# RINA

# DESIGN AND CONSTRUCTION OF VESSELS OPERATING IN LOW TEMPERATURE ENVIRONMENTS

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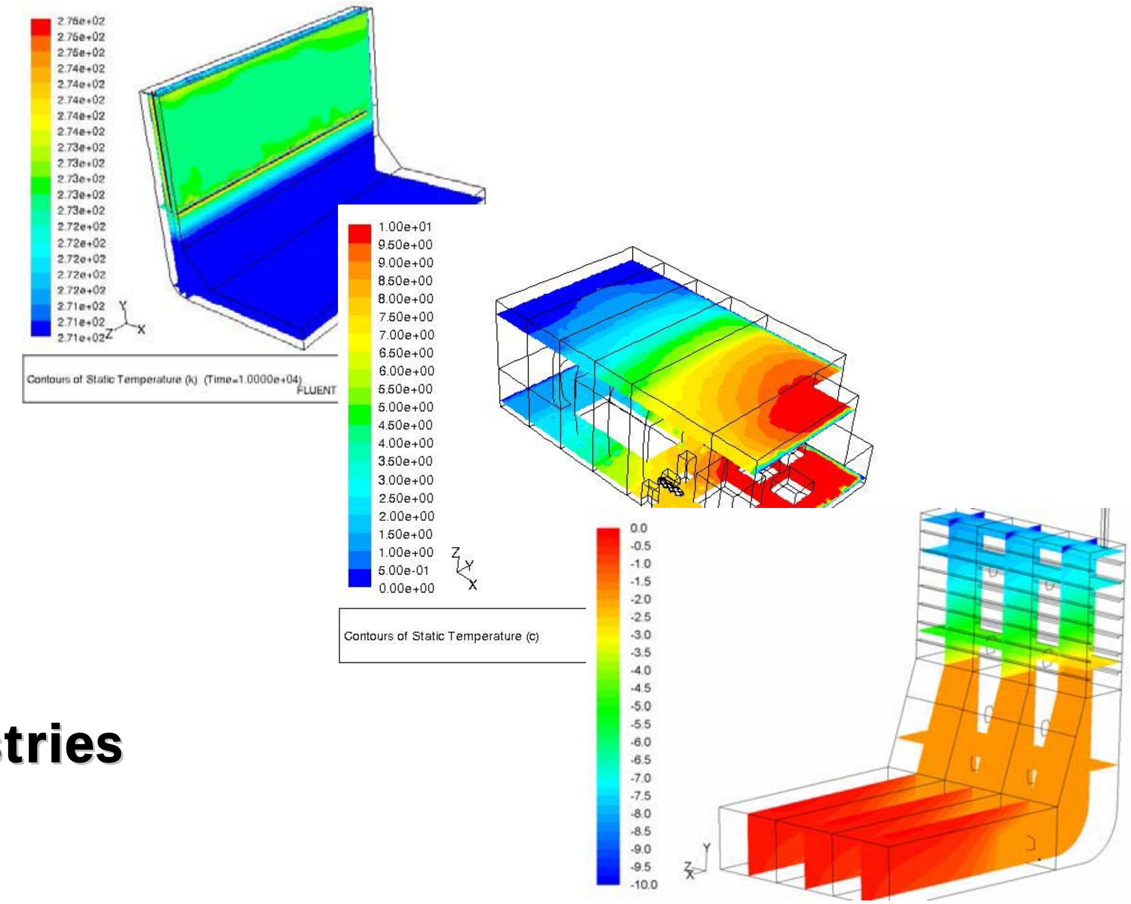
# Space and Tank Winterization Techniques for Vessels Operating in Cold Regions

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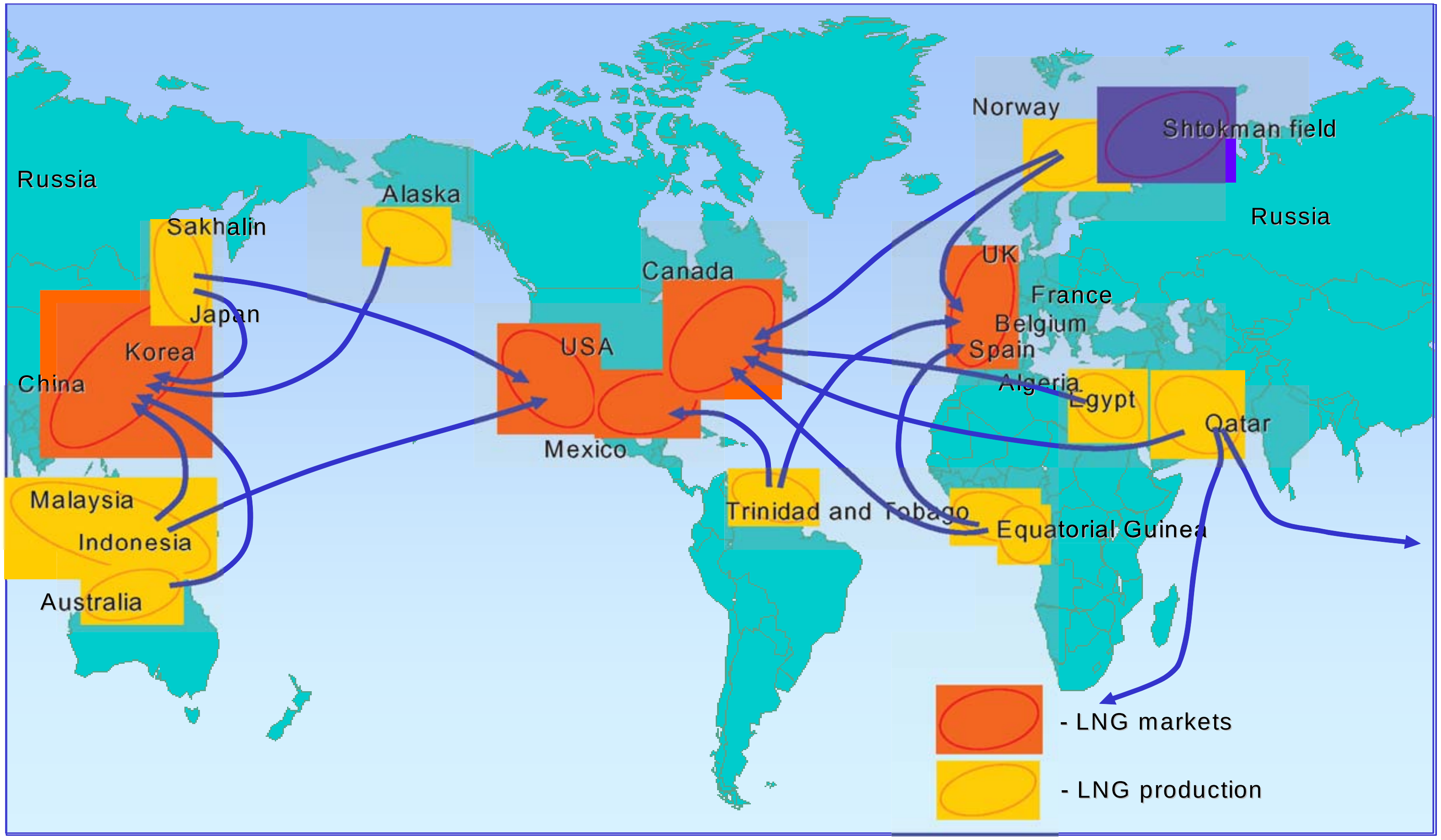
**Samsung Heavy Industries**

# Backgrounds

**World LNG transportation**



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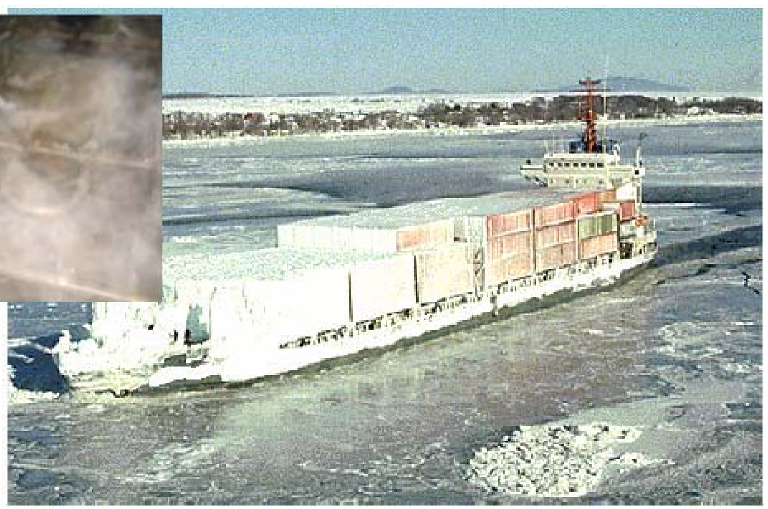
# Backgrounds

**From [Winter navigation on the river and gulf of St. Lawrence, 2005, Canada]**

Figures 5.3A– Severe case of ice accretion in bow area

Figures 5.3F– Deck, piping and valves inaccessible



Figures 5.3M– Effect of freezing spray on a container ship

Figures 9.1B – Deficient wheelhouse heating combined with a high rate of humidity

# References for winterization

What is the effect by the
temperature limitation

| | Space heating | Tank heating | Reference Cert. |
|---|---|---|---|
| ABS | Required certain value of heating to keep operation and safety of crews<br><br>-> need to define design tool for insulation and heating capacity | . Design temp. (-30~-10℃):<br>Heating or turbulence devices including bubbler system<br>. Design temp (Down to -30℃):<br>Steam heating coil | CCO : for vessel<br>CCO+ : CCO and for crew/training<br>GN, "Vessels operating in low temperature environments" |
| DNV | | . Design temp. (Down to -10℃):<br>Heating devices required (For ballast tank, fresh water tank & fuel oil tank) | Winterized (°C)<br>Winterized Arctic (°C)<br>DAT(-X°C)<br>"Rule for classification of ships: Newbuildings" Part 5 chapter 1 |
| LR | | . Design temp. (Down to -30℃)<br>Circulation of air bubbling<br>. Design temp. (-31~-45℃)<br>Circulation or heating coil<br>. Design temp. (-46℃ below)<br>Heating coil (For ballast tank, fresh water tank) | Part 7 chapter 15 :<br>Winterization (Section 2), "Rule Amendment (marine) for the consideration of the November 200x technical committee" |

# Objectives

- To introduce winterization engineering methods
- To show comparative studies on proposed methods
- To propose a standardization of winterization technology

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# Category of engineering works

| Level | Required Analysis | Application | Tool |
|---|---|---|---|
| 1 | Simplified heat balance in a steady state<br>-Thermal analysis<br>-Condensation analysis | . Insulation design<br>. Condensation avoidance<br>. Heating capacity for simplified geometrical spaces (tank, room etc.) | Simplified design tool |
| 2 | Simplified heat balance in transient | . Insulation design for recovery from black-out | Simplified Commercial design tool |
| 3 | Detailed heat balance<br>-Full-scale steady and transient analysis | . Air bubbling capacity for ballast tank<br>. Complex geometrical space (ballast tank with stiffeners and stringers) | Advanced Commercial tool<br>/FLUENT |

*** Levels 1 and 2 have been verified by level 3 tool.**



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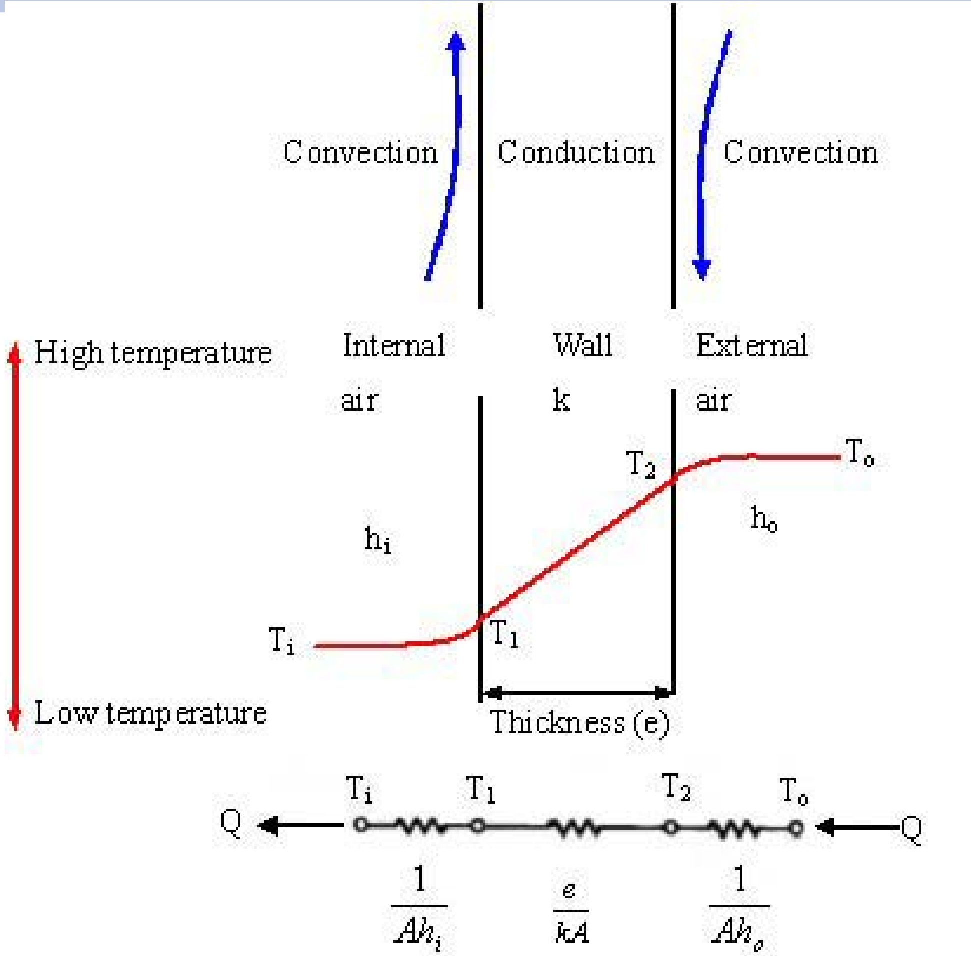
# Level 1: Fundamental level

- Work scopes
  - To derive design tool based on fundamental thermal analysis
  - To make simplified Excel chart for calculation
  - To verify the simplified model by the FLUENT calculation

- Considered parameters
  - Conduction with various materials
  - Convection with natural and forced air flow
    - Natural convection will be estimated by the direct calculation to consider real situation, compared with the given values in ISO code

- Application
  - Determination of heating coil capacity in ballast tank
  - Determination of heating capacity in APV room
  - Possibility check for condensation of engine room surface

# Governing equation




* **Conduction**

$$Q = kA\Delta T / e$$

* **Convection**

$$Q = hA\Delta T$$

* **Total heat transfer**

$$Q = UA(T_o - T_i)$$

$$U = \frac{1}{\frac{1}{h_i} + \frac{e}{k} + \frac{1}{h_o}}$$

Overall heat transfer coefficient

# Heat transfer coefficient: ISO based approach (1/2)

**ISO 7547 Ships and marine technology- air conditioning and ventilation of accommodation spaces- design conditions and basis of calculations**

**...**

**4 Design conditions**

**4.2 Summer temperatures and humidity**

- **(a) outdoor air: +35°C and 70% humidity**
- **(b) Indoor air: +27°C and 50% humidity**

**4.3 Winter temperatures**

- **(a) Outdoor: -20°C**
- **(b) Indoor air: +22°C**

**Note This international standard does not specify requirements for humidification in winter.**

**5.2.4 Calculation of heat transfer coefficient**

**...**

**h=80W/m²K for outer surface exposed to wind (20m/s)**

**h= 8 W/m²K for inside surface not exposed to wind (0.5m/s)**

**...**

**Rough estimation of heat transfer**

**Need to clarify in engineering works to find optimum solution**

# Heat transfer coefficient: ISO based approach (2/2)

…

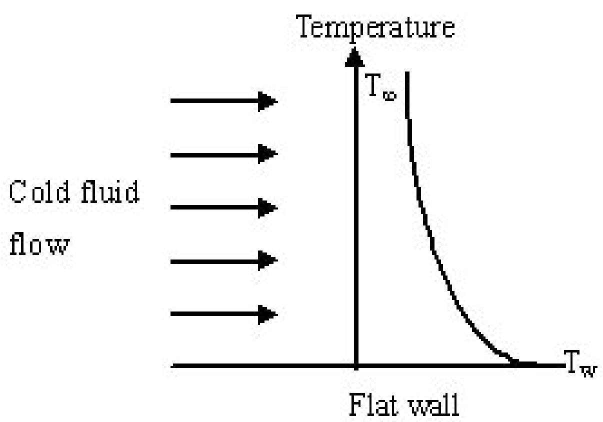
**Table 2 – Total heat transfer coefficient**

| Surface | Total Heat transfer ($kW/m^2K$) |
|---|---|
| Weather deck not exposed to sun's radiation and ship side and external bulkheads | 0.9 |
| Deck and bulkhead against engine-room, cargo space or other non-air conditioned spaces | 0.8 |
| Deck and bulkhead against boiler-room or boiler in engine-room | 0.7 |
| Deck against open air or weather deck exposed to sun's radiation and deck against hot tanks | 0.6 |
| Side scuttles and rectangular windows, single glazing | 5.5 |
| Side scuttles and rectangular windows, double glazing | 3.5 |
| Bulkhead against alleyway, non-sound reducing | 2.5 |
| Bulkhead against alleyway, sound reducing | 0.9 |

# Heat transfer coefficient: Theoretical approach

**Nusselt number: dimensionless number that measures the enhancement of heat transfer from a surface when convection takes place, compared to the heat transferred if just conduction occurred.**

$$Nu_L = \frac{hL}{k_f}$$




**Natural convection coefficient at a vertical wall**

$$\overline{Nu_L} = 0.68 + \frac{0.67 Ra_L^{1/4}}{[1+(0.492/Pr)^{9/16}]^{4/9}} \quad Ra_L \le 10^9$$

**from Churchill and Chu**

**Natural convection coefficient at a horizontal plate (For the top surface of a hot object in a colder environment )**

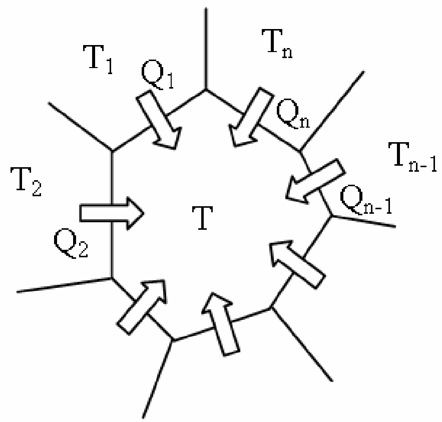
$$\overline{Nu_L} = 0.54 Ra_L^{1/4} \quad 10^4 \le Ra_L \le 10^7$$

$$\overline{Nu_L} = 0.15 Ra_L^{1/3} \quad 10^7 \le Ra_L \le 10^{11}$$

**Natural convection coefficient at a horizontal plate (For the bottom surface of a hot object in a colder environment )**

$$\overline{Nu_L} = 0.27 Ra_L^{1/4} \quad 10^5 \le Ra_L \le 10^{10}$$

# Determination of heating capacity




$$U_1 A_1 (T - T_1) + U_2 A_2 (T - T_2) + \cdots + U_n A_n (T - T_n) = 0$$

**Temperature (T) : required temperature to keep**

$$\sum_n Q_i = 0$$

$$T = \frac{U_1 A_1 T_1 + U_2 A_2 T_2 + \cdots + U_{n-1} A_{n-1} T_{n-1} + U_n A_n T_n}{U_1 A_1 + U_2 A_2 + \cdots + U_{n-1} A_{n-a} + U_n A_n}$$

$$\dot{m} = [\sum_j U_j A_j (T_j - T)] / i \qquad [kg/h]$$

$i$ : Enthalpy of internal medium ($W/kg$)

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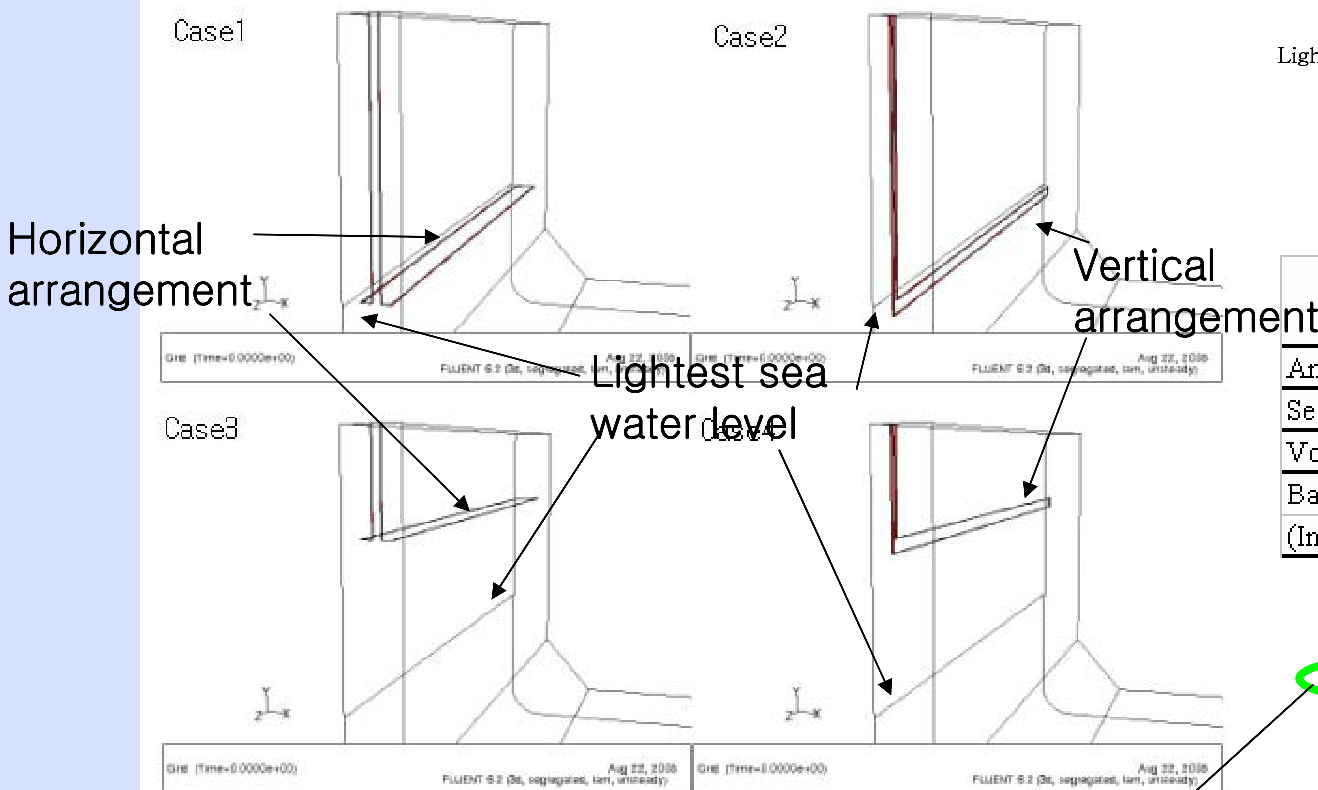
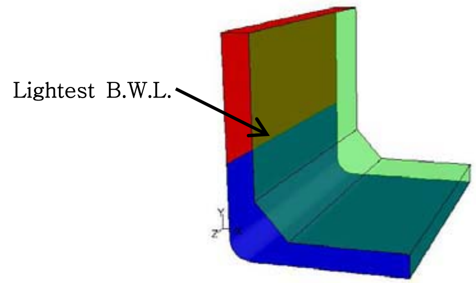
# Application 1: (Using proposed approach) Heating Coil Capacity in Ballast Tank

**Geometrical model**




**Boundary condition**

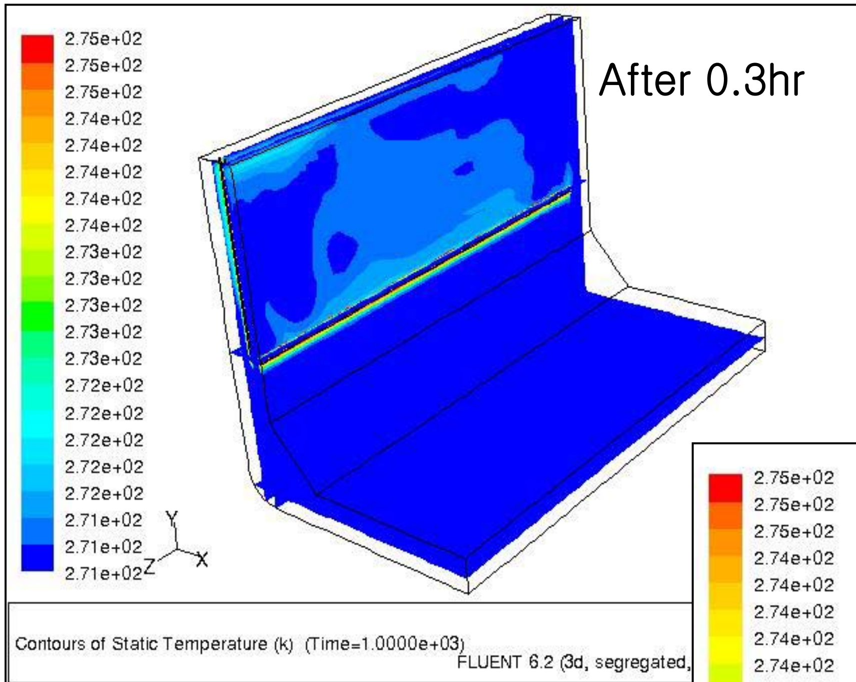


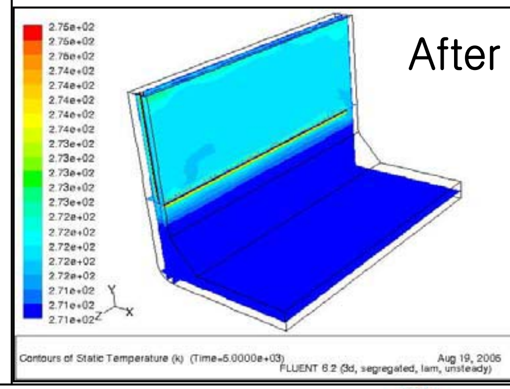

| | Temperature (°C) | Notice |
|---|---|---|
| Ambient air | -40 | Red face |
| Sea water | -2 | Blue face |
| Void space | 0 | Green face |
| Ballast water (Initial condition) | -2 | |

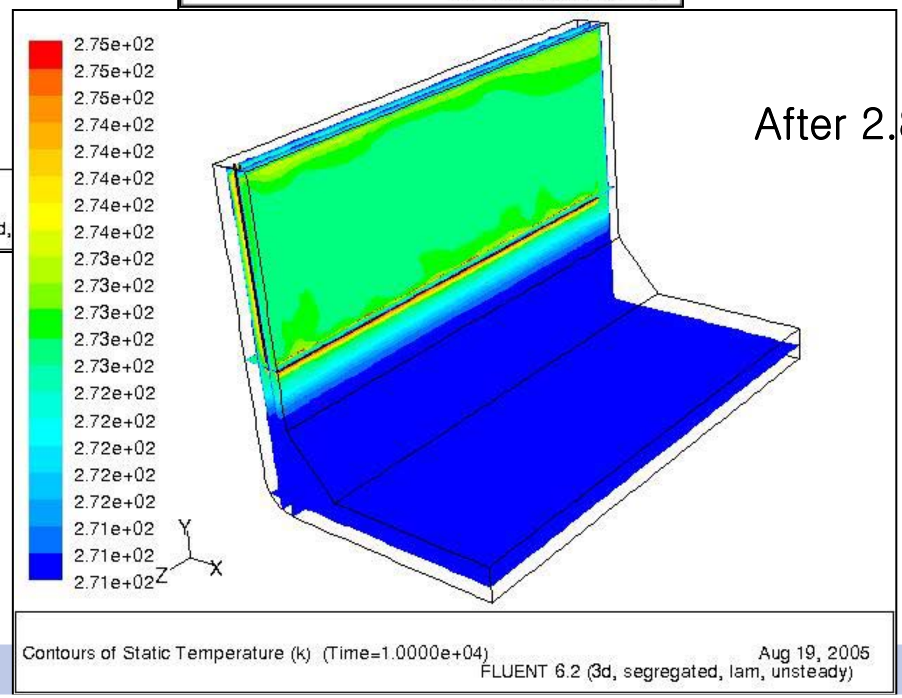
| | Heating coil area ($m^2$) | Heat flux ($W/m^2$) |
|---|---|---|
| Case 1 | 17.2 | 17710 |
| Case 2 | 16.8 | 18132 |
| Case 3 | 14.3 | 21301 |
| Case 4 | 14.0 | 21758 |

Add 10% margin for design safety : ~ 195kW

# Application 1: (CFD results)
# Heating Coil Capacity in Ballast Tank

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# Application 1: (Comparison) Heating Coil Capacity in Ballast Tank

Heater capacity estimation

| | Expected heating capacity for ballast tank | | | Difference ratio | |
|---|---|---|---|---|---|
| | (1) Convection by ISO | (2) Covection by fomula | (3) Convection by CFD analysis | (1)/(3) | (2)/(3) |
| Water ballast tank | 283 kW | 195 kW | 179 kW | 58% | 9% |

Estimated Temperature using derived heating capacity

| Cases: different location of heating coil | Expected heating capacity for ballast tank(°C) | | | |
|---|---|---|---|---|
| | Convection by ISO | | Covection by fomula | |
| | Design Temp | Est. Temp. by CFD using esti. heat capacity | Design Temp | Est. Temp. by CFD using esti. heat capacity |
| Case 1 | 2 | 8.4 | 2 | 2.1 |
| Case 2 | 2 | 8.3 | 2 | 2.1 |
| Case 3 | 2 | 8.1 | 2 | 2.1 |
| Case 4 | 2 | 8.2 | 2 | 2.1 |

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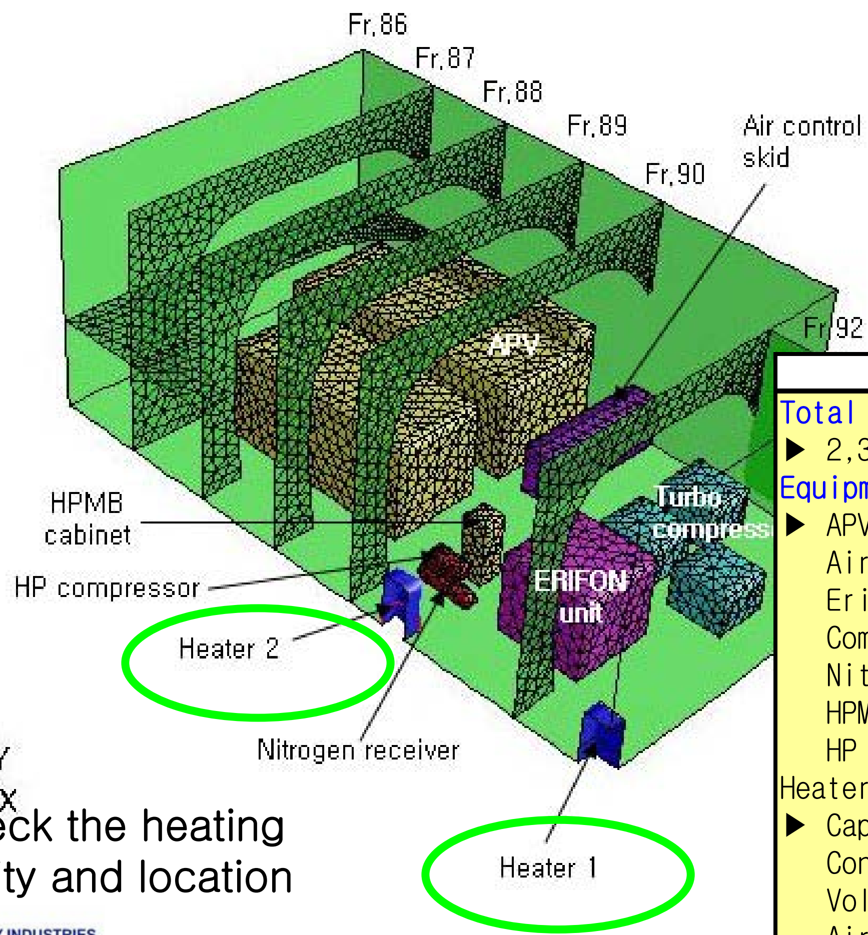
Quite similar results for simplified geometrical model

# Application 2: (Using Proposed approach) Heating Capacity in Air Pressurized Vessel Room

**Geometrical model**




**Boundary condition**

Boundary condition

Inlet condition

- ▶ Heating air(2500m$^3$/hr, 60℃)

Outlet condition

- ▶ Returned air to heater(2500m$^3$/hr)

Surrounding space temperature

- ▶ Ballast tank(0℃)
  - Upper deck air(-20℃ with 5knots)
  - No ventilation opening and holes
  - All of doors and vents are closed

APV room

Total voulme

- ▶ 2,323.28m3

Equipments

- ▶ APV(2units, 0kW)
  - Air control skid(1unit, 0kW)
  - Erifon unit(1unit, 5kW)
  - Compressor skid(3units, 27kW)
  - Nitrogen generator(1unit, 1kW)
  - HPMB cabinet(1unit, 0kW)
  - HP compressor(1unit, 0kW)

Heater

- ▶ Capacity : 34kW
  - Control mode : 3P
  - Voltage : 440V
  - Air flow rate & setting tempe.
  - 2000m$^3$/hr(100℃)~2500m$^3$/hr(60℃)

To check the heating capacity and location

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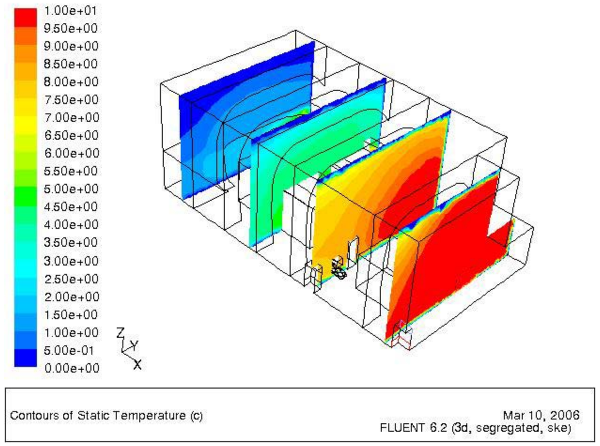
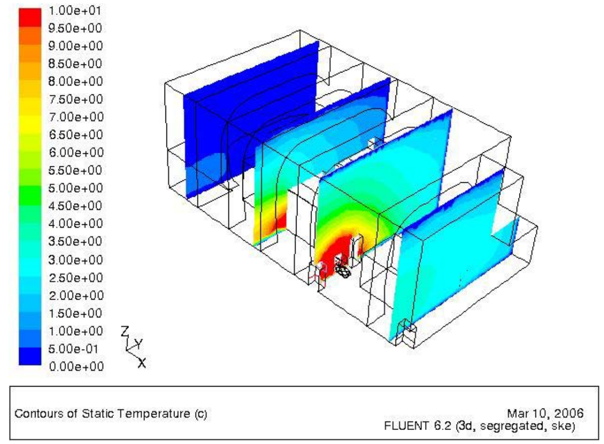
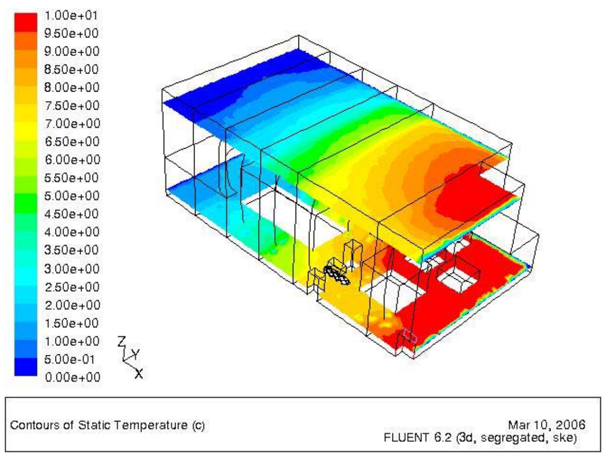
# Application 2: (CFD results) Heating Capacity in APV Room

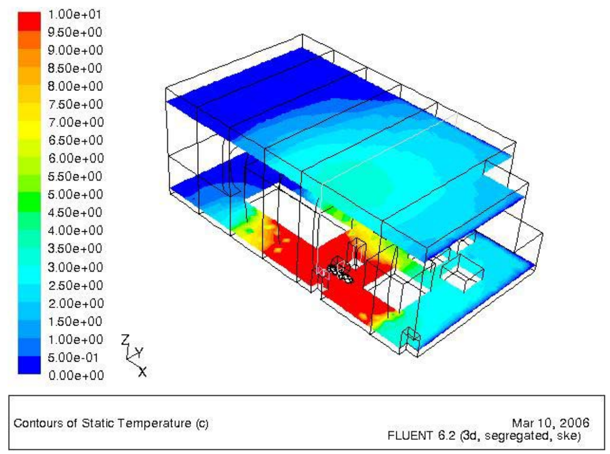
**Heater 1 (Operating)** **Heater 2 (Operating)**

*Temperature range of from 0 to 10 ℃*



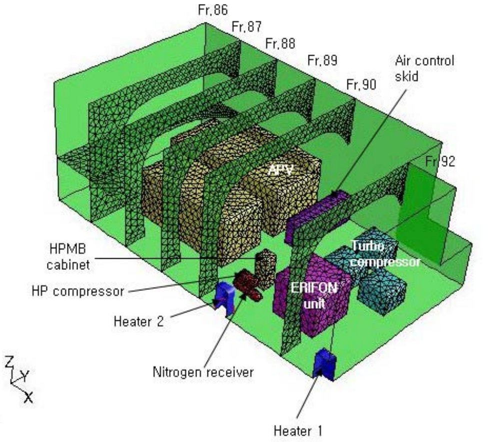
# Application 2: (Comparison) Heating Capacity in APV Room

Heater capacity estimation

| | Expected heating capacity for APV room | | | Difference ratio | |
|---|---|---|---|---|---|
| | (1) Convection by ISO | (2) Covection by fomula | (3) Convection by CFD analysis | (1)/(3) | (2)/(3) |
| APV room | 33 kW | 21.4 kW | 23 kW | 40% | 9% |

Estimated Temperature using derived heating capacity

| Operating heater with fan stopped condition for ventilation | Expected Temperature in APV room(°C) | | | |
|---|---|---|---|---|
| | Convection by ISO | | Covection by fomula | |
| | Design Temp | Est. Temp. by CFD using esti. heat capacity | Design Temp | Est. Temp. by CFD using esti. heat capacity |
| Heater 1 | 0 | 6.8 | 0 | 2.1 |
| Heater 2 | 0 | 2.6 | 0 | 0.6 |




Similar results with CFD, but less estimation due to the difficulties in interaction by geometrical complexity

# Level 2: Intermediate level

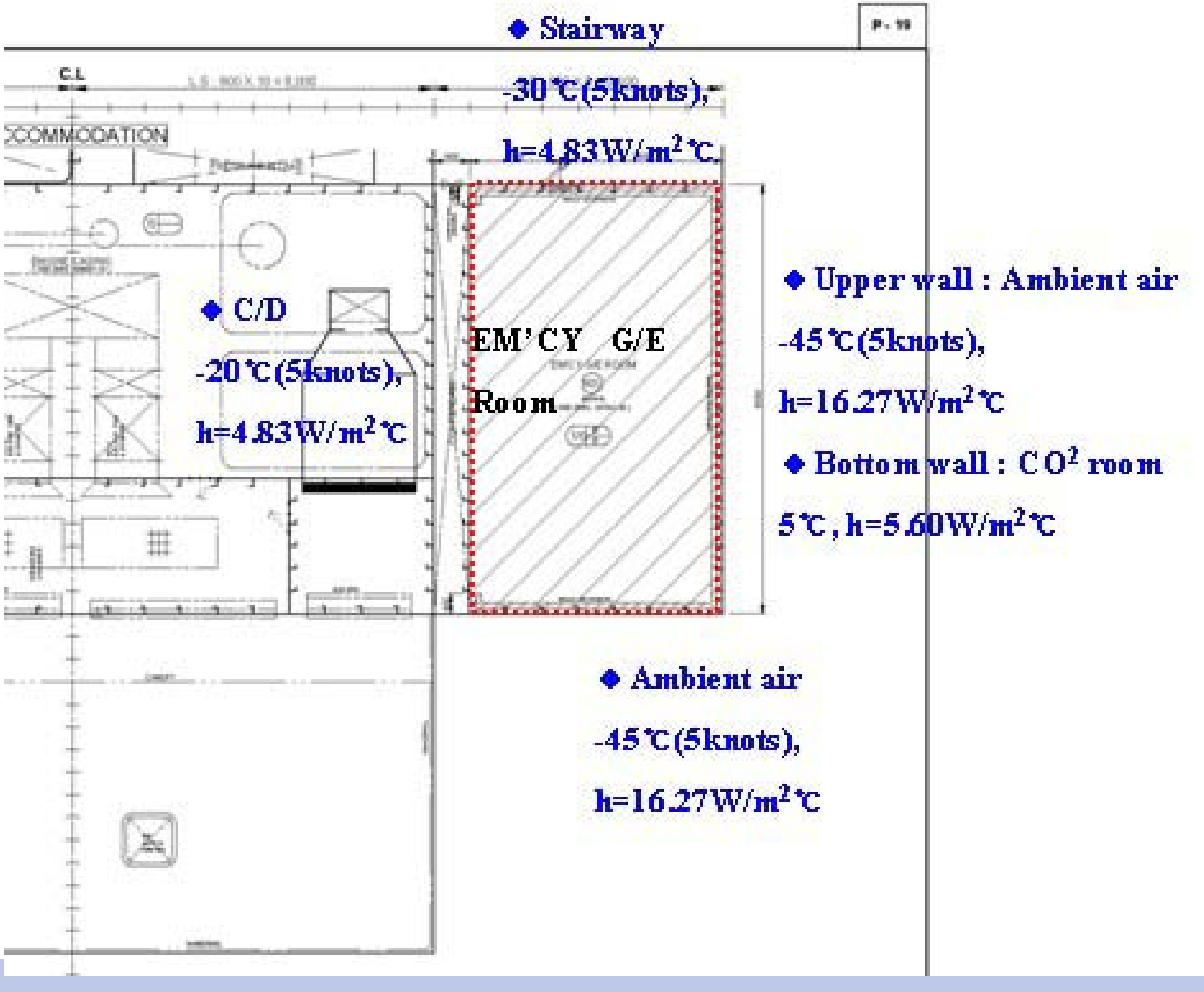
- **Work scopes**
  - **To check the transient analysis performance for a space**
  - **To compare CAPSOL (simplified approach) and FLUENT (3D CFD approach) results in time domain analysis**

- **Considered parameters**
  - **Conduction with various materials**
  - **Convection with natural and forced air flow**
  - **Thermal bridge in-between the insulation panel**

- **Application**
  - **Evaluation of temperature drop in emergency generator room due to blackout**
  - **Evaluation of temperature increase in Cold chamber due to blackout**

# Application:
# Temperature Drop in EM’CY Gen. Room

Geometrical model and Boundary condition

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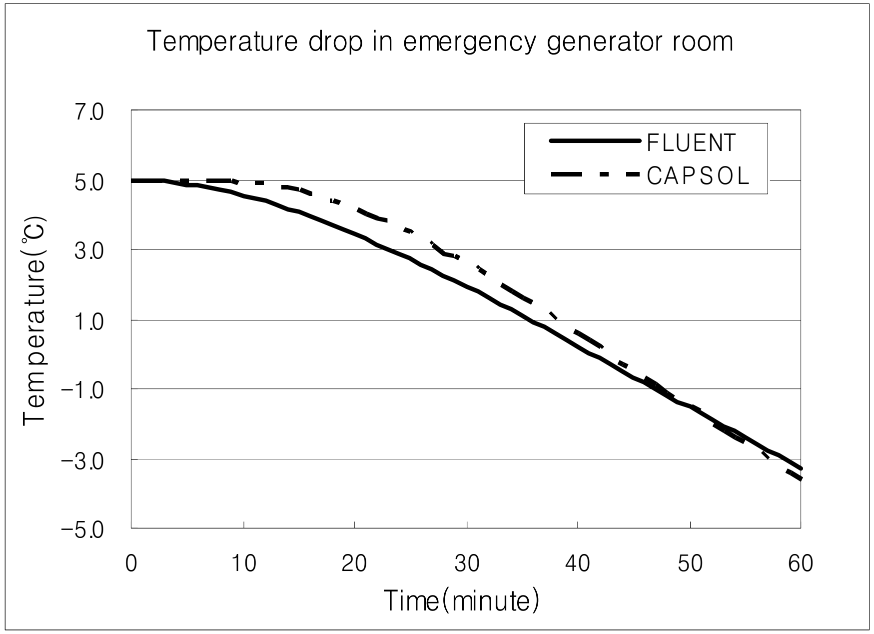
# Application:
# Temperature Drop in EM'CY Gen. Room




**Parameters to consider**

1. **Conduction of wall**
2. **Convection both sides**
3. **Heating capacity of wall consisting of insulation and steel**

**Based on the selection of heat transfer coefficients, estimation quality by simplified approach will be decided.**

# Level 3: Advanced level

- Work scopes
  - To validate design concept by considering geometry and systems
  - To check the winterization guidance by classes

- Considered parameters
  - Conduction with various materials
  - Convection with natural and forced air flow
  - Including additional systems such as air-bubbling or heating coil etc.

- Application
  - Evaluation of air-bubbling system for anti-freezing performance of ballast tank

# Numerical method & modeling : Governing equations etc. (Fluent Ver. 6.2)

- **Mass conservation equation**

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0$$

- **Momentum conservation equation**

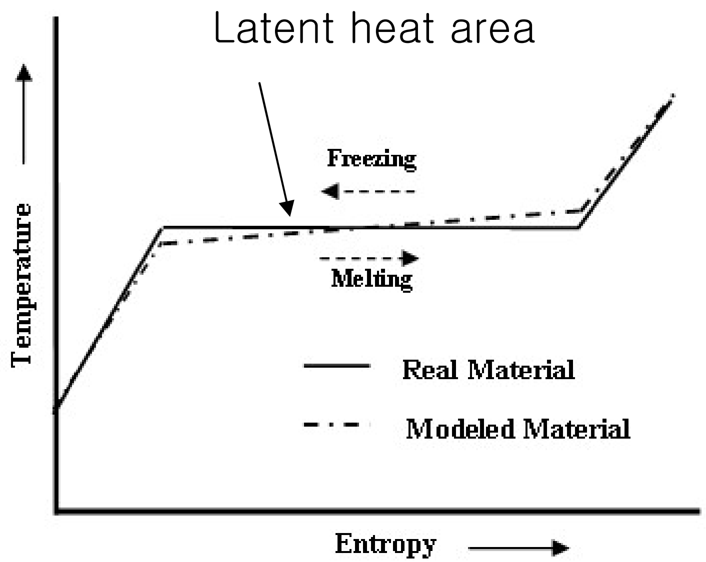
$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i \qquad \tau_{ij} = \left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] - \frac{2}{3}\mu \frac{\partial u_i}{\partial x_i}\delta_{ij}$$

- **Energy conservation equation**

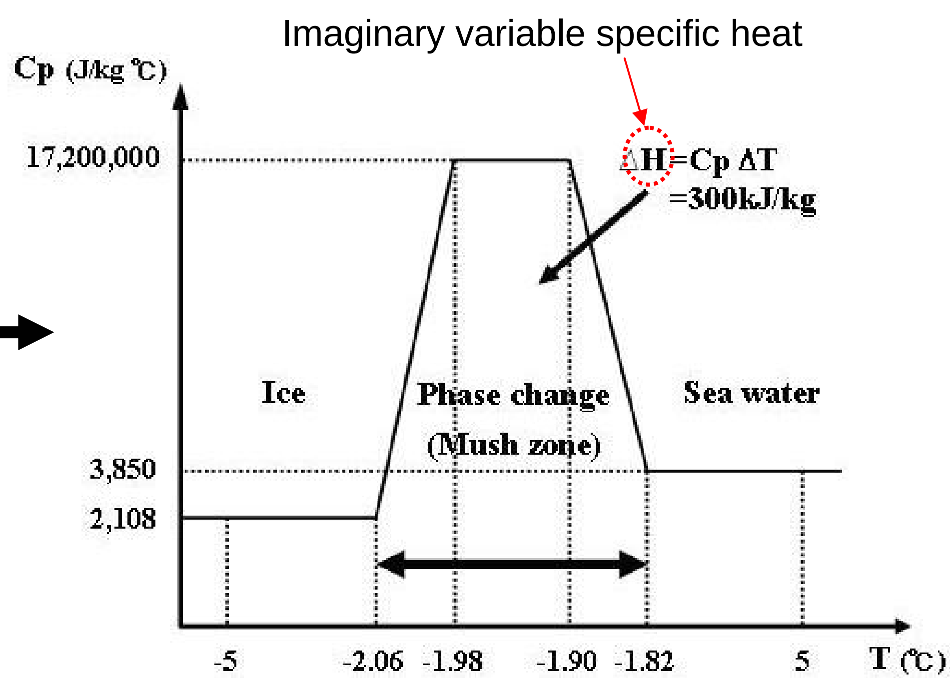
$$\frac{\partial}{\partial t}(\rho h) + \frac{\partial}{\partial x_i}(\rho u_i h) = \frac{\partial}{\partial x_i}\left(K \frac{\partial T}{\partial x_i}\right) + \frac{\partial p}{\partial t} + u_i \frac{\partial p}{\partial x_i} + \tau_{ij}\frac{\partial u_i}{\partial x_j} + S_h \qquad h = \sum_{i'} m_i h_{i'} \int_{T_{ref}}^{T} C_{p,i'} dT$$

- **Turbulence model : Standard k-ε turbulence model (with standard wall function)**
- **Multiphase model : VOF model (with Geo-Reconstruct)**
- **Freezing model : Imaginary variable specific heat at the range of freezing point**

# Numerical method & modeling : Modeling for heat transfer derived from freezing




Entropy – Temperature curve while phase change




Modeled latent heat defined by imaginary variable specific heat at the range of freezing point

To consider latent heat model in CFD,
we introduce imaginary variable specific heat model.

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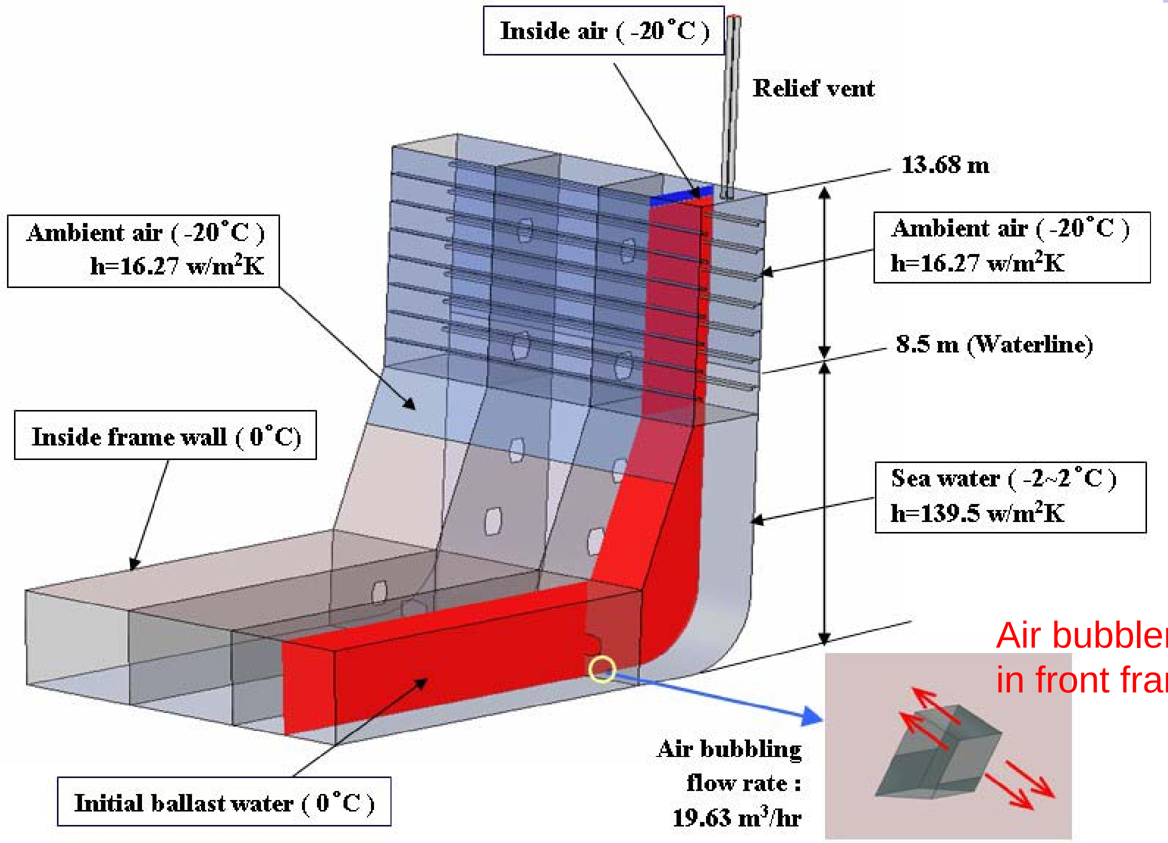
# Numerical method & modeling : Modeling of water ballast tank (Boundary conditions)




Air bubbler only equipped in front frame

Schematics of water ballast tank with detailed boundary conditions



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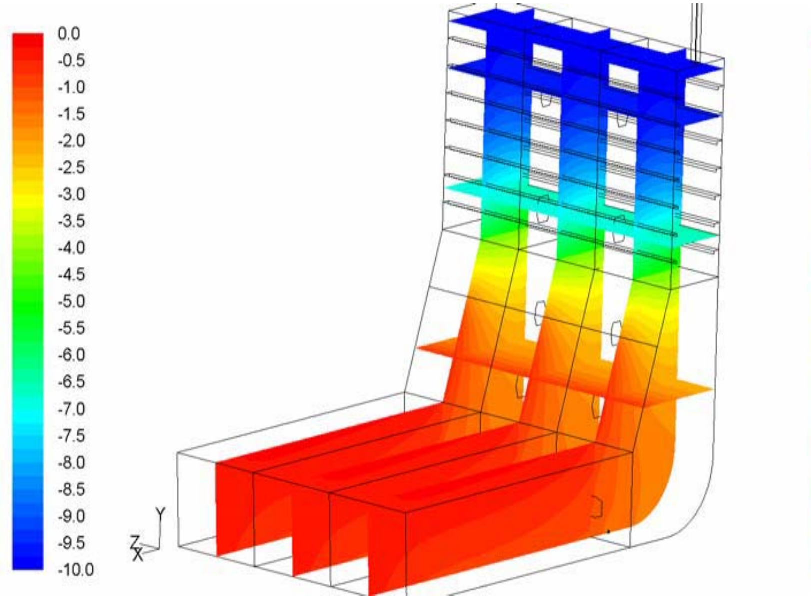
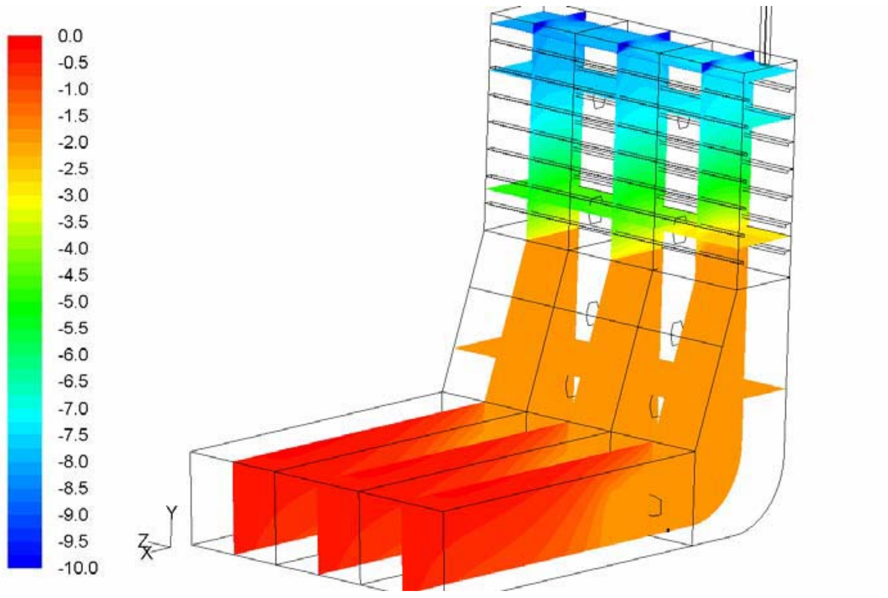
# Analysis results : Estimation of modeled phase change approach (without air bubbling)

Temperature (℃)




**With constant Cp (Cp =const = 3850 J/kg K)**

**With variable specific heat to consider latent heat**

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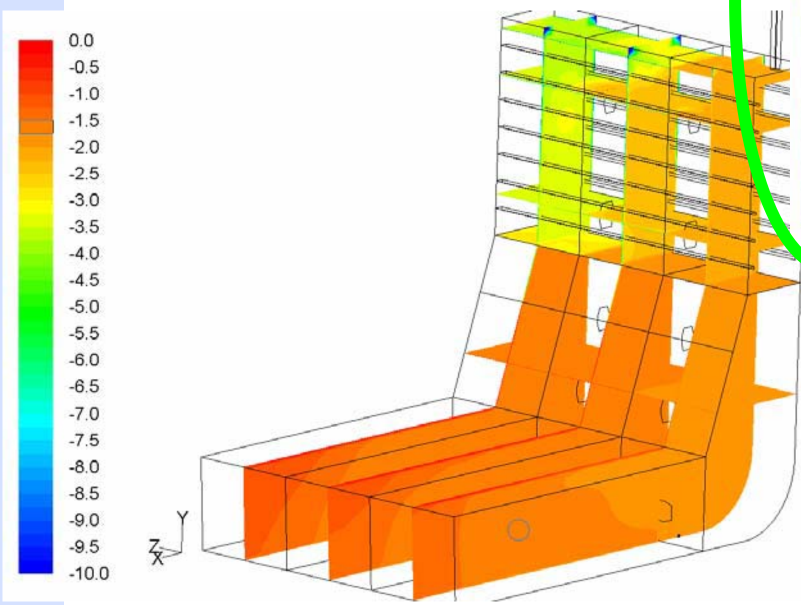
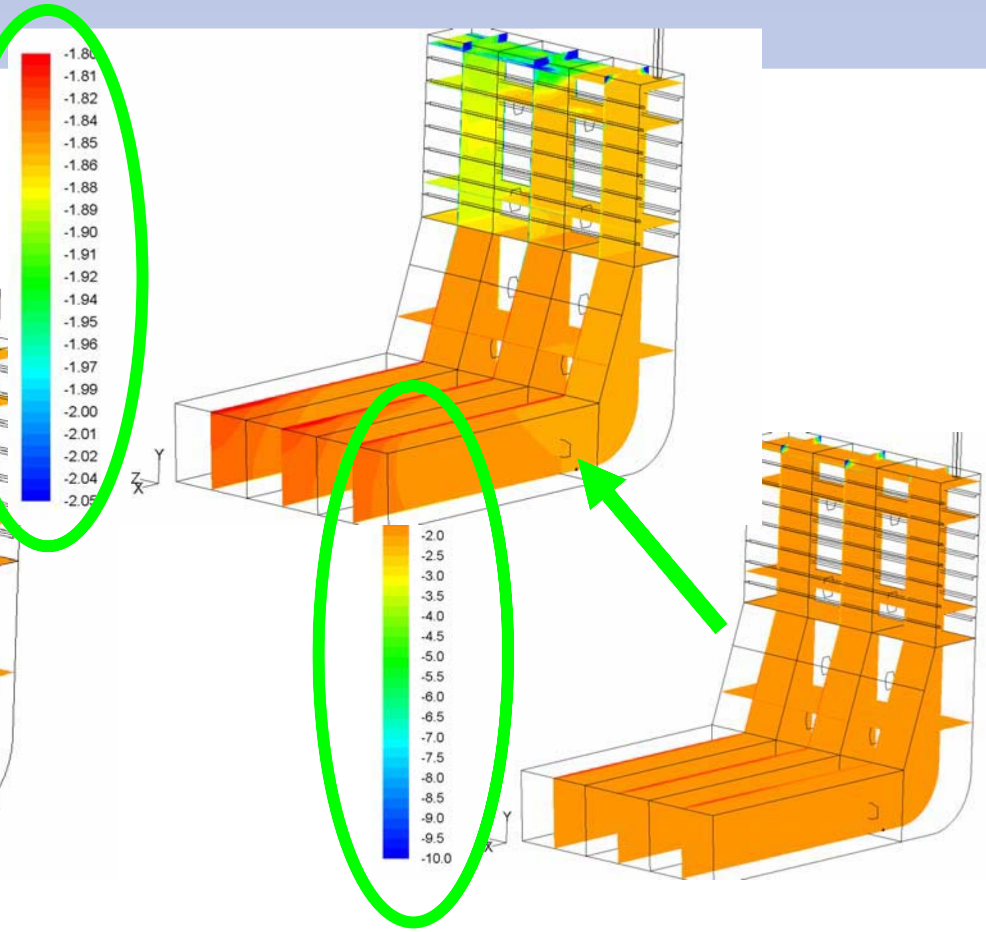
# Analysis results : Estimation of modeled phase change approach (<u>with air bubbling</u>)

Temperature (℃)




**With constant Cp (Cp =const = 3850 J/kg K)**

**With variable specific heat to consider latent heat**

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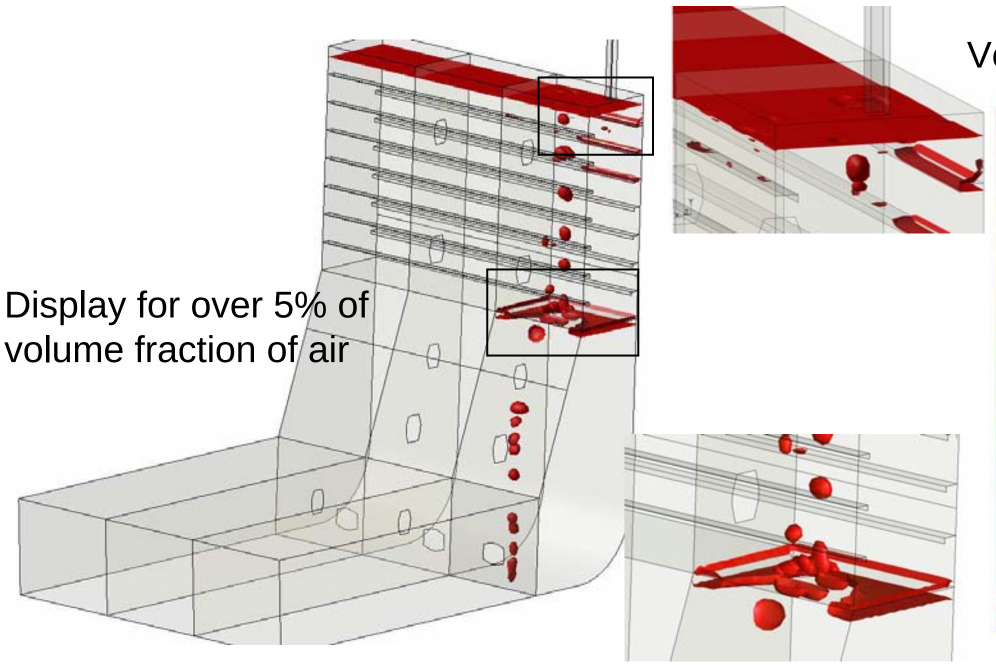
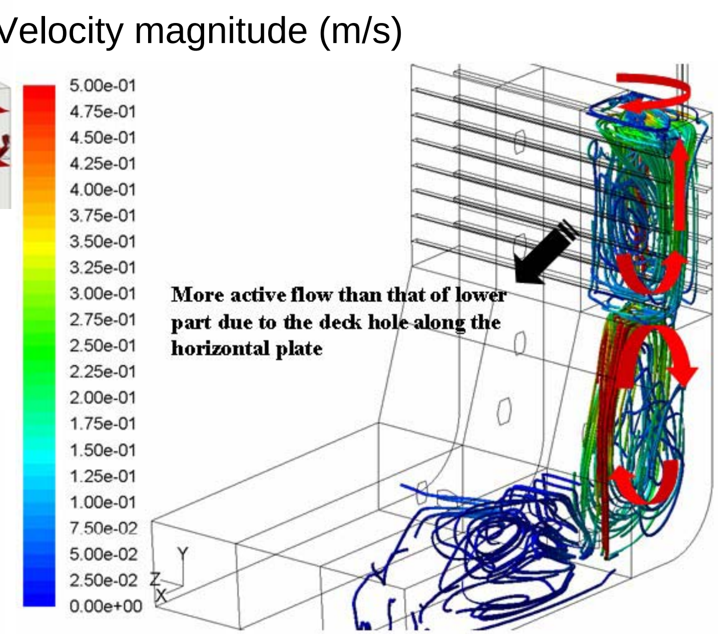
# Analysis results : Estimation of modeled phase change approach and air bubbling system

* **Air bubbles trajectories and path line of flow in water ballast tank**




Air bubbles trajectories in water ballast tank, represented by volume fraction for air




Path line of flow in water ballast tank with color legend of velocity magnitude with unit of m/s

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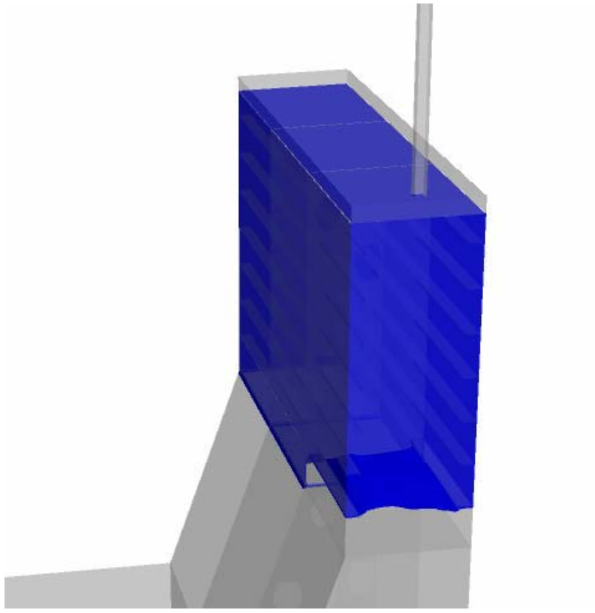
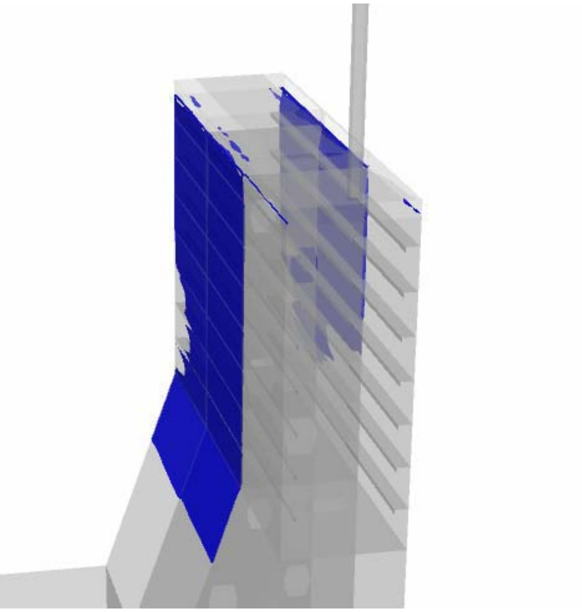
# Analysis results : Capability anti-freezing effect of air bubbling system

⁕ **Comparison of freezing and mush zone as installation of air bubbling system**

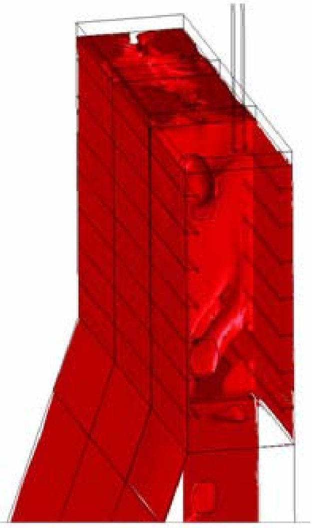
**Freezing zone : Down to -2.06℃**
**Mush zone : -2.06 ~ -1.82℃**



**Freezing zone without air bubbling system**

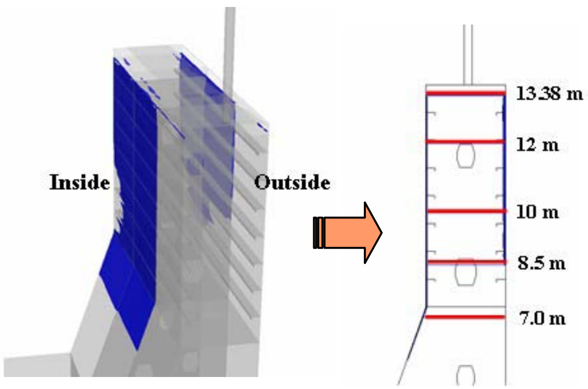


**Freezing zone with air bubbling of front frame**

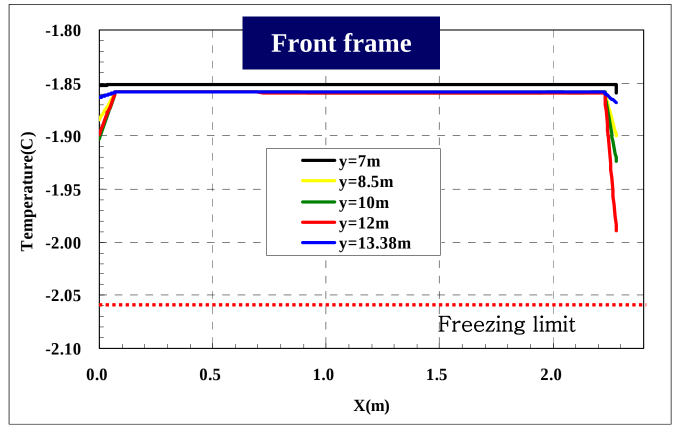


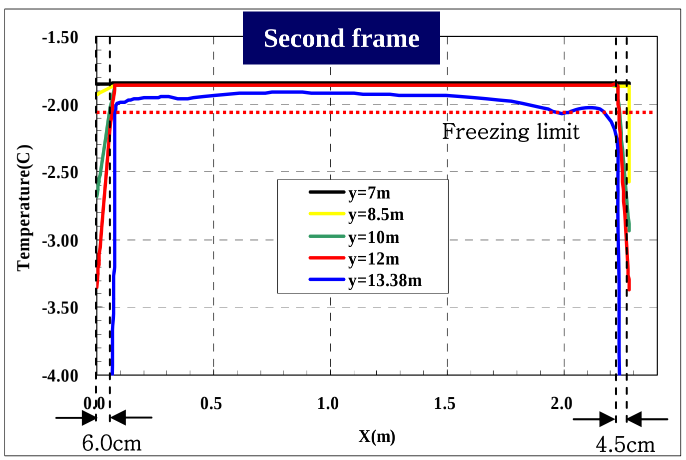
**Mush zone with air bubbling of front frame**

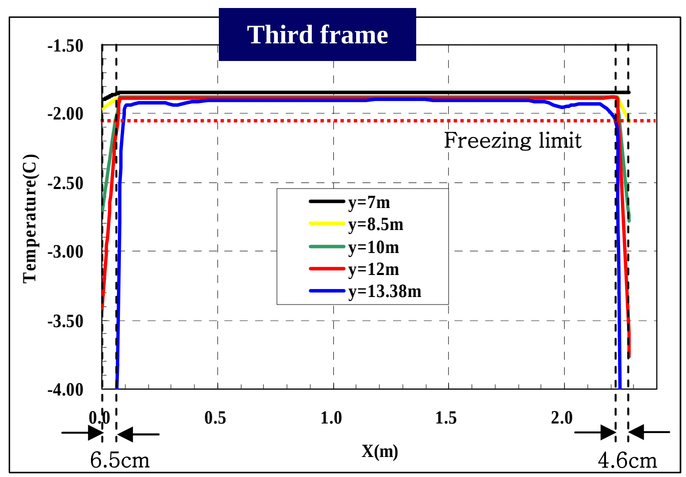
# Analysis results : Measures of thickness of ice sheet for each frame




Measuring points



# Concluding Remarks

- **Concerned works**

  **(1) Space and tank winterization**

  **(2) Propose 3 level categories for evaluation process**

- **Results**

  **(1) Reasonable prediction of heating capacity for Space (Level 1)**

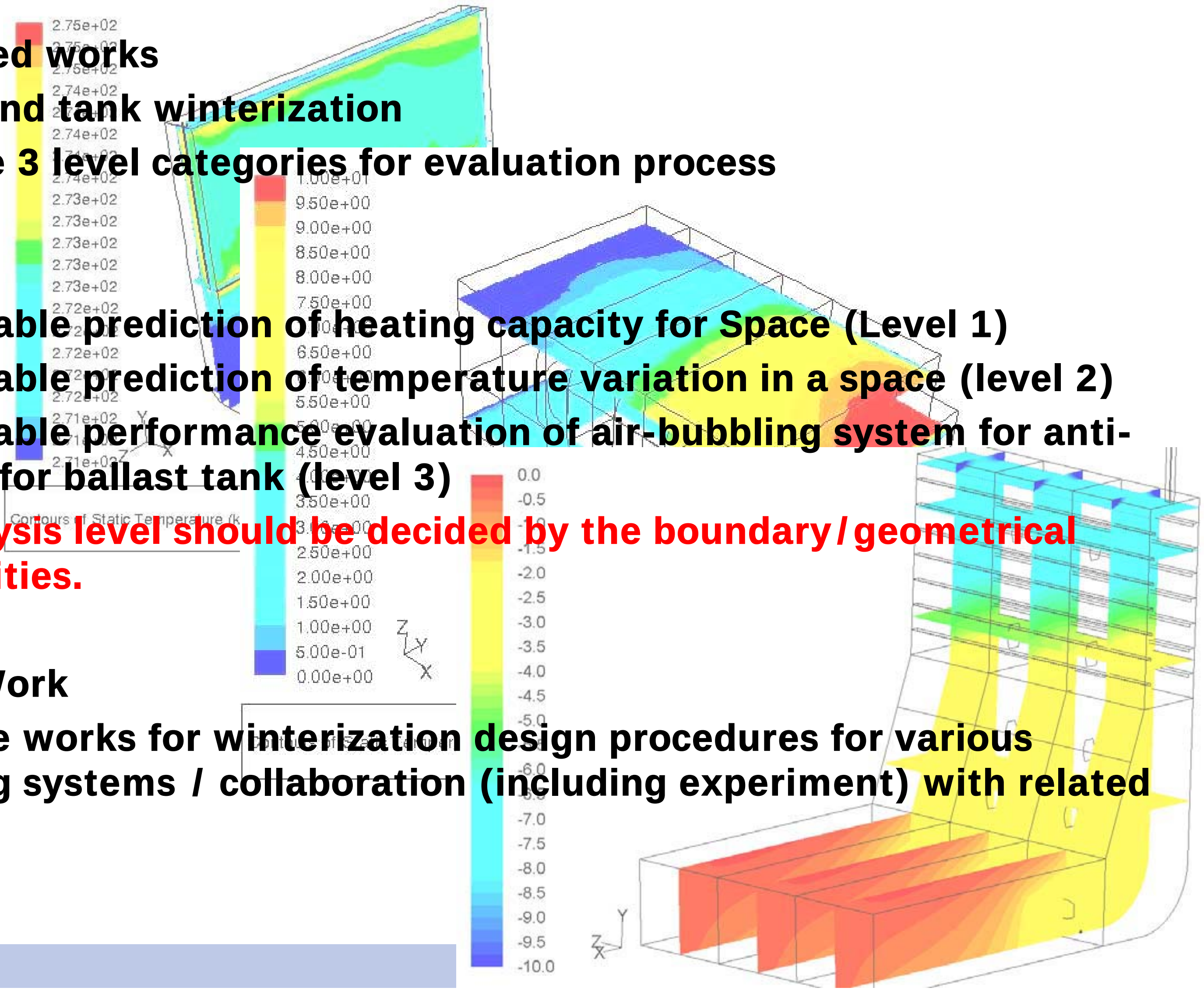
  **(2) Reasonable prediction of temperature variation in a space (level 2)**

  **(3) Reasonable performance evaluation of air-bubbling system for anti-freezing for ballast tank (level 3)**

  *** The analysis level should be decided by the boundary / geometrical complexities.**

- **Future Work**

  **Extensive works for winterization design procedures for various outfitting systems / collaboration (including experiment) with related parties**



# CONTENTS

*Design and Construction of Vessels Operating in Low Temperature Environments, London UK.*

# FACTORS INFLUENCING THE CHOICE OF AN ICE CLASS

**R. Bridges**, Lloyd's Register, UK

## SUMMARY

Ice class rules provide a fundamental level of safety for a ship when navigating in ice covered seas. There are a number of different ice classes that exist and the choice of the appropriate one is a complex decision. In a simplistic form, an ice class may be chosen based on the maximum ice thickness the ship is intended to operate in. In reality there are more factors to consider than just the ice thickness alone. This paper will put forward the proposal of three principal factors to consider in choosing an ice class; environmental, operational and ship design factors. Each factor has a number of issues which will be explored and discussed in detail. The paper is intended to provide a better understanding of the factors involved and to enable owners and builders to select an ice class that is suitable for the anticipated operation.

## 1. INTRODUCTION

Ships are principally designed, built and operated for service in water, such as in seas, lakes and rivers. However, in some regions around the world the air temperature may drop below zero and cause the water to freeze and become ice. When the low temperatures persist over a long period of time even the sea will freeze over. This ice poses a formidable barrier to the navigation of ships.

Historically, voyages into these waters were of discovery and exploration, and indeed, even today many voyages are research orientated and are still in the pursuit of greater knowledge. However, since the 1900's commercial ships have been navigating in ice infested waters and, especially in recent years, there has been a noticeable increase in traffic operating in cold climates, due to reasons such as globalisation, the search for hydrocarbon resources in remote locations, advances in ship design and global warming. This trend seems to be continuing into the future, which will result in an increase in shipping activity in ice covered waters, and hence, an increasing importance in ensuring these ships are suitable for the harsh environment.

To protect ships when navigating in ice, suitable strengthening of the ship is required. This is achieved by a special set of Classification Rules, the ice class rules. Ice class rules provide standards for additional strengthening of the hull structure and machinery, and increased engine power to enable the ship to force its way through the ice.

Ice classes are historically based on ice thickness. The lowest ice class is intended for navigation in thin ice, whilst the highest ice class is suitable for very thick ice conditions. However, there are more factors to consider than just the ice thickness. For example, National Administrations may impose limitations to ships outside a certain ice class, or an owner may wish to have a higher ice class from a commercial point of view. These factors may be broadly divided into three groups, which form the criteria for the selection of ice class:

- *The environment*

This includes the ice thickness intended to navigate in, but also comprises of ice properties, ice ridges, sea states, and geological features.

- *The operational scenario*

This includes factors such as icebreaker assistance, convoys, speed when navigating in ice and the National Administration requirements.

- *The ship design*

A factor that influences the selection of ice class is the capability of the ship in ice, which depends on the size of ship, the hull form and propulsion.

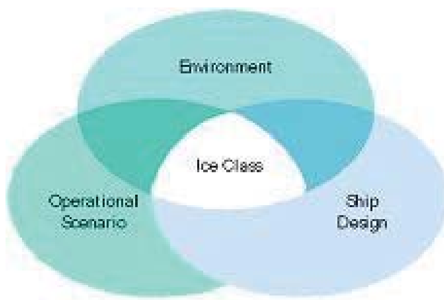



Figure 1: Criteria for ice class selection

This paper explores and discusses some of these factors, without being too detailed in the theoretical mechanics of ice class, with the intention to improve the awareness of the issues and to assist owners and builders in recognising some of the factors that need to be considered when choosing an ice class.

## 2. PAST AND PRESENT ICE CLASS RULES

### 2.1 CATEGORISATION OF ICE CLASS

Ice class rules are usually divided into two sections; first-year ice and multi-year ice:

2.1 (a) First-year ice

This is sea ice that is present during the winter only, as in the summer the ice melts entirely. Hence, the ice is only formed over one year. Since the ice has only one winter to accumulate, the ice is typically a maximum of 2m thick with low ice strength properties [1]. Typical examples of first-year ice conditions are in the Baltic and St Lawrence Seaway.

2.1 (b) Multi-year ice (or Polar ice class)

Multi-year ice has survived at least one summer and as such may be 3m or more in thickness [1]. In addition, the ice is usually much stronger than first-year ice. Multi-year ice class rules are for ships intended for areas of operation such as the Arctic or Antarctic.

## 2.2 BRIEF HISTORY OF LLOYD'S REGISTER ICE CLASS RULES

The following historical review of Lloyd's Register's rules is included to provide some context on how the ice class rules have developed over the years and to provide an insight into some of the inherit assumptions and underlying basis in the rules that are existing today.

Lloyd's Register's first ice class rules were developed in the 1920's and were published in the 1924 Rules and Regulations for the Classification of Ships [2]. These rules introduced requirements that were principally for reinforcement to the bow region. Ships complying with these rules were assigned the notation of 'Strengthened for Navigation in Ice'.

The major content of the ice class rules did not change in the next thirty years and it was only until 1958 when they were next updated [3]. At this time Lloyd's Register introduced three ice classes; ice class 1, ice class 2 and ice class 3. The requirements were far more detailed and graded depending on the severity of ice conditions. This was a significant step and is argued to be the origination of the rules today, since they provided a tiered system based on different ice thicknesses. These rules were simple in application, called percentage rules, due to the structure being increased by a percentage amount. In 1968 ice class 1* was added in response to the increasing ice going capability of the ships being built.

In 1971 the Finnish and Swedish Maritime Administrations published a set of ice class rules that were seen as another significant step [4]. Rather than a simple percentage increase in the structure, an ice pressure corresponding to the ice thickness was used to determine the structure. This provided a technical justification for the increased structure. In 1972 requirements for compliance with the Finnish-Swedish Ice Class Rules of 1971 were included in Lloyd's Register rules, assigning notations 1AS, 1A, 1B and 1C.

The Lloyd's Register ice class rules were then reviewed and developed in 1981 and were introduced into the rules in 1985 [5]. At this time there was considerable navigation in first-year ice, but also in multi-year ice found in the Polar Regions. The aim of these rules was to provide a rational approach to all ice classes, both first-year and multi-year ice class rules. The requirements were based around a principal equation:

$$t = K\,\alpha\,\beta\,y$$

Where $K$ = theoretical equation using span, spacing, nominal ice pressure and yield stress, $\alpha$ = correction factor for longitudinal position and class notation (rationalised to fit with Canadian and Finnish Swedish rules), $\beta$ = correction factor for vertical position (eventually determined to be 1.0 for first-year ice), and $y$ = correction factor for power and mass.

The Finnish-Swedish ice class rules were updated in 2002 [6] and these became the de-facto set of rules for ships to be built to for first-year ice. They included an engine power calculation recognising the capability of the hull form in ice. These rules were implemented in the Lloyd's Register 2003 rules.

In 2006 after more than ten years of development, the IACS Polar Class Ship Rules [7] were finalised. These rules were developed in conjunction with the IMO Arctic Guidelines for Ships Navigating in Ice Covered Seas [8]. The intention of producing these two sets of requirements was to provide a harmonised set of rules and regulations for Polar ships using state-of-the-art techniques.

## 2.3 WHAT IS AN ICE CLASS?

| Lloyd's Register Ice Class | Finnish-Swedish Ice Class | Ice thickness (metres) | | Polar Ice Class | Ice Description (Based on WMO Sea Ice Nomenclature) |
|---|---|---|---|---|---|
| | | | | 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 which may include multi-year ice inclusions. |
| | | | | PC 4 | Year-round operation in thick first-year ice which may include old ice inclusions |
| | | | | PC 5 | Year-round operation in medium first-year ice which may include old ice inclusions |
| 1AS FS(+) 1AS FS | IAS | 1.0 | | PC 6 | Summer/autumn operation in medium first-year ice which may include old ice inclusions |
| 1A FS(+) 1A FS | IA | 0.8 | | PC 7 | Summer/autumn operation in thin first-year ice which may include old ice inclusions |
| 1B FS(+) 1B FS | IB | 0.6 | | | |
| 1C FS(+) 1C FS | IC | 0.4 | | | |
| Note, although PC6 is equivalent to 1AS FS (and PC7 to 1A FS), the requirements will differ based on their intended operation. For example, the extents of reinforcement will be greater on the multi-year ice class due to the larger variation in ice conditions, escort operations etc. | | | | | |

Table 1: Finnish-Swedish and IACS PC ice classes [6, 7]

Lloyd's Register first-year ice class rules (and those of the Finnish-Swedish ice class rules) correspond to nominal ice thicknesses. Unlike the first-year ice class rules, the Polar ice class rules are related to a description of the ice conditions, as shown in Table 1.

Ice class rules provide protection of the ship when navigating in ice. As the ship sails in ice, the ice will come in contact with the ship's hull and this contact will apply a substantial pressure to the hull structure. To resist this, the strength of the hull needs to be increased. Likewise the propeller and rudder will be subjected to ice piece impacts and require additional strengthening to prevent damage. The ship will also need an enlarged engine power to overcome the resistance of the ice and in some cases, to break the ice. As such, the ice class requirements are split into three parts:

- Hull strengthening
- Increased engine power
- Machinery strengthening

2.3 (a) Hull strengthening

The hull strength is increased along an icebelt. For first-year ice class this is taken between the maximum and minimum draught waterlines which the ship is intended to navigate in ice, as shown in Table 2 below. For multi-year ice class, this also includes the bilge and bottom regions. The hull strengthening is increased in relation to the ice class and location (usually divided into forward, midship and aft regions).

2.3 (b) Engine power

There is a marked increase for engine power with ice class to provide additional power to navigate through ice. For the first-year ice class rules, the engine power calculation is sensitive to hull angles, so that ships designed with an icebreaking hull form receives a reduced engine power.

Ice model tests are often performed to provide an indication of the ice capability of ship and an accurate level of engine power. They also provide the opportunity to optimise the hull form for icebreaking.

2.3 (c) Machinery

The propulsion train of the ship is subject to additional loads, and hence, the propeller, shaft, and reduction gears are increased to provide protection against ice loads.

The steering gear, sea water inlets, overboard discharges and fire pumps also require arrangements to protect against ice damage and blockage.

## 2.4 WHAT IS THE DIFFERENCE IN ICE CLASSES?

The steel weight, machinery and engine power increase for each advancement in ice class. The increased steel weight and requirements will have a subsequent increase in the cost of the ship. This cost is usually offset by the enhanced operation and/or lower icebreaking fees. The added steel weight is approximately 30% for each ice class [9]. However, this depends on size and arrangement of the ship. For example, a large tanker will have a larger draught variation than a passenger ship and hence, a greater icebelt region to be strengthened.

With an increasing ice class the hull structure is increased, both for the area of reinforcement and also the thickness of the steel within this region. For example, Table 2 shows for ice class 1C, the extent of the icebelt is 0.4m above the load waterline. Whilst for 1AS this is increased to 0.6m. The extent below the waterline will follow the same principal. Also, the structure is increased within this region. For example, the shell plating thickness may be 20mm on a conventional ship; but this will be increased to, say, 22mm for 1C, and 25mm for 1AS.

| *Plate extents (in metres)* | | |
|---|---|---|
| *Ice Class* | *Above LWL* | *Below LWL* |
| 1AS | 0.6 | 0.75 |
| 1A | 0.5 | 0.6 |
| 1B | 0.4 | 0.5 |
| 1C | 0.4 | 0.5 |

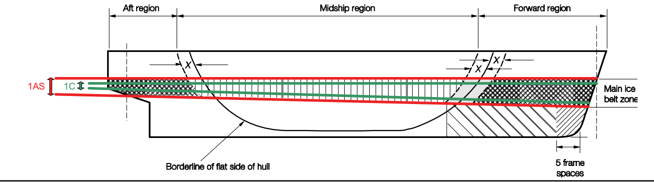



Table 2: Hull reinforcement regions

## 3. THE ENVIRONMENT

The region where the ship is intended to operate, and the environment therein, is one of the major contributing elements in the choice of an ice class. Ice classes are predominantly chosen for the maximum ice thickness in the area of operation. However, there are a variety of different ice conditions, which is partly due to the fact that each region has very different metrological and geographical features that influence the formation of ice. In addition, the prevalent weather conditions in each location will have a bearing on the design of the ship, for example icing, snow, lack of daylight and fog are often experienced in these remote locations, although they are usually dealt with outside the scope of ice class.

The sea ice conditions can be composed from three fundamental features; air temperature, sea conditions and geographical features:

*Air temperature*
When the air temperature drops below zero, sea ice forms, and with a lower temperature, a greater ice thickness is produced. Thus, regions of extreme low temperatures are likely to have the most inhospitable ice conditions. Other considerations include the duration of time at low temperatures (which defines the length of the ice season) and the rate of temperature drop (which effects the ice formation, thickness and strength).

*Sea conditions*
Sea conditions will dictate many of the ice characteristics. For example, regions of low salinity, such as the Baltic and Great Lakes, will form ice earlier and also be a stronger ice. The sea state and currents will also greatly alter the shape of the ice, moving icebergs and creating ice ridges.

*Geological features*
Geological features include items such as the depth of water (shallow ice is generally more susceptible to temperature) and proximity to land (for example, shore-fast ice is formed when the cold land cools the water). Ice may also form on geological features, which create navigation restrictions, such as reducing the width and draught of channels and in ports.

### 3.1 ICE CHARACTERISTICS

A number of characteristics of sea ice [10] are integrated into the ice class rules. The following are some of the main characteristics that have an impact on the ship and operations:

*Strength*
The strength of ice is determined by many factors, although the age tends to be the most important, as with time salt seeps from the ice, which increases the strength [11] and also makes the ice clearer (multi-year ice is often seen as a transparent/blue colour). The strength of the ice will have a direct impact on the ice class. Inherent within the ice class rules is the ice pressure which assumes a certain level of ice strength. The strength of ice is divided into a number of constituents such as compression, tensile, shear and flexural (with flexural being the most important due to the ship-ice interaction commonly causing the ice to fail in bending).

*Thickness*
Ice class rules allow an ice thickness to be chosen and used within the calculations and so this has a direct bearing on the requirements. It is generally assumed that a thicker ice will have a larger contact area and hence, exert a greater pressure on the hull or propeller [6, 12].

*Ice drift (also called ice field pressure)*
The movement of an ice field, due to currents or wind, will generate pressure acting on the ship and/or closing leads and ice channels. The effect of ice drift is not accounted for explicitly in the ice class rules. In the extreme case of the ship being stuck in ice, and subject to compressive forces from the ice movement, the loads would be too high for an economical merchant ship to be designed to.

*Ice ridges*
Ice is almost never level and forms in a multitude of shapes. For ships, one of the most important ice formations is ice ridges. Ice ridges are usually formed when two ice sheets are pushed together and fuse into one. The resultant pressure forces the ice into walls or lines, which extend above and (significantly) below the waterline. These ridges may be many metres thick, for example 14 or 15m, and pose a significant challenge for the ship. In most cases the ship will try to navigate around them. Where this is not possible they may need to ram or flush the ice, to allow the ship to navigate through. Because of this, discontinuities, such as ice ridges, are included in the rules. However, due to the nature of the randomness of the discontinuities, the exact effects are normally accounted for by safety factors. But some effects are explicitly defined, such as the longitudinal strength (hull girder) calculations in the IACS PC rules, which account for ramming against ice ridges where the bow rises out of the water onto the ice and causes a global bending of the ship.

*Icebergs*
Icebergs are usually excluded from the ice class rules. They may be treated as a separate scenario when it is necessary, and calculated as a collision impact resulting in strengthening in the bow region.

*Ice channels*
When a ship navigates in ice, the broken ice left in its wake is often referred to as an ice channel. This channel is made up of many ice pieces which have a different load pattern on the ship, when compared with navigating in level ice conditions. In the regions where icebreakers operate, such as the Baltic, parts of the rules (engine power) have been developed using these ice conditions.

*Ice extent and duration*
The extent, or area, which the ice covers and the duration of time that the ice is present, will differ from region to region, and year to year. Therefore it will affect the length of time the ship is in ice, and consequently the range of impacts and the likelihood of encountering ridges.

## 4. THE OPERATIONAL SCENARIO

The ice class rules cover a vast range of ice conditions and hence, a vast range of operations. Examples of operations include; acting in convoys, ramming against ice ridges, manoeuvring in channels and berthing, all of which will have a different ice loading. Additionally, the way the ship operates in these ice conditions will have a significant impact on the integrity of the ship. Therefore, extreme care and caution is required and the Master should be experienced in navigating in ice.

### 4.1 SPEED IN ICE

The ice class rules are intrinsically linked with the speed of the ship. A ship with a larger engine power will be able to travel faster in ice, and will therefore impact the ice at a higher speed, or will be able to navigate into thicker ice conditions, or provide greater inertia to proceed through ice discontinuities. Thus, the ship will be subjected to larger ice forces.

In the Finnish-Swedish ice class rules, the ice/speed relationship is represented by the engine power in the function, *k*, which influences the ice pressure [13]. The function *k* is often termed the “aggression factor” and includes the size of the ship:

$$k = \frac{\sqrt{\Delta P}}{1000}$$

Where, $\Delta$ is the ship displacement and $P$ the installed engine power.

There is a similar formula included in the IACS PC rules [12]. It uses an assumed value for ship speed and ice thickness (which increases with ice class):

$$ice\ force = fa\ P^{0.36} \Delta^{0.64}\ V^{1.28}$$

Where, *fa* is a factor to account for the hull form at the bow, $P$ is the ice crushing strength, $\Delta$ is the ship displacement, and $V$ is the speed.

Above all, it should be noted that the ship speed in ice is an operational consideration since it is at the discretion of the master and/or icebreaker captain. The navigational experience of the master in ice conditions is particularly important. An experienced master will be able to navigate around the worst ice conditions, knowing when to speed up and slow down. Indeed, an experienced master will provide a measure of safety. Previous ice class rules account for this by calibration based on service experience and damage studies [14]. The rules do not account for inexperienced masters which might appear due to increasing demand for ice class ships. Although this situation may also be offset by improved ship technology, such as satellite information which will allow better selection of routing, and provide tools to mitigate this. Hence, the speed in ice is variable and very difficult to define within the ice class rules.

One concept that has been developed to accommodate this is the speed/ice curves [15]. Speed/ice curves are graphs that plot the ship speed versus the ice thickness. The curves are plotted for an assumed ice condition, and a number of graphs are developed to cover a range of ice conditions. See Figure 2 below.

