to back and ram again (or after going astern go around the ridge). Good backing performance is reached by avoiding blunt lines at the stern. Many merchant ships that are only ice strengthened need not go astern in ice but can count on icebreaker escort in heavier ice conditions. In this case the design of the stern shape is less important.

The propellers encounter those ice floes that have made their way under the ship (flat) bottom. Propeller–ice interaction threatens the integrity of the propulsion but also decreases the propulsion efficiency; generally the required torque is increased while the produced thrust is decreased if much ice interacts with the propeller. Hull shape influences the amount of ice that gets under the ship bottom and consequently impacts on the propeller(s). Bow shape should be such that it allows the ice floes to float towards the surface before getting under the bottom. One way to do this is to make a bow plough, see Fig. 11. When the ice floes follow the buttock lines, they hit the bow plough that pushes the floes aside. There is an ice thickness limit up to which the bow plough is efficient, thicker ice will go under the bottom. The disadvantage of the bow plough is that it increases somewhat the ice resistance and also the open water resistance – in ships that are required to do heavy ice breaking the advantages outrank the disadvantages of a bow plough.

There are several points that should be checked in finalizing the hull lines for good ice breaking performance. At present there is no alternative to ice model testing as analytical methods are not so advanced that they could predict the effects which can be deemed local. The operation of the bow plough in deviating ice has already been mentioned. Here two other effects are mentioned. The first one concerns the bulbous bow. It has often been stated – and this has been reflected in some classification society ice rules – that bulbous bow is not good in an ice going ship. This is because the bulb itself does not break ice very well and also because there is one frame that is vertical at the bow, if the ship has a bulbous bow (at the bulb ice is bent up and at the shoulders down; thus there is a vertical frame in between).

Present experience shows that most merchant ships need not break ice as they either sail in broken channels or follow an icebreaker. Thus ice strengthened ships often have bulbous bow which is not a handicap in broken ice. The reason for this is that broken ice is displaced around the hull in a way that resembles the hydrodynamic flow, see Fig. 12. Only in ice going ships and icebreakers which must break ice themselves the bulbous bow is not appropriate. By shaping the bulbous bow for ice, much of the additional ice resistance can be avoided. An ice bulb is shown in Fig. 13.

Fig. 12. Bow wave created in a brash ice channel in front of a ship having a bulbous bow.

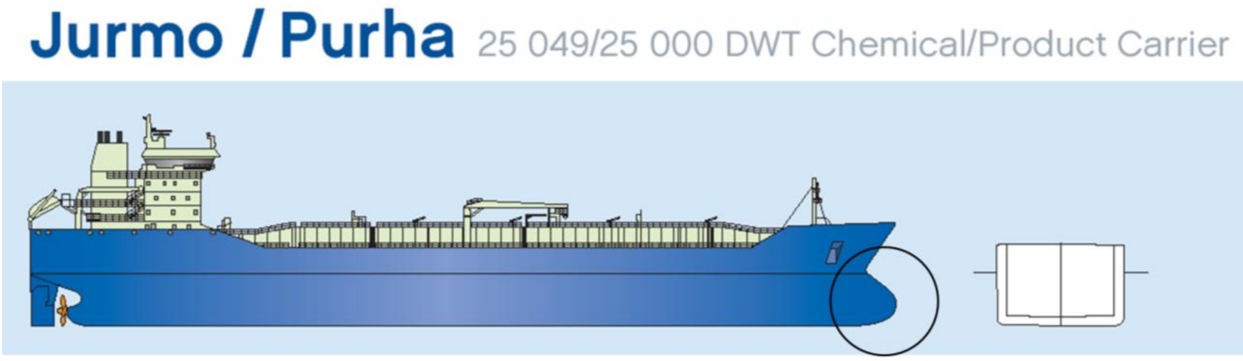


Fig. 13. A ship having an ice bulb, circled in the figure (www.nesteoil.com).

The other necessary check in hull lines is the presence of so called shoulder crushing. This phenomenon is created if the bow breaks in bending a narrower channel than the ship beam. In this case the ship has to force herself into a narrow channel by crushing the rest of the channel width close to the maximum beam (forward shoulders). The crushed ice extruded on top of the ice indicating the presence of shoulder crushing is shown in Fig. 14. If shoulder crushing is present, the ice resistance is increased much and it can even lead to a hull shape that is considered a failure. The only way at present to detect this phenomenon is to conduct ice model tests. In these the visual observation of the breaking is the best way to detect shoulder crushing – especially as the present model ices are somewhat weak in comparison with the bending strength. Thus the shoulder crushing is not revealed as a much increased ice resistance in ice model tests. There are simulation tools under development (Su et al. 2009) which may be able to predict the crushing. These simulate the breaking pattern assuming a certain size and shape of the broken ice floe. The first results of these are encouraging as Fig. 15. shows.

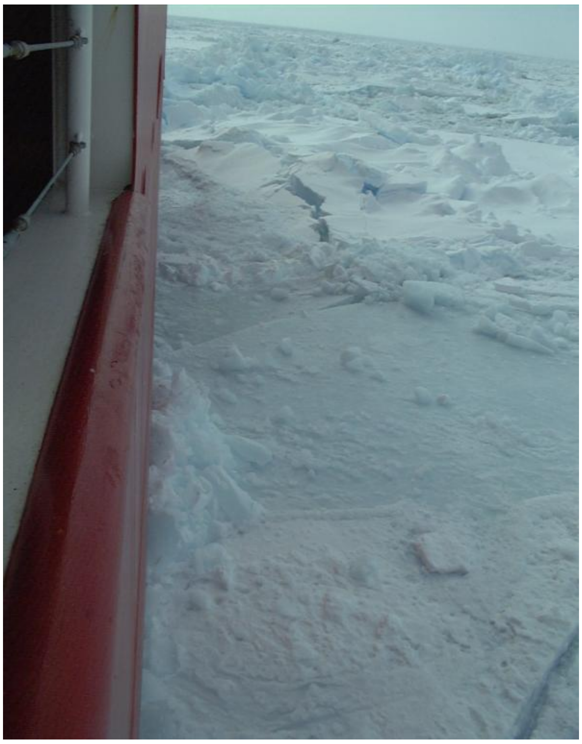


Fig. 14. Shoulder crushing at the bow of an icebreaker. The image is taken looking forward.

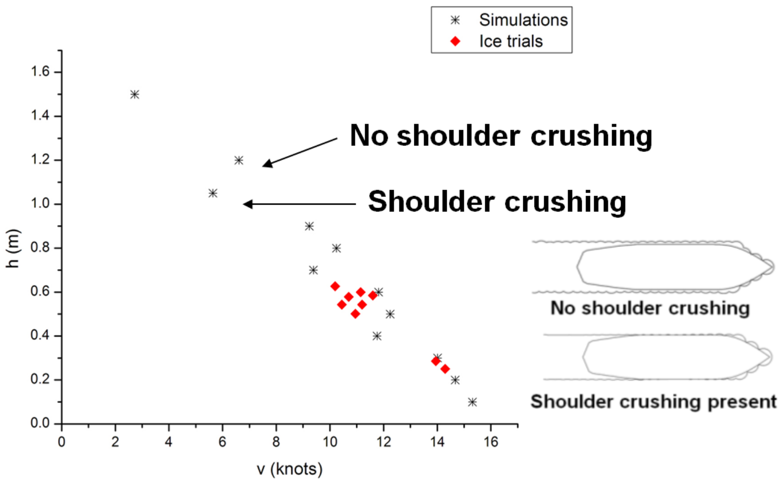


Fig. 15. Observed and simulated points of the $h_i - v$ plot for the Swedish icebreaker Tor Viking and the simulated breaking pattern. Measured points are from Riska & al (2001) and simulated from Su et al. (2009).

4. Machinery Layout

Main task of ship machinery is to produce the required thrust for ship propulsion. Main components of the ship propulsion machinery are the main engine, power transmission and the propeller. Each of these is described with the point of view of design for ice.

Machinery Alternatives

There are many alternatives for how the machinery layout can be realized. Most common machinery layout in ice classed tonnage a diesel engine or engines with a direct shaftline transmission (with or without gears) and a fixed pitch propeller (FPP) or controllable pitch propeller (CPP). An alternative is to use diesel electric propulsion where the power transmission is electric and separate electric propulsion motors supply the torque to FP propellers. Also gas turbines have been used as main engines or combined diesel/gas turbine solutions, where the gas turbine is used as a booster when high power is required.

The main difference in the machinery layout of an ice classed (ice going) ship as compared with open water ships is the operating regime of the propeller – in ice going ships the propeller load varies a lot depending on the ice conditions. The continuous load could be anything between the open water load to torque in bollard condition. If much ice is acting on the propeller, the ice torque can exceed the torque given by the engine; then the propeller slows down. Diesel engines have a relatively small *RPM* range where they can deliver full power, see Fig. 16, and thus direct drive diesel solutions may stall which often leads to stopping of engines in situations with heavy ice load. An improvement is to use a CP propeller that can adjust to the increased torque by decreasing the propeller pitch and in this way maintain the *RPM*. Even better solution is to use electric propulsion motors with diesel main engines and generators. Electric motors can maintain the torque in a large RPM range, thus the diesel-electric machinery is very efficient in conditions where occasional high torque loads are encountered. Diesel electric propulsion is used in most ice-going ships.

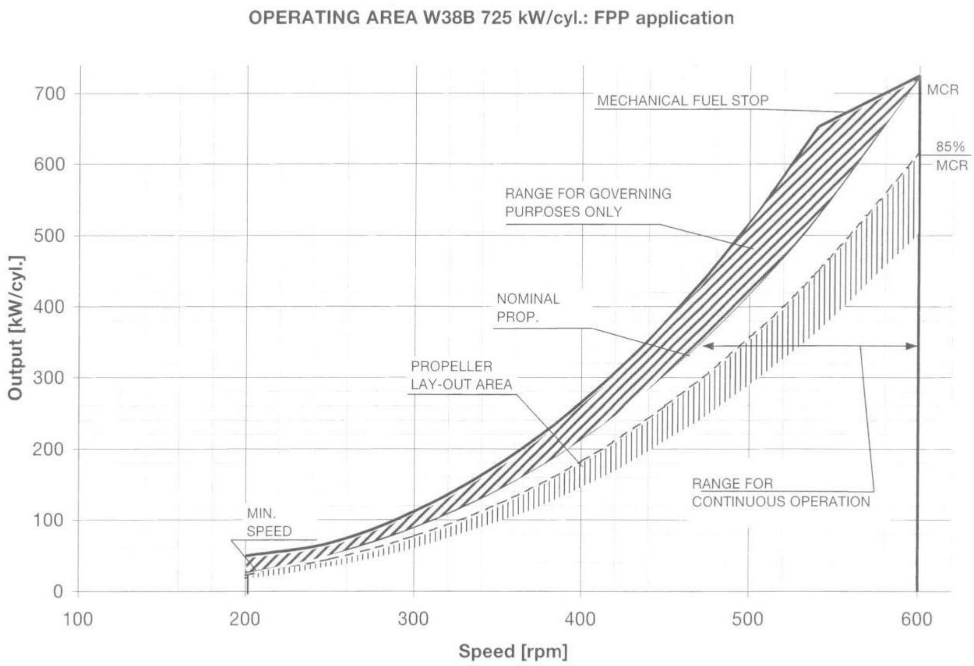


Fig. 16. Operating area of a Wärtsilä W38B main engine (Project Guide for Marine Applications, Wärtsilä 38B – 1/2002).

Use of the diesel electric propulsion must be accompanied with a control system that allows an over torque i.e. torque above the maximum torque absorbed by the propeller in the bollard pull condition. If the control system is such that immediately when the 100 % torque is exceeded it would start reducing the *RPM*, the performance of the ship would suffer much. Thus when over-torque is allowed for short periods of time (the limit comes from machinery heating), the ship can overcome short bursts of propeller-ice interaction without losing the thrust. Several knots in ship speed can be gained by allowing 40 % of over-torque as the icebreaker Fennica full scale trials demonstrated.

The over torque question can be presented in several ways; one of them is presented in Fig. 17. The delivered power is

$$P_D = 2\pi n \cdot Q\,,$$

where Q is torque and n number of revolutions. Thus on *power–RPM* chart the constant torque yields a straight line. Now if a higher torque is required because of the ice interaction (curve A in the diagram) and if there would not be any over torque allowance, the propulsion would develop less than 100 % power, which is not good for the ship performance. The action of the over-torque allowance can be measured in full scale trials if the torque and *RPM* are measured in the trials. The measured (*RPM*,*Q*) points with the time as a parameter are shown in Fig. 18. This plot is from a test where the ship started from standstill and accelerated to full power and then stopped again. The required over-torque moved roughly along the maximum power curve (hyperbola in these coordinates) and fluctuated up to about 115 % over-torque.

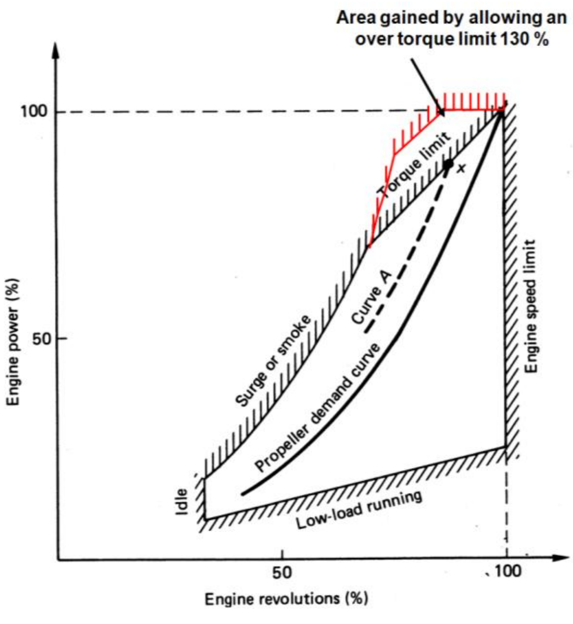


Fig. 17. Over-torque area above the nominal torque area.

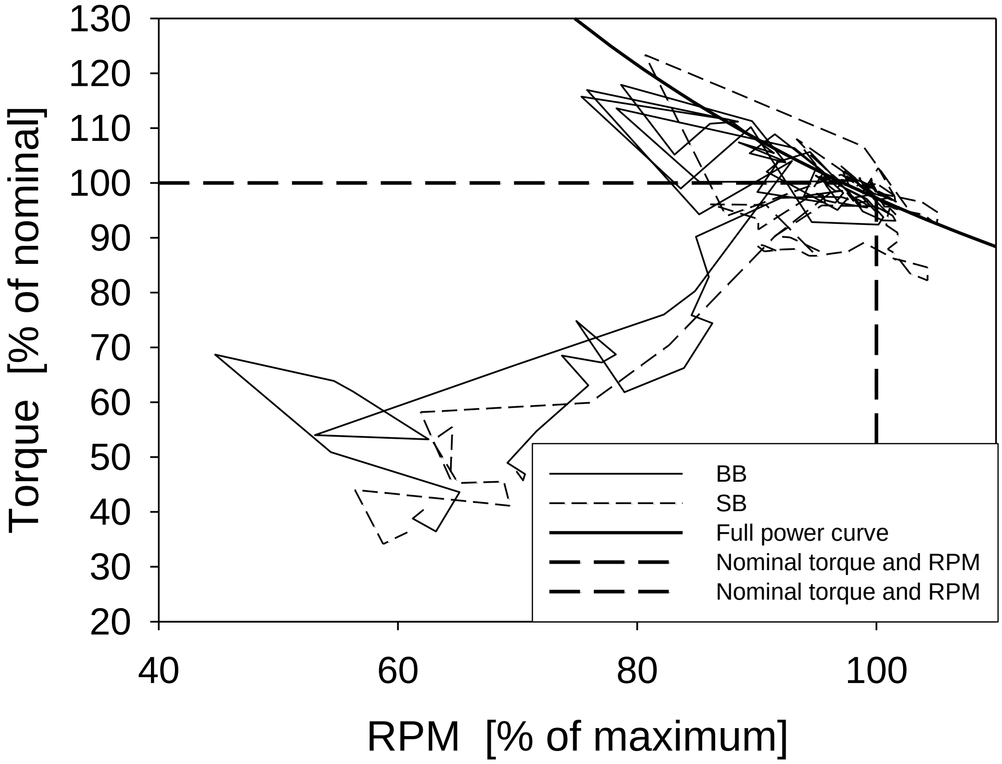


Fig. 18. Measured shaft torque and *RPM* values in ice tests showing clearly the over torque values experienced.

Propulsion Design

Classic propulsion system layout in icebreakers is single screw with a single rudder. The increased power led to twin screw solutions and then to introduction of bow propellers. The advantages of bow propellers were noticed in road ferries in the USA. The decrease of ice resistance due to bow propellers is attributed partly to a lubrication effect and partly to decrease in breaking resistance. The development of propulsion systems led finally to Urho class icebreakers with two bow and two stern propellers, two rudders and diesel electric propulsion. These ships that were constructed in mid 1970's can be considered the last conventional icebreakers. The development of the propulsion arrangement of icebreakers is shown in Fig. 19 where the first icebreakers are from late 19th century.

The design of propulsion systems for ice going ships has experienced two large steps towards more advanced systems since 1970's. The strength of CP propellers became adequate for ice conditions in 70's and now CPP's are widely used in ice going merchant ships. CPP's are applied with a direct shaft drive in merchant vessels while in diesel electric machinery applications there is no need for a CPP. Another major step forward occurred with the construction of the Finnish multipurpose icebreakers, MSV Fennica and Nordica, in 1990's, see Fig. 20. In these icebreakers azimuthing thrusters were used first time in icebreakers. Azimuthing thrusters offer a superb manoeuvring capability and they also replace the bow propellers because the propeller wash of the azimuthing thrusters can be used to flush ice along the ship hull. Since the construction of Fennica and Nordica, azimuthing thrusters have become most common propulsion system in icebreakers and ice breaking supply ships.

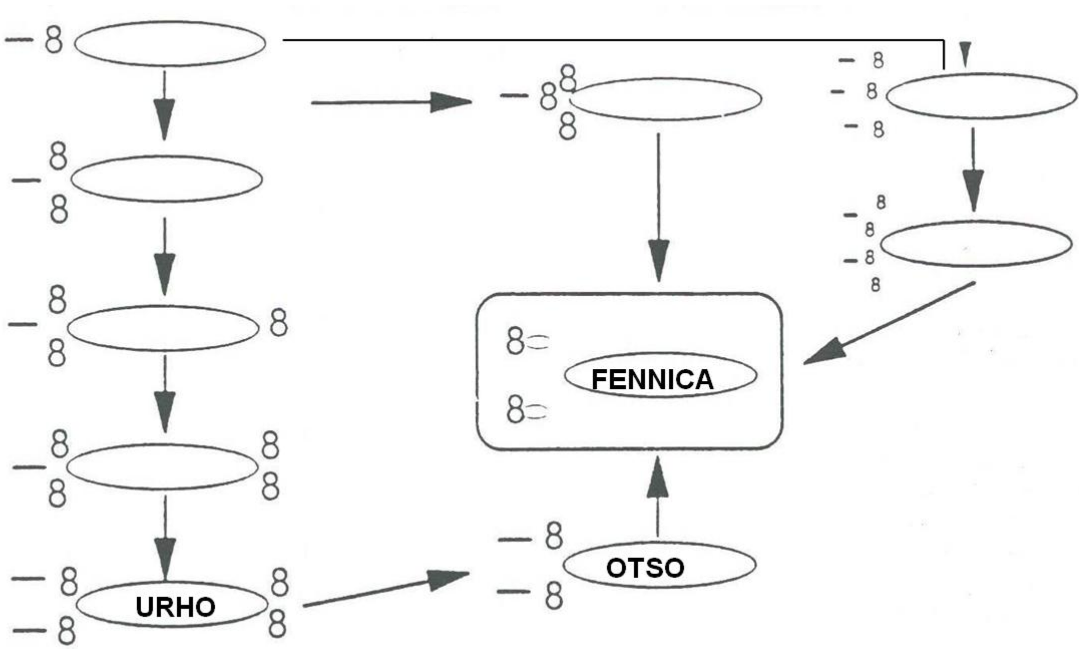


Fig. 19. The development of propulsion systems for icebreakers (modified from the original by G. Wilkman).

Fig. 20. MSV Fennica in drydock azimuthing thrusters clearly visible. These thrusters with nozzles were manufactured by Aquamaster, at present part of the Rolls-Royce group.

Fennica and Nordica were pioneering also in the sense that propellers in nozzles were used. The nozzle offers an added thrust at lower speeds (about 35 % increase in thrust compared with an open propeller at the same power in slow speeds) but their drawback is that they tend to get full of ice. When this happens, the thrust disappears and either the thrust must be reversed to flush the nozzle or – in case of azimuthing thrusters – the units turned around. Nozzles have been used earlier (e.g. Canadian bulk carrier MV Arctic and the supply ship CANMAR Kigoriak) but Fennica was first escort icebreaker where nozzles are used.

Since the advent of the azimuthing thrusters, also podded drives have been developed. Podded propulsion unit is an azimuthing thruster where the electric propulsion motor is in the hub of the propeller. The ABB Azipod was the first manufacturer of podded drives with many deliveries to ice going ships. The advantage of the podded drive as compared with other azimuthing solutions is that the space required for the engine room is smaller. A sketch of the diesel electric machinery layout with podded drives in shown in Fig. 21. The first Azipod icebreaker, MSV Botnica, was delivered in 1998 and is shown in Fig. 22.