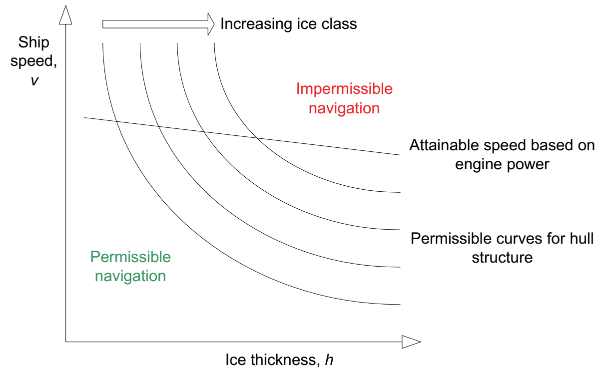



Figure 2: Speed/ice curves

### 4.2 DURATION OF VOYAGE IN ICE

One key operational aspect is the expected length of time spent navigating in ice. A ship on an international voyage, say from the US to the Baltic will spend a relatively short time in ice. Whilst, conversely a ship that operates on regional voyages, say all year round in the Baltic, will spend all winter in ice. Since the design of an ice class ship is detrimental to the open water performance (as the ship becomes heavier and slower), the ice class should be kept to a minimum so the ship is still competitive and safe in open water. Therefore, a ship making an occasional visit could select a lower ice class, assuming there was adequate icebreaking support to ensure the ship navigates safely.

### 4.3 ICEBREAKERS - ESCORT AND CONVOYS

Navigation in ice can be broadly categorised into five modes of sailing:

- Independent — Level ice
- Independent — Channel ice
- Icebreaker escort — Singularly
- Icebreaker escort — In convoy
- Towed by an icebreaker

Implicitly, the mode of sailing is assumed within the ice class rules. The IACS PC rules are founded on a glancing impact to the bow, and therefore based on an icebreaking capability, i.e. without escort. However, the mode of sailing is usually related to the operational ethos of Maritime Administrations, who provide icebreakers to operate in that region. For example, the Finnish-Swedish ice class rules are based on a continuity of trade principal, so there are always icebreakers present to make and maintain ice channels, and escort ships when necessary. Thus, the rules have been calibrated for these modes of operation. Should a ship desire to act independently with icebreaking capability, then these rules may not be particularly applicable, e.g. due to the additional ramming when acting independently the ship will be frequently impacting the ice in the bow region, and therefore a substantial bow structure will be required that may otherwise not be. Equally, on the approach to the

Russian Baltic port of Primorsk there are only limited numbers of icebreakers operating, so the voyages tend to be made in convoys. Therefore, the ice loads experienced by a ship operating in an ice channel will be different to those of operating in level ice or in convoy.

Additionally, there is an inherent link with the merchant fleet that operates within the region, see Figure 3 below. For example, a region where a large number of small ships (with a small engine power) are operating may require a large number of icebreakers to ensure the safe passage of these ships. Thus, the rules are developed to reflect this scenario.

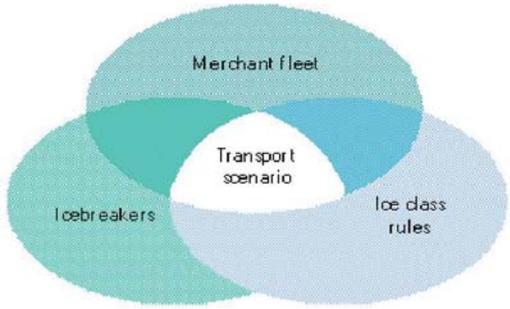



Figure 3: Relationship between ice class rules, icebreakers and with the merchant fleet

### 4.4 ADMINISTRATION REQUIREMENTS

Maritime Administrations provide icebreakers to ensure safe and effective navigation for their territorial waters. They require fairway dues to reinforce traffic restrictions. The fairway dues are usually in relation to the ice class [16]. A high ice class ship will have a low fee, and a low ice class ship will have a high fee. Thus, there is a financial incentive to have a higher ice class. It also needs to be considered in conjunction with the estimated frequency of visits and ice conditions predicted in the area of operation.

Restrictions are imposed by Administrations when ice conditions are above that of a certain ice class. For example, the ice thickness maybe 0.7m and therefore, only ships with an ice class of 1A or above would be suitable. The restrictions are determined historically or on the current ice conditions. For instance, the Finnish and Swedish Administrations set ice class restrictions on a weekly basis, whilst in Canada there are two systems that exist, one with historically set zone restrictions and the other that assesses the actual conditions with the vessels capability (ice shipping regime [17]). Previous restrictions can be obtained for the intended area of operation to predict the ice class required to enable the ship to visit when desired.

### 4.5 OWNER REQUIREMENTS

A ship owner may want to have a preferred ice class to improve the desirability of the ship since this will provide additional flexibility to the capability of the ship. The owner may also want to fulfil a particular requirement formed by a Charter party. These requirements will often differ on a case-by-case basis and are therefore, variable in nature. They are difficult to quantify and prescribe.

## 5. SHIP DESIGN

### 5.1 HULL STRENGTHENING

The design of a ship in ice is a balance of optimisation between icebreaking performance and open water performance [18]. For example, a bulbous bow is fitted to achieve good open water performance; but it has poor icebreaking capability. Conversely, if a sharp icebreaking bow is fitted the ship would have poor qualities when navigating in open water due to the increased resistance and seakeeping motions. Therefore, a compromise must be sought.

The level of hull strengthening in ice class rules is related to the ice conditions, where a higher ice class ship will be expected to navigate in thicker and more difficult ice conditions. Most rules use an ice scenario as a design basis. For the IACS PC rules this is a glancing impact to the bow region. For the Finnish-Swedish ice class rules this is contact with level ice, where the process assumes the ship contacts the ice at an angle to create a force which precipitates bending of the ice until breakage, see Figure 4 below. The exact contact loads vary due to ice conditions, ship speeds, hull angles and hydrodynamic components [19]. The greater the side shell angle is, the improved probability of promoting bending of the ice and breakage. Thus, a ship that has hull angles to efficiently break the ice would experience lower ice loads and would be a lower ice class.

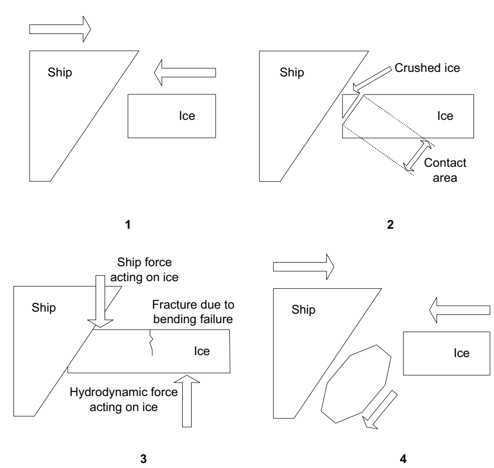



Figure 4: Ice breaking mechanism

Both the Finnish-Swedish engine power requirements (and by interference $k$ for the hull strength) and IACS PC hull requirements are determined taking into account the bow shape. Hence, for a highly optimised icebreaking ship, lower loads will be applied and a higher ice class attained, compared with a corresponding ship with the same scantlings but with no icebreaking optimisation.

Another consideration is the extent of reinforcement to the hull. The ice class dictates the extents of reinforcement and the application to other areas of the hull by using hull area factors. There is no implicit design scenario for the other regions. For example, in the aft region, one of the principal scenarios is going astern in the ice (perhaps after stopping due to a ridge) and the strengthening should be designed for such a manoeuvre, rather than being based on damage correlation factors.

This principal also applies to the five modes of navigation in ice (as given in 4.3). The Finnish-Swedish ice class rules are based on the operation with icebreakers. Therefore, the frequency of loads, the ice pressure, patch load size and location on the hull, are all based on the same scenario. For example, when operating in an ice channel many small ice pieces will impact the bow region and have large loads on the shoulders (due to the relative large thickness at ice channel edges and subsequent impact when turning in the channels). These factors will be changed for independent operation in level ice.

The ice pressure (and thus ice class) is also linked to the shape of the hull. For example, a small icebreaking ship may be designed to push the ice underneath the hull requiring hull strengthening in the bottom region, whereas a large merchant ship may have a blunt hull form, a deep draught and flat sides to produce ice pressure largely on the shoulders and sides rather than the bottom regions.

The design equations integrated within the ice class rules will also generate an influence. Most hull calculations in ice class rules are determined by plastic, rather than elastic, deformation theories. Plastic deformation theories assume the ships shell plate will deform (rather than revert to the same shape as would in elastic deformation theories), and to ensure no deformation in the shell plating, a higher ice class should be considered.

## 5.2 PROPULSION

The selection of the propulsion system is an important part in determining the capability of the ship in ice. If the ship possesses too large an engine power, it will go faster, which will increase the ice loads, and/or be able to force its way into very thick ice, which may cause damage to the hull. Whilst conversely, too small engine power will lead to the ship becoming entrapped in ice and subjected to high compressive ice forces. Furthermore, excess power is necessary to maintain inertia over ice ridges. So there is a delicate balance between the engine power and hull strength.

The ship speed is relatively slow when navigating in ice. Therefore, there is a need to have power at low speeds and in particular torque. However, increasing the ice performance of the propulsion system will usually decrease the open water performance and increase the cost. The simplified diagram below, Figure 5, illustrates some of the optimisation choices, where the choice of propulsion is a combination of the engine power and propeller, and each will have a different driving factor. For example, the propeller may be tuned to mill ice, but this would require additional power from the engine.

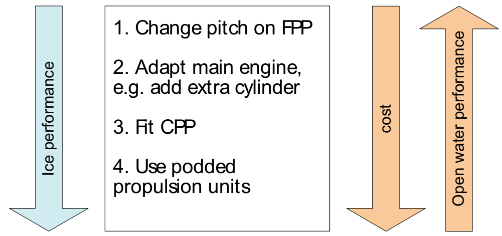



Figure 5: Propulsion optimisation for ice navigation

Another consideration is the mode of operation (from the five modes given above). The Finnish-Swedish ice class rules are based on navigation in the Baltic. The engine power formula is based on the research carried out on the traffic in the Northern Baltic [20]. The research showed that the majority of time was spent by merchant ships in independent operation in a channel made by an icebreaker when navigating in the Baltic. The research report developed the engine power calculation based on a continuity of trade, so the ship must be able to proceed at five knots in these ice conditions. Additionally, Lloyd's Register ice class rules provide an alternative formula for icebreaking capability in first-year ice. So ships have different engine power requirements for different operations in ice.

Other considerations that are not explicitly included in the rules are:

- The manoeuvrability of the ship. This includes the ability of the propulsion system to change the direction of the ship. A ship with a CPP will be able to reverse quicker than a FPP, and therefore will be able to ram against ridges more efficiently. Navigating in ice is not usually in a straight line, so the manoeuvrability of the ship is a major issue. Ships that can manoeuvre around ice features, such as ridges, will have lower ice loads and can therefore select a lower ice class.
- Another consideration is the hull form. As explained above, a small icebreaking ship may push more ice under the hull and cause more ice impacts to the propeller.
- Finally, the propulsion system should be selected to accommodate vibrations due to fluctuations and repetition of ice impacts. This is not usually included within the ice class rules, but it has a significant affect on the performance of the ship, crew and environment.

## 6. SIZE OF SHIP

Traditionally, ice class ships were small ships, in comparison with worldwide shipping standards, and the predominant nature of transportation was limited to feeder service. This is due to it being advantageous to use ships that are optimised for ice, predominantly in ice which only covers small regions. Nowadays, the size of the ship is vital in the selection of the ice class. Some of the reasons are discussed below.

Small ships have a relatively smaller engine power compared with larger ships, and they are more likely to become entrapped in the ice due not having enough power to force their way out of the ice. The exception is of course the icebreakers, being small size, but having large power to compensate. Equally, larger ships are less likely to become trapped in ice due to the greater inertia they have to move through ice ridges and obstacles. However, larger ships are less manoeuvrable, and thus turning in a channel and following leads in ice becomes more arduous, exposing the hull to larger ice loads.

The hull form also changes following the increase in size of ship. Small ships will tend to have more curved surfaces, which may push more ice underneath. Since, the extent of reinforcement in the ice class rules is based on numerical dimensions, rather than hull form, this may lead to damages in the lower regions.

Due to economies of scale and ship building practices, there is a general trend towards bigger ice class ships. An example of this is the emergence of aframax and suezmax oil tankers. These ships have a wider beam than icebreakers [21]. When they are designed to navigate in the Baltic, and according to the Finnish-Swedish ice class rules, they would require either escort with two icebreakers to provide an ice channel of sufficient width, or additional strengthening and engine power to navigate independently.

Another issue with the larger ships is the calibration of the existing requirements, which has been carried out on small ships. For example, the engine power formula was developed based on ships with proven service experience, and applying the formula to large tankers the values may give unrealistic results. Figure 6 illustrates this trend. For the small ships the results are relatively tolerable. While for larger ships, the formula increases substantially. This is more noticeable with higher ice class. One solution is to carry out ice model tests, which provide an alternative means for the calculation of engine power. Although a direction comparison should not be made with the formula results as these ships were optimised by refining the hull form for icebreaking, and enhancing the engine power for high torque at low speeds.

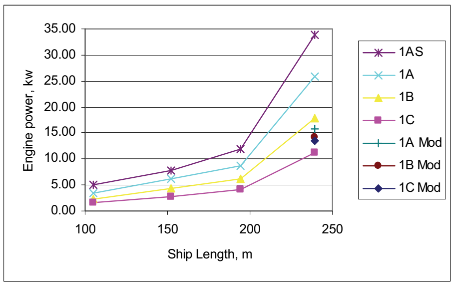



Figure 6: Engine power requirements (Finnish-Swedish ice class rules)

## 7. FRAMEWORK FOR SELECTING AN ICE CLASS

Based on the above, the factors influencing an ice class may be divided into three elements; the environmental factors, the operational factors and the ship design factors.

These factors may be further sub-divided into individual components, which may be considered separately. For example, ice strength is a factor of the environment and for the same operating conditions a higher ice class ship can navigate in a stronger ice.

Each component can have a scale and a corresponding description. This provides a framework for the choice of an ice class. See Tables 3a and 3b below.

| | Environment (ice properties) | | | | | Operation | | | | | Design | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C1a | C1b | C1c | C1d | C1e | C2a | C2b | C2c | C2d | C2e | C3a | C3b |
| | **ice strength (flexural)** | **ice thickness, h** | **ice drift** | **ice ridging** | **ice extent** | **speed** | **frequency** | **operation** | **Crew** | **administration restrictions** | **hull form optimisation** | **ship size and power** |
| Low | Weak | Thin | no compression | no ridges | small area | slow | Occasional | escorted | highly experienced | low ice region & time | Non ice class | Small |
| Moderate / Low | Medium weak | Medium thin | slow closing | few small ridges | moderate small area | meduim slow | Regularly | Occasional independent | moderate experience | moderate low ice region & time | Ice optimised low | Medium small |
| Moderate / High | Medium Strong | Medium thick | quick closing | large ridges | moderate large area | meduim fast | Often | Occasional ramming | little experience | moderate high ice region & time | Ice optimised high | Medium large |
| High | Strong | Thick | high compression | many and large ridges | large area | fast | Continuous | Icebreaking operations | no experience | high ice region & time | Icebreaking | Large |

| | Environment | | | | | Operation | | | | | Design | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C1a | C1b | C1c | C1d | C1e | C2a | C2b | C2c | C2d | C2e | C3a | C3b |
| | **flexural strength Mpa** | **m** | **m/s** | **concentration %** | **nm2** | **knots** | **days in ice** | **operation** | **Crew** | **administration restrictions** | **bow angle** | ***k*** |
| Low | 0.2 | 0.4 | 0.2 | 2 | 200 | 5 | 10 | escorted | highly experienced | 1C | 20 | 11 |
| Moderate / Low | 0.3 | 0.6 | 0.5 | 4 | 250 | 7 | 40 | Occasional independent | moderate experience | 1B | 30 | 12 |
| Moderate / High | 0.4 | 0.8 | 0.8 | 6 | 300 | 9 | 80 | Occasional ramming | little experience | 1A | 40 | 13 |
| High | 0.5 | 1.0 | 1.1 | 8 | 350 | 11 | 120 | Icebreaking operations | no experience | 1AS | 50 | 14 |

Table 3a and 3b: Factors influencing the ice class

| | Environment | | | | | Operation | | | | | Design | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C1a | C1b | C1c | C1d | C1e | C2a | C2b | C2c | C2d | C2e | C3a | C3b |
| Low | 1 | 1 | 1 | 1 | 1 | 1 | 1 | **1** | 1 | **1** | 1 | 1 |
| Moderate / Low | 2 | 2 | **2** | 2 | **2** | **2** | **2** | 2 | 2 | 2 | **2** | 2 |
| Moderate / High | **3** | **3** | 3 | **3** | 3 | 3 | 3 | 3 | **3** | 3 | 3 | **3** |
| High | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 |

Table 4: Numerical example for Baltic LNG carrier

## 7.1 EXAMPLE - BALTIC LNG CARRIER

As an example, the framework above has been applied for a hypothetical LNG carrier operating between the Gulf of Finland to USA. See Table 4. To quantify the framework, the factors have been given numerical values as far as practicable.

It should be noted that the determination of each value is given by simple engineering judgement with the aim of illustrating the process, rather than fully analysing for the solution. However, some factors cannot be easily expressed in numerical form, such as the crew competence and experience, and therefore are retained as descriptions.

It should also be noted that the upper and lower limits for each factor are difficult to define accurately. It is an aspect requiring careful consideration. In this instance, the Baltic and Finnish-Swedish ice class rules are used as the basis for selection. For example, the ice conditions along the route (Gulf of Finland) are used in comparison with those in the whole of the Baltic.

*The environment*
The ice conditions in the Gulf of Finland are capricious in comparison with the Baltic as a whole. The fresh/brackish water makes relatively strong ice and level ice thickness may reach 0.8m in the Gulf of Finland. Ice ridges are often encountered in the Gulf due to the prevailing southwesterly winds.

*The operation*
LNG carriers usually operate on specific trade routes and have fixed schedules. Therefore it is envisaged that the LNG carrier would require icebreaking assistance to ensure the ship is not trapped in ice and prevent the delivery of gas.

Also due to the fixed schedule, the ship would be expected to operate throughout the winter. However, the time in ice would be short compared to that in open water. It is also envisaged that the crew would be comprised mainly from open water operations rather than from ice operations.

*The ship design*
Since a majority of time would be spent in open water, the hull form would not be designed for ice navigation. However, LNG carriers have a slender hull form (due to the requirement for high open water speed), so moderate ice performance could be achieved. LNG carriers are normally larger ships in comparison with traditional ice class ships, and due to the open water requirements, the ship would have a relatively high engine power.

*Results*
For the purpose of illustration, each ice class has been assigned a numerical value: 1AS = 4, 1A=3, 1B=2 and 1C=1. An indication of the suitable ice class has thus been achieved by comparing the average numerical values for each factor with the numerical values for each ice class. See Table 5 below.

It can be seen that if the selection of ice class were based solely on either the environmental conditions or the design of the ship, the ice class would be 1A. But by including the operational factors, the methodology indicates an ice class of 1B.

| *Element* | *Average* | *Ice Class* |
|---|---|---|
| Environment | 2.6 | 1A |
| Operations | 1.8 | 1B |
| Ship design | 2.5 | 1A |
| *Total* | *2.17* | *1B* |

Table 5: Numerical averages for three elements and corresponding ice classes

## 8. THE FUTURE OF ICE CLASS

As greater knowledge of the environment, ice operations, ship performance and, machinery and structural analysis is gained, the requirements for ice class will be developed to reflect this learning. The development of ice class rules is complex and involves many parties including (but not limited to) Classification Societies, Maritime Administrations, research institutes, merchant ship owners and operators. The future of ice class rules is uncertain, but some of the possible avenues of development are discussed as follows.

As explained above, the ice class rules are dependent on three key elements, the environment, operations and ship design. By describing in detail how these factors are represented in the ice class rules, users will be provided with a clear understanding of the ships capability in ice. The Russian ice class rules provide basic guidance in the form of a table. This table prescribes the seasons in which ships are allowed to navigate in the Russian Arctic. The table also covers permissible operations (with and without icebreakers) and ice conditions (related to each region in the Russian Arctic) [22]. This is used as a simplistic approach. A unified and detailed description, in terms of ice conditions (ice ridges, ice drift, etc) and operations (ramming, escort, etc) should be developed for the purpose of providing additional consistency to the ice class rules.

The speed when navigating in ice plays a significant role. The development of speed/ice curves for ice classes will enable a better awareness of the relationship between the speed and ice class. Some examples of how this may be achieved are provided in Figure 7 below.

These speed/ice curves can be developed for individual scenarios. For example scenarios of operating in brash ice, in level ice, ramming ice and going astern. Each scenario can then have a separate set of requirements for engine power and hull reinforcement. This will provide requirements that give improved reflection of the ship operations. In association, the frequency of occurrence can be investigated for each scenario. This would allow the setting of goals and rules in terms of risk (frequency of occurrence) and consequence (how the ice will impact the ship).

In particular, one future development is to provide requirements for double acting ships. Double acting ships are able to proceed forwards in thin ice and astern in thick ice, by utilising an azipod system (which acts as a pump to flush the hull). Particular operation modes for double acting ships can be integrated with individual scenarios.

The added benefit of the scenario-based rules is the user can choose specific operations to suit their needs. For example, an experienced user may or may not include ramming as a scenario. The scenario-based rules may also be adapted to take into account the localised ice conditions and thus allow the optimisation of the ship for dedicated routes. See Figure 8 below.

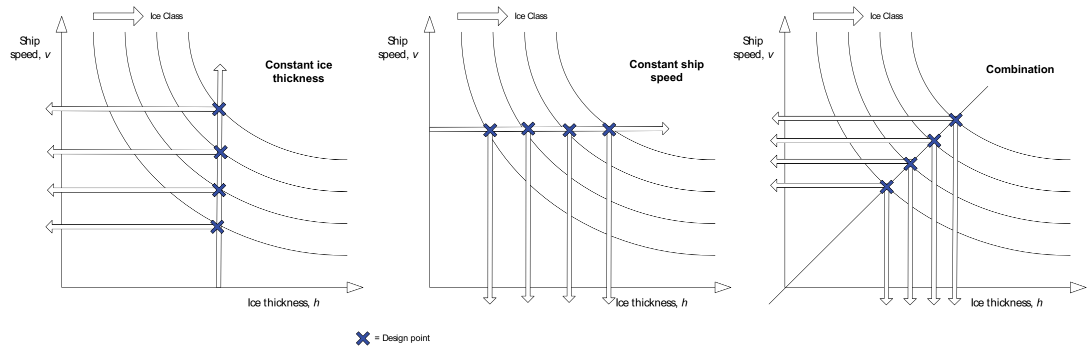



Figure 7: Various design points on speed/ice curves for the determination of ice classes

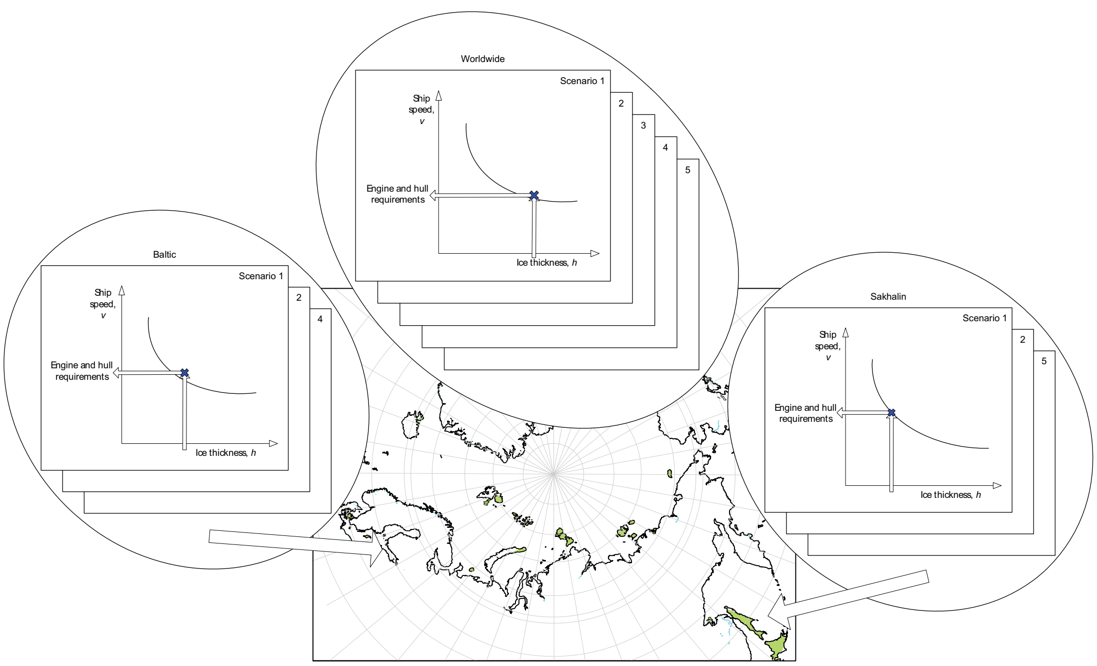



Figure 8: Localised scenarios for ice class

Another issue is the cold operations, or winterisation. Operating at low temperatures effects a number of items on the ship, such as hull material grades, engine air intakes, ballast tank heating, deck equipment, sea chest icing and stability.

The development of these requirements should be incorporated using the same principals as above, i.e. developing the requirements based on individual scenarios combining the three factors; environmental, operational and ship design.

## 9. CONCLUDING REMARKS

There is no simple answer in how to choose an ice class. Making the decision is complex and is based on many factors. All factors can be categorised into three groups; the environment, the operational scenario and the ship design. Each group of factors can then be sub-divided and quantified, so a means to choose the ice class can be achieved.

The future development of ice class has also been explored. As the number and different types of ice class ships increases, for example, LNG carriers and large tankers, the need to have a better understanding of ice class increases as well. Choosing the correct ice class is especially important. It is certainly a challenge to ensure the safety of the future generations of ice class ships.

## 10. ACKNOWLEDGEMENTS

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*Design and Construction of Vessels Operating in Low Temperature Environments, London, UK.*

# REVIEW OF AMERICAN BUREAU OF SHIPPING'S GUIDE FOR VESSELS OPERATING IN LOW TEMPERATURE ENVIRONMENTS

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**SUMMARY**

Vast reserves of gas and oil are expected to be developed in the offshore areas of the Russian Arctic. Atlantic states in North America and Europe are expected to be the chief consumers of this energy. Oil and gas operation in these environments at sea will place extreme challenges on the associated facilities and their crews. To address these challenges, the American Bureau of Shipping developed criteria by publishing the ABS Guide for Vessels Operating in Low Temperature Environments for the winterization of vessels so as to be suitable for continuous operation in arctic and polar regions.

This paper discusses the criteria to address design and operation challenges for these services in low temperatures. There are two related perspectives for this paper:

- Winterization of the vessel related to hull structure and machinery,
- Implications for design and operations due to the needs of humans.

From the first perspective, winterization issues such as: de-icing, ice effects mitigation (such as sea chest designs), piping arrangements, fire fighting arrangements and main/auxiliary machinery, are discussed.

The second perspective considers the implications on design, winterization, and operation to meet the requirements and needs of the crew. These include concerns related to: environmental controls, cold weather clothing, crew support and habitability, human performance in cold weather, safety and medical issues, personnel selection, and machinery operation and maintenance.

This paper summarizes the ABS development of standards for the winterization and operation of vessels operating in low temperature environments.

**NOMENCLATURE**

| | |
|---|---|
| DST | Design Service Temperature |
| IACS | International Association of Classification Societies |
| IMO | International Maritime Organization |
| INSROP | International Northern Sea Route Programme |
| LNG | Liquefied Natural Gas |
| LTE Guide | Guide for Vessels Operating in Low Temperature Environments |
| MAT | Minimum Anticipated Temperature |
| MARPOL | International Convention for the Prevention of Pollution from Ships |
| SOLAS | International Convention for the Safety of Life at Sea |
| WMO | World Meteorological Organization |

## 1. INTRODUCTION

The North-East Passage or Northern Sea Route has been seen as a shortcut for shipping between Europe and the Far East beginning with exploratory voyages in the 16th Century.

Since the end of the 19th Century, Russia has put considerable effort into developing the infrastructure of its Arctic Regions by developing marine transport systems. A significant experience in organizing Arctic navigation has been accumulated; however, the difficult ice conditions and the geo-political difficulties of the region have prevented the use of the Northern Sea Route by international shipping [1]. Still, Russian commercial shipping has been navigating Arctic waters for decades. Nickel has been shipped from northwest Russia year-round since the 1970's - with the help of a fleet of powerful ice-breakers [2].

Because of the increasing demand for oil and natural gas in Europe and North America, the transportation of these commodities originating from Russia will grow considerably. Huge oil and gas reserves onshore and offshore, are being made available to the international market. New gas fields are being developed in Norway (Snøhvit) and Russia (Shtokman). The LNG liquefaction plants under construction will supply the gas markets in Europe and North America within the next five years; however, this development may merely be the tip of the iceberg: potential gas reserves in the Barents and Kara Seas may dwarf the current proven reserves. It has been estimated that up to 25 percent of the world's undiscovered gas reserves may be located in the Arctic Regions.

New LNG liquefaction plants are being considered or are under construction in areas with harsh winter conditions. A plant for the Sakhalin II project in Eastern Siberia may be delivering LNG to the Far East markets by 2008, while another plant at Ust-Luga near St. Petersburg may be completed by 2009 to supply the North American market.

It is envisaged that the Arctic gas fields will be developed by using a combination of pipeline network and LNG marine transport system. Despite the use of gas pipelines, it is anticipated that a considerable number of LNG carriers will be needed for the Arctic gas trade. These ships will be required to operate in severe weather conditions with very low air temperatures, and in some routes, to navigate in ice-covered waters either during the winter months or year-round. Additionally, the Arctic natural environment is highly sensitive to pollution, and due to the remoteness of the region clean-up operations would be difficult and costly.

Vessels operating in the Arctic Regions are exposed to unique risks for which classification societies are establishing new guidance to meet the challenge of maintaining the necessary levels of safety and environmental protection for the operation [2].

## 2. ICE CLASS SELECTION FOR DIFFERENT REGIONS

The *Guide for Vessels Operating in Low Temperature Environments* (*LTE Guide*) was developed to coordinate with vessels seeking ice class. Vessels operating in areas where ice may be encountered require additional strengthening of the hull and propulsion system. Several administrations (e.g. Canada, Finland, Russia and Sweden) have additional regulations applicable for vessels operating in their waters. Other ice classes have been developed to address needs for operation in other parts of the world.

Selection of an appropriate ice class depends on the expected ice cover the vessel will encounter and the age of the ice. For example in Table 1, extracted from a proposed change to 6-1-1/Table 2 of the ABS *Steel Vessel Rules*[3], the ice designator is selected based on the first year ice thicknesses and ice concentration. Referring to Table 2, which is from a proposed change to 6-1-1/Table 1 of the ABS *Steel Vessel Rules*, one may select the appropriate general ice class. Note that these ice classes are for vessels operating independently of ice breakers.

Table 1: Ice Conditions of First-Year Ice Versus Concentration and Thickness of Ice Cover

| *Thickness of First-Year Ice Cover in m (ft)* | *Concentration of Ice (1)* | | | |
|---|---|---|---|---|
| | *Very Close and Consolidated Ice, Fast Ice (from 10/10 to 9/10 or from 8/8 to 7/8)* | *Close Ice (from 9/10 to 6/10 or from 7/8 to 5/8)* | *Open Ice (from 6/10 to 3/10 or from 5/8 to 2/8) and Fresh Channel(2) in Fast Ice (more than 6/10 or 5/8)* | *Very Open Ice (less than 3/10 or 2/8), Fresh Channel(2) in Fast Ice (6/10 or 5/8 and less) and Brash Ice* |
| 1.0 (3.3) and above | | | | Severe |
| from 0.6 (2) to 1.0 (3.3) | | | Severe | Medium |
| from 0.3 (1) to 0.6 (2) | | Severe | Medium | Light |
| less than 0.3 (1) | Severe | Medium | Light | Very light |

Notes:

1. These ratios of mean area density of Ice in a given area are from the "World Meteorological Organization Sea Ice Nomenclature", Appendix B.7, and give the ratio of area of Ice concentration to the total area of sea surface within some large geographic locales.
2. Provided the channel is wider than the ship

Table 2: Regions and Periods for Navigation in Ice for Selecting Ice Class

| *Ice class* | *Navigating independently* | *Year around navigation in water with first-year ice with the ice conditions given in Table 1* |
|---|---|---|
| A0 | Independently | Severe |
| B0 | Independently | Medium |
| C0 | Independently | Light |
| D0 | Independently | Very Light |

In July 2006, the International Association of Classification Societies (IACS) adopted unified requirements to apply to vessels constructed of steel and intended for navigation in ice-infested polar waters, with the exception of ice-breakers [6, 7, 8]. IACS defines an ice-breaker as any ship having an operational profile that includes escort or ice management functions, having powering and dimensions that allow it to undertake aggressive operations in ice-covered waters, and having a class certificate endorsed with this notation. These uniform requirements state only ice-breakers may be subject to additional requirements and are to receive special consideration for this service [6]. Table 3 (from Appendix 10/Table 1 of the *LTE Guide*) lists sea ice descriptions using World Meteorological Organization (WMO) terminology. The two most fundamental properties of ice cover are thickness and age. In Table 4, the corresponding Polar Class for the ice description is provided.

Table 3: Sea Ice Stages of Development

| *Description* | *Thickness* | *WMO Code* |
|---|---|---|
| New ice | < 10 cm | 1 |
| Nilas; ice rind | 0 – 10 cm | 2 |
| Young ice | 10 – 30 cm | 3 |
| Grey ice | 10 – 15 cm | 4 |
| Grey-white ice | 15 – 30 cm | 5 |
| First-year ice | 30 – 200 cm | 6 |
| Thin first-year ice | 30 – 70 cm | 7 |
| Thin first-year ice first stage | 30 – 50 cm | 8 |
| Thin first-year ice second stage | 50 – 70 cm | 9 |
| Medium first-year ice | 70 – 120 cm | 1. |
| Thick first-year ice | 120 – 200 cm | 4. |
| Old ice | | 7. |
| Second-year ice | | 8. |
| Multi-year ice | | 9. |
| Ice of land origin | | . |
| Undetermined or unknown | | x |

Table 4: Polar Class Descriptions

| – *Polar Class* | – *Ice Description (based on WMO Sea Ice Nomenclature)* |
|---|---|
| – 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 which may include multi-year ice inclusions. |
| – PC 4 | – Year-round operation in thick first-year ice which may include old ice inclusions |
| – PC 5 | – Year-round operation in medium first-year ice which may include old ice inclusions |
| – PC 6 | – Summer/autumn operation in medium first-year ice which may include old ice inclusions |
| – PC 7 | – Summer/autumn operation in thin first-year ice which may include old ice inclusions |

The classification societies generally have adopted the Baltic Ice Class regulations developed by Finland and Sweden for vessels intended to trade in the northern Baltic area. Table 5 lists the Ice Class and the corresponding ice thickness for first year ice [9]. Table 6 lists the Finnish-Swedish Ice Classes with the equivalent Ice Class notation of several classification societies and the Canadian administration. These equivalencies are approximate because there may be differences with the Finnish-Swedish Ice Classes [4, 5, 10].

## 3. WINTERIZATION

For the purpose of this paper, winterization is defined as the preparation of a ship for safe operation in extreme cold weather conditions by adapting the design and operation procedures to the requirements imposed by the intended service. Mean daily temperatures below 0°C are expected to be encountered by the ship during the voyage or in port. The American Bureau of Shipping recently published the *LTE Guide* to address various design, operation and crew requirements related to extreme cold weather conditions.

The *LTE Guide* addresses many design, operational and human factors aspects for vessels operating in extreme cold weather conditions. The *LTE Guide* has requirements addressing:

- Materials and coatings;
- Hull construction/arrangement and equipment;
- Vessel systems and machinery;
- Safety systems for personnel;
- Specific vessel requirements for four vessel types;
- Crew considerations;
- Crew training; and
- Supplementary Information for Weather Conditions, Vessel Operations, Administrations and Meteorological Organizations

If a vessel is to be designed to meet ice class requirements, the user is directed to the Strengthening for Navigation in Ice chapter in the ABS *Steel Vessel Rules* [3]; however, vessels operating in particular areas of the world such as the Baltic or the Russian Northern Sea Route are required to be designed in accordance with the local administration's regulations. The *LTE Guide* recognizes these requirements. IACS has recently completed Polar Ice Class Rules to unify the requirements to meet higher safety standards and changing demands of trading in the Arctic. These unified requirements will be incorporated in the ABS *Steel Vessel Rules* next year.

From the classification perspective the *LTE Guide* addresses a number of issues not covered by the Rules or vessels receiving ice class notations.

Table 5: Finnish-Swedish Ice Class Notation and Ice Thickness

| *Ice Class* | *For Navigation In:* | *Ice thickness – First Year Ice (cm)* |
|---|---|---|
| IA Super | Extremely difficult ice conditions | >100 |
| IA | Difficult ice conditions | >50 - 100 |
| IB | Moderately difficult ice conditions | 30 - 50 |
| IC | Easy ice conditions | 15 - 30 |
| Category II | Very easy ice conditions | 10 - 15 |

Table 6: Finnish-Swedish Ice Class Notation Approximate Equivalencies Between Classes

| *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 (Canadian Artic Shipping Pollution Prevention Rules) | A | A (B)[1] | 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 |

Note: 1. Finnish Icebreaking Service lists "A", [4] and [5] list "B".

## 4. CLASS NOTATION

The *LTE Guide* is applicable for vessels operating in cold climates where design service temperatures of -10°C or less are anticipated. Vessels designed, built and surveyed in accordance with the requirements of the *LTE Guide* will be assigned either of the two class notations: CCO-HR(TEMP) or CCO-HR(TEMP)+. The notation symbols signify the following:

- CCO: Cold Climate Operation
- HR: The number of hours of emergency services time, either 18 hours based on the SOLAS regulations or 36 hours for vessels operating in remote areas where rescue efforts may be delayed.
- TEMP: The Design Service Temperature (DST) for the vessel is listed. DST is defined as the lowest mean daily average temperature in the area of operation for data taken over at least a 20 year period. This definition is from International Association of Classification Societies, Unified Requirement S6, Use of steel grades for various hull members-ships of 90 m in length and above, section 6.3 [11].
- **+**: This symbol indicates the vessel's crew has been trained, and loose gear necessary for operation in low temperatures is onboard. It is recognized that in some cases a vessel may not trade in cold climates at the time of delivery. Therefore, an owner may delay crew training and installation of loose gear until such time.

Engineering plans must be sent to an engineering technical office for approval. A Surveyor will confirm the required systems are installed and are functional at the initial survey. Verification of continued functionality of the systems will occur at subsequent annual surveys. Owners who delay crew training and provision of loose gear may contact the Survey Department at any time to arrange a survey to change the class notation to indicate "**+**".

Additional resources and elaboration have been provided in the Appendices to provide users of the *LTE Guide* guidance for meeting the requirements. Contacts for national administrations that have additional requirements for vessels operating in their territorial waters are provided. Guidance in the form of temperature charts in the Arctic region along with a listing of meteorological organizations is also provided.

## 5. MATERIALS AND COATINGS

Vessels seeking a low temperature environment certification are required to have their hull structural materials selected based on the DST and appropriate material class in accordance with 6-1-1/35, Hull Structural Materials of the ABS *Steel Vessel Rules* [3]. The *LTE Guide* recognizes material requirements of other Ice Class Rules.

Any area of the vessel exposed to low air temperatures must be constructed with ductile materials suitable for operation in this environment. It is recognized that some vessels seeking a low temperature certification will not be ice classed. For design purposes, the vessel must be assigned a DST.

Structural steel class and grades for weather exposed plating and for inboard framing members attached to this plating may need to be upgraded if the design service temperature for the vessel is below the calculated design temperature of the material in the specific location.

Materials used for essential equipment exposed to the weather must be of steel or other suitable material with ductility properties at the Minimum Anticipated Temperature (MAT) for which the equipment is to operate. This is approximately 20°C below the DST. Exposed machinery includes anchoring and mooring equipment, lifting appliances such as cranes, etc. Obviously, materials intended for cryogenic service which are exposed to the weather will remain suitable for the service.

There are no additional requirements concerning welding. Welding requirements are in Part 2, Chapter 4 of the ABS *Steel Vessel Rules* [3].

Requirements regarding coatings alert users that such coatings are to be durable and resistant to abrasion or other degradation of the coating performance because of low temperatures. Appendix 2 of the Guide includes some additional information for coatings including a chart for the coefficient of friction for various coatings and ice.

## 6. HULL CONSTRUCTION / ARRANGEMENT AND EQUIPMENT

The *LTE Guide* requirements for hull construction and arrangements address issues related to prevention of tank contents freezing, protection of the personnel working on deck, protection of the environment, arrangements to reduce ice build-up on deck, and vessel stability.

### 6.1 BALLAST TANKS

Means must be provided to prevent freezing of the ballast water in the fore peak, after peak and wing tanks. For DST of -30°C to -10°C, these arrangements may be in the form of heating systems or turbulence-inducing systems such as continuous circulation of the ballast water in the tanks. However, steam heating coils are required to be installed if the DST is less than -30°C.

### 6.2 SUPERSTRUCTURE AND DECKHOUSES

The *LTE Guide* recommends the bow area be fitted with a forecastle to deflect waves and spray away from the deck areas aft of the bow. Alternatively, the shell plating in the bow area is to be flared to produce a similar effect.

Bridge wings are to be enclosed or designed to protect navigational equipment and operating personnel. External access to the navigation bridge windows is to be provided to facilitate ease of cleaning. Alternating navigation bridge windows are required to be heated.

Personnel required to perform external duties such as being a lookout when underway, security at the gangway when in port, or being on deck during loading operations are to be provided with a heated deckhouse.

### 6.3 ICE LOADS ON DECK

The *LTE Guide* requires stability information be available onboard to allow masters to recognize with the onset of icing what the consequences are for continued operation and what measures can be taken to mitigate the situation.

In particular, one of the potentially significant consequences for any ship in transit through cold weather waters is the concentration of ice on deck. Figure 1 shows representative ice build up on the deck of a tanker. While the amount of ice concentrated on deck will not normally exceed the design loads used in the analysis of the deck local strength, LNG carriers have deck features that can result in higher ice loads on deck.

Figure 1: Ice build up on a tanker

LNG carriers with either spherical tanks or membrane containment systems have large deck surfaces with a pronounced angle of inclination. Ice accumulating in such surfaces may constitute a hazard to the deck structure if the ice layer becomes so heavy it detaches and falls on the flat deck, and the impact, or the sudden ice accumulation, may exceed the deck design loads. Although the deck may be strengthened to withstand these additional ice loads, a more economical solution may be fitting external obstacles upon the inclined surfaces, such as horizontal flat bars, to prevent large ice layer detachment.

Consideration may be given to provide accessibility by the crew to these inclined surfaces in order to remove the ice layer before they become a hazard.

## 7. VESSELS SYSTEMS AND MACHINERY

The intent of this section is to address various areas related to systems and machinery where unfortunate arrangements and design lead to difficult operation and reduced reliability. Additional guidance has been provided for piping systems, such as sea chest arrangements. Additional requirements for fire safety systems and electrical systems have been provided.

### 7.1 ICE-STRENGTHENED PROPULSION SYSTEMS

For navigation in ice-covered waters, propulsion machinery, reduction gears, shaft lines, propellers and steering systems must be adequate to withstand the ice impact loads and their materials exposed to sea water temperature must be of steel or other ductile materials suitable for low temperature. Detailed requirements for the strengthening of the propeller and propulsion line as well as for propulsion machinery, reduction gears, related auxiliary systems and steering systems are available in the current ABS *Steel Vessel Rules* [3]. IACS requirements in this regard for Polar Class ships [8] have been recently made available to the industry.

Alternative propulsion systems may be proposed for vessels navigating in ice-covered waters. Azimuthing propulsors, normally known as pods, are used in Ice Class oil tankers in service and may be suitable candidates for LNG carriers. Non-conventional propulsion systems must be specially considered on the basis of their particular operational profile and loading cases.

Machinery arrangements may be required to be modified as a result of low ambient temperatures. For example, in many cases, combustion air for diesel engines is taken directly from the machinery space. In very cold climates this arrangement will cause the machinery space's temperature to become too low, possibly affecting equipment function and personnel comfort and ability to perform maintenance. Therefore, combustion air needs to be directly supplied to the diesel engines through duct work. The advantage of this ducting arrangement is the combustion air temperature can be better controlled.

Turbochargers must be designed to obtain surge free operation through the use of a blow-off air intake system. One arrangement is to install a blow-off valve in the charge air manifold of a diesel engine.

### 7.2 SEA WATER SUPPLIES

During navigation and at port in ice-covered waters, attention must be paid to sea water supplies for essential operational systems and safety systems. Current Classification Rules for Ice Class ships have requirements for sea chest inlets intended to prevent the clogging of sea water inlets by ingestion or accumulation of ice. Similarly the IACS requirements for Polar Class ships will cover these aspects. Sea water supplies are needed for the ballast system, the cooling water system serving propulsion machinery, inert gas cooling (if necessary), main and emergency fire pumps supplying the fire and wash deck system and the water spray system. The *LTE Guide* presents five sea water supply arrangements for guidance.

### 7.3 PROTECTION OF DECK MACHINERY AND SYSTEMS

Generally, deck machinery and systems are not prepared for freezing temperatures. Essential equipment and systems must be available at all times and in any temperature conditions. The methods to adapt this equipment to the Arctic environment may vary and will depend on the type of equipment and systems, their criticality for the safety of the ship and its crew, and the protection of the environment.

Essential equipment and systems ideally should be located in spaces protected from the extreme cold weather, however, it is recognized that exposure to extreme ambient weather will be unavoidable. For these situations the equipment is to be suitable for operation at the minimum anticipated temperature (MAT) which is $20^{o}C$ less than the DST.

In addition to the standard deck equipment and systems onboard any type of vessels, tankers and LNG carriers will have equipment on deck specific to their operation. This equipment must be considered essential, and therefore adequately protected and heated for operation under anticipated weather conditions. Particular attention must be made to safety systems and components such as cargo tank pressure relief valves and deck water spray systems.

Steam or thermal oil tracing and heating may be used for essential deck machinery and piping and safety systems and components, provided that the adequate redundancy is built up in the heating system to prevent its unavailability after a single failure. The maximum temperature of the steam or the heating media within the cargo area must take into consideration the temperature class (i.e., auto-ignition temperature) of the cargo being carried.

Heating and ventilation in the accommodations are to be designed for satisfactory distribution of heating at the DST. Spaces bordering exterior bulkheads may be provided with supplementary heating. In any case, the accommodation spaces are to be able to be heated to $20^{o}$C at the DST.

The lubricating oil and hydraulic oil used in rotating machines exposed to the weather must be suitable for low temperatures. The *LTE Guide* assumes heated lube and hydraulic oil sumps are provided to accomplish this. Lube oil sumps are prohibited from being heated with steam in order to prevent contamination of the oil in the event of a steam coil leak. Heat tracing or alternative means to maintain the hydraulic oil temperature may be considered. In some cases, synthetic lubricants with suitable viscosity at low temperatures may be used.

For the case of tankers and LNG carriers' spaces on deck such as, the motor room and the compressor room, the spaces should be maintained at a temperature above the minimum operating temperature of the equipment and systems contained therein. Continuous temperature monitoring with remote readings transmitted to the cargo control room would be expected.

## 8. SAFETY SYSTEMS

The Arctic and polar environments present many significant challenges to the design and use of emergency, evacuation, and rescue devices. Much of the hardware devised for such use is designed for more temperate climates, and Arctic deployment presents dilemmas. Fire mains can freeze. Use of free fall life boats, in ice conditions, may have lamentable results. Materials (such as used in life vests) become brittle. Working devices (such as sheaves, blocks, and davits) can freeze in place – refusing to move.

In this section various representative requirements addressing: protection of personnel, survival of personnel until help arrives, design of life saving/rescue equipment and personnel are discussed.

### 8.1 LIFE SAVING EQUIPMENT

Arctic conditions lead to many special considerations and risks related to life saving equipment, including:

- The presence of ice on the sea surface may inhibit deployment of life rafts and rescue boats, and also in making distance from a ship in distress
- The presence of ice on deployment mechanisms such as davits that may interfere with lowering of boats and rafts
- Crew survival/rescue time in life boats and rafts in Arctic temperatures is limited
- The thermal insulating qualities of immersion suits
- Operability of escape chutes, hatches, chutes, and doors, in conditions of ice and snow may be limited.

The types of appliances addressed include lifeboats, life rafts, rescue boats, launching stations, ice gangways, immersion suits, alarms, escape routes, and access routes. The *LTE Guide* requires life boats to meet the requirements of

- MSC Circular 1056/MEPC Circular 399, Guidelines for Ships Operating in Arctic Ice-Covered Waters [12].
- IMO Life Saving Appliances Code (from MSC.81(70) (1998) and MSC.48(66) (1996)) [13].

Life saving appliances are to be of a type that is rated to perform their functions at a minimum air temperature of -30°C in accordance with SOLAS or at the lower MAT if applicable.

In addition to the Guidelines and Code, the life boats are: to be totally enclosed, to be sized for 125% of crew size owing to bulky cold weather clothing, engine suitable for cold starting, provided with radio equipment, along with other features.

The flag state administration and the administrations responsible for the coastal areas that the vessel will be operating in may have additional requirements to those listed.

### 8.2 EQUIPMENT DEPLOYMENT

Launching life boats and rafts offer numerous risks:

- Entrance to boat stations can be obstructed by snow and ice
- De-icing equipment (steam hoses) may freeze
- Freezing of hinges, lashes, gaskets, brake guide wires, and sheaves
- Snow and ice on winches may interfere with their use
- Ice on hooks, latches and hydrostatics release couplings may interfere with their use
- Freezing of winches
- Frozen surface to which a boat is deployed (and hence laid over of her shear)

Accordingly, life boat releasing gear is to be shielded or protected from freezing for ready release or attachment. If free-fall life boats are provided, a secondary means of lowering onto ice or ice covered waters are to be provided.

### 8.3 FIRE FIGHTING EQUIPMENT

Significant risks are associated with fire fighting equipment, the most significant being the potential freezing of fluids in lines, thereby depriving crew of the use of the firefighting systems. Specific risks include:

- Freezing of fire water hoses, piping, nozzles, etc. Fire mains are charged and pressure is maintained with a topping-off pump. At –30°C, this may have to be changed and the fire mains drained until needed.
- Portable fire extinguisher storage may be obstructed or frozen
- Fire dampers may freeze in the stowage position (generally closed in temperate climates)

The *LTE Guide* requires fire extinguishing systems to be designed or located so that they are rendered inaccessible or inoperable by ice/snow accumulation or low temperature. Accordingly, equipment must be protected from freezing or located in heated compartments. Sea suctions must be designed to be capable of being cleared of ice.

### 8.4 HEATING FOR SURVIVAL

The *LTE Guide* lists a dozen spaces that must be supplied with heating in the event of an emergency. The heating system must be able to maintain a minimum of 10°C at the DST. The SOLAS regulations and the ABS Rules require emergency services to be a minimum of 18 hours duration. As an option for vessels operating in remote regions, this time can be increased to 36 hours and reflected in the classification notation (e.g. CCO-36(-40°C)+). Vessels operating in arctic regions may experience delays in rescue and medical services.

### 8.5 NAVIGATIONAL EQUIPMENT

The *LTE Guide* lists various equipment that must be installed onboard to aid navigation:

- Weather telefax receivers or similar,
- Radar systems capable of picking up ice targets,
- Adequate communications and signalling equipment,
- High powered search lights for navigating in darkness,
- Sound reception system for navigation bridge for exterior noises/signals.

## 9. SPECIFIC VESSEL REQUIREMENTS

Many vessel types have design and operational characteristics requiring special consideration. The *LTE Guide* lists additional requirements for:

- Liquefied Gas and Liquefied Natural Gas Carriers
- Bulk Carriers, Ore Carriers and Bulk/Ore Carriers
- Offshore Supply Vessels
- Oil Carriers and Fuel Oil Carriers

These vessels were chosen because they are the most likely vessel types to be constructed for increased oil and gas development in the Arctic Regions. Additional vessel types will be added when necessary.

Generally, these vessel types have particular systems and equipment onboard. The additional requirements address the functioning of these systems and equipment to ensure satisfactory operation in low temperatures.

## 10. ENVIRONMENTAL PROTECTION

In addition to the current Regulations in the IMO MARPOL Convention [14], it is anticipated that the increase of maritime traffic in the Arctic Regions would bring in the future the declaration of part of or the whole Arctic Regions as a Special Area under MARPOL Annex I and SOx Emission Control Area (SECA) under MARPOL Annex VI. LNG carriers intended to trade in the Arctic Regions should be designed to take into account all the current and foreseeable statutory Regulations for environmental protection in addition to coastal state requirements related to the same issue.

The IMO Guidelines [12] make a strong statement in this regard by referring in a considerable number of its sections to the need for preventing pollution from ships navigating the Arctic Regions.

The *LTE Guide* references the optional classification notation, POT, Protection of Fuel and Lubricating Oil Tanks which is in 4-6-4/17 of the ABS *Steel Vessel Rules* [3]. These requirements provide additional protection to these tanks in the event of vessel collision or grounding affecting tanks in the after area.

## 11. CREW CONSIDERATIONS

Working in cold weather environments has significant implications on human capabilities, and unless proper precautions are made, these can be hazardous to a person's health. In recognition thereof the *LTE Guide* also provides:

- Requirements for clothing to protect personnel,
- Basic supplemental information on human performance and health hazards when working in Arctic conditions.

The supplemental information is provided for those owners, designers, or operators who would like information of the sort provided as a reference to consider in the course of ship design, outfitting, and ship operation.

### 11.1 CLOTHING AND PERSONAL PROTECTIVE EQUIPMENT

For appropriate protection/isolation against cold climate conditions, adequate clothing must be provided for personnel. Requirements for the following clothing are listed:

- Hand Protection
- Head and Eye Protection
- Foot Protection
- Immersion Suit

There is also a requirement that the personnel protective equipment be properly maintained and stored.

The *LTE Guide* does not list specific requirements for the clothing types as the temperature conditions will vary depending on the route the vessel is trading in. Guidance for selection of the appropriate equipment is provided in Appendix 8.

### 11.2 ADDITIONAL INFORMATION

Appendix 8, Guidance Notes on Crew Considerations, provides supplemental information addressing the following areas:

- Decreases in cognitive/reasoning ability due to cold exposure
- Health hazards related to cold exposure
- Monitoring environmental conditions
- Clothing and personal protective equipment
- Nutrition considerations in cold climates
- Workstation design and operational considerations
- Accommodations and environmental control

This summary information is provided for the use of designers, owners and operators to be informed of the various issues that need to be satisfactorily addressed to protect personnel from unfavourable environmental conditions and to minimize risk of injury.

Several tables providing information for topics such as: suggested maximal allowable work times, wind chill and exposure danger, or protective and functional properties for outdoor work garments are listed. A representative example is Table 7 presenting the symptoms of hypothermia.

Table 7: Relationship between Wind Chill and Exposure Danger

**WIND CHILL CHART**

| | | Ambient Temperature (°C) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 4 | -1 | -7 | -12 | -18 | -23 | -29 | -34 | -40 |
| Wind km/h | Velocity mph | Equivalent Chill Temperature (°C) | | | | | | | | |
| Calm 0 | 0 | 4 | -1 | -7 | -12 | -18 | -23 | -29 | -34 | -40 |
| 8 | 5 | 3 | -3 | -9 | -14 | -21 | -26 | -32 | -38 | -44 |
| 16 | 10 | -2 | -9 | -16 | -23 | -30 | -35 | -43 | -50 | -57 |
| 24 | 15 | -6 | -13 | -20 | -28 | -36 | -43 | -50 | -58 | -65 |
| 32 | 20 | -8 | -16 | -23 | -32 | -39 | -47 | -55 | -63 | -71 |
| 40 | 25 | -9 | -18 | -26 | -34 | -42 | -51 | -59 | -67 | -76 |
| 48 | 30 | -16 | -19 | -22 | -36 | -44 | -53 | -62 | -70 | -78 |
| 56 | 35 | -11 | -20 | -29 | -37 | -46 | -55 | -63 | -72 | -81 |
| 64 | 40 | -12 | -21 | -29 | -38 | -47 | -56 | -65 | -73 | -82 |

**Little danger** in less than one hour exposure of dry skin

**DANGER** – Exposed flesh freezes within one minute

**GREAT DANGER** – Flesh may freeze within 30 seconds

Source: Threshold Limit Values (TLV™) and Biological Exposure Indeces (BEI™) booklet; published by ACGIH, Cincinnati, Ohio

## 12. TRAINING AND RELATED DOCUMENTATION

Vessels operating in low temperature environments are exposed to a number of unique circumstances. Among them, weather conditions are poor, navigation charts are unreliable, local ice conditions may differ significantly from those depicted on charts, and route planning is difficult. Therefore, specialized crew training must be undertaken and appropriate operations manuals developed.

The *LTE Guide* requires training in ice operations, navigation and winterization be provided. This training is also to address means to prevent and treat potential cold weather related injuries. An operating and training manual must be developed and submitted to the ABS Technical Office for review. This manual is to include instructions for vessel operations and personnel training.

## 13. SUPPLEMENTARY INFORMATION

Summary information has been provided in the *LTE Guide* appendices for designers and vessel operators unfamiliar with vessels operating in low temperatures. Information on weather conditions of interest is listed. An addendum will be released in early 2007 providing isotherms for regions north of $60^o$ latitude for the period 1981 through 2005 in conformance with the definition of DST. Guidance is provided for vessel operations related to deck access, vessel machinery systems and safety systems for personnel. There are numerous administrations in the Arctic and Antarctic regions for which the *LTE Guide* lists name, address and web site information, where available. As stated earlier, many administrations have additional requirements for vessels operating in their waters. A similar listing for meteorological organizations is also provided.

## 14. CONCLUSIONS

The development of new oil and gas fields in inhospitable regions of the Earth brings extraordinary challenges to the industry. The Arctic Regions are likely the next target of the petroleum industry to supply North America, Europe and the Far East.

Classification societies are teaming with industry to ensure that safety and environmental protection in maritime transportation is maintained even in extremely harsh conditions. Classification Societies have been participating actively in Joint Industry Projects and Joint Development Projects with designers, ship owners, operators, energy majors, shipyards and Regulatory Bodies to provide practical solutions to the challenges of ice navigation. New Rules and Guides are being published by individual Classification Societies and IACS to provide the industry with the basis for the construction of Ice Class and Polar Class vessels that can operate in Arctic waters safely while respecting the pristine natural environment.

The *LTE Guide* and soon to be published Polar Ice Class Rules provide vessel operators the requirements to successfully operate in the Arctic and other cold regions along with additional requirements for personnel to work in these harsh environments.

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## 16. AUTHORS' BIOGRAPHIES

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*Design and Construction of Vessels Operating in Low Temperature Environments, London, UK.*

# CLASSIFICATION OF POLAR SHIPS

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## SUMMARY

A lengthy process to develop harmonized requirements for Polar Class ships culminated in the approval by IACS Council in August 2006 of the new Unified Requirements (URs) I1, I2 and I3 for class descriptions, structures, and machinery respectively. All IACS members will incorporate these URs within their rule systems by March 2008. However, a number of classification societies may retain their existing ice classes, for reasons including compatibility with national regulations. There is also deliberate overlap between the Polar URs and the traditional Baltic higher ice classes. Designers, shipbuilders and ship owners will therefore need to understand how the new URs and these other systems correspond.

This paper describes the background to the UR polar classes, and how the URs are intended to complement the IMO Guidelines for Ships Operating in Arctic Ice-Covered Waters. It also discusses how the design principles and the resulting requirements of the URs compare to those of other rule systems, including the Canadian and Russian national regulations. Finally, it addresses areas in which there are significant uncertainties in ice loads, and where future research is needed to enhance the state of the art.

## NOMENCLATURE

| | |
|---|---|
| ABS | American Bureau of Shipping |
| DE | Design and Equipment |
| DNV | Det Norske Veritas |
| GL | Germanischer Lloyd |
| IACS | International Association of Classification Societies |
| IMO | International Maritime Organization |
| LR | Lloyd's Register |
| UR | Unified Requirements |
| | |
| $f_a$ | hull shape coefficient |
| $\sigma_y$ | yield stress [N/mm$^2$] |
| a | main frame span [m] |
| $A_1$ | load location factor: |
| | $a_1$ = shear area ratio |
| | $k_w = 1 / (1 + 2 * A_f / A_{mFIT})$ |
| | $A_f$ = flange area of main frame web [cm$^2$] |
| | $k_z = z_p / Z_p$ |
| | $z_p$ = sum of plastic section modulii of flange and shell plate [cm$^3$] |
| | $= w_f * t_f^2 / 4 + b_{eff} * t_{net}^2 / 4$ |
| | $w_r$ = width of flange [cm] |
| | $Z_p$ = plastic section modulus of main frame as fitted [cm$^3$] |
| AF | Hull Area Factor |
| $A_m$ | minimum shear area of main frame |
| b | height of design ice load patch [m] |
| $F_n$ | maximum force |
| LL | length of loaded portion of span = lesser of a and b [m] |
| $M_{ship}$ | ship displacement |
| $P_{avg}$ | average pressure within load patch |
| $P_O$ | ice strength |
| $PPF_m$ | Peak Pressure Factor |
| s | main frame spacing [m] |
| $V_{ship}$ | ship velocity |
| Y | 1 – 0.5 (LL / a) |
| $Z_{pm}$ | minimum plastic section modulus of the plate/stiffener combination |

## 1. INTRODUCTION

In August 2006 the Council of the International Association of Classification Societies (IACS) approved a new set of Unified Requirements for Polar Class ships (available under www.iacs.org.uk):

I1 – Polar Class Descriptions
I2 – Structural Requirements
I3 – Machinery Requirements

As noted in these new requirements, all classification societies are required to incorporate them within their rule systems by March 2008. However, societies are free to retain their existing ice class rules, and to supplement the URs with other provisions. All the societies are also expected to retain their Baltic Classes, which are an essential component of the winter navigation system developed and maintained by the Finnish and Swedish authorities.

No ice class rules are intended to be applied blindly. Selection of class can rarely be matched with any real precision merely to the route a ship is intended to follow – other operational and environmental factors also have to be considered.

Ship owners, shipbuilders and designers therefore need to have an understanding of the principles underpinning the URs, and other ice class rule systems before specifying and applying these to new construction projects, or to the redeployment of existing vessels on new services. This paper is intended to provide an introduction to this complex subject. It focuses mainly on I1 and I2, with limited discussion of the machinery requirements. These deserve their own paper.

## 2. BACKGROUND TO THE POLAR RULES

### 2.1 POLAR SHIPPING

The 1980s were the last 'golden age' for polar shipping. Offshore developments in the Canadian and US Arctic generated a number of new and innovative designs, both for the smaller vessels that were actually built; and also for larger transports for oil, gas, and other cargoes, which were not. Meanwhile, transportation along the Northern Sea Route of the Soviet Union reached its peak levels, supported by a powerful icebreaker fleet. Much research and development was undertaken at this time by governments and by private industry, including the development of large ice testing facilities in several countries.

By the early 1990s, activity levels had collapsed around the Arctic, due to a variety of political and economic shocks. However, there was a recognition that much knowledge had been gained, and many gaps in knowledge had been revealed by recent activities. This spurred a number of national administrations (notably Canada and Russia) to revisit their regulatory regime for Arctic shipping, and several classification societies to promulgate their own new Ice, Arctic or Polar Class rules. Unfortunately, this resulted in a plethora of completely incompatible approaches.

In the early 1990s, several proposals were made to the International Maritime Organization (IMO) for the development of a harmonized approach to polar ship classification, and a working group under the Design and Equipment (DE) subcommittee was formed to study the issue. It was rapidly agreed that no existing rules could be adopted as the basis for an international system, and that a holistic approach to design, equipment, and operation was required. In 1996, the way forward was formalized. IMO's working group was to develop a 'Polar Code' that would provide the overall framework for the initiative. A new IACS ad-hoc group would take responsibility for formulating unified requirements for structural and machinery design. The two groups would hold joint meetings to coordinate strategy, and separate sessions to address specific issues. Both groups would include representatives from government, class, industry and academia in order to benefit from a full spectrum of expertise.

### 2.2 IMO APPROACH

The IMO working group aimed to produce a set of guiding principles for the safe design and operation of all ships in ice-covered polar waters; and to offer specific guidance only where other organizations and/or standards were considered unable to do this.

It was rapidly agreed that national systems of navigation control would not be supplanted by the IMO initiative, though it was anticipated that the challenges of such control would be simplified. As an example, the Canadian administration has been faced with the need to set safe access and operating limits for a variety of vessels designed against different ice class systems (see below). This has led to delay, cost, and in some cases controversy. A move to a single set of polar classes was expected to simplify future approvals, and facilitate delegation to class of certain responsibilities.

It was originally envisaged that IMO would develop a "Polar Code" covering both Arctic and Antarctic waters. Some jurisdictional concerns over the Antarctic Treaty led to the initial scope being restricted to the Arctic (though more recently there have been moves to re-introduce waters south of $60^0$S). As there is no common international definition of Arctic waters, another early challenge in the process was to develop a map and supporting text. The map is shown in Figure 1. In many ways, the notional areas of applicability are unimportant, as the principles are equally relevant to any sea areas where ships may encounter ice.

The final IMO documents were published in December 2002, as MSC Circular 1056/MEPC Circular 399, "Guidelines for Ships Operating in Arctic Ice Covered Waters". They contain provisions for design, equipment, operation, and environmental protection. The Structures and Main Machinery sections are quite brief, and reference the IACS Unified Requirements as means of demonstrating compliance with the design principles therein.


Figure 1: Arctic Waters Covered by IMO Guidelines

### 2.3 IACS WORKING GROUP

The IACS work was split between Structural and Machinery Working Groups, each consisting of up to 20 experts depending on the stage of the development process. The bulk of the work – as described below - was completed by 2000, at which time a draft set of proposed requirements and a number of background papers were circulated to a broader stakeholder group and made available through the internet for review and comment.

Finalization of the proposals was somewhat protracted. A number of concerns were raised with the drafts. A serious concern related to how Polar Class and Baltic class were aligned, and how equivalencies between them could be assigned. This latter subject led to a series of meetings involving the Finnish and Swedish administrations; the development of an agreement on equivalency procedures, and a further 'fine tuning' of the proposed polar classes. Once all of this work was completed, the draft URs were submitted to the IACS General Policy Group and then Council for final approval, which was received in August 2006.

## 3. THE URS AND OTHER ICE CLASS RULES

### 3.1 OVERVIEW

The major thrust for the development of the URs arose from the fact that, prior to the harmonization efforts, a fairly large number of completely different sets of polar ice classes had been developed by national administrations and by various classification societies. From an owner or builder's point of view, this made it challenging to decide how to select an appropriate ice class. For a regulator, the challenge was to determine when and how ships of different class and capability could be permitted to operate on Northern routes.

The 'Baltic' rules developed by the Finnish and Swedish administrations for their own specific requirements tended to be used as a 'default' for many ships operating in lighter summer polar conditions. The Russian Register, initially with Soviet and subsequently with the Russian Federation authorities developed requirements for the Northern Sea Route and other ice-infested waters. Canada produced its own structural design standards in the early 1970s, and revised these substantively in the early 1990s based on much research during the early phase of development in the Beaufort Sea. Several of the leading classification societies produced their own ice class rules in the same timeframe, with GL basing their approach on the later Canadian system and ABS following some elements of the Russian approach. DNV and LR's polar ice classes provided additional rule systems.

In and of itself, this proliferation of rule systems would not necessarily be any more problematic than the similar situation for open water ships, where each classification society maintained its own rules and class notations. However, whereas existing IACS UR's harmonized many of the important aspects for open water design, in the case of ice classes there were fairly fundamental differences in both approach and outcome between many of the systems. This made it impossible to generate a simple system of rule equivalencies, and instead required the development of a whole new rule system.

### 3.2 DIFFERENCES IN APPROACH

A few examples of major differences between the rule systems are provided in Table 1 below. These are not exhaustive or detailed, but serve to illustrate the challenges involved in developing the consensus required under the UR process.

In addition, the basis on which previous rule systems had been developed was sometimes unclear, even to representatives of the organizations responsible for them. This was recognized as a potential impediment to future improvements in any new rule system, and so it was agreed early in the process that the basis for the Unified Requirements should be fully documented, and that the documentation should be provided in the public domain.

Table 1: Qualitative Rule Comparisons

| *Issue* | *Canadian* | | *Russian* | | *ABS* | *DNV* | *GL* | *LR* |
|---|---|---|---|---|---|---|---|---|
| | ASPPR | CAC | Old | New | | | | |
| No. of classes | 9 | 4 | 3+4 icebreaker | 6 | 5 (8 if escort available) | 6+3 icebreaker | 4 | 4 |
| Displacement dependency | Strong | Moderate | Strong | Strong | Strong | None | None | Moderate |
| Power dependency | None | None | Weak | None | Weak | None | None | Moderate |
| Structural design basis | Elastic | Elasto-plastic | Elastic | Elasto-plastic | Elasto-plastic | Elastic | Elasto-plastic | Elastic |

## 4. DEVELOPMENT OF POLAR RULES

### 4.1 DEVELOPMENT PRINCIPLES

The approach to the development of the new IACS Unified Requirements (hereafter referred to as the Polar Rules for convenience) was highly unusual. This was partly because they were intended to be a complement to the IMO Guidelines, and were developed in parallel with these. It was also due to the recognition that much of the expertise relevant to their development would come from outside the classification societies themselves. The working groups under IACS therefore had considerable external representation, and the work was extensively documented and debated. Working papers and supporting data extend to many metres of shelf space and gigabytes of electronic information.

It was agreed early in the overall program that no new research work would be commissioned to support the development of the rules. All of the organizations and administrations involved in the initiative provided access to their own previous R&D efforts and service experience; and in the event a number of detailed studies were conducted by various stakeholders in order to test the validity of some of the proposals. At the end of the process, it was acknowledged that there are still a number of significant unknowns in the prediction of ice loads and in the development of appropriate means for designing against them. Some of these are discussed later in the paper. However, it was also acknowledged that the current version of the Polar Rules represents a significant advance from past practice.

A key element of the overall development was to agree on the upper and lower capability bounds for polar ships, and to decide on the number of polar classes that would be appropriate. The polar classes had to be common to the IMO and IACS work, as they would be referenced in both sets of requirements [Kendrick & Santos-Pedro, 1999]. The high end capability was relatively easy to define. This PC 1 ship was to be capable of operating safely anywhere in the Arctic or Antarctic oceans at any time of year (though safe operation would still require due caution). At the low end, it was acknowledged that much current Arctic and Antarctic summer traffic is carried out by vessels with Baltic ice classes, but also that a number of experienced operators have added features over and above the notional Baltic class requirements. Therefore, the lower threshold was set at a capability level similar to Baltic IA. Between the upper and lower bounds, changes in capability (and therefore cost) should be at manageable increments.

Eventually, seven Polar Classes were adopted, as listed in Table 2. This table can be found both in the IMO Guidelines and in UR 1.2. As can be seen, the ice capability descriptions included are rather cursory. This was deliberate, as the wide variety of ways in which ships can be operated in polar waters precludes being overly precise when defining basic classes.

Table 2: IMO/IAC Polar Classes

| *Polar Class* | *Ice Description (based on WMO Sea Ice Nomenclature)* |
|---|---|
| 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 which may include multi-year ice inclusions. |
| PC 4 | Year-round operation in thick first-year ice which may include old ice inclusions |
| PC 5 | Year-round operation in medium first-year ice which may include old ice inclusions |
| PC 6 | Summer/autumn operation in medium first-year ice which may include old ice inclusions |
| PC 7 | Summer/autumn operation in thin first-year ice which may include old ice inclusions |

### 4.2 ICE LOADS

Ships interact with ice in various ways, each of which will produce some level of loading on the hull. During the development of the URs, over 70 ice interaction scenarios were defined. Many of these were considered to be unlikely, or avoidable. Ramming an iceberg, for example, is a hazard in polar operation but should not necessarily be treated as a design case.

Ice-capable ship performance usually refers to the level ice thickness that can be broken continuously. The ice loads from this type of icebreaking are typically not the worst case loads for the ship. These are rather represented by impacts against heavy ice features, which may either be deliberate or unavoidable. Ships often have to ram ridges or thicker ice, and the impact velocities will typically be higher than the level icebreaking speed in the same conditions.

An impact load model was therefore developed as the basis for the UR structural requirements. This builds on the classical work of the Russian scientist [Popov], extended by [Daley] to provide a more complete and general solution. The ship penetrates into the ice by crushing (Figure 2), and the maximum force is a function of ship (and ice) shape, velocity, and ice crushing strength, as shown in Equation 1:

$$F_n = fa \cdot Po^{0.36} \cdot V_{ship}^{\ 1.28} \cdot M_{ship}^{\ 0.64} \qquad (1)$$

Where *fa* captures the shape terms (full derivation can be found in Daley), *V* and *M* the ship velocity and displacement, and *Po* the ice strength.

For larger ships, the total force may be limited by breaking the ice in bending. Each polar class is therefore defined by ice thickness and crushing and flexural strength, and by an assumed ship speed. All of these parameters are combined into a set of class factors for the PCs 1 to 7. Hull (bow) shape and ship displacement are specific to the ship under consideration.

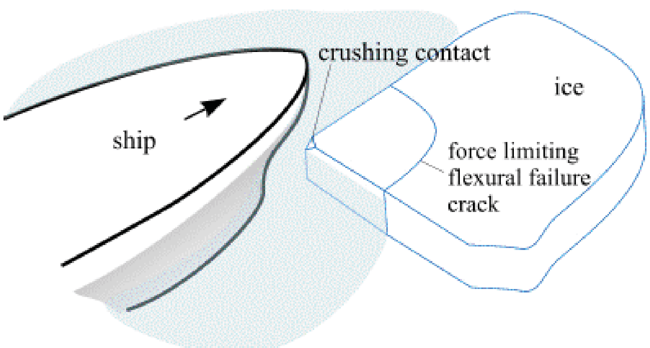


Figure 2: Design Ice Impact Scenario

Once the maximum force has been defined, it is applied to the structure through a load patch. The patch simplifies the indentation geometry, and accounts for the non-uniform distribution of pressure over the contact area; which is a key factor in ice-capable structural design. Over small areas, ice pressures can be extremely high (up to 50 MPa). However, at a structural design scale of 0.1 $m^2$ (e.g. a 0.3 x 0.35 m plate panel) or larger the average pressures are much lower; and by a 1 $m^2$ scale average pressures on even larger, high ice class ships are in the 5 MPa range. The UR ice load model provides an average pressure value for the load patch, and uses peak pressure (intensification) factors to adjust this average in the design of smaller structural components.

Loads are derived directly only for the bow region, which is the area of the hull that has been subjected to most extensive experimentation over the years. For other areas of the hull, as shown in Figure 3, reduced loads are defined using a set of hull area factors, . Area factors are common to most previous ice class systems, and the URs use those values that appear to have been appropriate, based on actual service experience. As an example, a class of vessels such as the SA-15 cargo ship or the 'Terry Fox' class icebreaker may have more - or less - damage in the bow than in the midbody. The former case would indicate that the midbody was relatively over designed; the latter that it was under designed.

A large database of ice-going vessels was assembled and assessed in this manner in order to select realistic area factors. The other major use of this ship database was to select the overall class factors that define each polar class. Existing ship structures were analyzed against the requirements for PC 1-7, and the known structural performance of the ship was assessed against the broad operational descriptions in Table 2. It was expected (and desired) that a ship such as a Russian nuclear icebreaker would turn out to comply largely with a PC 1 classification, and that some of the Baltic and 'Baltic plus' bulk carriers with successful Arctic service experience would meet PC 6 and 7 structural requirements. The bounding class factors were calibrated in this way (see also below). As outlined earlier, the intermediate ice classes/class factors were set at intervals that provide relatively consistent increments in capability.

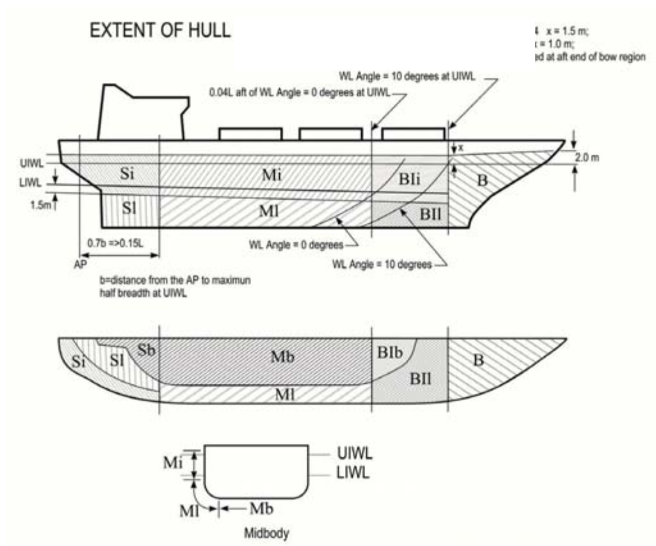


Figure 3: Hull Area Extents

## 4.3 STRUCTURAL RESPONSE

The URs approach structural design requirements with a similar philosophy to that utilized in the more Russian and Canadian ice class standards. Peak ice loads are expected to be relatively infrequent events for any given structural component. Therefore, rather than using first yield as a design point (as in most other ship design standards) the URs for plating and framing are based on the formation of elasto-plastic response mechanisms; i.e. the creation of systems of plastic hinges.

The plate response formulae look reasonably familiar to many structural engineers:

$$t = 0.5s\sqrt{\frac{p}{FY}} \cdot \frac{1}{1 + 0.5\dfrac{s}{b}} \qquad (2)$$

Equation (2) defines the onset of the system of hinges shown in Figure 4, which can vary depending on plate aspect ratio and on whether the plate is fully loaded or strip loaded, as shown.

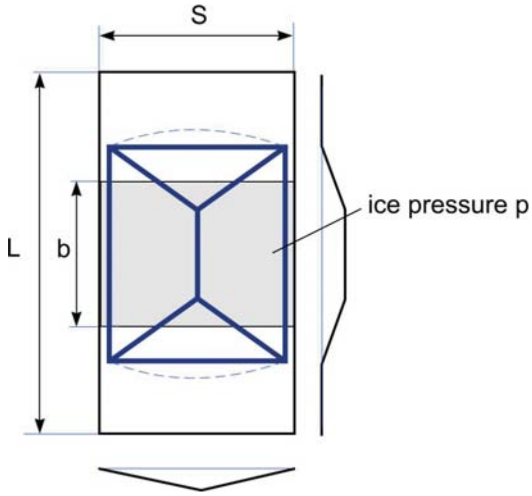


Figure 4: Plate Response

The frame design equations are more unusual. They reflect a system of hinges some of which combine bending and shear effects, which interact. The distribution of shear stress across a section will reduce its effective section modulus in bending, and bending stresses affect shear capacity. There is thus not a single design point for an ice frame, but a design domain in which various combinations of cross section shear area and section modulus are possible and can be selected on the basis of availability, configuration, and produceability. The system of equations (3) found in the URs (as equations 22/23 and 24/25 in UR I2) has to be solved iteratively, which can be done quite easily for example in an Excel™ macro.

Minimum shear area, $A_m = 100^2 * 0.5 * LL * s * (AF * PPF_m * P_{avg}) / (0.577 * \sigma_y)$ [cm²]

(3)

Minimum section modulus, $Z_{pm} = 100^3 * LL * Y * s* (AF * PPF_t * P_{avg})* a * A_1/ (4*\sigma_y)$ [cm³]

As the overall structural design approach was quite innovative, a considerable amount of work was done to ensure that no undesirable safety, serviceability or durability issues were likely to be encountered. For example, a range of finite element models of ice plate/frame systems were analyzed, with typical results as shown in Figure 5. In this figure, point A represents first yield in the frame, a typical structural design point (which should be predicted more or less identically by analytical and numerical methods). Point B represents the UR design point. The load carried is approximately double that at first yield, but as can be seen the peak deflection (a serviceability issue) is still very small. The residual deformation after unloading is immeasurably small, and well within normal fabrication tolerances.

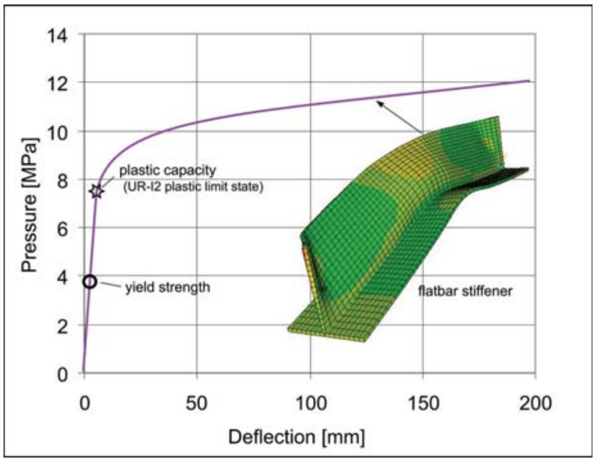


Figure 5: Frame Response

Equally importantly, there is still a substantial strength reserve above the design point. This is actually underestimated by the FE plot shown above, which uses relatively conservative geometric and material assumptions. Recent physical modelling work (Figure 6) has shown much larger strength reserve. Premature instability is prevented under the URs by a number of simple geometric constraints.

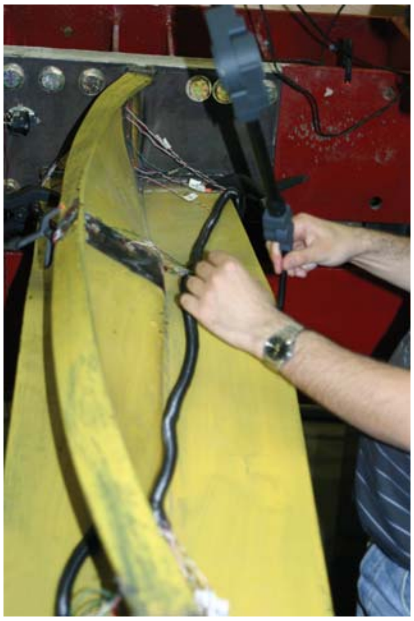

Figure 6: Ice Class Frame Tests

Fatigue was considered, but was not felt to be a serious issue. As noted above, loads at or approaching the design point are relatively rare events for any structural component, as ice loads (unlike wave-inducted loads) are quite localized and tend to have a random distribution. Service experience also indicated that fatigue has not been a problem for the hull. This is absolutely not the case for propeller design, where ice-induced fatigue can dominate. High ice milling loads at shaft rate frequencies form one of the UR propeller design criteria.

### 4.4 OTHER ISSUES

A notable deficiency of the URs is that they provide very limited guidance for the use of structure above panel scale, or for the design of structures such as double hulls with plate web frames and stringers. Rigorous analytical formulations for these elements are very difficult to develop, and are always likely to misrepresent real configurations with access openings, etc. Many designers are therefore likely to have to use FE modelling to develop efficient solutions. This can require a fairly sophisticated understanding of modelling approaches.

While FE models may be very effective in the design of larger components, they are not to be used to reduce the plate and frame scantlings derived from equations (1) and (2). This is partly because the application of the ice loads at a local level becomes important, and can easily be misrepresented. As noted earlier, the results from the combination of the UR load and structural response models have been calibrated jointly against actual service experience.

## 5. AREAS FOR FUTURE DEVELOPMENT

The Unified Requirements represent a significant advance for ice-capable ship design. They combine a modern understanding of ice loads with a state-of-the-art set of structural analysis approaches that provide for efficient and robust solutions. However, the URs are a long way from perfection.

Class factors and hull area factors have been calibrated against a very limited database of operating experience. Most of the vessels that have been operating in polar waters have been small. Almost 40 years on, the "Manhattan" is still by far the largest vessel to have encountered heavy ice conditions. A new generation of relatively large, higher ice class ships is now being designed and constructed, and there is a need to collect data on all aspects of their service experience to validate (or revise) the UR models.

Although steel thicknesses may be greatest in the bow, most of the steel weight, cost, and deadweight penalties are incurred in other ice-strengthening areas. Accordingly, there is a need to develop a better approach to the design of most hull areas than reliance on area factors. This will require both scenario modelling and data collection, using dedicated trials and long-term monitoring. Enlightened owners should be encouraged to provide access to their vessels, to help define their real safe operating limits and also to benefit future projects.

As noted earlier, there is a need to extend the scope of the URs to cover more structural components including decks, bulkheads, and double hulls. Individual classification societies are facing this need now, and are developing a range of approaches to the problem. Ideally, IACS should continue to coordinate these efforts, to avoid reverting to a multitude of approaches from the present uniform requirements.

## 6. SUMMARY

The new IACS Requirements concerning Polar Class consolidates the state-of-the-art knowledge on ice loads and structural responses to these. They were developed under a unique process that assembled an international group of experts and stakeholders under the IACS organization, and under an IMO umbrella. This process was lengthy, but highly effective in ensuring that the final outcomes were tested rigorously against theory and experience.

The URs themselves provide a good basis for the design of the next generation of ice-class ships. As with previous ice class rule systems, they should always be treated as a starting point for ship design, rather than a 'cookbook' solution to all aspects of the design.

A new generation of vessels is now being designed against the URs, within the 'goal-based' framework of the IMO Guidelines. Experience gained from these projects will be used to develop further refinements to the new international polar ship class system.

## 7. ACKNOWLEDGEMENTS

The authors would like to acknowledge the contribution of many others to the development of the Unified Requirements. In particular, Evgeniy Appolonov of the Krylov Institute added much clear thinking and mathematical rigour to many aspects of the work. Lefteris Karaminas, while at Lloyds Register, managed the initial stages of the overall work program with a combination of drive and diplomacy.

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## 9. AUTHORS’ BIOGRAPHIES

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**Richard Hayward** is a structural engineer with Germanischer Lloyd and has been engaged in rule development activities during the last 12 years. He was involved throughout the development of the IMO Guidelines and chaired the working group which developed the IACS Unified Requirements.

# MODERN RS REQUIREMENTS AND METHODS ENSURING OPERATING STRENGTH OF ICEBREAKING PROPULSION COMPLEX

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## SUMMARY

Ensuring the operating strength of propulsion complexes of ice-going vessels and icebreakers is one of the main problems facing the Arctic shipping and icebreaker shipbuilding. The existing requirements for the strength of the propeller and other propulsion complex components do not meet present-day operating requirements. The existing requirements need to be upgraded and new ones developed.

This paper sets forth the main results of research dedicated to the development of new requirements of Russian Maritime Register of Shipping (RS) for propellers aimed at ensuring both fatigue and static strength. The requirements cover double-acting icebreaking vessels.

The main outcome of development of the requirements for propulsion complex pyramidal strength has been stated. A new method has been put forth for determining propeller blade damage load. A method has been developed for ensuring the strength of the propulsion complex components at elastoplastic deformation in stress concentration areas.

## NOMENCLATURE

| | |
|---|---|
| RS | Russian Maritime Register of Shipping |
| ICB | Icebreaker |
| IGV | Ice going vessel |
| PC | Propulsion complex |
| AZT | Azimuth Thruster |
| IP | Icebreaking propeller |
| CPP | Controllable pitch propeller |
| CPM | Controllable pitch mechanism |
| NSR | Northern Sea Route |
| FEM | Finite element method |
| $D$ | Propeller diameter (m) |
| $\alpha(r)$ | Attack angle of blade section at radius $r$ (deg) |
| $\varphi(r)$ | Pitch angle at radius $r$ (deg) |
| $\bar{r} = r/R$ | Relative radius of blade section |
| $r$ | Blade section radius |
| $R$ | Propeller radius (m) |
| $\bar{r}_{hub}$ | Relative radius of hub or hub rigidly fixed root section |
| $c(\bar{r})$ | Chord length of cylindrical blade section at $\bar{r}$ (m) |
| $F_{ice}^{-}$ | Negative ice force on blade, opposite to hydrodynamic thrust (N) |
| $Q_{bend}$ | Blade bending moment caused by $F_{ice}^{-}$ (N m) |
| $Q_{spind}$ | Blade spindle torque caused by $F_{ice}^{-}$ (N m) |
| $(F_{ice})_{\max}$ | Extreme possible negative ice force, which is the upper limit of the $F_{ice}^{-}$ range (N m) |
| $Q_{bend}^{\max}$ | Blade bending moment caused by $(F_{ice})_{\max}$ (N m) |
| $Q_{spind}^{\max}$ | Blade spindle torque caused by $(F_{ice})_{\max}$ (N m) |
| $p_{ice}$ | Ice pressure in blade-ice contact area for ice milling regime (m Pa) |
| n | propeller revolution ( $s^{-1}$ ) |
| $\sigma_{yield}$ | yield stress |
| $\sigma_{tensile}$ | ultimate tensile strength |
| $\delta$ | relative elongation |
| $\delta_p$ | uniform relative elongation |
| $KV$ | impact energy for Charpy V-notshed test |

$(\sigma_{yield})_{\min}$, $(\sigma_{tensile})_{\min}$, $\delta_{\min}$, $KV_{\min}$ minimum value of $\sigma_{yield}$, $\sigma_{tensile}$, $\delta$, $KV$, guaranteed by delivery specification

| | |
|---|---|
| $\sigma_{perm}$ | permissible stress (Pa) |

## 1. INTRODUCTION

Ensuring the strength of propulsion complexes of ice-going vessels (IGV) and icebreakers (ICB) is one of the main problems facing the Arctic shipping and icebreaker shipbuilding. This is due to the heavy ice loads acting on their components during operation in ice.

Icebreaking propeller (IP) is one of the main components of the propulsion complex (PC). Ice loads on IP blades and their scantlings determine the operating strength of all the other PC components.

IP blade scantlings are to be assigned such as to ensure both fatigue and static strength. Scantlings of other PC components in lines of force (CPP components, propeller shaft, thrust bearings, etc.) are based on pyramidal and fatigue strength. The current requirements of classification societies to IPs are based on static strength. The reinforcements needed to withstand ice loads are taken into account as empirical coefficients developed on the basis of prior operating experience. The requirements for pyramidal strength in classification societies' rules are of general nature and do not provide for the rating of ultimate blade damage load (with the exception of DNV Rules). Operating experience has shown that the ultimate (pyramidal) strength criteria are also inadequate. These methods are not able to meet present-day design requirements. There is a need for developing new methods and requirements for ensuring the static and fatigue strength of IPs and other PC components under the action of ice loads. As a matter of priority, these requirements should be developed for the IP as the main PC component. In addition to that, PC pyramidal strength needs to be ensured.

## 2. PROPELLER AND PROPULSION COMPLEX OPERATING EXPERIENCE ANALYSIS

Development of modern requirements for icebreaker PCs would not be possible without taking account of the prior operating experience. The papers [1,2] are dedicated to the analysis of damages to IP and other PC components. The results and conclusions are set out below**:**

- for steel IPs, blade failures are caused by the loss of static and fatigue strength, the fatigue being the determining factor. Generally, a fatigue crack develops on the suction side of the blade root section in way of leading edge. Fatigue-related failures are not characteristic for IPs made of modern propeller bronzes**;**

- IP blade scantlings are generally to be assigned such as to ensure both fatigue and static strength;

- the minimum lifetime of blades should correspond to the ship's lifetime (20-25 calendar years) with 0.999 reliability**;**

- to ensure reliable operation of PC, consideration should be given to the possibility of occurrence of non-design IP-ice interaction modes where the ice loads are so great that it is not possible to ensure blade strength. Under these circumstances, the pyramidal strength principle should be employed to ensure the strength of PC components. For Arctic ICBs, the average blade damage frequency (average blade lifetime) of side IP blades at non-design modes is about 9 calendar years. The probability of a center IP being damaged is small. For Arctic IGVs on the NCR, the above value is equal to approximately 40 calendar years**;**

- for IGVs and ICBs operating in non-design modes, the blade damage frequency for side IPs is much greater than that of center IPs. In this regard, the risk associated with operating a twin-shaft PC may significantly exceed that of single-shaft PC. The latter consideration needs to be taken into account in designing icebreaker PCs.

For CPPs and AZTs, the main causes of failures in ice are underestimation of ice loads and the loads for ensuring pyramidal strength; errors in strength calculations; inadequate workmanship of CP components [1,2].

## 3. ICE LOADS FOR CALCULATING SCANTLINGS OF PROPELLER BLADES AND OTHER PROPULSION COMPLEX COMPONENTS

### 3.1 MAIN WAYS OF DETERMINING ICE LOADS FOR CALCULATING FATIGUE AND STATIC STRENGTH

Ice load parameters were determined on the basis of a comprehensive analysis of full-scale tests, model tests and theoretical research. Ice milling regime is taken as the main design mode for their determination, where the blade profile angle of attack $\alpha(r) \geq 0$ (see Fig. 1). In reversing mode, where $\alpha(r) < 0$, a direct ("flat") impact of ice fragment against blade surface can occur. When a ship is advancing with its IPs stopped, ice press on blade can occur. The above modes are treated as non-design modes. The ice force acting on the blade is so great as to make preservation of the blade impossible. In cases like these, pyramidal strength needs to be ensured.

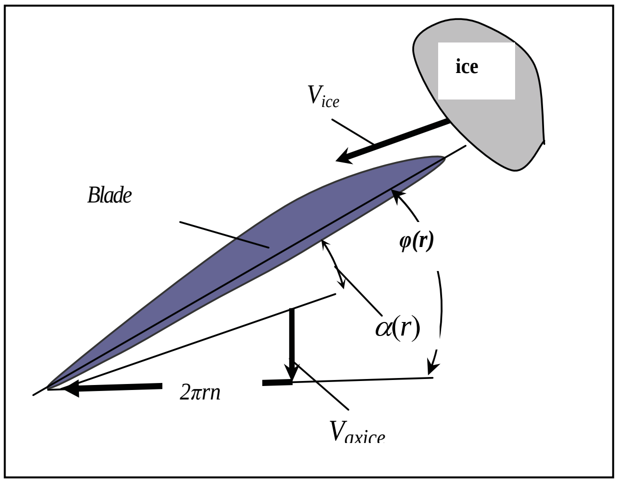


Figure 1: General scheme of ice blade section interaction for ice milling conditions at $r.V_{axice}$ – ice axial speed; $V_{ice}$ – ice speed.

In design milling regimes, operating ice forces acting on blades may be both positive and negative. The negative forces are directed opposite to the hydrodynamic thrust and they are applied to the IP blade leading edge. The positive forces are mainly caused by the interaction of blade peripheral sections with ice and they are applied to the centre [3]. The negative edge ice force $F_{ice}^{-}$ is a governing consideration for assigning IP blade scantlings, since it causes the bending moment $Q_{bend}$ and spindle torque $Q_{spind}$ to act on the blade. The value of $Q_{spind}$ is commensurable with that of $Q_{bend}$. The spindle torque $Q_{spind}$ results in restricted twisting of root sections and causes the occurrence of additional shear and normal warping stresses (normal restricted twisting stresses) [4]. The latter can exceed normal bending stresses. In this case, the maximum tensile stress point is on the suction side of the root section near the leading edge, which is confirmed by operating experience because it is in this area that fatigue cracks occur. Therefore *when determining blade scantlings, consideration should be given both to $Q_{bend}$ and $Q_{spind}$ caused by $F_{ice}^{-}$*. For milling regimes, the blade angle of attack is one of the main parameters governing the ice loads.

The frequency and level of ice loads acting on IP are random. To enable the assignment of scantlings of IPs and other PC components, their probabilistic and statistical parameters are to be determined (extreme possible value, distribution law, propeller-ice interaction time) based on ship category, propeller position, its main characteristics and propeller-ice interaction parameters. Based on the analysis of the results of full-scale and model tests and theoretical analysis, paper [2] examines the governing features of ice loads. Proposals regarding the assignment of their design values for ensuring strength of IP and other PC components have been developed and formulated (see below).