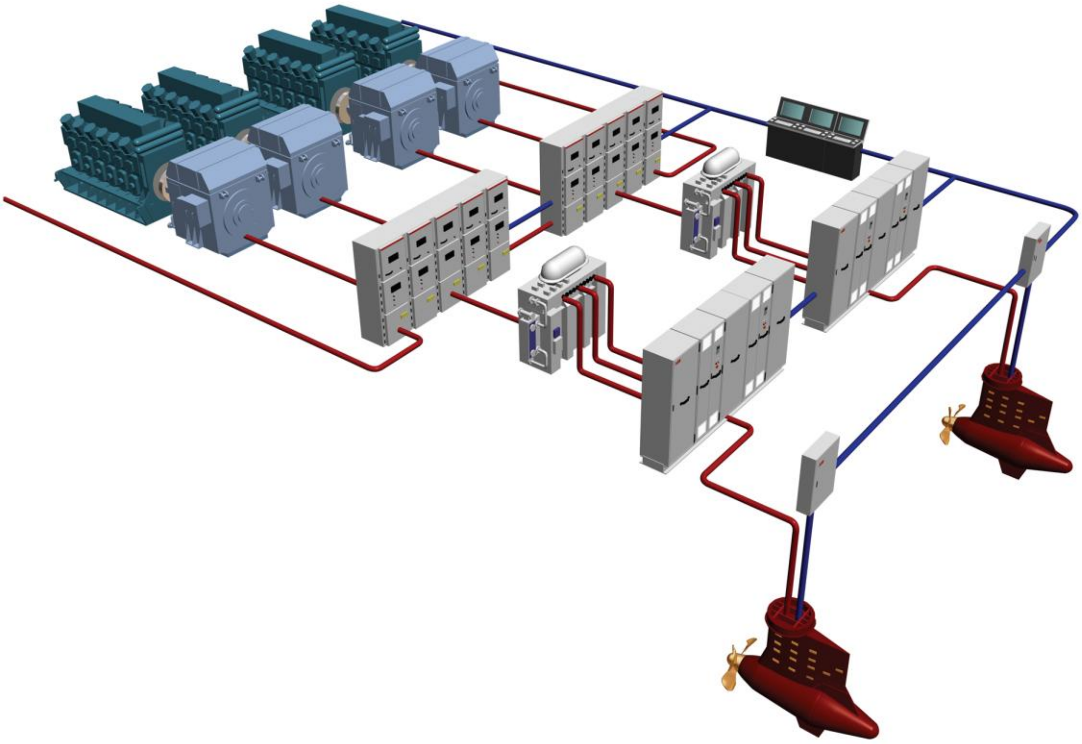


Fig. 21. Machinery layout in a twin Azipod system (ABB Marine).

Fig. 22. Multipurpose icebreaker Botnica.

Azimuthing propulsion improves the manoeuvrability of ships greatly as thrust can be directed in any direction. Typical manoeuvres that can be accomplished with ships having azimuthing thrusters are a turn on spot and ridge breaking moving astern. When MSV Fennica was in the first ice trials, it was noted that the ship can maintain a continuous speed in thick ridges by moving the thrusters from side to side. This way the ice floes forming the ridge break loose and the ridge can be dispersed. The effect is even more pronounced when moving astern as then the propeller wake flushes the ice floes very efficiently. This observation has led to the concept of dual mode ships; ships that go astern in heavy ice and forward in open water or light ice. In dual mode ships the bow and stern can be optimised to specific conditions. These ships have a cross over point on the thickness-speed plots where the stern first operation becomes more efficient, see Fig. 23. The cross over point can be explained in terms of the thrust deduction factor; in open water the thrust deduction is large when moving astern (or propeller first) whereas in ice the thrust deduction factors astern can be negative, see Leiviskä (2004).

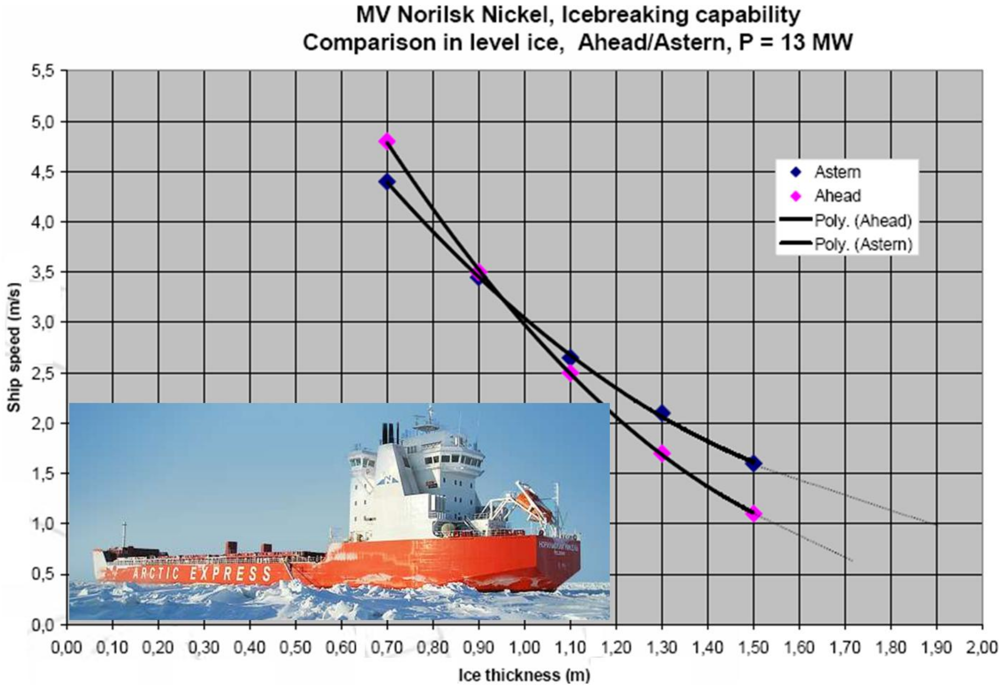


Fig. 23. The speed of the Norilsk Nikel ahead and astern showing the cross over point clearly (Gorshkovskij & Wilkman 2007).

The propulsion system general design is based on balancing economic matters with the required performance. Diesel electric machinery with azimuthing thrusters (either podded or direct drive) seems to offer clearly the most advantages – this solution is also most expensive. Thus the most common application in merchant ships is a diesel direct drive with a CPP. As the requirements for icebreaking ships are more stringent, the diesel-electric drive is common in these.

5. Hull and Machinery Strength

Design of ship hull structures requires knowledge of the ice loads acting on different regions of the ship hull. Even if the structural design usually follows class rules, it is important in conceptual design phase to have an idea about the magnitude of ice loads and quantities describing the loads.

Definition of Local Ice Load

As ice loads arise from contact with an ice edge, it is commonly assumed that the load acts mostly on a load patch (area of non-zero ice pressure) that is narrow in vertical direction and long in horizontal direction. In case of an impact with multi-year ice of rounded shape, the load patch can be of more irregular shape. Load patch is idealized as a rectangular patch for structural response calculation of local shell structures like plating, main frames, stringers and web frames. This idealization is sketched in Fig. 24.

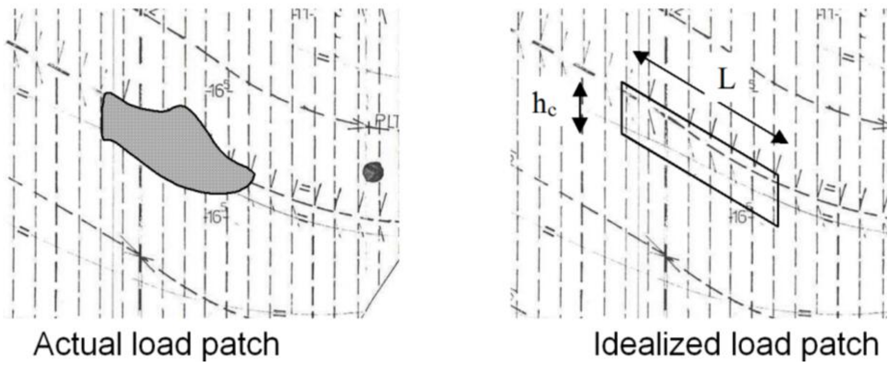


Fig. 24. Actual load patch and its idealization for structural design.

The nature of the load patch indicates structural idealizations that can be used in simple response calculations; these are shown in Figs. 25a,b. From this load patch idealization it is clear that vertical frames derive their loading from one frame spacing s; if the average ice pressure on the load patch is p_c, then the frame load is

$$F = p_c \cdot h_c \cdot s ,$$

where h_c is the load height. On the other hand, the horizontal frames may be loaded along the whole frame span L (L - web frame spacing) and thus the frame total load is

$$F = p_c \cdot h_c \cdot L .$$

The figures describing the design load patch show also that when designing any structural member, the load patch is placed at a location giving the largest response – for plating symmetrically at the centre of the plate field and for frames at the midspan. The Figs. 25a,b show also the structural idealization that can be used in estimating the response – attention should be given especially to the boundary conditions used. The use of simple structural idealizations is justified in case of ice loading as the advantage of more advanced methods disappears in the uncertainty concerning the ice load values.

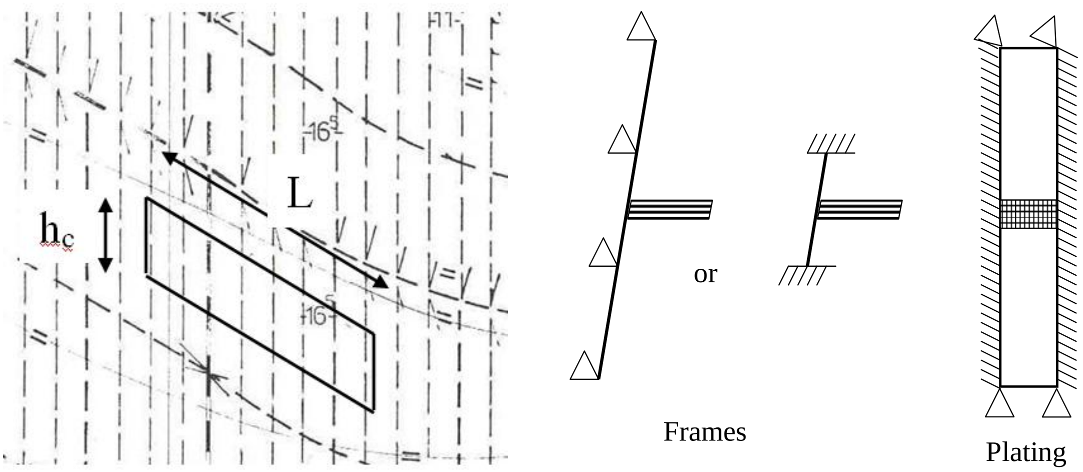


Fig. 25a. Ice load patch on a transversely framed shell structure, and the structural idealizations used in calculating the response.

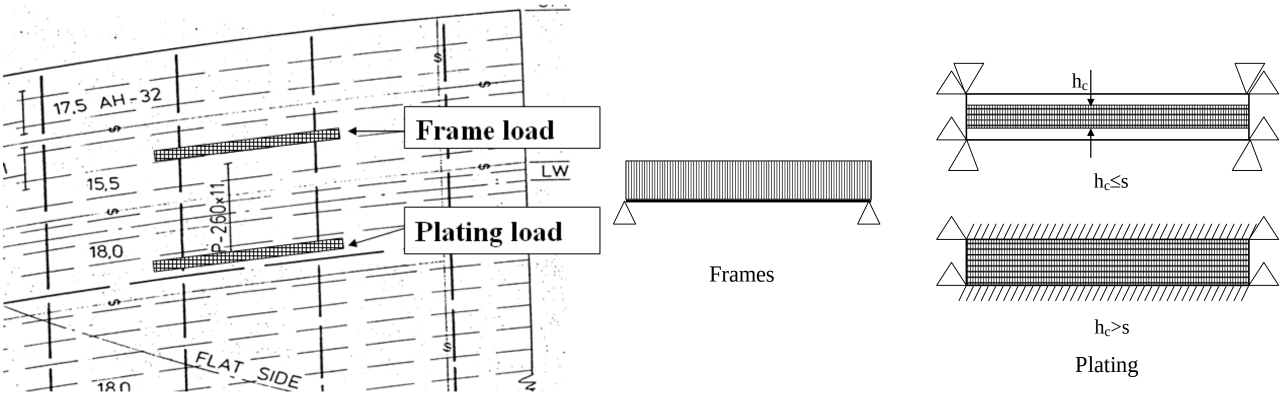


Fig. 25b. Ice load patch on a horizontally framed shell structure, and the structural idealizations used in calculating the response.

The simplification of the load patch suggests that there are three quantities describing the local ice load; pressure p_c, load height h_c and load length. These are analyzed briefly in the following but first a note about how to obtain these quantities. Ice pressure is not measured directly; it is always ice force F that is measured on a certain gauge area A_g and then the pressure is deduced as F/A_g. The force is mostly the normal force on the area. Gauge areas used have been very different, gauges of area about 1 cm^2 to several m^2 have been used. Load height and load length are difficult if not impossible to measure but the quantity of load length $q = p_c \cdot h_c$ can be measured based for example on the response of transverse frames as the load height does not influence the frame response much if it is clearly less than the frame span. The measurement of the line load value using frames is an example of measuring the whole load F. This can be done sometimes by measuring the response of the whole structure. It can thus be concluded that the quantities that can be observed relatively well are the ice force on certain gauge area and sometimes the line load value q. Other quantities are obtained based on reasoning concerning their physical nature. A short description of this reasoning is given.

Ice pressure

How the ice pressure is conceived has varied much and there still is quite large controversy how to treat it. Often ice pressure is described by the average pressure on the area considered. Usually this area is the gauge area but also some geometric considerations may determine the area – if for example the load is observed on a pile of straight face towards level ice, then this area can be assumed to be $D{\cdot}h_i$ where D is pile diameter and h_i ice thickness. Observations of the ice pressure on smaller areas have suggested that considerable variation in local ice pressure magnitude exists inside the nominal contact area. The nominal contact area is defined by the geometry of the cross section between the ice feature and the structure – like the area $D{\cdot}h_i$ mentioned above. Several different theories about ice pressure have been suggested.

The earliest model for ice pressure is to treat it uniform and proportional to compressive strength of ice (Korzhavin 1971). The proportionality factors depend on the shape of the contact surface and on ‘quality of contact’ whereas the dependence on ice temperature and strain rate was included in the definition of ice compressive strength. In the 1970’s much research was done to clarify these proportionality factors (see for example Cammaert and Muggeridge 1988) but when it was realized that the measured compressive strength of ice depends much on the testing methods and specimen preparation quality (see e.g. Kendall 1978 and Tuhkuri 1996), the popularity of the use of the Korzhavin Equation has diminished.

The highest values of ice pressure are coupled with ice failure by crushing. ‘Crushing’ is a general description of ice failure into small particles. As ice must be broken along the whole contact surface, it is clear that some flow of crushed ice from the centre of the contact must take place. Russian scientists have analyzed the flow of crushed ice assuming that the crushed ice is viscous fluid. The situation of the flow is depicted in Fig. 26. Based on this assumption and Reynolds thin film fluid flow equations the following form for the pressure have been derived (Kurdjumov & Kheisin 1976, Popov & al. 1968)

$$p \propto \left(\left(\frac{h_c}{2} \right)^2 - x^2 \right)^{1/4} .$$

The proportionality factor depends on (empirical) ice strength obtained from drop ball tests on ice, the indentation speed and film thickness. The fluid viscosity and ice strength have been combined into the empirical ice strength factor. This form of ice pressure has been used to develop a formulation for ice force using energy principles in an impact between an ice feature and a ship (Popov et al. 1968). The drawback of this ice pressure formulation is that many assumptions have been made (viscosity, uniform film thickness, uniform source of crushed ice, constant thickness of the film to mention a few). Thus this formulation has not gathered much use outside Russia except in the development of the new ice class rules of International Association of Classification Societies (IACS) – of these rules more later.

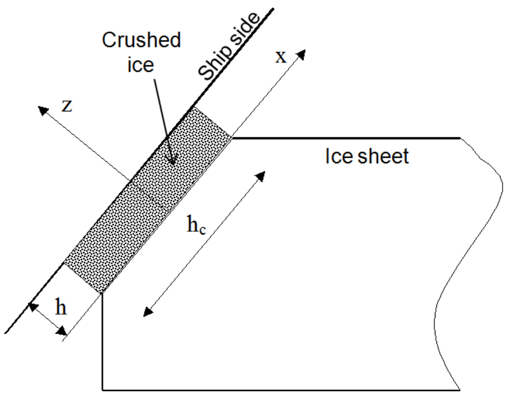


Fig. 26. Geometry of the assumed viscous layer of crushed ice.

The third formulation used for ice pressure is based on observation that the average ice pressure on an area is dependent on the magnitude of the area. Sanderson (1988) has collected many different results and then suggested the upper limit for this pressure-area relationship as

$$p_{av} = 8.1 \cdot A^{-0.57} ,$$

where p is in units of MPa and A in m^2. The constant and exponent in this pressure-area relationship has been studied for example by Riska (1987) and Frederking (1999) – the presented values for the constant vary between roughly 2 ... 10 and for the exponent between -0.3 ... -0.6. The Pressure-area relationship has been observed from very small areas (see Fig. 27.) to large areas of the magnitude about 100 m^2 and a similar trend prevails. The largest possible contact pressure most probably is set by the phase change of ice to liquid – this occurs at about 100 MPa in a temperature of about -10°C.

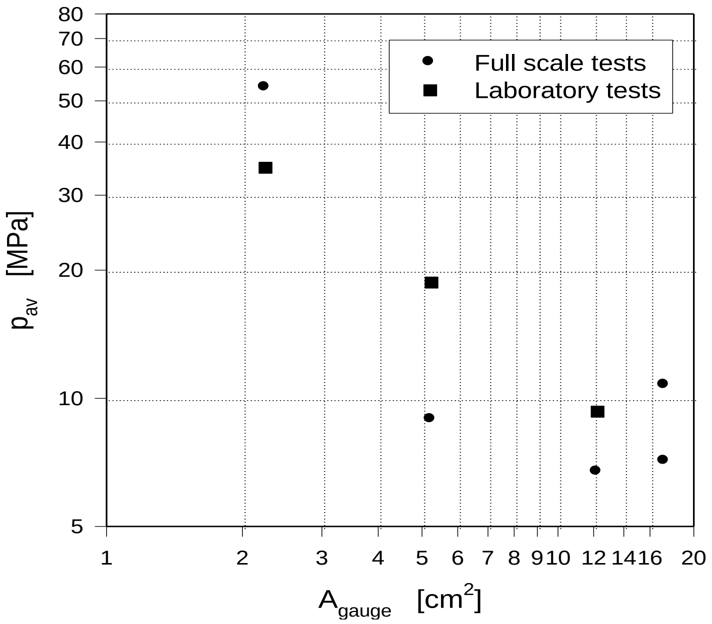


Fig. 27. Measured ice pressures in laboratory and in full scale ship trials on small gauge areas (Riska et al. 1990).

The drawback of the pressure-area relationship is that it is empirical and little physical basis exists for the area dependence. One possible reason for the pressure-area relationship is based on the observation that within the nominal contact area there is a line-like feature along which the ice pressure is transmitted (for the first observations, see Riska et al. 1990). This phenomenon has been investigated more thoroughly in the JOIA-project conducted in Japan (Sodhi et al. 1998). In Fig. 28. shows a result of an ice indentation test (Frederking & Sudom 2008, Muhonen 1991) and an explanation for the line-like features (Daley 1991, Tuhkuri 1996). The line is produced by a flaking process leaving a line on which a high pressure is acting. The flakes seem to be created so that the line of high pressure is directed towards the corners of the nominal contact area (Riska et al 1990).

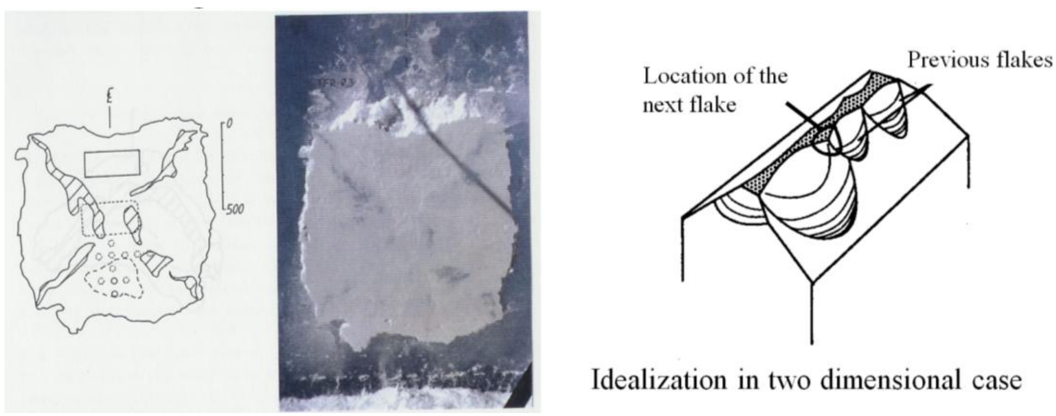


Fig. 28. Observation of structure-ice contact in the test at Hobson's Choice ice island (Muhonen 1991) and the idealization producing the line (Daley 1991).

A simplification of the situation in Fig. 28 is achieved by assuming that the line load (pressure x width) to be constant, say, q. This way the pressure and width need not be specified. This simplification leads to a line structure on a rectangular nominal area of width D and height H as shown in Fig. 29. The average pressure on the whole nominal area can now, if the line width is assumed to be small compared with other dimension, be presented as

$$p_{av} = q \cdot \left(\frac{2\sqrt{2}-1}{\sqrt{C}} - \sqrt{C} \right) \cdot A_{nom}^{-0.5},$$

where the aspect ratio is $C=D/H$. Thus the pressure-area relationship results from the assumption of the line-like nature of the contact. This is still a crude simplification as there might be some pressure acting outside the line where, apart from flaking, extrusion of crushed ice may occur. The assumption of the line can also be investigated making a thought experiment where the ice pressure is measured with gauges of different area but symmetrically located, see Fig. 30. The ice pressure is assumed to act on a line of somewhat nonuniform width – here also the pressure-area relationship emerges with the pressure exponent of -0.5.

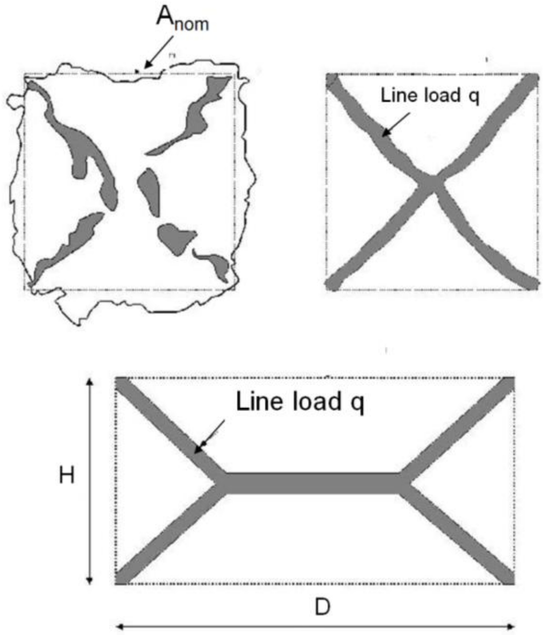


Fig. 29. The interpretation of the Hobson's Choice ice island results and the assumption of the line structure shape.