## 3.2 DESIGN MODEL FOR DETERMINING ICE PRESSURES IN BLADE - ICE CONTACT AREA

The design model for determining ice pressures $p_{ice}$ in blade-ice contact area is set forth in paper [5]. The model is based on the hydrodynamic viscous ice layer (powder) theory and was developed having regard to the research done by V.A. Beliashov, V.S. Shpakov, D. D.Heysin, H. Soininen, B. Veitch and E. M. Appolonov [6, 7, 8, 9, 10, 11]. The scheme demonstrating profile penetration in ice and extrusion of powder as a viscous layer is shown in Fig. 2.

1- ice crushing zone; 2 - ice powder; 3 – splitting element; 4 - line of attack angle; 5 - blade section chord; 6 - split crack; 7 – ice surface; 8 - direction of the ship motion; AB - ice contact zone from the direction of suction side; BC - ice contact zone from the direction of pressure side.

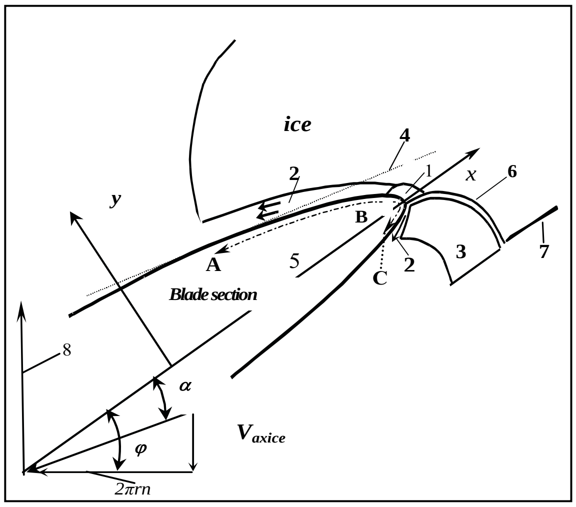


Figure 2: Ice blade section interaction for ice milling conditions.

A method for evaluating the powder layer thickness and form was proposed for determining $p_{ice}$. With this aim in view, the model was supplemented by the continuity condition – the condition of equality of the ice powder extruded and the ice fractured in way of profile tip. The characteristic dimension of the fractured ice area (crushing area 1 in Fig. 2) $l_{ice}$ and its mass were determined on the basis of linear fracture mechanics. The value of contact pressures $p_{ice}$ does not depend on the velocity of blade penetration in ice due to inversely proportional dependence of ice powder viscosity on penetration velocity. The latter was experimentally shown by D. Finn, and the relevant results are set out in paper [11]. The contact pressure is assumed to be constant in the crushing area (see crushing area 1 in Fig. 2). The contact pressure is not constant in the powder extrusion area (see area 2 in Fig. 2). Fig. 3 shows design and experimentally measured values of $p_{ice}$. The latter were obtained by H. Soininen [11]. The accuracy of design $p_{ice}$ is sufficient to ensure blade edge strength.

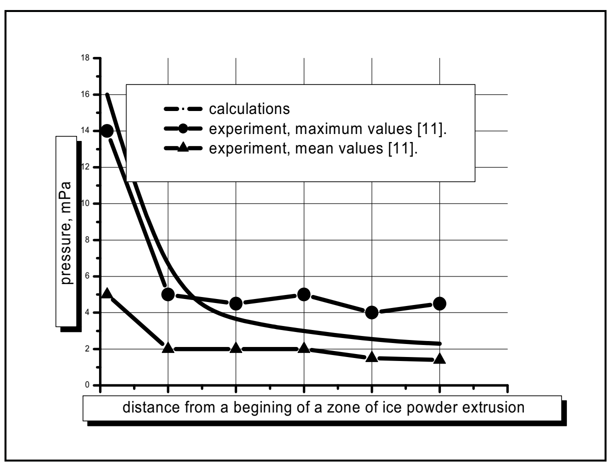


Figure 3: Ice contact pressure $p_{ice}$ for indenter suction side, zone of ice powder extrusion.

## 3.3 FULL-SCALE AND MODEL TEST RESULTS ANALYSIS

Papers [2,12] contain an analysis of full-scale data obtained from strain gauge measurements on IP blades of the icebreaking vessel and side CPP of the arctic icebreaker which were prepared within the framework of development of IACS Unified Requirements. Additionally, an analysis of ice loads on propellers of icebreakers type “Arktika” was carried out [2].

The results of model tests conducted in the ice tank of the Krylov Central Research Institute were used in the paper for examining ice loads on IP [13]. A scheme of model test on propeller in ice milling regime is shown in Fig. 4. The ducted IP - ice interaction process during self-propulsion tests on an Arctic icebreaker model is shown in Fig. 5. Methods were developed for determining operating ice loads on propellers based on the results of model tests (a method for converting ice loads on IP from model scale to full scale, techniques of model tests for evaluating ice loads intensity) [13,2].

Figure 4: Propeller model test for ice milling mode in ice tank of Krylov Shipbuilding Research Institute

Figure 5: Blocking of nozzle by fragments of ice. Self-propulsion test in ice tank of Krylov Shipbuilding Research Institute

As part of work on the above methods, an investigation was carried out into the effect of the following factors on ice loads intensity: ice strength characteristics, velocity of blade penetration in ice, scale effect due to no simultaneous fracture of ice along blade, IP and duct arrangement.

## 3.4 DESIGN MODEL OF ICE LOADS FOR ASSIGNING SCANTLINGS OF PROPELLER BLADES AND OTHER PROPULSION COMPLEX COMPONENTS

Theoretical and experimental research has made it possible to establish the following main features of ice loads on propellers in ice milling regimes**:**

- blade angle of attack $\alpha$ is the main parameter governing the ice loads. Ice loads increase as it decreases due to an increase in the blade-ice contact area. An example in Fig. 6 shows the values of $F_{ice}^{-}$ in relation to $\alpha$ for the side propeller of the arctic icebreaker. The blade angle of attack is determined by the blade pitch angle, IP speed, the axial velocity of its interaction with ice and the ship’s speed. The above parameters need to be taken into account when assigning ice loads. In harsh ice conditions, the minimum angles of attack and the maximum ice loads are realized for the maximum operating speeds in ice. For Arctic ICBs and IGVs, the maximum speeds correspond to the ramming regimes and are equal to around (10-12) and 8 knots respectively. For the above operating conditions, the blade angle of attack $\alpha$ for peripheral radii is not to be less than zero to preclude non-design ice interaction mode and blade failure;

- in ice milling regimes, the ice loads on IP do not depend on its speed (blade profile penetration velocity), provided that the other conditions remain constant;

- ice loads are directly proportional to ice crushing strength and uniaxial compression strength;

- taking into account the scale (reduction) factor due to no simultaneous ice fracture along blade, ice forces and moments on IP for ice milling regimes are proportionate to $S_c^{1.6}$ and $S_c^{2.6}$, respectively, where $S_c$ is the scale;

- ice force on IP blade is directly proportional to the length of its chord;

- the distribution of contact pressures along the profile surface depends on its shape only to a small extent;

- statistical distributions of ice loads, including the greatest values, conform to the third asymptotic law (3FFT) with the distribution function (1). The distinguishing feature of the 3FFT law is that the load range has an upper limit represented by the parameter $x_{\max}$, which corresponds to the maximum possible load.

$$F(x) = \exp\left[\left(\frac{x_{\max} - x}{x_s}\right)^{\beta}\right], \qquad (1)$$

where $x_{\max}$ - the extreme possible ice load value; $x$ - ice load value; $\beta$, $x_s$ - parameters.

For ice loads, the parameter $\beta$ is stable and $\beta \cong 4.3$ .

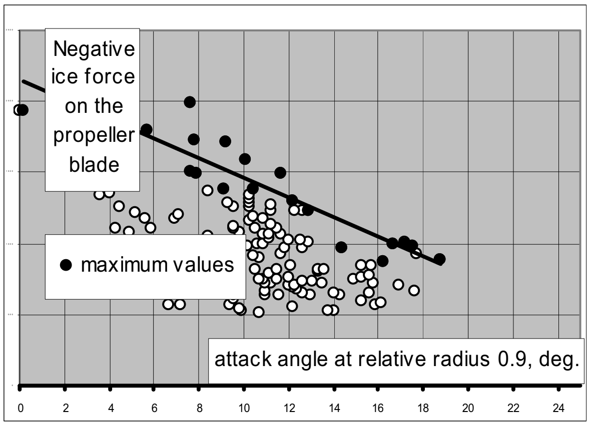


Figure 6: Negative ice force on the propeller blade depending on attack angle. Arctic Icebreaker.

Methods were developed for the evaluation of parameters of the distribution law (1) using bounded distributions [14]. The extreme possible negative ice force $(F_{ice})_{\max}$ (parameter $x_{\max}$), which is the upper limit of the $F_{ice}^{-}$ range, is taken as the main design value for assigning scantlings of IP blades and other PC components. The values of $(F_{ice})_{\max}$, N, are approximated to dependence (2).

$$(F_{ice})_{\max} = 10^3 \cdot \left[22 + 24 \cdot e^{-0.17\alpha(\bar{r}=0.9)}\right] \cdot D^{1.6}\, c_{mean}\, \sigma_{compr}(\bar{r}=0.8), \qquad (2)$$

where $c_{mean} = \int\limits_{\bar{r}=0.6}^{1} c(\bar{r})\,d\bar{r}/(0.4R)$ is the mean dimensionless blade breadth; $\sigma_{compr}(\bar{r}=0.8)$ - ice uniaxial compression strength at the depth corresponding to blade penetration in ice on relative radius $\bar{r}=0.8$, MPa.

Requirements were developed for determining the parameters of the equation (2) for assigning $(F_{ice})_{\max}$. The ice bending moment $Q_{bend}^{\max}$ and spindle torque $Q_{spind}^{\max}$ caused by $(F_{ice})_{\max}$ are determined having regard to blade characteristics. For fatigue strength calculation purposes, data were obtained on the frequency of ice loads acting on IP based on ship's ice category and IP position [2].

## 4. STRENGTH SCANTLINGS OF PROPELLER BLADES

According to the traditional practice of the classification societies the strength scantlings of propeller blades are assigned for the root section and peripheral section on relative radii $\bar{r}=0.6$, $\bar{r}=1$.

### 4.1 STRENGTH SCANTLINGS OF ROOT SECTIONS

The most frequent and typical type of blade damage is breakage of root sections. Usually, blades break at relative radius $\bar{r}_1$, where fillet joins the blade. The said root section is taken for the main design one. For IP blades $\bar{r}_1 \cong \bar{r}_{hub} + 0.05$. Strength scantlings of root sections are assigned on the basis of joint impact of the bending moment $Q_{bend}$ and spindle torque $Q_{spind}$ caused by ice. Methods of evaluation of stressed conditions of blade root sections caused by $Q_{bend}$ and $Q_{spind}$ are set forth in [15, 16]. It is shown that $Q_{spind}$ is the reason for occurrence of additional normal stresses of restricted twisting $\sigma_{dep}$ which maximum is on the suction side and shifted to the leading edge of blade. For icebreaker propellers the value $\sigma_{dep}$ exceeds normal stresses caused by bending $\sigma_{bend}$ which leads to shift of the maximum of total of normal stresses $(\sigma_{bend} + \sigma_{dep})$ and equivalent stresses $\sigma_{\Sigma} = \sqrt{(\sigma_{bend} + \sigma_{dep})^2 + 3\tau^2}$ to the leading edge of blade where $\tau$ is the shear stress. The point of maximum stresses corresponds to the approximate coordinate of $\xi_{0.6}(\bar{r}) = 0.6\,[c(\bar{r})/2]$ see the root section scheme, fig. 7. This point is taken as the main design one. The root section thickness in the design point $t_{0.6}(\bar{r}_1)$, m, is calculated by formula (3)

$$t_{0,6}(\bar{r}_1)=\left[\frac{\sqrt{A(\bar{r}_1)^2+39\left(\dfrac{Q_{spind}^{\max}(\bar{r}_1)}{c(\bar{r}_1)}\right)^2}}{\sigma_{perm}}\right]^{0,5}, \qquad (3)$$

where $A(\bar{r}_1)=\left[\dfrac{Q_{bend}^{\max}(\bar{r}_1)}{0.118\,c(\bar{r}_1)}+24.6\dfrac{Q_{spind}^{\max}(\bar{r}_1)}{D\cdot\Delta(\bar{r}_1)}\right]$;

$\Delta(\bar{r}_1)=\dfrac{Q_{bend}^{\max}(\bar{r}_2)\;c(\bar{r}_1)}{Q_{bend}^{\max}(\bar{r}_1)\;c(\bar{r}_2)}$ ; $Q_{spind}^{\max}(\bar{r}_1)$ - the spindle torque about the centre of coordinates of the expanded blade root section at $\bar{r}_1=\bar{r}_{hub}+0.05$, Nm; $Q_{bend}^{\max}(\bar{r}_1)$ and $Q_{bend}^{\max}(\bar{r}_2)$ - the ice blade bending moments about the neutral axis of the expanded blade section at $\bar{r}_1$ and $\bar{r}_2=\bar{r}_1+0.05$, Nm.

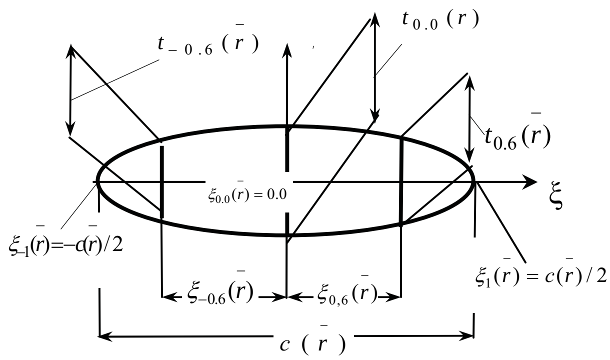


Figure 7: Blade root section scheme at $\bar{r}$.

Methodology for evaluation of permissible stresses based on conditions ensuring fatigue and static strength $\sigma_{perm}$ is given below. Rigidity of the root section to twisting is to be close to the ellipse rigidity. Therefore the thickness of the design root section on coordinates $\xi_{0.0}(\bar{r}_1)=0.0$, $\xi_{0.6}(\bar{r}_1)=-0.6(c(\bar{r}_1)/2)$ (see fig. 7) shall be at least

$t_{0.0}(\bar{r}_1)=1.19\cdot t_{0.6}(\bar{r}_1)$ (4) and $t_{-0.6}(\bar{r}_1)=0.75\cdot t_{0.0}(\bar{r}_1)$ (5)

Requirements for the trailing edge at the coordinate $\xi_{0.6}(\bar{r}_1)=-0.6(c(\bar{r}_1)/2)$ also ensures blade strength for astern movement. For CPP and non-reversible propellers condition (5) may be reduced and the trailing edge thickness may be taken equal to $t_{-0.6}(\bar{r}_1)=0.70\cdot t_{0.0}(\bar{r}_1)$.

## 4.2 STRENGTH SCANTLINGS FOR PERIPHERAL SECTIONS

Strength scantlings for peripheral sections at $\bar{r}=0.6$ are assigned on the basis of ensuring strength at "oblique" bending of its blade edge as well as "right" bending of edge sections due to ice load [2]. Relevant blade loading schemes and typical breakages of its peripheral sections are set forth on fig. 8, 9.

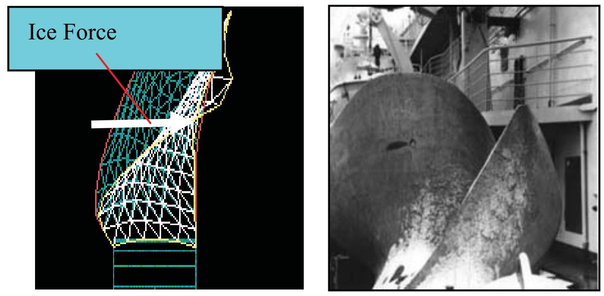


Figure 8: Blade load scheme and typical blade damage under the action of skew blade edge bending.

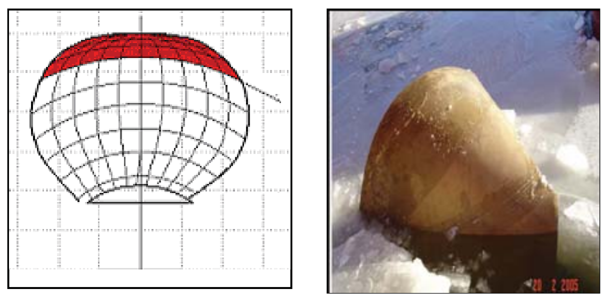

Figure 9: Blade load scheme and typical blade damage under the action of peripheral section bending.

At "oblique" bending of blade edge (fig. 8) the maximum thickness of blade peripheral section $t_{0,0}(\bar{r}=0.6)$, m, at $\bar{r}=0.6$ is to be at least the value calculated by formula

$$t_{0,0}(\bar{r}=0.6)=\left[\frac{17.4\cdot k_{\exp}\cdot(F_{ice})_{\max}\cdot\cos\left(\varphi_{design}(\bar{r}=0.8)\right)\cdot dc}{\sigma_{perm}\sqrt{D^2(1-\bar{r}_{hub})^2+\left(c(\bar{r}_{hub})\right)^2}}\right]^{0,.5}, \qquad (6)$$

where $dc=\sqrt{(0.1\cdot D)^2+0.25\left(c(\bar{r}=0.8)\right)^2}$ ; $k_{\exp}=0.7$ - the coefficient considering evaluation of design geometrical characteristics in the plane of expanded blade contour.

The maximum thickness at $\bar{r} = 0.6$, m, is to be not less than that calculated by formula for the "right" bending of tip blade sections (see fig. 9)

$$t_{0.0}(\bar{r} = 0.6) = \left[ \frac{0{,}12 \cdot D \cdot (F_{ice})_{\max}}{0{,}085\, c(\bar{r} = 0.6) \cdot \sigma_{perm}} \right]^{0{,}5}, \quad (7)$$

The maximum value of those calculated by formulas (6) and (7) is taken as a design thickness at $\bar{r} = 0.6$.

Requirements for the blade strength scantlings at $\bar{r} = 1$ as well as thicknesses of leading and trailing edges are developed on the basis of the operating experience. However, these strength scantlings can be assigned more exactly on the basis of detailed calculation of stressed condition using FEM.

## 5. PERMISSIBLE STRESSES FOR ASSIGNMENT OF STRENGTH SCANTLINGS FOR PROPELLER BLADES

Permissible stresses for assignment of strength scantlings for propeller blades $\sigma_{perm}$ are determined on the basis of ensuring both fatigue and static strength, $\sigma_{perm} = \min((\sigma_{perm})_{st};(\sigma_{perm})_f)$, where $(\sigma_{perm})_{st}$ and $(\sigma_{perm})_f$ permissible stresses based upon conditions ensuring both static and fatigue strength correspondingly.

### 5.1 STATIC STRENGTH OF PROPELLER BLADES

Static strength is to be ensured on the basis of an assumption of saving blade form and prevention of its breakage (splitting into parts) caused by a single ice load. These two factor [2] were studied to develop requirements for the static strength. It is shown that for the bronze blades the design value of permissible stresses $(\sigma_{perm})_{st}$ is to be taken equal to $(\sigma_{perm})_{st} \approx (\sigma_{yield})_{\min}$ while for the blades made of martensitic steels $(\sigma_{perm})_{st} \cong 0.8(\sigma_{yield})_{\min}$

### 5.2 FATIGUE STRENGTH OF PROPELLER BLADES

Blade strength calculation on the basis of fatigue conditions is to be made at the first stage when cracking starts. It is prohibited to use propeller blades with cracks (defects) which size and number exceed permissible level. Permissible level is calculated on the basis of condition of non-propagation of defect [2]. Blade strength scantlings based upon ensuring fatigue strength are to be assigned on the basis of equal term of service life of blade and ship. Meanwhile, it is necessary to take into account the random nature of the blade lifetime. In this case the strength scantlings are to be assigned on the basis of equality of the lower boundary of blade lifetime to the ship service life which is taken equal to 25 years. Compliance with this condition demands a study of the blade lifetime $T_{blade}$ distribution, including its minimum values. The probability and statistical analysis of $T_{blade}$ for IP blades made of steels with various properties including fatigue strength in sea water is made in [17]. IP location, blade surface treatment have been considered. $T_{blade}$ has been determined by the moment of origination of fatigue macrocrack in a blade. The time of macrocrack origination was found during diving inspections and surveys not less than once a month. Origination of fatigue macrocrack is mostly typical in the area of root section close to leading and trailing edges because stresses caused by ice loads during ice milling modes reach the maximum in this area. Blade surface shot peening increases average blade lifetime by 2 times meanwhile reducing durability variation. Blade lifetime of the side IP is 1.5-2.3 times less than for an central one which is conditioned by increase of time of ice interaction and ice loads [17]. 1.5 times increase of conventional limit of the fatigue strength leads to significant increase of blade life time. The latter shows that the fatigue strength is a decisive factor for assigning steel IP strength scantlings.

It is established that $T_{blade}$ distribution corresponds to the three-parameter Weibull law with distribution function

$$F(T_{blade}) = 1 - \exp\left[ -\left( \frac{T_{blade} - (T_{blade})_{\min}}{S(T_{blade})} \right)^{\gamma} \right], \quad (8)$$

where $(T_{blade})_{\min}$ - the minimum possible blade lifetime which limits the lowest boundary of variation of random values $T_{blade}$; $S(T_{blade})$, $\gamma$ – parameters.

$(T_{blade})_{\min}$ is taken for the basic design characteristic for assigning permissible stresses based on ensuring fatigue strength. Such approach guarantees failure free operation of the blade during ship service life. Based upon the rule of linear summation of damages the maximum blade stresses $\sigma_{\max}$ caused by design ice loads $(F_{ice})_{\max}$ shall not exceed the value

$$\sigma_{\max} \le (\sigma_{perm})_f \overset{def}{=} \left(1/k_l\, T_{ice}\, n\right)^{1/m} \cdot \psi(m)\ (\sigma_{-})_{blade}^{design}, \quad (9)$$

where $(\sigma_{-})_{blade}^{design}$ – is the design fatigue strength of blade components consisting of blade at a number of loading cycles equal to $N_0 = 5 \cdot 10^7$; $T_{ice}$ - the relative time of ice propeller interaction; $k_l$ – coefficient considering influence of propeller location on the interaction frequency with ice; $m$ – the constant of material determined upon results of fatigue tests of standard specimen in 3% NaCl sea water in accordance with the fatigue strength curve $\sigma^m N = \sigma_{-1}{}^m \quad N_0$; $\sigma_{-1}$ - an average value of conventional limit of the fatigue strength of standard specimen in sea water for the symmetrical loading cycle at a number of cycles $N_0 = 5*10^7$; $\psi(m)$ – function of *m*.

Parameters $k_l$, $T_{ice}$, $\psi(m)$ are determined and presented in draft RS requirements [18].

Design fatigue strength of blade component $(\sigma_{-})^{design}_{blade}$ is determined by formula:

$$(\sigma_{-})^{design}_{blade} = k^e \ k_{surf} \ \varepsilon \ k_{\mathrm{var}} \ \sigma_{-1}, \tag{10}$$

where $k^e$ - the effective stress coefficient by the symmetrical cycle considering asymmetric properties of the real cycle of loading; $k_{surf}$ - the coefficient considering blade surface treatment; ε – scale coefficient or influence of detail absolute dimensions on fatigue strength; $k_{\mathrm{var}}$ - the coefficient considering the probability and statistical range of conventional limit of the fatigue strength of blade component.

Values of these coefficients are set forth in [18]. $k_{surf}$, ε, $k_{\mathrm{var}}$ coefficients were calculated on the basis of the complex analysis of strength properties of smooth and notched specimen of various thicknesses, strength characteristics of propeller casting material. It is worth noting that $k_{surf}$, ε, $k_{\mathrm{var}}$ values were also calibrated by well-known distributions of random $T_{blade}$ for real blades (see above).

## 6. ENSURING PYRAMIDAL AND FATIGUE STRENGTH OF THE PROPULSION COMPLEX COMPONENTS

### 6.1 REQUIREMENTS FOR PYRAMIDAL STRENGTH

Ensuring pyramidal strength is one of the basic principles underlying design of modern PC for ICB and IGV. The pyramidal strength principle means that if a blade is broken during a non-design mode, all other components of the propulsion complex are to remain intact. This problem is studied in [2, 19]. It is shown that the blade is broken under the plastic deformation conditions. Plastic deformations occur during blade breakage in the areas of stress concentrations of PC components. Propulsion complex component material capability to withstand plastic deformation without breakage is a crucial factor determining pyramidal strength. Considering the above facts defining the ultimate (critical) elasto-plastic deformation of material $\varepsilon_{cr}$, corresponding to its breakage is a key task for development of requirements for the pyramidal strength.

Solution of this task is proposed forth in [2, 19]. Viscous and viscous-brittle breakage modes are to be taken for the design scenarios for assigning $\varepsilon_{cr}$ in order to ensure strength of PC components and determine the ultimate blade damage force. Depending on the viscous and plastic properties of material expressions for $\varepsilon_{cr} = \varepsilon_{cr}(\sigma_{yield}, KV, \delta, t)$ have been gained ($t$ - typical detail thickness) [2,18,19].

Findings are used for development of requirements to the ultimate blade damage force and to strength of PC components in the areas of strength concentration during elasto-plastic deformation.

### 6.2 REQUIREMENTS FOR ULTIMATE BLADE DAMAGE FORCE

Dr. G.V. Boytsov, and the author have prepared method to assign the blade damage force $F^{damage}_{spind}$ under action of spindle torque and bending moment [19]. In addition, another unique method for solving this task has been developed [2]. Two methods give almost similar results. Considering these general approaches laid in the basis of the first method, the last one was taken as a normative one and implemented into draft RS requirements.

According to this method $F^{damage}_{spind}$ is applied at $\bar{r} = 0.8$ and at a distance (2/3) from the blade axis to the leading edge in the plane of its expanded blade contour. $F^{damage}_{spind}$ is opposite to the hydrodynamic thrust. The weakest root section at $\bar{r}$, $\bar{r}_1 \le \bar{r} \le \bar{r}_1 + 0.05$, corresponding to the transition of fillet into blade is taken for the main design one. Breakage occurs during elasto-plastic deformation of root sections and corresponds to $\varepsilon_{cr} = \varepsilon_{cr}(\sigma_{yield}, KV, \delta, t)$. It is supposed to make linear-elastic strengthening of material in the area of plastic deformation. Considering the above approaches the design force $F^{damage}_{spind}$, N, is determined by formula.

$$F^{damage}_{spind} = 0{,}26 \cdot 10^6 \rho_{\mathrm{кр}} \cdot \frac{1}{\ell_{arm}} \cdot c(\bar{r}) \cdot t^2{}_{\max}(\bar{r}) \cdot \left(\beta(\bar{r})\right)^{1{,}5} \cdot \sigma^{damage}_{design}. \tag{11}$$

Ultimate spindle torque $Q^{damage}_{spind}$, Nm, about the blade spindle axis, applied to CPM, shall be determined from the formula

$$Q^{damage}_{spind} = 0{,}166 \cdot 10^6 \cdot k_{frict} \cdot \rho_{\mathrm{cr}} \cdot \frac{C_p(\bar{r} = 0{,}8)}{\ell_{arm}} \cdot c(\bar{r}) \cdot t^2{}_{\max}(\bar{r}) \cdot \left(\beta(\bar{r})\right)^{1{,}5} \cdot \sigma^{damage}_{design} \tag{12}$$

where $\rho_{cr} = \frac{1}{\left(1+k_{cr}^{1,5}\right)^{2/3}}$ ; $k_{cr} = \frac{3\frac{C_p(\bar{r}=0,8)}{c(\bar{r})}}{1+4,7\left(\frac{\ell_{arm}}{c(\bar{r})}\right)^2}$ ;

$k_{frict}$ - friction factor between the blade flange and hub; $C_p(\bar{r}=0,8)$ - distance by the chord of the expanded blade section at $\bar{r}=0,8$ from the blade spindle axis to the leading edge or half length of the chord along this section, whichever is greater, m; $\ell_{arm}=(0,8-\bar{r})\cdot R$, m; $t_{\max}(\bar{r})$ and $\beta(\bar{r})$ - the maximum thickness, m, and fullness coefficient of the design blade section at $\bar{r}$ ; $\sigma_{design}^{damage}$ - design yield stress of the blade material, MPa;

Design yield strength of blade material $\sigma_{design}^{damage}$, MPa, shall be determined from the formula

$$\sigma_{design}^{damage} = k_{stat}\cdot(\sigma_{yield})_{\min}\cdot\left[1+\frac{2}{3}\cdot(\bar{\varepsilon}_{cr})_{\max}\cdot\left(\frac{(\sigma_{tensile})_{\min}}{(\sigma_{yield})_{\min}}-1\right)\right] \quad (14)$$

where $(\bar{\varepsilon}_{cr})_{\max} = (\varepsilon_{cr})_{\max}/\delta_p$, equal to 0,35 for martensitic steel and 0,75 for austenitic steels and bronze; $(\varepsilon_{cr})_{\max}$ - the maximum possible value of $\varepsilon_{cr}$; $k_{stat}$ - the safety coefficient considering probability and statistical spread of strength properties of material of propeller casting; $(\sigma_{yield})_{\min}$, MPa.

Coefficient $k_{stat}$ is set on the basis of probability and statistical analysis of strength properties and it is taken equal to $k_{stat}=1.3$

### 6.3 STRENGTH REQUIREMENTS FOR THE AREAS OF STRESS CONCENTRATION DURING ELASTO-PLASTIC DEFORMATION

Strength requirements are presented in the form of dependency [18,19]

$$\varepsilon\left((\sigma_{yield})_{\min},\delta_{\min}\right)\le k_{safety}\,\varepsilon_{cr}\cdot\left((\sigma_{yield})_{\min},\delta_{\min},KV_{\min},t\right), \quad (15)$$

where $\varepsilon$ - the value of the elasto-plastic deformation in the area of stress concentrations,

Elasto-plastic deformation $\varepsilon$ is determined on the basis of Neiber approach.

Condition of ensuring pyramidal strength of PC components is added with compliance with the traditional criterion based on consideration of nominal stresses as well as requirements for materials.

### 6.4 ENSURING FATIGUE STRENGTH OF PROPULSION COMPLEX COMPONENTS

Ensuring fatigue strength of the CP components (CPM, AZT fastenings to hull, shafts, gears etc) is the prerequisite for ensuring failure free operation in ice conditions. Strength scantlings of a PC component with stress concentrator are to be assigned in such manner that stresses caused by ice loads shall not exceed the yield strength. Possibility of occurrence of plastic deformation is to be limited by single loads during blade breakage. For the time being, RS has developed requirements for ensuring fatigue strength of CPM components [18]. This approach can be used for other PC components also, for instance, blade-hub fastenings (bolts).

For the time being, studies on improvement of fatigue strength requirements to the components of the propulsion line are underway. It is necessary to use the method of straight calculation of dynamic ice loads in the screw-shaft system in order to calculate the fatigue strength of propulsion line components.

It is necessary to perform an in-depth R&D to develop requirements for the main thrust bearings of azimuth thrusters. Basic approaches to ensuring operating strength of rolling bearings have been developed. It is shown that fatigue caused by joint impact of ice and hydrodynamic loads is a decisive factor for choice of type of bearing and evaluation of necessary durability.

## 7. VERIFICATION CALCULATION AND CALIBRATION OF DEVELOPED METHODS

Development of modern requirements is impossible without their verification and calibration on the basis of the operating experience. For the time being, the developed requirements for propeller blades and other components of the propulsion complex have been tested.

IP blade strength scantlings have been calculated for ships of Russian and foreign construction with the ЛУ4, ЛУ5, ЛУ7 ice-strengthenings and their equivalents as well as for the icebreakers. Calculations were made for the Russian propeller steels as well as for *NIAL* bronzes. Surface treatment was taken into account for steel blades.

It is shown that strength scantlings of steel IP are determined by fatigue while the *NIAL* bronze blades scantlings are determined by static strength. Application of shot peening allows to reduce steel blades scantlings by 10-15%. The latter is of utmost importance to ensure pyramidal strength of PC of modern DAT ice going vessels.

Submitted findings of calculation of IP components strength scantlings correspond to the operating experience in full. Design strength scantlings agree with strength scantlings of safely operating propulsion complexes with an accuracy sufficient for modern design and operating practice.

## 8. CONCLUSION

On the basis of results shown above a new draft of the RS Rules for the Classification and Construction of Sea-Going Ships were prepared for:

- IP blades for arctic ships and icebreakers;
- controllable pitch mechanism for IP;
- strength of main components of azimuth thrusters located in the lines of force considering dimensional characteristic of components (details), visco-plastic properties of material comprising part of this material as well as stress concentration.

The said drafts are presented in the Collection of Regulating Documents of the Register, Book#12 and 13, edition 2004.

Draft requirements have been successfully tested. Within a scope of special consideration the draft have been used for designing and approval of technical documentation for the PC of modern ice going vessels and icebreakers, including Double-acting ships.

## 9. ACKNOWLEDGEMENTS

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## 11. AUTHORS BIOGRAPHY

**Mr. A.V. Andryushin** graduated from the Saint Petersburg Marine Technical University in 1983. 1983-1999 was employed by the Krylov Shipbuilding Research Institute. Specialized in the study of ice loads on the components of propulsion complexes of ice ships and icebreakers as well as in the area of development of probability and statistical methods for risk evaluation.

In 1995 he drafted and offered for defence his first Doctor's Thesis on experimental methods of designing icebreaker's propulsion complexes. Since 1999 he has been working in Russian Maritime Register of Shipping. The specialization is ensuring strength of components of propulsion complexes of ice ships.

# UPGRADE OF CANADIAN COAST GUARD TYPE 1100 ICE CAPABILITY

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## SUMMARY

The Canadian Coast Guard's Type 1100 vessels were designed in the early 1980s with Canadian Arctic Class 2 capability, and are operated in the summer in the Western Arctic. Over the years, experience has shown a problem with the design which induces violent decelerations during ice ramming events. CCG has decided to take advantage of a life extension program in order to introduce a number of upgrades to the ice capability. The first vessel in the program, the CCGS Sir Wilfrid Laurier, is currently undergoing its modernization.

BMT Fleet Technology Ltd was contracted on behalf of CCG to develop a set of modifications; including a redesigned stem and ice knife. This was done using mathematical modelling of the ice ramming process in order to reduce the impact loads and decelerations. The new IACS Unified Requirements for Polar Class ships were then used to verify the new structure against PC 5 strength requirements. This paper describes both the theoretical approach and also the practical challenges of retrofitting the modifications to an existing vessel

## 1. INTRODUCTION

The Canadian Coast Guard's Type 1100 vessels were designed in the early 1980s with Canadian Arctic Class 2 capability. CCG has decided to take advantage of a life extension program in order to introduce a number of upgrades to the ice capability. The first vessel in the program, the CCGS Sir Wilfrid Laurier, is currently undergoing its modernization. This vessel is annually deployed to the Western Arctic from its home base in British Columbia. During this voyage, the vessel frequently encounters heavy ice condition and experiences operations that cause the vessel to impact ice features that bring it to a stop and backing and ramming evolutions are necessary. Over the years, experience has shown that these operations can cause violent decelerations during the ice ramming events particularly in lighter draft conditions. These can cause intolerable motions on the bridge and in the accommodation/working deck.

The design of the Type 1100 includes a large ice skeg of an 'appendage' type ahead of the main hull. The skeg incorporates the bow thruster in this class of vessel. The skeg has a wide, face and is near vertical in profile. This impact profile induces the very high decelerations during rammings when the skeg impacts the ice edge. It also tends to push large ice features ahead of the ship, especially in lighter conditions, rather than breaking or deflecting these. Neither trait is operationally desirable.

BMT Fleet Technology Ltd. (BMT FTL) proposed a modification to the current skeg design that would mitigate both of these traits at acceptable cost and this paper outlines the development of the conceptual design considerations, and describes the challenges of implementing such a modification into an existing vessel.

## 2. ICE RAMMING IMPACT

The Sir Wilfred Laurier's skeg is shown in Figure 1 as she entered drydock in early 2007.


Figure 1.

The ice impact simulation of the type 1100 bow profile immediately after the bow impact can be described as a three phase evolution as shown diagrammatically in Figure 2.1 At initial contact, the vessel has a velocity $V_{ship}$. The initial kinetic energy of the ship is eventually completely lost during the impact.

During the first phase, the bow indentation phase, the 'normal' kinetic energy is lost and a notch in the ice edge is created. The 'normal' velocity $V_n$ is the component of the ship velocity that is normal to the stem. It is assumed that this initial penetration occurs without pitch.

During the next phase, the ship slides up the notch created in the ice edge, converting kinetic energy to potential energy. The ice knife then contacts the ice and creates a second notch. The ice knife penetration continues until all kinetic energy is used. The ice force will be largest at the end of each phase. The accelerations at the end of each phase can be calculated from the force vector and mass values.

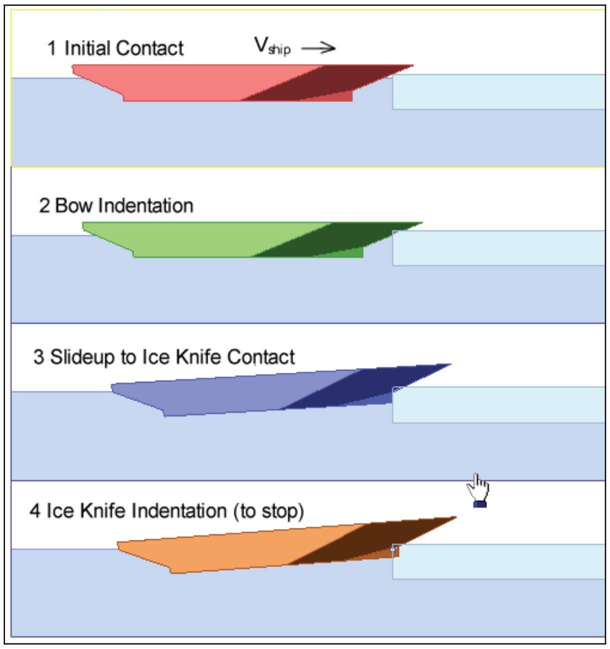


Figure 2: Ramming Sequence

The following analysis is an examination of the forces and motions that occur during ramming. The analysis is based on a sequence of contact events. An energy approach is used to determine the extent of each phase of the process.

## 3. IMPACT ANALYSIS

The sequence shown in Figure 2 simplifies the ice knife indentation component of the ram. The ice knife is also an inclined wedge and Figure 3 shows this interaction in more detail when impacting thick ice (deeper that the skeg). The ice knife will start to interact with the ice face (Figure 3 (b)) as the bow slides up the main bow imprint. The first phase of the ice knife interaction ends when the bottom of the knife contacts the ice (Figure 3.(c)). If the interaction continues (Figure 3 (d)), the bow portion of the area will diminish and may become too small to support the vertical force. If this occurs, the bow and ice knife will begin to indent horizontally (Figure 3 (e)).

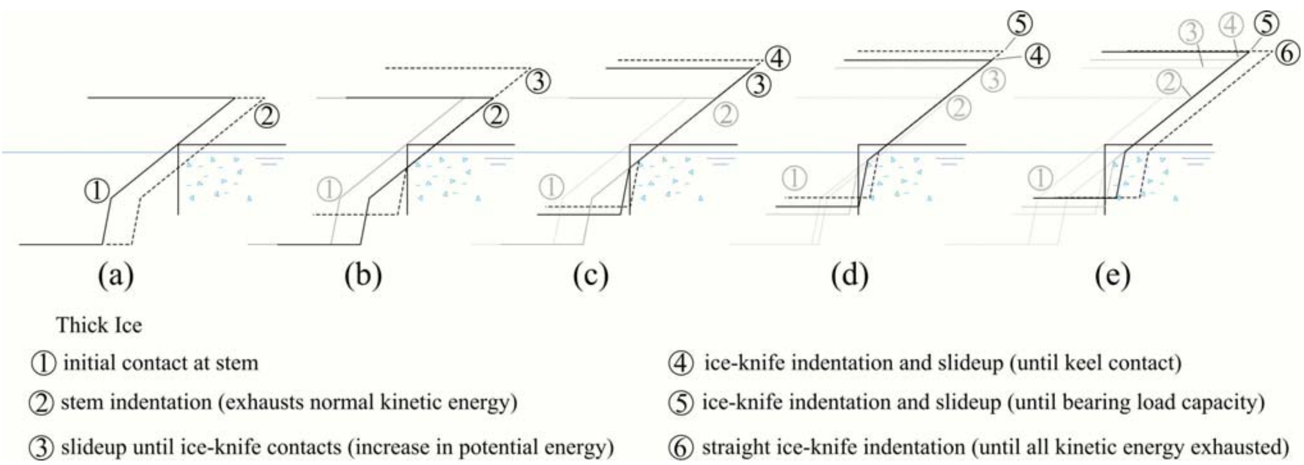


Figure 3: Ramming Sequence in Thick Ice in Detail

Figure 4 shows a perspective of the ice indentation caused by the assumed geometry and leads to the formulation of the ice impact simulation created for this analysis.

A numerical simulation was developed which calculates the peak accelerations at the bow impact and knife penetration stages based on the forgoing process and hull form geometry. The absolute magnitude of the answers derived from this calculation is dependent on the ice property assumptions, ship geometry and added mass calculations. They have not been verified in full-scale trials, however, the general magnitude of the answers is correct and when systematic changes in input values are exercised the trends of resultant accelerations are correct.

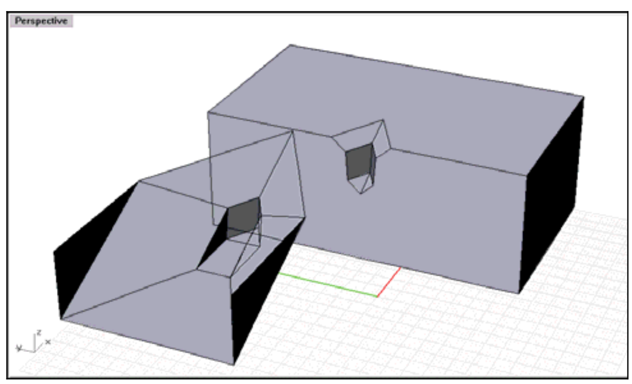


Figure 4: Sketch of Indentation

Figure 5 shows the speed / deceleration relationship at the first phase or "Stem" impact of the bow and at the final phase or "Knife" impact. Clearly the lightship condition where the draft reduction means that the knife impacts sooner in the ram, results in high decelerations, note this condition is also a reduced mass condition.

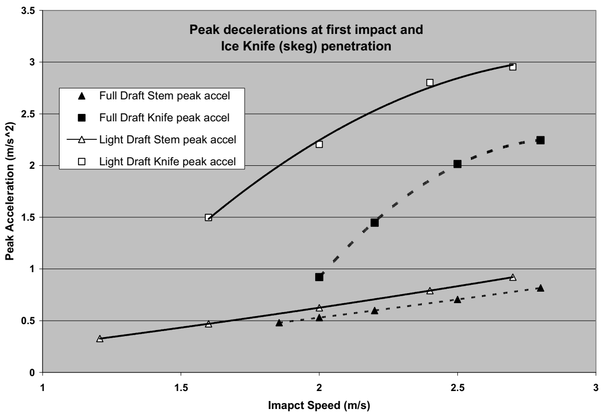


Figure 5: Peak decelerations

There are two options to redress the difference in peak decelerations, firstly to increase the bow stem angle, this will increase the rate at which the ship rises in the ice notch and converts kinetic energy, secondly to change the shape of the skeg impact face so as to change the impact force. The latter could be done by making the skeg sharper in plane or by changing the angle of the profile. The presence of the bow thruster fitted in the skeg constrains the extent by which modifications can be made to the skeg itself. Changes to the stem angle would also incur significant consequential changes in the ship structure. Therefore, a "knee" section was proposed to fit between the near vertical ice knife skeg and the bow stem-line as an add-on.

The concept of the knee or "fill in" piece is to effectively increase the stem angle from the point of contact with the ice in the light draft condition to the skeg. This will enlarge the ice notch necessary to be made in the ice feature and force the bow to ride up more rapidly increasing the first stage energy loss and thus redistributing the deceleration. A sketches of a variation in skeg-to-bow knee is shown in the following figure. For ease of analysis, a flat plane or straight line geometry is assumed, thus these rendering diagrams do not display fair or smooth curves as seen at the ship

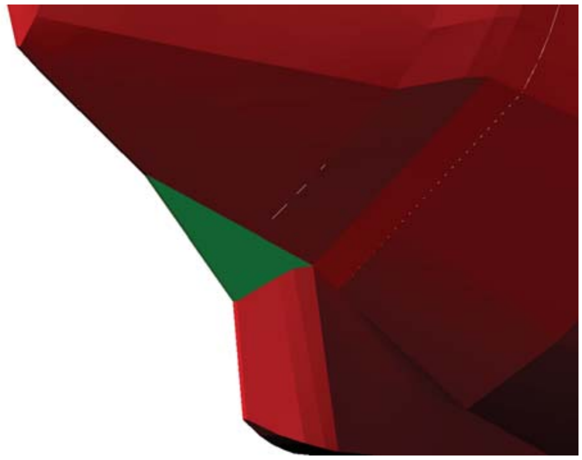

Figure 6: Skeg Knee from 2.5 Meter to 4.5 Meter Waterlines

The geometry of the changed bow shape was analyzed using the same logic model for peak accelerations using the light draft condition. The skeg knee increases the first phase peak decelerations and reduces the final stage decelerations as predicted.

A series of such knee shapes were investigated and the effect on peak acceleration. Figure 7 shows the results for a knee connecting the 2.5 meter water line knife intersection with the 4.5 meter waterline to stem intersection and a second connecting intersections at 2 meters and 4.5 meter is shown.

Fitting a knee between 2.5 and 4.5 meter waterlines reduces the maximum deceleration in the light draft condition to that level experienced in the full draft condition as built now, i.e., around the 2.5 meters/second (approximately 5 knots) impact velocity the peak ice knife deceleration in the as built deep draft condition is the same as that with the ice knee. However, fitting a knee between the 2 and 4.5-meter waterlines, the reduction in ice knife peak deceleration is reduced to a level close to that experienced in the initial contact and an overall smoother stopping action is predicted.

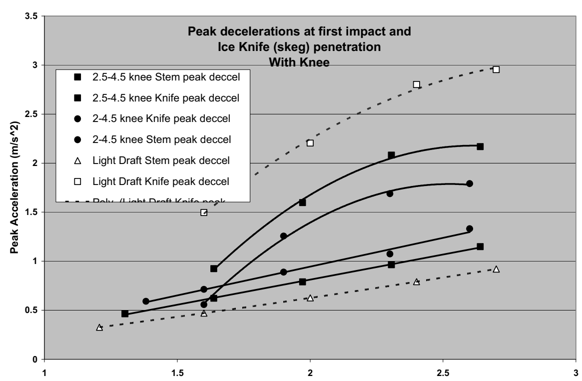


Figure 7: Redistribution of peak decelerations

In each case the light condition is shown and clearly the effect of adding a stepper section to the stem redistributes the decelerations from the final knife interaction to the earlier ride up phase.

A second ice performance related phenomena reported of the Type 1100 hull form in smaller ice flow conditions (less than ramming conditions), is that it tends to push a large amount of ice ahead of the bow rather than deflecting it around the hull. This is especially the case in the lighter draft conditions. It is suspected that this is also a function of the ice skeg shape and location. There is essentially zero downward force acting from the bow on an ice flow in the light condition as the bow impact area is small, i.e., the ship's motion is reacted on by the flow in the waterline plane by the near vertical skeg. The knee described above will provide a downward force component at the lighter drafts. It is predicted that this will significantly reduce the tendency for the ice flow to be pushed ahead of the ship. This is illustrated in the following Figure 8.

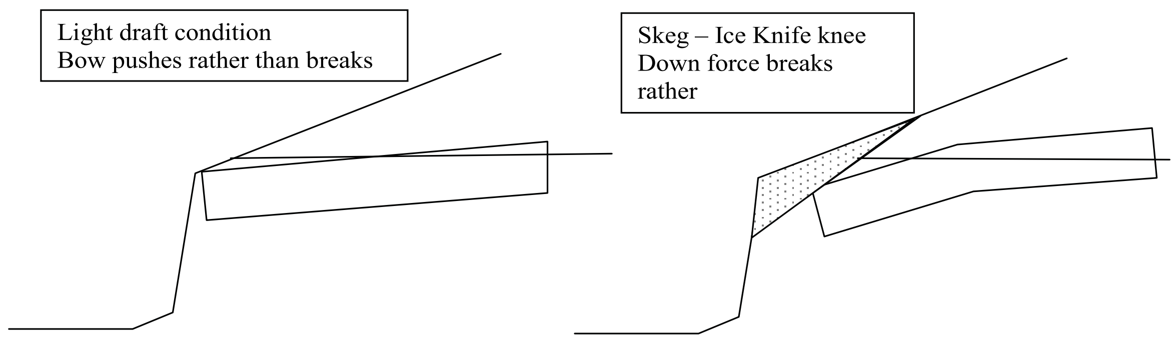


Figure 8: Increase Down Breaking Force of Knee

The other effects of a knee will be negligible.

Hydostatics will be such that the additional steel weight of any added structure will counter any added buoyancy therefore trim effects will be very small.

Change in open water resistance will be marginally to the better as it is expected that flow around the skeg will be improved.

Impact on the performance of the bow thruster will be zero if the lower part of the knee is kept above the 2.0-meter waterline.

## 4. STRUCTURAL INTEGRATION

After demonstrating that some considerable improvement in the ice interaction decelerations were achievable, the issues surrounding integrating the additional shape and volume of the knee into the existing structure was addressed.

The structural arrangement of the final concept is shown in figure 9. The stem bar and the face plate of the ice knee are built from a 250*120 solid bar. A casting integrates the stem and knife and the hull plating is scarfed into the side of each component. As noted the knife also houses the bow thruster and the structure forward of the thruster housing comprises plate floors which are covered with a 35mm thick plate plug welded to the floors. This internal area is virtually un-inspectable from the inside and it was filled with a foam substance. To form a Knee in the join of the stem, a single vertical plate of the desired profile was inserted on the centreline. Orthogonal plate stiffeners with a flange on the outboard edge were added to form the shape and a 25mm plate warped to each side to form the volume of the new knee. The closures were made using the same plug welding technique as on the knife.

The design of the structural components was based on IACS Unified Requirements for Polar Class ships were then used to verify the new structure against PC 5 strength requirements.

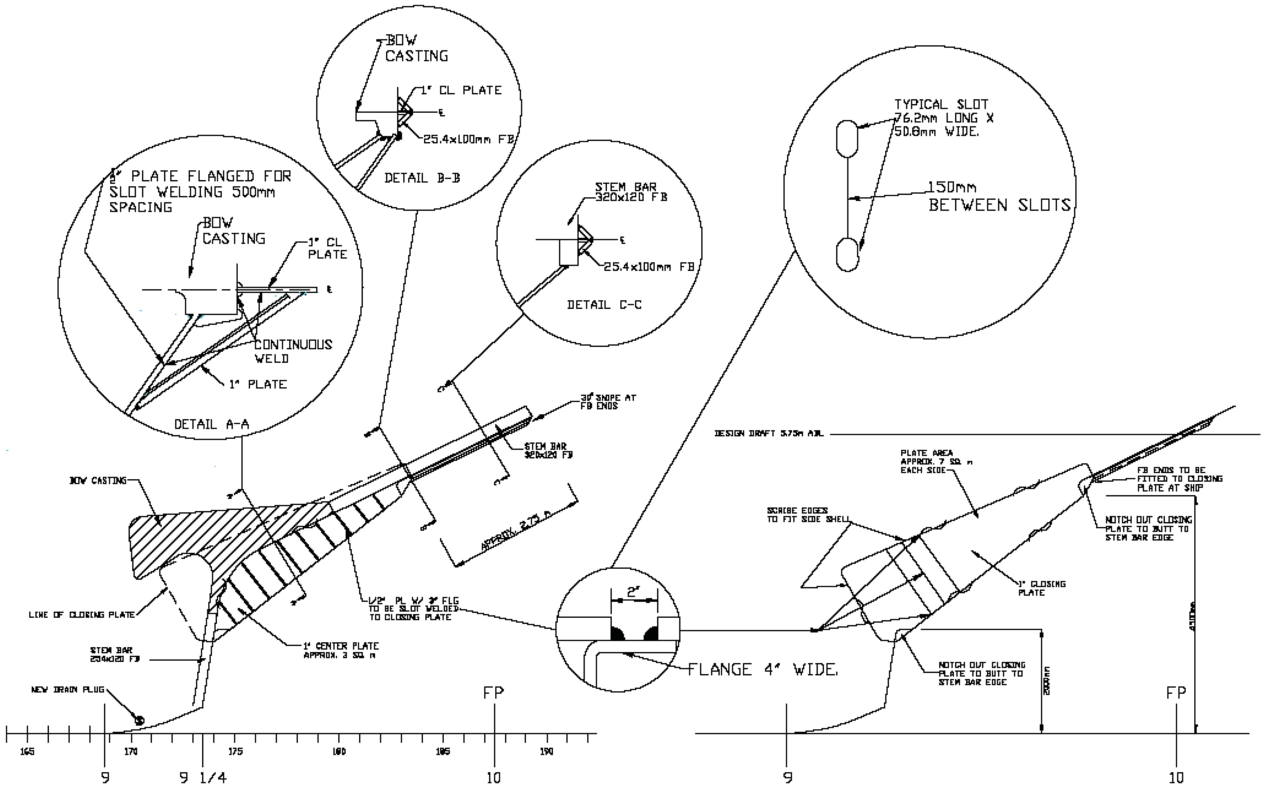


Figure 9: Concept structural drawing

At the time of drydocking the vessel was inspected and a mould lifted of the bow shape, the side plates of the new knee were arranged to be a flat as possible and the line of intersection with the existing bow arranged to meet along the line of a weld seem separating a change of plate thickness on the bow.

The form of the knee was also modified at the aftermost outboard ends to facilitate ease of construction. It was seen that the shape of the current bow and the new knee would form a space which would be virtually impossible to arrange stiffening for and was therefore sniped short of the intersection of the knife and bow.

Another issue which arose was access to the docking pug for the bow void. In this case the solution was to weld a pipe over the current docking plug and fit a closing piece on the knee surface which could be cut out when needed.

The following figures show some of the difficulties and solution adopted to fit the new knee.

Figure 10: Fitting the warped side plates

Figure 11: Snipe at the after intersection

Figure 12: Access to the bow void docking plug

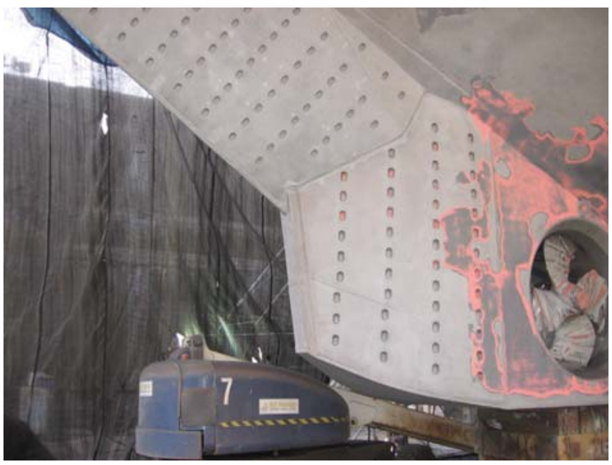

Figure 13: The finished plug welds

Figure 14: Ready to Float.

## 5. CONCLUSIONS

The inclusion of a bow shape modification to the type 1100 has been successfully implemented. The success of the new bow form in heavy ice will be tested this summer during the vessels annual voyage to the western Canadian Arctic.

## 6. ACKNOWLEDGEMENTS

The Authors would like to acknowledge the Canadian Coast Guard personnel who advanced the idea of this improvement and also the staff at Victoria Shipyards for their assistance in completing the difficult job of fitting new to old.

## 7. REFERENCES

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## 8. AUTHORS' BIOGRAPHIES

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# MODEL EXPERIMENTS TO SUPPORT THE DESIGN OF LARGE ICEBREAKING TANKERS

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## SUMMARY

In 1997, Samsung Heavy Industries became interested in applying its expertise in the design and construction of oil tankers to the specialized construction of ice class vessels for oil transportation in the Arctic Ocean and Baltic Sea. This interest was motivated by the potential development of several offshore and near shore oil and gas reserves together with increased shipping of oil through the Baltic Sea from Russia. Since at that time, Samsung Heavy Industries had little experience with performance prediction for ships in ice, they entered into a collaborative project with the Institute for Ocean Technology to apply and refine the modelling techniques required for predicting the performance of large tankers in ice.

This paper describes the modelling methods used. One important technique is the preparation of the model ice, and the scaling of the ice forces. Equally important is the preparation of the ship model and its propulsion system. The two models are combined to predict the powering and manoeuvring performance for large tankers in a range of ice conditions including level first year ice, pack ice and rubble ice. The results of experiments on four hull designs, with single and twin-screw propulsion arrangements, are presented and discussed and some suggestions are made for refining the modelling techniques for future projects.

## NOMENCLATURE

| | |
|---|---|
| $B$ | Maximum beam of the model, m |
| $C_b$ | Coefficient of the buoyancy resistance, $R_b$ |
| $C_{br}$ | Coefficient of the breaking resistance, $R_{br}$ |
| $C_c$ | Coefficient of the clearing resistance, $R_c$ |
| $C_o$ | Ice concentration |
| $F_p$ | Average pack ice force, N |
| $g$ | Acceleration of gravity (9.808 m.s$^{-2}$) |
| $h_i$ | Ice thickness, m |
| $R_b$ | Resistance due to buoyancy of the ice, N |
| $R_{br}$ | Resistance due to breaking the ice, N |
| $R_c$ | Resistance due to clearing of the ice, N |
| $R_{ow}$ | Resistance due to open water, N |
| $R_t$ | Total resistance in ice, N |
| $T$ | Maximum draft of the model, m |
| $V_m$ | Model velocity, m.s$^{-1}$ |
| $\Delta\rho_i$ | Difference in density between ice and water. |
| $\lambda$ | Linear scale of the model |
| $\mu$ | Hull-ice friction coefficient |
| $\rho_i$ | Density of the ice, kg.m$^{-1}$ |
| $\sigma_f$ | Flexural strength of the ice, N.m$^{-2}$ |

## 1. INTRODUCTION

In 1997, Samsung Heavy Industries (SHI) became interested in applying its expertise in the design and construction of oil tankers to the specialized construction of ice class vessels for oil transportation in the Arctic Ocean and Baltic Sea. This interest was motivated by the potential development of several offshore and near shore oil and gas reserves together with increased shipping of oil through the Baltic Sea from Russia. Since at that time, Samsung Heavy Industries had little experience with performance prediction for ships in ice, they entered into a collaborative project with the Institute for Ocean Technology (IOT) to apply and refine the modelling techniques required for predicting the performance of large tankers in ice.

A major portion of the effort for this project was for Naval Architects from Samsung Heavy Industries to become familiar with the challenges associated with designing and building ships for operation in ice covered waters. To achieve this objective, several staff from SHI's Marine Research Institute spent extended periods of time in St. John's, Newfoundland, Canada at the National Research Council's Institute for Ocean Technology (formerly Institute for Marine Dynamics). Staff from IOT explained the modeling processes for ships in ice and provided background literature on ship performance in ice-covered waters to the SHI staff.

The three main types of ice considered for this project were:

- Level first year ice,
- Pack ice in concentrations from 80% to 95%,
- Rubble ice, which consisted of multiple layers of ice, up to three times the initial thickness of the component ice sheet.

Brash ice, to simulate a channel broken by an icebreaker (or other ice capable ships) was also prepared by cutting a channel with straight edges. The ice within the channel was broken into small flows and evenly distributed over the area of the channel to give a nominal concentration between 90 and 100%. This required compacting the ice so that the final length of brash ice was less than the length of the un-broken ice sheet.

This paper presents a summary of the latest methods used by the Institute for Ocean Technology for predicting the performance of large tankers in ice, and presents the results of performance predictions for four ships designed as part of this project. Earlier procedures for model testing at IOT [1] were used as the starting point for developing and refining these techniques. The designs evaluated consisted of two twin screw Aframax sized tankers designed for Arctic ice conditions [2], a twin screw Suezmax sized tanker designed for Arctic ice conditions [3] and a single screw tanker, with bulbous bow, designed for Baltic ice conditions. All five icebreaking tanker designs were compared with a conventional tanker in open water, pack ice and level ice [4].

## 2. MODELING THE SHIP AND THE ENVIRONMENT

### 2.1 ICE

The EG/AD/S (CD) model ice prepared in the ice tank at IOT has been developed to provide the kinematic and mechanical characteristics required to model the ship-ice interaction correctly. The ice is grown at a carefully controlled temperature in a mild EG/AD/S (Ethylene Glycol/ Aliphatic Detergent/ Sugar) solution resulting in uniform thickness, with standard deviation normally less than 3%. Fine bubbles are selectively incorporated into the ice to produce the required ice density and plate stiffness. The ice is tempered for a period of time before the test, until the required flexural strength is achieved. Shear strength and compressive failure stresses are established as functions of the flexural strength, similar to the full scale relationships. The ice has a columnar grain structure as is normally found in nature.

Ice flexural strength is measured by sets of cantilever beam tests at different times and locations in the tank. For each ice sheet, flexural strength-time curves are developed, and strength is interpolated to test time and location. Ice thickness is measured every two metres along the ship track after a test. Ice density, shear strength, and compressive failure stress are determined from flexural strength relations, calibrated by measurements in each ice sheet. Pack ice concentration is determined from digitized overhead photographs of each ice sheet.

Additional ice conditions can be prepared from the level ice sheets, after completion of tests in this ice condition. Brash ice is prepared on the centreline of the ice tank, by cutting a channel with straight edges. The width of the channel is determined to be some fraction of the ships beam, and will vary depending on the requirements of a particular project. The ice within the channel is broken into small flows and distributed evenly within the channel. Nominal concentration within the channel should be between 90 and 100%. This requires compacting the ice so that the final length of brash ice is less than the length of the unbroken ice sheet. Photographs of the brash ice are taken and analyzed to estimate the concentration of ice within the channel.

Pack ice can be prepared in a similar manner by breaking the ice sheet into approximately uniform floes, and distributing them evenly over the test area. Photographs of the pack ice are taken and analyzed to estimate the concentration of ice within the test area. Two concentrations of pack ice were used in this project (95% and 75% nominal values).

### 2.2 SHIP MODELS

A typical scale for a tanker model at IOT is approximately 1:35. This provides an adequate compromise between the size of the model hull and propeller, together with the required ice thickness and flexural strength at model scale. Ice thickness and strength both scale linearly with the scale factor. Model hulls are constructed from a Styrofoam™ Hi 60 polystyrene foam core with a ¾” plywood floor and Renshape™ for areas requiring reinforcement. An internal structure of wooden frames and a deck provide additional strength. The foam is milled, with a 5-axis computer controlled milling machine, to the required shape of the hull. After hand smoothing the foam is covered with 3 layers of 10oz glass fibre cloth and epoxy resin. The internal surfaces of the model are covered with one layer of glass fibre cloth and resin to bond the structure together.

The external surface is primed with Duratec™ Primer Surfacer, sanded to 80 grit, followed by Duratec™ Primer sanded to 220 grit. The model and appendages are painted with 3 coats of Imron™ Caterpillar yellow finish, with the final surface finish having a friction coefficient to match the nominal value between a new ship and sea ice. A wooden board finished with the same surface preparation as the model is made at the time of model construction. This can be used to determine the hull-ice friction coefficient. The model is fitted with a propeller shaft, rudder, ice knife and any other appendages. The model is marked with 11 stations, the centerline and the design waterline.

Power to the propellers is provided by an electric motor fitted to the propeller shaft. A strain gauge dynamometer is used to measure thrust and torque on the shaft. The model is towed with a tow post incorporating a gimbal, which allows the model freedom to sink and trim, but restrained in yaw. Tow force is measured using a load cell built into the gimbal. Rotation rate on the propeller shaft was measured by a tachometer.

## 3. DESCRIPTION OF MODEL EXPERIMENTS & ANALYSIS METHODS

### 3.1 RESISTANCE IN LEVEL ICE

The method used for carrying out resistance experiments in ice assumes that four different forces occur when a ship moves through ice. These forces are due to the breaking the ice, the movement of the ice pieces around the hull, the friction of the ice against the hull, and the open water resistance (which is itself probably modified by the presence of the ice). These forces all scale differently to full-scale. Therefore, tests are conducted in open water, in level ice, and in pre-sawn ice in order to determine the resistance due to the different processes. Also, by using non-dimensional coefficients, it is easy to extrapolate the results to full-scale.

Therefore, we have,

$$R_t = R_{br} + R_c + R_b + R_{ow} \tag{1}$$

Note that the breaking resistance, $R_{br}$, is the only term that cannot be measured directly in the ice tank.

The open water term, $R_{ow}$, is determined by first testing the model in open water at the same speeds as those used in the ice tests.

The theory of the pre-sawn test is that it measures everything except the breaking term, ($R_c$+$R_b$+$R_{ow}$). Since $R_{ow}$ is known, the pre-sawn test determines $R_c$+ $R_b$ at each speed. By conducting a pre-sawn test at very low speed, $V_M$=0.02 ms$^{-1}$, the dynamic forces associated with ice block rotation, ventilation, and acceleration are negligible, leaving only buoyancy, and a sliding friction term which is included in $R_b$. Having measured $R_b$, which is independent of velocity, it is subtracted from $R_c$+$R_b$ to give $R_c$, which is velocity dependent. $R_{ow}$, and $R_t$ are also measured for each velocity. Thus $R_{br}$ can be calculated from equation (1) above, and all components can be determined.

In order to scale the model results to full-scale it is convenient to deal with non-dimensional coefficients for the resistance terms. These are defined as:

$$C_{br} = \frac{R_{br}}{\rho_i B h_i V_M^2} \tag{2}$$

$$C_c = \frac{R_c}{\rho_i B h_i V_M^2} \tag{3}$$

$$C_b = \frac{R_b}{\Delta \rho_i g B h_i T} \tag{4}$$

A non-dimensional strength number is defined as;

$$S_n = \left[ \frac{\rho_i B V_M^2}{\sigma_f h_i} \right]^{1/2} \tag{5}$$

Natural logarithms of $C_c$ are plotted against natural logarithms of $Fn_h$, where $Fn_h$ is the depth Froude number

$$Fn_h = \frac{V}{\sqrt{g h_i}} \tag{6}$$

and natural logarithms of $C_{br}$ are plotted against natural logarithms of $S_n$. Linear equations are fitted to both these relationships, and these equations are used to predict the effect of ice strength, thickness, densities of ice and water and ship speed within the range of the data obtained from the experiments.

The resulting force components are scaled from model to full scale by $\lambda^3$, except for the open water resistance, which includes a viscous scaling factor, based on the ITTC 1957 line. Figures 1 and 2 show a model in level ice and pre-sawn ice.

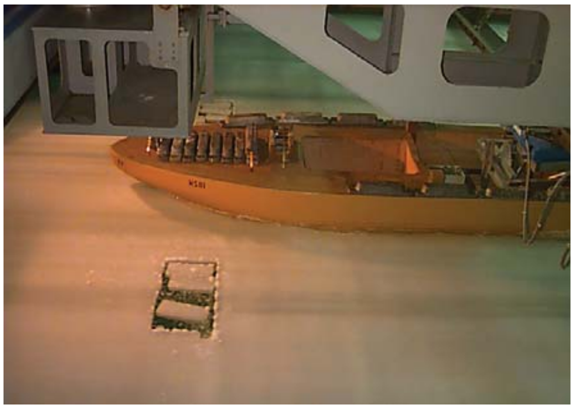

Figure 1: Model tanker in level ice

Figure 2: Model Tanker In Pre-Sawn Ice

### 3.2 RESISTANCE IN PACK ICE

A method for analyzing the results of resistance in pack ice has been presented [6] which considers only the buoyancy and submergence forces caused by the ice on the ship's hull. This method is the same as the analysis of the pre-sawn resistance component used in level ice resistance analysis, with the addition of an ice concentration component. For pre-sawn ice (100% concentration) this factor has a value of 1.0. A model tested in pack ice is shown in Figure 3.

Figure 3: Model Tanker In Pack Ice, 95% Concentration

In the analysis of level ice resistance (presented above) it was assumed that there were four force components, all of which scale separately. In the case of resistance in pack ice, provided that the flow sizes are small and there is very little breaking component, the ice breaking forces can be ignored. Resistance forces on a ship model due to pack ice are determined by subtracting the hydrodynamic resistance, determined from the open water experiments, from the total measured resistance.

The remaining force component can be non-dimensionalized using

$$C_p = \frac{F_p}{1/2\rho_i B h V_i^2 C_o^n} \qquad (6)$$

Velocity can be non-dimensionalized using Pack Ice Froude Number ($Fn_p$,). The linear function

$$Fn_p = \frac{V_m}{\sqrt{g h_i C_o}} \qquad (7)$$

was found to be the most appropriate.

The two coefficients are related by a function derived from the measured data. Experience has shown that $Ln(C_p)$ is a linear function of $Ln(Fn_p)$

Colbourne [5] recommended a value of 3 for n in equation (6), based on data for speeds appropriate for moored ships or FPSOs, where the only flow component was caused by a current. Analysis of the arctic tanker data for SHI, together with other ships tested in pre-sawn ice and pack ice, suggests a value of 2 collapses pack ice and presawn ice resistance onto a single line, with the smallest error band.

### 3.3 DELIVERED POWER IN ICE

The principle of IOT's method for predicting delivered power in ice is that overload experiments in open water can be used to predict the hydrodynamic torque required to develop a thrust sufficient to move the hull against a force equal to the hull resistance in ice. Because such open water tests cannot take account of any ice-propeller interaction, it is necessary to conduct a corresponding experiment in ice to determine the increase in torque due to propeller-ice interaction. It is assumed in this method that propeller-ice interaction has a negligible effect on the thrust developed by the propulsion system. This has been shown to be true for small values of $h_i/D$ where $h_i$ is the ice thickness and $D$ is the diameter of the propeller. The torque due to ice is considered a function of the ice parameters (thickness, strength etc.) and added to the open water values. This method is applicable to all types of ice, provided overload experiments in are carried out in each ice condition.

This method has the practical advantage that because the towing carriage arrangement for resistance in ice tests and overload propulsion in ice tests are identical, it is possible to change quickly from one to the other. Thus, resistance and propulsion experiments in the same ice sheet are possible.

For overload experiments in open water the model is towed, as in resistance experiments, but the with the propellers operating. The speed range of interest was the ice-breaking condition from zero up to 8 knots. Thrust, torque and revolutions were measured, together with model resistance, which for low speeds and high delivered power was a towrope pull. Five different rates of shaft revolutions up to approximately maximum delivered power for the ship were tested at each forward speed. Measured torques were corrected to the value at the propeller by carrying out experiments before and after the propulsion experiments to determine the mechanical friction in the stern tube bearings.

Self-propulsion experiments in ice using an overload method were conducted in a similar manner to open water experiments. It was not necessary to predict exactly the ship self-propulsion point, but the experiment was carried out at a rate of propeller rotation as close to that point as possible. The required rate of shaft rotation was estimated from the results of the resistance in ice experiments and the open water overload experiments by equating the tow force to the resistance in ice. The shaft revolutions were set and the model was towed through the ice sheet. Values of thrust and torque in ice were measured on each shaft, together with tow force and shaft revolutions. The total torque was analyzed to determine the mean value for each ice condition, relative to the open water value determined above.

Video records of four views of the model were made of all experiments in ice. These views covered underwater, bow and stern, and above water bow and beam views. The underwater views are necessary for observing ice flow around the hull and through the propellers.

### 3.4 PERFORMANCE IN OTHER ICE CONDITIONS

Resistance and propulsion experiments in brash ice were carried out in a similar manner to those in level ice and pack ice. Initial concentration of ice floes within the channel was approximately 95%, which was the same nominal value as the heavy pack ice condition. First the model was towed at three speeds through the channel filled with ice floes, and resistance was measured. Friction tests were carried out, the propeller was fitted, and propulsion experiments were carried out to obtain the level of propeller-ice interaction at the same three speeds.

Results were presented in non-dimensional form, of $C_p$ against $Fn_p$, where the coefficients have the same definition as for the analysis of resistance in pack ice, with ice concentrations estimated from photographic records of the brash ice before testing. Power equations were fitted to the data and these equations were used to predict the resistance values at which the open water overload data were interpolated.

## 4. MODEL-SHIP CORRELATION

The primary objective of carrying out model tests in ice is to make realistic predictions of the performance of the ship in the expected full-scale ice conditions. This requires the measurement of the same ice properties and ship performance data for the ship as were measured for the model. Ship performance parameters can be measured using the same approaches as those used for open water performance measurement [6]. Propeller shaft torque can be measured by either strain gauges on the propeller shaft, or more complex Acurex torsion meters fitted to the shaft. Rotation rate can be measured by tachometers fitted to the propeller shaft. Thrust can be obtained from thrust blocks in the propeller shaft or from strain gauges oriented for thrust rather than torque. Ship speed is commonly measured by differential GPS.

Measured ice properties are an essential part of the model-ship correlation process. Ice thickness can be measured directly by drilling holes along the projected track of the ship, using either a trial party deployed on the ice, or by means of an automated auger system deployed using the ship's crane. The disadvantage of this approach is that ice thickness is only available at the specified measurement points. A continuous record of ice thickness can be obtained from a video view of the ice pieces turned on their side as part of the breaking process. This view must be calibrated using a grid of known dimensions at the level of the unbroken ice sheet. The temperature and salinity profiles of ice core samples are used to obtain the estimated values of flexural strength. An alternative method is direct in-situ measurements of flexural strength using a cantilever beam test, similar to the one used in the model basin, but this is much more time consuming and expensive to use. A photograph of a trials team in action is given in Figure 4.

Figure 4: Trials team making full-scale ice properties measurements

Table 1: Summary of Ship Dimensions in Model-Ship Correlation Studies

| | CCGS type 1200, R-Class medium icebreaker | USCGC Healy |
|---|---|---|
| Length, O. A (m) | 98.2 | 128.0 |
| Beam (m) | 19.1 | 25.0 |
| Draft (m) | 7.2 | 8.9 |
| Displacement (tonnes) | 7,800 | 16,000 |
| Power (MW, total) | 10.14 | 11.2 |
| No. of propellers | 2 | 2 |
| Diameter[1] | 4.12 | 4.70 |
| P/D | 0.775 | 0.775 |
| Direction of rotation | Outwards | Outwards |
| Service speed, open water (knots) | 16.2 | 17.0 |
| Model scale | 1:20 | 1:23.7 |

[1] The same propellers were used on both model hulls. The linear scale of each hull was changed to match the required diameter for the ship.

Since there have been relatively few icebreaking ships built in North America in recent years, IOT has attempted to take a rigorous scientific approach to modeling and model-full scale correlation. The most recent studies comparing the results of model tests in ice, using the methods described above, with full scale data are given in [7] and [8].

In both cases, the ships for which full-scale data were available were government owned icebreakers. The principle particulars of each ship are given in Table 1. The advantage of using this type of ship is that there are typically very extensive acceptance trials, including many more data points for ice conditions and ship performance data than would be typically obtained for a commercial merchant ship.

The conclusions from these studies were that a hull-ice friction coefficient of 0.05 gave acceptable correlation between model predictions and full-scale measurements for cases when the hull is in good condition (typically freshly painted) and there is negligible snow cover. In cases where the hull roughness has increased above this level, or when the snow cover is significant this coefficient should be increased to 0.065. At model scale, changing the hull-ice friction coefficient from 0.03 to 0.09 resulted in doubling the delivered power required to propel the ship, and illustrated the importance of maintaining a low value of this coefficient.

The very nature of the material properties of ice (at model and full scale) results in much more uncertainty in measurements compared to traditional hydrodynamic testing. The flexural strength of ice is very sensitive to variations in thickness and tempering temperature, as well structural imperfections within the ice sheet. As a result, uncertainty in full-scale measurements is expected to be within 15% and measurement at model scale within 8% [7].

## 5. PERFORMANCE PREDICTIONS FOR FIVE TANKER DESIGNS

The detailed descriptions of the development of the hull designs used for illustration have been given [2, 3, 4]. A summary of the ship particulars is given in Table 2. The predictions of ship resistance in level ice against speed are given in Figure 5 for ice 0.75m thick and Figure 6 for ice 1.4m thick. Delivered power predictions are plotted against speed for ice 1.0m thick in Figure 7. Figure 8 shows resistance in pack ice against speed, at 95% coverage, for ice 1.0m thick. Figure 9 shows delivered power in pack ice against speed, for 95% coverage at a thickness of 1.0 m.

These results are helpful to determine which hull features result in the lowest resistance and delivered power. It is particularly important to note that the lowest resistance in ice need not necessarily result in the lowest delivered power. The amount of ice broken by the bow of the ship that interacts with the propellers is a key factor in determining the delivered power. The Aframax tankers were relatively shallow draft, which resulted in the propellers being close to the surface, and as a result, there was a high degree of ice contact with the propellers. The Suezmax tanker for heavy ice had a deeper draft, and as a result the propellers could be further below the water surface, and as a result avoid ice contact. The single screw ship with a bulb, had relatively high ice resistance, but the bow shape was very effective at deflecting the ice away from the propeller.

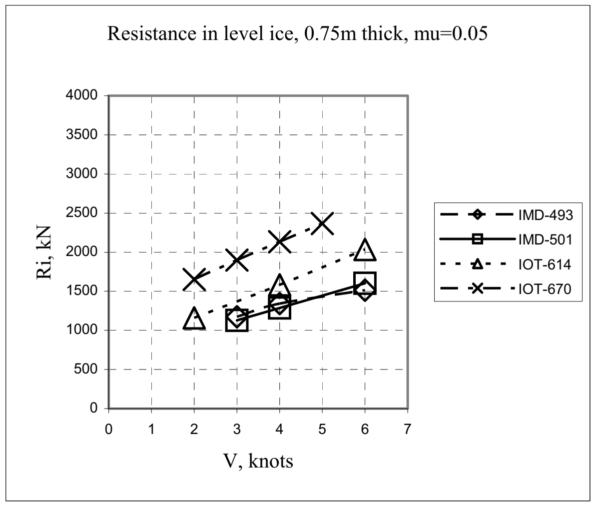


Figure 5: Comparison of resistance in level ice, *Hi*=0.75 m, $\mu$=0.05

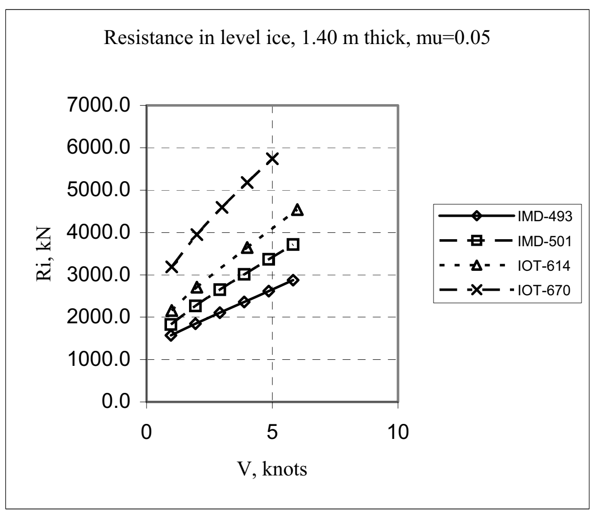


Figure 6: Comparison of resistance in level ice, *Hi*=1.4 m, $\mu$=0.05

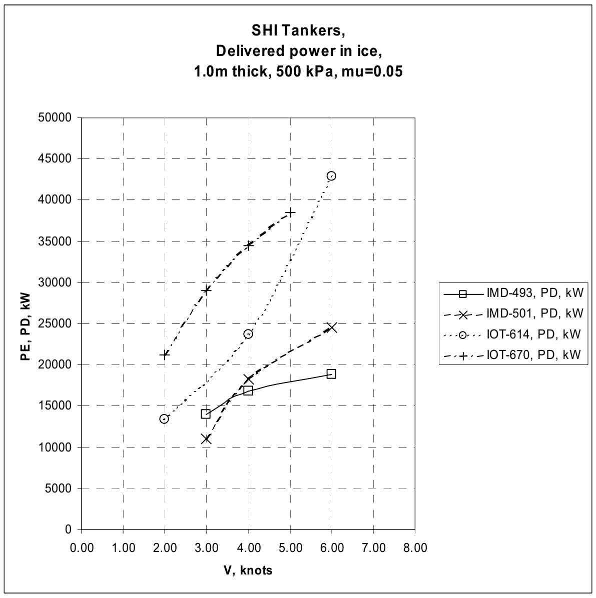


Figure 7: Comparison of delivered power in level ice, *Hi*=1.0 m,  =0.05

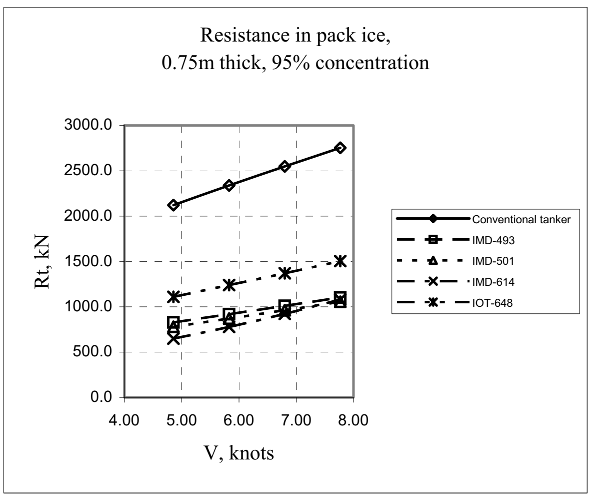


Figure 8: Resistance in 0.75m pack ice, 90% concentration

There are some areas where the modelling process could be improved. The correlation studies discussed above for government icebreakers were based on results for a model scale of approximately 1:22. The tanker models were on average a scale of 1:34. The material structure of model ice does vary with ice thickness. Model ice has a layer of small crystals close to the surface, with longer dendritic crystals growing downwards into the water. The thickness of the layer of small crystals is not a constant proportion of the ice thickness, and tends to be a greater percentage of the total thickness for lower ice thicknesses. This may have some effect on the most appropriate value of the hull-ice friction coefficient. However, obtaining full-scale trial data from an oil tanker is the key element in this evaluation.

Further study of the concept of the pre-sawn resistance experiment is also required for unconventional icebreaking hull forms. In the pre-sawn experiment, it is assumed that the ice has zero strength, and that no breaking of the ice occurs. In the case of the bulbous bow, the pre-sawn ice floes were clearly breaking on the upper surface of the bulbous bow. The magnitude of this effect may be reduced if the size of the ice floes is reduced.

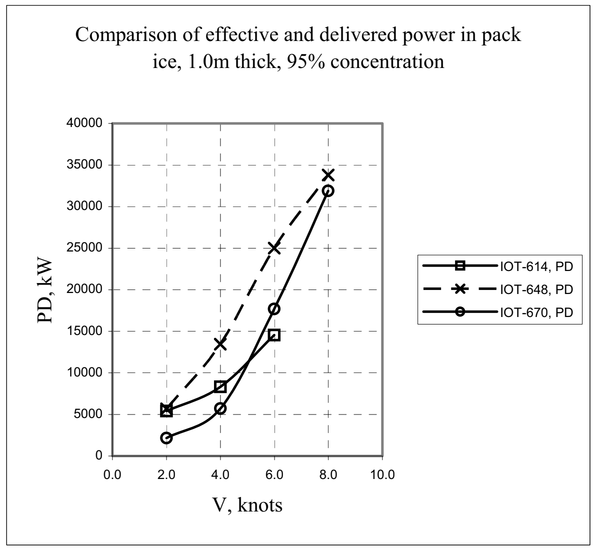


Figure 9: Delivered power in pack ice

## 6. CONCLUSIONS

The results of this research have been extremely important to SHI's strategy to become the world leader in the construction of large icebreaking merchant ships. Model testing has been an essential element of this strategy, since it is the opinion of the authors that at the present time analytical methods are not sufficiently well developed for accurate performance predictions.

The results of the research have shown that for large tankers in ice:

i) Quite different bow shapes can result in similar resistance in ice, once allowances for hull-ice friction coefficient and ice thickness have been included.

ii) Icebreaker designs have the lowest resistance in level ice and pack ice. Such a design is characterized by a raked bow with a long overhang. This type of bow is effective at breaking the ice, and directing the broken pieces around the hull. However, this type of bow has relatively poor performance in open water.

iii) Bulbous bows in ice have distinctive properties, compared to conventional icebreaker bows. The bow shape results in a lot of secondary breaking where the ice floes come into contact with the upper surface of the bulb. When the ice is already broken before it comes into contact with the ship, this penalty is removed.

iv) It is possible for a ship with a bulbous bow to be effective in light ice conditions, especially pack ice. The ice breaking performance is clearly much worse than a bow designed for heavy ice, but the improvement in open water performance compensates for this. It may be particularly effective in an area with extensive icebreaker support.

## 7. ACKNOWLEDGEMENTS

The methods of model testing in ice that are described in this paper have been developed at IOT over the last 25 years. Many people have contributed to this work over the years, but some deserve a special mention. Dr. Bruce Colbourne, Dr. Stephen Jones, Dr. Bruce Parsons, Mr. Don Spencer and Dr. Mary Williams have all made extensive theoretical and experimental contributions to the understanding of ice mechanics and ship-ice interaction at model scale and full scale. The technical staff at IOT have developed and refined the methods for modelling the ice and the ship. Mr. Brian Hill is singled out for his expertise in the management of IOT's model ice making capability and Mr. Craig Kirby for his expertise in making full-scale measurements.

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## 9. AUTHORS' BIOGRAPHIES

**David Molyneux** is a Senior Research Officer at the Institute for Ocean Technology in St. John's, Newfoundland, Canada. He obtained his Master of Applied Science Degree from University of British Columbia, and his Bachelor of Science Degree in Naval Architecture for University of Newcastle-Upon-Tyne. Since 1987, he has carried out research into ship performance in ice, for private and public sector organizations from Canada, United States, Germany and Korea. All of these projects have included model experiments as part of the performance prediction process. During the course of carrying out these projects, scientific ship modelling techniques for predicting powering and manoeuvring performance in a range of ice conditions have been developed and refined.

**Hyun Soo Kim** is a Senior Engineer at the Marine Research Institute of Samsung Heavy Industries Co. Ltd., Korea. He obtained his Ph. D. of Naval Architecture from Pusan National University, and his Master and Bachelor Degrees in Naval Architecture for the University of Inha. Since 1991, he has performed research on ship hydrodynamics. His major research fields included sloshing analysis, the prediction of ship maneuvering performance, sea trial measurement and analysis and ice technology to predict power of the icebreaking vessels. Recently his interest has included the subject of risk management.

## 10. APPENDIX

Table 2: Summary of Principal Dimensions, Tanker Designs

| Design | | Aframax Arctic 1 | Aframax Arctic 2 | Suezmax, Arctic | Suezmax No ice | Suezmax Baltic |
|---|---|---|---|---|---|---|
| Bow shape | | R-Class #1 | R-Class #2 | Spoon | Bulb | Ice bulb |
| Propulsion | | Twin gondola | Twin gondola | Twin shafts | Single | Single |
| Model number | | IMD-493 | IMD-501 | IMD-614 | SM 173 | IOT-648/ IOT-670 |
| Length, wl | m | 273.5 | 274.9 | 284.0 | 258.3 | 271.48 |
| B, wl | m | 43.6 | 43.6 | 42.8 | 46.2 | 44.0 |
| T, midships | m | 11.5 | 11.5 | 16.5 | 16.6 | 15.0 |
| Displacement | tonnes, SW | 100144 | 102145 | 161935 | 162001 | 145699 |
| Wetted area | sq. m. | 14720 | 14502 | 17689 | 17492 | 16746 |
| Propeller Diameter | m | 6.60 | 6.60 | 6.72 | 9.80 | 8.10 |
| Model scale | | 31.94 | 31.94 | 33.87 | 44.5 | 36.82 |

# TECHNICAL DEVELOPMENT OF LNG CARRIERS FOR HARSH ICE CONDITIONS

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## SUMMARY

At present times the interest for produsing gas in arctic remote areas has been increased. One alternative of transporting the gas would be the use of LNG Carriers. Last years succesfull development of highly capable and effective ice breaking ship solutions has opened an oppourtunity also for development of large size icebreaking LNG Carriers.

So far LNG transport in ice covered areas is done with small size vessels and reasonably easy ice conditions. The development of LNG Carrier capable of safe and economical operation in Arctic waters is a challenging task. By using the latest technology and recent experiences of smaller size technical solutions, Aker Arctic has built up a design and development program which resulted a higly ice capable large size (200 000m3) arctic LNG Carrier solution.

The development of the technial design included various versions and solutions for icebreaking operation. Evaluations and testing was made both for Double Acting solution, extreme icebow solution and icebreaker assisted ship version. Ship conceptual designs were created with similar main parameters and tank concepts and the test models were evaluated in the ice and open water test tanks. Furthermore the extensive simulations were carried out for finding the design alternatives capability to operate in Kara Sea ice conditions.

Different machinery and propulsion systems were evaluated including the diesel-electric, as well as steam turbine and slow speed heavy fuel engine solutions. The diesel-electric system can also be used as dual-fuel system using environmentally friendly gas as prime fuel. Environmental issues in the sensitive Arctic areas has a special importance in the design solutions. As propulsion alternatives the fixed pitch, controllable pitch and electric azimuthing propulsion systems were studied with different propeller diameters and different number of propellers.

The preliminary estimates and calculations also for the ship hull integrity in cold environments were done, as well as evaluation of different containment system suitability and risks involved in the arctic and North-Atlantic voyages.
Testing and analysis of this systematic developemnt has been completed in Aker Arctic and has resulted for Arctic capable large size LNG Carrier design including the novel solutions for breaking the ice and handling the cold enviroment challenges.

## 1. INTRODUCTION

Technical development both in conceptual designs and main propulsion systems lately has encouraged ship designers and operators to look new transport opportunities in Arctic harsh areas. Good examples of this development are multipurpose icebreakers, stern first icebreaking principles (Double Acting) and trend for ever larger vessels in ice class (tankers at Primorsk). After the development of the first Aframax size tankers (MT Mastera and Tempera) with individual icebreaking capability and good operational experiences, it was noticed the benefits and safe operation of big tankers in ice conditions. Furher experience gained from MV Norilsk Nickel, vessel operating in Kara Sea year around built to high Russian icelass LU7, have showed the usability of the technical solutions. These full-scale experiences and measurements has encouraged Aker Arctic Technology Inc. to further develop same principal solutions towards the LNG transport from northern Russia gas deposits. Idea is based of combining the existing large size vessel in Baltic with the high ice class vessel in Kara Sea, towards a new solution of icebreaking LNG Carrier design.

## 2. DESIGN CHALLENGES

### 2.1 CONCEPT SELECTION

The first main decisions when making a design for any Arctic transport is to choose the basic concept of the vessel’s icebreaking principle. The vessel could be designed for the operation combined with icebreaker fleet actually breaking the ice and making way for the cargo vessel. Alternatively the cargo vessel can be designed with high power, special hull form and suitable propulsion system for making icebreaking mainly by her own. However, real life practice has shown that not too many vessel’s are still able to comply with the most difficult ice conditions with ice compression and ice ridging. To obtain icebreaker like capabilities in the cargo vessel the Double Acting icebreaking principle has to be utilised. These three main categories vary both technically and therefore price wise and makes the concept selection important decision for future prospects. The development program done was to show the technical and economical feasibility of these three main concept solutions.

## 2.2 MAIN DESIGN CRITERIA

Each of the concepts developed in the program had the common basic criteria, which is following:

- Arctic LNG Carrier for operation in Kara Sea.
- Russian ice class level LU7.
- Cargo carrying capacity about 200 000m3.
- Year around traffic
- North Atlantic crossing
- Offshore loading capability
- LNG tanks type A or B

## 2.3 HULL FORMS

Hull form of the concepts must be different in order to meet the icebreaking requirements of the solutions. The truly icebreaking solution called Concept A in this paper, has a bulbous bow for efficient open water sailing and specially formed stern shape for icebreaking operation with the azimuthing propulsion devices. Concept B which is of type icebreaking vessel, has typical arrow shaped icebow form and stern shape with twin shaftline propellers and twin rudders. Concept C is vessel with the idea of using icebreakers for making the way through ice fields. This vessel has bulbous bow and twin shaftline propellers in aft.

Main dimensions of the versions are:

| | |
|---|---|
| Loa | 340 m |
| Lpp | 324.90 m |
| Breadth | 50.00 m |
| Depth | 22.90 m |
| Design draught | 12.00 m |
| Gross tonnage | abt. 133,000 |
| Cargo capacity | 206,000 m3 |

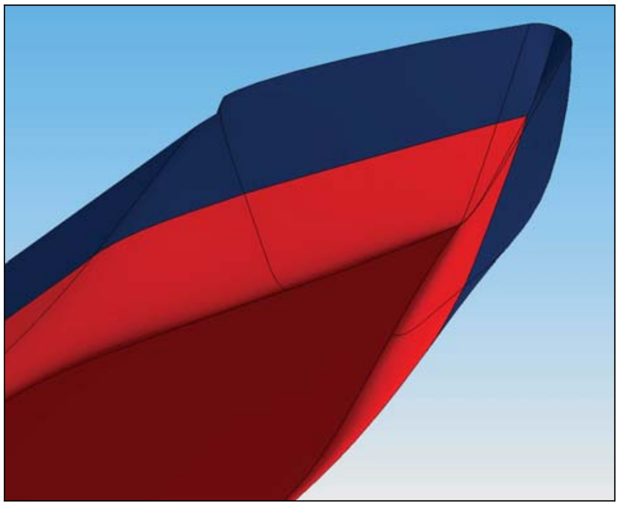

Figure 1: Form of the Concept B icebow

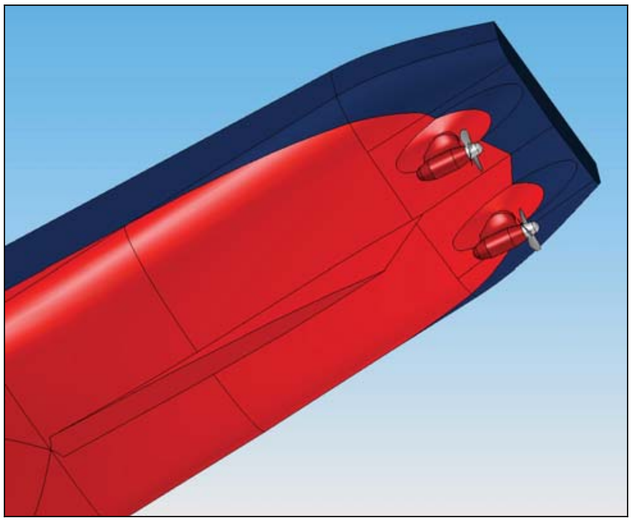

Figure 2: Form of the Concept A icebreaking stern

## 2.4 PROPULSION SYSTEMS

In the design criteria vessels route is determined to cross North Atlantic and thus the operation time in ice free sailing is rather dominant. This requirement has impact both on machinery selection, tank type selection and later final decision on concept selection. Also operational requirements in icebreaking or ice navigation in general have importance in the machinery choice. In LNG vessels one philosophical choice is whether to use boil-off gas for propulsion machinery or not. In this case it was evident that gas should be utilised as prime mover fuel. Then the main selections of the machinery solutions are:

- Diesel electric (gas driven)
- Steam turbine
- Gas turbine

For icebreaking concepts the dynamic behaviour of the propulsion system requires flexible machinery solution. Power must be easily adjustable in the ice navigation when at the same time ice contacts into propellers causes rapid changes in propeller rpm. This is the main reason why icebreakers have diesel-electric propulsion systems. Also the requirement to have a high power at low speed and rpm results high torque. All these facts make the electric propulsion an attractive choice. More freedom is available in the selection of electrical power plant system. Today’s trend in some LNG Carriers is to use gas driven dual-fuel medium speed diesel engines. This is fuel consumption and environmentally efficient selection with good efficiency. Also flexibility to run engines both with HFO oil and boil-off gas results this selection for the electric power plant.