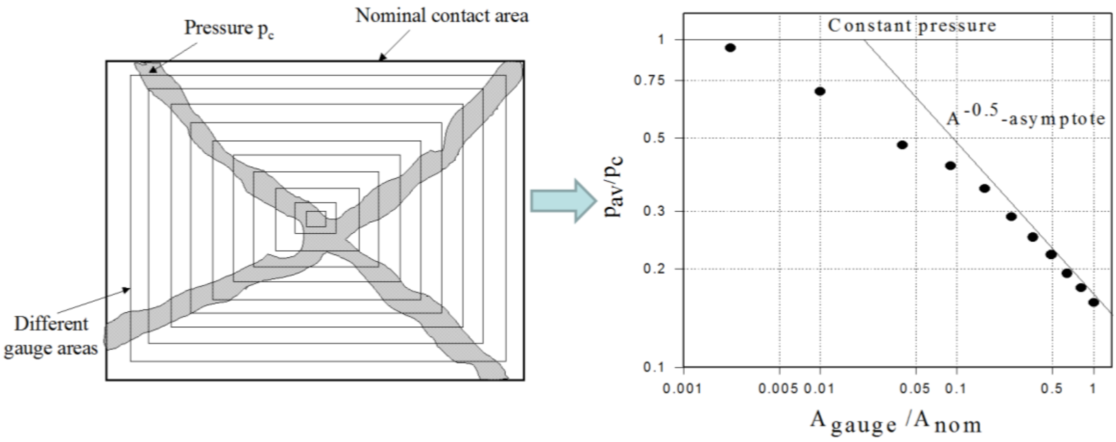


Fig. 30. A fictitious experiment of measuring the ice pressure with different gauge areas and the resulting average pressure results.

After presenting the three current views on ice pressure, it should be said that the pressure-area relationship is the one used most. Also some combinations to determine the pressure exist like a combination of the Korzhavin proportionality of ice pressure on compressive strength and the shape given by the hydrodynamic model. Apart from the hydrodynamic model there are not many suggestions for pressure distribution. The 'hot spot' theory where the ice pressure is conveyed by isolated small spots (Frederking & al. 1990) is one and another one suggests that the structural flexibility influences the pressure distribution (Riska et al. 2002). One additional problem for ice pressure is the observation that ice temperature and indentation rate influences the failure mode (see for example Sodhi et al. 1998). It has been suggested that in lower temperatures and/or higher indentation rates the ice fails in a brittle fashion forming flakes and a line-like contact whereas in higher temperatures and slower indentation rates the failure is ductile and the whole nominal contact area experiences pressure but much lower than that along the line. In the ductile case observations (Sodhi et al. 1998) suggest that the total load on the contact area is higher than in the brittle case but no conclusive results exists. Different values for the transition rate and temperature have been suggested and the present author concluded, based on a literature survey that the transition point is at -10^{o}C / 5 mm /s.

Load Height

The dimensions of the load patch are difficult to determine. In some cases an estimate of the load length L or the load height h_c can be given but usually the designer must assume most disadvantageous values for the dimensions. An example of the geometric reasoning for dimensions is given by the conceived load height in the Finnish-Swedish ice class rules. In the 1971 rules the magnitude of the load given by the line load q was determined based on an extensive ice damage survey (Johansson 1967). Damage survey gave the value of roughly q = 2 MN/m. In the rules the load height was assumed to be about the ice thickness and a value of 800 mm was selected. This selection caused an underestimation of the loading for plating, especially in longitudinally framed structure. Thus in the new rule set from 1985 the load height was reduced but the line load value - as this is based on observations - was kept constant, see Fig. 31. This meant increased ice pressure values. Now, if the ice load acts along a narrow line, the load height should be further decreased and pressures increased but this application has not been done.

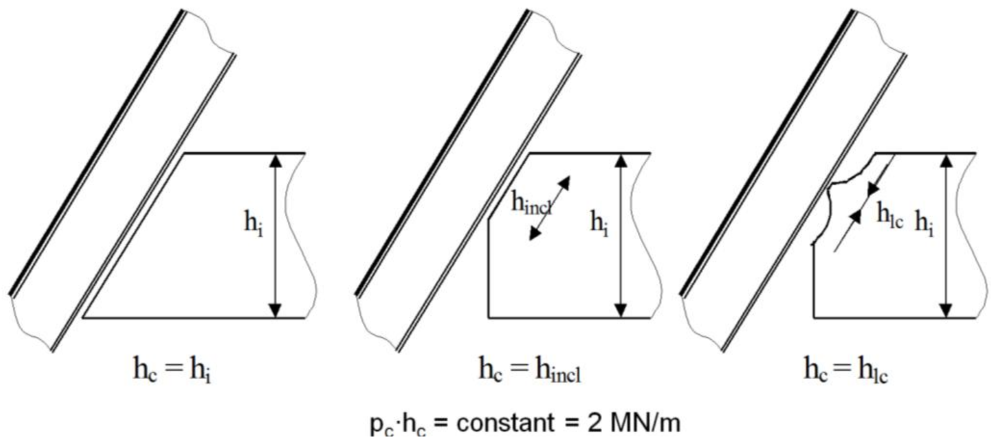


Fig. 31. Development of the concept of load patch height.

Total Ice Force

The total ice force F or its normal component F_n may be determined by analysing the motion of the colliding bodies (ice feature and ship). The analysis must include some concept of the variation of the ice load versus the indentation depth. Two such cases for calculation of the total ice load (and partly disregarding the load patch dimensions) have been given. The first one (Popov et al. 1968) investigates the collision of two bodies and deduces an equation for indentation along the normal to the contact area. The collision case for which this model is suitable is so called oblique collision where the ship collides with a smaller ice floe and the collision area is on one side of the bow. This equation that is essentially one dimensional is then solved and the maximum force is deduced. This simplified approach assumed a constant pressure and Daley (2001) incorporated the pressure-area relationship into this essentially energy based approach. The resulting force is

$$F_n = C \cdot p_0^{\ 0.36} \cdot v_{ship}^{\ \ 1.28} \cdot \Delta^{0.64},$$

where C is a factor containing the geometric information at the contact, p_0 is the constant in the pressure-area relationship, v_{ship} ship speed and Δ ship displacement.

Another case where the total ice load has been calculated is normal collision on a multi-year ice floe. This case includes crushing of the ice edge followed by the ship sliding up onto the ice. For a collision where the ice mass is assumed large compared with the ship displacement, the force has been deduced as (Riska et al. 1996)

$$F_n = C \cdot \sin^{0.2} \varphi \cdot \sqrt{\Delta A_{wp}} \cdot v_{ship},$$

where the constant C contains the dependency on ice strength. This calculation is based on the theory developed in Riska (1987). The above calculation of the collision force can be used to calculate the shear forces and bending moments on the ship hull.

Design Point

The concept of the design point includes the allowed structural response and how frequently it is reached. The allowed structural response may be maximum stress up to yield point (or somewhat smaller stress if safety factors are used), fully plastic stress which gives the onset of permanent deformation or also some minor but clearly defined damage. Once load is of statistical nature as most marine loads are, the allowed structural response must be coupled with some estimate of the frequency of the design loading. This process of finding the design point involves a description of the statistics of the loading. The description should include a prediction of extreme values coupled with the return period of these values; this process is illustrated in Fig. 32 for frame ice loads in the Baltic. Here an important observation can be made viz. for wave loading the statistical quantity is the wave spectrum whereas for ice loads there does not, as yet, exist any relationship between the ice conditions and ice loading. In this case the statistical analysis of ice loading is mostly carried out using measured ice loading.

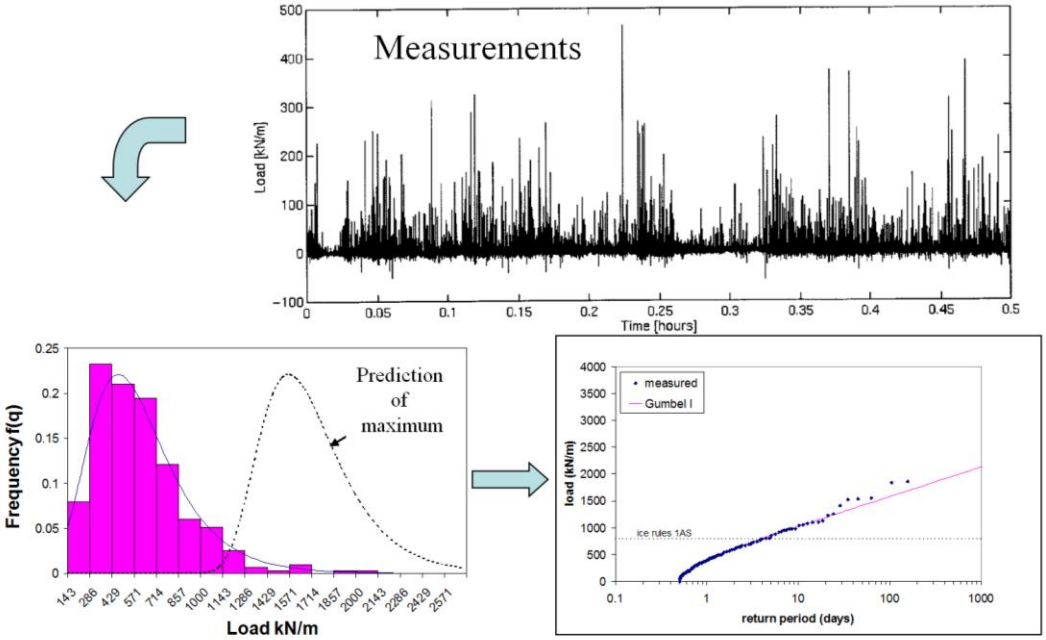


Fig. 32. Statistical analysis of frame ice loads based on measurements of frame ice loads in the Baltic, data from Muhonen (1991).

Next step in the development of the design point is to analyse the structural response for the foreseen loading. One question here is how the loading is to be described. For ice loads it would be natural to use the ice pressure as the quantity for statistical analysis but the problems associated with the ice pressure – and the need for long time series of measurements – makes the frame ice load more suitable for statistical analysis. The structural analysis gives the relationship between the maximum allowed response, say, w, the load quantity, say, q and the structural dimensions i.e. scantlings (for plating the plate thickness t). This relationship can be stated as

$$w = f(q;t)$$

where the function $f(\cdot)$ is determined by structural analysis using e.g. FEM. Using this relationship, the scantlings resulting from certain structural limit and return period of load can be determined. This process of obtaining the plate thickness is sketched in Fig. 33. It is a matter of opinion what structural limit to use and how often it is allowed to be reached. Fig. 34 shows that it might happen that different limits (in the figure w_0 and w_1) give similar scantling values if the frequencies are selected in a balanced way.