The re-liquefaction solution could be considered as alternative. Biggest motivation for this is to reduce the service speed and thus gain savings in fuel cost. For icebreaking vessel the high power is preferred and thus this option did not prove to be optimum for Arctic trade.

### 2.4 CONTAINMENT TYPE

A lot of discussion about the most suitable containment selection for icebreaking LNG Carriers have arisen. Concerns have been set for extensive vibrations and deceleration of the vessel when hitting ice. Also other thoughts of sloshing in the North Atlantic route have been in favour of Type B containment system.

Some concerns related to ice navigation have been studied in the development program.

The practical decelerations which the vessel might obtain in the navigation in ice have been looked both by simulations and model testing methods. The simulations were run with the vessel hitting most severe ice ridges what have been measured in the Kara Sea. Simulations indicated the obvious result that the acceleration values are clearly on the safe side. The dimensioning accelerations from wave conditions are nearly 10 times higher than the ones obtained in the ice. Also in the model testing the vessel was rammed directly to consolidated ice ridges with the speed up to 12 knots. The maximum measured deceleration was 0.11 m/s2.

Typically breaking of the ice causes low frequency vibrations to the ship hull. In the development of the ships, which are larger than any ships before with practical operation in heavy ice conditions, there are not much tools available to estimate this behaviour. However, with help of previous experience the new hull forms are developed in a way the ice induced vibration is minimised. Future work regarding the vibration issue is to make full scale measurements on the Aframax size oil tankers in the Gulf of Finland.

Based on the available information the ice operation does not have any restriction of using the membrane type containment system. The selection of using the Moss-type tanks in this design is based on more sloshing safe design in wave conditions, mainly for North Atlantic trades.

## 3. RESULTS

### 3.1 TANK TESTING FOR OPEN WATER

Tank testing of the hull versions were carried out in the VTT model basin in Espoo, Finland. The models were equipped with scaled Azipod units. The test program consisted:

- Resistance tests
- Open water test for propeller
- Open water tests for Azipod unit
- Wakefield determination
- Tuft tests
- Propulsion tests

Main objective of the tests was to find out the required propulsion power of the models. The hull forms were unconventional and therefore not well predicted. Before the testing and drawing of the final hull lines several CFD calculations were carried out with potential flow method. This was mainly used to make the bulb form well working both for ice and still water. Also the icebreaking bow form was optimised for wave making behaviour.

Resistance test revealed expected values for Concept hull A (DAT Carrier), but showed clearly higher values for hull B than indicated by CFD analysis. Resistance at design speed of 19.5 knots was 16% higher than for hull A.  As the wetted surface of the models are nearly the same the increase was in residual resistance. Visually could be obtained clearly larger waves from the bow region. This is illustrated in Figures 3 and 4.


Figure 3: Waves of the Concept A

Figure 4: Waves of the Concept B icebreaking bow

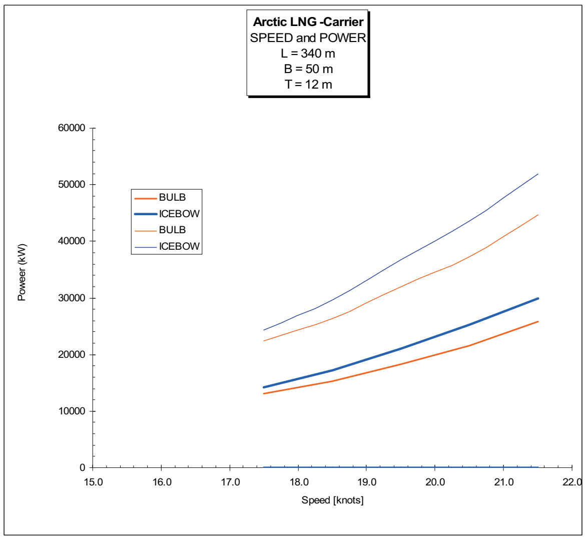


Figure 4: Resistance and effective power difference

Propulsion test shoved low thrust deduction values for podded propulsion system, but at the same time also low and smooth wake of the hull. This will obviously indicate lower pressure pulses from propeller and good comfortable level. Hull efficiency is not particularly good but is due to special forms required for icebreaking duties in the stern.

### 3.2 TANK TESTING IN ICE

Tank testing of the hull versions were carried out in the Aker Arctic Technology model basin in Helsinki, Finland. The same models than used in open water were used in ice tank. The test program consisted:

- Propulsion towing force tests without ice
- Self propulsion tests in ice
- Resistance can be derived from above tests

Different type of ice conditions were tested:

- Unbroken level ice 120cm
- Unbroken level ice 150cm
- Unbroken level ice 170cm
- Ridged ice 12-15m thickness with consolidation
- Broken ice to simulate icebreaker assistance

Set point for icebreaking criteria in level ice was to achieve 5 knots speed in 150cm ice thickness. The nominal power (100%) at the model setup was corresponding to 36 000kW at propeller shaft. In this case obviously 2 x 18 000kW as the model had twin Azipod propulsion system. The propulsion tests at different ice thicknesses were carried out at different power levels corresponding 80%, 100% and 120%.

Model A with running the vessel in Double Acting mode stern first to ice proved to overcome the initial requirement. The results showed the vessel was able to reach 5.4 knots speed in 1.5m thick unbroken ice sheet at 100% nominal power. Flexural strength of the ice was corresponding to 500 kPa in full scale.

Model was also tested bow ahead and with bulb the vessel was able to break 70cm of ice at 5 knots speed. The icebreaking capability ahead with the bulb is very good comparing to standard bulb design.

The Model B was not able to meet the initial performance criteria despite the icebreaking bow form worked well. The speed achieved in 1.5m ice was only 0.5 knots. All the models were equipped with 36MW total propulsion power.

In the test tank a ridged ice conditions were simulated with a long one uniform thickness ice formation. The length of the ridge was nearly one ship length and dept varied from 11-13 metres in different tests. Model A with Double Acting icebreaking operation managed to go through the ice ridge slowly with average speed 0.5 knots. Icebow and bulbous bow versions stopped in the ridge at least once. Additional ramming was required. However, the big vessel is quite capable to pass through even thick ridges of 13-15 metres when they are shorter than the ship length, which is typically the case in nature. It is expected that vessels can operate quite long periods in ahead mode until the thickness and concentration are getting heavier during the winter.

### 3.3 ICEBREAKING SIMULATION

In order to model more complex ice conditions than level ice and individual ridges the simulation technique can be used. With the rather straightforward method the simulation tool can model the propulsion system, icebreaking forces, hydrodynamic forces and operational commands. Simulation model can not handle manoeuvring of the ship.

So called ice profile is created which represents typical conditions in the studied sea area. In this case the seas of Eastern Barents Sea, Kara Strait and Kara Sea. Ice profiles for each of the sea were created based in historical information and latest ice charts from few years back.

Then the resistance values measured in the tank testing were implemented on the simulation model and vessels were sailed through the sea areas with ice profiles representing different months of the winter. As a result the vessel operational speed was determined and the fuel consumption could be derived.

Based on the simulation it is expected that average speed through the Kara Sea will be minimum 4.0 knots in April/May period. However, this simulation does take into account the ice compressions which may stop the vessel for few hours at a time, thus resulting slightly lower overall average speed.

## 4. FUTURE WORK

During the research work several issues came up which would need further development and solutions. One of the main importance was the power requirement to reach the service speed. Initial power was not sufficient and there will be need to have more power in the vessel. More power off course also benefits the icebreaking capability. It was envisaged that 20MW propulsion units would be suitable for this size of the vessel.
Other future studies are listed below:

- Optimisation of the hull form
- Follow-up work with reference vessels
- Handling of the load variations in power plant and propulsion drive system, sensitivity of DF engines
- Ice induced vibration effects
- Ice deflections in the hull versus tank system
- Offshore loading in Arctic environment

## 5. CONCLUSIONS

The main benefits of this development was to confirm the oil/gas industry that shipping solutions are existing today even for harsh ice conditions. The initial development of the main concepts is done with an extensive testing program. Some challenges are still unresolved, but based on existing vessels these can be considered to overcome.

The calculations and simulations indicate the LNG Shipping from remote Arctic areas can be done with cost efficient way, without risking the environment. Modern diesel technology provides economically friendly opportunity for arctic operations.

During the winter time the speed drop due to ice can be kept on such a level that over numbering the fleet just a little, the constant deliveries on LNG can be operated.

Icebreakers are mainly needed nearby the terminals to keep the port and loading operations running smoothly. By utilising the DAT technology there is no need for constant escort icebreaking. It could be considered to position one strong icebreaker in the Kara Gate area to secure the passing of the strait during heavy ice compression periods.

Feasible solutions were created, main challenges solved for ship concept level technology and next step would be further develop the operational issues especially regarding the offshore loading in Arctic extreme coldness.

## 6. AUTHORS' BIOGRAPHY

**Reko-Antti Suojanen** (born in 1969, Naval Architect M.Sc.) has worked in marine and computer technology organisations. His positions in marine organisations have been a concept designer, researcher, and project manager. In computer technology organisations his positions was a CFD application specialist. He started his work career 1996 at Kvaerner Masa-Yards Turku Shipyard as a hydro dynamist and project engineer. Later he served the Finnish IT centre for science (CSC), governed by the Ministry of Education. After that he joined the team on Masa-Yards Arctic Research Centre (MARC) and started to work as an arctic marine engineer and a project manager. He has been working in numerous projects in the arctic shipping area. In the beginning of 2005 he joined newly founded company Aker Arctic Technology Inc. where under his responsibility is running the arctic consultation group. The main activities have been in the shipping operational, technical and economical consultation for the arctic and sub-arctic waters for both the governmental and company customers. He has also published in international journals and given presentations in international maritime conferences.

*Design and Construction of Vessels Operating in Low Temperature Environments, London, UK.*

# STRUCTURAL DESIGN OF HIGH ICE CLASS LNG TANKERS

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**B-J Noh**, Hyundai Heavy Industries, Korea

## SUMMARY

The future development of LNG reserves in remote Arctic and sub-Arctic areas will require a new generation of highly ice-capable LNG tankers, capable of providing continuous throughput of gas at all times of the year. These ships will need to be able to travel faster in heavy ice than all but the largest icebreakers, which poses challenges for both hull and machinery design. The American Bureau of Shipping (ABS), BMT Fleet Technology Limited (BMT) and Hyundai Heavy Industries (HHI) are currently undertaking a joint research project aimed at addressing these design challenges. This involves risk assessment of operational scenarios, and the development of methodologies for direct calculation of loads on all areas of the hull. The project is also addressing the need for new techniques for the analysis of the outer hull, double hull and cargo containment systems of these ships under design and accidental loads; areas in which 'rule design' can only provide a starting point. This work presents some of the results to date of the research project, and highlight their significance for the design and operation of future ships.

## NOMENCLATURE

| | |
|---|---|
| $KE_e$ | effective kinetic energy |
| $IE$ | indentation energy |
| $PE$ | potential energy |
| $F_n$ | normal force |
| $\zeta_c$ | indentation depth |
| $A$ | contact area |
| $A_n$ | normal contact area |
| $P_{av}$ | average ice pressure on A |
| $ex$ | ice pressure area exponent |
| $f_A$ | ice contact geometry function |
| $d$ | ice contact function parameter |
| $P_o$ | ice pressure index |
| $R$ | ice edge radius |
| $\phi$ | ice wedge angle |
| $\theta$ | ice eccentricity angle |
| $\beta'$ | contact plane angle |
| $w$ | width of ice contact |
| $h$ | height of ice contact |
| $h_{ice}$ | height of ice |

## 1. INTRODUCTION

Demand for oil and gas are making frontier area reserves increasingly attractive. Estimates suggest that around 25% of undiscovered global hydrocarbons are located in Arctic areas such as Russia, Canada, and Alaska. The distances to market and resulting cost of pipelines makes marine transportation an attractive option for many fields. Figure 1 illustrates several potential LNG shipping routes and projects.

Ice-capable oil tankers already exist, and there is some experience of their operation in arctic conditions. However, to date no arctic-capable LNG tankers have been built, and there is very limited experience with sub-Arctic LNG. LNG tankers are quite different from crude carriers; in terms of form, containment system, and preferred operating speeds – speeds typically being relatively high. Before safe designs can be developed, research is needed into the quasi-static and dynamic loads that they may need to withstand.


Figure 1: Potential LNG Shipping Routes.

## 2. DESIGN CHALLENGE

Obviously the primary challenge in the arctic is ice. A related challenge is the cold, as are darkness, great distances and lack of infrastructure. The present discussion will focus primarily on matters of structural capacity to resist the ice loads. There are two structural aspects that require study. The ship hull itself, including the outer hull, the inner hull and the hull girder integrity, constitutes the primary structural system. As well, there is a complex cargo containment system, designed to keep the LNG cargo, at -160° C, from coming into contact with the steel hull of the ship. The containment system is multilayered and must not be (permanently) deformed in the event of contact with ice. Figure 2 sketches a cross section of an LNG vessel (membrane type).

Present ice-going ships are typically designed to a standard ice class. In most cases, similar ships with similar ice class are available for reference. The ice classes themselves are strongly based on the experience from existing vessels. Ice-going LNG Ships are unusual in several ways. Their size is well above the range of most ice-going ships. Operating speeds may also be higher than normal. These two issues add uncertainty to the loads. Existing ice classes, while size and speed dependent, have not been verified by experience with ships like high-arctic LNG ships. Another key issue is the LNG containment structure. The containment structure is crucial to the safety of the vessel, and any potential impact from ice operations must be carefully studied. For these reasons, there is a need for a careful appraisal of the ice strengthening needs of ice-going LNG ships. The aim of the project described here is to develop the appropriate loads models and analysis tools to allow the development of a safe and effective high arctic LNG vessel.

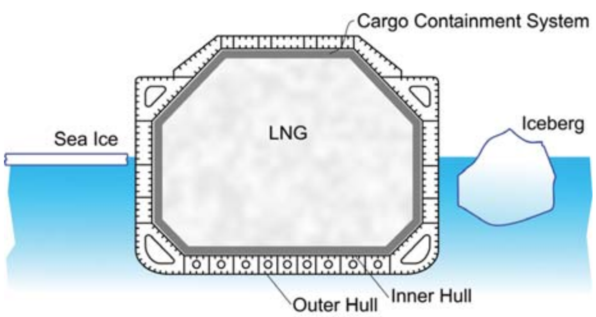



Figure 2: X-section of ice-worthy LNG Vessel.

## 3. ICE CONDITIONS

### 3.1. BALTIC SEA

The Baltic route from a hypothetical terminal near Primorsk in Russia, through the southern Baltic to the North Sea and beyond is already very widely used, though not by LNG traffic. The special nature of the Baltic may raise safety and political issues which preclude it from being used for LNG. However, as Baltic ice transit is “routine” it is well worth considering the challenges of an ice-going LNG ship in this area. Baltic ice is all first year, and the level thickness in the Gulf of Finland and southwards rarely exceeds 70 cm. The extent of the ice in the Southern Baltic varies year to year, but the typical icebreaking distance in the worst winter month (February) would be in the order of 150 nm. Tides in the Baltic are small, and currents are relatively weak. Ridging and pressure are therefore generated mainly by wind. The proximity of the shoreline means that pressure events and ridging are relatively frequent.

### 3.2. YAMAL PENINSULA, KARA SEA

A route from the Yamal Peninsula in Russia through the Kara Gate at the south end of Novaya Zemlya island and on out through the Barents Sea will normally encounter mainly first year ice, with some multi-year incursions in the Kara Sea. The first year ice can reach thicknesses approaching 2 m, and the shallow water in much of the Kara Sea can generate grounded ridges. Pressure is most likely around the Kara Gate, where tidal currents are the dominant factor. In the worst month of the year (May) the total distance in ice will be in the order of 850 nm.

There is also a Northern route around Novaya Zemlya which has approximately the same distance, but with much greater risk of encountering the predominantly multi-year ice of the polar pack. When the Arctic gyre brings this towards the Russian coast, there is also likely to be high ice pressure near Novaya Zemlya. Conversely, when the pack moves away, conditions can be lighter than the southern route due to the smaller number of freezing degree days away from the mainland. Henry Hudson, later to perish in the Canadian Arctic, tried to venture north of Novaya Zemlya in 1607. He was turned back by the polar pack that had come south that year. It must have been very challenging in a small wooden ship. Although chartered to try again the next year, he used his ‘initiative’ and went instead to Manhattan.

The western part of the Yamal route passes the Shtokman field. This area is sheltered from the polar pack by a series of island archipelagos, and sees lighter winter ice for a shorter season than in the Kara Sea. Conditions vary considerably year by year, and the location of the ice edge can change quickly under wind effects. Typically, the distance from the Shtokman development area to the ice edge in the worst months of April-June will be in the order of 200 nm in maximum ice thicknesses of 1.2 – 1.5m.

### 3.3. CANADIAN ARCTIC AND EAST COAST

There is a large gas discovery on Melville Island in the Canadian arctic. The route east through the Northwest Passage towards the Eastern Seaboard of North America contains first year, multi-year and glacial ice; on average the most difficult of any of the routes under consideration, with the longest total distances in ice of up to 1200 nm in April-June. The best route towards Baffin Bay will change year-to-year, as the concentrations of multi-year ice in any of the straits varies considerably. The first year ice thickness will reach 2 m.

On the Canadian East Coast there are potential LNG routes from offshore Labrador with medium size discoveries and from the Grand Banks, where gas reserves are comparable to oil. Offshore Labrador has highly dynamic and mixed ice conditions. The main issue in the Grand Basks is icebergs. Potential terminals for all these routes include the Gulf of St. Lawrence, which has

2-3 months of its own ice cover. This light ice would not be a serious issue for vessels capable of dealing with the conditions further North.

### 3.4. CENTRAL ARCTIC

Global warming, and the diminution of the polar pack, have led to a number of proposals for trans-polar shipping routes. One of the most practical of these may be from Eastern Siberia to the Western North American Arctic, where pipelines would take gas into North America. As can be seen, the sea distances are much shorter than those for several other services, and the conditions are no worse. While the polar pack is still relatively thick, it is rarely under high pressure in the mid-Arctic areas, and routings through thinner ice can generally be found using advanced ice navigation techniques.

### 3.5 SAHKALIN

A large set of offshore developments with associated transportation requirements is currently taking place around Sakhalin Island in the Russian Far East. Conventional LNG will soon be shipping from Aniva Bay at the south end of the island to markets in Japan and India. This gas comes from the Sakhalin 2 projects offshore mid-North Sakhalin, transported south by pipeline. Future developments are planned for further North (Sakhalin 5) where different approaches to transportation may be utilized. LNG may be shipped southwards by a number of routes, including down the East Coast of Sakhalin, down the West Coast (Tatar Strait), and into the more central areas of the Sea of Okhotsk. Ice in the Sea of Okhotsk is all first year ice – it melts completely in summer, and there is no glacial ice present. Thicknesses can reach 1.5 m in March/April, and there is a 5-7 month ice season, depending on location and the severity of the winter.

## 4. ICE LOADS

To develop the ice loads, it is necessary to consider a variety of design scenarios and for each, develop an appropriate contact model.

### 4.1 ICE INTERACTION SCENARIOS

To begin, an extensive list of ship-ice interaction scenarios will be presented. Many of these interactions are well understood. Some are less well understood. Tables 1 to 4 show twenty interaction scenarios. The scenarios cover first year, multi-year and glacial ice, and a variety of ice edge geometries and ship responses. These 4 tables only cover loads from a quasi-static and strength perspective.

Table 1: Ice Load Scenarios 1 to 5

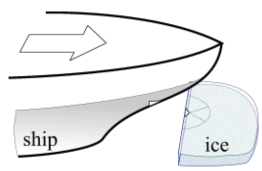
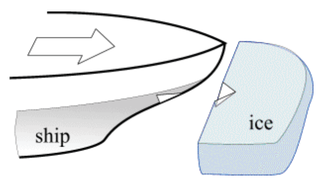
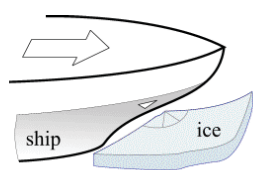
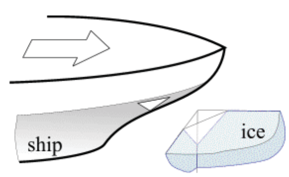
| # | Scenario | Description | Ice Type | Description | Sketch |
|---|---|---|---|---|---|
| 1 | Stem 1 | Icebreaking | Thin M/Y | This is a head-on collision with a thin ice edge. This involves either Contact Case #1 or #2, limited by flexural failure. |   |
| 2 | Stem 2 | Ramming | Thick M/Y | This is a head-on collision with a thin ice edge. This involves either Contact Case #1 or #2, limited by either Vn=0(initial impact) or Vs=0(beaching). |   |
| 3 | Bow 1 | Icebreaking | Thin M/Y | This is an oblique bow collision with a thin ice edge. This involves either Contact Case #3,#4 or #5, limited by flexural failure. |   |
| 4 | Bow 2a | Glancing (Russian) | Thick M/Y | This is an oblique bow collision with a thick ice edge. This involves either Contact Case #3,#4 or #5, limited by Vn=0. This is essentially the same as Case #3 above, without the flexural limit. |   |

Table 2: Ice Load Scenarios 6 to 11

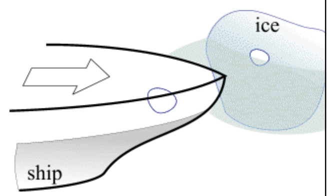
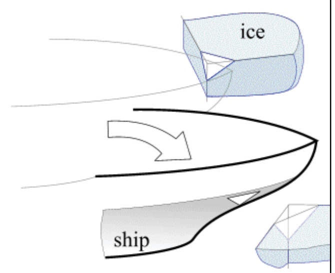
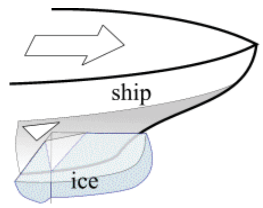
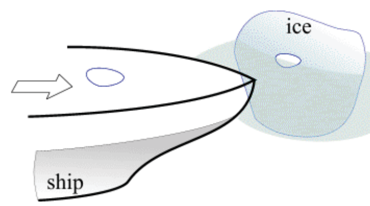
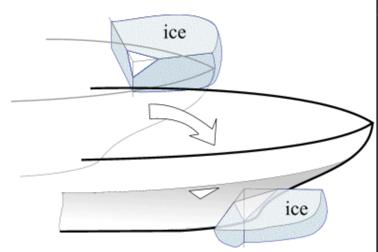
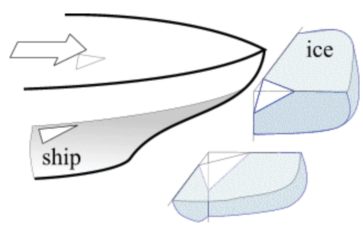
| # | Scenario | Description | Ice Type | Description | Sketch |
|---|---|---|---|---|---|
| 5 | Bow 2b | Glancing (Canadian) | Small glacial | This is an oblique bow collision with a glacial ice mass. This involves either Contact Case #8, limited by Vn=0. |  |
| 6 | Bow 3 | Reflected | Thick M/Y | This is a 2nd oblique bow collision with a thick ice edge. This involves either Contact Case #3,#4 or #5, limited by Vn=0. The case requires the specification of all 6 velocities (surge, sway, heave, pitch, roll, yaw), just prior to the 2nd impact. |  |
| 7 | Shoulder 1a | Glancing (Russian) | Thick M/Y | Similar to Case #4 above, but further aft. |  |
| 8 | Shoulder 1b | Glancing (Canadian) | Small glacial | Similar to Case #5 above, but further aft. |  |
| 9 | Shoulder 2 | Reflected | Thick M/Y | Similar to Case #6 above, but with contact further aft. |  |
| 10 | Shoulder 3 | Wedging | Thick M/Y | This is a symmetrical bow collision with two thick ice edges. This involves either Contact Case #3,#4 or #5, limited by Vn=0. This case combines elements of Cases #2 and #4 above. |  |

Table 3: Ice Load Scenarios 12 to 17

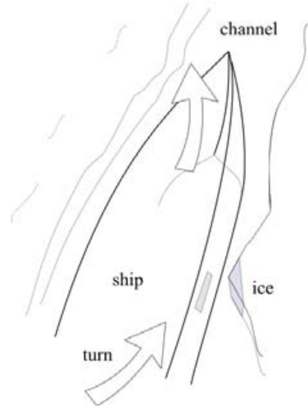
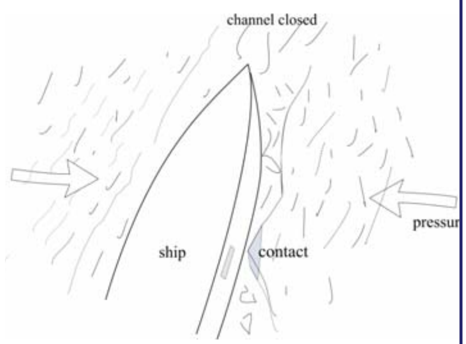
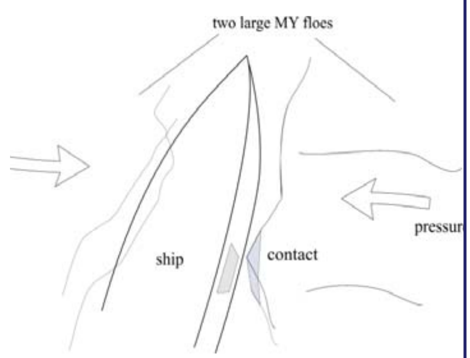
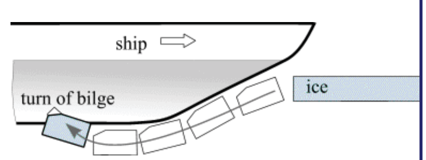
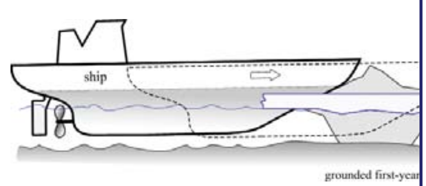
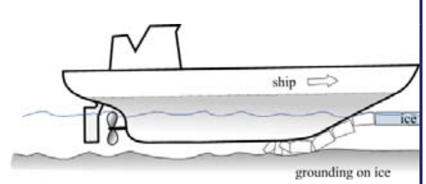
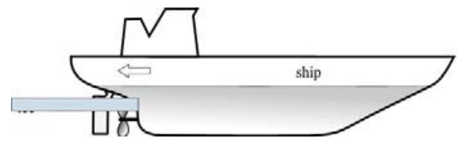
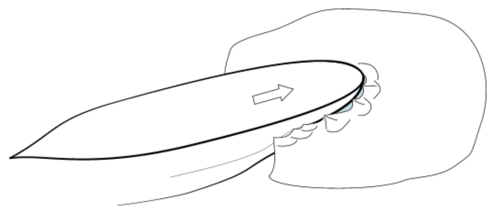
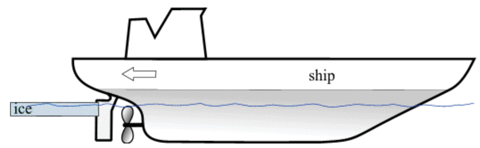
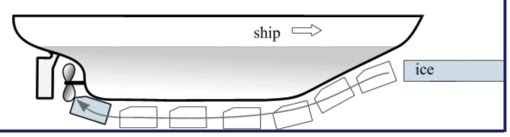
| # | Scenario | Description | Ice Type | Description | Sketch |
|---|---|---|---|---|---|
| 11 | Midbody 1 | Glancing | Thin M/Y | This is an oblique midbody collision with an ice edge. This involves either Contact Case #6 or #7, limited by Vn=0. |   |
| 12 | Midbody 2 | Pressure | Thick F/Y | This is a midbody static contact an ice edge (no kinetic energy issues). This involves either Contact Case #6 or #7, limited by either force in the ice field or contact stress on the whole ship side. The force limit requires a separate analysis. |   |
| 13 | Midbody 3 | Pressure | M/Y | This is a midbody static contact an ice edge (no kinetic energy issues). This involves either Contact Case #6 or #7, limited by either force in the ice field or contact stress on the whole ship side. The force limit requires a separate analysis. |   |
| 14 | Turn of Bilge 1 | Impact | Thin M/Y | In this case, a single block impacts the hull on the bottom or bilge. This involves either Contact Case #3 or #4, limited by Vn=0. The ice mass is quite small. |   |
| 15 | Bottom 1 | Beaching | Grounded ridge | This case involves a head-on ram into a complicated ice shape, up to an including the interaction of the bottom. This has elements of Case 2 above, but with both solid and granular ice. This case will need a new solution. |   |
| 16 | Bottom 2 | Impact | Thin M/Y | This case involves a kind of grounding on blocks of ice. This case will need a new solution. |   |

Table 4: Ice Load Scenarios 18 to 20

| | | | | | |
|---|---|---|---|---|---|
| 17 | Stern 1 | Backing | Thin M/Y | Similar to Cases #1, #3 and #4 above, but with mass and geometric parameters for the stern region. |   |
| 18 | Stern 2 | Astern Icebreaking | Thin M/Y | Similar to Cases #1 or #3 above, but with mass and geometric parameters for the stern region. |  |
| 19 | Stern 3 | Appendage Impact | Thin M/Y | Similar to Cases #1, and #4 above, but with mass and geometric parameters for the appendage. |   |
| 20 | Stern 4 | Prop. Induced | Thin M/Y | This involves a free ice block similar to Case #14 |   |

Another set of load cases will be developed to examine dynamic and vibration effects. While dynamic loads are not typically a concern for ice-going ships, the issue of the special containment system requires that this topic be given special attention, though there may ultimately be no special implications.

## 4.2 ICE COLLISION MECHANICS

The scenarios shown above each involve ice interaction. Each case depends on the geometry of the indentation (ship and ice) and the ice strength and failure mechanism. For many of the scenarios in Tables 1 to 4 the problem is one of impact between two objects. It is assumed that one body is initially moving (the impacting body) and the other is at rest (the impacted body). The solution is found by equating the available (effective) kinetic energy with the energy expended in ice crushing and the changes in potential energy (if any):

$$KE_e = IE + PE \tag{1}$$

The available kinetic energy is the difference between the initial kinetic energy of the impacting body and the total kinetic energy of both bodies at the point of maximum force. If the impacted body has finite mass it will gain kinetic energy. Only in the case of a direct (normal) collision involving one infinite (or very large) mass will the effective kinetic energy be the same as the total kinetic energy. In such a case all motion will cease at the time of maximum force. The indentation energy is the integral of the indentation force $F_n$ on the crushing indentation displacement $\zeta_c$ ;

$$IE = \int_0^{\zeta} F_n \cdot d\zeta_c \tag{2}$$

The potential energy is the energy that has been expended in recoverable processes, which can be either rigid body motions (pitch/heave) or elastic deformation (of either body). The potential energy is the integral of the indentation force $F_n$ on the recoverable displacement $\zeta_e$ :

$$PE = \int_0^{\zeta} F_n \cdot d\zeta_e \tag{3}$$

Equation (1) can be solved for $F_n$ provided that the required kinematic and geometric values are known. The general approach to determining *IE* and *PE* will be described next, with specific geometric examples further on. After that the determination of collision forces will be discussed.

## 4.3 ICE CONTACT EQUATIONS

The energy equations require an equation that relates force to indentation. By using the pressure-area relationship to describe ice pressures, it is easy to derive a force-indentation relationship. This assumption means that ice force will depend only on indentation, and the maximum force occurs at the time of maximum penetration. The collision geometry is the ice/ship overlap geometry. The average pressure $P_{av}$ in the nominal contact area $A$ is related to the nominal contact area as;

$$P_{av} = P_0 \cdot A^{ex} \tag{4}$$

where $Po$ is the pressure at $1m^2$, and $ex$ is a constant. Equation (4) is a 'process' pressure area model, in contrast to a 'spatial' pressure area model. For further explanation of this concept see Daley (2007).

The ice force is also related to the nominal contact area;

$$F_n = P_{av} \cdot A_n = P_0 \cdot A_n^{1+ex} \tag{5}$$

The available kinetic energy may be the total kinetic energy, in the case of a head-on collision, in which all motion ceases at the point of maximum force. Alternatively the available energy may be the 'normal' or 'effective' kinetic energy, as in the case of a glancing collision.

For each contact situation, there is a relationship between the normal indentation $\zeta_n$ and normal contact area. Assuming that the function can be expressed as;

$$A_n = f_A \cdot \zeta_n^d \tag{6}$$

where $f_A$ is a function that depends on the contact geometry and d is a scalar (typically .5, 1 or 2). This results in a function relating force to indentation;

$$F_n = P_0 \cdot (f_A)^{1+ex} \cdot (\zeta_n)^{d(1+ex)} \tag{7}$$

The next step is to determine the indentation energy $IE$, which is found by integrating the force;

$$IE = \int F_n d\zeta_n = P_0 \cdot (f_A)^{1+ex} \cdot (\zeta_n)^{d(1+ex)+1} \tag{8}$$

As an example of the approach, the values for one type of contact will be derived in detail. Many other geometry cases have been developed (see Daley, 1999, 2001)

Figure 3 shows a general wedge-shaped edge indentation (normal to hull). The indentation energy is derived as follows. The projected areas, vertical, horizontal and normal are;

$$A_v = \frac{\zeta_n^2 \cdot \tan(\phi/2)}{\cos^2(\beta`)} \tag{9}$$

$$A_h = \frac{\zeta_n^2 \cdot \tan(\phi/2)}{\sin(\beta`)\cos(\beta`)} \tag{10}$$

$$A_n = \frac{\zeta_n^2 \cdot \tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)} \tag{11}$$

Substituting (11) into (5) we arrive at:

$$F_n = p_o \left( \frac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)} \right)^{1+ex} \cdot \zeta_n^{2+2ex} \tag{12}$$

The indentation energy is found by substituting (12) into (8), to give:

$$IE = \frac{p_o}{(3+2ex)} \left( \frac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)} \right)^{1+ex} \cdot \zeta_n^{3+2ex} \tag{13}$$

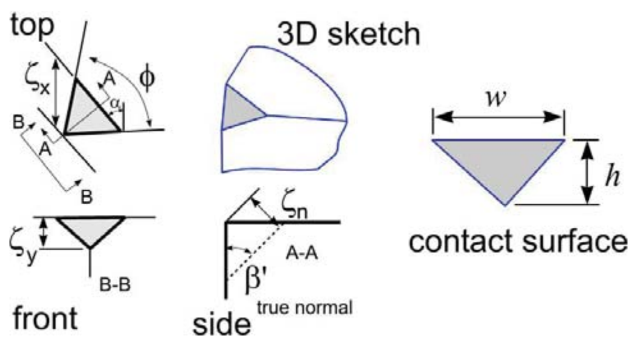



Figure 3: General Wedge-shaped Edge (normal to hull).

For this case the shape of the load patch (see Figure 4) is an isosceles triangular with horizontal extent $w$ and height (along the hull) $h$. The value for normal penetration is determined from the solution of equation (1) with equation (41). The patch dimensions shown in Figure 4 are;

$$w = \frac{2 \cdot \zeta_n \tan(\phi/2)}{\cos(\beta')} \tag{14}$$

$$h = \frac{\zeta_n}{\sin(\beta')\cos(\beta')} \tag{15}$$

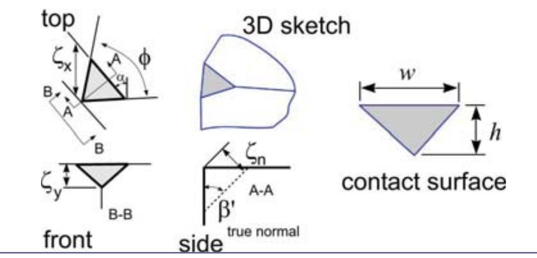
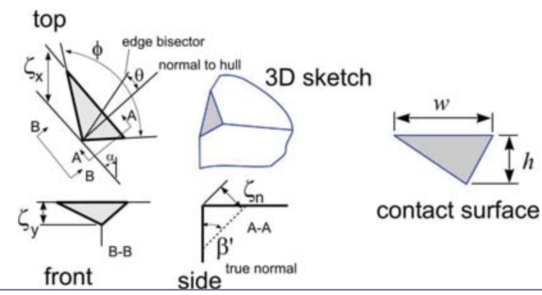
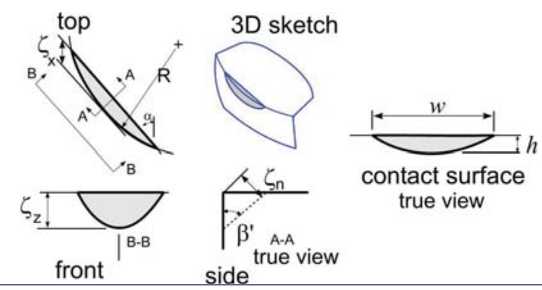
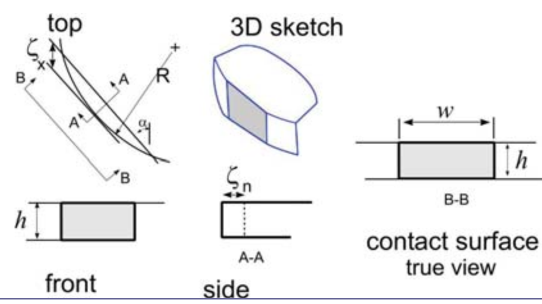
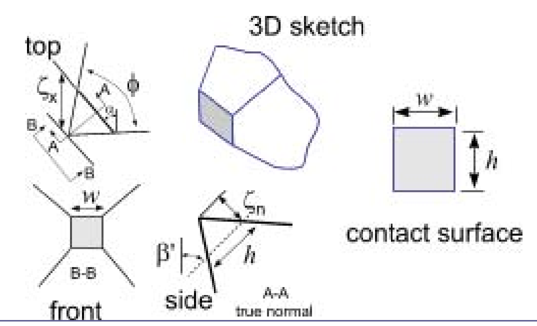
Note that for this case;

$$f_A = \frac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)} \tag{16}$$

and $d = 2$

Table 5 summarizes 4 particular ice indentation geometry cases and fives the shape functions. These can be used with equation (7) to get the force equations.

Table 5: Several ice contact geometry cases.

| | | |
|---|---|---|
| Case 1 :<br>General Wedge<br>(Normal to hull) | $f_A = \dfrac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)}$<br>$d = 2$ |  |
| Case 2 :<br>General Wedge | $f_A = \dfrac{(\tan(\phi/2-\theta)+\tan(\phi/2+\theta))}{2\cdot\sin(\beta`)\cos^2(\beta`)}$<br>$d = 2$ |  |
| Case 3 :<br>General Round<br>Edge | $fa = \dfrac{4}{3\cdot\cos^{1.5}(\beta`)\sin(\beta`)}\sqrt{2R}$<br>$d = 1.5$ |  |
| Case 4:<br>Vertical<br>Cylinder | $fa = 2h\sqrt{2R}$<br>$d = 0.5$ |  |
| Case 5:<br>Horizontal<br>Pyramid | $f_A = 4\tan(\phi/2)$<br>$d = 2$ |  |

## 4.4 NORMAL IMPACT COLLISIONS

A wide variety of collision scenarios can be analyzed as 'normal' collisions. A general oblique collision is shown in Figure 4. The load acts on the hull, primarily normal to the hull plane at that point.

The general approach is presented followed by the force values that occur for the set of contact geometry cases described in section 1. Start by equating the normal kinetic energy with the ice crushing energy.

$$KE_e = IE \tag{17}$$

where

$$KE_e = \frac{Me}{2} \cdot Vn^2 \tag{18}$$

which ,using equation (8) can be stated as;

$$KE_e = P_0 \cdot (f_A)^{1+ex} \cdot (\zeta_n)^{d(1+ex)+1} \tag{19}$$

Solving for the normal indentation:

$$\zeta_n = \left( \frac{KE_e}{p_o \cdot f_A{}^{1+ex}} \right)^{\frac{1}{d(1+ex)+1}} \tag{20}$$

The normal force can be found by substituting eqn. (20) into (7) to give

$$F_n = p_o \cdot f_A{}^{1+ex} \cdot \left( \frac{KE_e}{p_o \cdot f_A{}^{1+ex}} \right)^{\frac{d(1+ex)}{d(1+ex)+1}} \tag{21}$$

The values from Table 5 can be substituted into eqn. (21), together with (18) to get impact force equations for each case. The effective kinetic energy depends on the nature of the collision. For the simplest direct collisions the effective kinetic energy (eqn. (18)) is the total kinetic energy. For ship-ice collisions (see Figure 4) , the effective mass and velocity properties at the point of impact are determined as follows (see Daley 2001 for the *lx* and *Co* terms) ;

$$V_n = V_{ship} \cdot lx \tag{22}$$

where $V_n$ is the normal velocity at the point of impact
$V_{ship}$ is the forward velocity (all others zero)
$lx$ is the x-direction cosine

$$M_e = \frac{M_{ship}}{Co} \tag{23}$$

where $M_e$ is effective mass at the point of impact
$M_{ship}$ is the ship's mass (displacement)
$Co$ is Popov's mass reduction factor

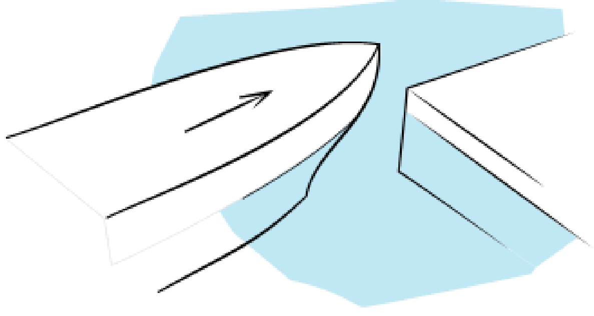


Figure 4: General oblique collision

The solution of the impact gives several results. With the penetration from equation (20), the nominal contact area can be found. This gives an initial load patch that can be used for structural assessment. As well, it is possible to make an estimate of the impulse in the collision and thus find the changes in velocity. This is useful when calculating velocities for reflected collisions.

## 5. STRUCTURAL ANALYSIS

In order to execute a structural analysis, a detailed load description is required. In the previous sections, the approach to find the peak collision force, together with the nominal load patch was presented. It is well known that ice loads are internally complex, with very high local pressures on small portions of the nominal contact area. In order to reflect this mechanics in a reasonable and practical way, a method has been developed to convert the nominal load to a load for structural analysis purposes. The procedure reflects the methodology developed for the new IACS Unified Requirements for Polar Ships. (see Daley 2000 and 2001 for details). In the procedure the nominal load is converted to a rectangle and reduced in size, with a sufficient pressure to ensure that the force is as calculated. The design load patch can then be applied to a finite element model of the hull. Figure 5 illustrates the procedure being followed in the current project.

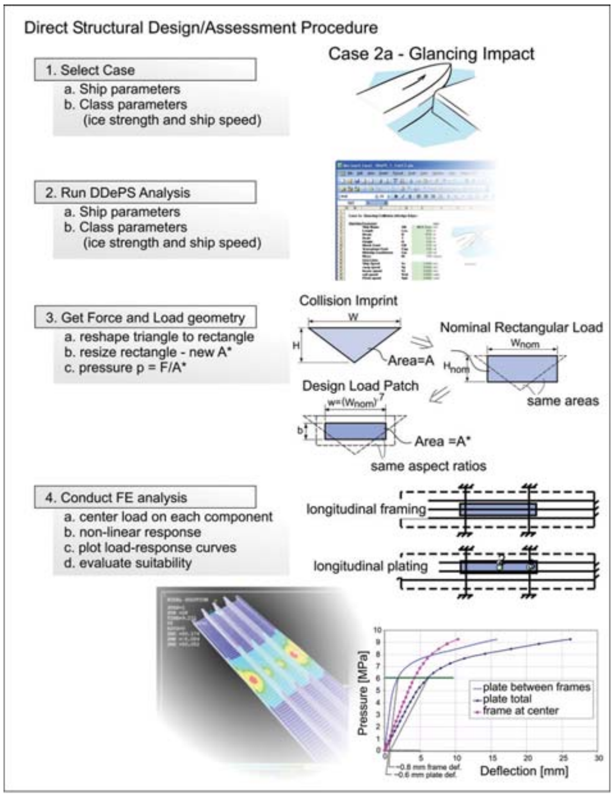



Figure 5: General Wedge-shaped Edge (normal to hull).

The structural analysis aspect of the joint research project is just getting underway. There are several challenges which will need to be addressed. Figure 6 shows a cross section through the hull and cargo containment system. Structural analysis will be used to assess not only the hull integrity, but the impact of ice loads on the cargo containment system (CCS). It is crucial that the CCS not be a risk, even when there is minor structural damage. The system behaviour, through the elastic range and into the plastic and large deformation range will be examined.

In assessing the results, it will be necessary to keep the work grounded by any and all previous experience. To do this, one strategy being used is to make use of the methods and philosophy of the new IACS Unified requirements for Polar Ships. In areas where the LNG ships will differ in operations or layout from the standard ships envisaged in the Polar Rules, comparable or better safety can be achieved by building upon the Polar Rules.

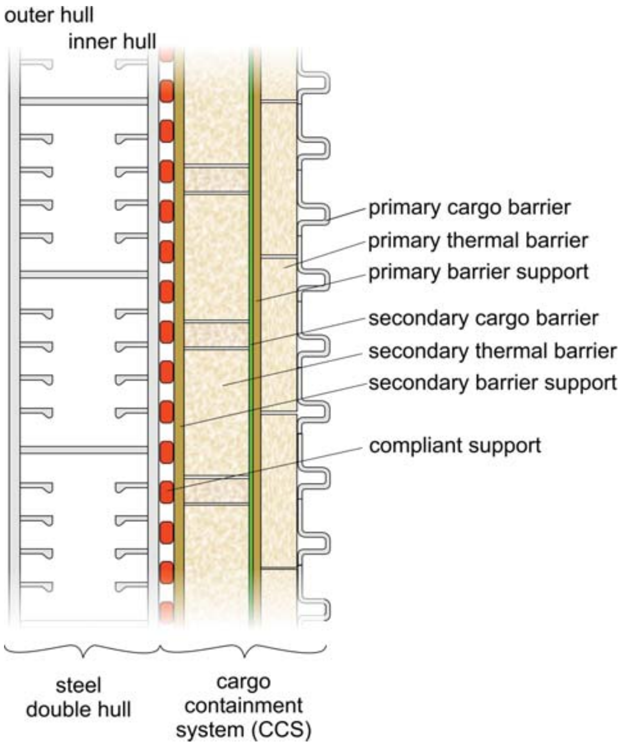



Figure 6: Concept Sketch of LNG Cargo Containment System (CCS) (represents HHI GTT Mark III)

## 6. CONCLUSIONS

The paper has presented some of the initial results and the current directions of the joint research project on high-arctic LNG ship design. The novelty and significance of these ships requires that the design proceed from the basics. As well, a strategy of adopting the concepts embedded in the IACS Polar Rules makes it possible to craft a novel design with the benefit of all the scientific, safety and operational experience that went into the Polar Rules.

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# PRACTICAL DESIGN OF LNG CARRIERS IN LOW TEMPERATURE ENVIRONMENTS

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## SUMMARY

The demand for LNG Carriers sailing in low temperature environments is expected to increase. In order to meet this increase in demand, Winterization for large LNG Carriers is being studied by classification societies, shipowners and shipyards. In this paper, the practical design of LNG Carriers will be discussed from a shipbuilder's point of view.

Since there is a large variation in "low temperature environments", it is important to optimize design specifications for Winterization, taking into consideration the initial investment required and the range of sailing areas to be covered. In this paper, key issues for the design of LNG Carriers operating in cold climates, such as the definition of design temperature for material and equipment, practical countermeasures against icing and so on, will be discussed. Furthermore, the selection of the propulsion system will be discussed in view of environmental protection, which is especially important in arctic areas. Focusing on the above issues, this paper will propose practical solutions, utilizing design knowledge of existing LNG Carriers, while also covering the wide range of cold climate conditions expected in LNG trading routes.


## 1. INTRODUCTION

There is a wide range of low temperature environments, from Arctic and Antarctic areas, where temperatures are very low with thick sea ice, to areas where temperatures only reach minus for a short time in winter, not long enough to form sea ice.

The voyage profile of LNG Carriers also varies widely, with some vessels continuously operating in low temperature environments throughout the winter, whilst other vessels, engaged in long distance trade, only experience cold climates near to their terminal ports. Furthermore, ship owners have their own policy for operation in low temperatures. For economic reasons, some ship owners continue to use normal vessels in low temperature environments, relying on the skilled seamanship of experienced crews. Other ship owners however, prefer to invest more money to combat the effects of Winterisation, in order to reduce the crew's burden and the risk of damage to their vessels and equipment.

Due to the varieties of environment, voyage profile and owner's policy, it is therefore very difficult to say what the optimum Winterisation solutions should be (refer to Figure 1). It goes without saying that, if a vessel operates frequently in lower temperature environments, the risk of damage to the vessel and its equipment will be increased.

LNG Carriers have a good safety record in normal climate operations, therefore the challenge for Winterisation solutions is to maintain the same level of safety whilst operating in lower temperatures. In doing so, we must also pay particular attention to the natural environment in colder climates, which are very vulnerable to pollution. To this end, in this paper we will consider the propulsion system, suited for environmental protection, as well as the practical design approach for strengthening the hull structure against ice pressure.

However, in a competitive market, vessels specifically designed for operating in low temperature environments must be economically competitive with normal, existing vessels. In this paper we will try to present Winterisation solutions for Merchant Vessels, which can best meet the varied requirements of the modern world.

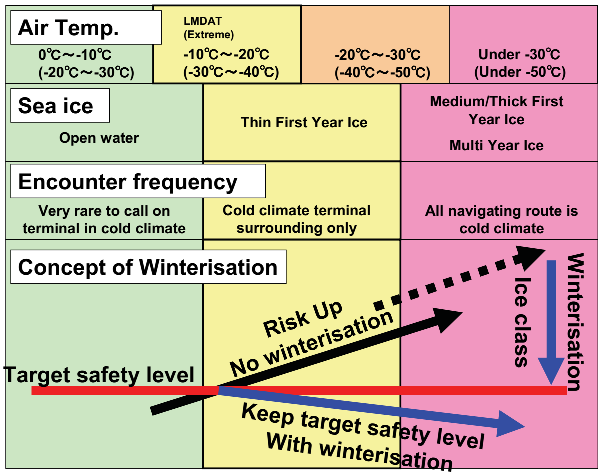



Figure 1: Concept of winterisation

## 2. CLASSIFICATION OF WINTERISATION

### 2.1 DESIGN TEMPERATURE

Temperature is a variable unit or measurement, therefore a definition of design temperature is necessary, in order to start discussion on Winterisation. For example, for selection of hull steel, the IACS definition S 6.2 is the "Lowest Mean Daily Average Temperature" (LMDAT) [5]. (refer to Figure 2.)

For the function of equipment (air conditioner, propulsion plant etc.), several definitions can be considered as follows:

- "LMDAT" as explained above.
- "Extreme Temperature" corresponds to the minimum temperature to which the ship is exposed during its operational life, but only for short periods. This is roughly considered to be 20 degrees C lower than LMDAT.
- "Temperature of the Coldest Day with Probability x.xx", "Temperature of the Coldest Five-Day Period with Probability x.xx", according to Russian Industrial Standards.
- "Absolute Minimum Temperature", which may be the lowest minimum temperature in 100 years.

Sometimes "Extreme Temperature" may be the basis for equipment design, however, depending on the importance of the equipment, we consider design based on the "Extreme Temperature" definition to be somewhat excessive, and a temperature between "LMDAT" and "Extreme temperature" may in fact be more suitable. A correct definition of design temperature and its probability is an important discussion item regarding functional design for Winterization.

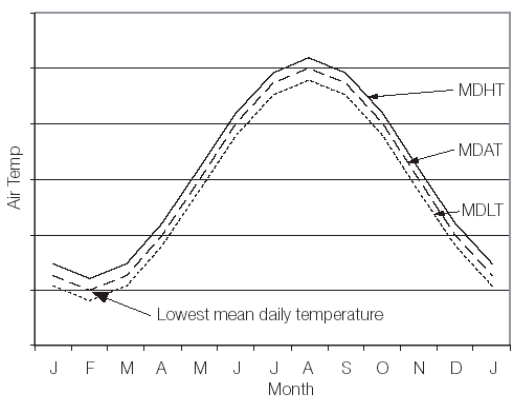



Figure 2: Definition of design temperature [5]

## 2.2 CATEGORIES OF WINTERISATION

We consider that the level of Winterisation can be roughly divided into three groups, taking into consideration the voyage profile and environmental types of merchant vessel (refer to Figure 3).

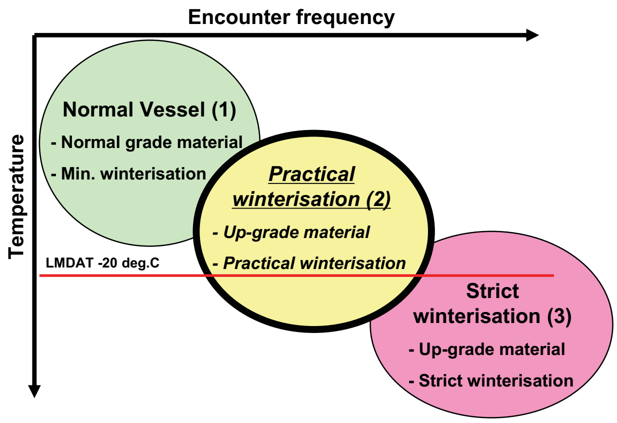



Figure 3: Portfolio of Winterisation

- <u>Normal Vessel with Minimum Winterisation</u>

Existing vessels operating in the Baltic Sea and LNG Carriers for the Snohvit project may be classified in this category. The ambient temperature is around 0 to -10 degree C in LMDAT, equal vent to -20 to -30 degrees C in extreme temperature condition. Normal steel grade should be applicable for this type of vessel, therefore countermeasures for cold climate conditions can be limited, to minimize cost increase and additional maintenance work.

- <u>Vessel with Practical Winterisation</u>

Vessels, constantly operating in areas where ambient temperatures go down -10 to -20 degrees C in LMDAT definition and -30 to -40 degrees C in extreme temperature, may be classified in this category. It is important to minimise damage and maintain functional operation in low temperature environments with adequate, practical and effective winterisation. However Winterisation solutions should be carefully selected so as not to significantly deteriorate the vessel's cost effectiveness.

Winterisation specifications in this category of Vessel may vary considerably, depending on the ship owner's preference, because this category is in-between that of a normal vessel and that of a polar class vessel, described hereunder.

Natural Gas Fields are scattered in Arctic region of Russia and Sakhalin Island, however, we believe that new terminals for catering such resources will be built at the area where the vessels of this category can suitably operate. In this paper, we discuss Practical Winterisation for this category.

- <u>Vessel with Strict Winterisation</u>

Vessels operating in Arctic regions, including "Polar Class" [6] Vessels may be classified in this category. These areas are covered with thick ice and reach very low temperatures. The safety of both vessels and crews is the top priority in determining Winterisation specifications for this category. Although the concrete design of the vessel in this category is under investigation, the big gap in design and cost is expected as compared with ordinary commercial LNGC.

## 3. HULL STRUCTURE

### 3.1 MATERIAL FOR HULL STRUCTURE

Regarding the mandatory requirement for material selection for use in cold environments, rules and regulations are unclear. It may be understood that basic class notation can cover unlimited sea trade provided "good seamanship" prevails, on the other hand, the IACS UR S 6.2 requirement stipulates, "For ships intended to operate in areas with low air temperatures (below and including -20 degree C), e.g. regular service during winter seasons to Arctic or Antartic waters, the materials in exposed structures are to be selected based on the design temperature…..". [5]. The "design temperature" is defined as LMDAT in this requirement. In this case, hull structural material should be selected in accordance with the tables given in S6.2. The technical background of the material selection is explained based on fracture mechanics [7].

The IACS S6.2 case is clear, however, most of the sailing routes for LNG Carriers are in much milder temperatures. In such cases, there is no clear class (mandatory) requirement and the following three options can be considered.

- <u>Normal Steel Grade</u>

It is considered that the normal steel grade (of basic class notation) can cover up to LMDAT -10 degrees C, extreme temperature -30 degree C.

- <u>Partial Upgrade of Hull Steel</u>

For example, an upgrade of side shell plating above water line from Grade A to AH or D/DH is considered a practical solution to obtain extra steel toughness compared to normal grade steel.

- <u>Application of DAT Notation</u>

Material selection based on tables in IACS S6.2 can be applied to milder environments rather than the areas described in S6.2 ("regular service during winter seasons to Arctic or Antartic waters"). Most classification societies provide additional notation for DAT (LMDAT) of less than -20 degree C. This is the most expensive option.

### 3.2 STRENGTH AGAINST ICE PRESSURE

Structural design against ice pressure is based on the Ice Class Rule, for example FS rule [8]. Furthermore, the strength of the "as built" structure depends on the structural arrangement. For example, in the case that a Russian Authority "Ice Passport" concept is applied, obtained "safe speed" (permissible speed) may depend on the design with the same ice class [4]. Figure 4 shows flow chart of strength design against ice load.

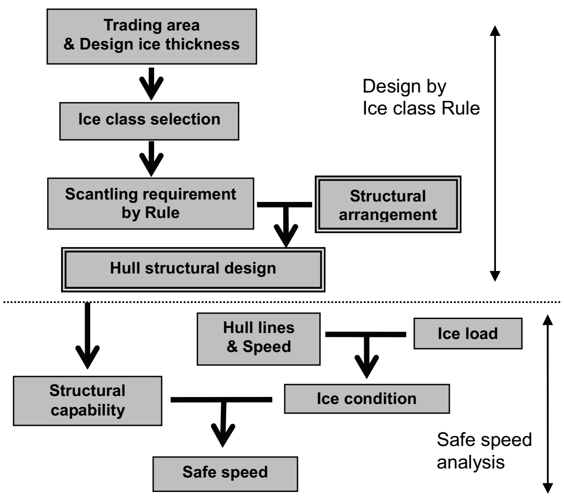



Figure.4: Flow Chart of Strength Design against Ice Load

Since ice pressure varies according to hull form, the greatest pressure location needs to be investigated. Before F.E.Analysis is carried out, ice pressure distribution of the given hull form is estimated. Figure 5 is a schematic figure showing the relation between hull form and ice pressure, and selection of F.E.Model location.

For structural analysis against ice load, a non-linear FEM program such as "ABAQUS" is used. Based on such analysis, optimal design can be obtained in terms of better "as built strength" with limitation of hull steel weight.

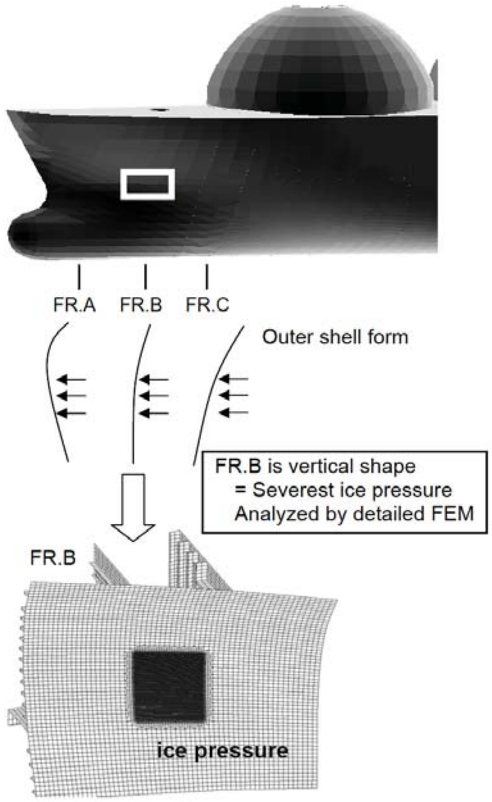



Figure 5: Structural Analysis against Ice Pressure (schematic figure)

3.3 FATIGUE DESIGN

LNG Carriers in cold regions often sail in relatively severe wave environments, therefore fatigue design should be based on the dedicated sailing route of the ship.

Figure 6 gives an example of fatigue analysis with direct analysis of wave load. MHI has developed a system called DILAM (Direct Loading Analysis Method) in which spectral fatigue analysis can be carried out in an efficient way. [9]

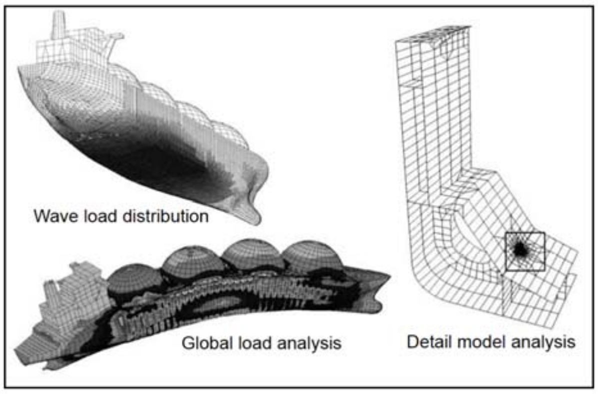



Figure 6: Example of Fatigue Analysis

## 4. OUTFITTINGS

### 4.1. GENERAL

Applying Practical Winterisation, outfittings should be designed to maintain safe and reliable operation in cold environments, and to incorporate anti-icing and de-icing measures, taking into consideration the severity of the cold environment. Typical examples of such measures are shown as follows:

### 4.2 ANTI-ICING FUNCTION

#### 4.2 (a) Ballast tanks

Ballast tanks should be protected against total icing, otherwise dead ballast may occur or ballast tanks may collapse, due to a vacuum effect at de-ballasting. In case of Practical Winterisation, dead ballast will only occur when the vessel is in colder areas and the ice would be expected to melt once the vessel leaves the port. Therefore, the primary aim is to design a countermeasure, to prevent collapse of ballast tanks, due to vacuum effect caused by icing at de-ballasting.

Steam heating can be considered as one countermeasure for anti-icing of ballast tanks, but if many heating pipes have to be fitted in the ballast tanks, this may result in increased maintenance work or require expensive piping made from high grade anti-corrosion material.

However, first of all it is necessary to assess the probability of ice forming in the ballast tanks in any designated cold environment, before deciding on the necessity of any anti-icing measures. Figure 7 gives one estimate of the time required for ice to form and in this case, the ice block formed in 18 days. Therefore, it can be seen that, from this example, unless the vessel stays in that particular area for 18 days or more, no additional anti-icing measures, such as steam heating, are necessary.

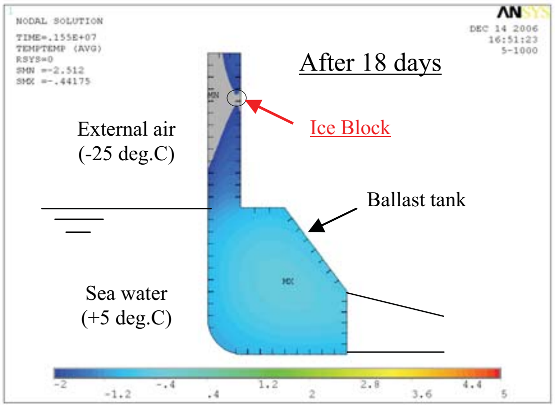



Figure 7: Estimation of Ice Formation in Ballast Tanks

A suitable anti-icing measure, if required, would be Air Bubbling as shown on Figure 8. With this system, air bubbles injected at the bottom of the tank will go upward, making holes on ice surface, to ensure continuous breathing. Another merit of this system is that the air bubbles slow down ice formation, by continuously breaking the ice.

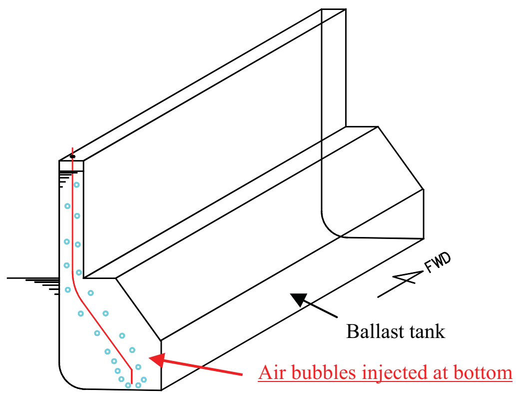



Figure 8: Air Bubbling System

4.2 (b) Fire Hydrant Line

The normal anti-icing measure for the fire hydrant line, is the complete drain off and dry up of the line. However, for safety reasons, LNG carriers are required to keep fire hydrant lines pressurized all times, ready for immediate use if necessary.

Therefore, a circulating system, designed on the principal of total heat balance, is recommended as shown in Figure 9. Sea water supplied by the fire pumps circulates in the fire hydrant line, thus maintaining the pressure. Depending on the temperature of the operating environment, sea water heaters can be installed at the discharge side of the fire pumps as an additional safety feature.

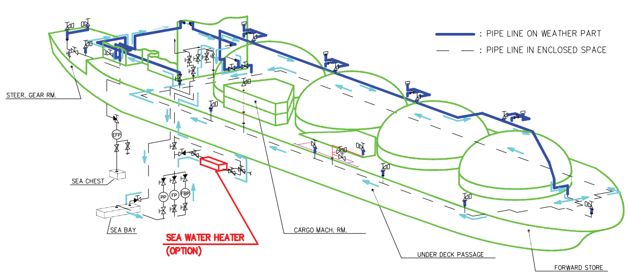



Figure 9: Circulating System for Fire Hydrant Line

4.3 DE-ICING FUNCTION

4.3 (a) De-icing measure

Several factors, such as the severity of ice accumulation, safety concerns, frequency in operation, access etc., should be considered when deciding the appropriate de-icing measures for exposed areas of the vessel. High priority areas should be fitted with permanent heating fixtures, such as heat trace, whereas lower priority areas can be covered with steam blowing as necessary.

4.3 (b) Heating Arrangement

In the case of vessels categorized under practical Winterisation, it is expected that heavy ice accumulation will be limited to the forward mooring deck area. It is unlikely that thick ice will accumulate in other areas, due to features such as the grating structure (shore manifold area), or areas protected from sea spray (flying passage level) and areas behind large structures (aft mooring deck).

Therefore, the forward mooring deck should be provided with some kind of fixed heating system, arranged under the deck. The aft mooring deck can be provided with the same heating system if required, but it should be limited. (refer to Figure.10)

For the other areas, steam blowing connections should be arranged as necessary from view point of maintenance.

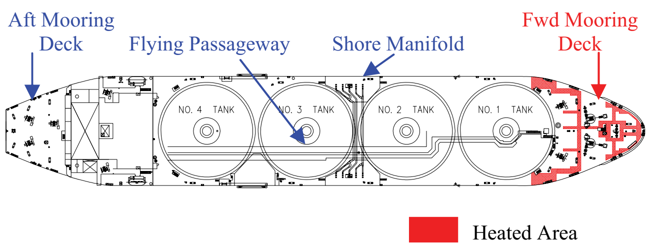



Figure 10: Recommended Heating Arrangement

4.4 MATERIAL FOR DECK EQUIPMENT

Regarding material selection of deck equipment, there seems to be no clear guidance such as in IACS S6.2 for hull steel material. Therefore, appropriate material selection of deck equipment is not straightforward. Furthermore, it should be noted that selection of normal grade material ("normal ship" without DAT notation) depends on the building practice of individual shipyards.

Taking this into consideration, we would like to propose a basic procedure as follows:

- The applied material for existing vessels should be the starting point.
- The Difference in design temperature (LMDAT) between a normal vessel and the newly designed vessel should be assessed based on fracture mechanics. As a result, Charpy energy at a specified test temperature is given.
- The material, which satisfies the charpy energy in the above, should be selected.

It should be noted that since the combination of Charpy energy and test temperature represents required toughness, the test temperature itself will not necessarily be the design temperature of the ship. Furthermore, the required toughness depends on working stress and importance of the equipment, similar to the hull structure. Figure 11 illustrates the proposed procedure, using the example of anchor chain material.

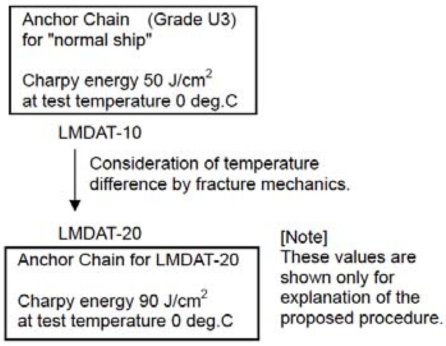



Figure 11: Example of Material Selection (Anchor chain)

## 5. MACHINERY

The required features for propulsion plant will be discussed in this section.

For the Sakhalin project, LNG Carriers will be escorted by two ice breakers and will navigate in the ice channel created. We presume that carriers for other LNG projects will also opt for escorted navigation in waters covered by thin first year ice.

Since the ship would be forced to manoeuvre in the ice channel, the propulsion system should be suitable for slow speed operation such as "dead slow". A propulsion system which enables quick and frequent changeover between ahead and astern may assist the operation, however it is not essential for escorted ice navigation. Furthermore, the low temperature environment would force the vessel to counteract against the cold air suction for engine and cold cooling seawater.

### 5.1 PROPULSION OPTIONS

The five propulsion options can be seen in Figure 12 as follows:

From the inception of the LNG carrier until several years ago, the steam turbine plant (CST) has been the main propulsion system, due to their high reliability and capability to use cargo boil-off gas (BOG) as the main fuel.

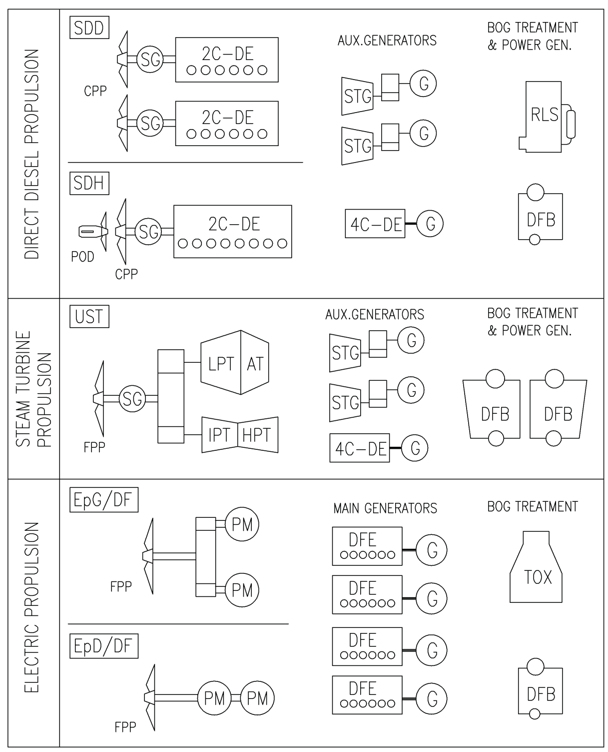
*The Ultra Steam Turbine plant (UST)* has been developed by MHI, based on CST, to achieve higher fuel efficiency corresponding approximately to a 15% reduction in fuel consumption. This system is similar to CST in that the propeller shaft is geared by the cross-compound of high speed turbines, while the difference is the shaft inline generator/motor (SGM) for power generation and back-up slow steaming propulsion operation.

*EpG/DF* is well-known as second propulsion option these days. The propeller shaft is geared by electric motors powered by a dual fuel engine (DFE) generating system.

*EpD/DF* is direct drive by electric motors with the same power plant as EpG/DF, however, it has failed to survive in the market so far.

*SDD* of Slow-speed Diesel Direct oil fuel drive, with boil-off gas re-liquefaction system, is another well-known alternative.

*SDH* of Slow-speed Diesel with electric drive Hybrid developed by MHI has a boil-off treatment system to use gas for electric power and to save boil-off by re-liquefaction.




2C-DE: Slow speed diesel engine
4C-DE: Medium speed diesel engine
RLS: BOG re-liquefaction system
ST: Steam turbine generator
PM: Propulsion motor
DFB: Dual fuel boiler
TOx: BOG incinerator
DFE: Dual fuel engine

Figure 12: Propulsion Options

### 5.2 ICE NAVIGATION

Figure 13 shows the allowable torque range of continuous operation for each option. They are categorized mainly by geared drive or direct drive. CST, UST and EpG/DF are categorized in the geared drive and have the same allowable torque range. The plots for ice navigation are also shown in the figure where these options have enough allowance for escorted operation in ice conditions of 0.6m.

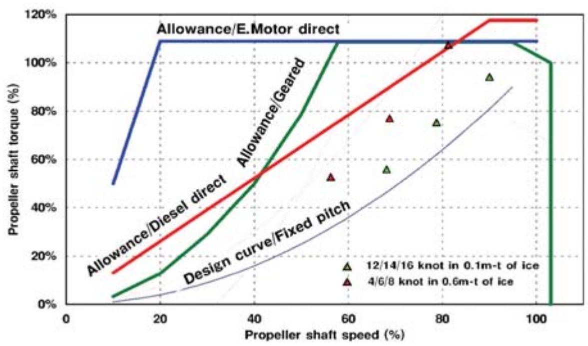



Figure 13: Operating Zone (rpm - torque)

However, when operating in thicker ice conditions or self ice navigation, a special appliance would be required, such as a Controllable Pitch Propeller (CPP), to buffer the high torque or shock load generated, when the propeller comes into contact with the ice.

With the UST, SGM would share the load during slow speed operation, therefore the UST covers wider allowable torque range than the other geared options.

On the other hand in electric propulsions, wider coverage of EpD/DF shall be advantageous.

For SDD, CPP would be unavoidable since the engine itself has narrow allowable torque range.

For SDH, however, a POD drive, which has the same characteristics as a direct motor drive, would be very effective for ice navigation and has the potential in the future to power ice-breaking LNG vessels.

## 5.3 COUNTERACTING THE EFFECTS OF COLD WEATHER

### 5.3 (a) Ice Sea-Chest & Sea-Bay (Refer to Figure14)

In the ice channel, seawater with a minimum temperature of -2 degree C is the primary cooling source and sometimes is made up of slush ice. For all propulsion options, an Ice Sea-Chest & Sea-Bay System would be recommended to prevent slush ice from clogging the flow and from causing serious under-cooling of the secondary cooling source.

### 5.3 (b) Direct Air Suction (Refer to Figure14)

Direct air suction for large output of internal combustion engines would be recommended to avoid an unsuitably cold temperature in the engine room. The engine room would be able to maintain an acceptable temperature for machinery of 0 degree C or above.

### 5.3 (c) Dual Fuel Boiler

Measures to counteract the effects of cold weather require a large amount of steam and hot water to be produced, compared to a conventional vessel. The CST/UST is equipped with dual fuel boilers for main propulsion, hence its ability to use the surplus BOG is expected at slow steaming during ice navigation.

Similarly, even for other propulsion alternatives, dual fuel auxiliary boiler installation is recommended.

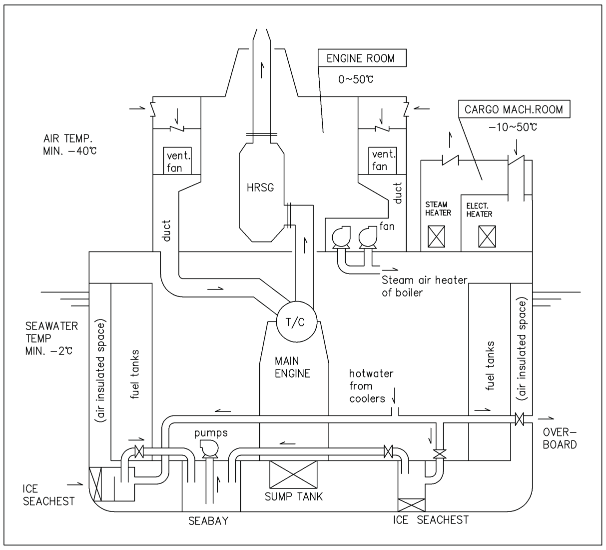



Figure 14: Concept of machinery space

## 5.4 EVALUATION

From the view of ice navigation, the UST, the EpD/DF or the SDH would be good choices for propulsion system. In order to further compare the three options, it is important to look closer in economical and environmental aspects which are very important considerations for vessels sailing in cold climates.

### 5.4 (a) Economical aspect

Total expense per transport quantity is one of the economic indexes and the total expense is made up of fuel cost, maintenance fee, payment for initial investment, etc., however, we focus on fuel cost of the most important factor. Figure 15 shows the fuel gas consumption of UST and EpD/DF.

The required power range for escorted ice navigation and estimated BOG under the cargo-loaded is illustrated in the figure. For both options, fuel gas would be fully provided by natural BOG, in other word, substantial amount of surplus gas should be wasted especially when lower load operations.

At higher load such as ocean going, there's no significant difference between them. And the UST can realize not only mono fuel operation but also dual fuel operation using Heavy Fuel Oil (HFO) of wide mixing rate.

For the EpD/DF, on the other hand, the mono fuel operation is not realized in gas-fuel mode since DFE requires pilot fuel of MDO (Marine Diesel Oil), and use of HFO is only permitted for designated engines as oil-fuel mode.

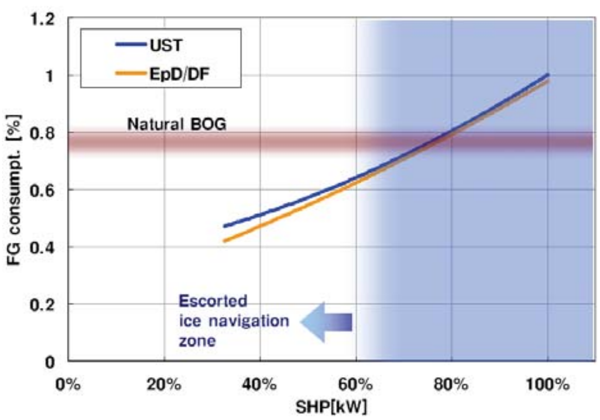



Figure 15: Fuel gas consumption for UST and EpD/DF

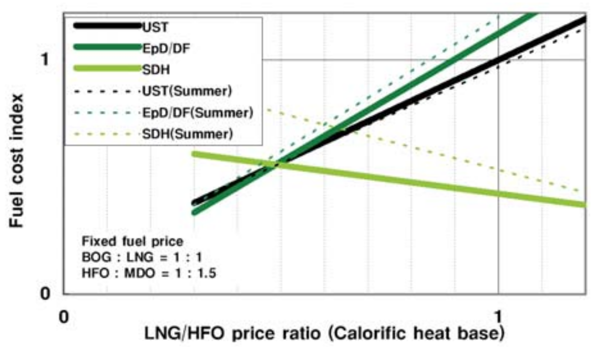



Figure 16: Fuel cost comparison

Figure 16 is an example of the fuel cost evaluation for an LNG carrier engaged between Baltic and Canadian port for winter and summer.

Since each option uses various kinds of fuel throughout a voyage, the fuel cost index on the basis of UST as 1.0 is calculated depending on the LNG/HFO price ratio. For simplicity, MDO price is fixed with the price of HFO as shown in the figure. And for SDH, which has ability to re-liquefy the surplus BOG, the effect of saving LNG is taken into account by subtract equivalent value of cargo LNG from the expense.

The result shows that there is slight difference in fuel cost between UST and EpD/DF throughout the seasons, and wide range of LNG price.

Furthermore, SDH is deserved to achieve the highest economy by using a part of BOG and saving the rest of them, especially in winter operation, because of the longer slow steaming and rest period.

5.4 (b) Environmental aspect

Most coastal areas where you would expect to find ice in the winter are close to countries which will have strict emission restrictions in near future.

Figure 17shows the significant emission factors for each propulsion option. $CO_2$ is the main concern for sailing in the open sea and the emission quantity is rated by propulsive power. On the other hand, NOx is the main concern for port operations, which deeply depends on the availability of BOG.

The environmentally friendly operation would be expected if enough BOG is available, even as the case may be in vaporization of LNG. If not available, considerable NOx is unavoidable for EpD/DF.

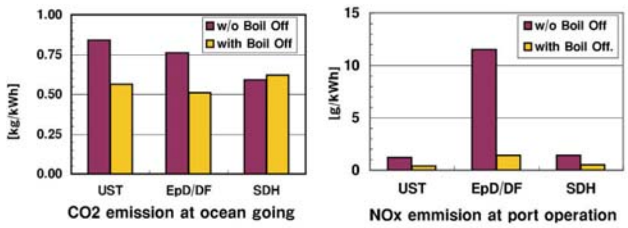



Figure 17: Emission

5.5 SUMMARY

From the above, we can conclude that there is great flexibility in selection of both propulsion system and fuel type to meet the logistic requirements of any vessel.

For Bunker-free vessels, especially in Arctic areas, the UST would be a good choice, since Marine Oil is not always readily available in these areas.

There’s no remarkable difference in economics between UST and EpD/DF, but UST realizes no oil assisted operation, and the better emission performance even when fuel oil operation.

To maximize LNG being transported, SDH would be the best solution by minimizing BOG wasting, while at the same time being environmentally friendly and providing good manoeuvrability in icy waters.

## 6. CONCLUSIONS

We have discussed the practical design of LNG Carriers in low temperature environments, based on our experiences of designing and constructing vessels for the Snohvit and Sakhalin Projects.

Practical design will begin from an understanding of the meaning of design temperature and the effects of low temperature on the vessel & its equipment. As pointed out in this paper, ship owners have their own design policies, therefore, detailed discussions between owners and builders would lead to beneficial solutions.

For vessels in the “Practical Winterisation” category, we believe that the Ultra Steam Turbine (UST) will prove to be both a very economical and environmentally-friendly propulsion plant at the same time also providing the required manoeuvrability.

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# New concept for arctic transportations

**Oskar Levander**
Conceptual Design
Wärtsilä Corporation

DESIGN & CONSTRUCTION OF VESSELS OPERATING IN LOW TEMPERATURE ENVIORNMENTS

London, 30.5.2007

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# Conceptual Design

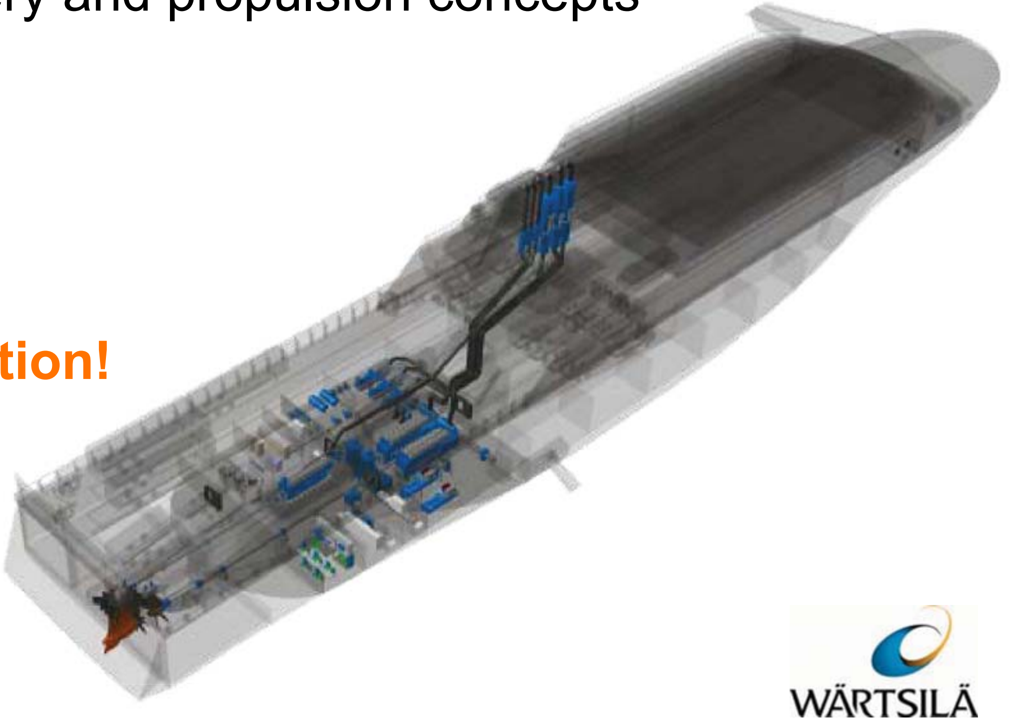
**Conceptual Design's focus areas:**

– Development of new machinery and propulsion concepts

– New ship designs

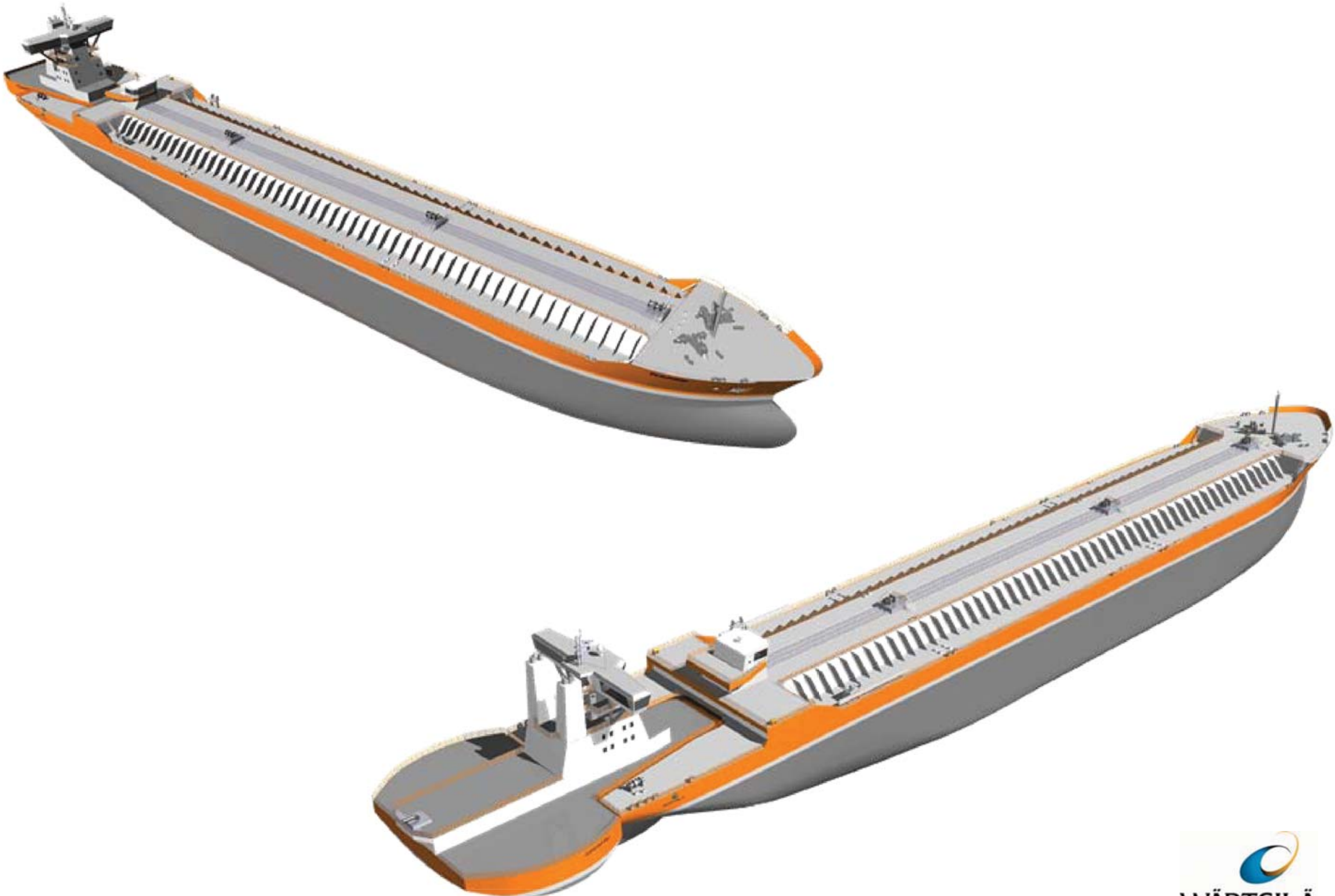
– Customer R&D tasks

**The spearhead of ship innovation!**



# New concept for artic transports



# Contents



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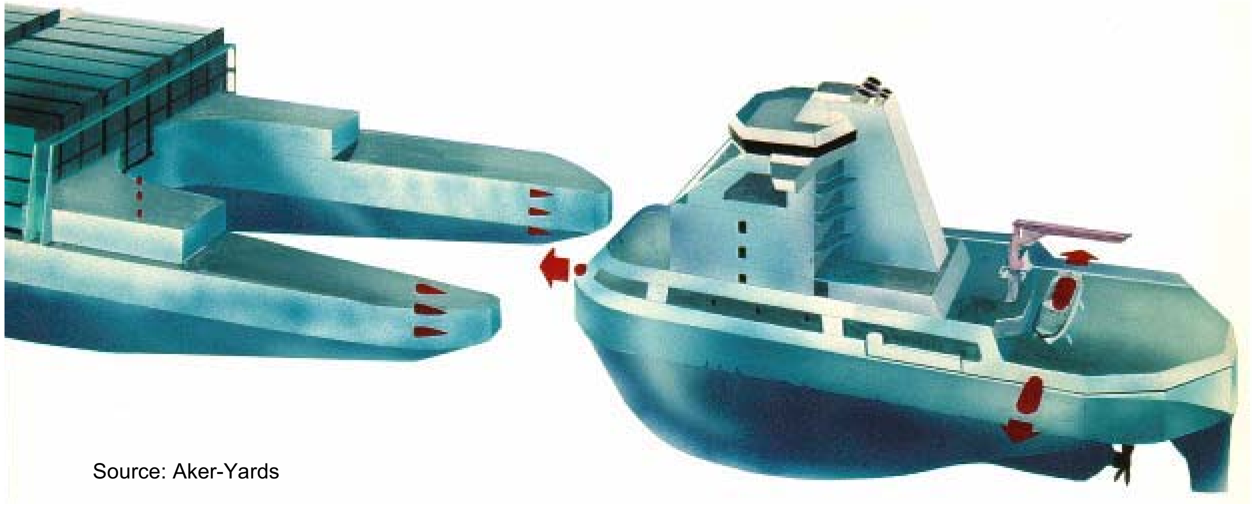
# Background – Pusher-Barge

Pusher-barge systems

- Concept
  - Tugboat with propulsion machinery and accommodation
  - Several barges
- Operation philosophy
  - The barge is loaded and unloaded in port while the pusher / tug pushes another barge at sea
  - The machinery and crew is utilized efficiently at sea. Less non productive time in port



Source: Aker-Yards

# Total economy – Pusher-Barge – Fleet

– The maintenance cost is reduced, because the total number of propelling engines is smaller for the fleet.

– As barges can stay long at terminals for cargo-handling and the on-shore cargo-handling facilities can be operated practically without interruption, their loading and unloading rates can be lower

– It is sometimes possible to reduce the running speed without affecting the total transporting capacity of the fleet to save fuel consumption

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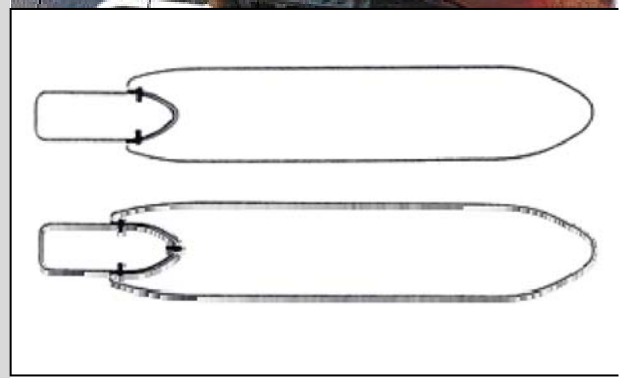
# Pusher-Barge - Connection

– Flexible connection
  - Ropes or wires

– Mechanical connection
  - Articulated (2 points)
  - Integrated (3 points)

# Background – Artic shipping

The is a large interest for artic shipping

- The high oil price increases the interest to exploit artic oil and gas reserves in the Barents Sea and Okhotsk Sea
- Artic ice class ships are needed
  - Tankers
  - LNG carriers
  - Supply vessels


Source: Aker Arctic

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# Background – DAS

- **DAS - Double Acting Ship***, a ship that operates with the stern first when operating in ice.
- This saves in installed power, and fuel, and makes it possible to optimize the bow of the ship for open water performance. A bulbous bow can be used, which is not otherwise suitable for ice operation.
- The flushing of the propellers reduces ice friction against the hull

* developed and patented by Aker Arctic

Source: Aker Arctic

 

# Background – DAS

The DAS concept is successfully used in some artic vessels

- Tankers
- Container vessels
- Supply vessel
- Ice breakers


Source: Aker Arctic

Source: Aker Arctic

 

# Problem

Despite the clear advantages of the DAS concept, there are still some drawbacks:

- Electric propulsion is expensive
- Electrical transmission losses in open water conditions
- Optimising stern for both operation ahead in open water and astern for icebreaking
  - Flow into propeller
  - Course stability
  - Resistance


Source: Aker Arctic


Source: Aker Arctic

 

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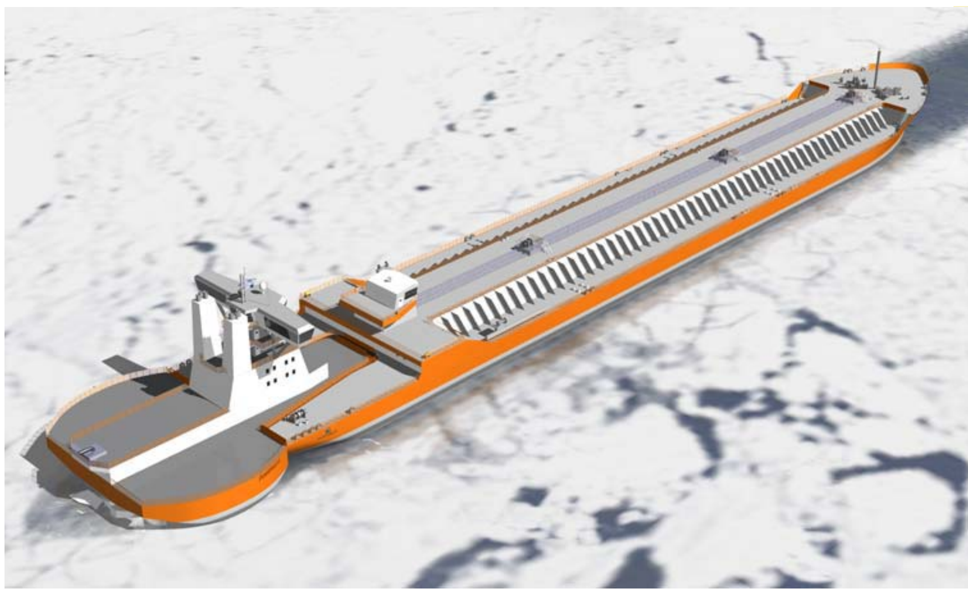
# New concept

The idea is to combine the advantages of the DAS concept and the pusher barge combination:

- There are many identical barges
- Many pushers optimised for open sea use
- One or more “pulling tugs” optimised for moving forward in ice



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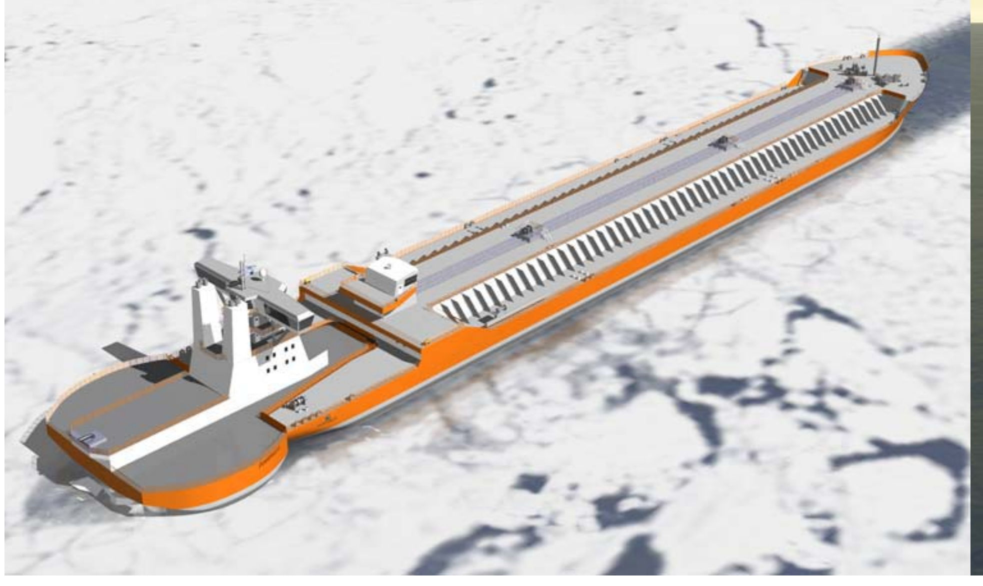
## New concept

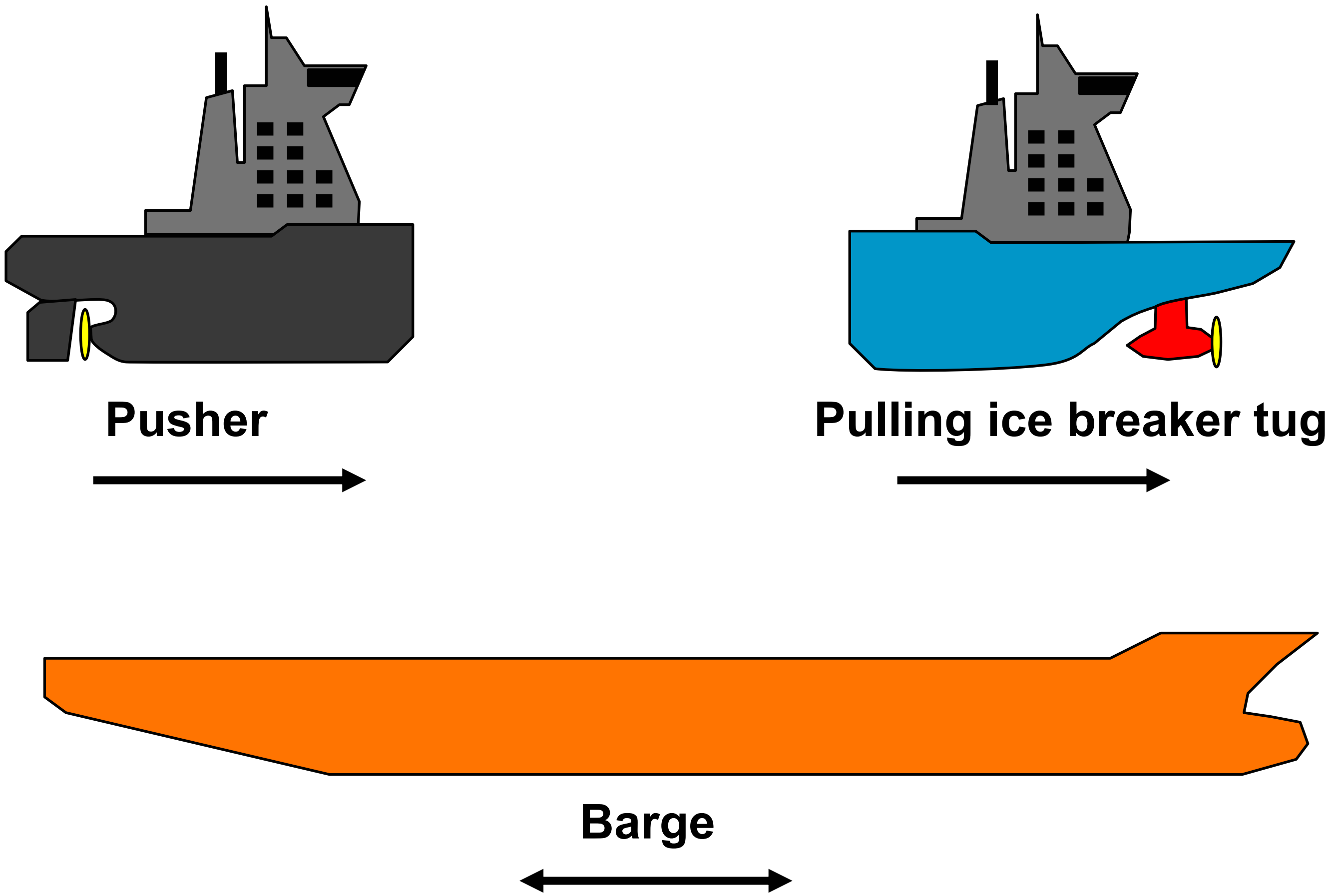
# Double Acting Pusher-Puller Barge system

=

# DAPPB



# Double acting pusher-puller barge concept



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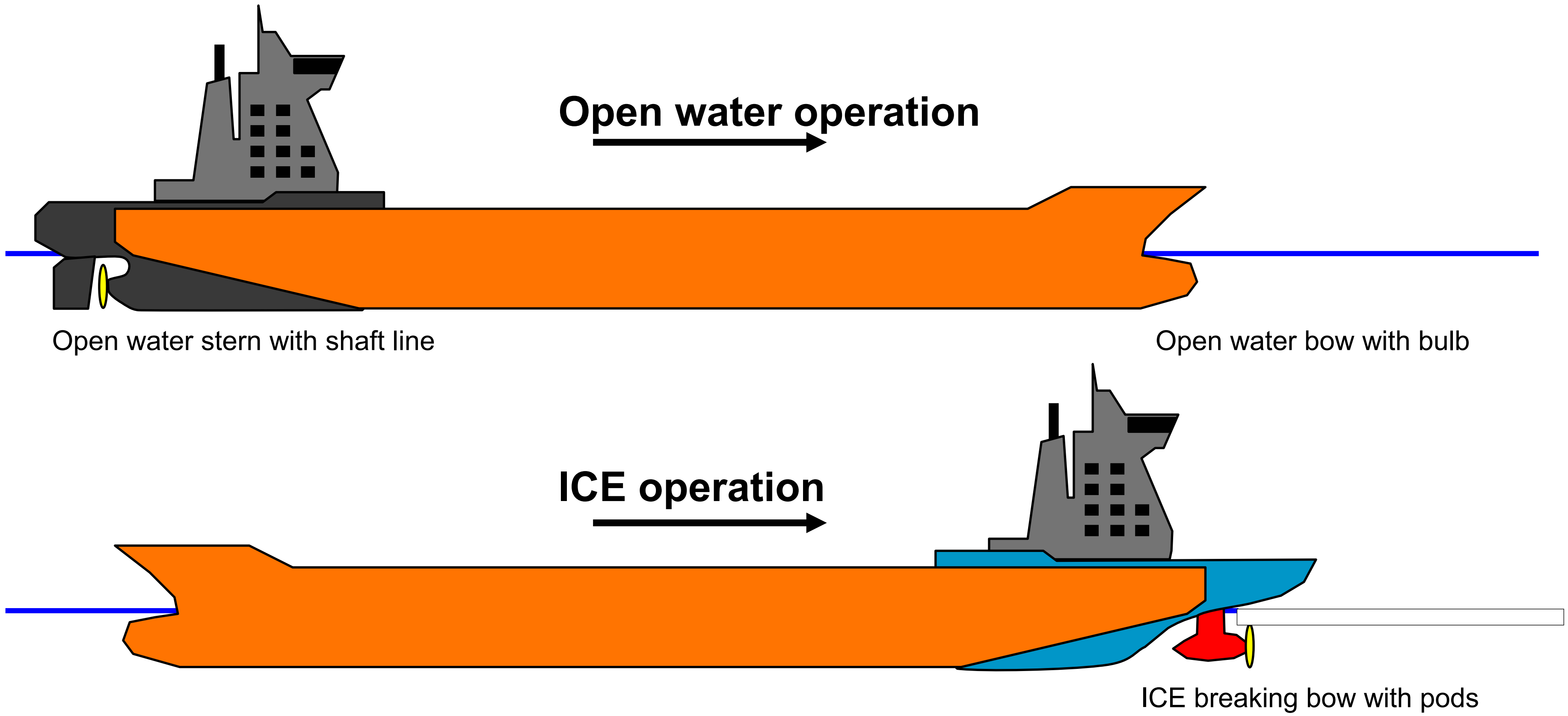
# Double acting pusher-puller barge concept

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# Operation

The new idea is to use a pusher at open sea and switch to pulling tug when approaching the ice:

- Many arctic vessels often operate only a short distance in ice. Most time is at open sea

➔ The expensive ice features are not utilised for more than part time of the operation

 

# Operation - Example


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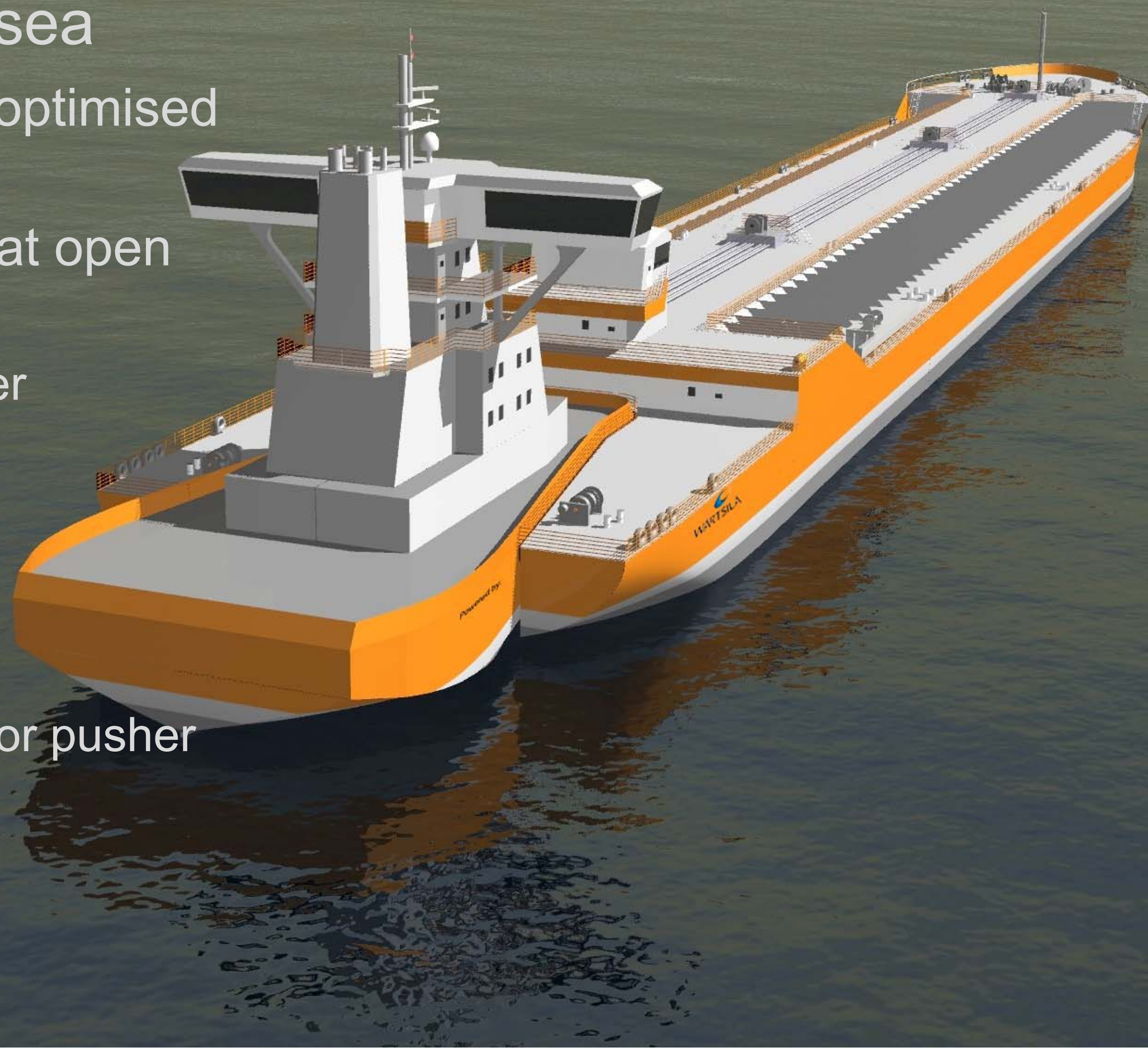
# Benefits – Pushing mode

## Pushing mode – open sea

- Both bow and stern is optimised for open sea operation
- Low fuel consumption at open sea
  - Good inflow to propeller
  - Mechanical drive
  - Less ballast water
- Lower cost machinery
  - No extra power
  - No ice class required for pusher unit



# Benefits – Pulling mode

## Pulling mode - ice

- Ice breaking hull form, no compromise for stern shape
- Pods for efficient flushing of hull
- Expensive machinery is utilized well for the intended use

- Wide ”aft” in puller enhances icebreaking capability
- Ice breaking knowledge of the crew is used efficiently in arctic operation

# Applications

The new Double Acting Pusher-Puller barge concept can be used in the following applications

- Arctic tankers
- Arctic LNG carriers
- Arctic container vessels
- Arctic general cargo vessels
- ?

The same puller/pusher units can serve barges with different cargo

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# Tanker DAPPB - Main dimensions

| | | |
|---|---|---|
| Deadweight | 70 000 | ton |
| LOA | | |
| Pusher-Barge | 241 | m |
| Puller-Barge | 250 | m |
| Beam | | |
| Pusher-Barge | 34 | m |
| Puller-Barge | 40 | m |
| Draught | 13 | m |
| Depth | 19 | m |
| Speed (open water) | ~ 15 | kn |

 

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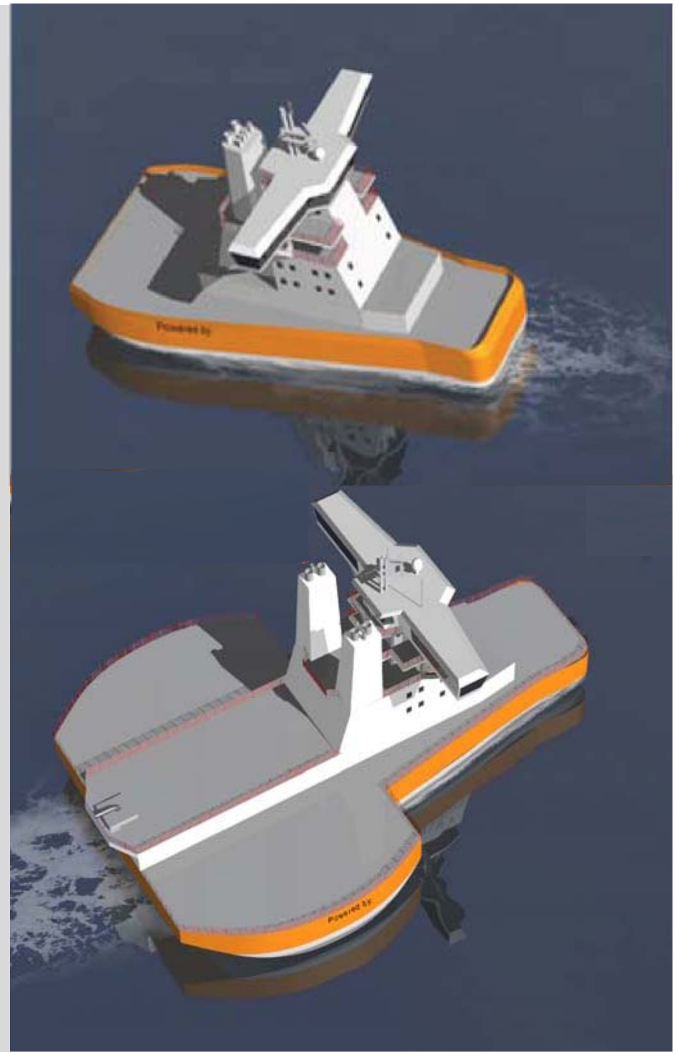
# Main dimensions – Pusher and Puller units

| | Pusher | Puller | |
|---|---|---|---|
| Length oa | 48 | 57 | m |
| Length wl | 47 | 53 | m |
| Beam | 23 | 40 | m |
| Draught | 9 | 11.5 | m |
| Depth | 15 | 17.5 | m |
| Propulsion Power | ~11 | ~17 | MW |



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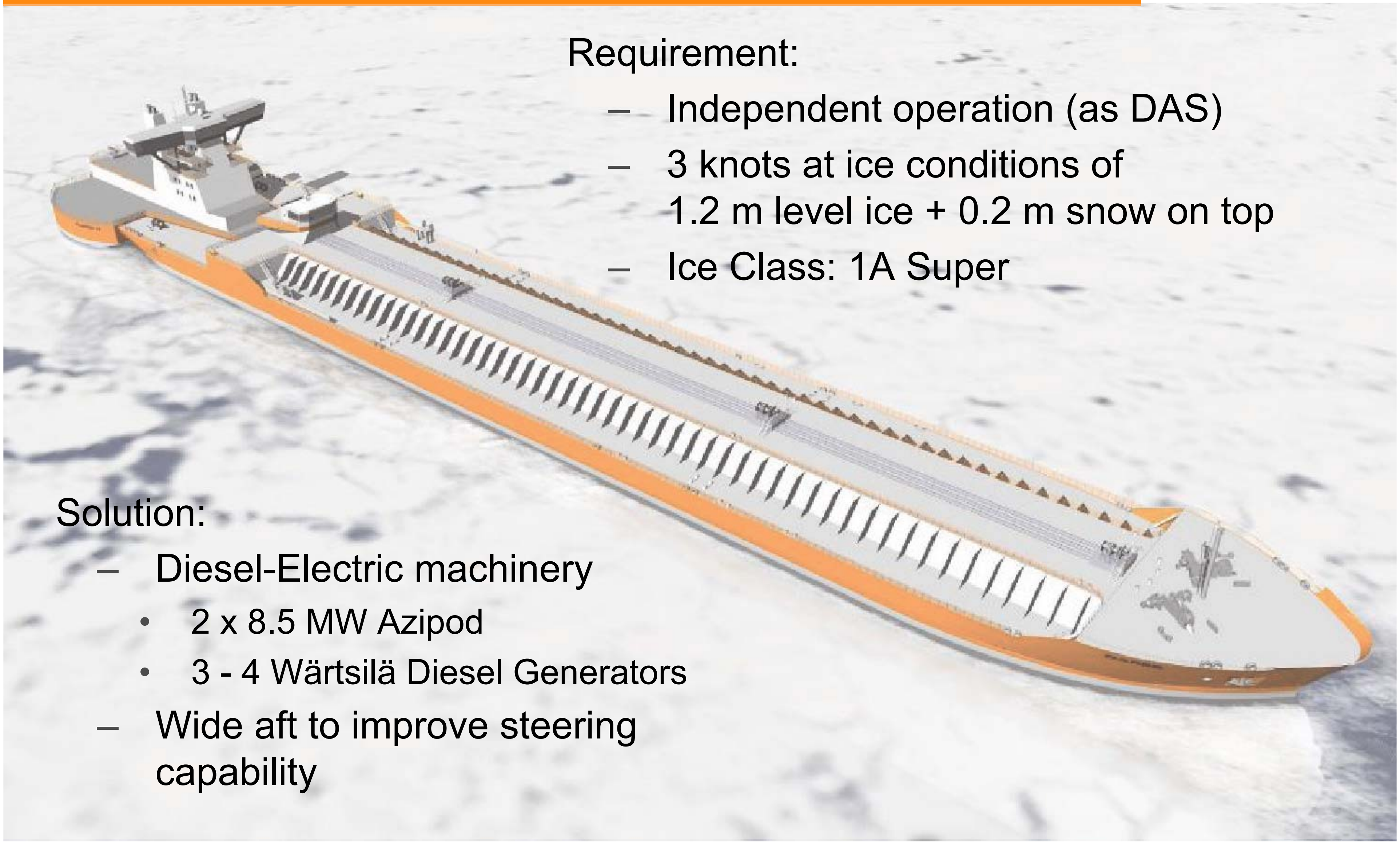
# Ice Performance for Puller-Barge combination



Requirement:

- Independent operation (as DAS)
- 3 knots at ice conditions of
  1.2 m level ice + 0.2 m snow on top
- Ice Class: 1A Super

Solution:

- Diesel-Electric machinery
  - 2 x 8.5 MW Azipod
  - 3 - 4 Wärtsilä Diesel Generators
- Wide aft to improve steering capability

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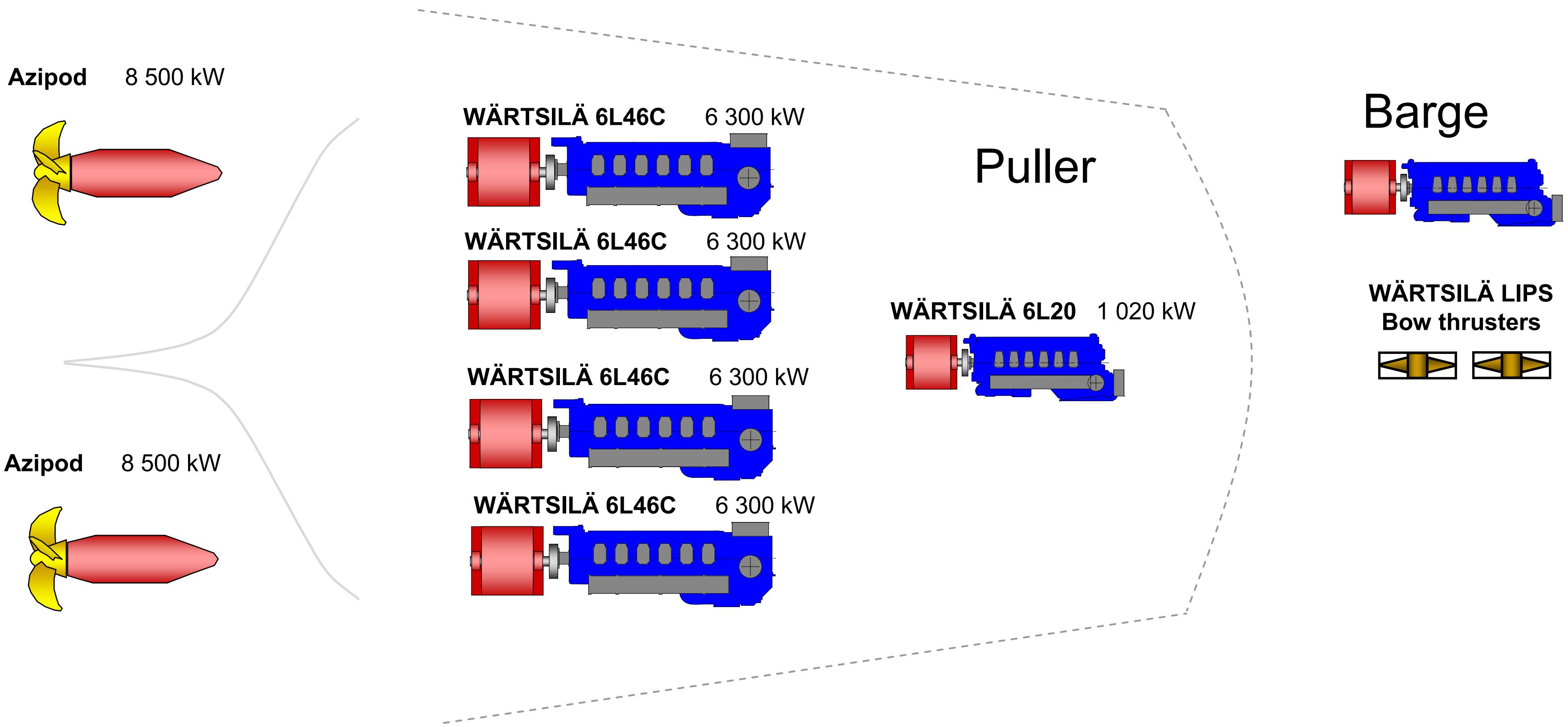
# Machinery example for ice Puller



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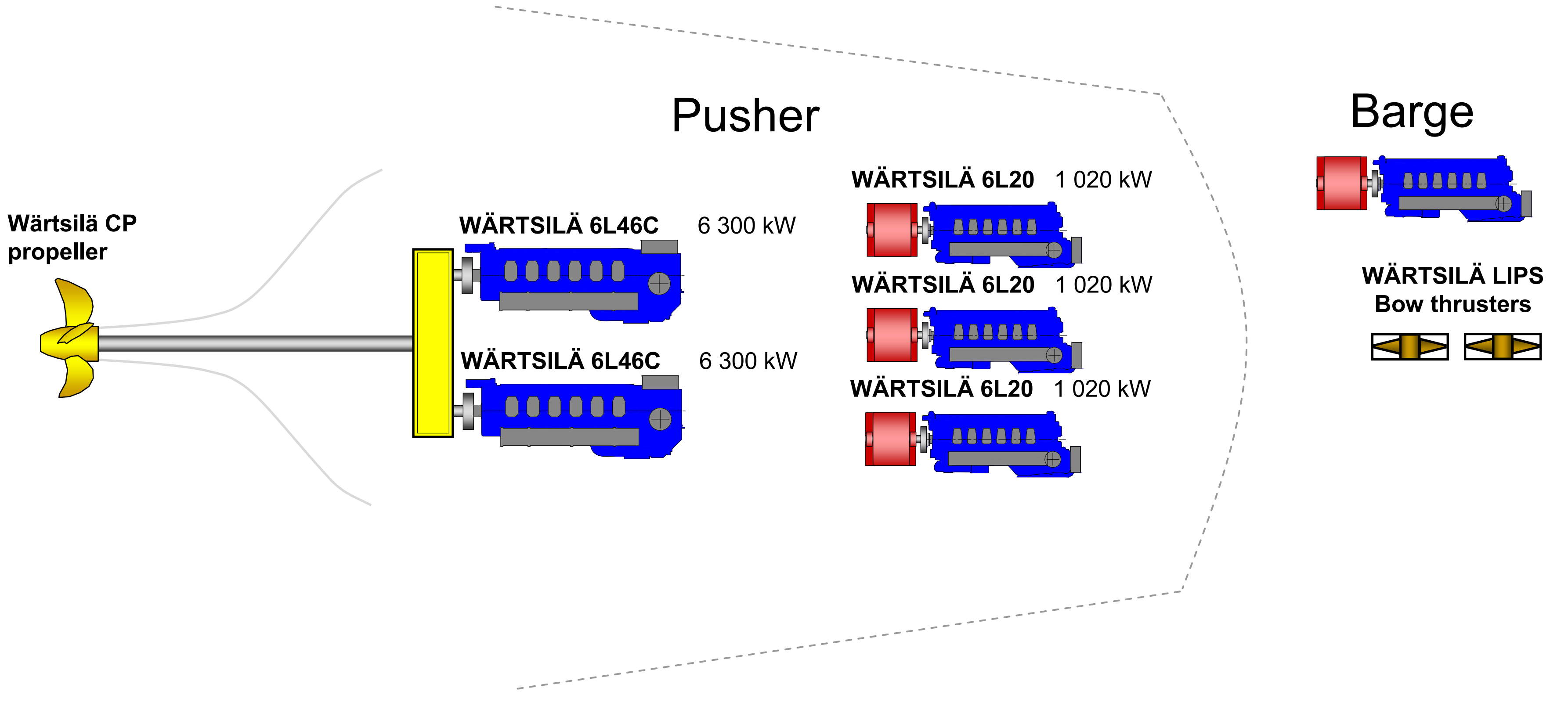
# Machinery example for Pusher



# Pusher change

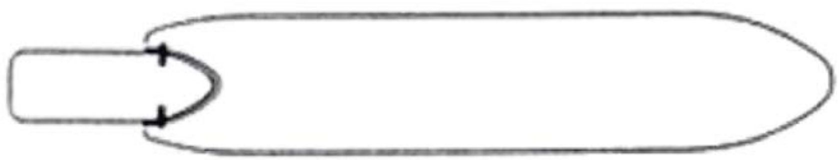

 

# Connection: Ice puller – Barge

The units can not be disconnected at loading terminal

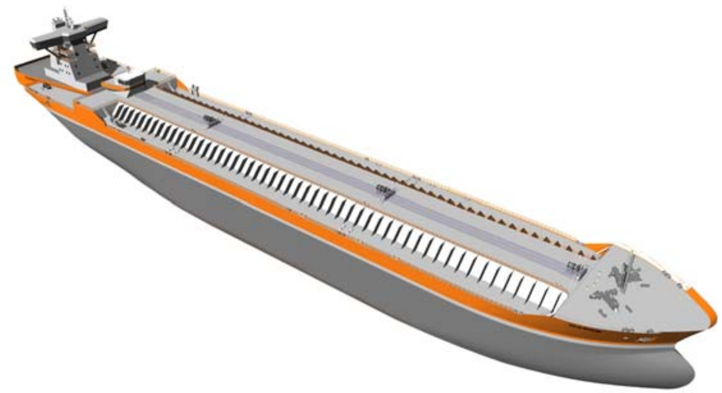
→ Articulated –type connection enabling vertical movement

- Pitching is allowed in ice / heavy sea conditions to reduce forces
- Gap between Puller and Barge required
- Flexible connections on pipes etc. between Puller and Barge required



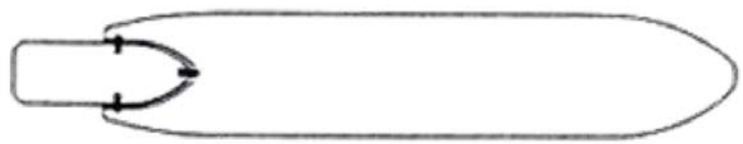
 

# Connection: Pusher – Barge



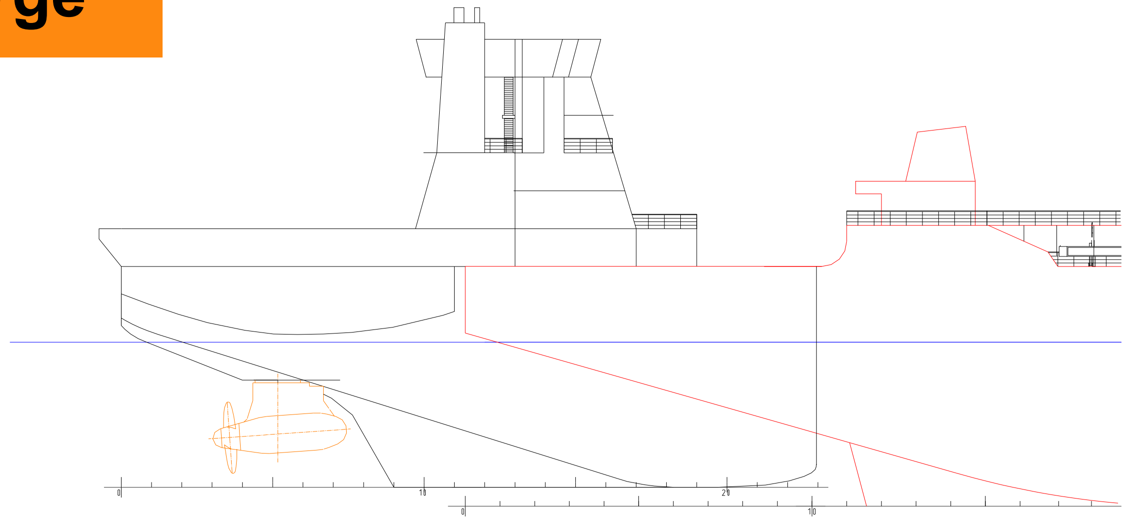
Integrated -type connection on two levels

– Gap between Pusher and Barge can be minimized to achieve continuity on underwater hull form
– During the unloading connection height should be changed
  → Temporary wire connection during unloading

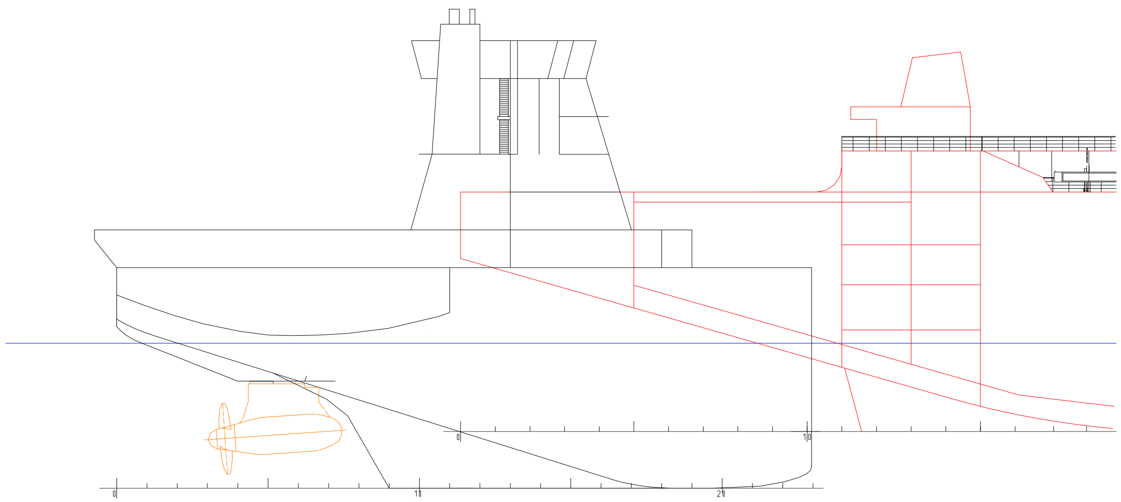


# Ice Puller – Barge



BARGE IN LOADED
CONDITION



BARGE IN BALLAST
CONDITION
CA. 6 M UP

Puller unit floats at ”fixed” draught to ensure the propeller immersion

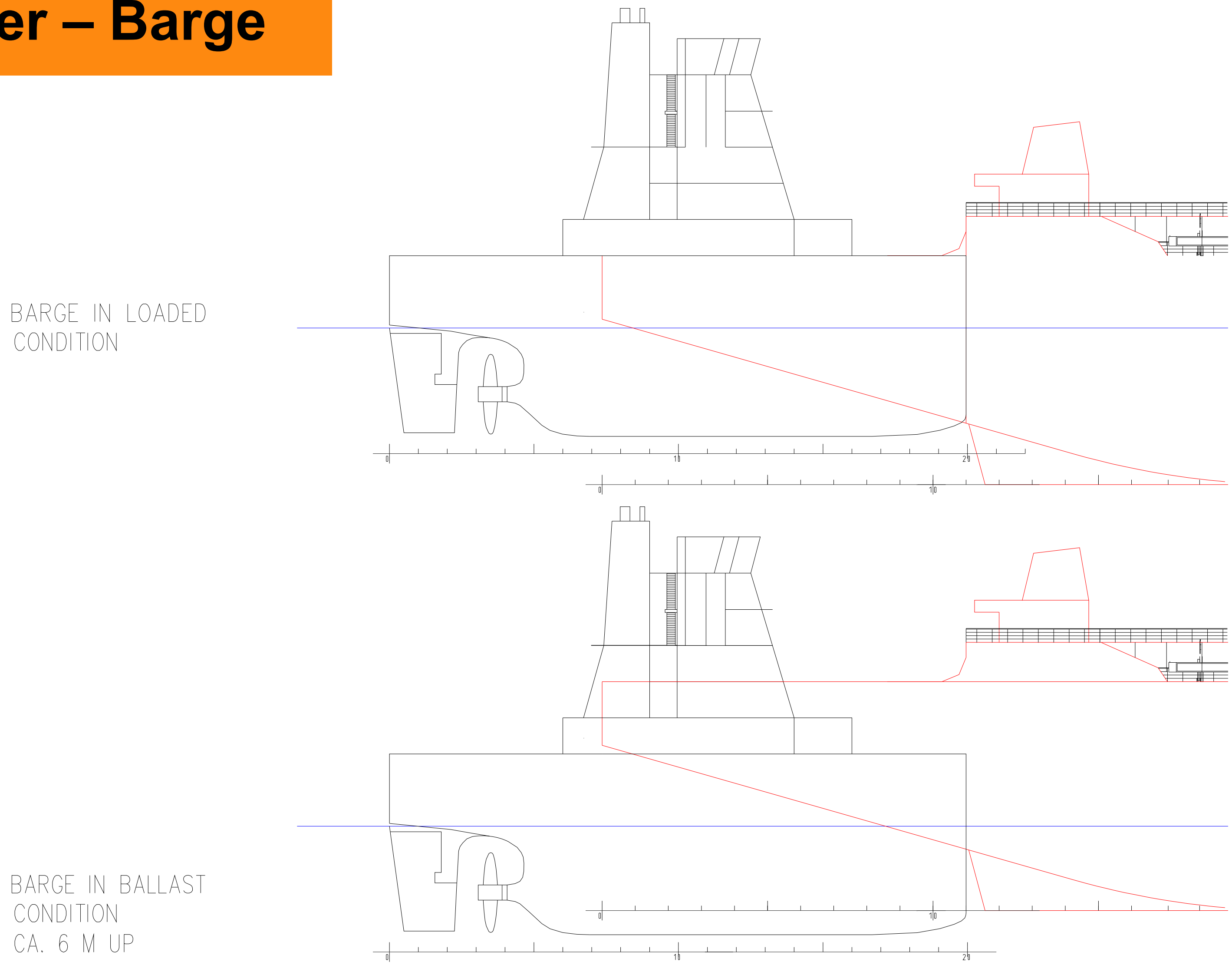
 

# Ice Puller – Barge

Puller is always at optimum ice breaking draft

# Pusher – Barge



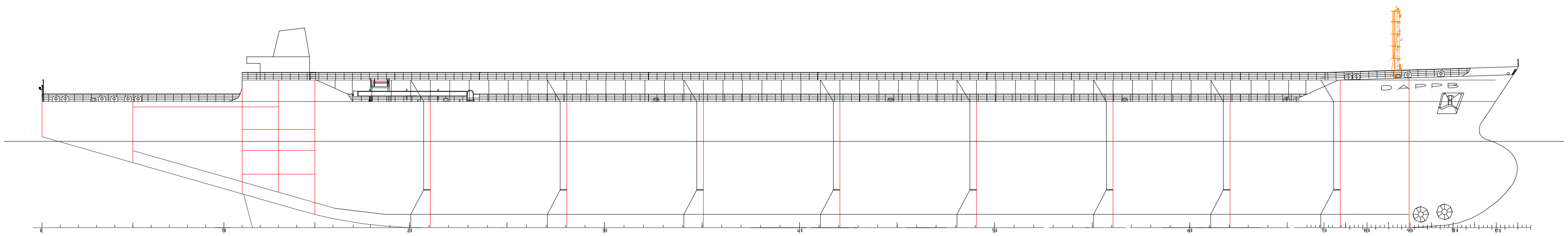

Pusher unit floats also at ”fixed draught”. With this system it may be possible to reduce the amount of ballast water

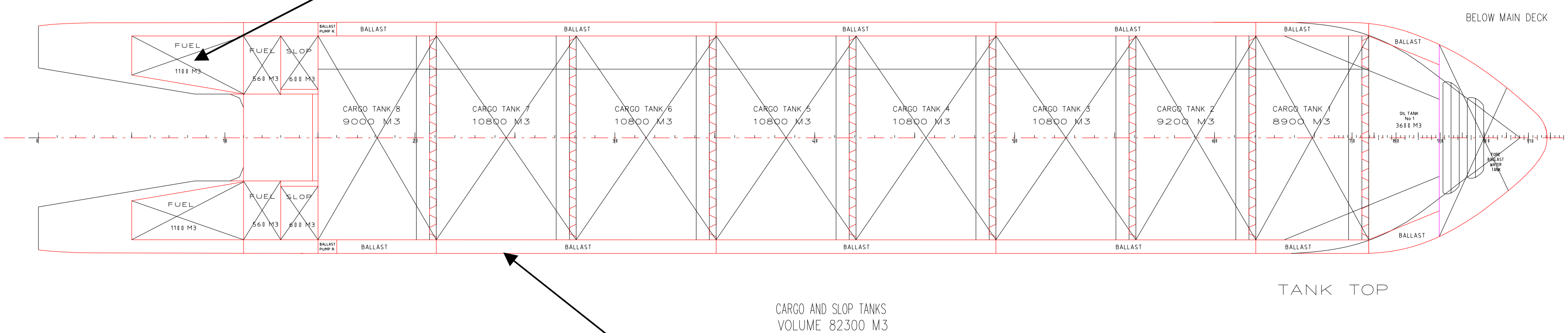
# Barge with Pusher

# Barge




Fuel tanks are located in barge



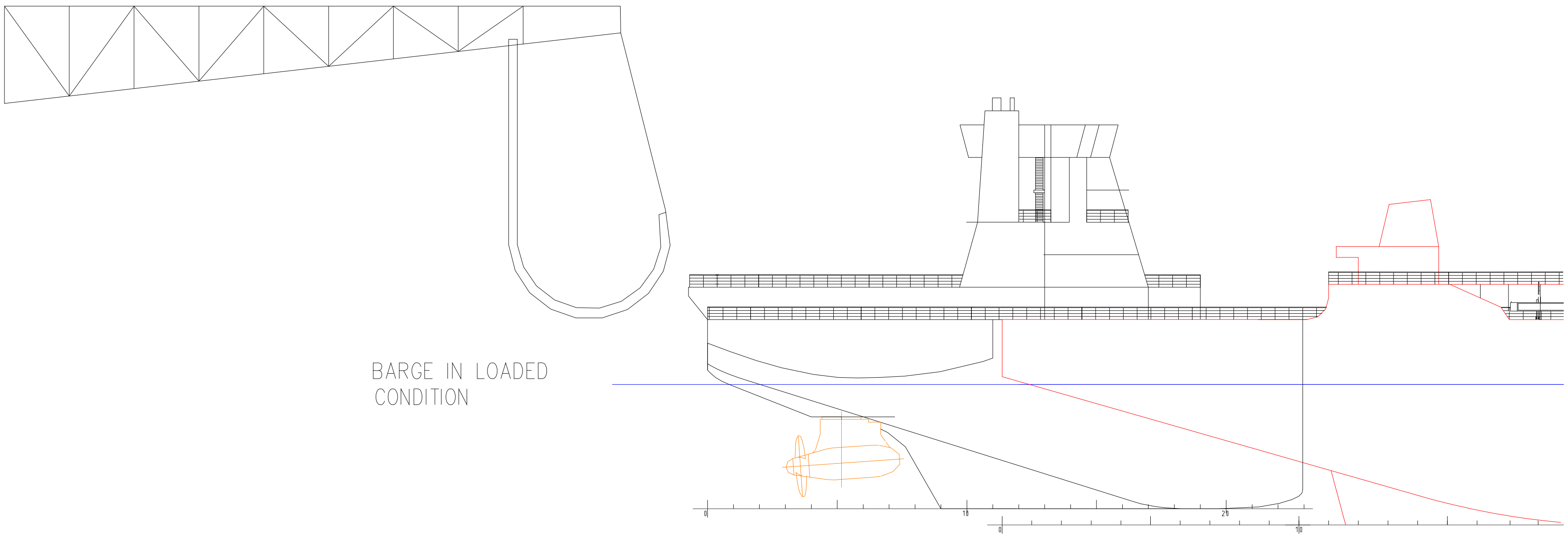

”Normal” ballast water capacity

# Loading from offshore terminal

The loading manifolds are only in ice pullers

→ Switch is made also in summertime



# Simulation

- Three different routes were simulated to investigate the transportation chain economy

1. Varandey – Murmansk (no pushers, only ice pullers)
2. Varandey – Murmansk – Rotterdam
3. Varandey – Murmansk – Port Fourchon (USA)

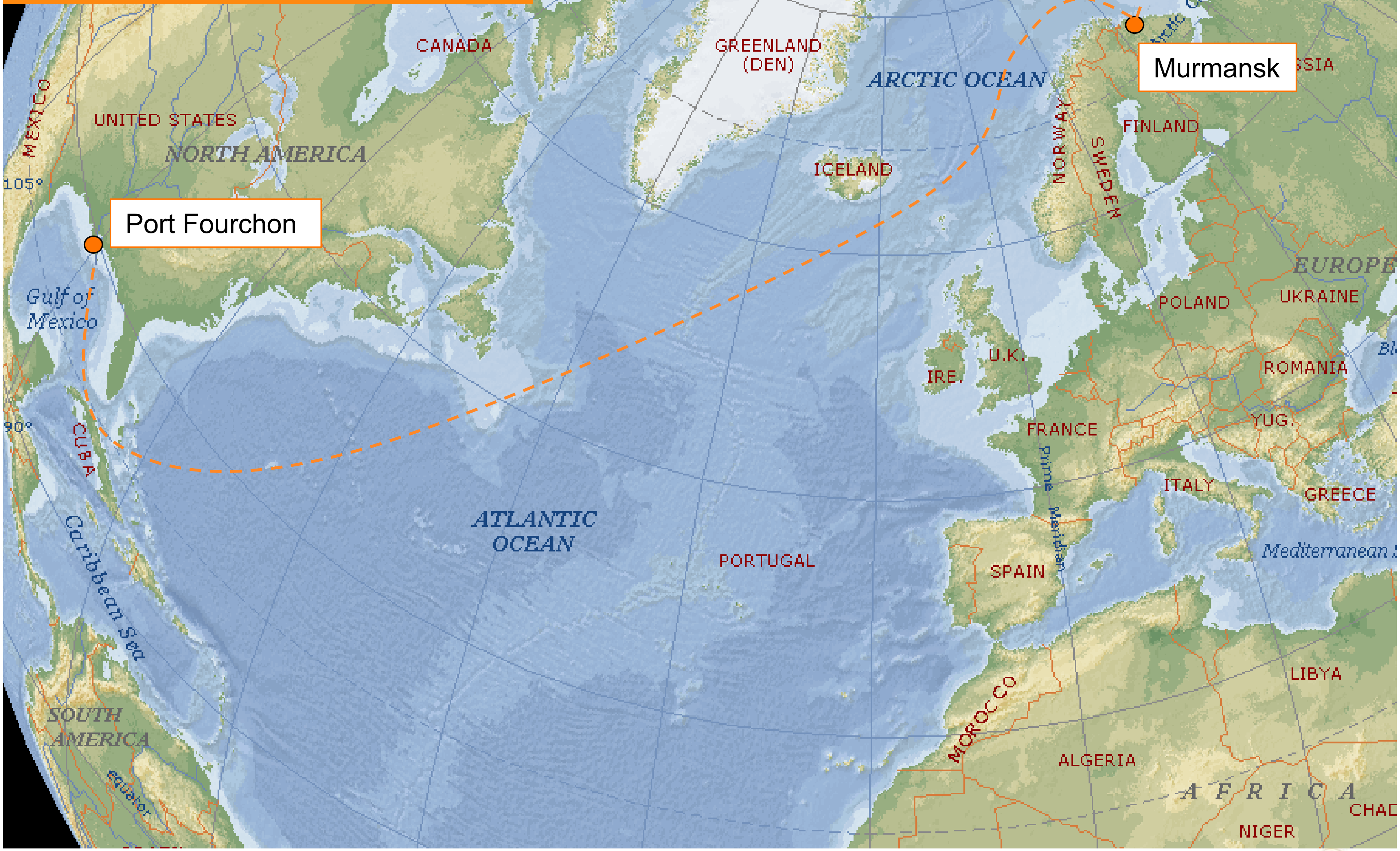
- The economy was compared to a 70 000 dwt Double Acting Tanker

  30 August 2007 Oskar Levander

# Simulation Routes 1 & 2

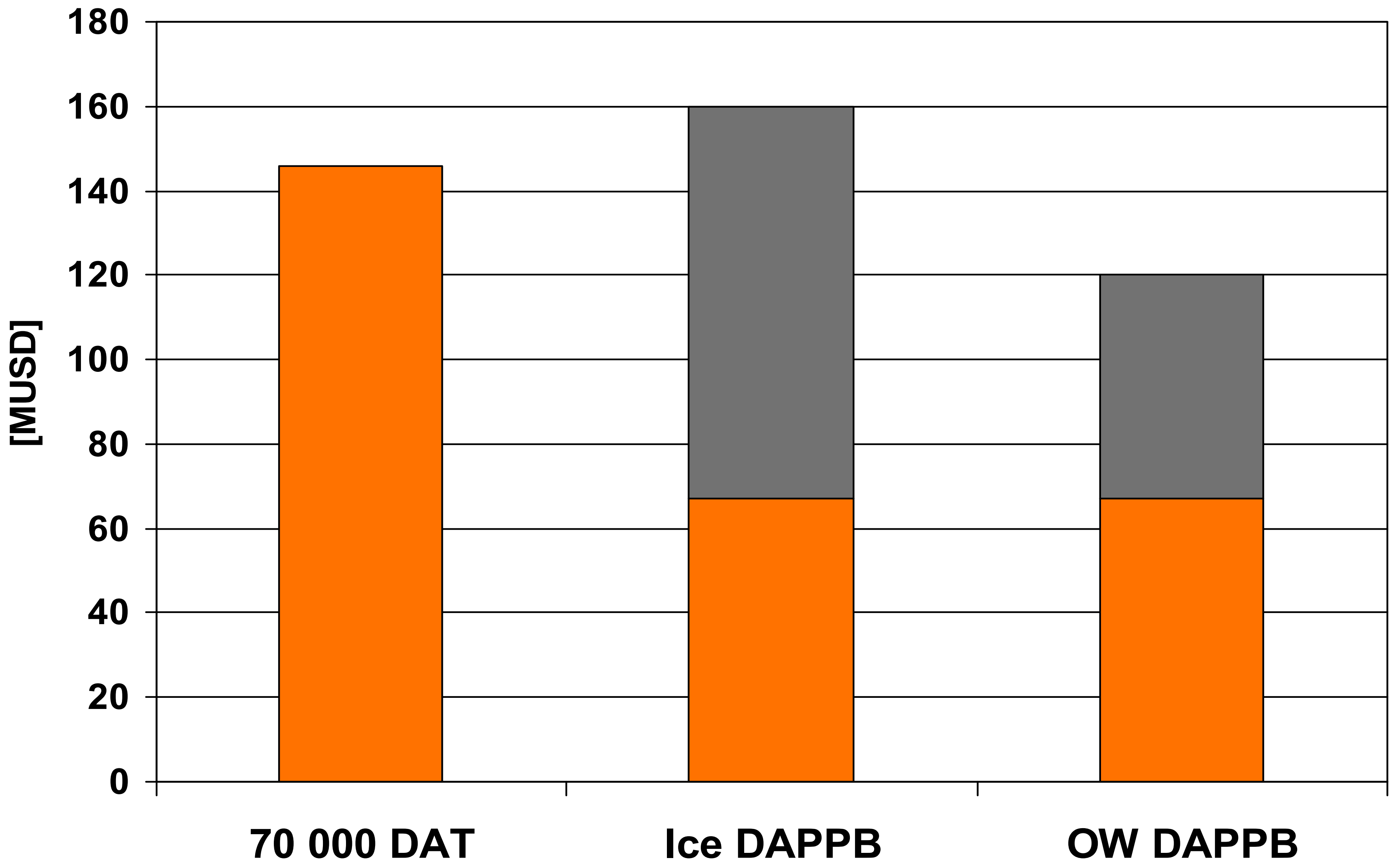

 

# Simulation Route 3



# Estimated investment cost



WÄRTSILÄ

# Simulation Parameters

| | DAPPB | DAT | |
|---|---|---|---|
| Deadweight | 70 000 | 70 000 | dwt |
| Cargo capacity | 69 700 | 69 700 | ton |
| Service speed (open water) | 15 | 15 | kn |
| Sea Margin | 15 | 15 | % |
| Engine power | 11/17 | 17 | MW |
| | | | |
| Cargo transported annually | 12.5 | 12.5 | Mton/y |
| Cargo loading/unloading time | 10.6 | 10.6 | h |
| Mooring time | 1.5 | 1.5 | h |
| Pusher – Puller changing time | 2 | - | h |
| Dry docking time | 14 | 14 | days |
| Dry docking period | 36 | 36 | month |

WÄRTSILÄ

# Simulation Parameters – Route 1

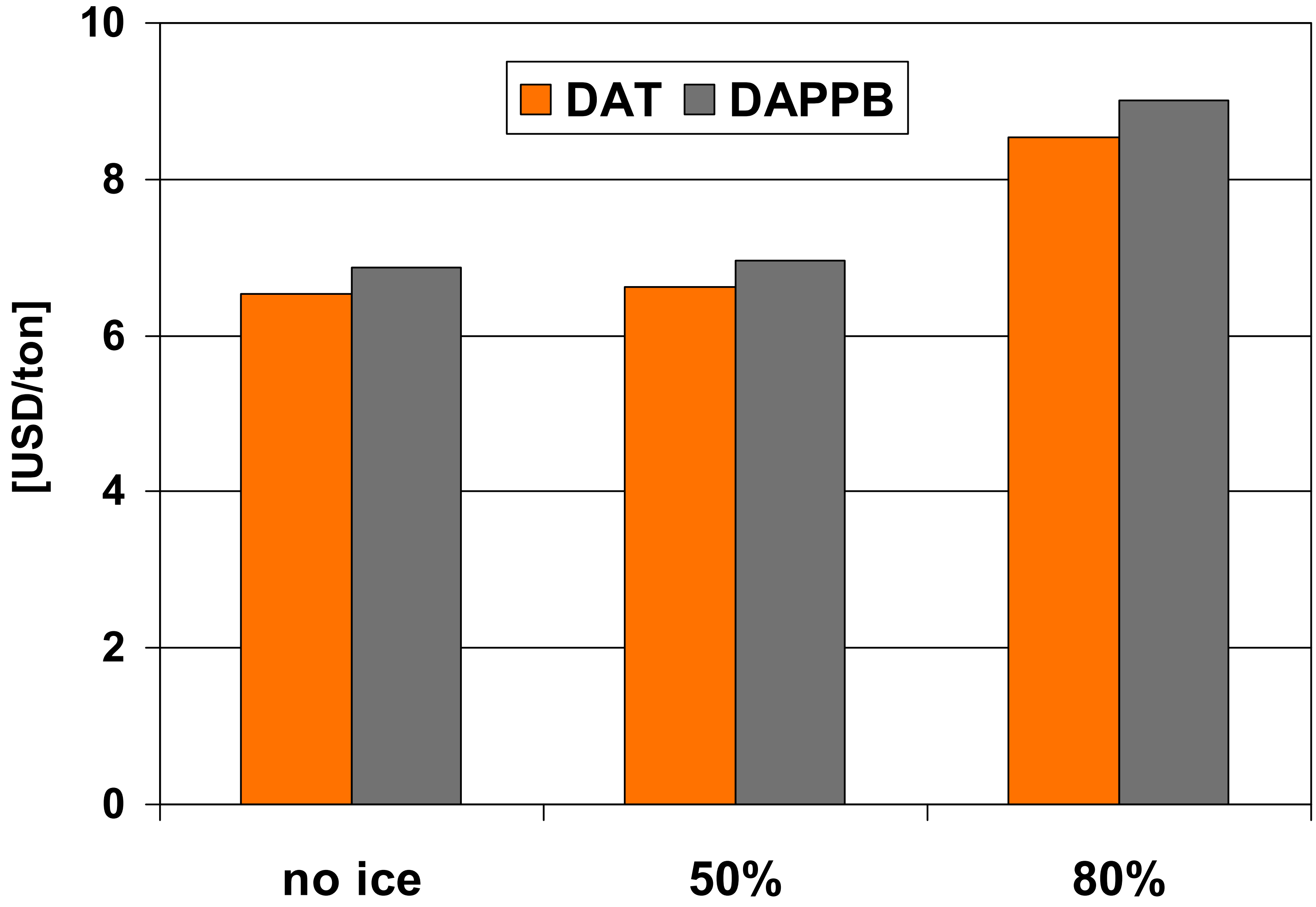
Varandey – Murmansk (no open water pushers)

| Winter type | no ice | 50% | 80% |
|---|---|---|---|
| **70 000 DAT** | | | |
| Number of ships in fleet | 3 | 3 | 4 |
| **DAPPB** | | | |
| Number of ice pullers in fleet | 3 | 3 | 4 |
| Number of barges in fleet | 3 | 3 | 4 |

WÄRTSILÄ

# Required Freight Rate – Route 1



# Simulation Parameters – Route 2

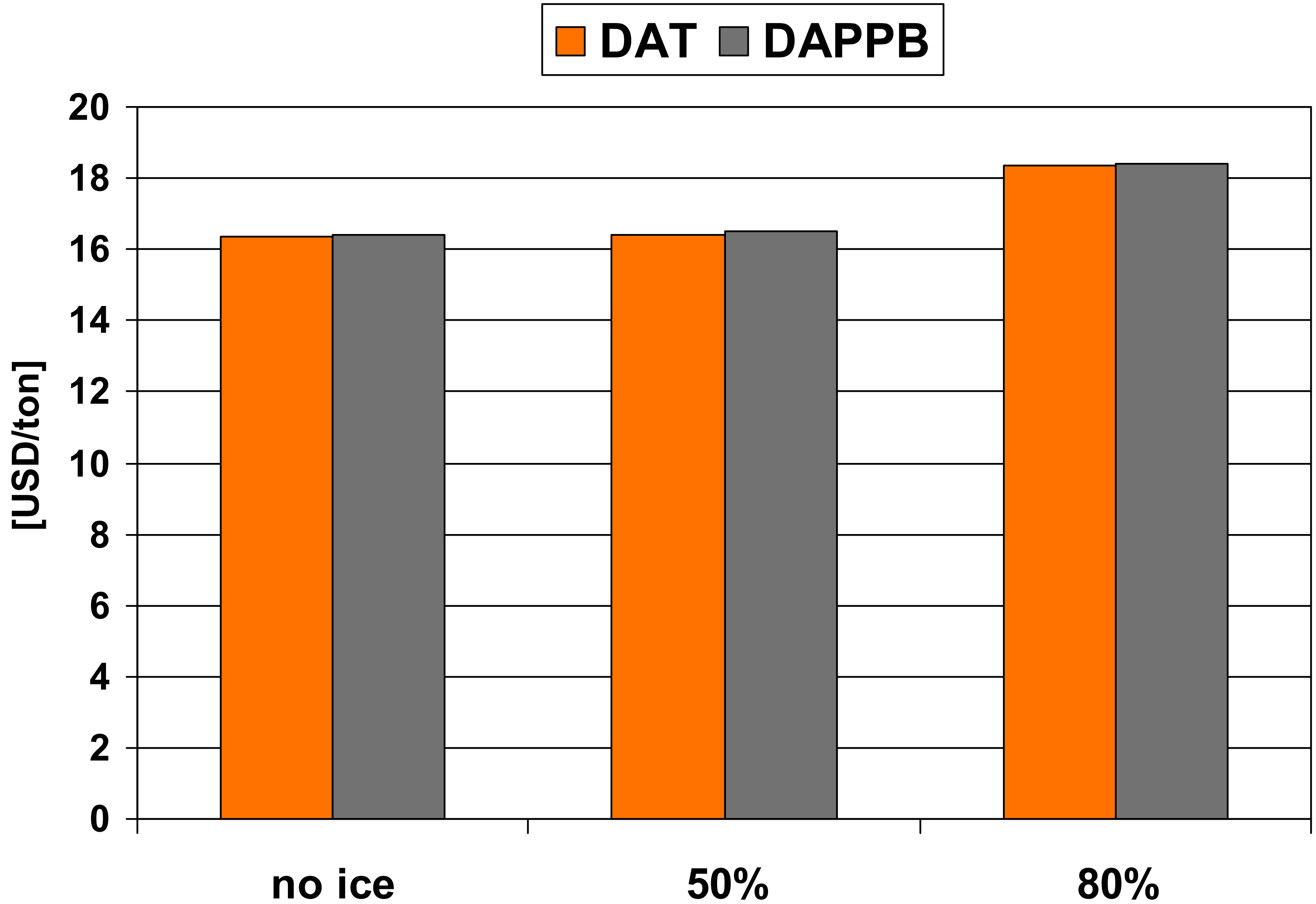
Varandey – Murmansk - Rotterdam

| Winter type | no ice | 50% | 80% |
|---|---|---|---|
| **70 000 DAT** | | | |
| Number of ships in fleet | 7 | 7 | 8 |
| **DAPPB** | | | |
| Number of ice pullers in fleet | 3 | 3 | 4 |
| Number of pushers in fleet | 5 | 5 | 5 |
| Number of barges in fleet | 8 | 8 | 9 |




WÄRTSILÄ

# Required Freight Rate – Route 2



# Simulation Parameters – Route 3

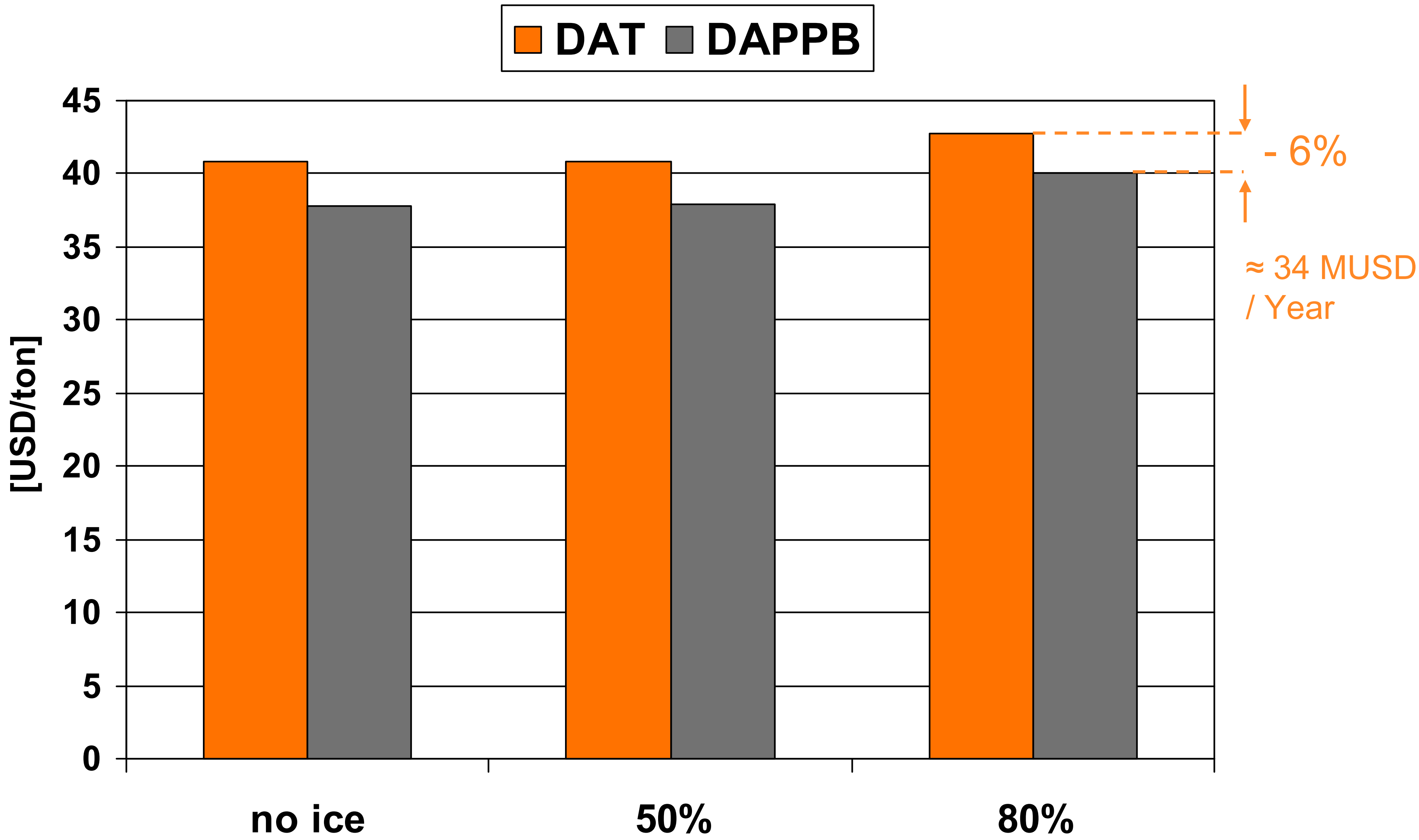
Varandey – Murmansk – Port Fourchon

| Winter type | no ice | 50% | 80% |
|---|---|---|---|
| **70 000 DAT** | | | |
| Number of ships in fleet | 17 | 17 | 18 |
| **DAPPB** | | | |
| Number of ice pullers in fleet | 3 | 3 | 4 |
| Number of pushers in fleet | 16 | 16 | 16 |
| Number of barges in fleet | 19 | 19 | 20 |

WÄRTSILÄ

# Required Freight Rate – Route 3



# Simulation Parameters – Route 3

DECIMAL COMPARISON

Varandey – Murmansk – Port Fourchon

| Winter type | no ice | 50% | 80% |
|---|---|---|---|
| **70 000 DAT** | | | |
| Number of ships in fleet | 16.3 | 16.7 | 18.0 |
| **DAPPB** | | | |
| Number of ice pullers in fleet | 2.2 | 2.5 | 3.8 |
| Number of pushers in fleet | 15.0 | 15.0 | 15.0 |
| Number of barges in fleet | 17.2 | 17.5 | 18.8 |

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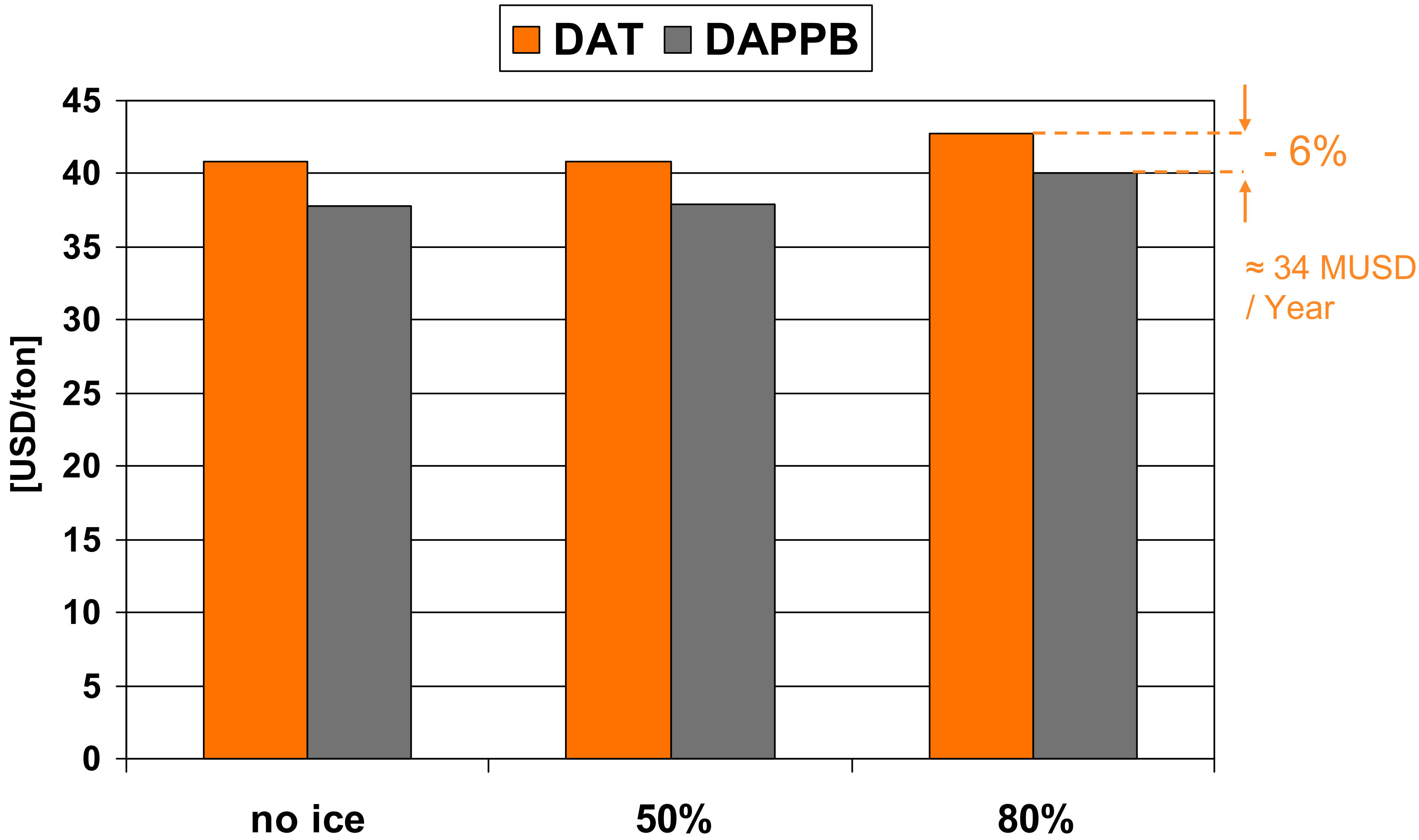
# Required Freight Rate – Route 3

DECIMAL COMPARISON



# Conclusion of simulation

- The DAPPB system is economically feasible at route that contains both ice and open water conditions
- When open water part exceeds abt. 1 600 nm, DAPPB concept is more economical compared to Double Acting ship
- At route to US (3 600 nm), the DAPPB gives 6% lower Required Freight Rate that corresponds 34 000 0000 € on annual basis

WÄRTSILÄ

# Thank You for Your Attention!

Oskar.levander@wartsila.com

# ICE LOAD MONITORING

**M. Mejlaender-Larsen** and **H. Nyseth**, Det Norske Veritas, DNV, Norway

## SUMMARY

With the coming growth in the transport of oil and gas in Arctic areas, DNV has increased the focus on safe ship operations in cold climate and ice infested waters. Studies have revealed the increased risk compared to world wide operation and identified additional risk elements related to the cold climate operation. One of the identified challenges has been the lack of information to the bridge about the actual load on the hull when operating in ice. Today the evaluation of the actual ice condition and corresponding load on the hull is mainly based on the navigators' judgement of what he can see from the bridge. Shifting ice conditions and long periods with darkness also reduces the ability to get a correct picture/overview of the actual condition. The consequences of lack of ice information may lead to loads on the hull exceeding the elastic capacity which may lead to permanent deformations of the hull structure that have to be repaired.

DNV is together with partners carrying out a project including design, installation and testing of a system for Ice Load Monitoring, ILM. The project is aiming to determine whether the selected technology is suitable for this purpose, and during the project learn about the systems possibilities and limitations. The system is based on 66 fibre optic sensors located inside the hull structure in the bow section and an electro magnetic device for measuring the actual ice thickness. The Norwegian Coast Guard vessel KV "SVALBARD" was equipped with the ILM instrumentation and tested in the Barents Sea. The test voyage was carried out in March 2007.

## 1. INTRODUCTION

With the expected increase in demand for oil and gas and decrease in production in traditional producing areas in the period 2010-2020, there is an enhanced pressure on developing the Arctic region. It is reported that a quarter of the world's total undiscovered petroleum resources could lie in the Arctic. Thus, a significant increase in oil and gas transport is expected from areas with ice covered waters and extreme low temperatures.

Today there is export of oil from the Barents Sea, White Sea, Pechora Sea, Sakhalin and Baltic Sea. Transport of oil from Arctic areas north of Canada and Alaska is mainly trough pipelines.

The Arctic is now experiencing some of the most rapid climate changes on earth. Melting of sea ice may also open entirely new possibilities in the future with respect to new shipping routes and extended use of existing routes.

The extreme conditions with ice and very low temperatures will put stricter requirements to the ships, equipment and crew than normally required. The Arctic areas are defined as particular sensitive, and the vision of zero discharge is commonly accepted and different organizations and authorities are preparing for the increased activity. Increased focus in IMO, EU and other Arctic organisations working with environmental issues and safe operations will influence on the operation of the vessels.

It is therefore of great importance to develop good, innovative and cost effective solutions to improve environmental and maritime safety. There are today several ongoing projects supported by IMO, EU and ESA working to develop and apply modern maritime electronic solutions. The use of electronic charts, information from vessel traffic control centres (VTS), and a more active use of Automatic Identification Systems (AIS) are tools being developed for increasing the safety of navigation.

Based on a study of different risk elements involved when operating in ice, one of the main challenges found, was to assess the actual load on the hull when operating in ice. Even officers with long experience expressed uncertainties with regard to the actual load when operating in different ice conditions. With more precise information about the actual load on the hull, the officers can adjust the speed and course in order to reduce the risk for damages to the hull.

The ILM system will also include detailed information about the weather and ice conditions displayed on the electronic chart at the bridge, (ECDIS) to be used for route planning before and during the voyage. The information about the ice will be based on satellite images and merged with the existing meteorological information. If this system is used in connection with a shuttle traffic along the same route, then the information recorded in one ship can be used to update the information for the next vessel to carry out the voyage.

The information will also enable the officers to operate the ship with correct speed under different ice conditions which will result in a more effective operation. Improved regularity and reduced cost for repair by avoiding damages to the hull will be the profit for the owner and operator.

The main challenge is therefore to gather information about the actual ice load condition and to operate the ship within the strength and operational limits of the vessel, i.e. within the limitations of the given Ice Class. Experience has shown that vessels operating in ice are exposed to damages on both hull and propeller, and the main cause is assumed to be that the ship is operated in more severe ice conditions than designed for.

A prerequisite for reducing this type of damages, which in worst case can lead to total loss of the vessel, is to get information about both the actual ice condition and the load the hull is exposed to. The aim is to adjust the speed and course according to the actual ice conditions to avoid possible damage to life, property and environment. The geographically remote location of these areas and extreme meteorological conditions will require special designed ships and equipment, as well as crew (skills) to reduce the operational risk.

The ILM project is sponsored by the Norwegian Research Council and headed by Det Norske Veritas, DNV. Other partners are:

- Light Structures AS
- C-MAP Marine Forcast
- The Norwegian Meteorological Institute, met.no
- TEEKAY
- STATOIL
- Norwegian Coastguard

## 2. THE ICE LOAD MONITORING SYSTEM

### 2.1 DESCRIPTION OF SYSTEM

The Ice Load Monitoring (ILM) system is built up of different components which are described in the following chapters. The system is schematically shown in figure 1 and described more in detail below.

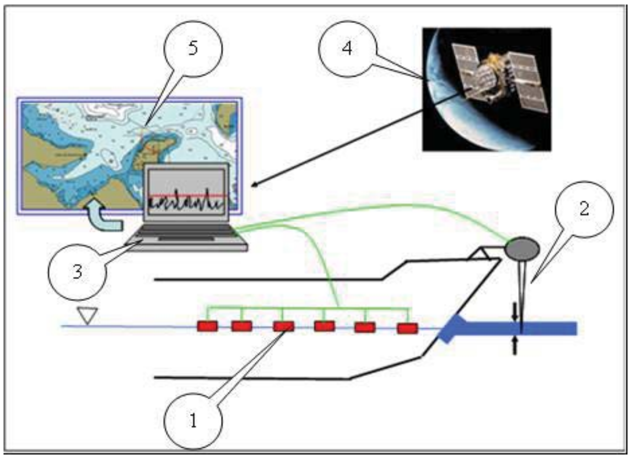



Figure 1: The Ice Load Monitoring system

The system to be mounted onboard contains the following items:

1. Strain sensors to measure the shear at the frames, i.e. actual structural response of local members exposed to ice load. The sensors are mounted on the frames in a limited area in the bow area.
2. Electro Magnetic ice thickness measurement equipment.
3. Computer and software to analyse and display measured data at bridge.
4. Utilize meteorological and satellite data and apply these data on the electronic chart.
5. Display and update the ice information and forecast continuously.

### 2.2 FIBRE OPTIC STRAIN SENSORS

The applied fibre optic strain sensors are based on Fiber Bragg Gratings (FBG) and come with individual temperature compensation. They are approved for mounting at up to 6 bar of water pressure. Their relative small size means that they are easily installed on girders and stiffeners in all parts of the hull. Mounting the sensors on girders and stiffeners ensures that measurements are not contaminated by local vibrations caused by equipment on deck, etc.

Sensors are fastened with adhesives for optimal stress transfer from the hull surface and can be incorporated in the vessel's coating regime with an unbroken membrane covering the sensor and the surrounding metal.

Fibre optic sensors offer an electromagnetically passive and intrinsically safe solution with no electrical power outside the central data processing unit, which can be placed in equipment rooms near the bridge. Signals are not affected by electrical fields, and all cables can follow either signal or power cable paths.

The fibre-optic sensors have a number of advantages over the electrical alternatives, especially in harsh environments:

- High sensitivity
- Good resistance towards water and chemicals
- Signal is wavelength coded, and unaffected by the environment along the cable path
- Immunity toward electromagnetic interference
- Do not contribute to the total surrounding electromagnetic field
- EX-safe
- Multiplexing: Many sensors on a single cable

The sensor system is a combination of a light source, sensors and an analyzer that receives the optical signals from the sensors and converts them to a format suited for digital signal processing. The Light Structures FBG Analyzers are based on a scanning filter that gates the light from a broadband source. The FBGA determines the Bragg wavelength of each grating with high precision.

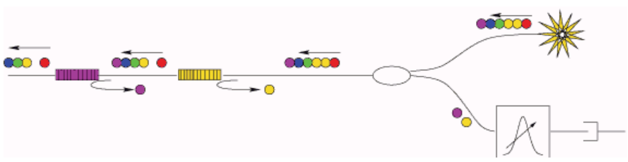


Figure 2: Principle of optic strain sensors

The equipment used for the strain measurements and signal analysis is delivered by Light Structures AS located in Oslo Norway, specialists in delivering equipment for hull monitoring. Additional information about the hull monitoring system may be found at http://www.lightstructures.biz/home.html

## 2.3 ELECTROMAGNETIC ICE THICKNESS MEASURING DEVICE

Electromagnetic (EM) sea ice thickness sounding has become one of the most powerful tools for systematic thickness profiling, both for climate studies as well as for engineering applications. Initial work using a surface-based EM induction system goes back to the 1970s and this work showed the effectiveness of this approach in principle. Additional work led to a system based on the Geonics EM-31 ground conductivity meter, since then being used on an operational basis on the ice surface as well as suspended from ship cranes. Due to its semi-regional applicability, ground based EM has been used to study changes in the regional sea ice thickness distribution.

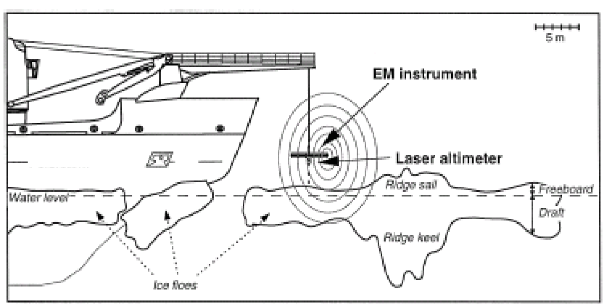



Figure 3: Operation of EM system in front of an icebreaker. Note that the laser altimeter can also be replaced by a sonic distance meter.

In general, EM sounding is used to measure the electrical conductivity structure of the underground. An EM instrument generates a low-frequency (e.g. 9.8 kHz) EM field, which penetrates into the underground. Here, the underground is composed of a sea ice layer above a deep sea water layer. As the conductivity of sea ice is very low (only between 0 and 50 mS/m), the EM field penetrates the ice layer almost unaffectedly into the underlying sea water. In sea water electrical eddy currents are induced due to its high conductivity of 2400 to 2800 mS/m. In turn, these eddy currents result in the induction of a secondary EM field, which is sensed by the EM instrument. The strength of the secondary EM field is directly related to the conductivity of the sea water layer, and to its distance to the EM instrument. When the EM instrument rests on the ice surface, the measured secondary EM field strength is thus directly a measure of ice thickness.

As EM thickness measurements can be performed without a direct contact to the ground, they can also be applied from an icebreaker while steaming through ice, thus providing continuous, along-track ice thickness information. This is particularly valuable for ship-in-ice studies and ice load monitoring.

A ship-based EM ice thickness system is composed of two instruments, an EM instrument to measure the distance between the EM system and the water surface $d_{EM}$, and a laser or sonic altimeter to determine the height of the EM system above the ice surface ($d_{Laser}$; Figure 4). Ice thickness $Z_i$ is then obtained as the difference between those two measurements:

$$Z_i = d_{EM} - d_{Laser}$$

Figure 4 also shows $d_{Instr}$, which is the vertical distance between the EM instrument and the altimeter, a technical specification of the mounting / suspending construction. However, note that consideration of $d_{Instr}$ is only important if the data are to be compared with results from numerical modelling of the EM response. $d_{Instr}$ is irrelevant when the EM measurements are calibrated over open water. For ship-based measurements, the EM instrument should be operated in vertical magnetic dipole (VMD) mode, as this result in the highest sensitivity.

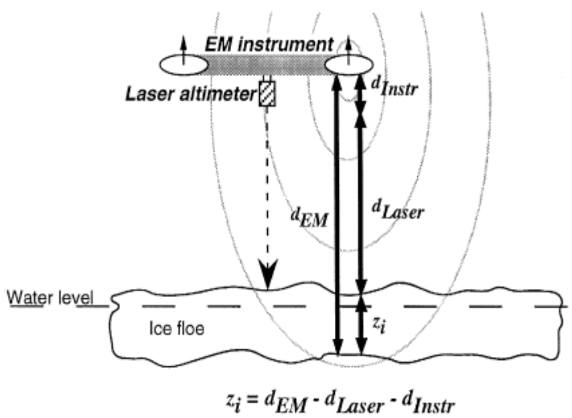



Figure 4: Illustration of ice thickness $Z_i$ computation from measurements of EM system height above the water ($d_{EM}$) and ice surfaces ($d_{Laser}$).

It should also be noted that $Z_i$ is the total ice thickness, i.e. the sum of ice plus snow thickness. So far there is no means to measure snow thickness independently and coincidentally while the ship is steaming. Additional snow thickness radar added to the instrument package would have potential to tackle this issue.

## 2.4 DATA ACCURACY OVER LEVEL ICE

The accuracy of the EM ice thickness measurements depends on the sensitivity of the instrument on changes in ice thickness, on the accurate knowledge of ice and water conductivity, and on the validity of the assumption that the ice is actually level. Even in the absence of pressure ridges and deformed ice it is shown that due to the special situation in front of the ship problems due to the footprint of the measurement can cause problems.

The sensitivity of the measurements depends on the actual ice thickness and instrument height above the water surface. The sensitivity decreases with increasing instrument height above the water. I.e. at a larger height or greater ice thickness, a certain ice thickness results in a smaller signal than at low instrument heights or with thin ice. For example, instrument heights of 4 m (above the ice surface) ice with a thickness of 2 m can be measured with an accuracy of 0.1 m.

## 2.5 ACCURACY LIMITATIONS

In front of the ship, ice conditions are normally very variable, as small floes are passed or broken, and because the bow is very often over floe edges. Due to the footprint of the EM measurements, which is between 12 and 16 $m^2$ for instrument heights of 4 m, this can result in deviations of the ice thickness retrieval from the real ice thickness.

Pressure ridges are another condition which will create inaccuracies of EM profiling. Result from measurements over deformed ice, will due to the footprint underestimate the actual thickness.

Another problem of measurements over ridges is the large porosity of ridge keels due to the mixture of unconsolidated ice blocks and water. The porous nature of ridge keels results in a highly increased conductivity, which invalidate the assumption of negligible ice conductivity, and therefore make simple thickness retrieval impossible. Consequently, also the measured apparent conductivity is much higher than what would be expected for solid level ice, leading to large thickness underestimates.

On the other hand, for engineering studies often the true ridge thickness is irrelevant, because ice blocks are only loosely consolidated and do not exert strong ice forces when broken by an icebreaker or structure.

Onboard KV “SVALBARD” the EM device is placed in the end of a wooden beam in front of the bow. The beam is made of wood to avoid any magnetic interference. Figure 5 below shows the EM device in front of the vessel. The horizontal distance from the bow is 6 m.

The EM device mounted in the bow of KV “SVALBARD” is provided and operated by Alfred Wegener Institute (AWI) and Pfaffling Geophysics.

Figure 5: EM device mounted in the bow of KV “SVALBARD”

## 2.6 SATELLITE AND METEOROLOGICAL INFORMATION

WeatherView™ from C-MAP Marine Forecast was installed onboard and used during the voyage. Information about the ice conditions was downloaded from Met.no, the home page of the Norwegian Meteorological Institute in addition to Satellite images from other suppliers which were used for route planning and validation. A part of the project will be to merge the weather and ice information and present it as a single source of information at the ECDIS, to be used for optimal and safe route planning.

## 2.7 DISPLAY AT BRIDGE

A screen is located at the bridge to display the estimated utilisation of the hull structure as well as all the other measured parameters. Both the instant values and statistical values are available. A separate window showing the time history and trends of different parameters and the correlation of different parameters can also be displayed. The parameters are displayed in real time and can be used for displaying the ice thickness and corresponding utilization factors for the different sensors. Figure 6 shows an example of the display presented at bridge.


Figure 6: Example of display at bridge

## 3. KV SVALBARD – TEST PLATFORM

As the Norwegian Coast Guard is a partner in the project, it was decided to instrument and use KV "SVALBARD" for testing of the ILM system. KV "SVALBARD" is a coast guard vessel designed for operation in ice with the DNV notation POLAR-10 ICEBREAKER. The vessel is 103 m long with a displacement of 6500 ton. The beam is 19.1 m and the draft is 6.5 m. The vessel operates as a coast guard vessel in the Barents Sea and around Svalbard islands. Even though the vessel's main tasks are related to uphold of sovereignty, fishery inspections, search and rescue etc, the vessel is also used for Arctic research cruises every year.


Figure 7: The KV "SVALBARD"

## 4. APPLICATION OF STRAIN SENSORS

As mentioned above, 66 fibre optic sensors have been mounted onboard KV "SVALBARD". The strain measuring arrangement is based on spot checks of critical frames mainly in the bow area. The locations of the measured frames are shown in Figure 8. A total of nine frames are instrumented. The measuring arrangement is designed to provide information of the actual response and the corresponding residual strength of plates and frames forming the hull structure directly exposed to ice loads. A typical sensor arrangement mounted on the frames is shown in Figure 9.

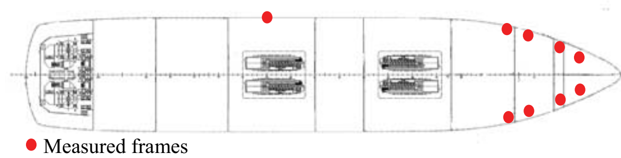



Figure 8: Positions of strain sensors

Figure 9: Mounting of fibre optical sensors

The basic monitoring arrangement is based on the assumption that the measured deformations of the frames can be extracted and converted into known response parameters as a measure of the total force acting on the structural member. In this case, the primary parameter to be determined is the shear response at the upper and lower supports of the frames, which, based on shear difference principles, gives an indication of the total integrated force acting on the frame.

Based on finite element analysis, a procedure has been developed to convert the measured response signals into predefined response patterns to be further used for the structural utilisation assessment. As the internal stiffening of the ice-reinforced bow structure is very complex in addition to the natural randomness of the ice loads, the structural response is not explicitly determined and the procedure has to be developed accordingly.

The utilisation assessment is based on the assumption that the calculated load may be located at any parts of the surrounding structure which are exposed to ice loading. The ultimate strength including relevant "safety" factors is calculated for each individual structural member. The calculated force from the measured response is compared to the ultimate strength of the frames or plates, whichever is the smaller, giving an indication of the actual utilisation of the given strength members, as given below:

$$\eta_i = \frac{F_{calc,i}}{C_i}$$

where

$\eta_i$ = usage factor for shell plate/framing, frame i

$F_{calc,i}$ = Calculated force acting on frame i

$C_i$ = Predefined capacity of shell plate/framing, frame i

Numerous finite element analyses of the framing structure are carried out to investigate the frame response for different load combinations. Based on these analyses, algorithms for recognition of load patterns/locations and corresponding responses have been established as basis for the further response assessment.

The response assessment procedure may be summarized in figure 10.

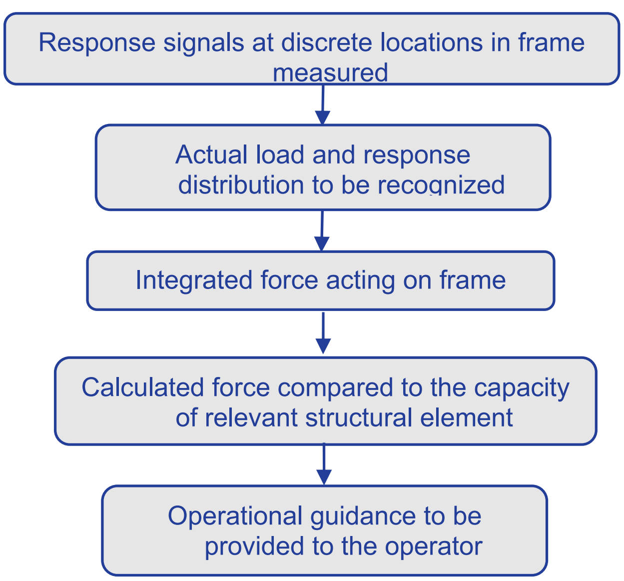



Figure 10: Procedure for determination of structural utilisation

In addition to the basic arrangement mounted on the frames, additional sensor packages are mounted to provide added information of the load distribution and extensions, temperature distributions and actual frame response. These additional sensors may act as basis for any further improvement of the measuring system in the future.

## 5. PRELIMINARY RESULTS

A few examples of the results from the first full-scale measurements recorded are shown below. The first data set covers 10 days of ice going service, including level ice, first year and multi year ice. As the vessels operational limitations mainly are believed to be the lack of propulsion power rather than structural capacity, the structural responses measured are also seen to remain well below their predefined limits.

Figure 11 shows the response calculated for a given frame in the bow region relative to the predefined capacity of the structure. The curve consists of 5 minutes statistical maxima during 24 hours operation as the vessel was entering harsher ice conditions.

Figure 12 shows one hour recording of the structural response at the same day. The time history of the maximum peak pulse occurring at about 09:05 PM is shown in Figure 13. The sampling frequency is 678Hz.

In Figure 14 the structural response of the frame is compared to the actual measured ice thickness (5 min statistical maxima) during a 6 hour interval. The corresponding ice ridge thickness at the time of maximum response was measured to be about 5 m. The figure clearly shows the obvious correlation between the structural response and the actual measured ice thickness.

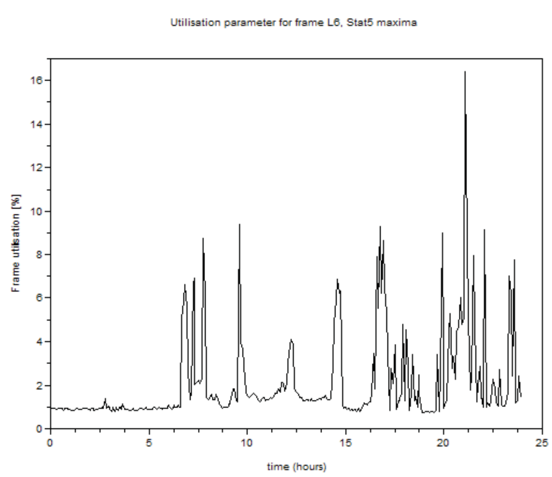



Figure 11: 24 hour recording of structural utilisation structural member in bow region

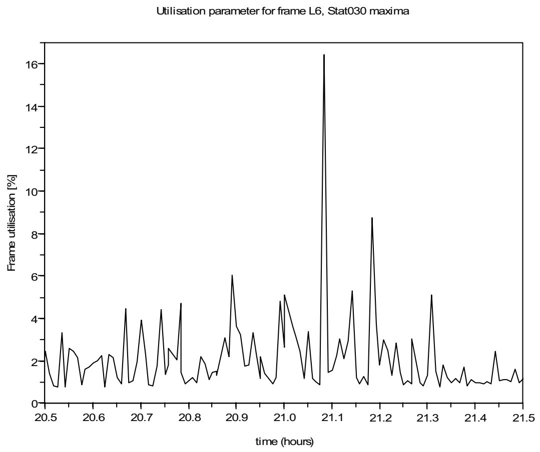



Figure 12: One hour recording of structural response, 30 seconds response maxima

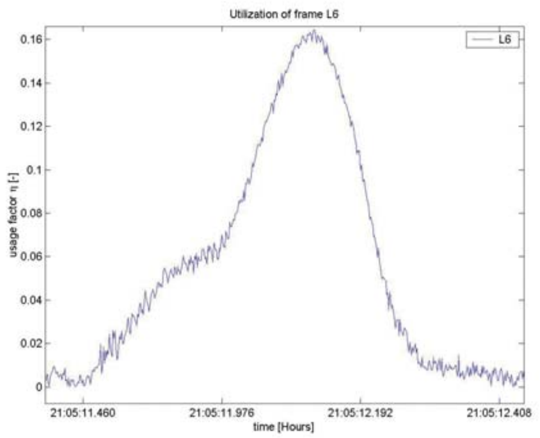



Figure 13: Load pulse at 2105 hr

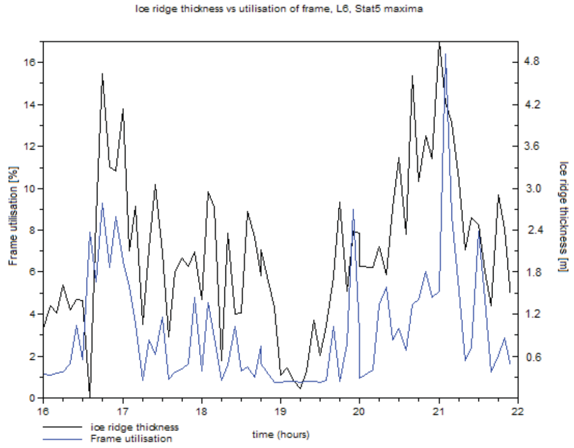



Figure 14: Calculated structural utilisation vs measured ice thickness

The next phase of the project will cover the in-depth analysis of the already recorded data and the data to be gathered during the next year, to determine whether the proposed monitoring system is suitable for this purpose.

An important factor is the response from the crew to the output shown at the bridge and the relation between the "sensed" impact and the system feedback.

## 6. CONCLUSION

This paper describes the different systems which are included in the Ice Load Monitoring Program (ILM). The objective of the project is to develop tools which can provide additional information of the actual ice conditions and the corresponding loads acting on the hull structure. The different systems should be integrated into a single source of information and act as a decision support system for safe and effective operation in ice infested waters. Up to the projects end in 2008 the system will be further evaluated and improved to see if the installed tools are suitable for this purpose.

Even though only selected parts of the recorded data have been analysed, the results clearly show that the instrumentation setup as mounted onboard KV "SVALBARD" has the potential of serving as a reasonable measure of the actual ice load to which the hull structure is exposed. A challenge with the monitoring system is to increase the efficiency of the system with regard to number of sensors etc. In addition, the ice and meteorological models will be further improved and incorporated into the existing systems.

One of the success criteria, and the ultimate intention of the ILM project, is to develop the tool to be a trustworthy and reliable system for the onboard personnel, with a feedback interface that renders it useful as a decision support tool with capability of giving instantaneous response pictures.