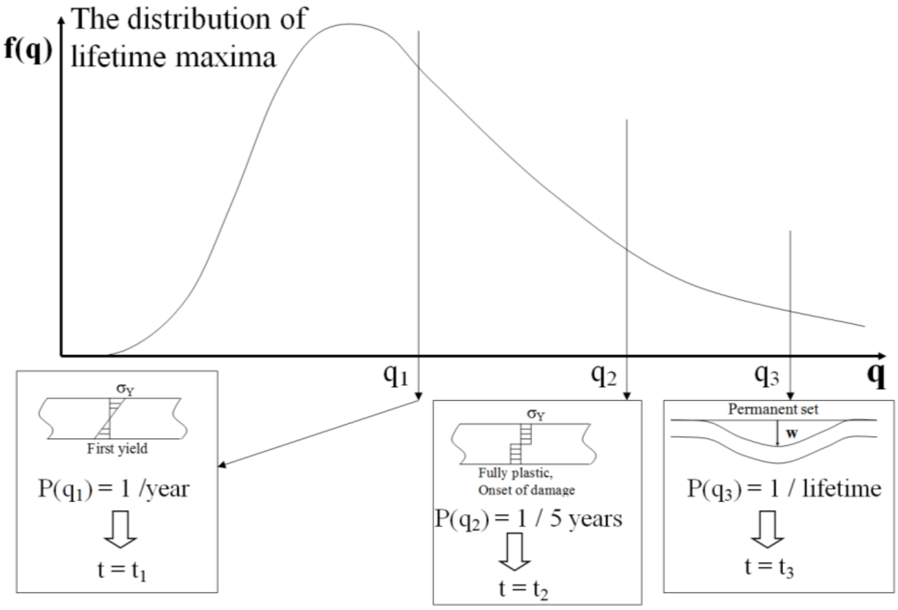


Fig. 33. Process of determining the structural dimension (scantling), which in this example is the plate thickness, using different structural limits and at the same time different occurrence frequencies for the limits.

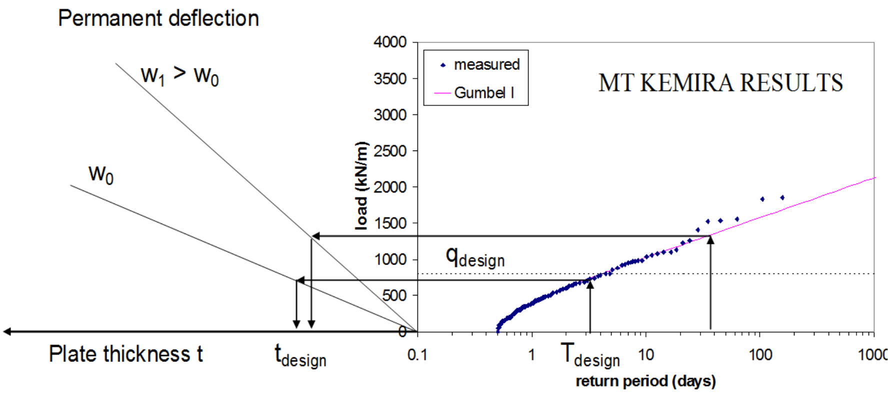


Fig. 34. Selection of scantlings using two different structural limits and two return periods for the loading.

Due to the longitudinally elongated load patch it is clear that vertical framing gives a lighter structure than longitudinal framing. The situation may change if the design point is based on plastic deformation or smaller damage as then the large plastic reserve in plating is utilized. In plastic design the load frequency and how often certain size of damage is allowed must be balanced; not much knowledge about this balance exists.

Machinery Loading

In order to achieve adequate strength in ice of the ship machinery i.e. the shaftline consisting of propeller(s), shafts, gears and couplings, and the main engine, all the shaftline components should have adequate strength. The ice loading on the shaftline stems from impacts of broken ice floes on the propeller blade. The design point of the propeller blades is thus an impact with an individual ice floe. The magnitude of the force on the propeller is determined by the mass of the ice floe and ice strength as well as the impact speed set by the rotation rate (n) and diameter (D) of the propeller. If the propeller is submerged at an adequate depth i.e. has an adequate ice clearance (see Fig. 35), the propeller will never be in contact with unbroken ice. Large propeller clearance with the hull (stern frame clearance in Fig. 35) reduces also propeller ice loading.

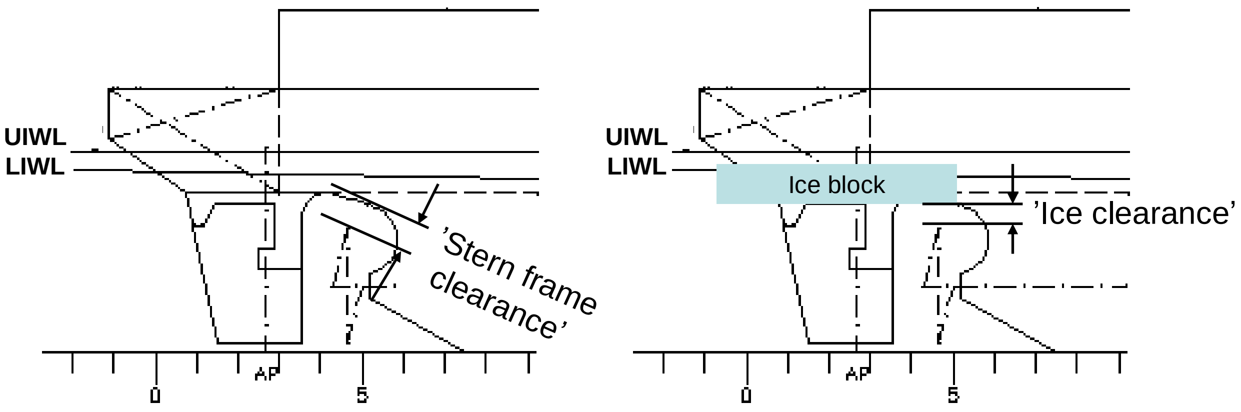


Fig. 35. Definitions of 'Stern frame clearance' and 'Ice clearance' (Guidelines 2010).

As *RPM* of the propeller is quite high, it often happens that several blades hit the same ice floe – this is called milling, see Fig. 36. Milling results in a pulsating load on the blade with a main frequency component determined by *RPM* and number of blades (Z) – this is called blade frequency. As the blade frequency is of the same order of magnitude as the lowest shaftline natural frequencies, the pulsating blade load excites a dynamic torque in the shaft. This response on the shaft is also shown in Fig. 36. It is clear from this discussion of the machinery ice loading that even if the propeller blades react in a static fashion to ice loads, the shaft reacts dynamically. As each shaft is different from others, it is difficult to give design torques for shafts, only the loads on the propeller and the torque excitation are given and the designer must then do the structural analyses for his/her design.

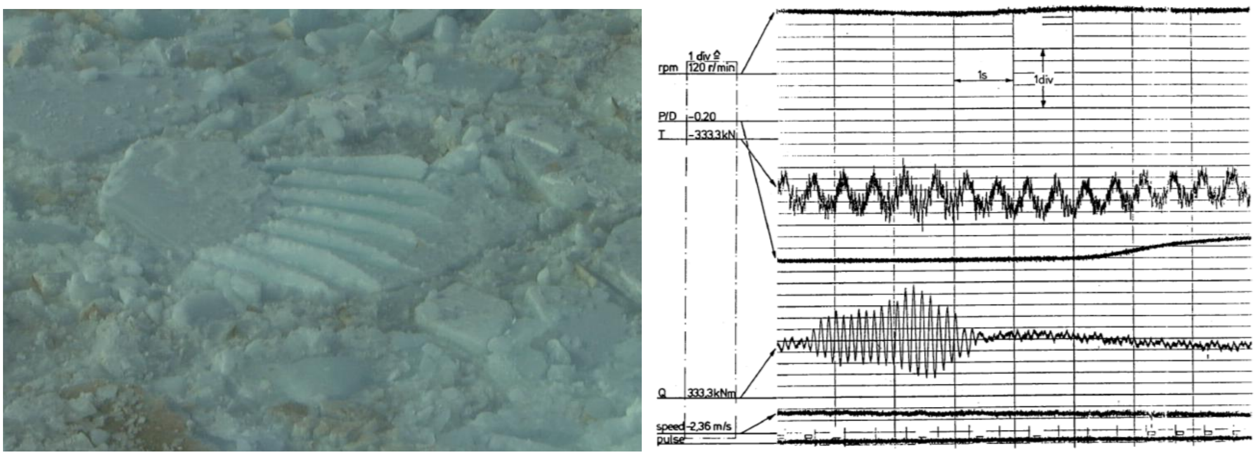


Fig. 36. An ice floe milled by several impacts of propeller blades (photo: K. Riska) and the resulting shaft torque signal Q (Koskikivi & Kujala 1985)

A formulation of the amplitude of the design torque at the propeller has been given in Browne & Norhamo (2007). The torque amplitude value is divided into two formulas according to the propeller diameter – the limit depends on the ice floe thickness (ice floe dimensions are assumed to be H_{ice} x 2 H_{ice} x 3 H_{ice}):

$$Q_{\max} = k_{open} \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot [n \cdot D]^{0.17} \cdot D^3 \quad \text{[kNm]}, \quad D < 1.8 H_{ice}$$

$$Q_{\max} = 1.9 \cdot k_{open} \cdot \left[1-\frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot [n \cdot D]^{0.17} \cdot D^{1.9} \cdot [H_{ice}]^{1.1} \text{ [kNm]}, \; D \geq 1.8H_{ice}$$

where d is the propeller hub diameter, $P_{0.7}$ propeller pitch [m] at 0.7·D/2 and n rotational propeller speed [rps] at bollard condition. The factor k_{open} describes the severity of the operational area; its value is between 10 ... 15.

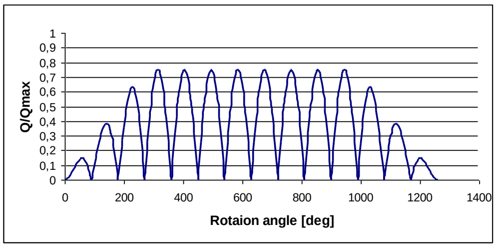


Fig. 37. Design torque time history at the propeller for shaftline design (FSICR 2008)

If the blade frequency is not very close to any shaft natural frequency, the following estimation of the maximum torque for any component in the shaftline (component 'r') can be used (FSICR 2008);

$$Q_r = Q_{e\max} + Q_{\max} \cdot \frac{I}{I_t},$$

where I is the equivalent mass moment of inertia of all parts on the engine side of the component under consideration and I_t is the equivalent mass moment of inertia of the whole propulsion system. The maximum torque given by the engine is denoted as Q_{emax}.