## 7. ACKNOWLEDGEMENTS

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## 9. AUTHORS' BIOGRAPHIES

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With the new focus on Cold Climate, DNV Maritime appointed Mr Mejlaender-Larsen as Focus area responsible for the cold climate activity in 2004. The activity includes R&D, rule development projects, JIP and other related activities.

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# SPACE AND TANK WINTERIZATION TECHNIQUES FOR VESSELS OPERATING IN COLD REGIONS

**J.W. Choi**, **M.J. Koo** and **M.K. Ha**, Samsung Heavy Industries Co. Ltd, Korea

## SUMMARY

To meet the increasing demands for vessels operating in cold regions, SHI (Samsung Heavy Industries) has contributed to the design and the construction of these vessels. In comparison with other ships, the main differences can be considered as winterization for outfitting systems to guarantee safe operation in cold environments.

In this study, a simplified evaluation technique has been introduced for thermal protection, based on the derivation of an overall heat transfer coefficients including suitable convection terms. It has been applied to evaluate the thermal protection performance in ballast tank and air-pressurized-vessel room and proved to be reasonable performance within 10%, comparing with the results by CFD (Computational Fluid Dynamics) using commercial code, FLUENT.

Since CFD technology has been widely used in various fields, anti-freezing mechanisms in ballast tank with an air bubbling system has been simulated to check the performance under -20℃ of ambient temperature. An imaginary variable coefficient of specific heat value was introduced to consider latent heat at the range of freezing point temperature. The simulation gives physically reasonable phenomena and confirms the applicability of air-bubbling system as an anti-freezing system for ballast tank.

## NOMENCLATURE

| | |
|---|---|
| $k$ | Thermal conductivity (W $m^{-1}$ $K^{-1}$) |
| $A$ | Area ($m^2$) |
| $T$ | Temperature (K) |
| $h$ | Convective heat transfer coefficient (W $m^{-2}$ $K^{-1}$) |
| Re | Reynolds number (dimensionless) |
| $u$ | Velocity (m $s^{-1}$) |
| $L$ | Characteristic length (m) |
| $\nu$ | Kinematic viscosity ($m^2$ $s^{-1}$) |
| $Gr$ | Grashof number (dimensionless) |
| $\beta$ | Coefficient of expansion ($K^{-1}$) |
| $U$ | Overall heat transfer coefficient (W $m^{-2}$ $K^{-1}$) |
| $e$ | Thickness of wall (m) |
| $Q$ | Heat transfer rate (W) |
| $\rho$ | Density (kg $m^{-3}$) |
| $t$ | Time (s) |
| $x$ | Displacement (m) |
| $p$ | Pressure (N $m^{-2}$) |
| $\tau$ | Stress tensor |
| $\mu$ | Viscosity (kg $m^{-1}$ $s^{-1}$) |
| $g$ | Gravity (m $s^{-2}$) |
| $\delta$ | Kronecker delta |
| $H$ | Enthalpy (J) |
| $S_h$ | Heat source (W) |
| $m$ | Mass (kg) |

## 1. INTRODUCTION

Market demand for vessels operating in cold regions has increased recently because of several reasons. One of the main reasons is high oil prices which could not have ever been seen before. High oil prices gave rise to the economic feasibility of the development of oil field in harsh regions, especially in cold regions, such as the North Sea, Sakhalin and the Arctic Sea. The second is increase of marine transportations passing the arctic routes such as the Barents Sea and the Kara Sea. These active marine transportations result from the rapid economic growth of Russia. The vessels operating in cold regions, therefore, have become interesting to the ship owners nowadays, and many orders of those vessels are predicted.

The main differences, compared with other ships in operating warm regions, can be considered as winterization for outfitting systems to guarantee safe operation in harsh environments with extremely low outside temperatures, strong winds and snow. According to many researches done by classification societies, one can easily find out the design guidance on the material selection and the thickness guidance of hull. However, the design tools for outfitting systems are not well known in the industry, except a general guidance note.

This paper introduces two approaches done by SHI to determine insulation thickness for spaces and anti-freezing mechanisms of an air bubbling system in a ballast tank. In this study, a simplified technique for insulation thickness, based on the derivation of overall heat transfer coefficients, is compared with the result by CFD (Computational Fluid Dynamics) using commercial code with FLUENT. The study proves that the overall heat transfer coefficients considering convective terms having the information of air flow velocity are crucial factors to reduce the estimation error of within ±10%.

The second approach is the investigation of anti-freezing mechanisms of an air bubbling systems using CFD. For this study, imaginary variable specific heat coefficients at the range of freezing point temperature have been introduced to consider latent heat due to phase change in numerical approach.

These design techniques, therefore, can be treated as useful design tools and also have been widely approved by engineering groups from classification societies and ship owners.

## 2. ESTIMATION OF HEATER CAPACITIES AND INSULATION THICKNESS FOR SPACES

### 2.1 THEORY

#### 2.1 (a) Conduction and Convection

Conduction is a mode of heat transfer in which energy exchange takes place from hot region to cold region by kinetic motion or direct impact of molecules, as in case of fluid at rest, and by the drift of electrons, as in case of metal. Heat transfer law states, the rate of heat flow by conduction is proportional to the area normal to the direction of heat flow and to the gradient of temperature. Heat flow due to conduction is the product of total heat transfer area, material thermal conductivity and temperature difference.

$$Q = kA\Delta T / e \tag{1}$$

When fluid flows over a solid body while temperatures of the fluid and the solid surface are different, heat transfer between the fluid and the solid takes place as a consequence of the motion of fluid relative to the surface. This mechanism of heat transfer is called convection. If the fluid motion is artificially induced by a fan etc., the heat transfer is said to be due to forced convection forces of the fluid flow over the surface.

If the fluid motion is set up by buoyancy effects resulting from density difference caused by temperature difference in the fluid, the heat transfer is said to be by free or natural convection. In engineering applications, a simplified equation is used to represent the heat transfer between a hot (cold) surface and a cold (hot) fluid.

$$Q = hA\Delta T \tag{2}$$

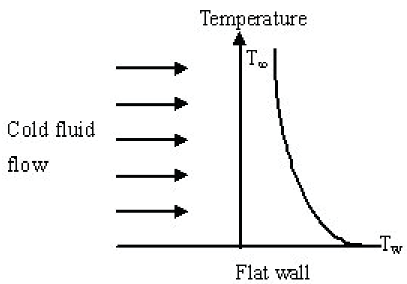


Figure 1: Heat transfer by convection from a hot wall to a cold fluid [1]

#### 2.1 (b) Dimensionless Parameters

Dimensionless parameters, such as the Reynolds number (Re) and Grashof number (Gr), imply the physical significances in the interpretation of the conditions associated with fluid flow or heat transfer.

The Reynolds number represents the ratio of the inertial forces to viscous forces. This implies that viscous forces are dominant for small Reynolds numbers and the inertial force are dominant for large Reynolds number. This parameter can be used to define laminar or turbulent flow and for the calculation of heat transfer in case of forced convection.

$$\mathrm{Re} = \frac{uL}{\nu} \tag{3}$$

The Grashof number represents the ratio of the buoyancy force to the viscous force acting on the fluid. The Grashof number in free convection enacts the same role as the Reynolds number in forced convection. As referred, in forced convection the transition from laminar to turbulent flow is governed by the critical value of the Reynolds number. Similarly, in free convection, the transition from laminar to turbulent flow is governed by the critical value of the Grashof number. Heat transfer, in case of natural (free) convection can be characterized by the Grashof number.

$$Gr = \frac{\rho g \beta L^3 \Delta T}{\nu^2} \tag{4}$$

#### 2.1 (c) Overall Heat Transfer Coefficient

Heat flow can be usually considered equivalent to electric flow. The reciprocals of convective and conductive heat transfer coefficients are the thermal resistances derived from the fundamentals of electronic resistance. Hence the heat flow through the wall can be demonstrated as per the following figure.

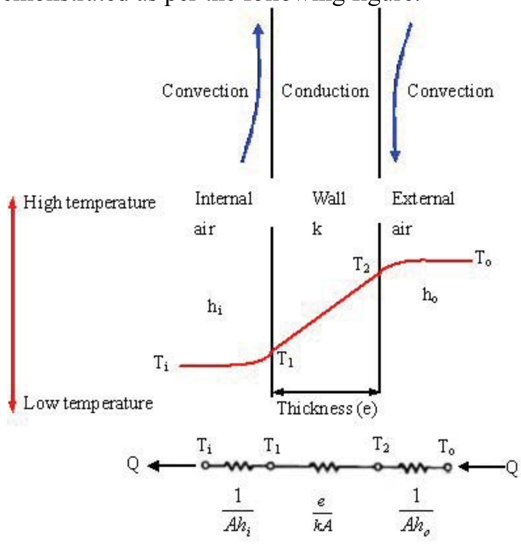


Figure 2: Schematics of thermal resistance Law of energy conservation summarizes to,

$$Q = h_i A(T_1 - T_i) = \frac{k}{e} A(T_2 - T_1) = h_o A(T_o - T_2) \tag{5}$$

For above equations, it is useful to demonstrate heat flow related to temperature components $T_i$ and $T_o$ which are often known values in the heat system required to be analyzed. So, coming equation is useful to calculate heat flow between a concerned space and adjacent spaces.

$$Q = UA(T_o - T_i) \tag{6}$$

where, $$U = \frac{1}{\frac{1}{h_i} + \frac{e}{k} + \frac{1}{h_o}} \tag{7}$$

Here, $U$ is called as overall heat transfer coefficient and its unit is $W / m^2 K$.

## 2.2 HEAT EQUILIBRIUM AND CALCULATION PROCEDURE

As per this law, the specific space will reach heat equilibrium with surroundings. This balance ultimately refers to the energy conservation. This heat equilibrium can be preserved unless change of flow and thermal conditions.

The heat equilibrium of the specific space can be calculated as next turns. Initially, the temperature of each boundary space contacted to the specific space is assumed as certain value. All convective heat transfer coefficients of walls passed by heat flow are calculated as per initial conditions of spaces. The overall heat transfer coefficients of each wall can be evaluated by means of combinations of calculated convective heat transfer coefficients and given conductive coefficients. So, the temperature of the specific space can be solved under given conditions of temperatures of boundary spaces and overall heat transfer coefficients. This procedure will be repeated until the energy sum for the specific space is to zero.

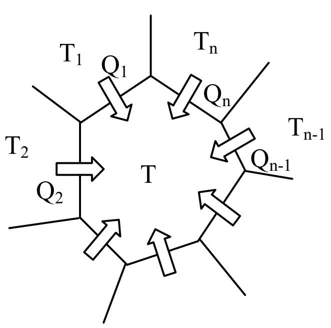


Figure 3: Heat flow for a space [2]

Heat transfer rates and temperatures from each boundary space are,

$$Q_i = U_i A_i (T - T_i) \tag{8}$$

$$T_i = \frac{U_i A_i T - Q_i}{U_i A_i} \tag{9}$$

By heat equilibrium,

$$\sum_n Q_i = 0 \qquad \text{for } i\text{=1 to } n \tag{10}$$

This can rewritten as

$$U_1 A_1 (T - T_1) + U_2 A_2 (T - T_2) + \cdots + U_n A_n (T - T_n) = 0 \tag{11}$$

So, the temperature of specific space can be obtained,

$$T = \frac{U_1 A_1 T_1 + U_2 A_2 T_2 + \cdots + U_{n-1} A_{n-1} T_{n-1} + U_n A_n T_n}{U_1 A_1 + U_2 A_2 + \cdots + U_{n-1} A_{n-a} + U_n A_n} \tag{12}$$

## 2.3 APPLICATIONS

Developed thermal calculation was used to design the capacity of heating coils of a ballast tank for arctic shuttle tanker. The design specification stated that the temperature of tank is maintained to 2℃ under outside air temperature of -45℃ and sea water temperature of -2℃. The overall heat transfer coefficients of heating coils and each area could be calculated with developed approach. Figure 4 illustrates the simplified approach can give less than 10% over-estimation in heater capacity, comparing with the results from CFD.

Actual temperatures according to the selected heater capacity were estimated by CFD. Four cases were simulated as per installation positions and arrangements for heating coils. Case 1 is that the position of heating coils is just below sea level and heating coils are arranged horizontally. Case 2 is that the position of heating coils is just below sea level and heating coils are arranged vertically. Case 3 is that the poison of heating coil is a just below full ballast level and heating coils are arranged horizontally. In the end, Case 4 is that the same position is as Case 3 but, heating coils are arranged vertically. Figure 4 illustrates the temperature differences are less than 0.14℃ deviation from the design value.

The second approach in this paper is APV (Air Pressurized Vessels) room with many air pressurized vessels and nitrogen receiver including several equipments for a drillship in the North Sea. Even though the ambient conditions of ocean, where the vessels operates, are very cold, temperature in the room have to be controlled over 0℃ for smooth operation of nitrogen receiver. The considered boundary conditions are -20℃ of ambient temperature and 5knots of wind speed.

Figure 5 shows the simplified approach can be about 10% under-estimation in heater capacity.

Figure 4: Thermal calculation of a ballast tank

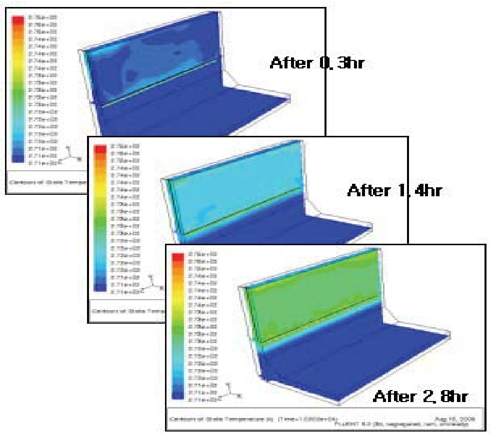


Figure 4 (a) Temperature distribution by increasing heating time (Case 1 in Figure 4(b))

Temperature

| | Expected ballast tank temperature (℃) | | Difference |
|---|---|---|---|
| | Design | CFD | |
| Case1 | 2 | 2.14 | ↑0.14 |
| Case2 | 2 | 2.12 | ↑0.12 |
| Case3 | 2 | 2.08 | ↑0.08 |
| Case4 | 2 | 2.13 | ↑0.13 |

Heater capacity

| | Expected inside ballast tank heating capa. (W) | | Difference |
|---|---|---|---|
| | Design | CFD | |
| No. 2~4 W.B.TK. | 194,572 | 178,880 | 8.8% |

Figure 4 (b): Tables of temperature and heater capacity

Actual temperature differences were estimated in two cases according to the heater positions. In Case 1, heater is located at the corner, for Case 2, operating heater is at the middle of line in one side of wall. The temperature deviations in Figure 5 show about 2℃.

## 3. ANTI-FREEZING MECHANISMS OF AN AIR BUBBLING SYSTEM FOR A BALLAST TANK

### 3.1 NUMERICAL METHOD

#### 3.1 (a) Governing Equations and Turbulence Modelling

The equations for conservation of mass or continuity equation, momentum conservation and energy conservation can be written as follows to describe the transport phenomenon of fluid and heat. [3]

<u>Continuity equation</u>

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \qquad (13)$$

Figure 5: Thermal calculation of APV (Air Pressurized Vessels) room

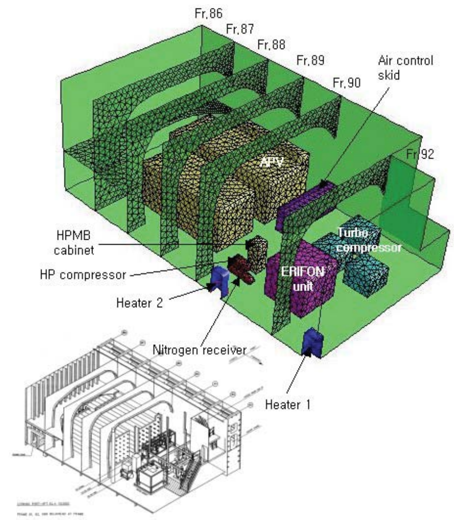


Figure 5 (a) Schematics and arrangements of APV

Temperature

| Cases | Operating heater | Expected inside room temperature (℃) | | Difference |
|---|---|---|---|---|
| | | Design | Simulation | |
| Case 1 | Heater 1 | 0 | 2.14 | ↑2.14 |
| Case 2 | Heater 2 | 0 | -0.57 | ↓0.57 |

Heater capacity

| | Expected inside ballast tank heating capa. (W) | | Difference |
|---|---|---|---|
| | Design | Simulation | |
| APV ROOM | 21.4 | 23.6 | 9.3% |

Figure 5 (b) Tables of temperatures and heater capacity

<u>Momentum conservation equation</u>

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i \qquad (14)$$

$$\tau_{ij} = \left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] - \frac{2}{3}\mu\frac{\partial u_i}{\partial x_i}\delta_{ij} \qquad (15)$$

Where $\tau_{ij}$ is the stress tensor, the indices $i$ and $j$ in $\tau_{ij}$ indicate that the stress component acts in the $j$ direction on a surface normal to the $i$ direction, $\delta_{ij}$ is Kronecker delta ( $\delta_{ij}$ =1 if $i$=$j$ and $\delta_{ij}$ =0 otherwise).

Energy conservation equation

$$\frac{\partial}{\partial t}(\rho h) + \frac{\partial}{\partial x_i}(\rho u_i h) = \frac{\partial}{\partial x_i}(k\frac{\partial T}{\partial x_i}) + \frac{\partial p}{\partial t} + u_i \frac{\partial p}{\partial x_i} + \tau_{ij}\frac{\partial u_i}{\partial x_j} + S_h \quad (16)$$

$$H = \sum m_i h_{i,T_{ref}} C_{p,i'} dT \quad (17)$$

The standard k-ε turbulence model (the most popular two equation model) was used to calculate turbulence kinetic energy and eddy viscosity in this analysis. Wall function predicts velocity and two turbulence parameters with the formation of the log layer solution by simply matching to the law of the wall using suitable value for the constant term. [4]

The VOF (Volume Of Fluid) model with a surface tracking technique applied to a fixed Eulerian mesh was considered to trace air bubbles from a pipe hole. It is designed for two or more immiscible fluids where the model, a single set of momentum equations is shared by the fluids, and the volume fraction of each of the fluid in each computational cell is tracked through domain. [5]

3.1 (b) Modelling of Heat Transfer for Phase Change

It is well known that phase changes need a certain amount of heat exchange called latent heat. This means that the heat transfer rate is much larger than that of sensible heat caused by temperature variation. For that reason, phase changes such as freezing, melting and vaporization are difficult to occur under given energy. [6] The sensible heat can be calculated the value of the specific heat (unit, J/kg-K), which is a property of particular fluid and the mass. The latent heat is constant value (unit, J/kg) for a fluid. In case of freezing of sea water, the freezing point can be defined for a given salinity. The freezing point of sea water is -1.94℃ in case of the salinity of normal sea water of 35%.

Theoretically, there is no temperature variation for a pure fluid on freezing, while there is small temperature variation for mixtures, particularly like sea water. The concept of variable specific heat does not exist for phase change but, imaginary specific heat could be simply used in order to compensate for heat transfer derived from freezing. Such a small temperature variation of the mixture can make it more reasonable to define the amount of energy to solve for the latent heat with offsetting of specific heat. Figure 6(a) illustrates mutual relations between entropy and temperature. Heat transfer rate is defined by specific heat, which is divided into three parts within the temperature range around freezing point. The specific heat is subject to liquid phase in sea water region and to solid phase in freezing region respectively. In mush zone, a large imaginary specific heat is used to offset the heat transfer derived from latent heat. Mush zone is the coexistence section with sea water and softly frozen ice. Variable specific heat values for each section are represented at the range of freezing point temperature in the graph (b) of Figure 6.

Figure 6: Numerical approach to describe latent heat for phase change

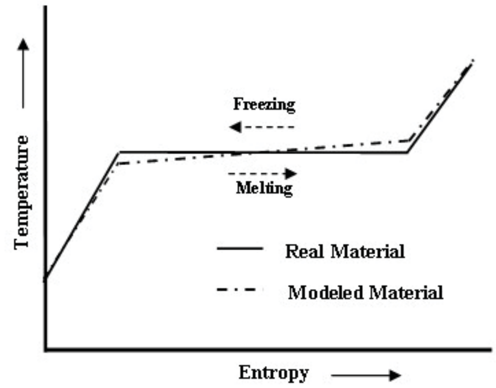


Figure 6 (a) Entropy – Temperature curve while freezing

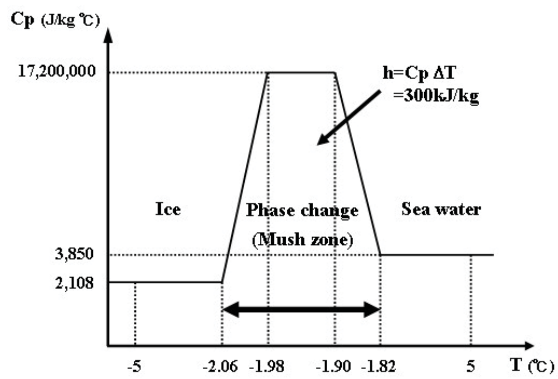


Figure 6 (b) Modelled latent heat defined by variable specific heat at the range of freezing point

3.2 MODELLING OF A BALLAST TANK

In very large containers and crude oil tankers, the ballast tanks are usually composed of many frames that are divided to increase hull strength. These frames are a critical factor that defines flow patterns and heat transfer in a tank. However, full modelling of a tank with many frames needs more cells and prevents precise description for hull structure. For that reason, a ballast tank with only three frames is modelled in this analysis. An air bubbling pipe is modelled to be placed in one frame only. The stiffeners installed over the water line are considered because they are expected to have some effect on flow patterns by rising air bubbles. The total number of cells is approximately 410,000 with shape of hexahedron.

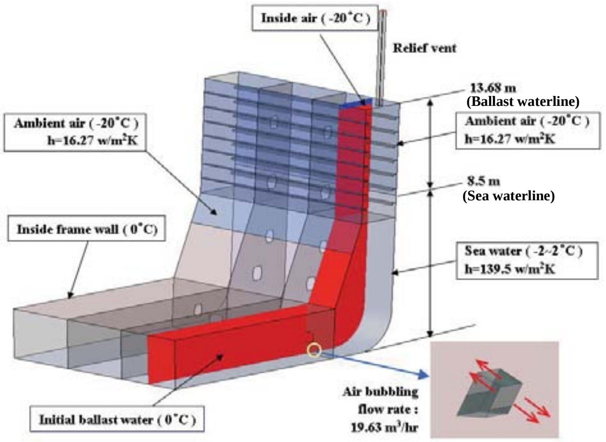


Figure 7: Schematics of a ballast tank including detailed boundary conditions

Boundary conditions are critical factors for analysis particularly, in case of the temperatures of outside air and sea water. The air temperature of -20℃ and the sea water temperature of 2℃ are referred to the environmental data at the North Sea. [7] Sea water temperature of -2℃ is used to describe frozen sea near the Arctic Sea. The convective heat transfer coefficients for outside hull exposed to ambient air and sea water are calculated under the designed velocity of air and current, respectively. The flow rate of air bubble is 19.63m$^3$/hr. Initially, a tank with three frames is filled with sea water under full ballast condition. There is only small gap filled with inside air between water line of ballast water and top of the tank. Discharged air bubbles off pipe could be exhausted to relief vent on top of the tank across ballast water in a tank. Used boundary condition and initial conditions are described in Figure 7.

## 3.3 ANALYSIS RESULTS

### 3.3 (a) Estimation of Effects of Modelled Phase Change Approach and Air Bubbling System

Simulations with latent heat and without latent heat are taken to compare and examine each other under the boundary conditions (sea water temperature of -2℃, ambient air temperature of -20℃). In case latent heat from ballast water to ice is not considered, any heat transfer is not needed for freezing. For that reason, the temperature of ballast water can be down easily. However, ballast water need much heat transfer rate in order to be frozen in real situation.

In Figure 8(b), the temperature distribution of using variable specific heat shows higher temperature values than that of using constant specific heat represented in Figure 8(a), outstandingly, at the upper area of the tank. Besides, temperature difference of ballast water is reduced as heat transfer derived from freezing is considered by using variable specific heat.

Figure 8: Comparison of temperature distribution of the tank as the condition of specific heats, represented by degree of Celsius

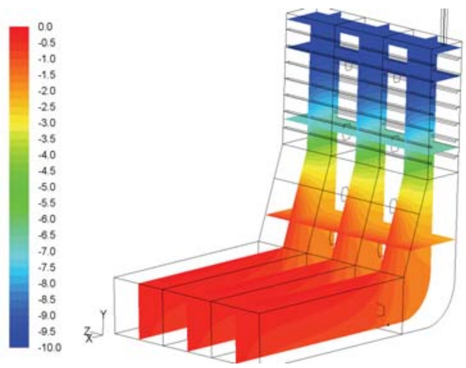


Figure 8 (a) Using constant specific heat for ballast water

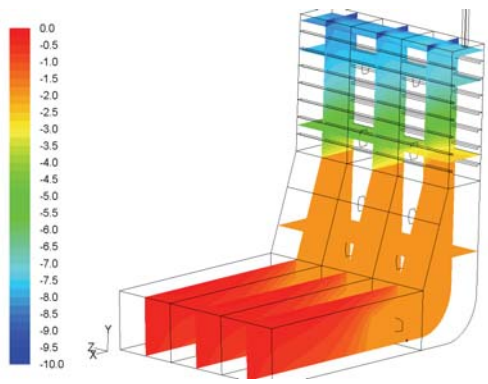


Figure 8 (b) Using variable specific heat to offset latent heat for ballast water

Figure 9 is the results of temperature distribution with and without air-bubbling system. Air bubbling pipe is only installed in front frame. The rest of frames would be flowed through the hole of structure from front frame. These simulations are for variable specific heat under same boundary conditions above. Temperature range of -7~-2℃ is marked at the upper area of tank for no air bubbling system because relatively low temperature of ambient air is contacted, while temperature distribution of with air bubbling system shows about -1.94℃ of tolerance 0.1℃, only small temperature difference can be checked in small figure with rearranged temperature section. Nevertheless, so similar temperature profiles can be investigated in the figures with different ranges of temperature of no air bubbling system. More uniform temperature distribution results from newly active internal flow induced by air bubbles off pipe. In this time, heat transfer subjected to conduction can start to be accompanied with convection, which makes temperature gap reduced.

Figure 9: Comparison of temperature distribution of the tank as per installation of air bubbling system, represented by degree of Celsius

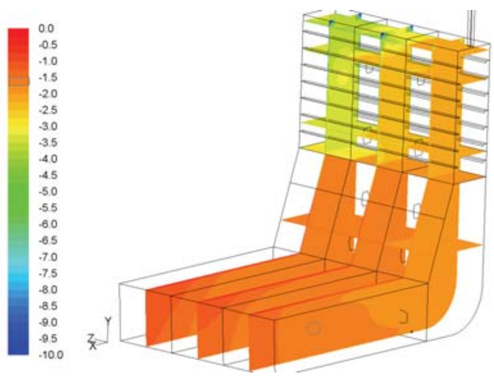


Figure 9 (a) Without air bubbling system

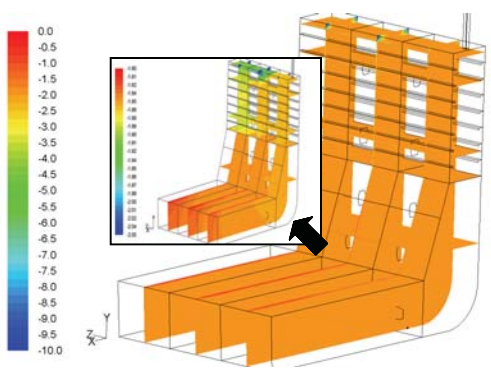


Figure 9 (b) With air bubbling system

Figure 10, 11 show measures of air bubbles trajectories and flow path line in ballast tank, respectively. Discordance for vertical positions between pipe hole of air bubbles and deck hole of horizontal plate brings on accumulation of emerging air bubbles under the horizontal plate and then the bubbles are released cross the deck hole, refer to B of Figure 10(a). This makes bigger volume and buoyancy force of bubbles on releasing from deck hole. In upper part, because the flow patterns are decided by rising air bubbles of deck hole, more complicated flow up and down with right and left is investigated. While, the main flow derives from released air bubbles at the corner side and consistent clockwise flow is formed in the lower part, going down at outer hull side and going up at inner hull side. Rising air bubbles hitting the water line induce fierce flow form side to side on water line. This expects to support delay of formation of freezing area, particularly, around the water line which is anticipated to form first freezing, under the law ambient temperature. Escaped air bubbles from ballast water to inner air layer of top of the tank are exhausted to the relief vent and, for that reason, the velocity magnitude near the relief vent marks high value.

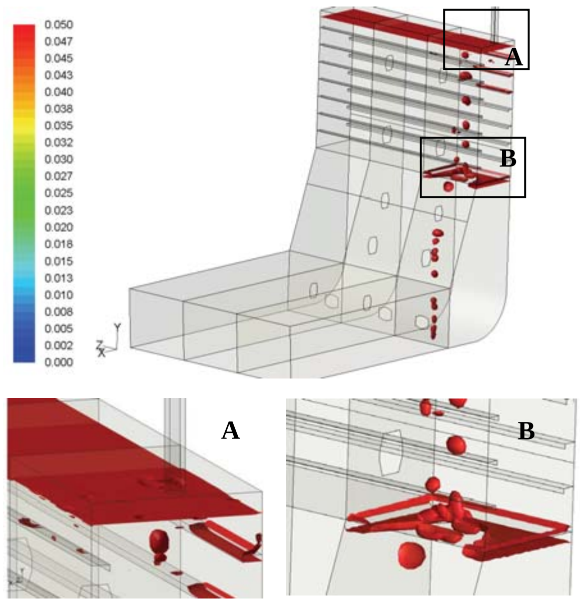


Figure 10: Air bubbles trajectories in ballast tank, represented by volume fraction for air

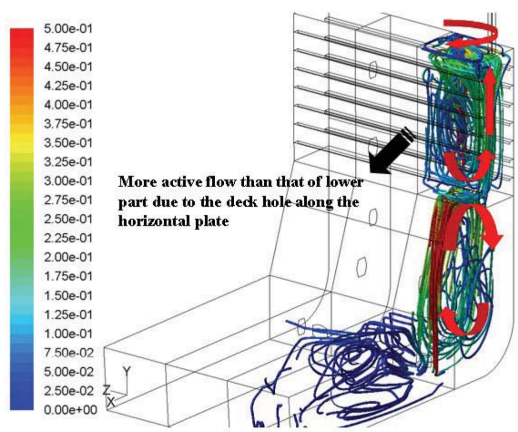


Figure 11: Path line of flow in ballast tank with colour legend of velocity magnitude with unit of m/s

Figure 12: Measures of thickness of ice sheet

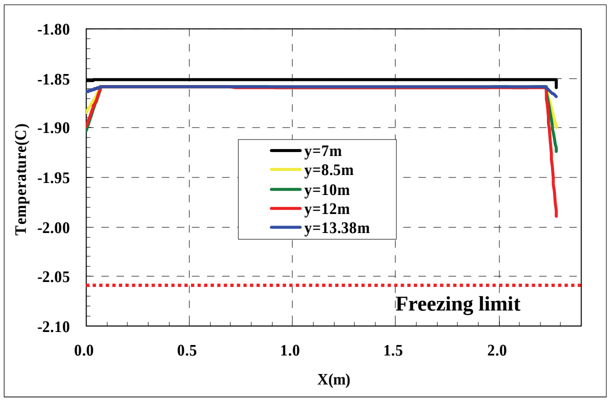


Figure 12 (a) Front frame

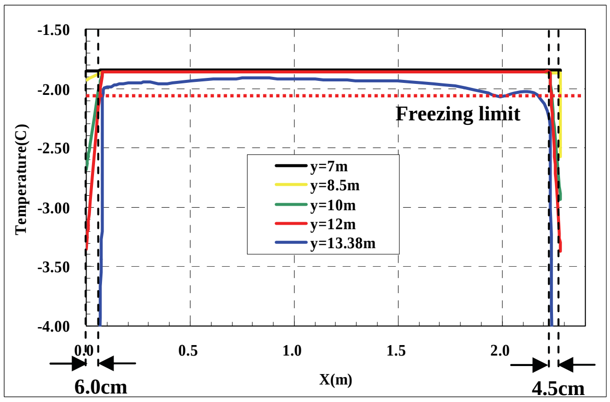


Figure 12 (b) Second frame

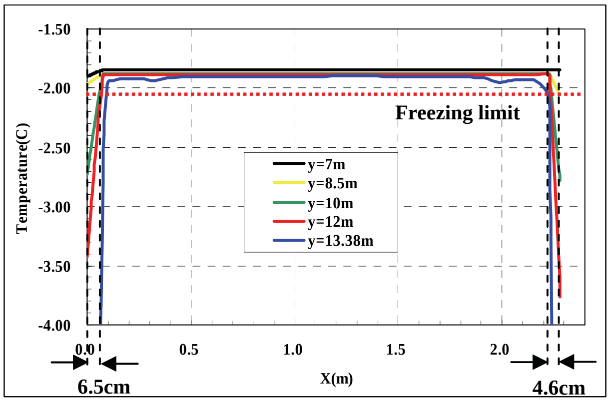


Figure 12 (c) Third frame

### 3.3 (b) Measures of Thickness of Ice Sheet for Each Frame

The thickness of ice sheet of each frame formed was measured for calculation when installing air bubbling system in front frame only. As aforementioned in Figure 6, the decision of forming icing is defined to temperature distribution of the tank. So, in case that the temperature of ballast water is less than -2.06, it can be regarded to appear icing.

Figure 12 shows the marked temperature values cross the vertical plane as per horizontal position for front, second and third frames which were represented according to priority of distance from installed air bubbling system. Remarkable temperature differences among the horizontal positions of the frames except for around water line (y=13.38m) and close regions to side hull surface can not be discovered. This is easily predicted to be due to activation of flow caused by air bubbles. The temperature of around water line is relatively low despite active flow. The reason of low temperature can be inferred from direct contact between water line and inner air of the ballast tank.

Front frame with air bubbling system does not have any icing area. The smallest temperature differences among horizontal positions are examined in the upper part of the tank. While, there are some icing areas in second and third frames. This means that active flow led by rising air bubbles in front frame does not affect significantly to other frames over the partitions with several small holes. The icing area can be only found on the side hull surface contacting with low temperature of outside air. Ice sheet starts to be formed over vertical height of 10m in both of second and third frames. The largest thickness of ice sheet on surface appears near water line for both of the frames. The thickness of ice sheet of second frame are 6.0cm at inside hull surface and 4.5cm at outside hull surface for vertical height of 12m, and those of third frame are 6.5cm at inside hull surface and 4.6cm at outside hull surface. The thickness of ice sheet of second frame is a little thinner than those of third frame due to relative smaller effect of circulation caused by proximity to front frame. Besides, the surface contacting to ambient air on inside hull is 56% larger than that on outside hull. So, the ice sheet on inside hull is shown to be formed 33% ~ 41% thicker in comparison with the ice sheet on outside hull.

## 4. CONCLUSIONS

Predefined overall heat transfer coefficients considering convection heat transfers related to flow patterns to inside and outside surfaces showed a more similar results with CFD simulation. It can be concluded that the estimated heating capacity can be considered as reasonable one within 10% of error.

For freezing analysis, the results show that when there is no external moving force action on the vessel, an icing area could be formed in all frames under an ambient air temperature -20℃ without any anti-freezing system. The air bubbling system has enough anti-freezing effect to the frames having air bubbling nozzle. However, frames without air-bubbling nozzle could bring on ice sheet along the side hull surface. Finally, it became clear that the air bubbling system installed in each frame of ballast tanks provides a reasonable countermeasure to prevent freezing along the side hull surface by promoting convection heat transfer between top and bottom section of a tank. Future work is expected to include experiments of freezing of ballast tanks. This can provide conclusive validation of the proposed model of internal freezing phenomenon.

Through a series of these researches from us, the developed design technologies have been guided us unique shipbuilder for arctic operating ships.

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# FERRITIC STEELS FOR LOW TEMPERATURE SERVICE

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**SUMMARY**

This document deals with the requirements of ferritic steels for low temperature service. Special attention is paid to the toughness properties and the measures for gaining these properties. With respect to the shipbuilding and operation of ships in low temperature environment the toughness properties of hull structural steels are discussed.

## 1. INTRODUCTION

Generally speaking, there are two main applications for low temperature steels in shipbuilding. One is the cargo tanks as well as the cargo and processing equipment on gas tankers. For this application, the use of specially designed low temperature steels is ruled in the Classification Society Rules. The other is the operation of ships in low temperature environments. In this case common normal and higher strength hull structural steels will be applied.

## 2. REQUIREMENTS CONCERNING THE PROPERTIES FOR USE

Low temperature steels are structural steels, which have good toughness properties at low temperatures. But beside these toughness properties, it is also important that low temperature steels have sufficient strength properties and good weldability. For the assessment of steels with respect of their suitability for low temperature service, the knowledge of how the material properties depend on the temperature is essential. This is precondition for selecting the appropriate material and to ensure the safety of the structure. The steel as well as the welded joint may show brittle behaviour at temperatures below minimum design temperature, only.

In case of the use of ferritic low temperature steels, it needs to be observed that it is typical for steels with body centred cubic lattice that the toughness properties decrease rapidly at a defined temperature range. This transition temperature depends not only on the material but also on the loading speed and on the degree of the multi-axial load condition.

## 3. CHARACTERISATION OF THE REQUIRED PROPERTIES

The toughness, which is the basis for the assessment of the suitability of the steel, will be preferably characterised by the notch bar impact test. However, due to the relatively low informative value of this test concerning practical load condition of steel structures, other test methods have been developed. For crack initiation, e.g. the notch tensile test and for crack propagation e.g. the drop-weight test can be applied [1].

The strength properties are based on the tensile test. Yield strength and tensile strength increase with decreasing temperature. As the calculation of structures is usually based on the values for room temperature, the strength properties at design temperatures will be not fully utilised.

The suitability for welding depends on the welding technology and will be checked indirectly by the procedure qualification tests performed by the welding shop or shipyard respectively.

## 4. MEASURES FOR GAINING THE REQUIRED PROPERTIES

### 4.1 CHEMICAL COMPOSITION

For receiving good toughness properties, a fine grain structure is important. In this respect, some alloying elements have significant impact.

Beside fine graining also other effects of the alloying elements need to be considered for toughness properties:

#### 4.1.1 Carbon

The carbon content should be limited to approximately 0.2% to reduce the pearlite content in order to reach sufficient toughness [2]. Furthermore, low carbon content improves the weldability properties and avoids hardening in the heat affected zone.

#### 4.1.2 Silica

The silica content shall be below 0.6% as higher contents have negative influence on the toughness properties.

#### 4.1.3 Manganese

Manganese contents up to 2% improve the toughness properties with respect to the transition temperature. The critical temperature for transition of shear fracture to brittle fracture will be shifted to lower values. [3]

However, the interaction with other elements, e.g. like Nickel, needs to be considered. Yield strength and tensile strength will be increased with higher manganese content.

For higher strength hull structural steels, the manganese content is restricted to a maximum of 1.60%.

#### 4.1.4 Nickel

Nickel has a special effect for decreasing the temperature of transition from shear fracture to brittle fracture. Low Nickel contents (< 2% Ni) lead to a reduced ferrite grain size. Higher Nickel contents result into the formation of bainite and martensite and in combination with appropriate heat treatment this leads to fine grain structure with high toughness. In case of Nickel contents higher than 5% (up to 9%) the toughness properties will be improved by the formation of homogeneous distributed secondary austenite.

Hull structural steels, except FH-grades, contain Carbon, Silica and Manganese but are not alloyed with Nickel. According to the specification, hull structural steel grade FH allows a maximum Ni-content up to 0.8%. In combination with defined manganese content and low carbon content, transition temperatures determined with ISO-V-notch bar impact tests down to -80°C are possible.

### 4.2 FINE GRAINING

Important for a fine grained structure is the alloying concept, the heat treatment and the rolling / forming process. Alloying elements for improvement of the grain size are elements, which form nitrides or carbo-nitrides such as Aluminium, Niobium, Vanadium or Titanium. Such precipitations hinder the growing of the austenite grains and therefore lead to a reduced ferrite grain size. [4-8] A steel grain size G (ASTM) of 6 to 10 is considered as being a fine grain steel.

### 4.3 DEGREE OF PURITY OF THE STEEL

Beside the alloying concept of the individual steel grade, the purity has significant influence on the toughness properties. The reduction of Sulphur and Phosphor leads to a decreased transition temperature [9-10]. The upper limit for Phosphor and Sulphur content of hull structural steel is 0,035% (0,025% for FH-grades). For structural steels or steels for pressure purposes for low temperature application the limits are often lower.

### 4.4 HEAT TREATMENT

The heat treatment is an additional measurement to adjust the microstructure and therefore the material properties. Normalizing of the steel results in a reduction of the grain size and in a more homogeneous microstructure. Also a quenching and tempering process or an accelerated cooling after rolling can improve the toughness properties.

### 4.5 ROLLING PROCESS

Due to the high impact of the grain size on the toughness of the steel, the rolling process is adjusted to have a grain fining effect and a resulting low transition temperature. The rolling process can be optimised for a following heat treatment or it can be optimised by gaining improved properties without further heat treatment. The latter can be achieved by a normalizing rolling or by thermo mechanical rolling. [11-12]

## 5. TESTING RESULTS OF HULL STRUCTURAL STEEL PLATES

Figure 1 shows impact test results of a 90mm thick grade E steel plate in a thermo mechanical rolled condition. The chemical composition is shown in Table 1 (Plate 1). The optimised chemical composition and high degree of cleanliness together with the rolling process leads to remarkable high impact values. The transition temperature for impact strength of 27J ($T_{Ü27}$) is ca. -50°C. The fibre fracture rate at -50°C is about 40% of the fracture area test specimen.

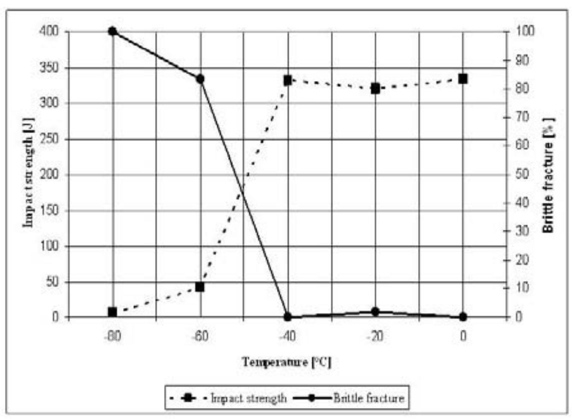


Figure 1: Impact test results – 90 mm plate

In Figure 2, the fracture areas of the notch bar impact test specimen are shown. However, different than the $T_{Ü27}$, the result of the drop weight test shows that the non-ductility temperature (NDTT) is at -40°C, already. Figure 3 and Figure 4 show the specimens tested at -35°C and -40°C.

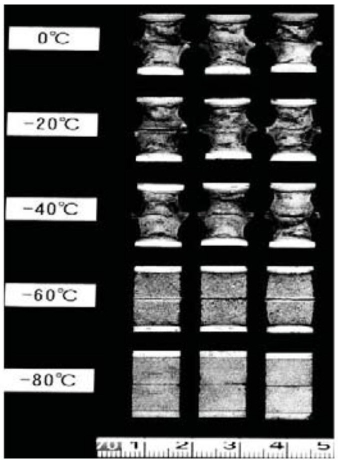


Figure 2: Fracture area of impact specimen – 90 mm plate

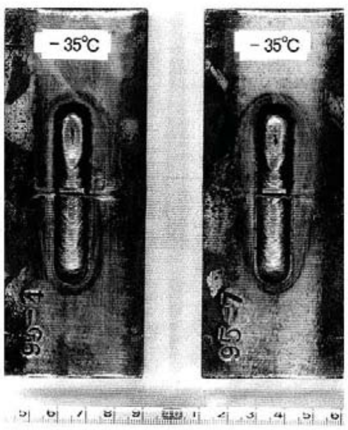


Figure 3: Drop weight test specimen – 90 mm plate (-35°C)

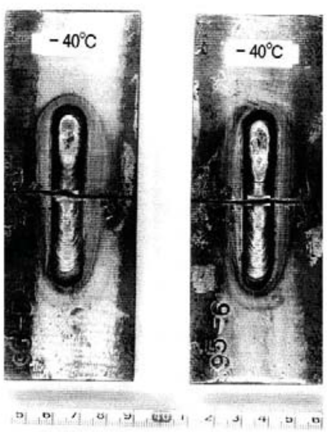


Figure 4: Drop weight test specimen – 90 mm plate (-40°C)

Figure 5 shows the results of impact tests of a 130mm thick grade E steel plate in the normalized and in the strain aged condition. The chemical composition is shown in Table 1 (Plate 2).

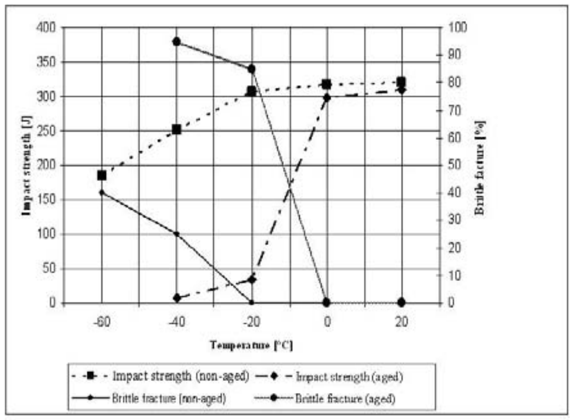


Figure 5: Impact test results – 130 mm plate

| Chemical composition (mass %) | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C | Si | Mn | P | S | Al | Cu | Ni | Cr | Mo | Nb | V | Ti | Ceq* |
| Plate 1 | 0.12 | 0.18 | 1.27 | 0.012 | 0.004 | 0.034 | 0.01 | 0.01 | 0.02 | <0.01 | <0.01 | <0.01 | 0.01 | 0.34 |
| Plate 2 | 0.10 | 0.25 | 1.29 | 0.009 | 0.004 | 0.037 | 0 | 0.02 | 0.03 | 0.01 | 0 | 0 | 0 | 0.33 |
| Plate 3 | 0.10 | 0.24 | 1.28 | 000.9 | 0.004 | 0.040 | 0.02 | 0.02 | 0.02 | 0.01 | 0 | 0 | 0 | 0.32 |
| Plate 4 | 0.15 | 0.19 | 0.49 | 0.011 | 0.019 | 0.040 | 0.04 | 0.023 | 0.03 | 0.00 | 0.001 | 0.003 | 0.005 | 0.26 |
| Plate 5 | 0.15 | 0.24 | 0.97 | 0.011 | 0.009 | 0.004 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.005 | 0.31 |

* : Ceq = C + Mn/6 + (Cu+Ni)/15+(Cr+Mo+V)/5

Table 1: Chemical composition

Also here, the impact test results are remarkable high down to -60°C. However in the strain aged condition, there is a significant shift of the transition temperature to higher temperatures. The area of brittle fracture corresponds to the impact test results.

The grain size of the material G (ASTM) is 6, which is the lower limit for fine grain steel. Again the results of the drop weight test differ from the result of the impact test. The specimen broke already at -40°C.

The third example is a 35mm thick plate of grade D in the as-rolled condition. The chemical composition of the steel plate is shown in Table 1 (Plate 3). The impact test results are shown in Figure 6. The steel shows remarkable good test results down to at least -40°C. Again the drop weight test seems to be stricter because it shows breaks of both specimens at -30C.

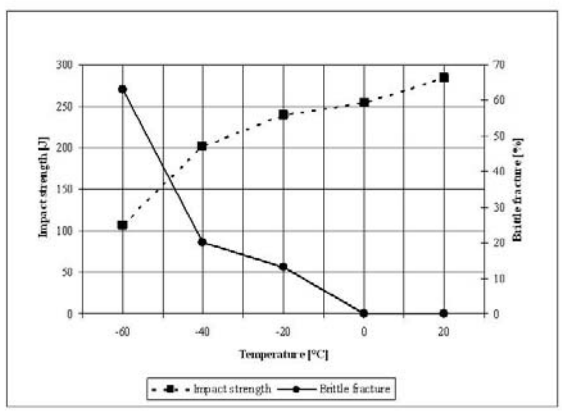


Figure 6: Impact test results – 35 mm plate

For grade GL-A, it is hard to find impact test results from the running production. Nevertheless, test results can be used from structural steel grade S235JR (+AR) according to EN10025 and comparable grades instead. Here the variation of the steel quality is very wide. Most often – depending on the manufacturer – impact values between 27J and 300J at room temperature can be found. As a result, the transition temperature is expected to be between +20°C and -20°C. Typical values for chemical composition are given in Table 1 for Plate 4 (18mm) and Plate 5 (20mm).

## 6. FORGINGS AND CASTINGS FOR SHIPBUILDING

Beside the rolled products discussed before, there are used a lot of forgings and castings in shipbuilding. The usual forging steels and cast steels used for shipbuilding e.g. for rudder parts or hatch covers are not specially tested concerning their low temperature properties. Many standards and rules do not require impact testing at all.

Impact testing results of forgings and castings at room temperature show, that their toughness properties at low temperatures are at least quite often questionable. In the replaced DIN17182 however, the transition temperatures, e.g. for GS-20Mn5 and GS-16Mn5 often used in ship building, are listed. Of course, there exist many standards and rules specially dealing with steel castings and forgings for low temperature service.

## 7. WELDING OF LOW TEMPERATURE STEELS

In general, ferritic low temperature steels have good welding properties. For the higher Ni-alloyed steel, special attention should be paid to the interpass temperature, which should not be higher than 80°C due to the susceptibility for hot cracks.

The welding consumables shall be adjusted to the base material and the hydrogen content shall be low. The use of ferritic welding consumables might be restricted by the design temperature and the load of the structure. Therefore dissimilar welding consumables might be necessary. Basic covered rod electrodes or relevant wire-flux-combinations should be preferably used. The thermal expansion and the magnetism (in case of higher Ni-alloyed steels) need to be observed.

## 8. CONCLUSIONS

Hull structural steels show a wide variation in their low temperature properties depending on the grade and on the plate thickness. Even for steels of the same grade and of similar thickness, the variation is quite high due to the wide range of chemical composition and the different production processes. Careful consideration is therefore needed for choosing the right grade and also for choosing the appropriate quality for the specific application.

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# BRITTLE FRACTURE IN SHIPS

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## SUMMARY

The majority of the world's civilian cargo vessels have side shells constructed largely with grade A steel which, by definition, is steel of unqualified toughness. In 2002 and 2003 respectively, samples of side shell steel from the *Lake Carling* and the *Ziemia Gornoslaska* (ex *Lake Charles*) were made available to the Transportation Safety Board of Canada (TSB). Metallurgical tests and analysis were conducted on this steel under the TSB's mandate to advance transportation safety. It was found that the steel from both vessels had very low Charpy Vee Notch (CVN) energies at temperatures near zero degrees Celsius. Because of the suddenness of brittle fracture and the very serious consequences for the vessel (water ingress and subsequent foundering), relevant information on such casualties has been unavailable or very limited. As such, the true causes of these vessel losses have perhaps been misclassified or misinterpreted in the past, thereby obscuring the true risk of brittle fracture to vessels. Given the uncertainties and variability of fracture arrest toughness for some grade A and B steels, it would appear that residual risks for unstable brittle fracture are still present in vessels with hulls constructed with these steels, especially when operating in colder climates. Since IACS Unified Requirements to use steel of qualified toughness in way of a vessel's side shell does not apply to the entire side shell, the risk of brittle fracture could be perpetuated in a significant proportion of new buildings. Given the gravity of consequences of brittle fracture of the side shell, the establishment of a toughness standard for this structural member is desirable. The standard should be rigorous enough to ensure adequate steel toughness to the new IACS qualitative benchmark of "North Atlantic 25 Years". As such, a reasonable damage tolerance could be predicted and relied upon. This paper brings forward the salient aspects of the TSB report on the brittle fracture of the *Lake Carling* and gives an overview of the TSB Safety Concern on brittle fracture in ships. Recent changes to the IACS Common Structural Rules for Bulk Carriers (April 2006) are also discussed as well as the residual risks seen to be remaining.

## 1. INTRODUCTION

On March 19, 2002 the loaded bulk carrier Lake Carling, while transiting the Gulf of St. Lawrence in benign wind and wave conditions, suffered a 6 metre brittle fracture of the port side shell in way of hold number 4. The vessel was loaded in holds 1, 3 and 5, and had been simultaneously de-ballasted during loading according to established procedures and within the parameters of the loading manual.

The vessel's loading history was examined, beginning with the most recent load of iron ore at Sept Iles the previous day, and extending back approximately one year before the accident. The only anomaly found was a load of potash taken at Thunder Bay some 15 weeks prior. At that time holds 1, 2, 3 and 5 were loaded such that at frame 86 in hold number 4 (empty), still water bending moment (SWBM) was about 103% of the maximum allowable. Temperatures during this time were cold – between 0 and 5 degrees C. In its report, the Transportation Safety Board of Canada [1] determined that at that time conditions were created for small initial cracks to form at the lower ends of some side frames between frames 85 and 96 in hold number 4 due to:

- Service loads greater than those approved for the vessel
- Probable presence of residual stresses
- Stress concentration factors due to discontinuity caused by scallop (cut-out) in the side frame
- The proximity of the frame end to the shell plate seam weld
- The change in plate thickness at the shell plate seam.

During the investigation pre-existing cracks were found in hold number 4; on the port side, at frames 89, 91 and 93, and on the starboard side, at frames 85, 91 and 96. All pre-existing cracks were located in H strake and appeared to originate near the base of the frame at the toe of the weld. Each location gave rise to two cracks, one forward and one aft of the frame, each some 75mm in length and generally in a characteristic "V" formation. All of these cracks were rusted and appeared to have been present for some time. The principal fracture was the forward run of one of these pre-existing cracks, that of frame 91 port.

Figure 1: Frame 91 Port (aft)

## 2. LAKE CARLING AND SISTERS

The Lake Carling was built in Turkey in 1992 to DNV 1A1 and Polish Registry specifications. The vessel was strengthened for carriage of heavy bulk cargoes and was DNV ice class 1C. Vessel specifications indicate that holds Nos. 2 and 4 may be empty (alternate loading). Strakes H, J, and K are all grade A steel, 19mm thick, with the rolling direction along the length of the ship. G strake, just below H, is similar in quality to the above-mentioned strakes but is 15 mm thick. In shipbuilding, grade A steel is often used in the majority of a hull structure, and this was the case for the Lake Carling. The shear strake (L strake) and strength deck were grade E steel 30 mm thick.

Two other vessels were constructed to the same plans and specifications as the Lake Carling (now Ziemia Cieszynska), and at the same shipyard. Hull number 14 was constructed in 1990 and later became the Lake Charles (now Ziemia Gornoslaska). Hull number 15 was constructed in 1992 and later became the Lake Champlain (now Ziemia Lodzka). The Lake Carling was hull number 16.

In December 2003, the Ziemia Gornoslaska was in Montreal for repairs to cracks found in the side shell in way of hold number 2.

Because samples of side shell steel from both the Ziemia Cieszynska and the Ziemia Gornoslaska were made available to the Transportation Safety Board of Canada, metallurgical tests and analysis were conducted. It was found that the steel from both vessels had very poor fracture toughness at temperatures near zero degrees Celsius. In the case of the Ziemia Cieszynska, the critical crack length was found to be about 100 mm – that is the length which, if reached, will result in unstable rapid propagation of the crack under normal service loads.

Figure 2 illustrates the Charpy Vee Notch (CVN) impact energies (in Joules) as recorded during testing at the TSB Engineering Branch facility. [2]

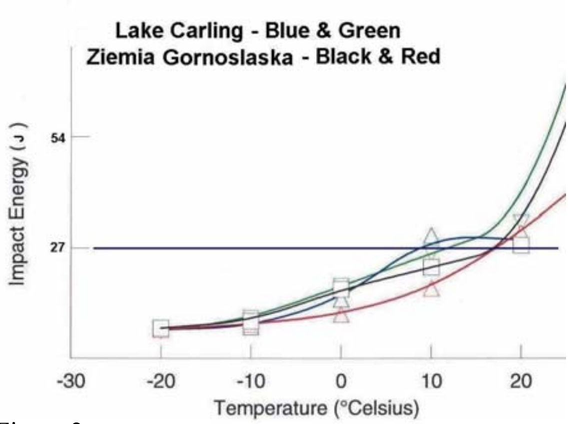


Figure 2:

## 3. TOUGHNESS, CVN AND FATT

Although the relationship between CVN energy and fracture toughness is not necessarily straightforward, this test has been used with relative success by all of the major classification societies for many years by providing a qualitative estimate of material toughness.

In a review of the fracture properties of Lloyd's Register (LR) grade A ship steel [3], LR found that from a total of 39 samples coming from a variety of steelmakers word-wide, the lowest average CVN recorded was 49 J at 0°C (from one sample), while the average value at this temperature amongst all 39 samples was much higher, at 134 J. Five samples (12.8%), however, had fracture appearance transition temperatures ($FATT^{50\%}$) above 0°C. (Where $FATT^{50\%}$ is the temperature at which the fracture surface shows 50 percent fibrous appearance and 50 percent crystalline appearance.) A reasonable assessment of these results should raise red flags insofar as toughness is concerned. The fact that grade A steel is, by definition, a steel without a toughness standard should raise concerns given that vessels can be expected to trade in areas where water temperature is at or near 0°C.

For the Ziemia Cieszynska, the FATT was determined to be 32°C. [4] In other industries, such as electric power generation, risks due to brittle fracture are reduced by ensuring that operating pressures are only permitted at component temperatures approaching or exceeding the component's FATT.

Nonetheless, tests such as those undertaken by LR have shown that the average CVN of grade A steel available worldwide is often quite high and grain size relatively small. This, in effect, sets a defacto standard - ship owners, ship constructors, and classification societies all expect and depend upon grade A steel having a fracture toughness that is sufficient for all operational conditions. However, without actual standards, expectations are not always enough to ensure adequate fracture toughness and damage tolerance.

## 4. WATER TEMPERATURE

Although toughness standards are imposed for vessels constructed to operate in areas with low air temperatures [5] there are no requirements to use steel of a known or verified toughness (or transition temperature) in way of the majority of a ship's side shell for vessels operating world-wide – hence Grade A is used almost universally for this application.

IACS's own new qualitative standard is stated as "25 Years Operation Life North Atlantic". [6] This standard therefore considers criteria such as the respective wave conditions, i.e. the statistical wave scatter takes into account the basic principle for structural strength layout. However, what is unmentioned in the standard but should be an implicit consideration is the water temperature. Figure 3 shows the average winter North Atlantic water temperature distributions. [7] Clearly, ships transiting this area will often encounter water temperatures between 5°C and 10°C, and in some cases, such as in the Gulf of St. Lawrence, close to 0°C.

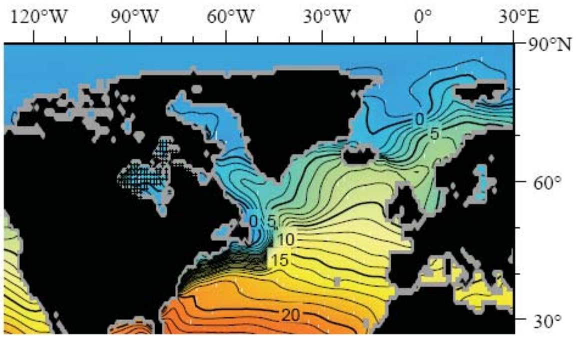


Figure 3:

## 5. HISTORICAL PERSPECTIVE

Historical data have revealed that nearly three quarters of all vessel loss-related fatalities on bulk carriers are attributable to vessel structural failure. Other data culled from Lloyd's casualty database indicates 23 bulk carriers foundered in cold water in a twenty year period, yet the cause of the losses are undetermined because the wrecks were never thoroughly investigated. Even the Derbyshire and the Prestige, both extensively investigated, did not produce CVN testing of the side shell steel. As such, the toughness of this steel cannot be appreciated and hence remains an unknown factor insofar as possibly contributing - or not - to the vessel's loss.

Of the four major accident reports on the loss of the Prestige, not one was able to identify the cause of the initial ingress of water on the starboard side – the significant event that began the vessel's slow fateful demise. Figure 4 below was taken on 17 November when further damage had occurred subsequent to the initial ingress of water but before the towing phase had commenced. Straight clean fractures, often associated with that of brittle fracture, are clearly visible. Yet the toughness of the Prestige steel in way of the side shell remains unknown. What is even more surprising, however, is that it remains unquestioned.


Figure 4 [8]

## 6. RISK REDUCTION MEASURES

Although the Enhanced Survey Program (ESP) and other initiatives more recently introduced to reduce risk for bulk carriers are continuing to increase safety, the Ziemia Cieszynska and her sister ship can be viewed as examples of residual risk for side shell integrity related to brittle fracture that remains in spite of these initiatives. Side shell integrity studies have identified several promising risk control options (RCO) for bulk carriers but increasing toughness requirements for side shell steel has not been one of them.

The appropriateness of using steel of unknown toughness in vessel construction has been raised in various reports and proceedings, including those concerning the loss of the Derbyshire, the brittle fractures of the Tyne Bridge and the breaking in two of the Kurdistan. In a major review of a vast amount of available literature concerning the fracture properties of grade A ship plate, it was concluded that "...the crack arrest ability of grade A plate is poor and probably inadequate for most ship applications". [9] Notwithstanding the average high toughness and quality found in some steels, a certain proportion of grade A and B steels are not suitable for all conditions. Yet these steels are still being produced and used in ship's hulls.

## 7. RISIDUAL RISKS

Cracks in ships, be they from greater than approved service loads, fatigue, loading/unloading equipment, or other sources are a fact of life in the marine world. All ships operating in cold waters and having their side shell of metal with characteristics similar to those of the Ziemia Cieszynska are therefore at risk. The damage tolerance could be less than adequate and cracks could remain unnoticed or discounted as insignificant, yet they would still pose a significant risk when exposed to low temperatures. Although the new IACS Common Structural Rules for bulk carriers (2006) have a more stringent standard than previously for single skin BC-A and BC-B ships, this grade D/DH requirement extends only to strakes in the proximity of the intersection of the side shell and bilge hopper sloping plate. Given the uncertainties and variability of fracture toughness for some grade A and B steels, it would appear that residual risks for unstable brittle fracture remain in bulk carriers and other vessels with hulls constructed with these steels, especially when operating in colder climates. Additionally, double side skin construction is not necessarily a panacea if the steel is of inadequate toughness for all conditions.

## 8. A STUBBORN DEFICIENCY

Even in an industry such as railway transportation, where every failure can potentially be studied in detail, the stubborn deficiency of steel with less than adequate toughness for the intended use has only recently been revived. [10] After the derailment and subsequent release of anhydrous ammonia near Minot, North Dakota (U.S.A.) in 2002, the National Transportation Safety Board recommended design-specific fracture toughness standards for steel pressure tank cars used to transport certain hazardous materials. [11]

Figure 5: NTSB/RAR-04/01

## 9. CONCLUSIONS

The use of steel of unknown and unqualified toughness and/or FATT in way of ships' side shells has allowed some vessels to be constructed of steel that is less than adequate for the ambient conditions they will be used in. Because a vessel's side shell, particularly bulk carriers, is prone to flexing, the side shell is more at risk to crack damage than any other area of the vessel. [12] Crack initiation is the first step towards a major fracture. Once a crack has initiated, only the material's critical crack length stands between a nuisance defect and disaster. The material's critical crack length is intimately related to its' inherent crack arrest toughness.

Ship construction standards continue to progress, as witnessed with the introduction of the new IACS Common Structural Rules for Bulk Carriers in 2006. However, in order for vessels to truly achieve the "25 Year North Atlantic" performance standard that was introduced in these rules, more can be done. If brittle fracture is to be definitively taken out of the 'lost vessels equation' it can be reasonably argued that the transition temperature of the side shell steel should be adequate to meet IACS's own qualitative standard of North Atlantic water temperatures. For this to be so, the transition temperature of the steel used for a vessel's side shell must be known.

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## 11. AUTHORS' BIOGRAPHIES

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# LIFEBOAT OPERATIONAL PERFORMANCE IN COLD ENVIRONMENTS

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## ABSTRACT

Shipping and offshore petroleum industry operations in Arctic and sub-Arctic regions have to account for an environment characterized by cold temperatures, remote locations, and a wide range of sea ice cover. To do so successfully, environmental factors must be addressed at the concept design stage. The environment affects operations on multiple levels: special structural design and steel grades to withstand ice loads under cold temperatures; robust propulsion systems to ensure reliability under propeller-ice interaction; winterization measures such as heating, insulation of fire mains and cooling water pipes, arrangement of access ways, icing, and extended low light conditions; and the human factors of working in a cold, remote, dark environment for extended periods. Design and operation in such environments requires special knowledge, skill and technology. This applies as well to the design and operation of the vessels' safety systems, including evacuation craft. An evacuation scenario must be executed in the ice conditions that prevail at the time of the emergency. In order to design an appropriately robust emergency response capability, it is essential to know what to expect of evacuation systems in terms of their utility in the presence of ice. This paper presents the results of an experimental campaign that investigated the performance capabilities of several lifeboats in ice. A series of model scale experiments was done in an ice tank to examine the effects of ice concentration, floe size and thickness on the lifeboats' abilities to launch and make way through the ice. Three different hull forms were tested to see how changes in shape might change performance. Likewise, changes in the delivered power were investigated in terms of simple performance benchmarks. Conclusions drawn from the model tests are presented and discussed.

## 1. INTRODUCTION

Most emergency response procedures and equipment have been designed for temperate regions and open water conditions. Indeed, evacuation craft on ships and offshore petroleum installations are overwhelmingly of the conventional lifeboat sort, whether configured as free-fall or davit launched. Such craft are clearly not suitable for operation in heavy ice cover, if for no other reason than they cannot be launched into the water. Still, even if used in cold regions, they are capable in open water conditions and have marginal utility when the ice conditions are not onerous, including during the freeze-up and break-up seasons that typically bracket the most severe winter conditions. The main aim of the work presented here is to provide some quantitative guidance on the utility of lifeboats in marginal ice conditions, including the expected limits of their abilities. The performance limits of conventional lifeboats define the minimum performance requirements of complementary innovative evacuation systems that can extend existing emergency response capabilities.

Hazards that may give rise to the need to evacuate, and the impact of such hazards on the means of evacuation, are not dealt with here, although they are recognized as being important, as is the integration of means of evacuation and rescue in the broader context of emergency response.

## 2. EXPERIMENTS

### 2.1 SCOPE

The main objective of the experiments reported here was to define the operating limits imposed by ice conditions on a conventional lifeboat in terms of ice concentration, ice floe size, and ice thickness. In addition, the effects of additional lifeboat power and different hull forms were investigated. The limits constitute a boundary beyond which there is a gap in evacuation capabilities that presents an opportunity for innovation.

The main set of tests involved a pair of 1:13 scale models of a generic lifeboat, fitted with propulsion systems that could be operated at 4 distinct power settings. Tests with these models were done in a range of ice concentrations from approximately 4/10$^{ths}$ up to 9/10$^{ths}$, and in ice of two thicknesses and two floe sizes. A second set of tests was done with three models of three different lifeboat types. These three models, which were built at a larger scale (1:7), were also tested in a range of ice conditions and at two separate power settings, although the range of test conditions was somewhat smaller than in the first set of tests. The main goal of the second set of tests was to evaluate the effect on performance of changes in the lifeboats' hull form.

### 2.2 SETUP: MODELS

Sketches of the three larger models are presented in Figures 1, 2 and 3, showing a conventional TEMPSC style displacement craft, a free-fall type lifeboat, and a hard chine TEMPSC displacement craft, respectively. All three vessels were of similar size, as indicated by the 1:7 scale model particulars in Table 1. Tests done with the smaller scale models (1:13) used the same geometry as the conventional lifeboat shown in Figure 1.

All the models were built in two sections (hull and canopy) using molded glass reinforced plastic. Each of the 1:7 scale models was fitted with an electric motor run on batteries. Two power settings were available. The main power setting corresponded to the power required to meet the regulatory requirement that the vessel make 6 knots in open water (IMO 1997), which was slightly different for each hull form. The second power level corresponded to the maximum available and was similar for each vessel. Using bollard pull as a benchmark, the second power setting provided an increase of about 10% to 25% over the main power setting. Each model was driven by a single screw. A small video camera was fitted in the coxswain's position and transmitted to a tank-side monitor that was used during the tests by a technician who operated the vessel by remote control. Each model was also fitted with motion sensing instruments, markers for optical tracking, remote control hardware, a radio transmitter, and a PIC acquisition system. More details about these models can be found in Mak et al. (2005) and Simões Ré et al. (2006).

The two smaller models were geometrically similar to each other (and to the model in Figure 1), but were configured differently for the two test types: one for launch tests and the other for powering tests. Instrumentation and outfit included a motor and battery pack, a propeller and rudder, and a wireless transmitter and camera. The maximum shaft speed used in the tests corresponded to a full scale forward speed in calm water of 6 knots. The model launch system was fitted to the carriage of the Ice Tank and consisted of a conventional twin falls davit arrangement with dual motors and winches. This was used to lower the model into the water from its stowed position, where the falls were released and the lifeboat was driven away at full power, with control being exercised by the remote control coxswain. Powering tests could not be done with the same model as there was insufficient room for the larger motor and battery pack. Likewise, this meant that the higher powered model was unable to be launched remotely and so started in the water. A fuller description of the models is given elsewhere (Simões Ré & Veitch 2003 and Simões Ré et al. 2003).

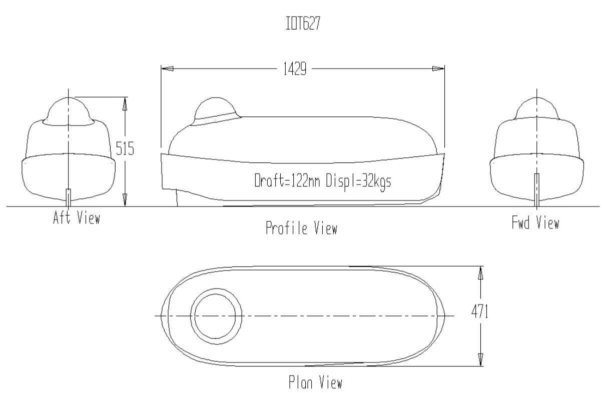


Figure 1. Conventional TEMPSC lifeboat model (627).

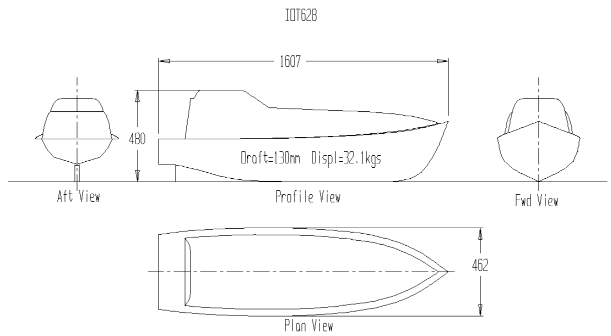


Figure 2. Free-fall lifeboat model (628).

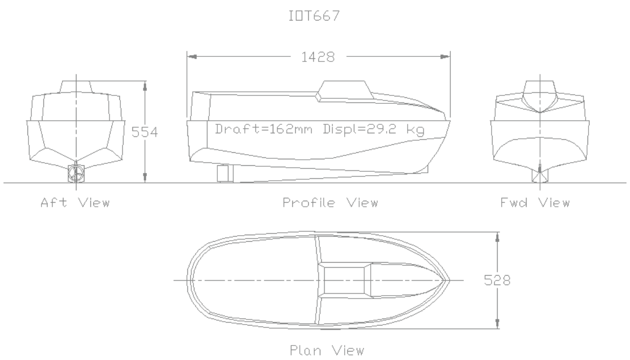


Figure 3. Hard chine TEMPSC lifeboat model (681).

Table 1. Model particulars, 1:7 scale.

| Condition | IOT627 | IOT628 | IOT681 |
|---|---|---|---|
| Length overall (m) | 1.429 | 1.607 | 1.429 |
| Length on water line (m) | 1.381 | 1.521 | 1.353 |
| Breadth overall (m) | 0.456 | 0.413 | 0.507 |
| Mass (kg) | 32.85 | 32.92 | 29.15 |
| Longitudinal centre of mass (m) | 0.720 | 0.709 | 0.740 |
| Vertical centre of mass (m) | 0.186 | 0.214 | 0.221 |

## 2.3 SETUP: ICE

All the experiments were done in the Ice Tank at the Institute for Ocean Technology. Separate ice sheets were grown for the thinner and thicker ice conditions. For the initial set of tests with the small model of a conventional lifeboat, the ice sheets were approximately 25mm and 50mm thick, corresponding to full scale conditions of 325mm and 650mm. For a given ice sheet, separate pools were cut for the smaller and larger floes. For each pool, the ice cover was cut into appropriate floe shapes and then a strip of ice was removed to reduce the concentration to the initial test conditions (typically $9/10^{ths}$). The remaining floes were distributed over the pool's surface area. Tests were then done in the pool for the given conditions, after which the concentration was adjusted and testing continued.

A similar process was followed for the ice sheet preparations in the tests with the larger lifeboat models. In those tests, the ice sheets were approximately 46mm thick, corresponding to a full scale ice thickness of approximately 325mm.

### 2.4 TEST PLAN

The small model used in the initial tests was tested first in thinner ice. 20 launches were made into a pool with smaller floes, followed by 12 launches into another pool with larger floes. Ice concentration in the smaller floe pool started at $9/10^{ths}$ and was reduced during the test series stepwise to $4/10^{ths}$. The corresponding range for the larger floe pool was $7/10^{ths}$ to $4/10^{ths}$. Following the series of tests in thinner ice, a second series of tests was done using the same model, but in thicker ice. 10 launches each were made into separate pools of smaller and larger floes over a range of concentrations.

After the launch and sail-away tests described above, the second 1:13 scale model of a conventional lifeboat was used for powering tests. The model was calibrated to provide approximately two, three, or four times the bollard pull compared to the power available from the basic configuration (that is, the power required to make 6 knots in calm water). The powering configurations are denoted, from basic to highest, as $T_1$, $T_2$, $T_3$, and $T_4$, respectively. 35 powering tests were done, consisting of 17 in the thinner ice sheet and 18 in the thicker ice sheet. Two pools were used, one for smaller floes and the other for larger floes. The range of ice concentration conditions was narrower in these tests as the focus was on the conditions that presented difficulty for the model with the basic installed power.

The three larger scale models were also tested in similar ice conditions consisting of combinations of floe size and ice concentrations. In addition, for several ice conditions, the model was tested at two different power settings. A total of 76 tests were done, 45 in smaller floes and 31 in larger floes.

## 3. RESULTS

### 3.1 ICE CONCENTRATION, THICKNESS, AND FLOE SIZE

The main measure of performance used throughout the tests was simple: was the lifeboat able to launch and sail away through the pack ice? Each test was given a pass or fail grade based on whether the boat made it through a distance of 75m (full scale).

Results are presented in Figures 4 to 7 in terms of the plotted paths taken by the lifeboat during each test. For example, Figure 4 shows the path taken by the lifeboat in each of the tests done in the thinner ice and smaller floes. A separate plot is given for each of the six concentrations, starting at the top with $9/10^{ths}$. In that top plot, two very short paths are presented, corresponding to the unsuccessful transits associated with the two test launches done in that condition. Similar results are shown in the second plot for the three unsuccessful transits in $8/10^{ths}$ concentration. The third plot shows the paths taken by the lifeboat in each of four successful transits in $7/10^{ths}$ conditions. Similarly successful transits are shown in the remaining plots of Figure 4 for the tests in lower concentrations. With reference to the figure, as the concentration decreases, the paths taken by the boat become increasingly more direct.

Figure 5 shows the corresponding results for the tests done in thinner ice and larger floes. Figures 6 and 7 show results of tests done in thicker ice and smaller and larger floes, respectively. Table 2 summarizes the results in terms of the simple pass (P) or fail (F) grades. The numbers in parentheses in the table refer to the number of tests done in those conditions. The results of repeated tests for each condition were consistent: all passed or all failed. Conditions became impassable at concentrations of between $6/10^{ths}$ to $8/10^{ths}$, depending on the thickness and ice floe size, with thicker ice and larger floes being more difficult to transit than thinner ice and smaller floes.

### 3.2 POWERING

The second model in the main test series had an adjustable setting to provide different thrust values, corresponding approximately to a doubling, tripling and quadrupling of the basic power configuration. Results of these tests are summarized in Table 3, again using pass and fail grades as the basic performance measure. For each combination of concentration, floe size and ice thickness, tests were done at one or more power settings. For each condition, results using the basic power setting are shown in the bottom right, results from the highest power ($T_4$) are in the upper left, and the intermediate powers, $T_2$ and $T_3$, are shown in the lower left and upper right corners, respectively. The results show that very significant increases in power yielded only marginal improvements in terms of extending the operational limits, for example from $6/10^{ths}$ to $7/10^{ths}$.

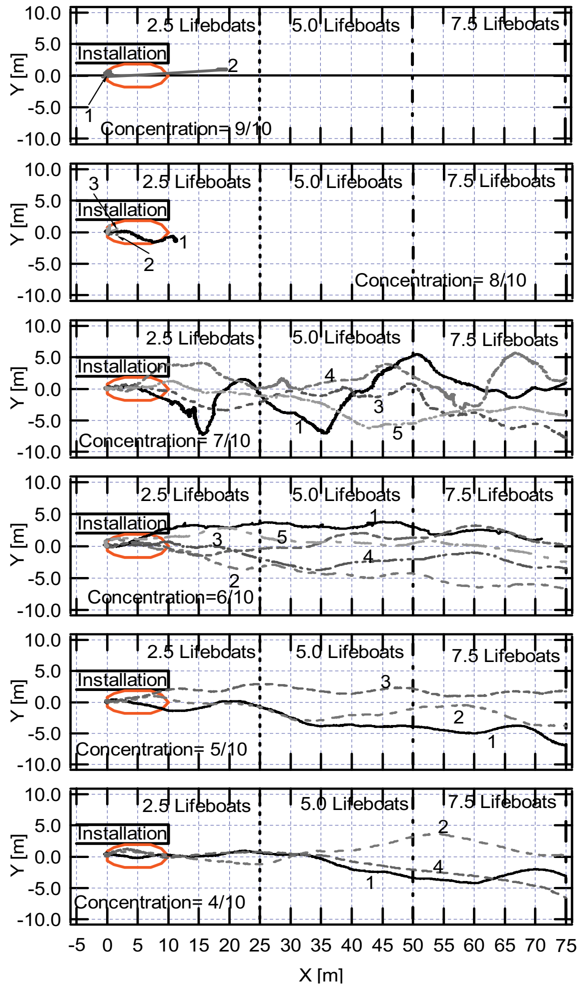


Figure 4. Paths: thin ice, small floes.

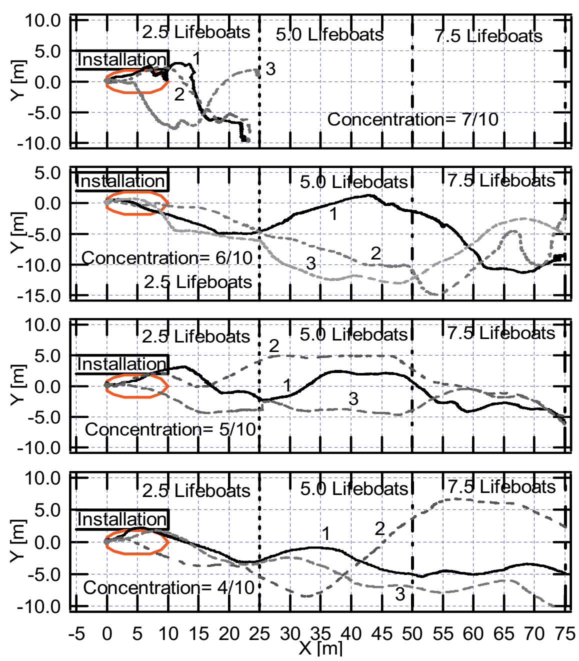


Figure 5. Paths: thin ice, large floes.

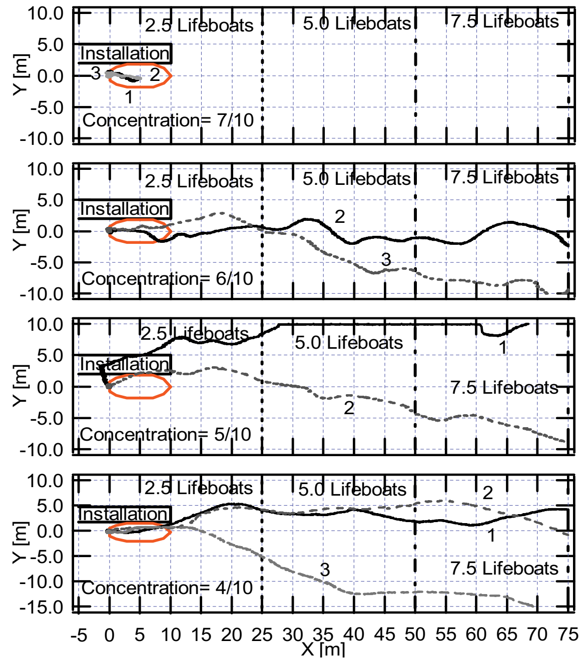


Figure 6. Paths: thick ice, small floes.

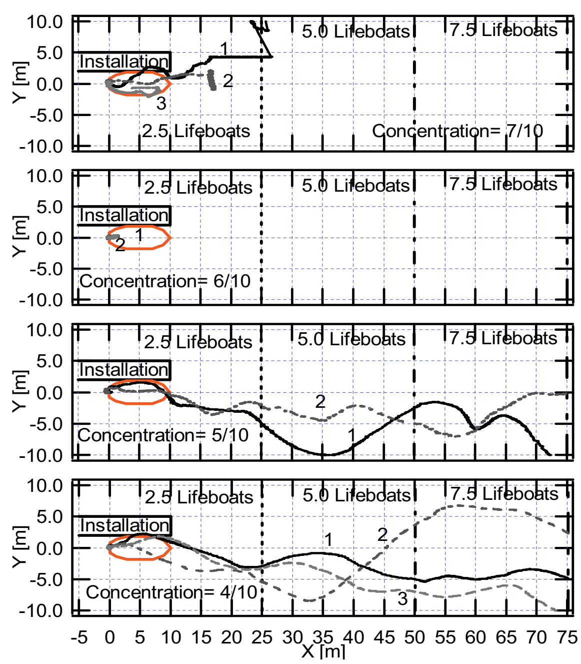


Figure 7. Paths: thick ice, large floes.

Table 2. Pass and fail results for the launch tests.

| Nominal Ice concentration [10ths] | | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|
| nominal thickness | nominal floe size | Pass or Fail | | | | | |
| 325mm | smaller | P(3) | P(3) | P(5) | P(4) | F(3) | F(2) |
| 325mm | larger | P(3) | P(3) | P(3) | F(3) | | |
| 650mm | smaller | P(3) | P(2) | P(2) | F(3) | | |
| 650mm | larger | P(3) | P(2) | F(2) | F(3) | | |

Table 3. Powering effects: conventional lifeboat.

| thick [mm] | floe size [-] | Ice concentration [10ths] | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 5 | | 6 | | 7 | | 8 | |
| 325 | smaller | | | | | | | 2F | |
| | | | 3P | | 5P | 2P | 3P4P | | 3F |
| 325 | larger | | | | | 2P | 3P | | |
| | | | 3P | | 3P | 2P | 3P3F | | |
| 650 | smaller | | | | | 2P | 2P1F | | |
| | | | 2P | | 2P | 1P1F | 3F | Power legend | |
| 650 | larger | | | 2P | 1P1F | 3F | 3F | $T_4$ | $T_3$ |
| | | | 2P | 1P | 2P2F | | 3F | $T_2$ | $T_1$ |

3.3 HULL FORM

Results of tests with the three larger models are shown in Table 4, where the model numbers corresponding to the boats are 627 for the conventional TEMPSC, 628 for the free-fall, and 681 for the hard chine boat. These tests were done in similar conditions as those done with the smaller models, that is, in pack ice comprised of smaller and larger floes with concentrations from $5/10^{ths}$ to $8/10^{ths}$. For each combination of ice thickness, floe size, and ice concentration, the table has two spaces, one on the right for the lower power (or thrust) $T_1$, and the other at left for the higher power, $T_+$.

The limiting ice concentration for all of the lifeboats was usually $7/10^{ths}$. Repeated tests done with the hard chine boat in $6/10^{ths}$ concentration and small floes in the basic power configuration had some mixed results – that is, 5 successful transits and 1 unsuccessful test. Similarly mixed results were observed for the conventional TEMPSC in larger floes at both $6/10^{ths}$ and $7/10^{ths}$. Overall, the lifeboats exhibited roughly similar behaviour; there was no clear evidence that one hull form performed better or worse than the others.

Table 4. Hull form effects (325mm full scale).

| Floe size | model | Ice concentration [10ths] | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 5 | | 6 | | 7 | | 8 | |
| small | 627 | | | 6P | 6P | 1F | | | |
| | 628 | | | 7P | 7P | 1F | | 1F | |
| | 681 | | | 6P | 5P1F | 2F | | 2F | |
| large | 627 | | 2P | | 2P1F | 5P2F | | | |
| | 628 | 3P | 4P | 3P | | 2F | | Power Legend | |
| | 681 | | | 3P | 4P | | | $T_+$ | $T_1$ |

The conventional lifeboat is 627, the free-fall lifeboat is 628, and the hard chine TEMPSC is 681.

Maneuvring characteristics were also examined in both open water and in ice using turning circle diameter as a performance benchmark. All three lifeboats had larger turning circles in open water than in the ice conditions in which tests were done, and although the diameters were different in open water for the three hull forms, they were almost the same in ice.

Results of additional tests done in combinations of pack ice and waves for all three models were reported by Sudom et al. (2006). The results indicated that the presence of waves in combination with pack ice helped the vessel make way through the ice, even in relatively high ice concentrations that would otherwise have prevented progress in calm conditions. The models' success in transiting pack ice in waves was found to depend mainly on the wave period and ice concentration, rather than on hull form.

## 4. CONCLUSIONS AND DISCUSSION

An experimental campaign using model scale lifeboats was carried out to investigate the performance of evacuation craft in a range of ice conditions, including combinations of pack ice concentration, thickness and floe size. These tests were used to estimate the limiting ice conditions for these evacuation systems. In addition, the effects of additional power and different hull form were investigated to determine if these might have mitigating effects on the performance limits.

In terms of ice conditions, concentrations of $6/10^{ths}$ to $8/10^{ths}$ were found to be impassable, with the limit being reached at lower concentrations for thicker ice and larger floes. Substantial increases in powering extended the performance limits in ice only marginally, although it is likely that increased power would offer more significant improvements to the lifeboats' open water performance in waves. As for the different hull forms, there was no discernable difference in performance during transits through pack ice, either in calm water or combined with waves, nor during maneuvering in pack.

These conclusions must be taken in the context of the experiments and their associated limitations. Ice conditions in the field are more complicated than those used in the tests. For example, ice conditions can be dynamic, such as when pack ice comes under compressive wind loads, causing its concentration to increase. As well, the tests did not look at ice features such as ridges or rubble embedded in intact ice sheets, because the evacuation craft examined are not capable of operating in such conditions; nor did the tests consider the capability of the evacuation craft in brash ice conditions that might occur in fairways or ports where ice is managed by icebreakers or similar vessels.

Still, the results provide some benchmarks of the utility of such craft in terms of pack ice conditions. By extension, they provide an indication of the requirements of either alternative or complementary evacuation systems for use in ice-covered waters.

The decision drivers for alternative or complementary evacuation systems are much broader than the performance requirements in pack ice, and it is worthwhile in this context to consider some of these. In general terms, the key aim of personnel with respect to evacuation systems in ice is similar to those in open water: access, embark, and launch an evacuation vehicle; get clear of the emerging hazard and survive until rescued; and do so in prevailing environmental and hazard scenarios without undue risk of harm.

From this overall goal, an evacuation system can be defined in other terms, such as functionality, operability, flexibility, and availability. Target functionality includes a specification of the environmental conditions in which the evacuation system must work, as well as issues such as number of personnel to be evacuated, and the distances involved. It also includes the identification of credible hazard scenarios under which evacuation might be required. Operability covers issues such as ease of use and extends to training; flexibility covers the interface between the functional requirements and the operational logistics; and availability includes issues such as maintenance requirements and reliability. To be fit for purpose, the solutions to these various needs must be compatible in a single coherent design.

What that design solution might look like depends on the specific situation. Limiting the discussion for the moment to petroleum installations, one example is a situation involving a very large number of offshore personnel working on a group of linked near shore installations in shallow water. This might be best served by fixed infrastructure that has multiple functions, including routine personnel transportation as well as evacuation. Roads in various configurations might fit this need, including ice roads, roads on pylons, in-filled causeways, and tunnels, to name a few. Another situation, say in relatively deep water and for a complement of only some 10s of personnel, might lend itself to consideration of means such as helicopters, or perhaps less conventional aircraft. While the use of roads in the first situation might negate the need for other forms of evacuation, the use of helicopters in the latter probably does not.

Indeed, there are many situations that would probably be well served by a surface vehicle of some sort. Conventional lifeboats have marginal utility once the ice season begins as they are not suited to operating in many of the ice conditions that would be common to most ice-covered regions. For heavy ice conditions, wheeled or tracked vehicles are possible solutions and have already been used for various functions in Arctic conditions. For lighter ice conditions, or during the freeze-up and break-up periods of the ice season, the weight of an ice surface vehicle will limit its utility. In these circumstances, a light weight vehicle that can operate on even rather thin ice, or a vehicle with amphibious capabilities might be an attractive solution. Various craft of this sort have already been used for roles in ice-covered water, including hovercraft for icebreaking and fan boats for search and rescue.

Space limitations prevent a fuller discussion of the merits and demerits of these and other evacuation system alternatives, but even this notional assessment highlights that there are a range of options that could be adopted or adapted to a wide range of circumstances. The best solution for a given situation will be that which stands up to a full assessment and is found to be most fit for the purpose.

## 5. ACKNOWLEDGEMENTS

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# ENVIRONMENTAL PROTECTION REGULATIONS AND REQUIREMENTS FOR SHIPS FOR ARCTIC SEA AREAS

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## SUMMARY

A challenge for shipbuilders and ship-owners when considering the construction of an anticipated new generation of ships for the commercial exploitation of the Arctic is "*what environmental protection requirements to specify*"?

An analysis of environmental protection requirements has been carried out to develop and offer considered opinions with specific reference to the following:

- *Current regulations for environmental protection applicable to Arctic shipping*
- *Possible future environmental protection requirements for Arctic shipping*

The basic premise for this analysis is that answers for this question are already in current regulations, or in the expected development of those regulations.

- *No waste discharges – of oily wastes, sewage and garbage. Requiring onboard holding tanks and spaces for the duration of the voyage*
- *No contact with exposed hull of any potential pollutants – including oil cargo, fuel oil as well as oily wastes, sewage and garbage*
- *Water ballast management by treatment*
- *Stringent limits on sulphur content of fuels – for future Sulphur Emission Control Areas (SECAs) in Arctic seas*

Overall the trend for environmental protection in Arctic sea areas is likely to be shipping specified and designed for zero emission and discharge of pollutants.

## 1. INTRODUCTION

On first impression there has been less emphasis placed on the regulation of environmental protection for shipping in Arctic sea areas than for ship safety.

This impression is reinforced in Arctic shipping conference circles with emphasis in public papers and discussion forums placed on ship safety regulation, a specific case in point being the frequent papers and discussions on efforts to harmonize the Polar Ice Classes between IACS Classification Societies.

However, in reality there have been significant efforts by international, regional and local bodies to address the environmental protection of shipping in Arctic sea areas. Furthermore a characteristic of these efforts is that they are being undertaken by a more diverse body of regulators, with both regional and local organisations playing an active role in the development of environmental protection regulation of shipping.

A challenge for shipbuilders and ship-owners when considering the construction of an anticipated new generation of ships for the commercial exploitation of the Arctic is "*what environmental protection requirements to specify*"? Such a question, if answered solely on the basis of published and implemented regulations is, in principle, relatively easy to answer.

However, in practice, the challenge lies in anticipating future regulations, and this is especially important for high specification ships being designed and built for the special trades and environments of the Arctic. High capital expenditure costs and the limited availability of substitute shipping mean that the ship's specification needs to consider, at the project concept stage, regulatory compliance for the lifetime of the vessel.

The purpose of this paper is to offer advice on the specification of environmental protection requirements, including consideration of "*future proofing*", for the following aspects:

- Hull structural arrangements
- Air pollution prevention
- Anti-fouling systems
- Ballast water management
- Processing and discharge of oily wastes, sewage and garbage

## 2. ARCTIC SEA ENVIRONMENTAL CONCERNS

The Arctic region, and its continental shelf and coastal sea areas, contain some of the largest known unexploited hydrocarbon reserves. However exploitation of these Arctic reserves has, until relatively recently, been restricted by technology, cost and the harsh climatic conditions.

Increasing global energy demands and rising crude oil and natural gas prices, combined with emerging technological advances, are now enabling exploration and production activities for hydrocarbons to commence in Arctic sea areas. Associated with these developments are expected increases in commercial shipping in the environmentally sensitive Arctic sea areas.

Arctic sea areas are some of the most environmentally sensitive and at the Arctic Council, a body which represents the eight Arctic nations, a commitment was made to:

"*Encourage continued and enhanced efforts of CAFF* [Conservation of Arctic Flora and Fauna] *in promoting the implementation of the Circumpolar Protected Area Network and relevant initiatives of the Arctic Marine Strategic Plan*." [1]


Figure 1 :Current status of the "*Circumpolar Protected Area Network*", protected parts of the Arctic coastline are shown in red, parts of the coastline which have no protected status but which have been identified for protection are shown in green [2]

The commitment of the Arctic Council to increase protection and conservation of Arctic coastal areas, as well as the general environmental sensitivity of Arctic seas, serve to highlight the importance of careful consideration of environmental protection in the specification and design of future Arctic shipping.

## 3. ORGANISATIONS AND ENVIRONMENTAL PROTECTION REQUIREMENTS

Table 1 summarises the organisations involved in establishing and implementing environmental protection regulations and guidelines for Arctic sea areas. This includes international and regional regulations.

Table 1 also summarises the other organisations involved in establishing and implementing regulations and guidelines for other (non-Arctic) environmentally sensitive sea areas with seasonal ice cover.

## 4. CURRENT REGULATIONS FOR ENVIRONMENTAL PROTECTION APPLICABLE TO ARCTIC SHIPPING

The following section considers current regulations and requirements for environmental protection of Arctic shipping for the following aspects:

- Hull structural arrangements
- Air pollution prevention
- Anti-fouling systems
- Ballast water management
- Processing and discharge of oily wastes, sewage and garbage

### 4.1 HULL STRUCTURAL ARRANGEMENTS

Current environmental protection regulations and requirements for hull structural arrangements are:

- *Fuel oil bunker tank protection* – implementation from 1st August 2010
- *Double bottom protection of oil tanker cargo pump rooms* – implementation from 1st January 2007
- *Double side and bottom cargo oil tank protection* – implemented for new ships, phase out of all existing single hull tankers by 2015.

International and Arctic area requirements are summarised in Table 3 with the most stringent requirements shown in bold text.

| Sea Area | Baltic | | | | Arctic Canada | | | | | | | Great Lakes Area | | | | | Arctic Russia | | | All Arctic Areas | | Alaska | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Organisation | FMA | SMA | Port of Primorsk | HELCOM | Transport Canada | | | | | | | Transport Canada | USCG | | | St Lawrence Seaway Dev. Corp | Russian Register | NSRA | Russian Federal Government | IACS | IMO | Alaska State | |
| Regulations | Finnish Swedish Ice Class Rules | | Port Regulation | Convention on the Protection of the Marine Environment of the Baltic Sea Area and subsequent recommendations | Joint Industry Coast Guard guidelines for the control of oil tankers and bulk chemical carriers in ice control zones of Eastern Canada | Arctic Shipping Pollution Prevention Regulations | Equivalent Standards for the Construction of Arctic Class Ships | Guidelines for the Operation of Tankers and Barges in Canadian Arctic Waters (Interim) | Ballast Water Control and Management Regulations | A guide to Canada's Ballast Water Control and Management Regulations | Pollution Prevention Guidelines for the Operation of Cruise Ships under Canadian Jurisdiction | Great Lakes Sewage Pollution Prevention Regulations | Reporting by Foreign flagged Vessels bound for the Great Lakes Federal Register Notice 19742 | Ballast Water Management for Control of Non Indigenous Species in the Great Lakes and Hudson River | USCG Ballast Water Management | The Seaway Handbook | Rules for the Classification and Construction of Sea-Going Ships | Requirements for the Design, Equipment and Supplies of Vessels Navigating the Northern Sea Route | Arctic Port Regulations | Unified Requirements for Polar Ships | Guidelines for ships operating in Arctic ice-covered water | Certain Alaskan Cruise Ship Operations | Commercial Passenger Vessel Environmental Compliance Program - 18 AAC 69 |
| Applicability of Regulations | Finnish Waters | Swedish Waters | Primorsk Waters | Baltic | Canada | Canada | Canada | Canada | Canada | Canada | Canada | Great Lakes | Great Lakes | Great Lakes | Great Lakes & Hudson River | Great Lakes / St. Lawrence | Northern Sea Route | Northern Sea Route | Russian Arctic | Arctic | Arctic | Alaska | Alaska |
| Hull Structural Arrangements | ✓ | | | ✓ | ✓ | ✓ | ✓ | | | | | ✓ | | | | | ✓ | ✓ | ✓ | ✓ | ✓ | | |
| Air Pollution Prevention | | | | ✓ | | | | | | | ✓ | | | | | | | | | | | | ✓ |
| Anti-Fouling Systems | | | | ✓ | | | | ✓ | | | ✓ | | | | | | | | | | | | ✓ |
| Ballast Water Management | | | ✓ | | | | | | | ✓ | | | ✓ | ✓ | ✓ | | | | ✓ | | | | |
| Processing & Discharge of Oily Waste, Sewage, Garbage | | | ✓ | ✓ | | ✓ | | | ✓ | | ✓ | ✓ | | | | ✓ | | ✓ | ✓ | | | ✓ | ✓ |

Table 1 : Organisations, regulations and scope of application of regulations for Arctic sea areas as well as other seasonally ice covered sea areas

### 4.1. (a) International Requirements – Hull Structural Arrangements

Requirements for preventing oil pollution of the marine environment through structural protection of tanks containing oils are found in *MARPOL Annex I* [4] and are indicated in Table 2. These regulations include the extent of wing tanks and the minimum double bottom height.

Regulation 12A of *MARPOL Annex I* enters into force for all vessels delivered after 1st August 2010 and for vessels with a contract date after 1st August 2007. This regulation includes provision for fuel oil tank protection by means of either side and bottom protection or by complying with an accidental oil outflow performance standard. Regulation 12A requires that oil fuel tanks be located above the moulded line of the bottom shell plating, or inboard of the side shell plating, nowhere less than a distance defined with respect to vessel beam, with an absolute minimum of 0.76 metre and 1.0 metre respectively.

| Reference | Details |
|---|---|
| Regulation 19 | Double hull |
| Regulation 12A | Fuel oil tank protection |
| Regulation 22 | Pump room bottom protection |
| Regulation 27 | Intact stability when in port |
| Regulation 28 | Damage stability |
| Regulation 23 | Accidental oil out flow performance and tank size limitations (for tankers delivered on or after 1 January 2010) |
| Regulation 25 | Hypothetical outflow of oil |
| Regulation 26 | Limitation of size and arrangement of cargo tanks |

Table 2 *MARPOL Annex I* main structural requirements for environmental protection ('new' regulation numbering)

Regulation 22 applies to the cargo pump rooms of oil tankers of 5000 deadweight tonnes and above, constructed after 1st January, 2007. The regulation requires the cargo pump room to be provided with a double bottom so that, at any cross section, the depth of each double-bottom tank or space is at least 1.0 metre from the bottom of the pump room to the ship's baseline. As stability and survivability of the ship is a necessary pre-requisite for environmental protection requirements for intact and damage stability requirements are also included in MARPOL.

Furthermore containing damage through subdivision is essential to restricting oil outflow, especially in hazardous ice covered Arctic sea areas.

International intact stability standards are those found in the *Intact Stability (IS) Code - Intact Stability for All Types of Ships (IMO Resolution A.749(18) as amended* [5]). The Code addresses ice accretion on deck, although this is the only part of the Code that specifically addresses operation in cold climates.

### 4.1. (b) Arctic Area Requirements – Hull Structural Arrangements

Limiting the potential risk of an accidental oil discharge following structural damage is addressed in all regulations. In the majority of regulations the provision for tank protection (be it bunker fuel or cargo oil) is required, although in Arctic areas it is only the Canadian requirements which specify actual tank dimensions.

For regulations where no dimensions are specified it can be assumed that the de facto standard will be IMO (*MARPOL Annex I* Regulation 19) for oil cargo tank protection and Classification Society Rules (where they exist) for double bottom requirements in other ship types. In the Canadian *ASPPR* the side tank dimensions are related to the Arctic Class of the vessel, although the requirement for protective tanks is limited, for example, a reduction in the required scantlings is permitted.

The Canadian *Equivalent Standards* and the IMO *Guidelines for Ships Operating in Arctic Ice Covered Waters (Arctic Guidelines)* [15] also stipulate that no oil should be carried in contact with the hull.

| *Regulation* | *All Arctic Sea Areas* | *Canadian Arctic* | *Russian Arctic* | |
|---|---|---|---|---|
| *Detail* | Guidelines for ships operating in ice covered waters (2002) | ASPPR (1978/1996)<br>§ - Equivalent Standards for Construction (1998)<br># - Interim guidelines for operation of tankers & barges in Canadian Arctic (1998) | Northern Sea Route Administration: Guide to Navigating the NSR (1996) | Russian Maritime Register of Shipping<br>Rules for the Construction of Ships (2005) |
| *Wing Tanks* | No pollutant (oil) to be in contact with hull. | **Side tank dimension dependent on Arctic Class (0.91 – 1.83m).**<br>**Waste (any pollutant) at least .075m from side shell.#** | Recommended double side between FP & AP, required iwo ER for some ice classes. | |
| *Double Bottom* | Double bottom between FP & AP bulkheads.<br>No pollutant (oil) to be in contact with hull. | Double bottom required iwo oil tanks.<br>**Waste (any pollutant) at least .075m from side shell.§** | Double bottom between FP & AP bulkheads. | Minimum double bottom height 0.65m. |
| *Bunker Oil* | No pollutant (oil) to be in contact with hull. | **Waste oil to be 0.75m from shell§**<br>**Fuel or Cargo oil to be 0.76 from side shell.#** | Double bottom & sides not to be used for oil products. | |
| *Stability* | Intact, damage. | **Ice accretion, Intact, damage, ramming.§** | References IMO for intact. Additional requirements for damage. | |

Table 3 : Hull structural arrangements

With respect to Arctic area regulations an appropriate standard to apply the requirement for no oil (be it cargo or bunkers) to be carried against the exposed hull would be the new Regulation 12A of *MARPOL Annex I*. This effectively ensures that bunker tanks are protected by a double skin in the same way as cargo oil tanks.

However, the probabilistic method permitted in Regulation 12A for bunker tank location could create, in theory, an allowable solution where there are some bunker tanks in direct contract with the vessel's hull (albeit small tanks, with low outflow rates). Hence care should be taken if the probabilistic approach is adopted for design to ensure the current Arctic sea regulation requirements for no oil carried against the exposed hull are met.

Stability is also a consideration for survivability and consequent environmental protection in Arctic sea areas. For navigating on the NSR the IMO *Intact Stability Code* is referenced as the standard of intact stability. For other Arctic sea areas the regulations give additional requirements for intact stability (in terms of metacentric height (GM) and area under the righting lever (GZ) curve).

Furthermore, there are important considerations for damage (ice induced) stability required for all Arctic sea areas. The Canadian requirements [6,7] and those contained within the IMO *Arctic Guidelines* are equivalent in terms of the damage dimensions, although the Canadian *Equivalent Standards* also include ramming stability criteria according to the vessel's operational profile. Northern Sea Route Administration (NSRA) requirements for damage [10] are generally less strict than the Canadian standards and the *Arctic Guidelines*.

## 4.2 AIR POLLUTION PREVENTION

Current environmental protection regulations and requirements for air pollution prevention are:

- *NOx emissions limitations* – IMO MARPOL Annex VI in force
- *Bunker fuel sulphur content limitations* – IMO MARPOL Annex VI in force

### 4.2. (a) International Requirements – Air Pollution Prevention

The requirements of *MARPOL Annex VI* [22] set out international regulations for the prevention of air pollution from ships and aim to regulate the emissions discharged from marine engines. In particular Regulation 13 limits diesel engine Nitrogen Oxide (NOx) emissions. An exhaust gas cleaning system which reduces NOx emissions to the required standard can also be employed if the engine itself is not suitably designed.
Regulation 14 of *MARPOL Annex VI* limits the sulphur content of marine fuel, depending on the operating area of the vessel. The limiting value of 4.5% sulphur content applies globally.

| *Regulation* | *Maximum Sulphur Oxide Emissions* | *Maximum Nitrogen Oxide Emissions* | *Maximum Sulphur Content of fuel* |
|---|---|---|---|
| MARPOL – Annex VI | - | 17g/kWh | 4.5% |
| MARPOL – Annex VI (SECA) | 6.0g/kWh | 17g/kWh | 1.5% |
| EC Directive 2005/33/EC | - | - | 0.2% (07/2006) 0.1% (07/2010) |

Table 4 Air Pollution Regulations

*MARPOL Annex VI* also designates the Baltic Sea as *'Sulphur Emission Control Area'* (SECA) where the maximum sulphur content of the fuel is reduced to 1.5%.

| ***MARPOL Annex VI*** **SECA** | *Implementation date* |
|---|---|
| Baltic Sea | May 19 2006 |
| North Sea | November 22 2007 |

Table 5 Current SECAs and *MARPOL Annex VI* implementation dates

### 4.2. (b) Arctic Area Requirements – Air Pollution Prevention

Within the regulations reviewed for Arctic sea areas there are no requirements for air pollution which exceed those set out in *MARPOL Annex VI*. The only regulations which consider air pollution are the *Pollution Prevention Guidelines for the Operation of Cruise Ships under Canadian Jurisdiction* [24] which refer directly to the requirements in *MARPOL Annex VI*. There are no SECAs within Arctic sea areas by international or Arctic area regulations.

The extension of an EU Directive (2005/33/EC) to include marine applications will, though, have an impact on both fuel quality requirements and ship borne engine emissions in the future with more stringent limitations on sulphur content of fuel as indicated in Table 4.

## 4.3 ANTI-FOULING SYSTEMS

Current environmental protection regulations and requirements for anti fouling systems are:

- *No exposed anti-fouling paint containing organotin compounds* – IMO anti-fouling system (AFS) Convention not yet in force
- *Total ban on tributyltin (TBT) contained within anti-fouling paints in European Waters* – in 2008 by EU Directives

4.3. (a) International Requirements – Anti Fouling Systems

The protection of the marine environment from tributyltin (TBT) contained within anti-fouling paints on vessel hulls is regulated by the *International Convention on the Control of Harmful Anti-Fouling Systems on Ships (AFS Convention)* [17] adopted by the IMO in 2001.

However, the convention has yet to be ratified by Member States, and is therefore not yet enforceable globally. The convention requires that by 1st January 2008 no vessel will use exposed anti-fouling paint containing organotin compounds. Paints containing these compounds must either be removed or sealed over by a barrier to prevent leaching.

4.3. (b) Arctic Area Requirements – Anti-Fouling Systems

Although the *AFS Convention* is not yet in force EC Directives have, on the other hand, made the application of such TBTs illegal from 2003 and a total ban will be in force in European waters by 2008. HELCOM has also urged its member states to ratify the IMO convention to encourage the *AFS Convention* to be enforced.

From available literature there is no indication of regulations relating to the control of TBT anti-fouling systems in Russian Waters. Currently the Russian Federation has not yet ratified the *AFS Convention*. However the unavailability of TBT containing paint has effectively implemented the *AFS Convention.*

4.4 BALLAST WATER MANAGEMENT

Current environmental protection regulations and requirements for ballast water management are:

- *Requirements for control and management of ballast water and sediments* – IMO ballast water management (BWM) convention not yet in force and enforceable
- *Ballast water treatment equipment to be provided and elimination of ballast water exchange* – by 2016 when the BWM Convention is ratified
- *Certain sea areas where water ballast cannot be exchanged* – by current USCG and Transport Canada regulations

4.4. (a) International Requirements – Ballast Water Management

IMO has adopted requirements for the management of water ballast and sediments in ballast tanks (and its associated discharge and treatment) through the *International Convention for the Control and Management of Ships' Ballast Water and Sediments, (BWM Convention)* 2004 [18].

However, at present this convention has not been ratified by sufficient countries to meet the entry into force criteria and is therefore not enforceable worldwide. The Convention contains standards to be achieved during water ballast exchange in regulation
D-1. An overview of the main requirements is given in Table 7.

| *Reference* | | *Details* |
|---|---|---|
| Ballast Water Exchange | Regulation D-1 | 95% volumetric exchange or pumping through three times the tank volume<br>At least 200 nm off shore and more than 200m in depth |
| Ballast Water Treatment | Regulation D-2 | Treatment to meet defined standards in regulation D-2 |
| Ballast Water Management | Regulation B-4 | Provision of ballast water management plan - |
| Ballast Water Treatment | Regulation D-4 | Prototype ballast water treatment technologies |
| Ballast Water Performance Standard | Resolution MEPC.126(53) | Performance in terms of monitoring the number of foreign organisms present |

Table 7 Key requirements of the *BWM Convention*

| *Regulation* / *Details* | *Great Lakes* | *Canadian Arctic* | |
|---|---|---|---|
| | USCG Ballast Water Management for the Control of Non indigenous Species in the Great Lakes and Hudson River (1993) | A guide to Canada's Ballast Water Control and Management (2006) | Ballast Water Control & Management Regulations (2006) |
| Area | **Exchange 200nm offshore in at least 2000m water depth before entering lakes. Requirements for Areas where Water Ballast should not be exchanged.** | **Areas for water ballast exchange before entering certain ports.** | Exchange 200nm offshore in at least 2000m water depth<br>Or Exchange 50nm in at least 500m water depth (if vessel does not navigate more than 200nm offshore) |
| Standard | Minimum salinity of 30 pp thousand. | References IMO Regulation D-2. | 95% Volumetric exchange<br>Specific concentrations of organisms<br>30 pp thousand salinity of conducted 50nm from shore. |
| Plan & Reports | **Submit info on last Water Ballast exchange before entering Great Lakes.** | | Requires Ballast Water Management Plan. |

Table 6 Ballast water management

The *BWM Convention* contains provision for the use of approved ballast water treatment equipment. Some treatment systems are installed on a limited number of ships engaged in trials of treatment technologies. Section C of the Convention enables flag states to require additional measures for ballast water management.

If ratified it is the intention of the *BWM Convention* to eliminate ballast water exchange as an option for ballast water management by 2016. This would leave ballast water treatment as the only solution for managing water ballast. However, this is also dependent on the continuing development of the emerging technology in ballast water treatment.

4.4. (b) Arctic Area Requirements – Ballast Water Management

Although the *BWM Convention* is not yet in force its development has produced a standard level of water ballast management throughout the world. This is equivalent to the previous requirements of some more specific and sensitive areas, including those in Arctic sea areas.

The Canadian Arctic area requirements for ballast water management are summarised in Table 6 and compared with some of the most stringent requirements, shown in bold text, i.e. USCG requirements for the Great Lakes.

However, there can be practical problems for implementation of ballast water exchange regulations. Specifically discussions held at HELCOM have raised doubts regarding the practicality of applying the *BWM Convention* to the Baltic Sea. There are no areas in the Baltic Sea with a water depth greater than 200 metres which are more than 50 nautical miles offshore. Consequently the Baltic Sea is unsuitable for water ballast exchange in strict compliance with the current *BWM Convention* and similar practical limitations exist in shallow Arctic sea areas.

Local distinctions exist in some sea areas between the requirements for ballast water management and exchange. In particular both the USCG [19] and the *Canada Shipping Act - Ballast Water Control and Management Regulations* [20] identify specific areas either where water ballast should not be exchanged (due to a sensitive local environment) or conversely areas where water ballast exchange may be carried out before entering certain ports. The volumes of the ballast water exchanged are generally consistent across the regulations. To regulate the ballast water being discharged into such sensitive environments the USCG requires reporting of the last discharge of water ballast before entering certain ports.

4.5 PROCESSING AND DISCHARGE OF OILY WASTES, GARBAGE AND SEWAGE

Current environmental protection regulations and requirements for processing and discharge of oily waste, garbage and sewage are:

- *15 ppm oil water separator* – IMO MARPOL Annex I in force
- *Requirements for processing of garbage* – IMO MARPOL Annex V in force
- *Requirements for management and treatment of sewage* – IMO MARPOL Annex IV in force
- *Options for the management and treatment of garbage and sewage* – IMO MARPOL convention Annexes IV and V

4.5. (a) International Requirements – Processing and Discharge of Oily Wastes, Garbage and Sewage

Table 8 is an overview of the IMO MARPOL requirements for processing and discharge of oily wastes, garbage and sewage. *MARPOL Annex I* requirements are for the performance of oily water separators with a limitation of 15 parts per million (ppm) oil permitted for discharge. In addition *MARPOL Annex I* includes requirements for the treatment of oily waters before discharge with the separator arranged such that oily waters with higher concentrations (than 15 ppm) are re-circulated into a holding tank.

Regulation 15 of *MARPOL Annex I* also indicates requirements for operating in special sea areas, including the Antarctic where no discharge to sea of oil or oil/water mixture is permitted.

Requirements associated with the processing of garbage (incinerators) are contained in *MARPOL Annex V* [25]; The *Guidelines for the Implementation of Annex V of MARPOL 73/78 (Annex V Guidelines)* [27] give a number of options for garbage handling, including the use of incinerators, compactors and comminuters, with notes regarding processing of common garbage items. The guidelines to Annex V give an indication to the process of handling and discharging garbage types, in addition to specifying where, if at all, such waste can be discharged.

The requirements for the management and treatment of sewage (sewage treatment systems or holding tanks) are outlined in *MARPOL Annex IV* [26]. Both *MARPOL Annex IV* and *Annex V* contain optional requirements giving some flexibility to the operator in relation to how sewage and garbage can be managed. For sewage, the waste can either be stored in a holding tank, treated using a sewage treatment plant, or disinfected. The regulations also dictate in which sea areas treated and disinfected sewage can be discharged.

| *Regulation* | *Reference* | *Details* |
|---|---|---|
| MARPOL Annex I | Regulation 15 | Methods for the prevention of oil pollution from ships while operating in special areas [requirements for the Antarctic that ships have sufficient capacity for retention of sludge and oily wastes while in that area] |
| | Regulation 14 | Oil filtering equipment and 15 parts per million alarm |
| MARPOL Annex IV | Regulation 9.1 | Requirement for a holding tank (as one of 3 options for sewage management) |
| | Regulation 9.2 | Disinfection systems |
| | Regulation 9.3 | Requirements for a type approved sewage treatment plant (as one of 3 options for sewage management) |
| | Resolution MEPC.157(55) | Recommendations for the minimum distance offshore and minimum speed for any sewage discharge |
| MARPOL Annex V | Guidelines for implementing MARPOL Annex V | Shipboard equipment for processing garbage (Incinerator, compactors, grinders and comminutors) |

Table 8 Overview of IMO requirements for sewage, oily water and garbage related equipment.

4.5. (b) Arctic Area Requirements – Processing and Discharge of Oily Wastes, Garbage and Sewage

The discharge of treated wastes, be it garbage, sewage or oily water is very much dependent on the area of operation. For example the requirements of the Canadian *ASPPR* state that untreated sewage, which has been produced on the ship, may be discharged in Arctic waters, consequently for Canadian Arctic waters the requirements of *MARPOL Annex IV* requirements would prevail. The *Guide to Navigating the Northern Sea Route* requires collecting tanks to have piping to both sides of the ship for discharge to shore.

Oily water filtering equipment are common amongst all ship types, with the standard performance set by *MARPOL Annex I* as 15 parts oil per million parts water as a maximum discharge limit.

Most Arctic regulations contain this performance figure, such as those of the NSR. However the Canadian Arctic requirements, although equivalent to MARPOL for external waters, have stricter standards for some internal waters.

Management of waste is a consideration for all areas, except the NSR. Recording of discharges has already been mentioned, but in addition to this there are requirements for incorporating waste management into the shipboard operations manual. MARPOL gives the ship several options for managing waste. It is important that these options are considered as complementary in some cases in order to ensure all the regulations are met.

## 5. POSSIBLE FUTURE ENVIRONMENTAL PROTECTION REQUIREMENTS FOR ARCTIC SHIPPING

Based upon the most stringent requirements from environmentally sensitive sea areas with seasonal ice cover, as well as expert opinion from specialist staff with domain and regulatory knowledge, a view of possible future environmental protection requirements for Arctic shipping is offered in the following sections.

| *Regulation* | *Sea off Alaska* | *Baltic Sea* | *Canadian Arctic* | *Great Lakes* | *Russian Arctic* |
|---|---|---|---|---|---|
| *Detail* | Commercial Passenger Vessel Compliance Program 18 AAC 69 (2006) | HELCOM (1992) | Pollution Prevention Requirements for Cruise Ships in Canadian Waters (2004) ASPPR (1978/96) - # Equivalent Standards§ | Canada Shipping Act – Great Lakes Sewage Pollution Prevention Regulations (2005) Seaway Regulations (2006)* | Northern Sea Route Administration: Guide to Navigating the NSR (1996) |
| Discharge | May discharge <6kts & within 1nm of shore subject to continuous monitoring levels. | Waste categories for discharge / retention in Baltic. | Grey water discharge: Speed 6 knots, >4nm from shore. Untreated sewage may be discharged in the Arctic.# | **Concentrations & quality of sewage discharged Sampling point on sewage tanks.** | Collecting tanks must have deck piping to both sides of ship. |
| Oily Water Separators | | **5.0 $m^3/h$ throughput (for ships >15,000GRT).** | **5ppm (internal waters). 15ppm (external waters).** | | Bilge water separator performance 15 ppm. |
| Retention | | | **No waste to be held in tanks against side shell.**§ | Double bottom not to be used for holding tank. | Sewage Collecting tank (if no treatment system) & oily water storage tank volume required for 30 day navigation. |
| Incineration | | **Incineration of ship generated wastes prohibited.** | Manage incinerator ash. | | Requires incinerator for contaminated wastes or a storage tank for 30 day navigation. |
| Management | **Maintain sewage discharge records for 12 months. Hazardous waste offloading plan. Non hazardous solid waste disposal plan Sampling Plan.** | | Incorporate waste management practices in to operations manual. | Records to be kept for location where bilge water is discharged.* | |

Table 9 Processing and Discharge of Oily Wastes, Garbage and Sewage

### 5.1 HULL STRUCTURAL ARRANGEMENTS

Possible future environmental protection requirements for hull structural arrangements include:

- *Zero discharge – requiring holding tanks (duration of voyage or 30 days)*
- *No oil/oily water to be in contact with vessel hull*
- *No wastes stored in contact with vessel hull* – oily wastes, garbage and sewage

The *Great Lakes Sewage Prevention Regulations* [30] require that the double bottoms should not be used as sewage holding tanks. This is the only regulation which stipulates location of such tanks at present. However, this points to the future, where a double hull may be required for all substances that are considered pollutants.

Disposal of garbage into the sea is currently not permitted in the Baltic under HELCOM. Incineration is also banned in the Baltic. It could be anticipated that this ban could be applied to Arctic sea areas, which are deemed as sensitive environments.

Regulations for the monitoring of discharge levels of waste vary with area. With a variety of different levels future vessel design should consider suitably large holding tank(s) to store wastes. The Russian regulations for the NSR require storage appropriate for 30 day navigation (if there is no treatment system) but this figure may also be appropriate as a secondary measure of ensuring compliance in the future. This is particularly relevant in the case of a vessel operating on a number of different routes in the Arctic – retaining waste will always achieve compliance.

### 5.2 AIR POLLUTION PREVENTION

Possible future environmental protection requirements for air pollution prevention include:

- *Arctic sea areas become SECAs*
- *EC Directive levels of fuel sulphur content adopted in environmentally sensitive sea areas in Arctic*
- *Reductions in fuel sulphur content* – under MARPOL Annex VI

The Baltic Sea has already been designated a SECA. For a sensitive environment such as the Arctic the designation of sea areas within the Arctic as SECAs seems likely by future regulation.

Similarly although the proposed EC directive to reduce the Sulphur content of marine fuel is unlikely to become an international standard in the near future, the standard may well be adopted for SECAs. Some Arctic areas would therefore see a requirement for reduced sulphur content fuels if SECA status (or SECA equivalency) is adopted. In addition there are proposals to reduce sulphur limits which could effectively supersede SECAs as they currently exist.

There is, however, some uncertainty on the outcome of revisions to MARPOL Annex VI with the agenda and debate at IMO in April 2007 being widened to include the consideration of green house gas and $C0_2$ emissions by an expert working group.

### 5.3 ANTI FOULING SYSTEMS

Possible future environmental protection requirements for anti-fouling systems include:

- *AFS mandatory in the Arctic areas* – before being adopted internationally

As TBT containing paints are no longer available the Convention is effectively implemented. A total TBT paint ban in sensitive areas, such as the Arctic, is likely before the AFS convention itself is ratified.

### 5.4 WATER BALLAST MANAGEMENT

Possible future environmental protection requirements for water ballast management include:

- *Water ballast treatment mandatory in Arctic sea areas* – before being adopted internationally

The practical problem of application of ballast water exchange regulations to the shallow sea areas of the Arctic is likely to lead to the implementation of ballast water treatment methods, once technology has been proven, in the Arctic in advance of the implementation of the BWM convention.

### 5.5 PROCESSING AND DISCHARGE OF OILY WASTES, SEWAGE AND GARBAGE

Possible future environmental protection requirements for processing and discharge of wastes include:

- *Retention of all wastes on-board*
- *Use of Oily Water separators in the Arctic banned*
- *Use of Incinerators in the Arctic banned*

For the Antarctic the *'no discharge to sea'* restriction indicates the environmental sensitivity of Antarctic sea areas. In addition the infrastructure in the Antarctic is limited and discharge/offloading of oily wastes to land facilities is practically unfeasible.

Consequently in the Antarctic it is required that the vessel has sufficient holding tanks for retaining oily wastes throughout the voyage, for discharge at a suitable facility once outside the area.

| Sea areas *(principal regulatory document)* | | All Arctic Seas *(Possible future regulation?)* | All Seas *(MARPOL)* | All Arctic Sea areas *(IMO Arctic Guidelines)* | Russian Arctic *(Guide to Navigating the Northern Sea Route)* | Canadian Arctic *(ASPPR & Equivalent Standards)* |
|---|---|---|---|---|---|---|
| Environmental Protection Issue | Detail | | | | | |
| Hull Structural Arrangement | Double hull | Required for any potential pollutant anywhere in length | For oil tankers | No oil pollutant in direct contact with side shell | Recommended double side. Required iwo ER for some ships. | Required side tank dimensions. Required iwo oil tanks. No harmful waste in direct contact with side shell |
| | Double bottom tanks | Required through ships length | For oil tankers. Pump Room bottom protection required. | Between FP and AP bulkheads | Between FP and AP bulkheads | Required iwo oil tanks No harmful waste in direct contact with side shell |
| | Bunker oil tanks | Not against side shell | Double skin protection required for tanks | No oil Pollutant in direct contact with side shell | Double bottom and sides not to be used for oil products. | Fuel or Cargo oil to be 0.76m from side shell. Waste oil to be 0.76m from shell |
| | Waste (garbage, sewage) tanks | Not against side shell | | No harmful pollutant in direct contact with side shell | Sewage collecting tank (if no treatment system) | No harmful waste in direct contact with side shell |
| | Stability | Intact, Damage, Ramming | Intact (only ice consideration is deck accretion ) and Damage | Intact and Damage | Intact and Damage | Intact, Damage, Ramming requirements. |
| Anti-Fouling Systems | TBT free coatings | Required | Effectively implemented | | | Required for cruise ships |
| Water Ballast Management | Water ballast control method | Treatment | Exchange or Treatment | | | Specific Areas for exchange |
| Air Pollution Prevention | Maximum Sulphur Oxide emissions | 6g/kWh | | | | |
| | Maximum Nitrogen Oxide emissions | 17g/kWh | 17g/kWh | | | |
| | Maximum Sulphur content of fuel | 0.1% | 4.5% | | | Average for cruise ships 1.5% |
| Processing and Discharge of Oily Wastes, Garbage, Sewage | Oily Water Separators | Not Permitted – Retain oil on board | Performance - 15ppm | | | 5ppm (some internal waters) 15ppm (external waters) |
| | Incineration | Not Permitted – Retain waste on board | Permitted | | Required or a storage tank | Not permitted in port for cruise ships |
| | Discharge (sewage) | Retain sewage on board | Storage or discharge | | Storage or discharge | Storage or discharge |
| | Discharge (garbage) | Retain garbage on board | Storage or discharge | | Storage or discharge | Storage or discharge |

Table 10 : Environmental protection for Arctic Shipping - current sea area requirements and a considered opinion on possible future requirements

Similar environmental sensitivities and limitations of shore discharge exist in Arctic sea areas and future regulation to retain oily wastes on board could reasonably be anticipated.

The use of incinerators is given as an option in *MARPOL Annex V* for managing garbage. In the *Guide to Navigating the Northern Sea Route* incineration is the only option, other than retention of garbage in a holding environment.

Russian Arctic Port entry regulations specify that incinerators must not be used within the port's limits, whilst there is a complete ban on incinerators in the Baltic Sea. Therefore incineration can not be relied upon for garbage disposal and a likely future requirement is for the retention of garbage onboard.

## 6. ENVIRONMENT PROTECTION FOR FUTURE ARCTIC SHIPPING

Table 10 offers a summary of possible future environmental protection requirements for Arctic shipping based upon the analysis carried out for this paper.

Current sea area requirements are also shown in Table 10 for Arctic sea areas, with requirements identified in the table which are more stringent or differ significantly from current international regulation.

The environmental protection requirements for future Arctic ships may be summarised as follows:

- *Hull structural arrangements* – Future regulations may require that all potential pollutants including all wastes be kept separate from the shell.
- *Air pollution prevention* – In future more stringent emissions limits for sulphur content and emissions may be applied for Arctic ships in association with Sulphur Emission Control Areas (SECAs) in Arctic seas. Furthermore with the consideration of $CO_2$ and green house gas emissions at IMO during 2007 the development of the MARPOL Annex VI regulations and requirements to address these issues could also be anticipated.
- *Anti-fouling systems* – In future a total TBT paint ban in Arctic seas is likely before the AFS convention itself is ratified.
- *Ballast water management* – The problem of application of ballast water exchange regulations to shallow Arctic seas areas is likely to lead to the early implementation of ballast water treatment methods in the Arctic.
- *Processing and discharge of oily wastes, sewage and garbage* – The holding of wastes and separation of waste holding spaces from the exposed hull may be applied in future for Arctic ships in addition to bans on the use of oily water separators and garbage incinerators.

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The opinions expressed in this paper are those of the authors and do not necessarily reflect the opinions of the Lloyd's Register Group.

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# FUTURE SEA ICE OPERATING CONDITIONS IN THE ARCTIC OCEAN

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## SUMMARY

The Arctic sea ice cover is undergoing an extraordinary transformation that has significant implications for marine access and shipping throughout the Arctic basin. The Arctic Climate Impact Assessment (ACIA), released by the Arctic Council in November 2004, documented that Arctic sea ice extent has been declining for the past five decades; sea ice thickness has been decreasing during the same period; and, the area of multiyear ice has also been decreasing in the Arctic Ocean. The five Global Climate Models (GCMs) used in the ACIA simulate a continuous decline in sea ice coverage through the 21$^{st}$ century; one model showed it is plausible that during mid-century the entire Arctic Ocean could be ice-free for a short period in summer. Recent research has indicated this ice-free state of the Arctic sea ice cover may occur as early as 2040, if not sooner. The resulting sea ice conditions for Arctic marine activities will be challenging and will require substantial monitoring and improved regional observations. This 'new' Arctic Ocean will require enhanced marine safety and marine environmental protection systems based, in part, on these changing conditions that allow increasing marine access.

## NOMENCLATURE

ACIA Arctic Climate Impact Assessment
AMSA Arctic Marine Shipping Assessment
FY first-year sea ice
GCM Global Climate Model
MY multiyear sea ice

## 1. INTRODUCTION

In assessments of ongoing and projected climate change, Arctic sea ice is a critical element. Observed sea ice data for the past five decades indicates a decrease of sea ice coverage over the Arctic Ocean, especially in summer. Interestingly, the five smallest September ice-covered areas for the Arctic Ocean during the modern satellite record (1979-2006) have been the most recent five seasons (2002-2006). The actual 'loss' of Arctic sea ice coverage in September during this 27-year period is estimated to be 100,000 square kilometers per year. Figure 1 shows the coverage at the time of minimum extent of Arctic sea ice on 6 September 2005. This 'snapshot' represents the minimum extent of Arctic sea ice in the satellite era of observations. Striking are the large ice-free areas across the Russian Arctic coastal seas (north of the Eurasian coast); the ice edge has retreated north of Svalbard and well north in the Beaufort and Chukchi seas. These extraordinary changes in the summer ice cover of the Arctic Ocean are major factors in the potential lengthening of the navigation season in regional Arctic seas.

## 2. ARCTIC CLIMATE IMPACT ASSESSMENT

The Arctic Climate Impact Assessment (ACIA) released in November 2004 was called for by the Arctic Council and the International Arctic Science Committee. ACIA found that the Arctic is extremely vulnerable to observed and projected climate change and its impacts. The Arctic is now experiencing some of the most rapid and severe climate change on earth. During the 21$^{st}$ century, climate change is expected to accelerate, contributing to major physical, ecological, social and economic changes, many of which have already begun. Changes in Arctic climate will also affect the rest of the planet through increased global warming and rising sea levels. Of direct relevance to future Arctic marine activity is that potentially accelerating Arctic sea ice retreat improves marine access throughout the Arctic Ocean [1].

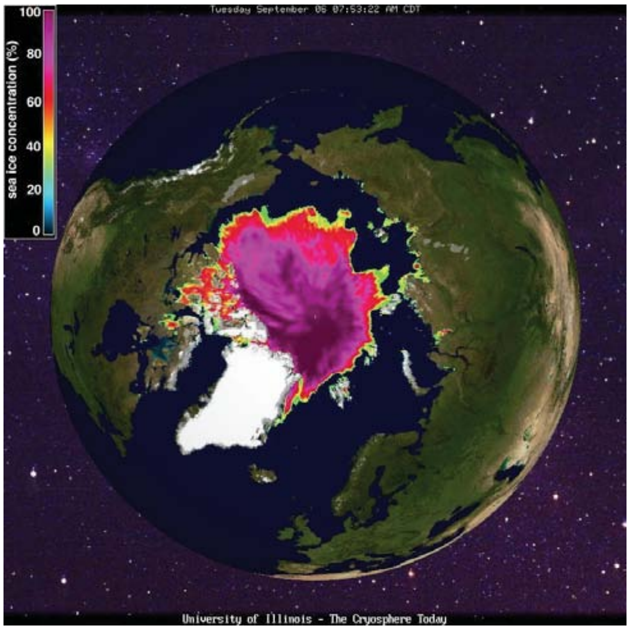


Figure 1: Summer minimum extent of Arctic sea ice on 6 September 2005 (a record summer minimum) (University of Illinois & NOAA).

ACIA documented that declining Arctic sea ice is a key climate change indicator. During the past five decades the observed extent of Arctic sea ice has declined in all seasons, with the most prominent retreat in summer. Each of the five Global Climate Models (GCMs) used in ACIA project a continuous decline in Arctic sea ice coverage throughout the 21st century. One of the models projects an ice-free Arctic Ocean in summer by 2050, a future scenario of great significance for Arctic marine shipping since multiyear (MY) ice could possibly disappear in the Arctic Ocean. All of the next winter's ice would be first-year (FY). GCM projections to 2100 suggest that Arctic sea ice in summer will retreat further and further away from most Arctic coasts, potentially increasing marine access and extending the season of navigation in nearly all Arctic regional seas. One limitation of the GCMs is that they are not useful for determining the state of sea ice in the Northwest Passage region. Their resolution is much too coarse to be applied to the narrow straits and sounds of the Canadian Arctic Archipelago. In ACIA the only reliable observed data for the region comes from the Canadian Ice Service and this information, archived since the late 1960s, shows a mean negative trend of sea ice coverage in the Canadian Arctic Archipelago, but very high inter-annual variability. The ACIA models, however, could be applied to the more open coastal seas of the Russian Arctic. ACIA sea ice projections for Russia's Northern Sea Route indicate an increasing length of the navigation season throughout the 21st century [1].

In summary, ACIA confirms that the observed retreat of Arctic sea ice is a real phenomenon. The GCM projections to 2100 show extensive open water areas in summer around the Arctic basin (Figure 2). Thus, it is highly plausible there will be increasing regional marine access in all the Arctic coastal seas. However, the projections show only a modest decrease in winter Arctic sea ice coverage; there will always be an ice-covered Arctic Ocean in winter although the ice may be thinner and may contain a smaller fraction of MY ice. The very high, inter-annual variability of observed sea ice in the Northwest Passage and non-applicability of the GCMs to the region, prevent an adequate assessment of this complex region. Although the ACIA projections indicate an increasing length of the navigation season for the Northern Sea Route, detailed quantification of this changing marine access is testing the limitations of today's GCMs. There is a definite need for improved Arctic regional models to adequately assess future changes in sea ice extent and thickness, and their considerable implications for expanded marine uses of the Arctic Ocean.

The final ACIA report lists ten major findings which are essentially the key impacts of climate change on Arctic people and the environment. ACIA key finding #6 states 'Reduced sea ice is very likely to increase marine transport and access to resources.' One of the follow-on Arctic Council activities addressing this finding is the ongoing Arctic Marine Shipping Assessment (AMSA). This effort will review the current levels of Arctic marine activity and create a set of scenarios (plausible futures) of Arctic Ocean marine use for 2020 and 2050. A range of environmental, social and economic impacts resulting from increasing Arctic marine use will be evaluated, and future marine safety and environmental protection strategies will be reviewed.

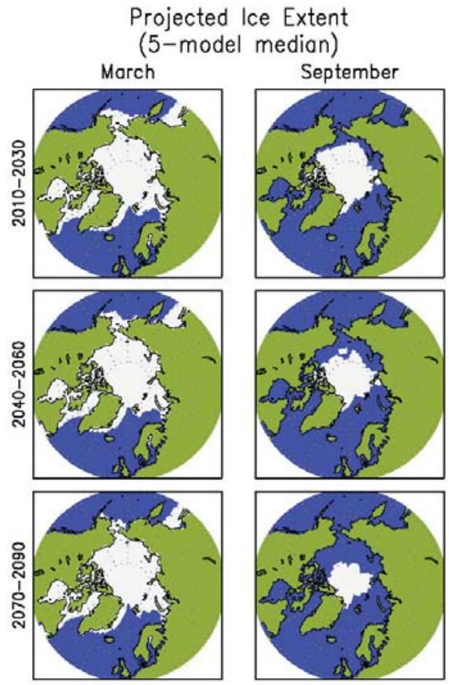


Figure 2: Arctic sea ice simulations for the 21st century. Not the projected ice-free coastal regions in September (Arctic Climate Impact Assessment).

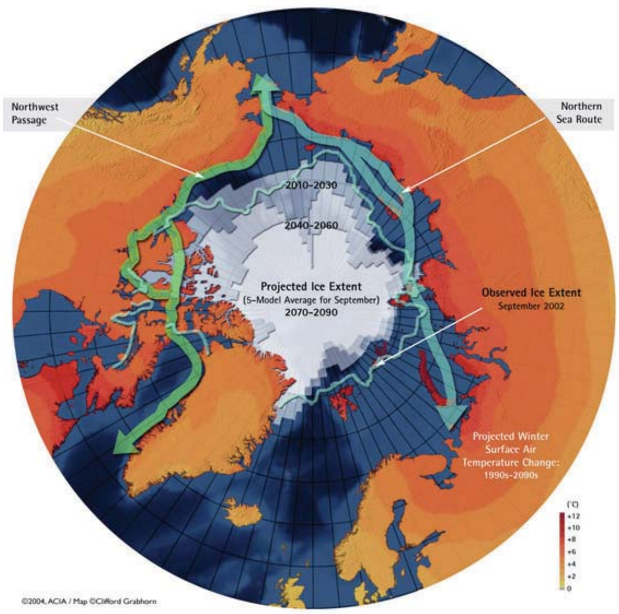


Figure 3: Changing Arctic sea ice and increasing marine access in the Arctic Ocean projected through the 21st century. This illustration represents ACIA finding #6 (Arctic Climate Impact Assessment).

## 3. RECENT SEA ICE TRENDS AND RESEARCH

Earlier observations from aircraft and ships, and nearly three decades of satellite observations, suggest that the September 2005 minimum sea ice extent (Figure 1) was the lowest since the early 1950's; Arctic sea ice extent at the end of the summer melt season has declined at a rate of 7.8% per decade for five decades (-9.1% per decade for 1979-2006) [2]. The Arctic sea ice cover is at a maximum extent in March and this maximum coverage has also been observed to decrease at approximately 2% per decade during the same period of observations [3]. These extent reductions have been observed in all seasons, but perhaps more significant have been observations of a rapid decline of thick, MY sea ice in the central Arctic Ocean. A study of satellite data for winter during 1978-1998 revealed that the MY ice cover had declined by 7% per decade [4]. A second trend analysis for 25 years of summer ice minimums (1978 to 2003) reports a decline of MY sea ice as high as 9.2% per decade [5]. One important result of these trends should be a decrease in the presence of MY ice in the Arctic's coastal seas where seasonal navigation is highest.

Arctic sea ice thicknesses have been much more difficult to monitor and evaluate during recent decades. Direct measurements of FY sea ice in the Arctic coastal seas, especially along the Russian Arctic, generally yield 1-2 m thicknesses. For the central Arctic Ocean, thicknesses of MY ice can be as high as 4-5 m. One pioneering study using sea ice draft data acquired on submarine cruises[data from 1958-1976 compared with cruise data for 1993-1996] indicated a decrease in thickness at the end of the melt season for the central Arctic Ocean from 3.1 m to 1.8 m [6]; this represented a volume decrease of 40% and a widespread decrease in sea ice draft. This 40% reduction was adjusted to 32% in a subsequent study once additional submarine tracks were added [7]. Other observations show regions of thinning and also regions with no recent changes. One key issue is that future sampling of Arctic sea ice thickness requires enhanced monitoring systems for more effective spatial and temporal measurements. Future Arctic navigation and all marine activity will depend on more frequent, reliable, near real-time, and improved sea ice thickness measurements.

As noted, one GCM simulation used in the ACIA indicated the possibility of an ice-free Arctic Ocean for a period in summer 2050. Recent analyses of GCM sea ice simulations using models for the Fourth Assessment of the Intergovernmental Panel on Climate Change [IPCC AR4] (applying global warming scenarios) show near-complete loss of Arctic sea ice in September for 2040 to beyond 2100 [8]. However, research also indicates abrupt reductions in sea ice coverage during the 21st century are a common feature in many of the GCM sea ice simulations [9]. Whether these periods of accelerated summer ice retreat might provide 'windows of opportunity' for improved marine navigation is unknown.

Continued research on the performance of the sea ice simulations of the IPCC AR4 models reveals that none of the GCMs have negative trends as large as the observed sea ice coverage trend for the period 1953-2006 (7.8% per decade reduction ; the observed trend is three times larger than the multi-model mean of -2.5% per decade loss [2]. This is an extraordinary development that also means the current summer sea ice minima are as much as 30 years ahead of the mean of the model simulations [2]. With continued green house gas emissions, it is highly plausible that the Arctic Ocean could become completely ice-free for a short summer period earlier than 2040. Just as important to ship navigation, these simulations show large areas of the coastal Arctic seas to be ice-free for longer periods in the spring and autumn months. Arctic marine access continues to increase in nearly all the scenarios posed by these global warming assessments.

## 4. REGIONAL TRENDS

### 4.1 CANADIAN ARCTIC & NORTHWEST PASSAGE

The observed record of minimum sea ice extent (coverage) for the eastern and western regions of the Canadian Arctic is illustrated in Figure 4. Although the observations for both regions show negative trends for the period 1969-2003, the year-to-year variability in coverage is quite extreme. Both regions also exhibit large differences for a given year; for example, in 1991 the western Canadian Arctic showed a one of the highest largest ice coverage areas, while in the eastern region a more normal coverage area at the summer minimum was observed [1]. While these observations indicate an overall decrease in the ice cover of the waterways that comprise the Northwest Passage, the two key variabilities ~ year-to-year and spatial ~ create challenges for planners judging risk and the reliability of an Arctic marine transportation system.

The five models used in ACIA revealed that the last regions of the Arctic Ocean with sea ice coverage in summer would be in the northern waterways of the Canadian Archipelago and along the northern coast of Greenland [1]. The flow of more mobile MY ice through the northern passages of the Canadian Arctic presents another challenge to marine operations. Enhanced satellite monitoring of this complex region will be a necessity if year-round marine activities are to be realized.

### 4.2 RUSSIAN ARCTIC & NORTHERN SEA ROUTE

Figure 1 indicates that a nearly ice-free passage could have been made from Kara Gate through to the Bering

Strait along the length of the Northern Sea Route. Passive microwave satellite observations of sea ice in the Russian Arctic seas from 1979 to the present show large reductions in sea ice extent in summer and reductions in winter extent in the Barents Sea. All of the ACIA model simulations and more recent IPCC AR4 model simulations confirm that large summer ice edge retreats should occur in the Laptev, East Siberian and western Chukchi seas. With a continued shrinkage of the MY fraction of sea ice in the central Arctic Ocean, fewer MY ice floes should be observed along the navigable eastern passages of the Northern Sea Route [1].

Long-term fast ice thickness measurements of the four Russian marginal seas (Kara, Laptev, East Siberian and Chukchi seas) have been analyzed to detect significant trends. However, the five, 65-year observational records indicate that long-term trends are small and inconclusive: the trends are small (approximately 1 cm per decade); the trends for the Kara and Chukchi seas are positive and the trends for the Laptev and East Siberian seas negative [10]

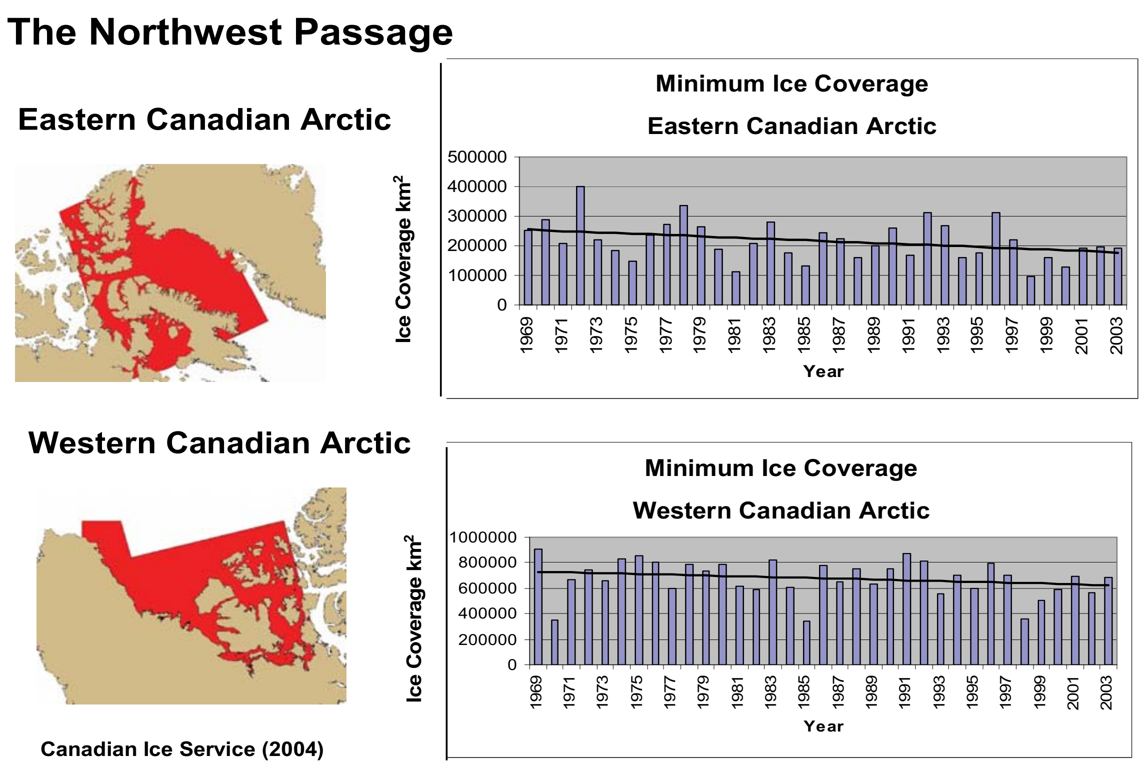


Figure 4: Sea ice variability in the Canadian Arctic and Northwest Passage (Arctic Climate Impact Assessment).

## 5. CONCLUSIONS

A review of recent assessments, observations, and studies indicate there remains much to understand about the present and future trends in Arctic sea ice. The operating conditions for Arctic ships will remain challenging, particularly in winter. There is also a high probability that Arctic sea ice will be more mobile and dynamic in the future. Several key conclusions from this review are:

1 Arctic sea ice has been observed to be diminishing in extent and thinning for five decades.

2 All GCM model simulations (based on a range of global emission scenarios) indicate a continuing retreat of Arctic sea ice through the 21st century.

3 Recent GCM simulations of sea ice cannot replicate the observed sea ice reductions for the period 1953-2006; the observed (negative) trend is three times *larger* than the multi-model mean.

4 One GCM simulation of sea ice in the Arctic Climate Impact Assessment indicated the possibility of an ice-free Arctic Ocean (for a short period in summer) by mid-century; more recent model simulations show that this Arctic 'ice-free state' could be reached by 2040, if not sooner.

5 Even a brief ice-free period in summer for the Arctic Ocean would mean the disappearance of MY ice in

the central Arctic Ocean; such an occurrence would have significant implications for design, construction and operational standards of all future Arctic marine activities.

6 Observed sea ice trends and GCM simulations show coastal Arctic regions to be increasingly ice-free, or nearly ice-free, for longer summer and autumn seasons; longer open water seasons increase the potential for greater coastal erosion which can impact support infrastructure for Arctic development and transportation.

7 The resolution of all GCMs that are applied to the Arctic region is generally too coarse for adequate coverage of complex geographies such as the narrow straits and waterways of the Canadian Arctic.

8 The observed record of sea ice extent in the Canadian Arctic displays very high inter-annual variability; such year-to-year variability, also observed in the Russian Arctic, is a serious challenge to risk and reliability of Arctic marine transportation systems.

9 Despite a small decrease in the maximum Arctic sea ice extent in March (seen in the observed record and model simulations), the Arctic sea ice cover will remain in winter and will continue to present unique challenges for all Arctic marine uses including commercial shipping.

10 It is highly plausible that Arctic sea ice will be more mobile, particularly in spring, summer and autumn, as the cover continues to retreat from Arctic coastlines; coastal seas may experience increased ridging of seasonal sea ice potentially creating more difficult conditions for marine navigation.

11 A key requirement is continued development of high resolution, regional sea ice models that can provide more robust and realistic forecasting of operating conditions.

12 The current GCM sea ice simulations are not yet robust enough to provide detailed information on future operating conditions such as the length of the navigation season and 'residence time' of ice-free regions that would allow faster ship transits.

13 There is a critical requirement for more real-time sea ice observations, especially ice thickness measurements, to support all future Arctic marine uses; the national ice centers are critical providers of such sea ice information and greater international collaboration among the centers will enhance the development of more integrated products; new satellite sensors hold the promise of providing greater, near real-time ice thickness information for Arctic ships that are underway on future voyages.

14 Although the future sea ice operating conditions in the Arctic Ocean are uncertain, there *is* greater marine access throughout the Arctic Basin and it is highly plausible longer seasons for access and navigation will be the norm throughout the 21st century.

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# ABS Requirements for Vessels Operating in Low Temperature Environments

**Design and Construction of Vessels Operating in Low Temperature Environments International Conference**

**30 – 31 May 2007**

**K Lilley, R M Conachey, C Baker, G Wang, and R Miguez**

**American Bureau of Shipping**

ABS

# Overview

- Challenges and demands for Arctic vessels
- Ice Class selection
- ABS LTE Guide
  - Material
  - Hull design
  - Arrangement
  - Systems
  - Human exposure

# Demand and Challenges

- Demands: ice-going vessels need to be
  - Powerful
  - Ice-strengthened
  - Winterized and suitable for cold weather operation
  - Operated by trained/experienced crew
- Environmental challenges:
  - Low temperature
  - Ice covered seas (some routes)
  - Extended days (summer) and nights (winter)
- Technical challenges
  - Limited design guidance
  - Limited operation experiences

# Ice Class Selection – ABS

- Selection of Ice Class dependent on:
  - Expected ice cover
  - Age of ice (e.g. First-year ice, multi-year ice, etc.)
  - Expected ice thickness
- Determine the Ice designator
  - Severe
  - Medium
  - Light
  - Very Light
- Select appropriate Ice Class

# ABS General Ice Class Selection

| *Ice class (ABS)* | *Navigating independently* | *Year around navigation in water with first-year ice* |
|---|---|---|
| A0 | Independently | Severe |
| B0 | Independently | Medium |
| C0 | Independently | Light |
| D0 | Independently | Very Light |

From 6-1-1/Table 1 of Steel Vessel Rules

# IACS Polar Ice Class

- Adopted Unified Requirements in August 2006
  - I1 – Polar Class Descriptions and Application
  - I2 – Structural Requirements for Polar Ice Class Ships
  - I3 – Machinery Requirements for Polar Ice Class Ships
- Intended for navigation in ice-infested polar waters
- Do not address "Icebreaker" notation
  - Icebreaker "refers to any ship having an operational profile that includes escort or ice management functions, having powering and dimensions that allow it to undertake aggressive operations in ice-covered waters,….."
  - Icebreakers are subject to additional requirements/special consideration

# Polar Class Descriptions

| Polar Class | Summer/ Fall ops | Year-round ops | Ice Description |
|---|---|---|---|
| PC 1 | | ✔ | all Polar waters |
| PC 2 | | ✔ | moderate multi-year ice conditions |
| PC 3 | | ✔ | second-year ice which may include multi-year ice inclusions |
| PC 4 | | ✔ | thick first-year ice which may include old ice inclusions |
| PC 5 | | ✔ | medium first-year ice which may include old ice inclusions |
| PC 6 | ✔ | | in medium first-year ice which may include old ice inclusions |
| PC 7 | ✔ | | thin first-year ice which may include old ice inclusions |

# Baltic Ice Class

- Finnish-Swedish Ice Class Rules 1985, as amended
  - Vessels trading in the Northern Baltic in winter
- Provide icebreaker assistance to vessels bound for Finnish-Swedish ports
  - Ice conditions determine restrictions that may apply to vessels depending on size and ice class


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# Equivalencies Between Classification Societies

## From HELCOM MARITIME 2/2004

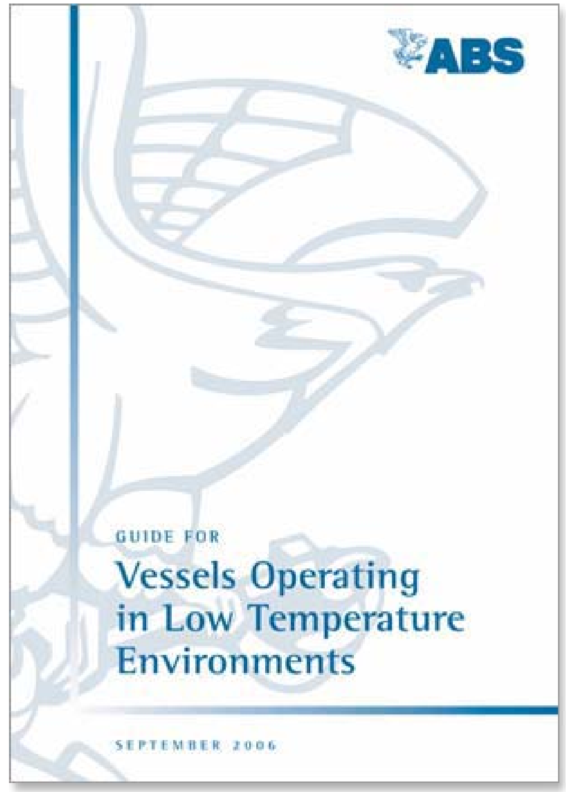
| Classification Society | Ice Class | | | | |
|---|---|---|---|---|---|
| Finnish Swedish | IA Super | IA | IB | IC | Category II |
| RMRS (1995) | UL | L1 | L2 | L3 | L4 |
| RMRS (1999) | LU5 | LU4 | LU3 | LU2 | LU1 |
| ABS | IAA | IA | IB | IC | D0 |
| DNV | ICE-1A* | ICE-1A | ICE-1B | ICE-1C | ICE-C |
| LR | 1AS | 1A | 1B | 1C | 1D |
| CASPPR (1972) | A | A | C | D | E |

# ABS LTE Guide

- Guide for Vessels Operating in Low Temperature Environments (LTE)
- Address winterization issues not covered by ice class
- Developed with:
  - Internal personnel with Baltic/polar experience
  - Input from ice experts/shipbuilders
- Published September 2006

# ABS LTE Guide

- Two documents in one
  - Guide –requirements
  - Guidance Notes – As appendices with additional explanations
- Supplementary information
  - Weather conditions
  - Additional reference materials
  - Administration listings
  - Meteorological organization listings



# ABS LTE Guide – Organization

- Section 2 – Materials, Welding and Coatings
- Section 3 – Hull Construction and Equipment
- Section 4 – Vessel Systems and Machinery
- Section 5 – Safety Systems
- Section 6 – Specific Vessel Requirements
- Section 7 – Survey Requirements
- Section 8 – Crew Considerations
- Section 9 – Training and Related Documentation

HARDWARE

SOFTWARE

# General Section – Notations


- CCO-HR(TEMP)
  - Sections 2 – 6
- CCO-HR(TEMP)+
  - Sections 2 – 6, 8 – 9
- HR – 18 or 36 hours for emergency services
  - Other hours > 18 can be considered
- TEMP – Design Service Temperature (DST)
- Examples: CCO-18(-30°C), CCO-36(-40°C)+

# General Section

- Application
  - Any vessel/marine structure operating in cold area
  - Exposed hull structure materials must be suitable
- Objective
  - Improve vessel/system performance
  - Design service temperatures ≤ $-10^{o}$C

# Design Service Temperature

- Per IACS UR S6.3
- At Least 20 Year Observation Period
- MDHT–Mean Daily High Temperature
- MDAT–Mean Daily Average Temp.
- MDLT–Mean Daily Low Temp.

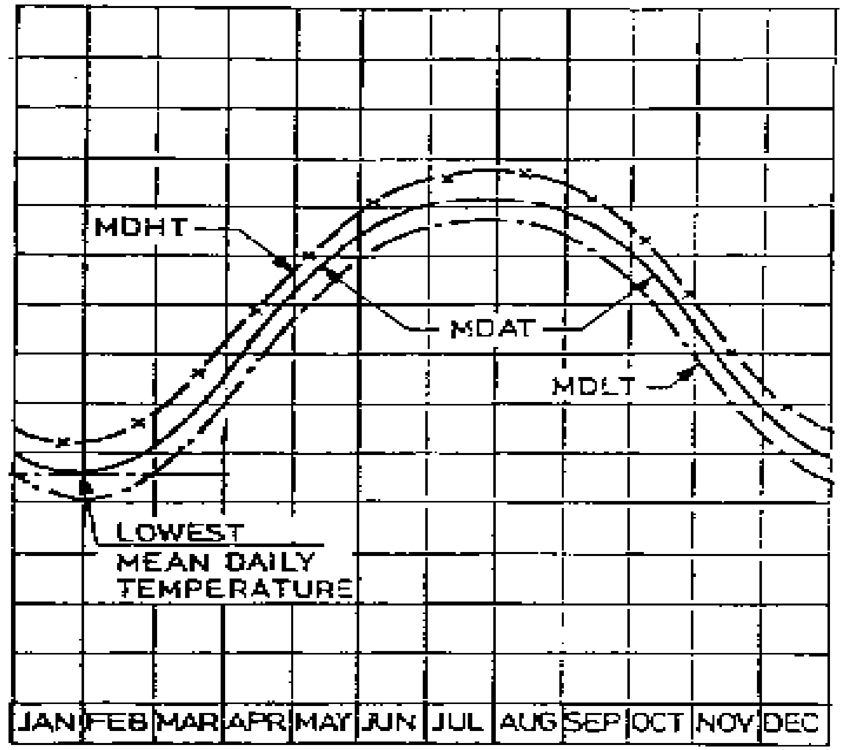



Fig. 1
Commonly used definitions of temperatures.

# Materials Section – Intent

- Provide requirements with references to appropriate paragraphs in Part 2 of SVR
- Not all vessels operating in LTE are Ice Class
- Non-Ice Class Vessels to meet Section 2
- Requirements of other Ice Class Rules may be applied
- No special welding requirements at this time
- Coatings – alert users concerning durability and abrasion resistantance

# Materials Section

- Selection of material grades based on:
  - Structural location of member/stress level
  - Design service temperature (DST)
  - Material thickness
- Selection of hull materials are based on
  - DST
  - Hull location – stress level
- Selection of materials for deck equipment, outfittings
  - Minimum anticipated temperature (DST – 20°C)

# Hull Construction and Equipment

- Prevent freezing of tanks containing liquids
  - Turbulence inducing systems/heating coils
- Protection of the environment
  - POT – Protection of Oil Tanks notation

# Hull Construction and Equipment

- Protection of personnel working outside
  - Superstructures – enclosed bridge wings
  - Deckhouses – heated for performance of deck duties
- Reduce spray and ice accumulation on deck
  - Forecastles – deflect spray
- Alert designers to stability issues related to ice build-up
  - Consider ice accretion when calculating stability

# Vessel Systems and Machinery – Intent

- Address areas where mistakes have been made in past due to poor arrangements/design
- Improve reliability and maintain operation of rotating equipment
- Provide additional guidance for piping systems not addressed in ice class Rules
- Address issues related to fire safety
- Additional requirements for electrical systems
- Appendix 4 based on current design/operational experience

# Vessel Systems and Machinery

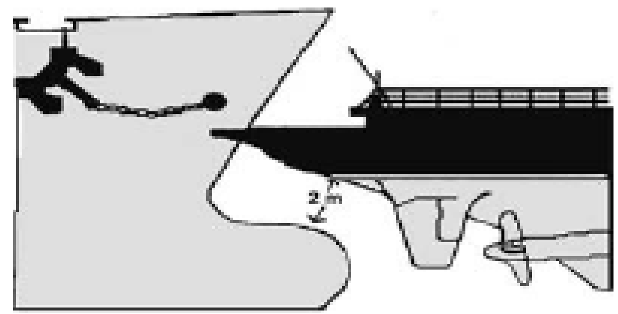
- Prime Movers
- Propulsion and Maneuvering Machinery
- Deck and Other Machinery
- Piping
- Fire Safety
- Electrical

# Vessel Systems and Machinery

- Prime Movers
  - Operation at low power outputs/low speeds
  - Low power operation considerations for auxiliary equipment
  - Thrust reversal for Ice Class vessels
- Propulsion and Maneuvering Machinery
  - Turbochargers
  - Lubricating oil systems
  - Propulsion shafting bearing lubrication

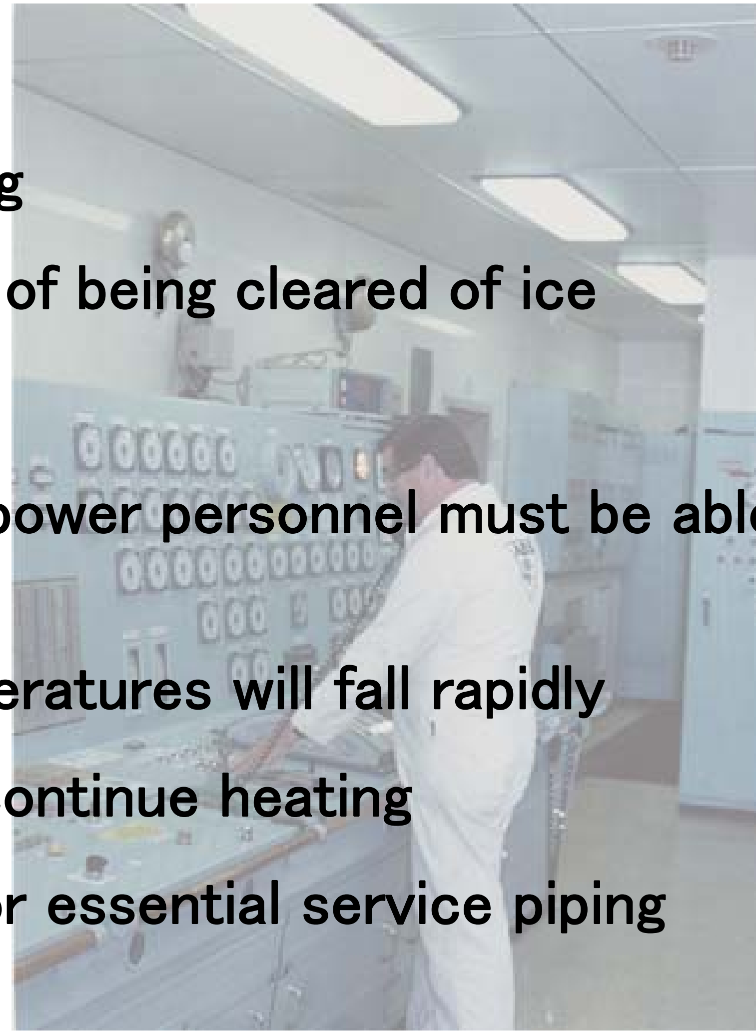
# Vessel Systems and Machinery

- Deck and Other Machinery
  - Anchoring arrangements for Ice Class vessels
  - Anchor windlass, towing winch
- Piping
  - Materials suitable for MAT
  - Piping arranged to drain fluids
  - Components designed to function – protection/heating
  - De-icing/heat tracing



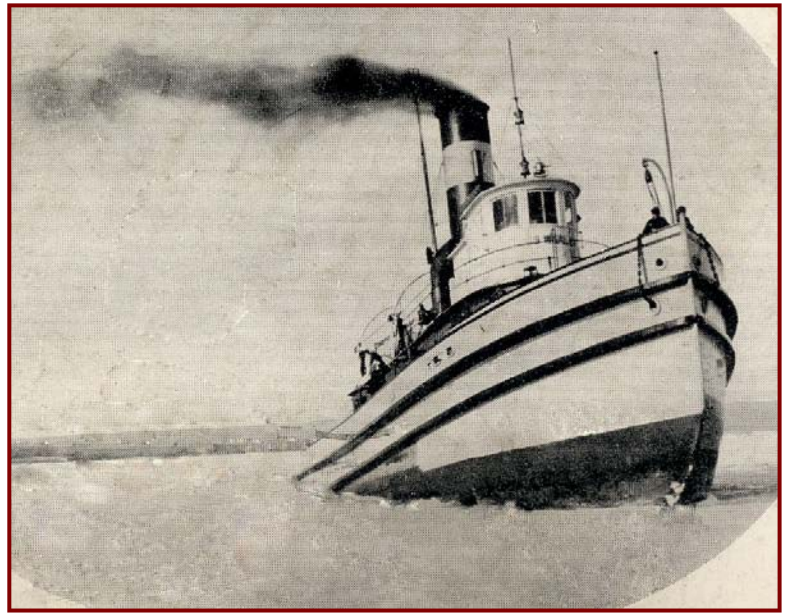
# Vessel Systems and Machinery

- Fire Safety
  - Protection from freezing
  - Pump suctions capable of being cleared of ice
- Electrical
  - In the event of loss of power personnel must be able to survive
  - Expected internal temperatures will fall rapidly
  - Boiler and controls to continue heating
  - Maintain heat tracing for essential service piping



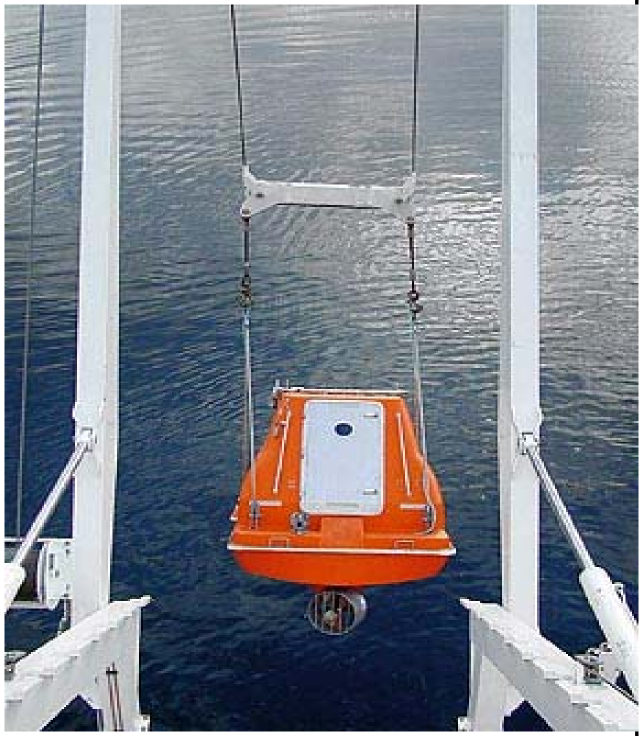
# Safety Systems – Intent

- Protection of personnel
- Survival of personnel until help arrives
- Life saving equipment for rescue or in event of abandonment
- Personnel training



# Safety Systems

- Life Saving Appliances
  - Life rafts / boats
  - Launching
  - Ice gangway, escape chutes
  - Immersion suits
  - Alarms and communications
- Heating for survival
- Navigational equipment



# Specific Vessel Requirements – Intent

- Select certain vessel types we believe will trade in Arctic
- Focused on
  - LNG
  - Tankers
  - Bulk Carriers
  - Offshore Supply Vessels
- Can add additional vessel types later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# Crew Considerations – Intent

- Crew remain effective in performing their duties
- Experience of committee members indicated vessel crew not provided with proper equipment
- Provide information human response to cold
- Provide requirements and guidance concerning cold weather equipment
- With some basic information – operators will take proper precautions

# Guidance Notes Crew Considerations

- Human performance and health hazards
- Guidance for design or selection of clothing
- Monitoring environmental conditions
- Nutrition
- Design of equipment for operation in cold conditions
- Accomodations/environ-mental control

Crewmembers of a BP Tanker seen clearing the "Frozen Spray" from the anchor windlass

# Tables/Charts – Performance

**WIND CHILL CHART**

| Wind km/h | Velocity mph | 4 | -1 | -7 | -12 | -18 | -23 | -29 | -34 | -40 |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Ambient Temperature (°C) | | | | | | | | |
| | | Equivalent Chill Temperature (°C) | | | | | | | | |
| Calm 0 | 0 | 4 | -1 | -7 | -12 | -18 | -23 | -29 | -34 | -40 |
| 8 | 5 | 3 | -3 | -9 | -14 | -21 | -26 | -32 | -38 | -44 |
| 16 | 10 | -2 | -9 | -16 | -23 | -30 | -35 | -43 | -50 | -57 |
| 24 | 15 | -6 | -13 | -20 | -28 | -36 | -43 | -50 | -58 | -65 |
| 32 | 20 | -8 | -16 | -23 | -32 | -39 | -47 | -55 | -63 | -71 |
| 40 | 25 | -9 | -18 | -26 | -34 | -42 | -51 | -59 | -67 | -76 |
| 48 | 30 | -16 | -19 | -22 | -36 | -44 | -53 | -62 | -70 | -78 |
| 56 | 35 | -11 | -20 | -29 | -37 | -46 | -55 | -63 | -72 | -81 |
| 64 | 40 | -12 | -21 | -29 | -38 | -47 | -56 | -65 | -73 | -82 |

Source: Threshold Limit Values (TLV™) and Biological Exposure Indeces (BEI™) booklet; published by ACGIH, Cincinnati, Ohio

**Little danger** in less than one hour exposure of dry skin

**DANGER** – Exposed flesh freezes within one minute

**GREAT DANGER** – Flesh may freeze within 30 seconds

**Suggested Maximal Allowable Work Times**

| Equivalent Temperature | Consequence - Action |
|---|---|
| Below -30 °C | No outdoor work performed unless deemed critical from a safety or operational perspective |
| Below -21 °C | Available outdoor working time is below 50% of working hour. |
| Below -12 °C | Available outdoor working time is below 75% of working hour. |
| Below -6 °C | Available outdoor working time is below 90% of working hour. |
| Above -6 °C | Normally 100 % Available working time |

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# Training/Documentation – Intent

- Vessels operating in LTE exposed to unique conditions:
  - Weather is poor
  - Navigation charts unreliable
  - Local ice conditions vary
  - Remote areas
  - Rescue difficult
  - Environmental clean-up is difficult
- Additional crew training necessary to address issues

# Summary

- ABS teaming with industry to maintain excellent record of safety
  - Joint industry projects
  - Joint development projects
- Publication of New Rules and Guides
  - Guide for Vessels Operating in Low Temperature Environment
  - Polar Ice Class Rules (based on IACS) Part 6 – SVR
  - Guidance Notes on Ice Class

Transportation Safety Board of Canada — Bureau de la sécurité des transports du Canada

TSB

# Brittle Fracture in Ships

Transportation Safety Board of Canada
Bureau de la sécurité des transports du Canada

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Canada

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# *Acknowledgements*


- **Lance Bedlington**
- **Mike Mathieu**
- **Bernard Breton**
- **Tim Lang**
- **Tony Gasbarro**




Canada

# Outline

- Lake Carling Hull Fracture – Genesis and Contributing Factors
- The Safety Deficiency – low (and unknown) toughness of vessel side shells
- Literature & Standards Review
- Residual Risks
- Conclusions

# From 10 cm to 6 meters

On 19 March 2002, brittle fracture was sustained notwithstanding correct loading at Seven Islands and benign seas.

- 1.5m to 2.5m amplitude
- 56m wavelength

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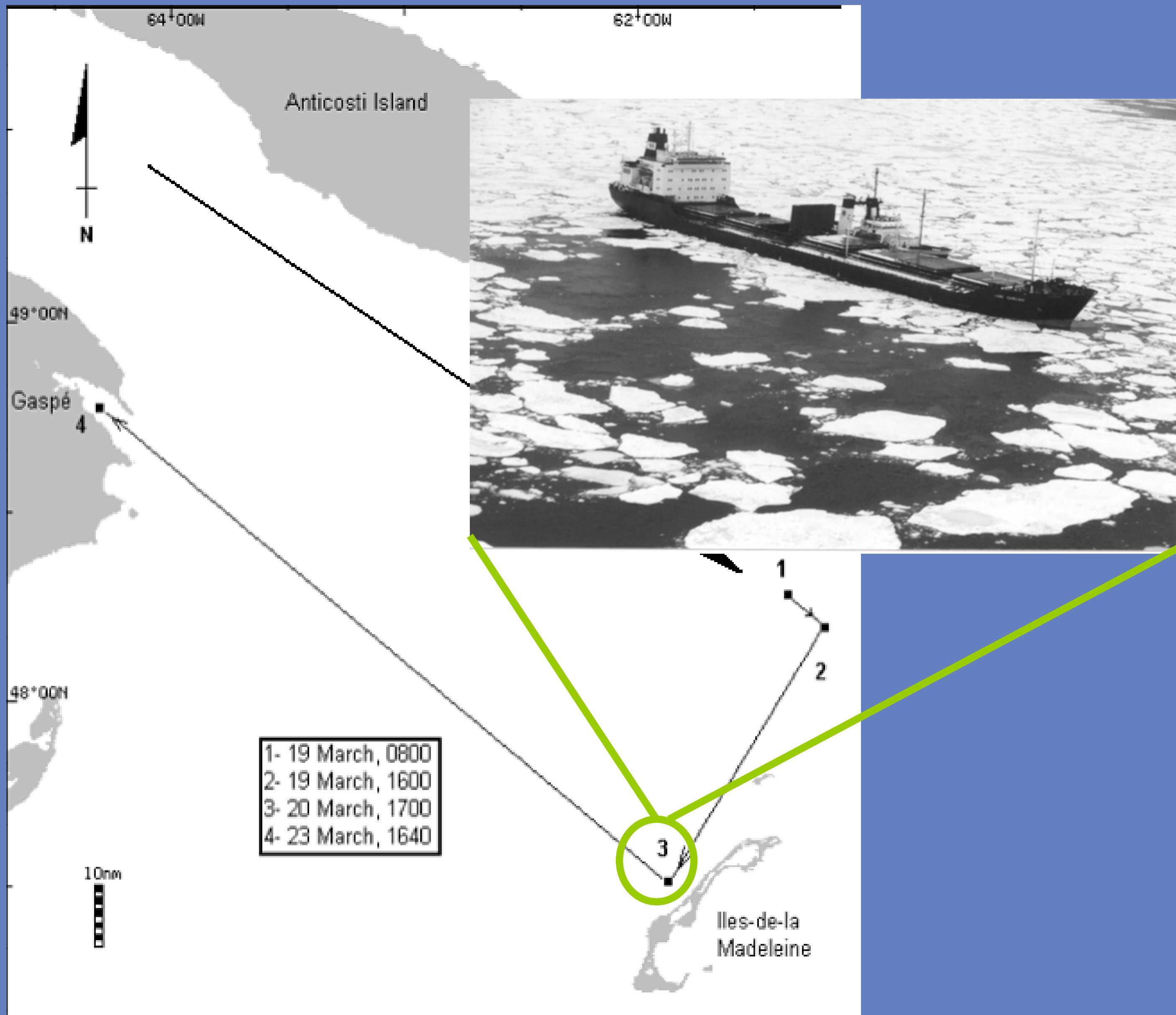




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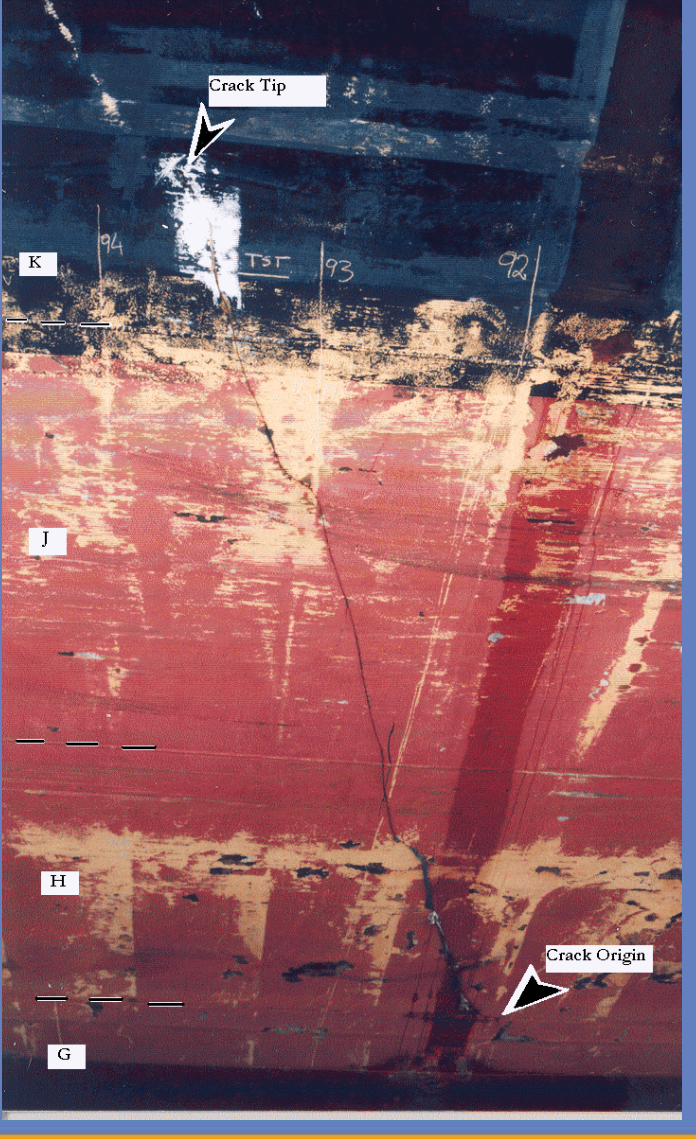
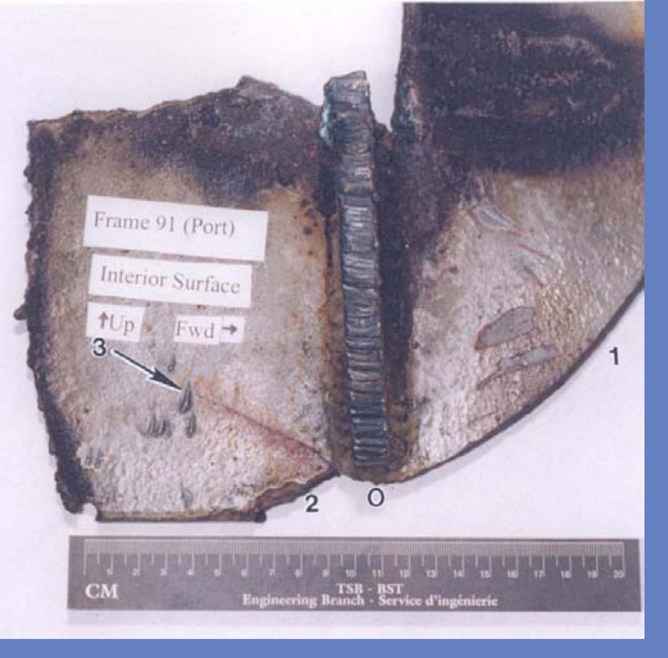
# Principal Fracture



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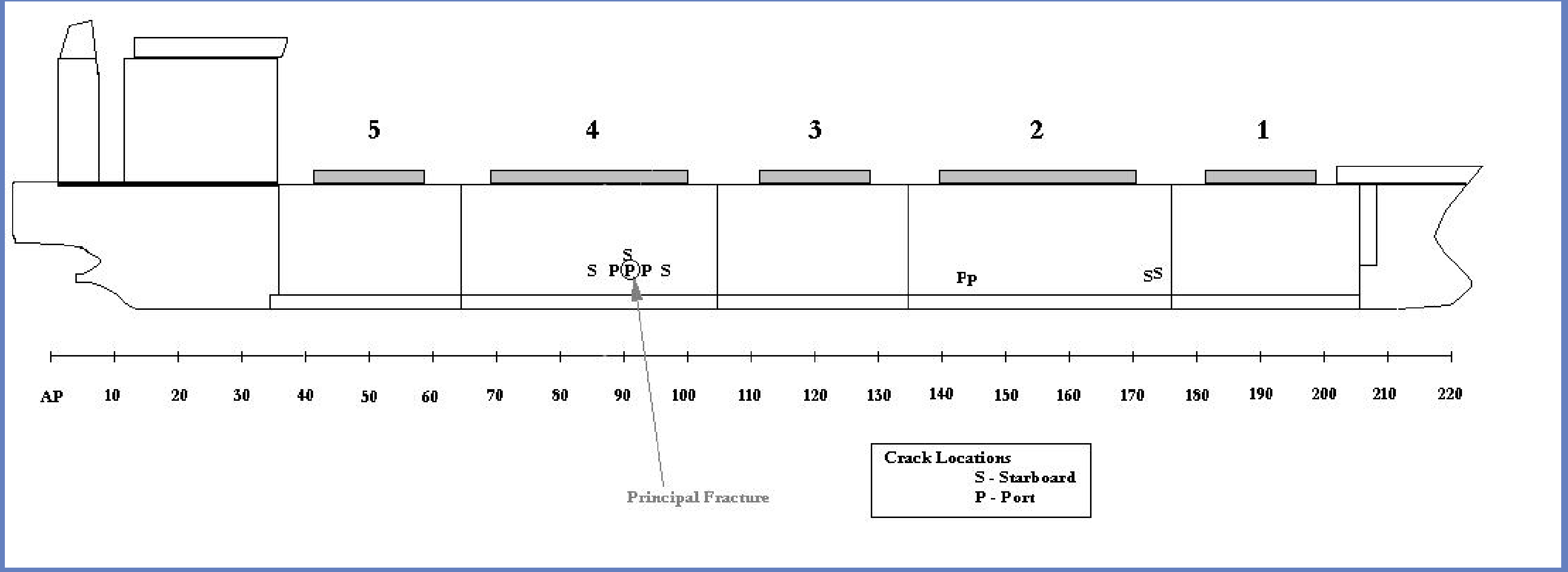
# Pre-existing Cracks

- March 2001 Drydock at Gdansk – no cracks
- November 2001 Vessel experienced greater than approved SWBM



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# Pre-existing Crack Details

**Frame Nº 89 port**

**Frame Nº 93 port**



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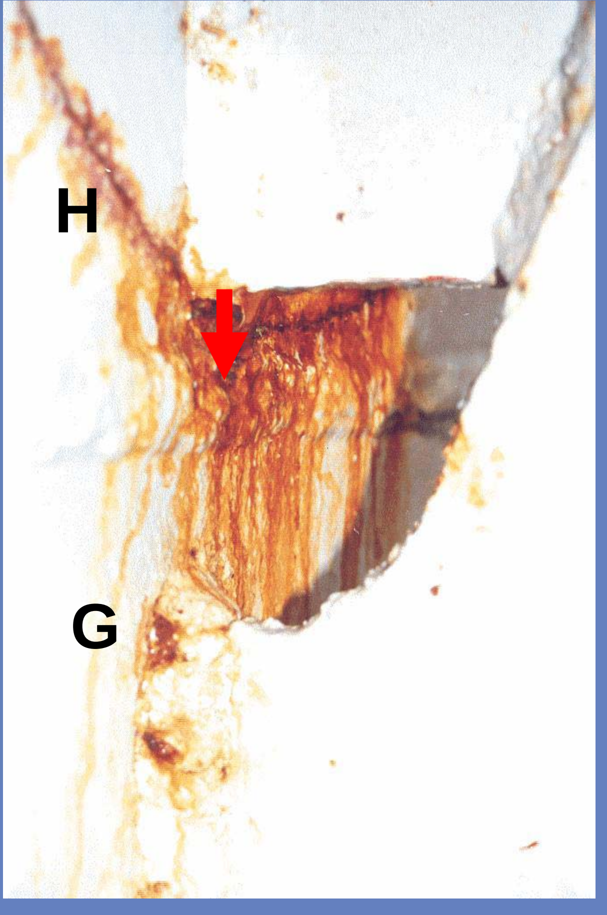
# Some Contributing Factors




- Lake Carling – 25 mm
- Lake Charles – 100mm
- Lake Champlain – 90mm
- H strake; 19 mm
- G strake; 15 mm

# Brittle Fracture on Ships… 3 Causes

- Abnormal forces in or on the ship structure
- Presence of flaws or notches
- Inadequate physical properties of the steel at service temperatures

# IACS Steel Toughness Standards

| grade | CVN (J) |
|---|---|
| A<br>Less than 50mm | -- |
| B<br>Less than 25mm | -- |
| D | 27 |
| E | 27 |

???

-20 C

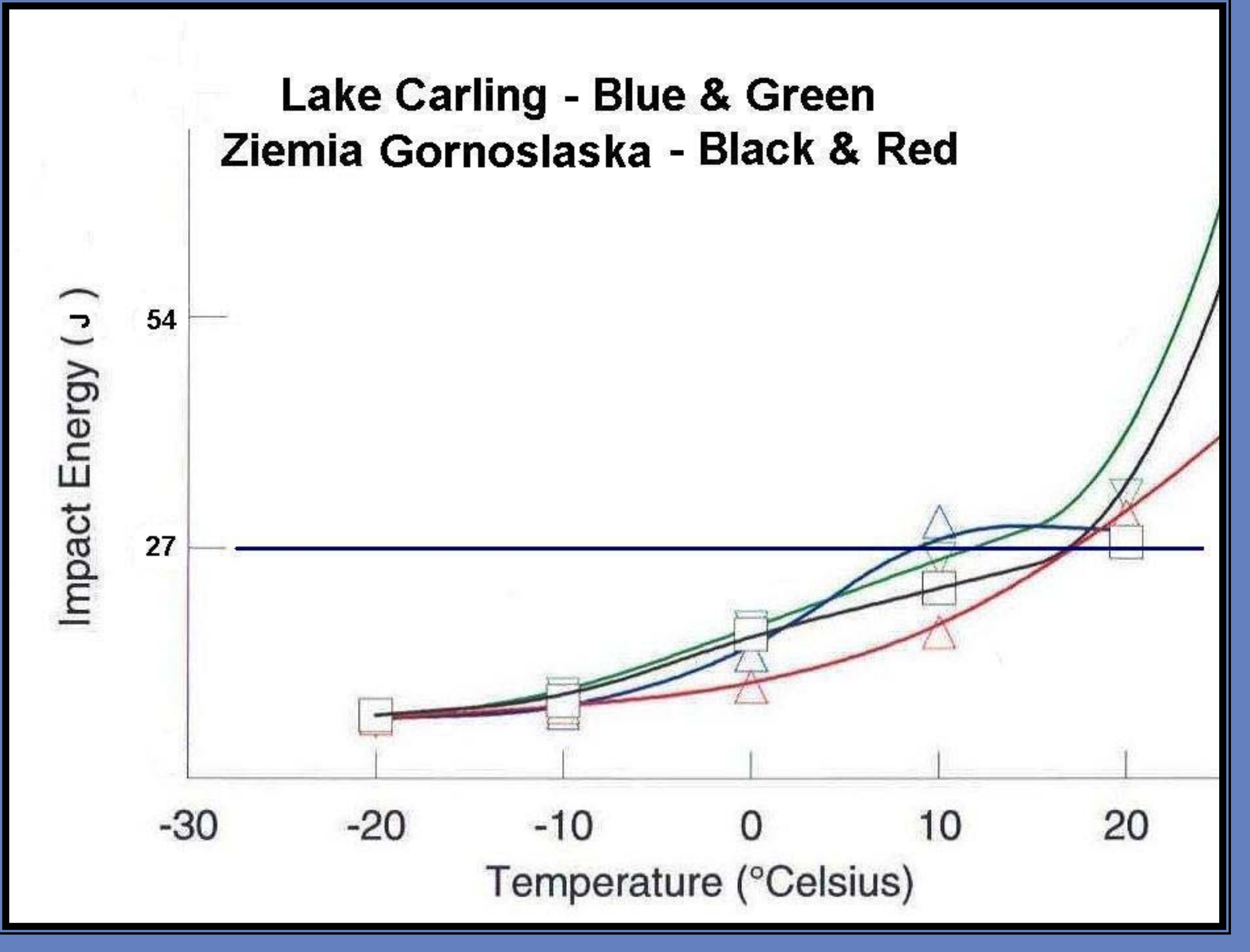
-40 C

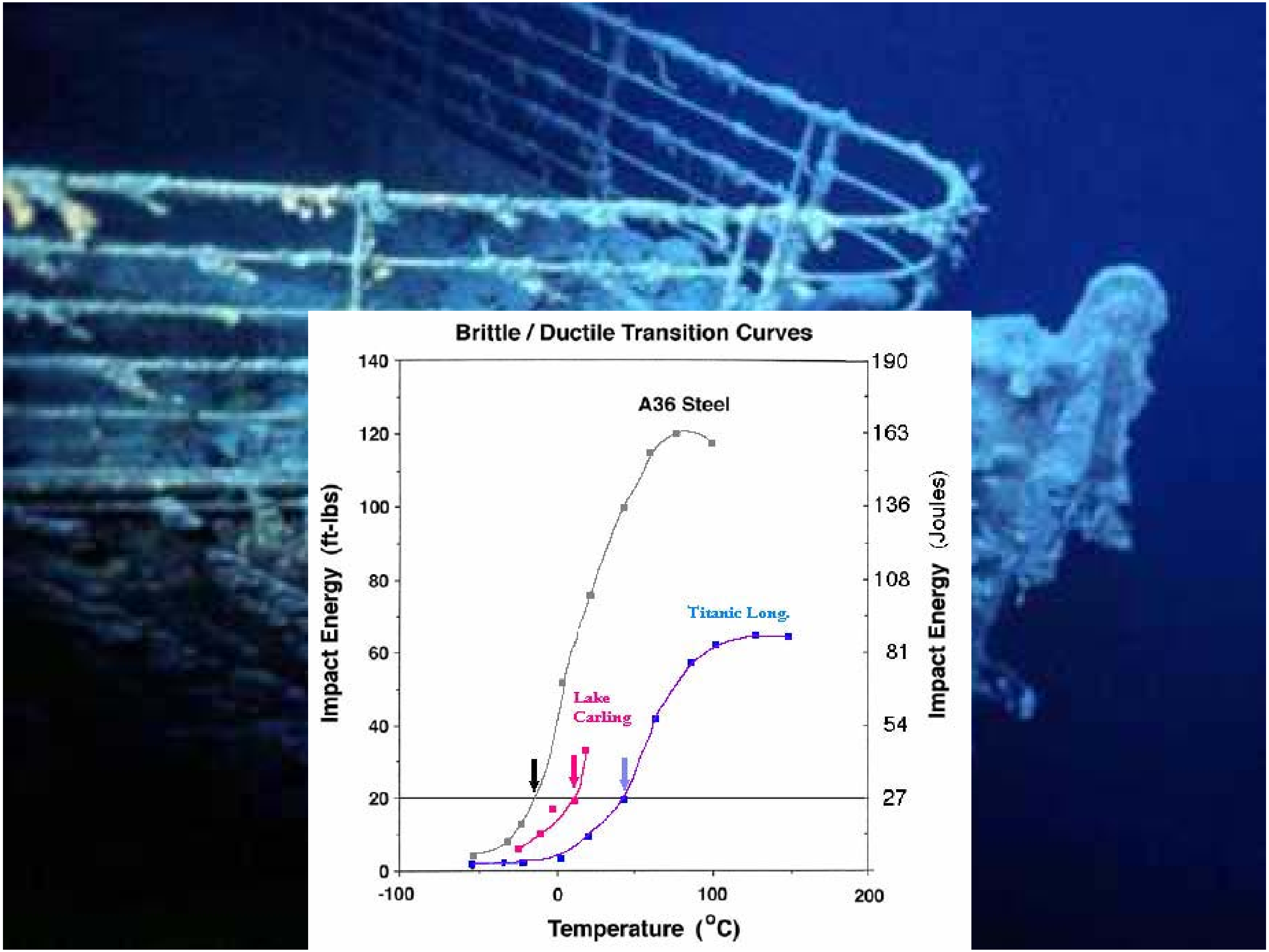
# Charpy – Lake Carling

| Temp (°C) | CVN (J) |
|---|---|
| +20 | 33 |
| **+10** | **26** |
| **0** | **18** |
| **-10** | **10** |
| -20 | 7 |

TSB Lab Report LP022/2002

# CVN Energy
# Lake Carling and Sister Ship




Brittle / Ductile Transition Curves
A36 Steel
Lake
Carling
Titanic Long.
Impact Energy (ft-lbs)
Impact Energy (Joules)
Temperature (°C)
140
120
100
80
60
40
20
0
190
163
136
108
81
54
27
-100
0
100
200

# CTOD – Lake Carling

- 0.07 mm @ -19°C
- 0.25 mm @ 0°C

Lloyd's Register tests 2004

# Fracture Appearance Transition Temperature (FATT)

- In many industries….brittle fracture is controlled by ensuring the 50% FATT is less than the operating temperature of the unit.
- For the steel used in ship's sides, FATT is not considered.