The design forces of the propeller blades are determined by the size of the impacting ice floe. Several, mostly empirical investigations of these propeller blade loads have been conducted (see Koskinen & Jussila 1991 and Marquis & al. 2008). Measurements have suggested a formulation for blade loading which has been adopted by several classification societies' ice rules (about ice rules more later). Again the force formulas are divided into two parts according to ice thickness – also different formulation is valid for forces forward and astern. The formulas are for backward blade force (FSICR 2008)

$$F_b = k \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot D^2, \text{ [kN], when } D \leq 0.85 \cdot H_{ice}{}^{1.4}$$

$$F_b = k \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot D \cdot H_{ice}{}^{1.4}, \text{ [kN], when } D > 0.85 \cdot H_{ice}{}^{1.4}$$

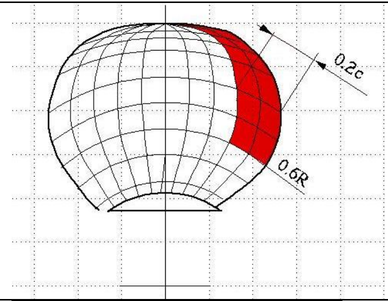
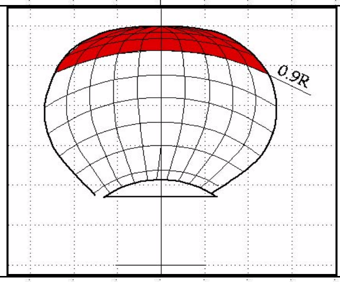
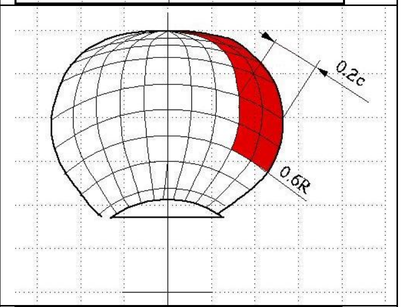
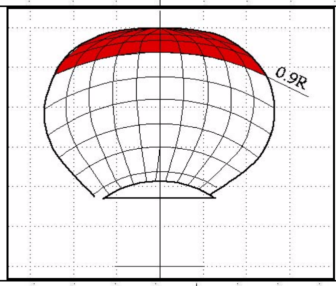
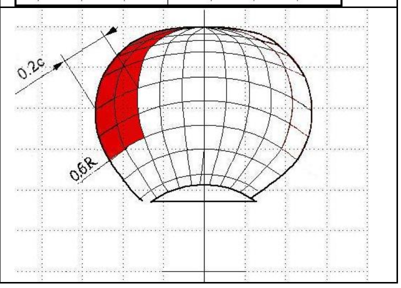
where EAR is the expanded blade area ratio and k a factor depending on the severity of ice conditions, $k = 20 \ldots 30$. Similarly for forward blade force

$$F_f = k \cdot \left(\frac{EAR}{Z}\right) \cdot D^2 \text{ [kN], when } D \leq \frac{2}{1-\frac{d}{D}} \cdot H_{ice}$$

$$F_f = k \cdot \left(\frac{EAR}{Z}\right) \cdot D \cdot \frac{1}{1-\frac{d}{D}} \cdot H_{ice} \text{ [kN], when } D > \frac{2}{1-\frac{d}{D}} \cdot H_{ice},$$

where the factor k depends on the severity of ice conditions and can have values k = 250 ... 500. The above ice forces are applied to the propeller blade and the resulting stresses should be lower than a preset limit. Modern propeller blades are not simple beams and thus simple beam idealization is not correct, some more advanced methods like FEM must be applied to determine the stresses. The distribution and location of the ice force has a large influence on the response. Thus the load patch size and location must be determined. The load cases and the associated load patch sizes are given in Table 1.

Table 1. Load cases for the propeller blade ice loading (FSICR 2008)

	Force	Loaded area	Right-handed propeller blade seen from behind
Load case 1	F_b	Uniform pressure applied on the back of the blade (suction side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 2	50% of F_b	Uniform pressure applied on the back of the blade (suction side) on the propeller tip area outside 0.9R radius.	
Load case 3	F_f	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 4	50% of F_f	Uniform pressure applied on propeller face (pressure side) on the propeller tip area outside 0.9R radius.	
Load case 5	60% of F_f or F_b, whichever is greater	Uniform pressure applied on propeller face (pressure side) to an area from 0.6R to the tip and from the trailing edge to 0.2 times the chord length.	

Determination of the propeller blade ice forces and the shaftline torque values forms the essence of ship machinery design for adequate strength in ice. There is, however, one final comment to be made about the design point along the shaftline. The principle of progressive strength (strength hierarchy or pyramidal strength) should be followed in designing the shaft and its components. Progressive strength means that the components get progressively stronger when proceeding from the propeller towards the main engine. Thus the propeller blade is the weakest link in the shaftline. If excessive loading is applied, the propeller blade fails and acts as a 'fuse' for the other components. The propeller blade is the easiest component to be replaced in the shaftline – the propellers of ice going ships are made of blades that are bolted to the propeller hub (ships commonly carry replacement blades).

Ice Class Rules

The determination of scantlings of ship structures and also more generally the design of ship structures follows some rules that classification societies have given. The classification societies and also some maritime authorities (Finnish and Swedish Maritime Administrations and Transport Canada) have developed rules for designing ice capable ships. These ice class rules define several different ice classes depending on the severity of ice conditions. Ice class rules define the scantlings of the hull and shaftline structures and give some requirements for ship performance in ice and structural arrangement. At present there are three main sets of ice class rules: the Finnish-Swedish Ice Class Rules (FSICR), the Russian Maritime Register of Shipping (RMRS) ice rules and the unified Polar Class (PC) rules of the International Association of Classification Societies (IASC).

The FSICR (2008) contain requirements for ship hull, ship machinery and also for ship performance in ice. Four different ice classes are defined and also the open water ships have their own ice class notations (II and III). This is because the fairway dues are dependent on the ice class – higher ice class ships pay less fairway due as these ships use less icebreaker support. The Finnish-Swedish ice classes are

1. ice class IA Super; ships with such structure, engine output and other properties that they are normally capable of navigating in difficult ice conditions without the assistance of icebreakers, maximum level ice thickness 1.0 m;
2. ice class IA; ships with such structure, engine output and other properties that they are capable of navigating in difficult ice conditions, with the assistance of icebreakers when necessary, maximum level ice thickness 0.8 m;
3. ice class IB; the same as above for ice class IA except that maximum level ice thickness 0.6 m;
4. ice class IC; the same as above for ice class IA except that maximum level ice thickness 0.4 m;
5. ice class II; ships that have a steel hull and that are structurally fit for navigation in the open sea and that, despite not being strengthened for navigation in ice, are capable of navigating in very light ice conditions with their own propulsion machinery;
6. ice class III; ships that do not belong to the ice classes referred to in paragraphs 1-5.

FSICR are intended for ships navigating in the Baltic following the operational practice used there i.e. ships are escorted by icebreakers in the worst ice conditions. The design point in the FSICR is the elastic limit; and the scantling equations have been modified through the years so that the damage frequency has reached an acceptable level. Measurements of the structural response of the hull structures have shown that the yield point is reached about once a week (Muhonen 1991); also that the yield point in plating is reached more often than in the frames – this suggests a correct structural hierarchy in FSICR. The highest machinery and hull loads and the performance requirement do not have a common design ship-ice interaction scenario as the largest ship response occurs in different kinds of scenario. The design scenarios for hull, machinery and performance are stated in Table 2.

Table 2. Design ship-ice interaction scenarios used in FSICR.

Hull	Impact with ice level ice of thickness h_0	The ship can encounter thick level ice in ridges where the consolidated layer can be 80 % thicker than the level ice thickness. Also channel edges can be very thick.
Propulsion machinery	Impact with large ice floes	Propellers encounter only broken ice and the design scenario is an impact with these floes. Large ice floes can be encountered among the level ice floes for example in old channels.
Propulsion power	Ship must make at least 5 knots in a specified brash ice channel	Ships must be able to follow icebreakers at a reasonable speed and also to proceed in old brash ice channels independently at reasonable speeds.

The Finnish ice class rules have evolved since the first rules published in year 1890. The first rules gave just requirements for the general arrangement. The first rules for scantlings were published in 1920. These were so called ‘percentage rules’ as the scantlings were increased a certain percentage from the open water values. These rules were slightly modified in 1932 and 1962. When the year-round navigation to all Finnish ports started in 1960’s, the ship damages due to ice started to increase sharply. This experience from ice damages led to new ice rules in year 1971 – these were the first joint Finnish-Swedish rules and also the first modern ice rules in the sense that the ice load was stated explicitly. These rules have been revised several times (1985, 2002 and 2005) and the present rules stem from 2008.

The requirements for scantlings are based on ensuring an adequate safety of ships. The performance requirement (stated also as a powering requirement) is based, on the other hand, on ensuring an efficient winter navigation system. All ships fulfilling the requirement for an ice class set by Finnish or Swedish maritime authorities and bound to/from Finnish or Swedish ports get icebreaker escort. If the ice capability of ships would be low, many icebreakers would be needed to escort all ships (or the waiting times would be intolerably long), and the winter navigation system would be very expensive to maintain. Thus the merchant ships are required to have some ice capability so that the escort distances in ice will be shorter and escort speed higher.

The Finnish-Swedish ice class rules have been adopted by most of the classification societies (all except RMRS) – the FSICR have been described as an ‘industry standard’ for first year ice conditions even if they are intended only to Baltic. The classification societies follow their own notations, but the basic rules are the same as FSICR. The equivalent notations are stated in the table 3.

Table 3. Equivalent notations for the Finnish-Swedish ice classes.

Rule System	**Corresponding classes, Notation**			
Finnish-Swedish Ice Class Rules 2008	IA Super	IA	IB	IC
American Bureau of Shipping 2010 Pt. 6, Ch. 1, Sec. 2	I AA	I A	I B	I C
Bureau Veritas 2010 Pt. E, Ch.8	IA Super	IA	IB	IC

Det Norske Veritas 2010 Pt. 5, Ch. 1, Sec. 3	ICE-1A*	ICE-1A	ICE-1B	ICE-1C
Germanischer Lloyd 2010 Pt. 1, Sec. 15	E4	E3	E2	E1
Lloyd's Register 2010 Pt. 8, Ch. 2, Sec. 7	1AS	1A	1B	1C
Nippon Kaiji Kyokai (Class NK) 2010 Pt. I, Ch. 5	*IA Super*	*IA*	*IB*	*IC*
Registro Italiano Navale 2010 Pt. F, Ch. 9 Sec. 1	IAS	IA	IB	IC
Korean Register of Shipping 2010 Pt. 3, Ch. 20, Sec. 2-6	IA Super	IA	IB	IC
China Classification Society 2006 Pt. 2, Ch. 4	B1*	B1	B2	B3

The RMRS ice rules consist of nine ice classes – and additionally four ice classes for icebreakers. The merchant ship classes are presented in Table 4. The ice classes up to Ice5 are intended for first year ice and higher classes to Polar operations. The RMRS ice rules contain also three parts; hull and machinery and powering. The powering requirements for the Baltic are the same as the corresponding FSICR ice classes. The structural limit in the design point is full plastic response for plating and frames. The limit for stringers and web frames is yield. The RMRS rules are mainly used for ships with the Russian flag or ships operating in Russian waters.

Table 4. Russian Maritime Register of Shipping ice classes.

Ship category	Permitted thickness of ice, in m		Type of operation
	Independent navigation in open pack ice at a speed of 5 knots	Navigation in channel following an icebreaker in compact ice at a speed of 3 knots	
Ice1	0,40	0,35	Episodically
Ice2	0,55	0,50	Regularly
Ice3	0,70	0,65	Regularly

Ship category	Permitted speed, in knots	Ice concentration and type	Ice thickness, in m		Methods of surmounting ice ridges
			Winter/spring navigation	Summer/ autumn navigation	
Arc4	6 — 8	open floating first-year ice	0,6	0,8	Continuous motion
Arc5		open floating first-year ice	0,8	1,0	
Arc6		open floating first-year ice	1,1	1,3	
Arc7		close floating first-year ice	1,4	1,7	Episodic ramming
Arc8	10	close floating second-year ice	2,1	3,0	Regular ramming
Arc9	12	very close floating and compact multi-year ice	3,5	4,0	Surmount of ice ridges and episodic ramming of compact ice fields

The third set of ice class rules are the harmonized ice rules developed by IACS. These rules have been under development since mid 90's and in 2008 the rules were finally accepted. At the moment all IACS members are adopting these rules into their rule structure and deleting their old versions for polar classes. There are seven polar classes in IACS ice rules. These classes are described in Table 5 where the ice description follows the World Meteorological Organization's practice. Hull design in PC classes is based on plastic structural limit and it has been stated that the return period of the loads causing response up to the limit is one year. The machinery rules for PC classes are based on the same theory of ice loads as in the FSICR.

Table 5. Ice classes of the IACS unified ice rules.

Polar Class	Ice Description
PC 1	Year-round operation in all Polar waters
PC 2	Year-round operation in moderate multi-year ice conditions
PC 3	Year-round operation in second-year ice with old ice inclusions
PC 4	Year-round operation in thick first-year ice with old ice inclusions
PC 5	Year-round operation in medium first-year ice with old ice inclusions
PC 6	Summer/Autumn operation in medium first-year ice with old ice inclusions
PC 7	Summer/Autumn operation in thin first-year ice with old ice inclusions

The short survey of ice classes show that it is difficult to select an ice class based solely on the ice class descriptions. The ice class that a ship should have is in principle set by the ice conditions and the required safety level – but in practice the required ice class is decided by the requirements of the maritime authorities. In Finland and Sweden the maritime authorities set the required ice class for each port in the Traffic Restrictions. These requirements develop when winter proceeds. Russian and Estonian authorities follow roughly a similar procedure; only the requirements are slightly lower than to Finland and Sweden. The Canadian system is called the Arctic Ice Regime Shipping System (AIRSS 1996) – in this system an Ice Numeral is calculated based on the prevailing ice conditions and ship ice class, and if the numeral is negative, the ship cannot enter the area. The selection of a suitable ice class must take into account what the authorities require in different ice conditions.

HELCOM (Helsinki Commission i.e. an intergovernmental co-operation body) Ice Expert Working Group (Ice EWG) addressed the question of transparency in the requirements of maritime authorities. The working group developed a recommendation for the required ice class to be applied in the Baltic. The ice class requirement is to be based on the maximum level ice thickness as follows (Ice EWG 4/2003):

Level ice thickness 10 – 15 cm	Required ice class Ice1 or II
Level ice thickness 15 – 30 cm	Required ice class Ice2 or IC
Level ice thickness 30 – 50 cm	Required ice class Ice3 or IB
Level ice thickness >50 cm	Required ice class Arc4 or IA

At the same time the working group agreed on equivalencies between ice classes. The equivalency means that the maritime authorities (and in Finland also the fairway due system) treat equivalent classes in the same way. An equivalency is at best approximate when the rule formulations are different like in RMRS ice rules compared with FSICR – the equivalency is just a practical way forward. The equivalencies agreed by HELCOM Ice EWG are presented in table 6.

Table 6. The Equivalencies agreed by HELCOM Ice EWG (Ice EWG 4/2003). Some class notations have been since changed, for example the Russian ice class LU1 is now Ice1.

Classification Society	Ice Class				
Finnish-Swedish Ice Class Rules	**IA Super**	**IA**	**IB**	**IC**	**Category II**
Russian Maritime Register of Shipping (Rules 1995)	UL	L1	L2	L3	L4
Russian Maritime Register of Shipping (Rules 1999)	LU5	LU4	LU3	LU2	LU1
American Bureau of Shipping	IAA	IA	IB	IC	D0
Bureau Veritas	IA SUPER	IA	IB	IC	ID
CASPPR, 1972	A	B	C	D	E
China Classification Society	Ice Class B1*	Ice Class B1	Ice Class B2	Ice Class B3	Ice Class B
Det Norske Veritas	ICE-1A*	ICE-1A	ICE-1B	ICE-1C	ICE-C
Germanischer Lloyd	E4	E3	E2	E1	E
Korean Register of Shipping	ISS	IS1	IS2	IS3	IS4
Lloyd's Register of Shipping	1AS	1A	1B	1C	1D
Nippon Kaiji Kyokai	*IA Super*	*IA*	*IB*	*IC*	*ID*
Registro Italiano Navale	IAS	IA	IB	IC	ID

The class equivalencies agreed for the ice classes used in the Baltic are not valid for Arctic ice classes. Many ships having a Baltic ice class (IA or IA Super) have navigated in the Arctic successfully. This experience has prompted an action to parallel the lowest PC classes with the highest Baltic classes and treat the classes PC6, Arctic5 and IA Super as equivalent (and also PC7, Arctic4 and IA). This equivalency is recognised by the Baltic authorities and also by the Canadian authorities in the following form: 'As an interim measure for navigation purposes, Transport Canada will consider that PC 6 and 7 vessels will be allowed to operate as Type A and B vessels (Baltic 1AS and 1A construction) respectively.' (Transport Canada Bulletin No. 04/2009)

6. Winterisation Aspects

Winterization refers to those design aspects that are influenced by cold weather or ice cover, but are not covered in the structural design of hull or machinery covered by the ice rules. Thus for example the ballast water heating, sea chests' operation without clogging by ice, deck equipment operation and avoiding or mitigating the effects of ice accretion are areas where the cold weather should be taken into account. The term 'winterisation' sometimes alludes to a situation where a ship is designed according to the open water practice and then the winterization aspects are added on top of this open water design. This does not result in good solutions as the cold weather and ice should be taken into account from the beginning.

In the conceptual design phase most of the winterization aspects are not very prominent as these can be dealt with during the basic design phase – this does not mean, however, that these aspects are somehow second in importance. If winterisation aspects are not taken into account, often the ship operability and functioning is impaired. Some winterization aspects should be, however, taken into account from the early design phases. One of these is the protection for icing. Often heating i.e. ice melting is offered as a solution. Heating can be a solution for icing only in small areas like control boxes, control equipment etc. Mostly a better solution for icing is protection; the forecastle can be covered so that winches are protected, outside gangways could be indented into the deckhouse etc.

Fig. 38. A covered forecastle (left, from IB Varandey) and covered and sheltered gangways (right, from IB Healy).

Another matter that influences the early design is the required visibility from the wheel house. Icebreakers and other ships that must navigate actively in ice must do many manoeuvres that bring them close to other ships and also other obstacles like offshore structures. In order to successfully operate in close proximity of other ships, visibility from the wheel house must be good in all directions. This has led to a cockpit concept in wheel house design, see Fig. 39. In this concept all the ship operation is concentrated at one location, usually on the starboard side. No helmsman is required as the officer of watch operates the ship from one position. The visibility in all directions, especially forward, aft and sideways to starboard must be good.

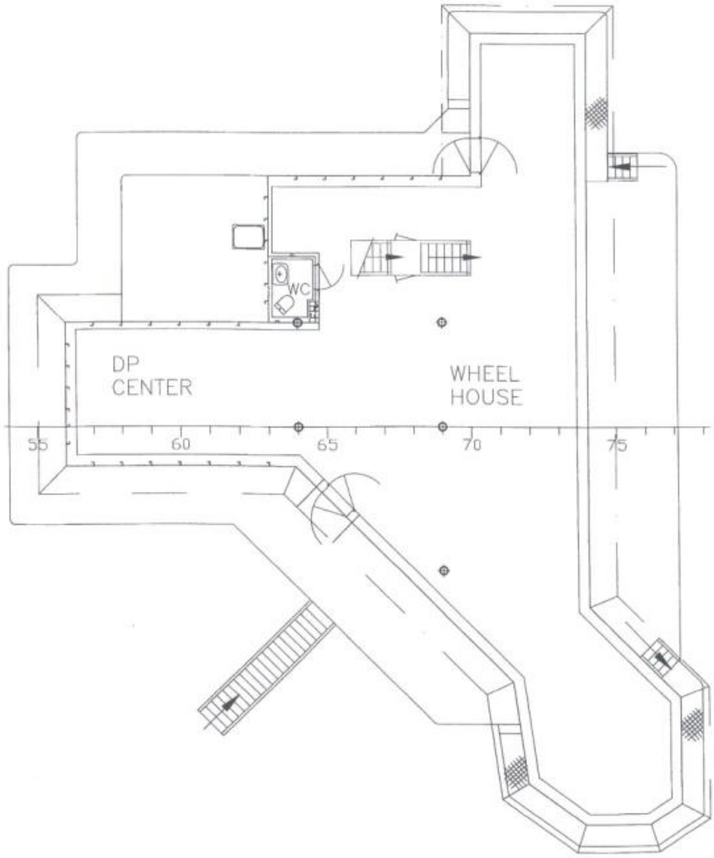


Fig. 39. Wheel house of the icebreaker Botnica.

The ship materials must be able to withstand the cold temperatures encountered without suffering any brittle damage. In the hull structure this is achieved by using higher grade steels in colder temperatures. An example of the steel grades required for different ice classes is given in Fig. 40. It should be noted that for lower ice classes PC6 and PC7 the required steel grade of thicker plating is grade D.

Thickness, t mm	Material Class II			
	PC1 to **5**		**PC6** and **7**	
	MS	HT	MS	HT
$t \leq 10$	B	AH	B	AH
$10 < t \leq 15$	D	DH	B	AH
$15 < t \leq 20$	D	DH	B	AH
$20 < t \leq 25$	D	DH	B	AH
$25 < t \leq 30$	E	EH, *see* Note 2	D	DH
$30 < t \leq 35$	E	EH	D	DH
$35 < t \leq 40$	E	EH	D	DH
$40 < t \leq 45$	E	EH	D	DH
$45 < t \leq 50$	E	EH	D	DH

NOTES
1. Includes weather exposed plating of hull structures and appendages, as well as their outboard framing members, situated above a level of 0,3 m below the lowest ice waterline.
2. Grades D, DH are allowed for a single strake of side shell plating not more than 1,8 m wide from 0,3 m below the lowest ice waterline.

Fig. 40. Steel grades required at different design temperatures of e.g. shell plating at bow area according to the Lloyd's Register Rules for Ice and Cold Operations (July 2010).

6. Conclusion

The design for performance and safety in ice has been covered in this chapter. The approach has been to give an overview of design aspects that must be taken into account rather than giving exact calculation methods. The design for ice is still more an art than a science and thus the designer must combine knowledge, sometimes conflicting, from different sources. Here the aim is to give some background for the designer and for a general interested reader of what the design for ice entails.

Several aspects in design for ice are still somewhat controversial; nozzles in ice, bulbous bows in ice, the pressure-area relationship to mention a few. The designer must use his/her own experience in making the design decisions. The design methods for ice do not have a single methodological background like the hydrodynamics where Navier-Stokes equations prevail. The approach used is mostly a collection of different methods from beam/plate theory, hydrodynamics, fracture mechanics etc. The methods in this approach are of an *ad hoc* type, all parts in the methods used should be roughly right. A first hand insight from ships operating in ice is invaluable in developing and using design methods incorporating results from several disciplines. The correct balance of different factors seems to be best achieved if the designer has some insight on ice action on ships. Thus designers who have gained insight from the feedback from the operation of earlier designs are in a good position.

Acknowledgements

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Glossary

Ship conceptual design: Determining the ship main parameters so that it fulfils the requirements.
Ice loads: Load acting on a ship when the ship collides with ice.
Ice pressure: Local ice force divided by the area the load is acting on.
Ice resistance: Time average of the longitudinal force from ice resisting ship motion.
Ship propulsion: The means to move the ship forward (or astern).
Ship-ice interaction: When a ship and an ice feature are in contact, the response (motions) and the contact force influence each other.
Lines: Ship hull shape drawing given usually at three projections
Shoulders: The location where the bow reaches the full breadth
Shaftline: Ship propulsion system from propellers, shafts, gears to the main engine
Bollard pull: The force the ship propulsion can develop at zero speed
Stringer: Main longitudinal frame supporting the transverse frames
Web frame: Transverse frame supporting the stringers or longitudinal frames
General arrangement: The layout of the ship spaces and machinery/equipment.
Ship ice performance: The speed that the ship can reach in ice of different type, the way the ship can manoeuvre in ice.

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Biographical Sketch

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