# Literature Overview

# Literature Review of the Fracture Properties of Grade A Ship Plate - OTH 95 489

**"... the <u>crack arrest</u> ability of Grade A plate is <u>poor and probably inadequate</u> for most ship applications. In the event of a fracture initiating, this factor will greatly contribute to the overall risk of subsequent vessel failure."**

A C Bannister / Investigator, SE Webster / Manager Engineering Metallurgy Department, DJ Price / Research Manager
General Steel Products

# Other Recognized Experts

- Dr. James Matthews (DREA- ret.)
  - "Grade E steel or better …" (27J @ -40°C)
- J.D.G. Sumpter (DRA-UK)
  - "FATT to be less than 0°C"
- D. Faulkner (Emeritus Professor of Naval Architecture, University of Glasgow)
  - "Grade A steel not to be used for ship's hulls"

# Lloyd's Tests
# Grade A Steel

- 1997, 39 samples evaluated
- CVN minimum of 49 J
- CVN (average) of 134 J
- 5 samples (~13%) with FATT greater than 0°C
- 8 other samples (20%) with FATT greater than -10°C

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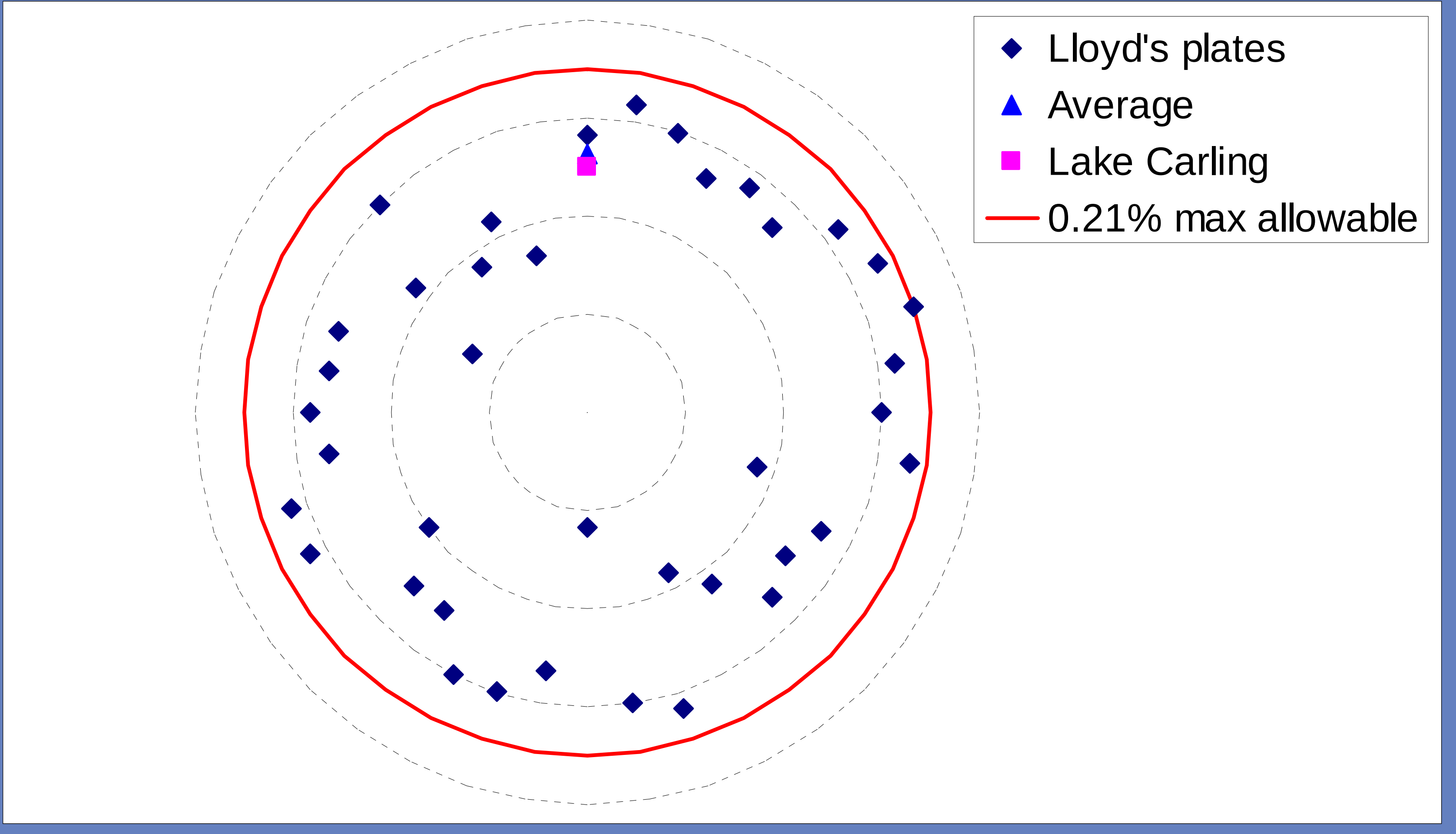
# Carbon Content



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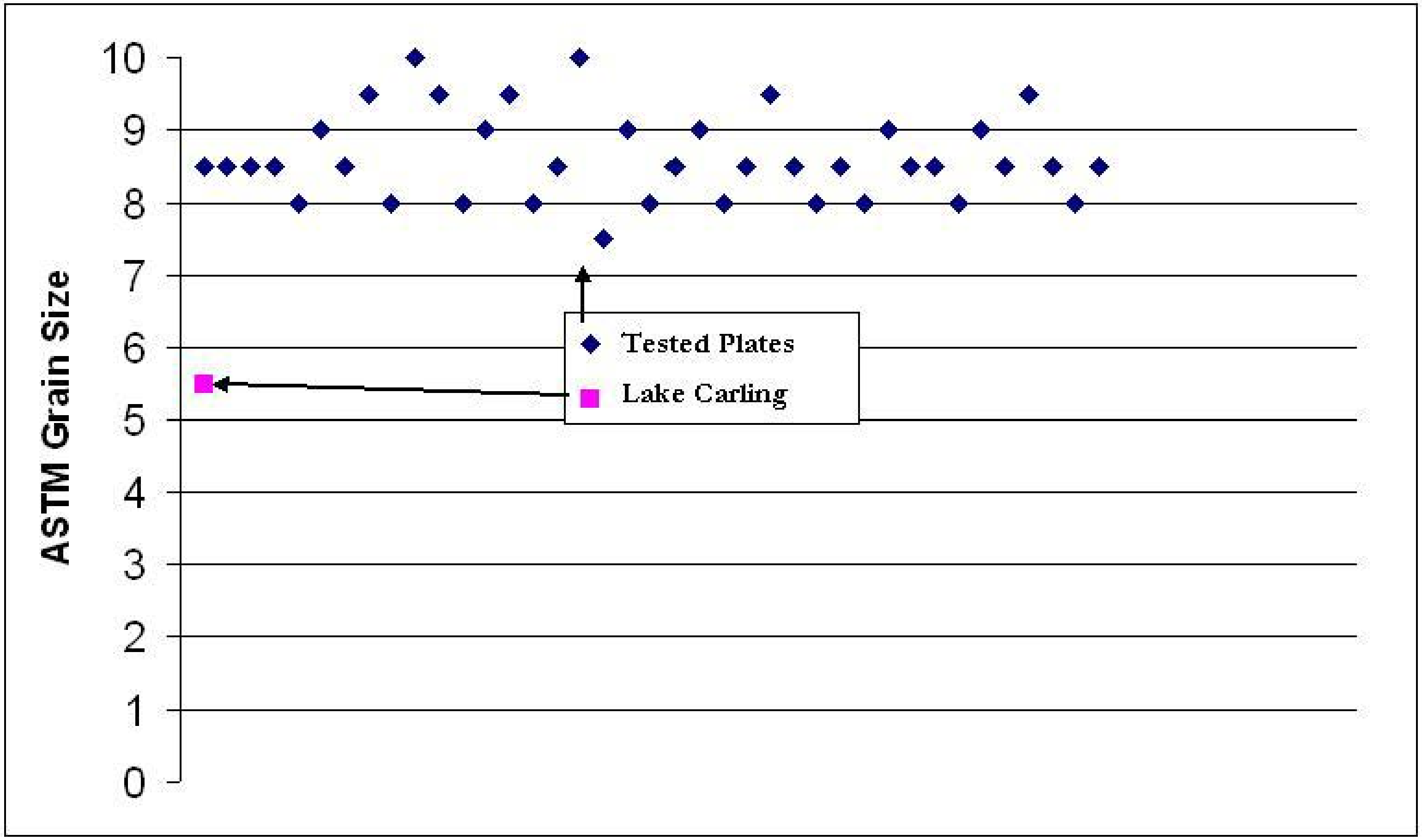
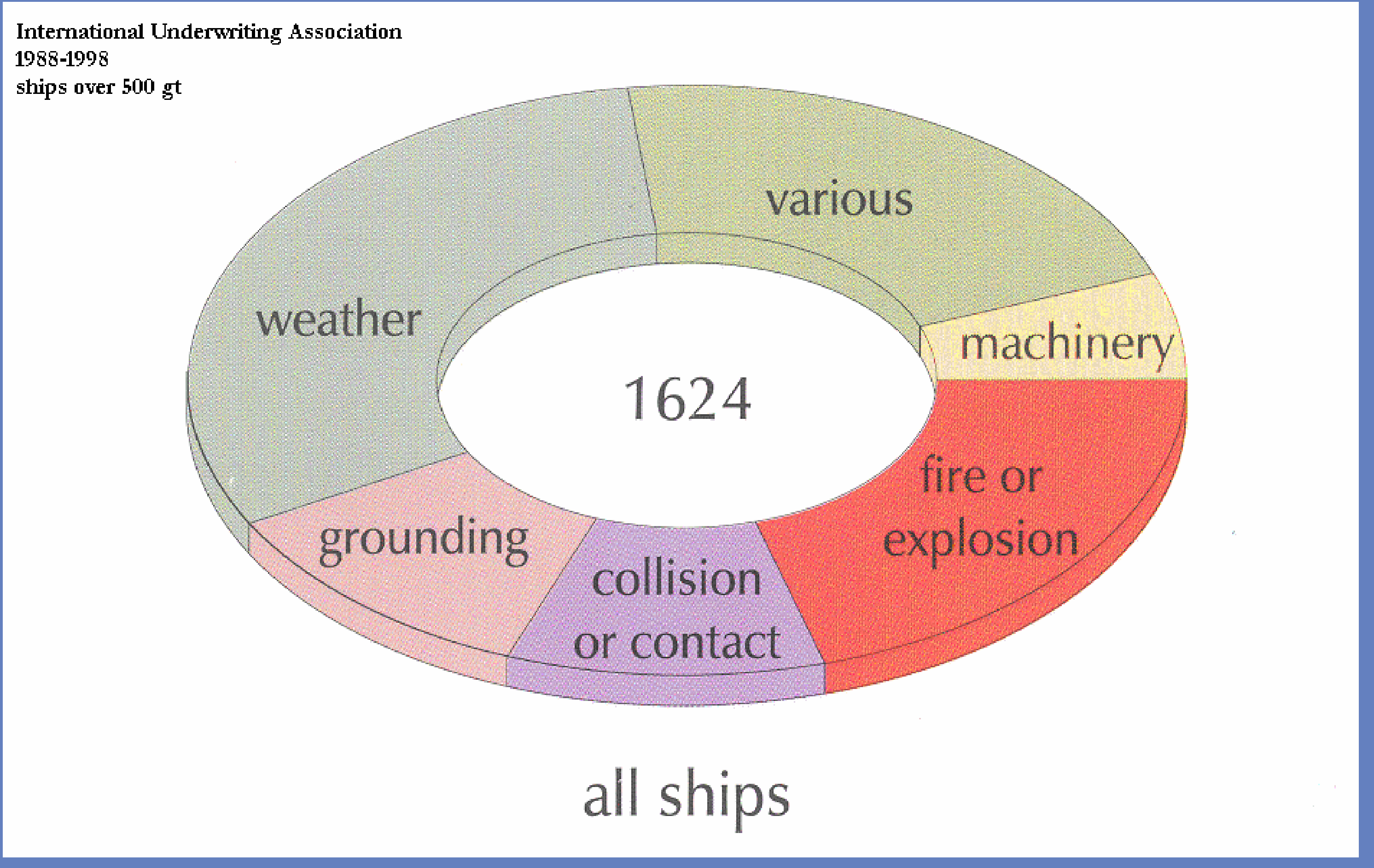
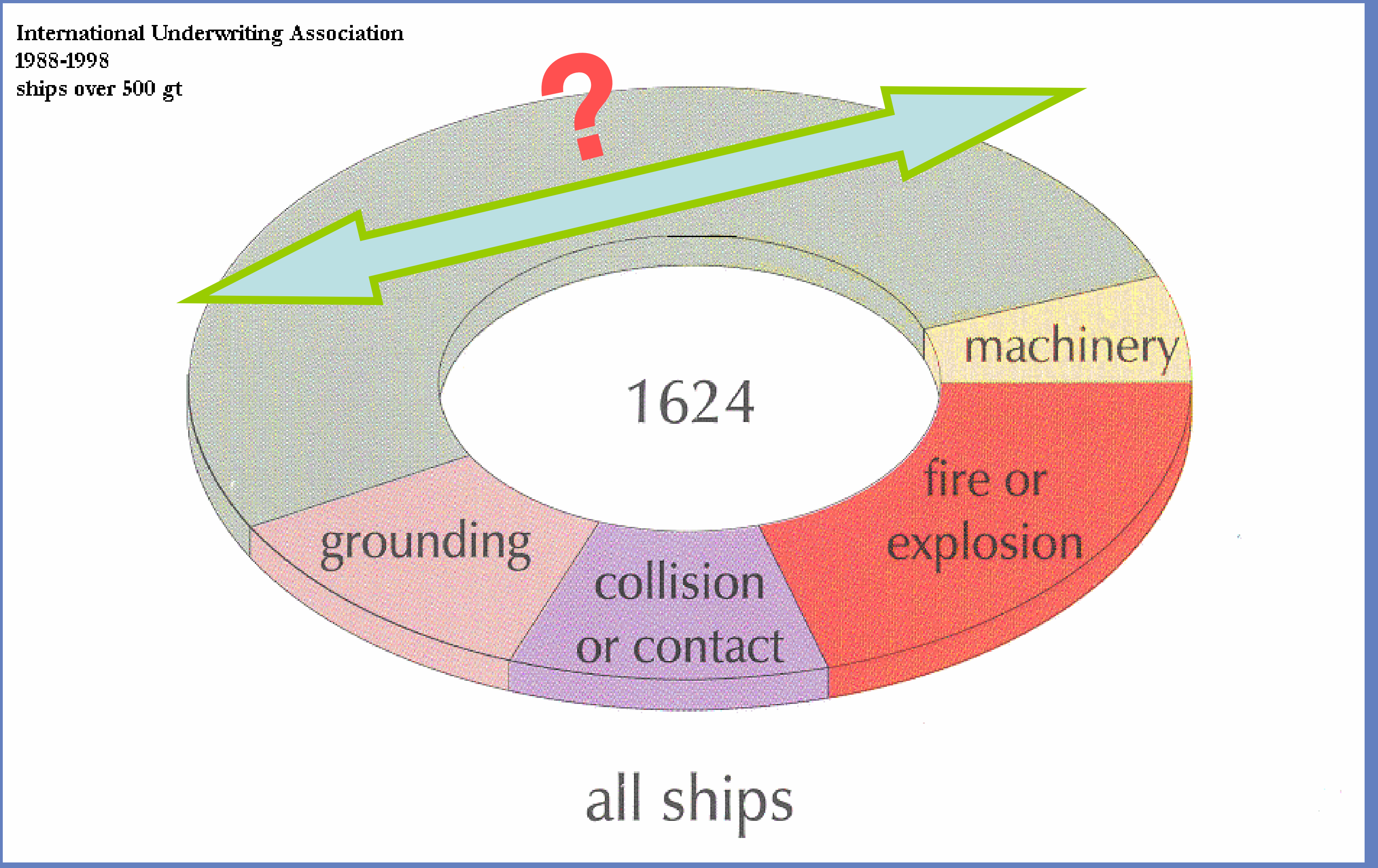
# Grain Size



# Statistics


International Underwriting Association
1988-1998
ships over 500 gt
various
machinery
weather
1624
fire or
explosion
grounding
collision
or contact
all ships


International Underwriting Association
1988-1998
ships over 500 gt
?
1624
machinery
fire or explosion
collision or contact
grounding
all ships

# Misclassified !!
# (ie: Weather & Various)

## **Dodsland** (1987)

- **Seaweb**…. SUSTAINED HEAVY WEATHER DAMAGE IN NORTH ATLANTIC OCEAN ON OR BEFORE 17/2/87. CONTINUED ON TO HALIFAX WHERE REPAIR EFFECTED.

- **TSB database**…. POOR FILLET WELD / STRESS CONCENTRATION / CRUCIAL CYCLIC SERVICE LOADING.

# Some Known Examples of Brittle Fracture or Low Toughness Side Shell

- *World Concord*, November 1954
  *Less than 27J at +12º C (temperature at the time of the brittle failure)*
- *Kurdistan*, March 1979
  *27J TT between +5º et +20º C !!*
- *Tyne Bridge*, Winter 1982 *(sister ship to Derbyshire)*
  *CVN reportedly very low*
- *Mesange,* July 1983
  *(ice damage in the Canadian Arctic) 5.6J at 0º C*
- *Kowloon Bridge, November 1986* *(sister ship to Derbyshire)*
- Dodsland, February 1987
  *27J TT about -1º C*
- Erika, December 1999
  *Low CVN contributed to brittle fracture growth. Often, transverse CVN was 50% of longitudinal CVN.*
- *Lake Carling,* March 2002
  *CVN very low … 27J TT +10.5º C*
- *Ziemia Gornoslaska,* December 2003 *(sister ship of Lake Carling)*
  *Transition temperature even higher than Lake Carling … 27J TT +17º C*

# Prestige

# Loss of the Erika

"An "undetectable weakness" in its hull and not metal corrosion was at the origin of the structural collapse of the Erika"

Massimo Gronda, Studio Tecnico Navale Ansaldo

Lloyd's List, 16 May 2007

A pre-existing crack up to 25 cm long spread vertically.

# Age is Not a Factor

| | |
|---|---|
| **Dodsland** | **2 years** |
| **World Concord** | **4 years** |
| **Kurdistan** | **6 years** |
| **Tyne Bridge** | **10 years** |
| **Lake Carling** | **10 years** |
| **Mesange** | **14 years** |
| **Ziemia Gornoslaska** | **13 years** |
| **Erika** | **24 years** |

# ? Some of the Unknown ?

- *Derbyshire*, September 1980
- *Jalamorari*, December 1982
- *Charlie*, January 1990
- *Protektor*, January 1991
- *Gold Bond Conveyor*, March 1993
- *Marika*, January 1994
- *Salvadore Allende*, December 1994
- *Leros Strength,* February 1997
- *Flare*, January 1998 (suffered brittle fracture, but toughness of side shell unknown)
- *Leader L*, March 2000
- *Prestige*, November 2002
- *Aurelia*, February 2005
- *Alexandros T*., May 2006

# IACS new Common Structural Rules for Bulk Carriers (2006)

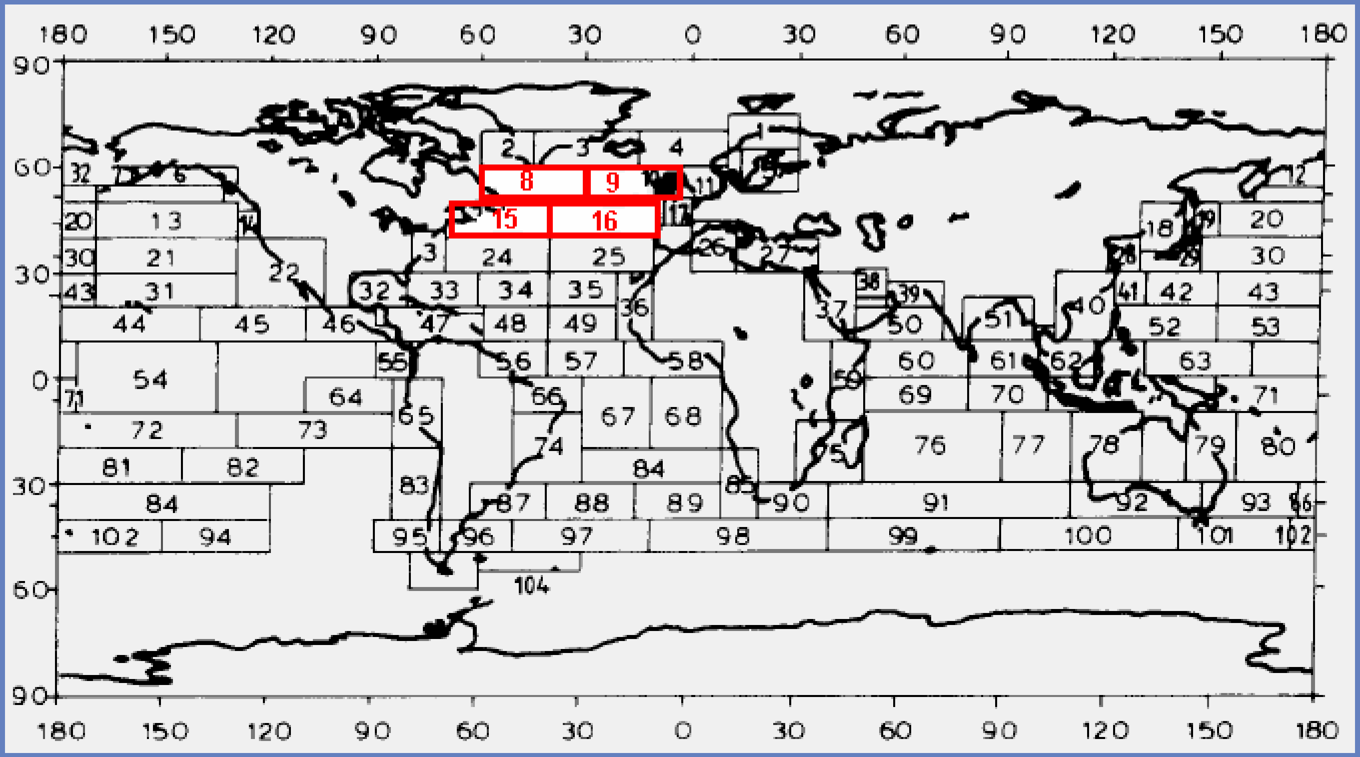
- These Rules have a more stringent standard than previously for single skin BC-A and BC-B ships;
  - but this new requirement for grade D/DH steel is only for strakes in the proximity of the intersection of the side shell and bilge hopper sloping plate.

# IACS new Common Structural Rules for Bulk Carriers (2006)

- New Rules have the qualitative criteria based on “25 Years Operation Life North Atlantic”

**But, at what temperature??**

# North Atlantic Zone





Canada

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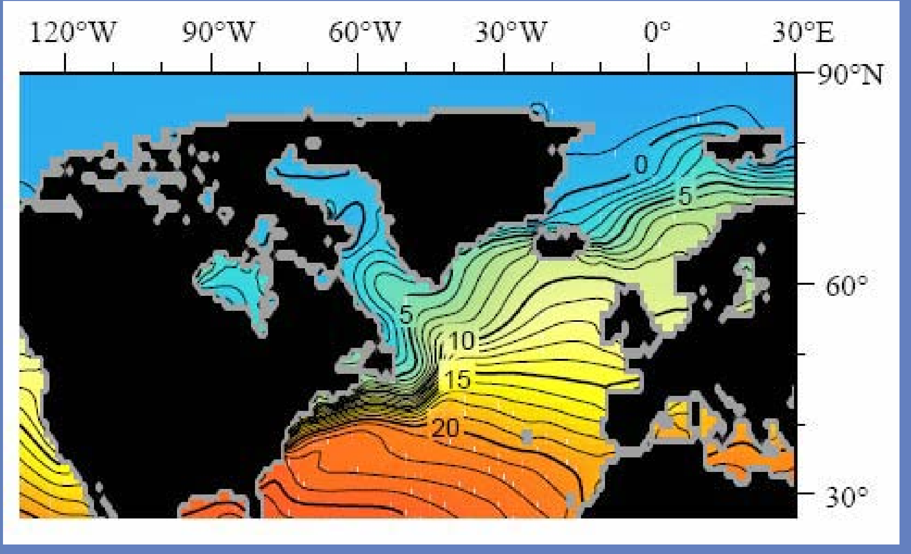
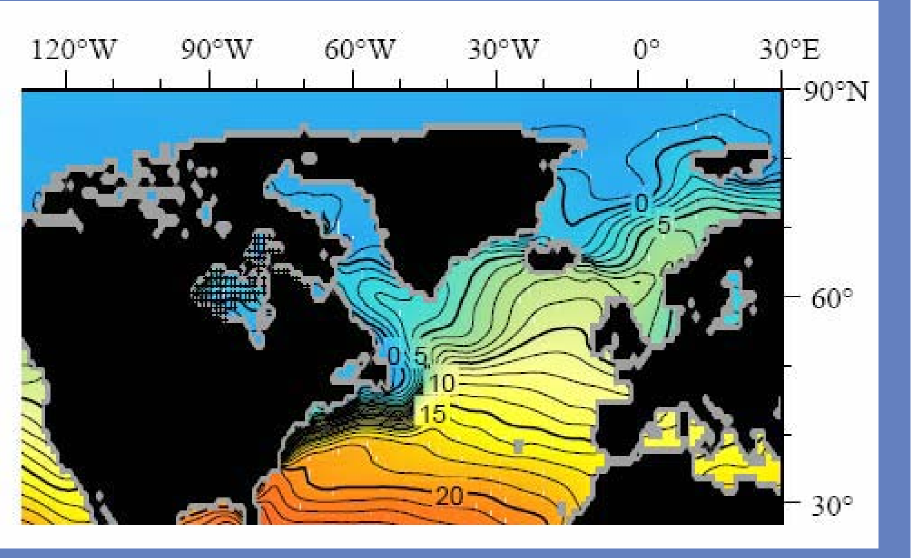
# Service Temperatures North Atlantic

## Annual




## Winter




NOAA Ocean Surface Temperatures, World Ocean Atlas 2005



Canada

# Global Warming as a Crack Driving Force?

**Paradoxically, will global warming bring more ships, more often, to colder waters?**

"Retreating sea ice in the Arctic region has created a 'new Arctic Ocean', with widespread implications for marine access at the top of the world. Interest is growing…in the potential development of Arctic shipping routes…".

Lloyd's List, 24 May 2007

# Residual Risks??

- **Toughness of grade « A » steel is generally high……but…**
- **…toughness of vessel side shells <u>remains unknown</u>.**
- **Nearly <u>13%</u> of samples tested (by Lloyds) exhibited a FATT at <u>0°C or warmer</u>. (Lake Carling was +32°C!)**
- **Another 20% of samples tested exhibited FATT greater than -10°C**

# TSB Safety Concern

- Vessels have been and continue to be constructed with steel of unknown toughness in way of their side shells.
- A significant proportion of these vessels may be exposed to unacceptable risks when operating in colder waters.

# Conclusions

- Steel ships continue to be built – their sides of unknown fracture toughness.
- A significant proportion of side shells have a FATT greater than 0º C !
- Each year, ships sink and lives are lost due to the structural failure of the vessel.
- Some of these structural failures are likely due to brittle fracture caused by the inadequate fracture toughness of the vessel side shell – yet this safety deficiency continues to be neglected.

# What We Have Seen

- Lake Carling Hull Fracture – Genesis and Contributing Factors
- The Safety Deficiency – low (and unknown) toughness of vessel side shells
- Literature & Standards Review
- Residual Risks
- Conclusions

Transportation Safety Board of Canada

Bureau de la sécurité des transports du Canada

Canada

# Thank You

BMT Fleet Technology

# CLASSIFICATION OF POLAR SHIPS

**Andrew Kendrick, BMT Fleet Technology Limited, Canada**
**Claude Daley, Memorial University of Newfoundland, Canada**
**Victor Santos-Pedro, Transport Canada, Canada**
**Richard Hayward, Germanischer Lloyd, Germany**

**R.I.N.A.**
**May 2007**

# Presentation Structure

**Background**

**Existing Ice Class Rules**

**Development of the Polar Rules and IMO Guidelines**

- **Ice loads**
- **Structural response**
- **Other issues**

**Continuing Development**

# Background

The 1970s and 1980s saw dramatic increases in Polar shipping, and many groundbreaking technological developments

In the late 1980s this fell away, due to the collapse in the oil price and the political situation in the ex-USSR

Knowledge gained, and the recognition of gaps in earlier rules and regulations led to the introduction/upgrading of new national and class rules

Several administrations made proposals to IMO to develop a harmonized system of ice class rules

IMO struck a Working Group, under the DE Subcommittee, to explore options and develop a way ahead

BMT Fleet Technology

# IMO and IACS Approach

**The IMO Working Group drew on expertise from stakeholders and experts, including representatives from Class**

**Consensus was reached early in the process to set up parallel groups with overlapping membership and meetings**

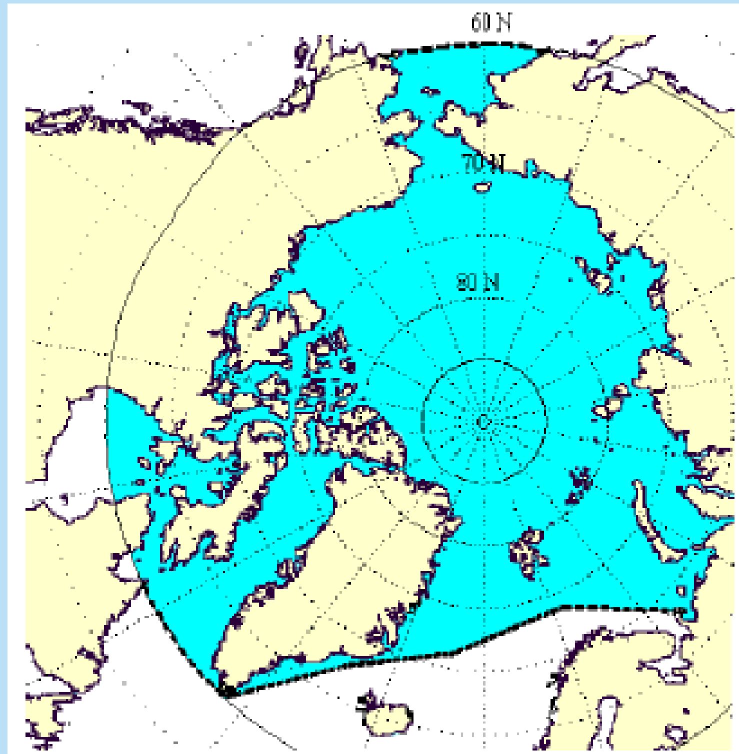
- IMO would develop the overall framework for the initiative
- IACS would produce detailed requirements for construction-related items

# IMO Arctic Guidelines

Initially, the IMO group intended to produce a “Polar Code”, including formalized requirements for ships in Arctic and Antarctic waters.

Concerns over jurisdiction and other issues led to the final version becoming Guidelines - MSC Circular 1056/MEPC Circular 399, “Guidelines for Ships Operating in Arctic Ice Covered Waters” - formally applicable only to Arctic waters.

Even defining the Arctic is not a simple task




BMT Fleet Technology

# Ice Classes – 1990s

**Polar classes only – Baltic system forms part of all IACS member rules (except RMR)**

| Issue | Canadian | | Russian | | ABS | DNV | GL | LR |
|---|---|---|---|---|---|---|---|---|
| | ASPPR | CAC | Old | New | | | | |
| No. of classes | 9 | 4 | 3+4 icebreaker | 6 | 5 (8 if escort available) | 6+3 icebreaker | 4 | 4 |
| Displacement dependency | Strong | Moderate | Strong | Strong | Strong | None | None | Moderate |
| Power dependency | None | None | Weak | None | Weak | None | None | Moderate |
| Structural design basis | Elastic | Elasto-plastic | Elastic | Elasto-plastic | Elasto-plastic | Elastic | Elasto-plastic | Elastic |

# IACS Unified Requirements Development

**Definition of Polar Classes was joint IMO/IACS issue:**

- Set lower bound to capture “Baltic-like” classes with successful polar operating experience
- Set upper bound to include realistic capability limits for economically-viable vessels

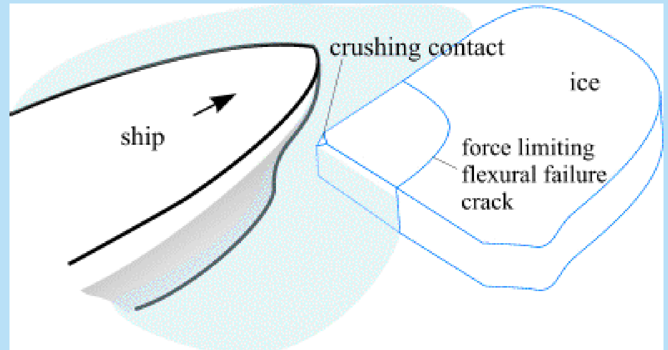
| Polar Class | Ice Description (based on WMO Sea Ice Nomenclature) |
|---|---|
| PC 1 | Year-round operation in all Polar waters |
| PC 2 | Year-round operation in moderate multi-year ice conditions |
| PC 3 | Year-round operation in second-year ice which may include multi-year ice inclusions. |
| PC 4 | Year-round operation in thick first-year ice which may include old ice inclusions |
| PC 5 | Year-round operation in medium first-year ice which may include old ice inclusions |
| PC 6 | Summer/autumn operation in medium first-year ice which may include old ice inclusions |
| PC 7 | Summer/autumn operation in thin first-year ice which may include old ice inclusions |

# Ice Loads

Group wanted to ensure that ice load models were explicit, physics-based, and validated (as far as possible)

More than 70 ship-ice interaction scenarios were considered, from level icebreaking to impacts with icebergs

Basic design scenario is a (glancing) impact with "thick" ice in any of the operating regimes summarized in the class definitions




# Ice Load Derivation

Normal Kinetic Energy = Ice Indentation Energy

$\Downarrow$

Find indentation → Find force, area, pressure.

$$\frac{1}{2}\frac{M_{ship}}{Co}\cdot\left(V_{ship}\cdot l\right)^2 = Po\cdot ka^{1+ex}\int_0^{\delta m}\delta^{2+2\cdot ex}\cdot d\delta$$

Solve for δ

– then solve for Force

# Ice Load Derivation

Exact Solution is:

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

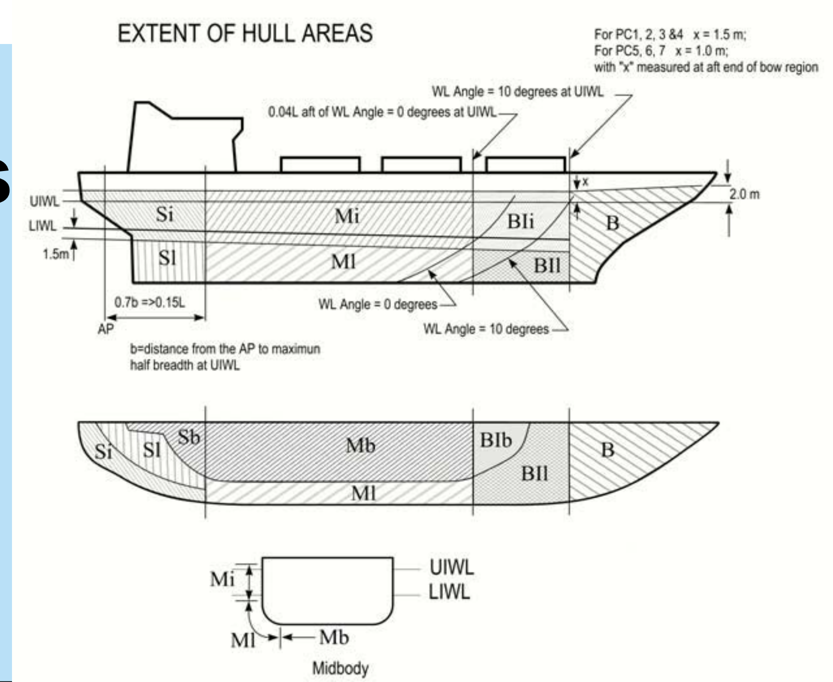
Simplifies to:

$$F_n = fa\cdot Po^{0.36} \cdot V_{ship}^{1.28} \cdot M_{ship}^{0.64}$$

Becomes Rule Equation:

$$F = fa\cdot CFc\cdot D^{0.64}$$

# Loads on other Hull Areas




| Hull Area | | Area | Polar Class | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | PC1 | PC2 | PC3 | PC4 | PC5 | PC6 | PC7 |
| Bow (B) | All | B | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 |
| Bow Intermediate (BI) | Icebelt | $BI_i$ | 0.90 | 0.85 | 0.85 | 0.80 | 0.80 | 1.00* | 1.00* |
| | Lower | $BI_l$ | 0.70 | 0.65 | 0.65 | 0.60 | 0.55 | 0.55 | 0.50 |
| | Bottom | $BI_b$ | 0.55 | 0.50 | 0.45 | 0.40 | 0.35 | 0.30 | 0.25 |
| Midbody (M) | Icebelt | $M_i$ | 0.70 | 0.65 | 0.55 | 0.55 | 0.50 | 0.45 | 0.45 |
| | Lower | $M_l$ | 0.50 | 0.45 | 0.40 | 0.35 | 0.30 | 0.25 | 0.25 |
| | Bottom | $M_b$ | 0.30 | 0.30 | 0.25 | ** | ** | ** | ** |
| Stern (S) | Icebelt | $S_i$ | 0.75 | 0.70 | 0.65 | 0.60 | 0.50 | 0.40 | 0.35 |
| | Lower | $S_l$ | 0.45 | 0.40 | 0.35 | 0.30 | 0.25 | 0.25 | 0.25 |
| | Bottom | $S_b$ | 0.35 | 0.30 | 0.30 | 0.25 | 0.15 | ** | ** |

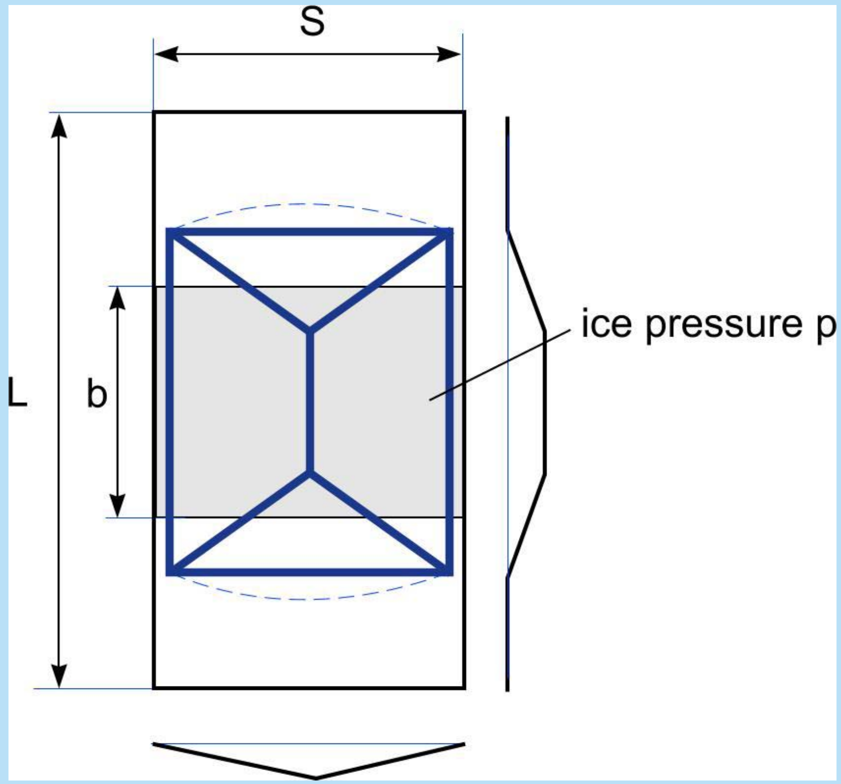
# Structural response

**URRs utilize advanced elasto-plastic response formulations:**

- Utilizes true capacity of structure
- Helps ensure robust designs
- Does not compromise safety or serviceability limit states

# Plate requirements

$$t = 0.5s\sqrt{\frac{p}{FY}} \cdot \frac{1}{1 + 0.5\frac{s}{b}}$$




**similar to plastic collapse formula for uniformly loaded plate**

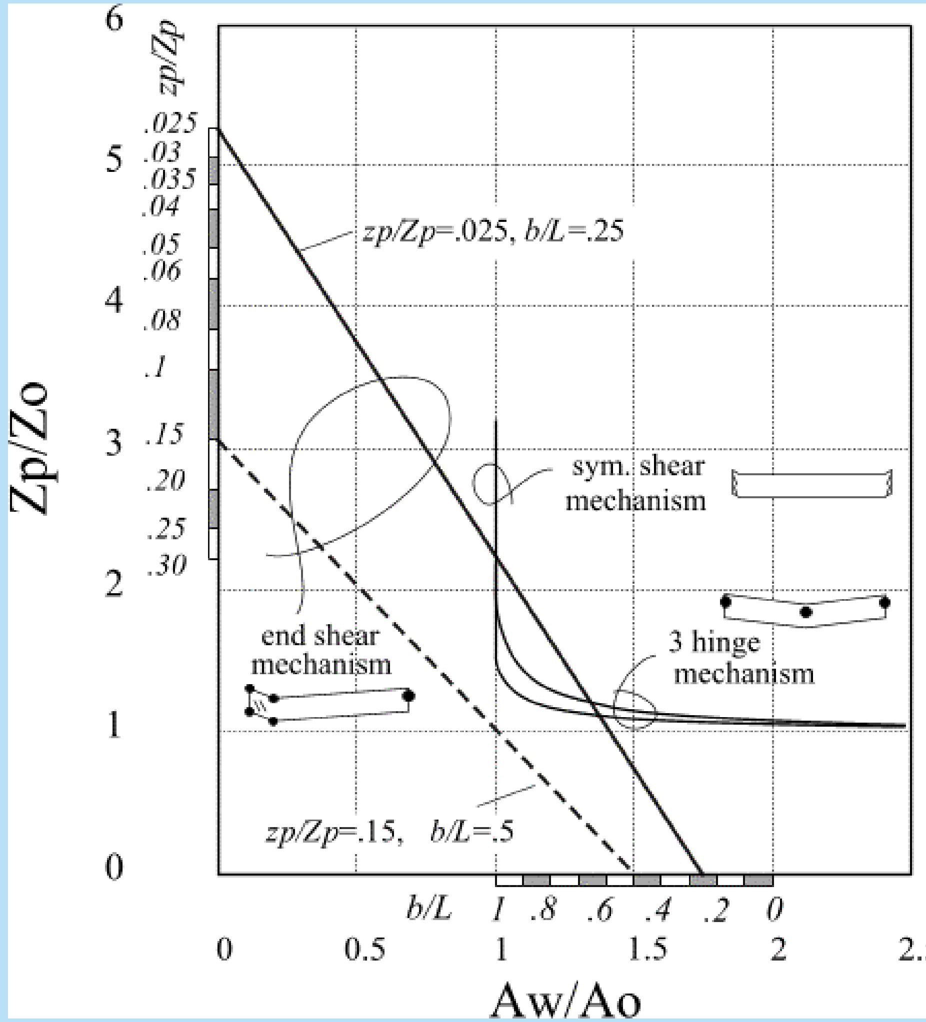
s/b term reflects load height effect

# Framing Requirements

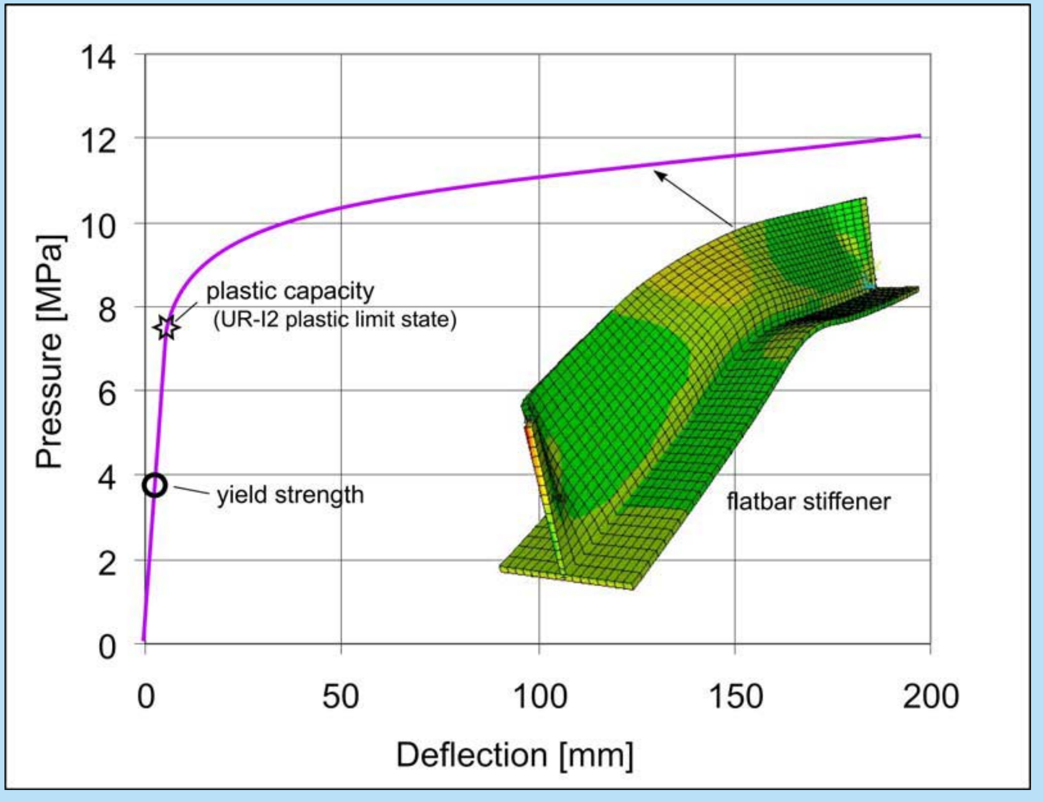
Considers bending/shear interaction effects

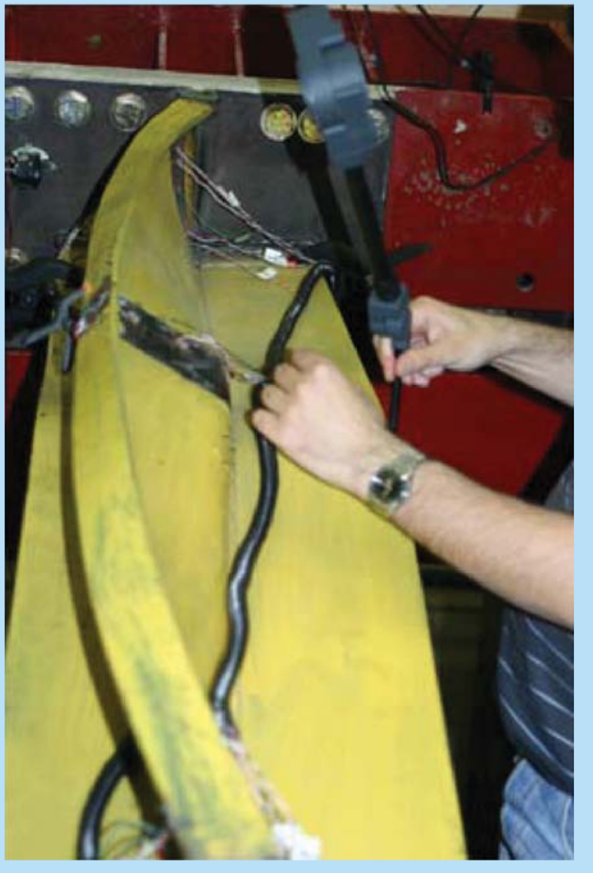
Includes range of load locations and failure mechanisms

Permits use of complete solution domain



# Framing Response



# Other Aspects of the URs

Hull girder strength – an issue for high polar classes only

Material grades – pragmatic, and based on successful experience

Stuctural instability – probably conservative for most sections

Corrosion and abrasion allowances – importance of effective coatings

# Uncertainties and Gaps

**Grillage design**

**Plate structures**

**Use of F.E. methods**

**Pressured ice loads**

**Extrapolation to non-traditional ship types**

# Summary

Development of the new Unified Requirements represents a unique collaboration between IACS and IMO, drawing in additional expert stakeholders

The URs are the best available basis for the design of the next generation of ice-capable ships

The URs should be used with caution.  No ship is safe in all ice conditions unless it is  operated with due caution and with respect for the conditions.

# Thank you – Questions?

# Environmental Protection Requirements for Arctic Shipping

Robert Tustin
Robert Hindley
Des Upcraft

May 31, 2007

DESIGN & CONSTRUCTION OF VESSELS OPERATING IN LOW TEMPERATURE ENVIRONMENTS
LLOYD'S REGISTER

# Introduction

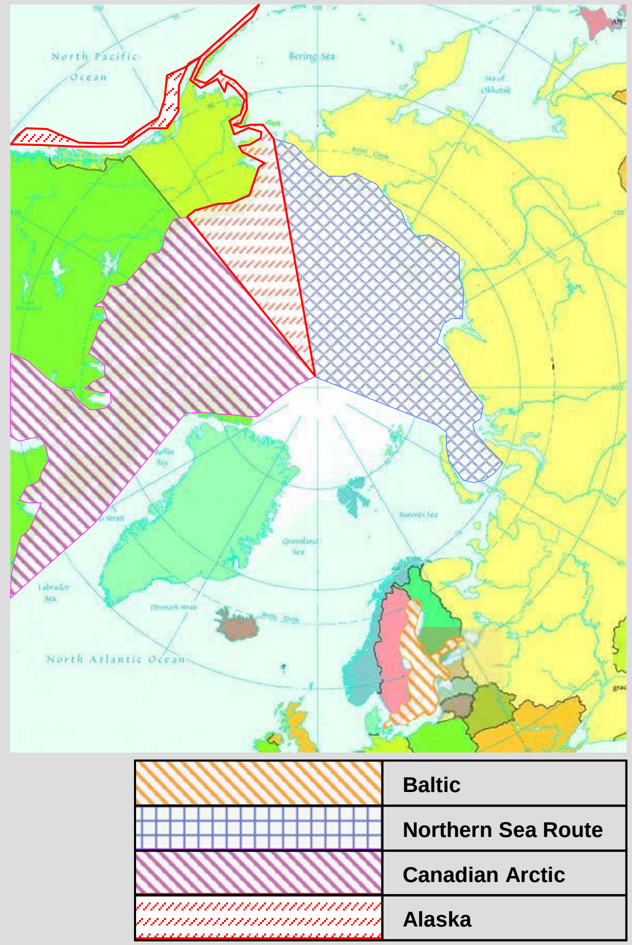
- ***A challenging question for future Arctic ships***
    - *"What environmental protection requirements to specify"*
- **An answer to this question has been derived from:**
    - Analysis of current environmental protection regulations & requirements
    - An opinion on future* environmental protection requirements
- **Premise behind our analysis of the future*:**
    - Answer is already in current regulations or expected development of those regulations

# Diverse body of regulators for environmental protection

- International *(IACS & IMO)*

- Regulators for Arctic seas with regional responsibilities:
  - Canadian Arctic (*Transport Canada)*
  - Russian Arctic *(NSRA)*
  - Alaska *(Alaska State)*

- *"Regulatory trend setters"* … for seasonally ice covered & environmentally sensitive seas:
  - Baltic *(HELCOM)*
  - Great Lakes *(USCG & Transport Canada)*
  - Alaska *(Alaska State)*
  - North Sea *(EU)*



6 Sea Areas

14 Regulatory Organisations

22 Regulations

Scope of Application

Geographical Limits

Aspect of Ship Design

Identification of Environmental Protection Requirements

| Sea Area | Organisation | Regulations | Applicability of Regulations | Hull Structural Arrangements | Air Pollution Prevention | Anti Fouling Systems | Ballast Water Management | Processing & Discharge of Oily Waste, Sewage, Garbage |
|---|---|---|---|---|---|---|---|---|
| Baltic | FMA / SMA | Finnish Swedish Ice Class Rules | Finnish Waters / Swedish Waters | ✓ | | | | |
| Baltic | Port of Primorsk | Port Regulation | Primorsk Waters | | | | ✓ | ✓ |
| Baltic | HELCOM | Convention on the Protection of the Marine Environment of the Baltic Sea Area and subsequent recommendations | Baltic | ✓ | ✓ | ✓ | | ✓ |
| Arctic Canada | Transport Canada | Joint Industry Coast-guard guidelines for the control of oil tankers and bulk chemical carriers in ice control zones of Eastern Canada | Canada | ✓ | | | | |
| Arctic Canada | Transport Canada | Arctic Shipping Pollution Prevention Regulations | Canada | ✓ | | | | ✓ |
| Arctic Canada | Transport Canada | Equivalent Standards for the Construction of Arctic Class Ships | Canada | ✓ | | | | |
| Arctic Canada | Transport Canada | Guidelines for the Operation of Tankers and Barges in Canadian Arctic Waters (Interim) | Canada | | | ✓ | | |
| Arctic Canada | Transport Canada | Ballast Water Control and Management Regulations | Canada | | | | | ✓ |
| Arctic Canada | Transport Canada | A guide to Canada's Ballast Water Control and Management Regulations | Canada | | | | ✓ | |
| Arctic Canada | Transport Canada | Pollution Prevention Guidelines for the Operation of Cruise Ships under Canadian Jurisdiction | Canada | | | ✓ | | ✓ |
| Great Lakes Area | Transport Canada | Great Lakes Sewage Pollution Prevention Regulations | Great Lakes | ✓ | | | | ✓ |
| Great Lakes Area | USCG | Reporting by Foreign Flagged Vessels bound for the Great Lakes | Great Lakes | | | | ✓ | |
| Great Lakes Area | USCG | Ballast Water Management for Control of Non Indigenous Species in the Great Lakes and Hudson River | Great Lakes | | | | ✓ | |
| Great Lakes Area | USCG | USCG Ballast Water Management | Great Lakes & Hudson River | | | | ✓ | |
| Great Lakes Area | St Lawrence Seaway Corp | The Seaway Handbook | Great Lakes / St. Lawrence | | | | | ✓ |
| Arctic Russia | Russian Register | Rules for the Classification and Construction of Sea-Going Ships | Northern Sea Route | ✓ | | | | |
| Arctic Russia | NSRA | Requirements for the Design, Equipment and Supplies of Vessels Navigating the Northern Sea Route | Northern Sea Route | ✓ | | | | ✓ |
| Arctic Russia | Russian Federal Government | Arctic Port Regulations | Russian Arctic | ✓ | | | ✓ | ✓ |
| All Arctic Areas | IACS | Unified Requirements for Polar Ships | Arctic | ✓ | | | | |
| All Arctic Areas | IMO | Guidelines for ships operating in Arctic ice-covered water | Arctic | ✓ | | | | |
| Alaska | Alaska State | Certain Alaskan Cruise Ship Operations | Alaska | | | | | ✓ |
| Alaska | Alaska State | Commercial Passenger Vessel Environmental Compliance Program - 18 AAC 69 | Alaska | | ✓ | ✓ | | ✓ |

**Five Aspects of Ship Design**

**Future Requirements** | **International Requirements** | **Arctic Sea Area Requirements**

**Detail Requirements for Environmental Protection**

| **Sea areas** *(principal regulatory document)* | | **All Arctic Seas** *(Possible future regulation?)* | **All Seas** *(MARPOL)* | **All Arctic Sea areas** *(IMO Arctic Guidelines)* | **Russian Arctic** *(Guide to Navigating the Northern Sea Route)* | **Canadian Arctic** *(ASPPR & Equivalent Standards)* |
|---|---|---|---|---|---|---|
| **Environmental Protection Issue** | **Detail** | | | | | |
| **Hull Structural Arrangement** | Double skin | Required for any potential pollutant anywhere in length | For oil tankers | No oil pollutant in direct contact with side shell | Recommended double side. Required iwo ER for some ships. | Required side tank dimensions. Required iwo oil tanks. No harmful waste in direct contact with side shell |
| | Double bottom tanks | Required through ships length | For oil tankers. Pump Room bottom protection required. | Between FP and AP bulkheads | Between FP and AP bulkheads | Required iwo oil tanks. No harmful waste in direct contact with side shell |
| | Bunker oil tanks | Not against side shell | Double skin protection required for tanks | No oil Pollutant in direct contact with side shell | Double bottom and sides not to be used for oil products. | Fuel or Cargo oil to be 0.76m from side shell. Waste oil to be 0.76m from shell |
| | Waste (garbage, sewage) tanks | Not against side shell | | No harmful pollutant in direct contact with side shell | Sewage collecting tank (if no treatment system) | No harmful waste in direct contact with side shell |
| | Stability | Intact, Damage, Ramming | Intact (only ice deck accretion ) and Damage | Intact and Damage | Intact and Damage | Intact, Damage, Ramming requirements. |
| **Anti Fouling Systems** | TBT free coatings | Required | Effectively implemented | | | Required for cruise ships |
| **Water Ballast Management** | Water ballast control method | Treatment | Exchange or Treatment | | | Specific Areas for exchange |
| **Air Pollution Prevention** | Maximum Sulphur Oxide emissions | 6g/kWh | | | | |
| | Maximum Nitrogen Oxide emissions | 17g/kWh | 17g/kWh | | | |
| | Maximum Sulphur content of fuel | 0.1% | 4.5% | | | Average for cruise ships 1.5% |
| **Processing and Discharge of Oily Wastes, Garbage, Sewage** | Oily Water Separators | Not Permitted – Retain oil on board | Performance - 15ppm | | | 5ppm (some internal waters)<br>15ppm (external waters) |
| | Incineration | Not Permitted – Retain waste on board | Permitted | | Required or a storage tank | Not permitted in port for cruise ships |
| | Discharge (sewage) | Retain sewage on board | Storage or discharge | | Storage or discharge | Storage or discharge |
| | Discharge (garbage) | Retain garbage on board | Storage or discharge | | Storage or discharge | Storage or discharge |

# Hull layout & arrangement for pollution prevention

- **Current requirements:**
  - Separation of cargo oil from the shell of Arctic ships required
  - Separation of fuel oil from the shell of Arctic ships by imminent regulation
- **Future requirements for Arctic seas:**
  - All potential pollutants may be separated from the shell including oily wastes, sewage and garbage

# Processing & discharge of oily wastes, sewage & garbage

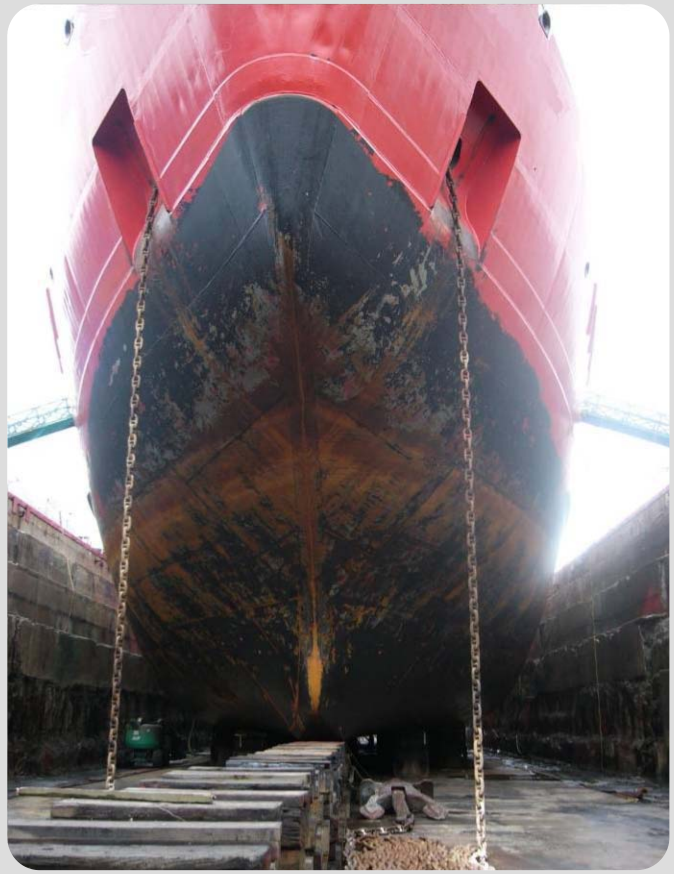
- **Current requirements:**
    - International requirements established for the processing and treatment of oily wastes, sewage and garbage
- **Future requirements for Arctic seas:**
    - Holding of wastes, and separation of waste holding spaces, from the exposed hull

# Air pollution prevention

- **Current requirements:**
  - NOx and SOx emissions limits have been established internationally
  - Limitations on sulphur content of fuel have also been established internationally
- **Future requirements for Arctic seas:**
  - More stringent sulphur emissions
  - More stringent sulphur content limits
  - SECAs in Arctic seas
- **Most uncertainty on future requirements:**
  - Further work needed to examine other *"regulatory trend setters"* e.g. California Air Resources Board
  - Develop opinions on future regulation of GHG emissions (post Stern report) …

# Anti Fouling Systems

- **Current requirements:**
  - No exposed anti-fouling paint containing organotin compounds
  - Total ban on tributyltin (TBT) contained within antifouling paints in EU Waters
- **Future requirements for Arctic seas:**
  - TBT paint ban in Arctic seas is likely before AFS convention ratified



# Ballast water management

- **Current requirements:**
  - Restrictions on ballast water exchange in sensitive sea areas
- **Future requirements for Arctic seas:**
  - Early implementation of ballast water treatment methods to Arctic … this as compliance with exchange methods in shallow Arctic seas is practically impossible

## *“What environmental protection requirements to specify”*
## - our opinion on the answer for Arctic shipping

- **For future Arctic ships:**
  - **No waste discharges** – of oily wastes, sewage and garbage requiring onboard holding tanks and spaces for the duration of the voyage
  - **No contact with exposed hull of any potential pollutants** – including oil cargo, fuel oil as well as oily wastes, sewage and garbage
  - **Water ballast management by treatment**
  - **Stringent limits on sulphur content of fuels** - for SECAs in Arctic seas

- **Further analysis needed on air pollution prevention some uncertainty about our conclusions**
- **Ultimate outcome of legislation for environmental protection in Arctic sea areas will probably be a “*zero emission/discharge*” ship**

# Environmental Protection Requirements for Arctic Shipping

Robert Tustin

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Lloyd’s Register Asia

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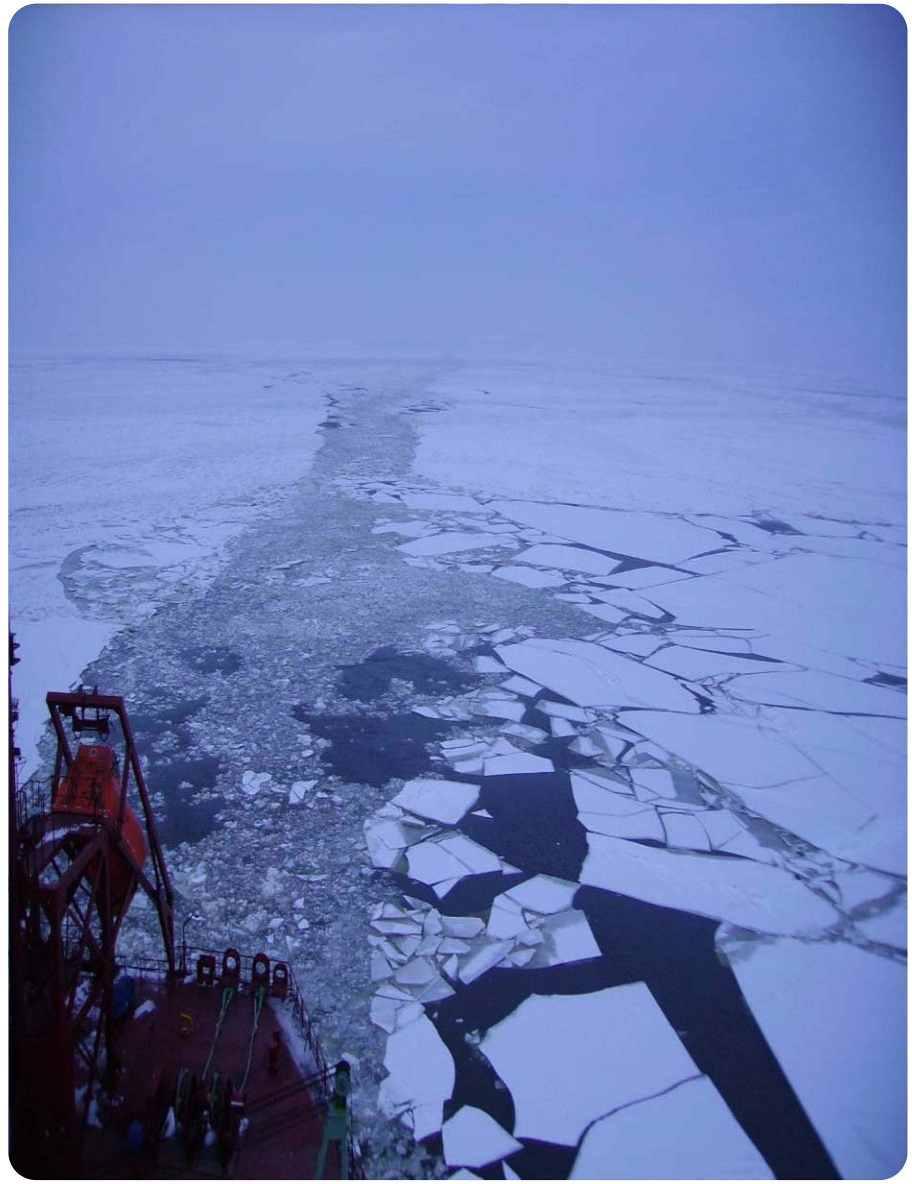


# Factors influencing the choice of an ice class

**Robert Bridges**
Research & Development

RINA - Design and construction of vessels operating in low temperature environments
30-31 May 2007

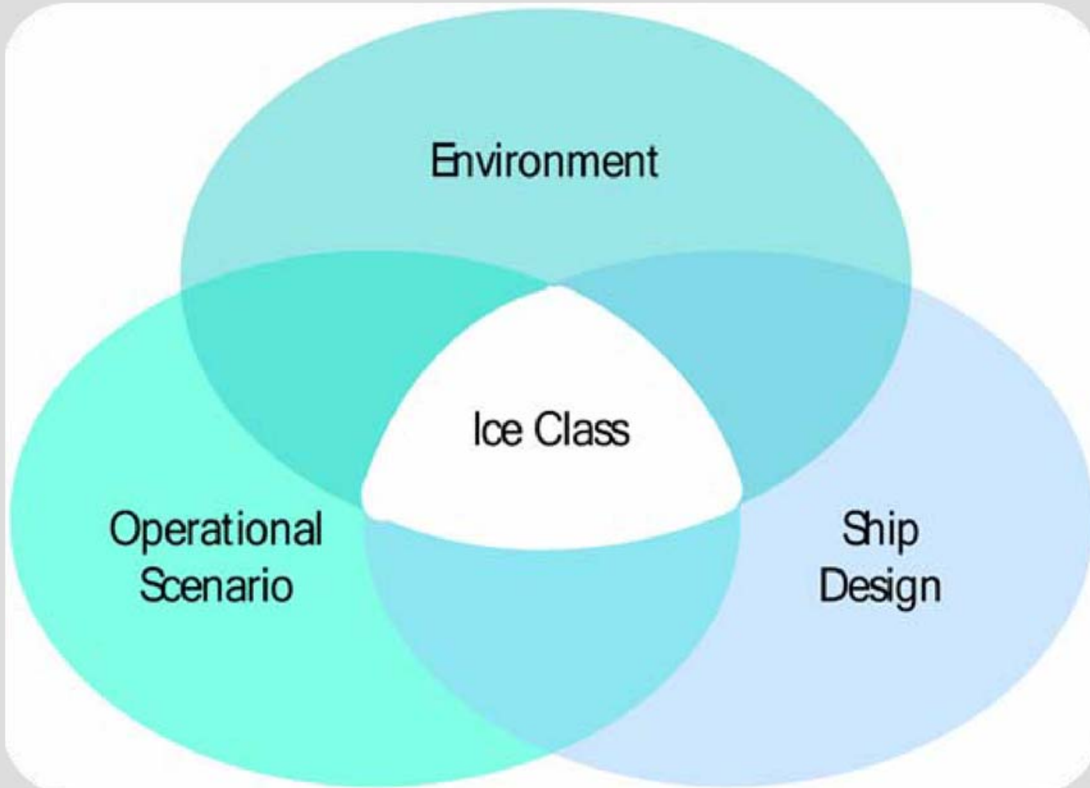
ICE & COLD OPERATIONS
LLOYD'S REGISTER

Lloyd's Register

# How to select an ice class?

- Owner requirements
- Administration requirements
- Duration of voyage
- Speed in ice
- Icebreaker assistance
- Hull form
- Ice thickness
- Ice concentration
- Size of ship
- Engine power
- Ice strength
- Crew experience

# How to select an ice class?



# Brief history

| Year | Description | Notation |
|---|---|---|
| 1924 | First notation | Strengthened for Navigation in Ice |
| 1958 | Percentage rules | Ice Class 1, Ice Class 2 and Ice Class 3 |
| 1968 | Increasing capability of ships | Ice Class 1* |
| 1971 | Ice pressure introduced in the Finnish Swedish Ice Class Rules | 1AS, 1A, 1B and 1C |
| 1985 | Multi-year ice classes | AC1, AC1.5, AC2 and AC3 |
| 2002 | Finnish Swedish Ice Class Rules update | 1AS FS, 1A FS, 1B FS and 1C FS |
| 2006 | IACS Polar Class Rules finalised | PC1, PC2, PC3, PC4, PC5, PC6 and PC7 |

# Current first-year and multi-year ice class rules

## First-year ice


- one winter's growth
- 120cm thick and low ice-strength properties
- **Finnish-Swedish Ice Class Rules**

## Multi-year ice


- survived at least one summer's melt
- 3m or more and high ice-strength properties
- **IACS Polar Ship Rules**

# What is an ice class?

| Lloyd's Register Ice Class | Finnish-Swedish Ice Class | Ice thickness (metres) | | Polar Ice Class | Ice Description (Based on WMO Sea Ice Nomenclature) |
|---|---|---|---|---|---|
| | | | | 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 which may include multi-year ice inclusions. |
| | | | | PC 4 | Year-round operation in thick first-year ice which may include old ice inclusions |
| | | | | PC 5 | Year-round operation in medium first-year ice which may include old ice inclusions |
| 1AS FS(+)<br>1AS FS | IAS | 1.0 | | PC 6 | Summer/autumn operation in medium first-year ice which may include old ice inclusions |
| 1A FS(+)<br>1A FS | IA | 0.8 | | PC 7 | Summer/autumn operation in thin first-year ice which may include old ice inclusions |
| 1B FS(+)<br>1B FS | IB | 0.6 | | | |
| 1C FS(+)<br>1C FS | IC | 0.4 | | | |
| Note, although PC6 is equivalent to 1AS FS (and PC7 to 1A FS), the requirements will differ based on their intended operation. For example, the extents of reinforcement will be greater on the multi-year ice class due to the larger variation in ice conditions, escort operations etc. | | | | | |

# What is the difference in ice classes?

- Added steel weight
  - Icebelt region
  - Plate and frame thickness increase

- Increased propulsion and machinery

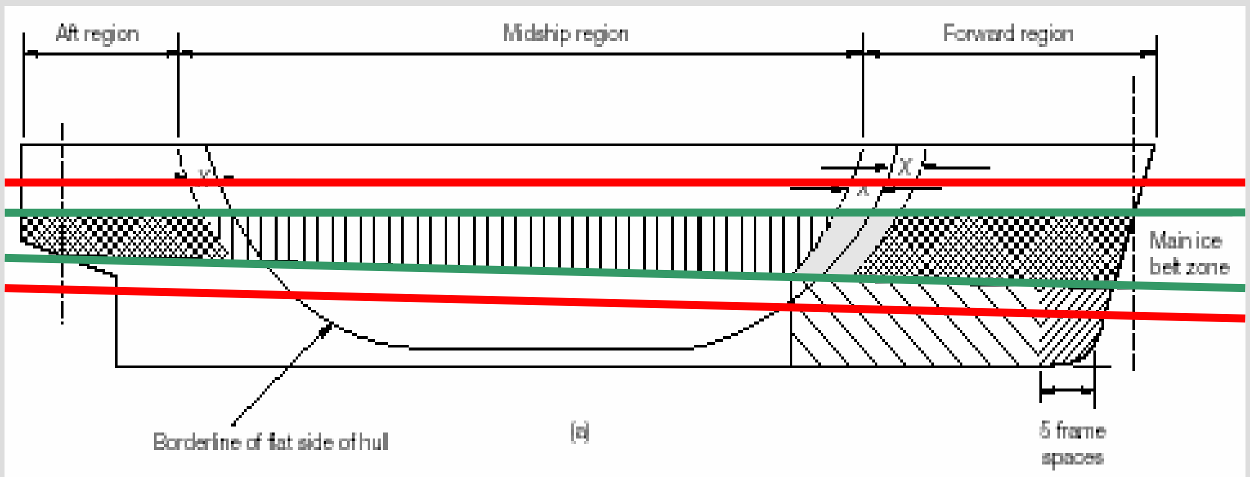



**Plate extents, m**

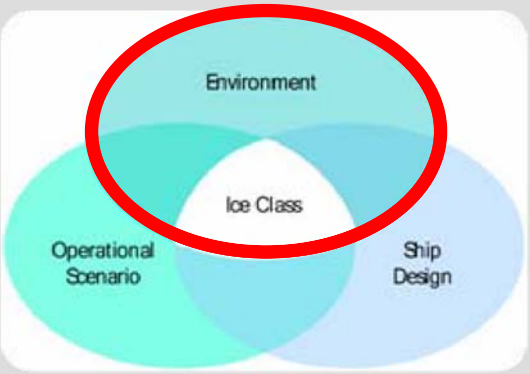
| Ice Class | Above LWL | Below BWL |
|---|---|---|
| 1AS | 0.6 | 0.75 |
| 1A | 0.5 | 0.6 |
| 1B | 0.4 | 0.5 |
| 1C | 0.4 | 0.5 |

# The environment

- Air temperature
  - Extreme low temperatures
  - Duration of time at low temperature
  - Rate of temperature drop

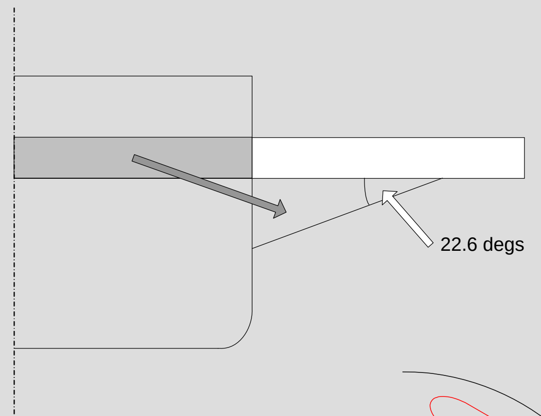
- Sea conditions
  - Salinity
  - Sea state
  - Currents

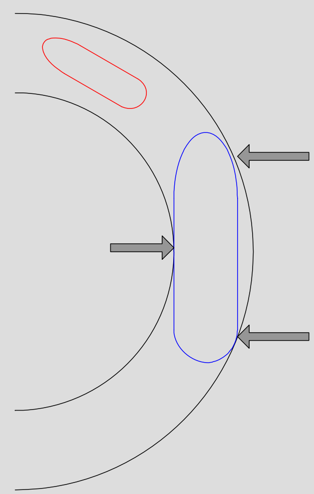
- Geological features
  - Depth of water
  - Proximity to land




# Ice characteristics

- Strength of ice
  - Tensile, compression, flexural, shear
- Ice thickness
  - Increase contact area with ship
- Ice drift
  - Movement of ice generate pressure acting on ship
- Ice ridges
  - Pose significant challenge for ships
- Icebergs
  - Not accounted for in ice class rules
- Ice channels
  - Particularly applicable to regions where icebreakers operate
- Ice extent and duration
  - Range of impacts and likelihood of encountering difficult ice features (ice ridges)



# The operational scenario


- Five main modes of navigating in ice:

1. Independent
   - 1a in level ice
   - 1b in channel ice
2. With icebreaker
   - 2a singularly
   - 2b in convoy
3. Towed by an icebreaker

- The influence of the master
- Duration in ice

# Speed in ice

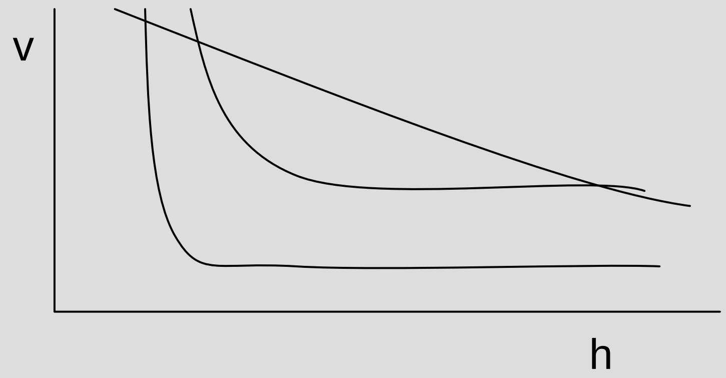
- Finnish Swedish ice class rules:

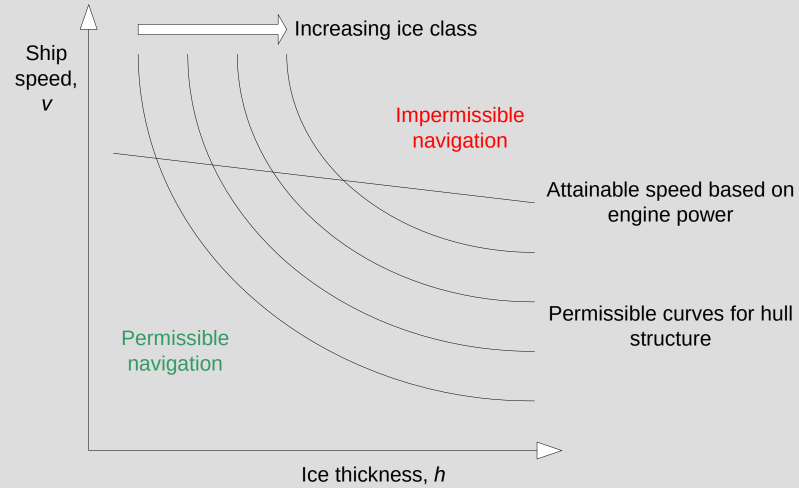
$$k = \frac{\sqrt{\Delta P}}{1000}$$

- IACS PC rules:

$$force = faP^{0.36}\Delta^{0.64}V^{1.28}$$

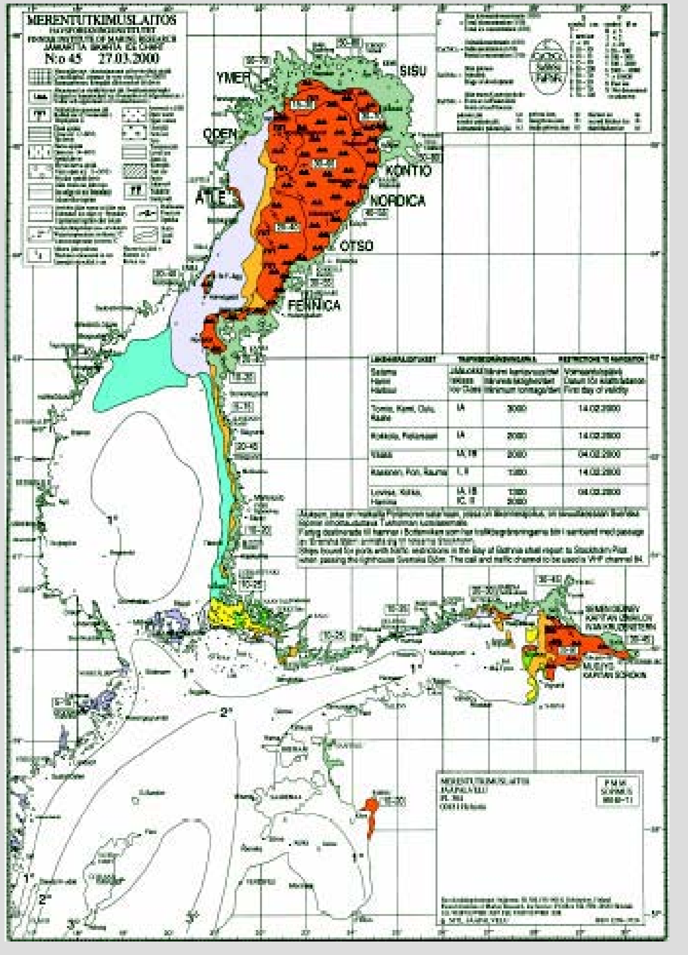
- **Speed/ice curves**
- The ‘Ice Passport’ was devised over 25 years ago in Russia by AARI
- Similar documents produced by Krylov and CNIIMF
- Today CNIIMF are the approved body within Russia to produce the ‘Ice Certificate’



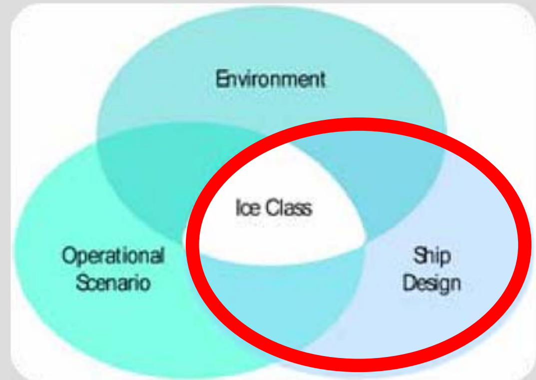



# Administration requirements

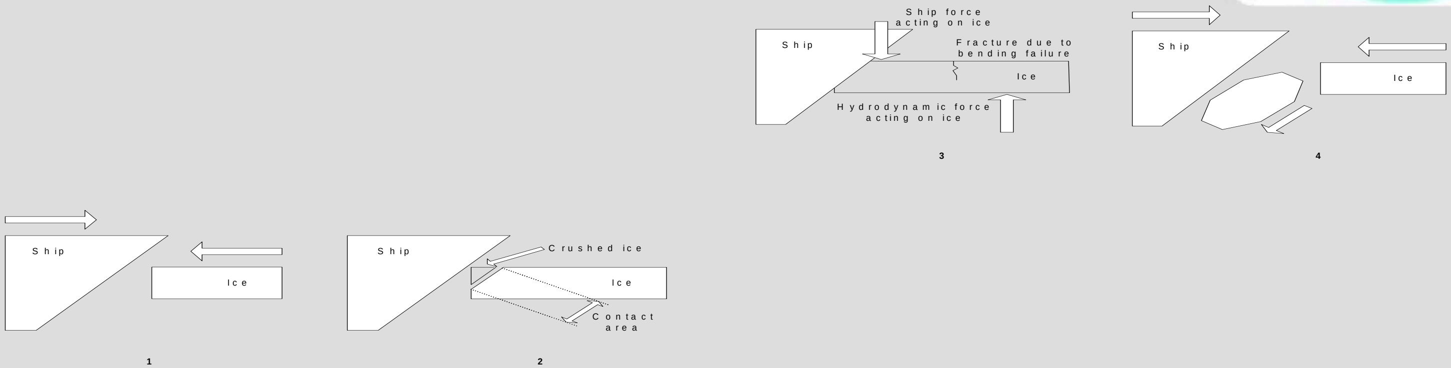
- The administrations of Sweden and Finland provide icebreaker assistance to ships bound for ports in respective countries in the winter season
- Continuity of traffic philosophy
- Fairway dues
- Depending on the ice conditions, restrictions are enforced by weekly traffic notices (restrictions)
    - Ice class
    - Tonnage
- Merchant fleet – Rules – Icebreakers



# Ship design

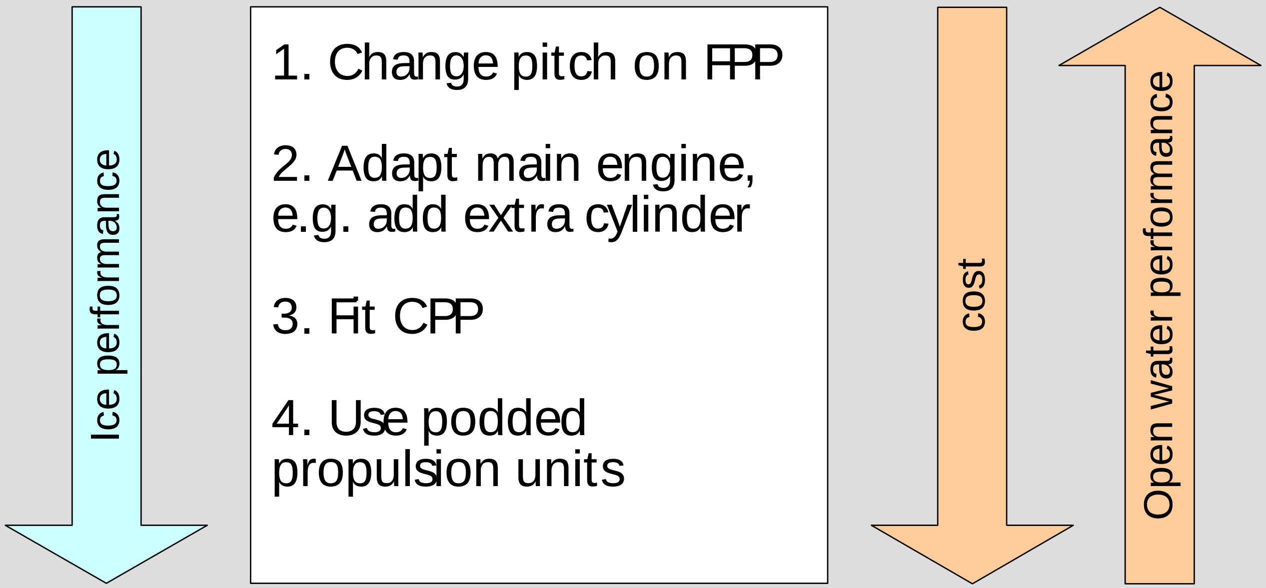



- Hull strengthening

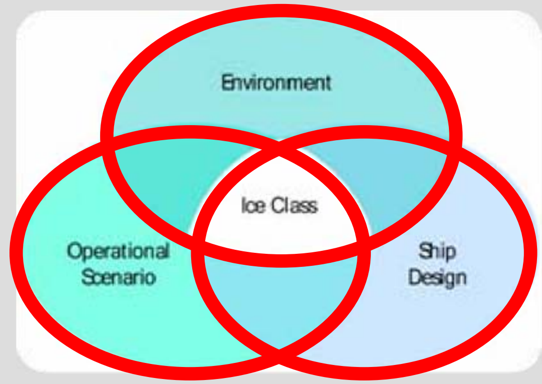


# Propulsion

- Balance between engine power and hull strength
- Balance between open water performance and ice going performance



# Framework for selecting an ice class

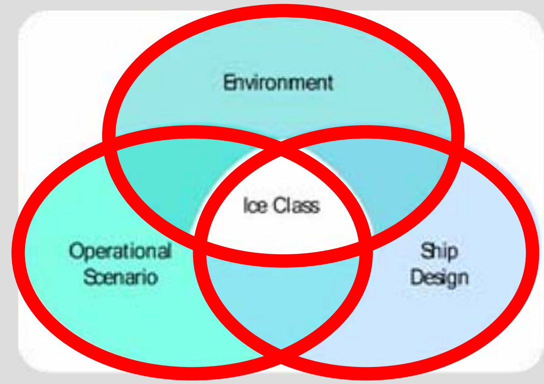



- Three factors divided into individual subcomponents
- Provided with a scale and description

| | Environment (ice properties) | | | | | Operation | | | | | Design | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C1a | C1b | C1c | C1d | C1e | C2a | C2b | C2c | C2d | C2e | C3a | C3b |
| | **ice strength (flexural)** | **ice thickness, h** | **ice drift** | **ice ridging** | **ice extent** | **speed** | **frequency** | **operation** | **Crew** | **administration restrictions** | **hull form optimisation** | **ship size and power** |
| Low | Weak | Thin | no compression | no ridges | small area | slow | Occasional | escorted | highly experienced | low ice region & time | Non ice class | Small |
| Moderate / Low | Medium weak | Medium thin | slow closing | few small ridges | moderate small area | medium slow | Regularly | Occasional independent | moderate experience | moderate low ice region & time | Ice optimised low | Medium small |
| Moderate / High | Medium Strong | Medium thick | quick closing | large ridges | moderate large area | medium fast | Often | Occasional ramming | little experience | moderate high ice region & time | Ice optimised high | Medium large |
| High | Strong | Thick | high compression | many and large ridges | large area | fast | Continuous | Icebreaking operations | no experience | high ice region & time | Icebreaking | Large |

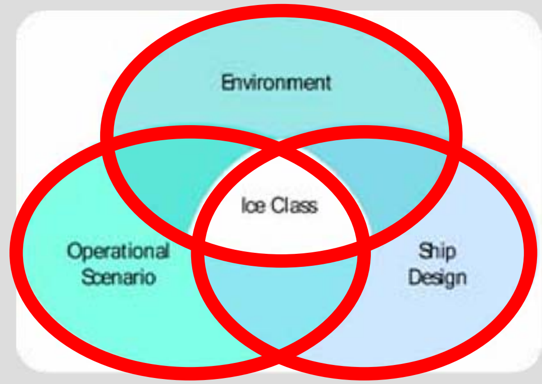
# Framework for selecting an ice class

- Descriptions replaced with numerical values




| | Environment | | | | | Operation | | | | | Design | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C1a | C1b | C1c | C1d | C1e | C2a | C2b | C2c | C2d | C2e | C3a | C3b |
| | **flexural strength Mpa** | **m** | **m/s** | **concentration %** | **nm2** | **knots** | **days in ice** | **operation** | **Crew** | **administration restrictions** | **bow angle** | ***k*** |
| Low | 0.2 | 0.4 | 0.2 | 2 | 200 | 5 | 10 | escorted | highly experienced | 1C | 20 | 11 |
| Moderate / Low | 0.3 | 0.6 | 0.5 | 4 | 250 | 7 | 40 | Occasional independent | moderate experience | 1B | 30 | 12 |
| Moderate / High | 0.4 | 0.8 | 0.8 | 6 | 300 | 9 | 80 | Occasional ramming | little experience | 1A | 40 | 13 |
| High | 0.5 | 1.0 | 1.1 | 8 | 350 | 11 | 120 | Icebreaking operations | no experience | 1AS | 50 | 14 |

# Example – Baltic LNG carrier




- Values are provided with numerical scale

| | Environment | | | | | Operation | | | | | Design | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | C1a | C1b | C1c | C1d | C1e | C2a | C2b | C2c | C2d | C2e | C3a | C3b |
| | **flexural strength Mpa** | **m** | **m/s** | **concentration %** | **nm2** | **knots** | **days in ice** | **operation** | **Crew** | **administration restrictions** | **bow angle** | ***k*** |
| Low | 1 | 1 | 1 | 1 | 1 | 1 | 1 | **1** | 1 | **1** | 1 | 1 |
| Moderate / Low | 2 | 2 | **2** | 2 | **2** | **2** | **2** | 2 | 2 | 2 | **2** | 2 |
| Moderate / High | **3** | **3** | 3 | **3** | 3 | 3 | 3 | 3 | **3** | 3 | 3 | **3** |
| High | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 |

# Example – Baltic LNG carrier

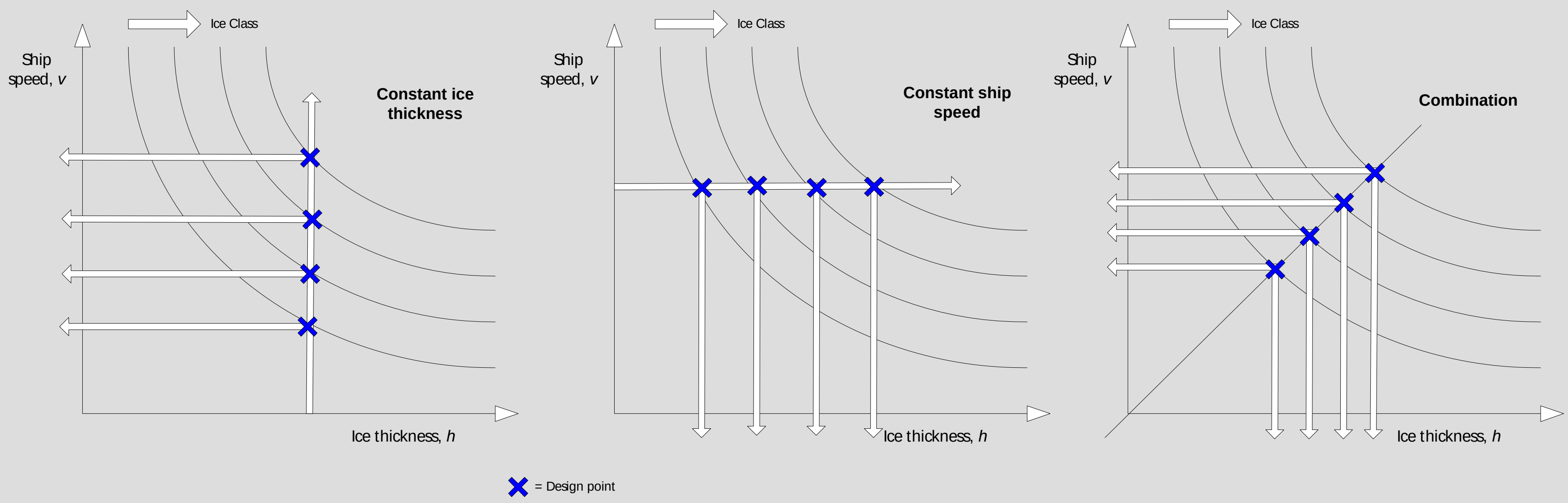
- Each ice class assigned numerical value
- Average numerical values for each factor
- Comparison

| **Ice Class** | |
|---|---|
| **1** | **1C** |
| **2** | **1B** |
| **3** | **1A** |
| **4** | **1AS** |

| **Element** | **Average** | **Ice Class** |
|---|---|---|
| Environment | 2.6 | 1A |
| Operations | 1.8 | 1B |
| Ship design | 2.5 | 1A |
| **Total** | **2.17** | **1B** |

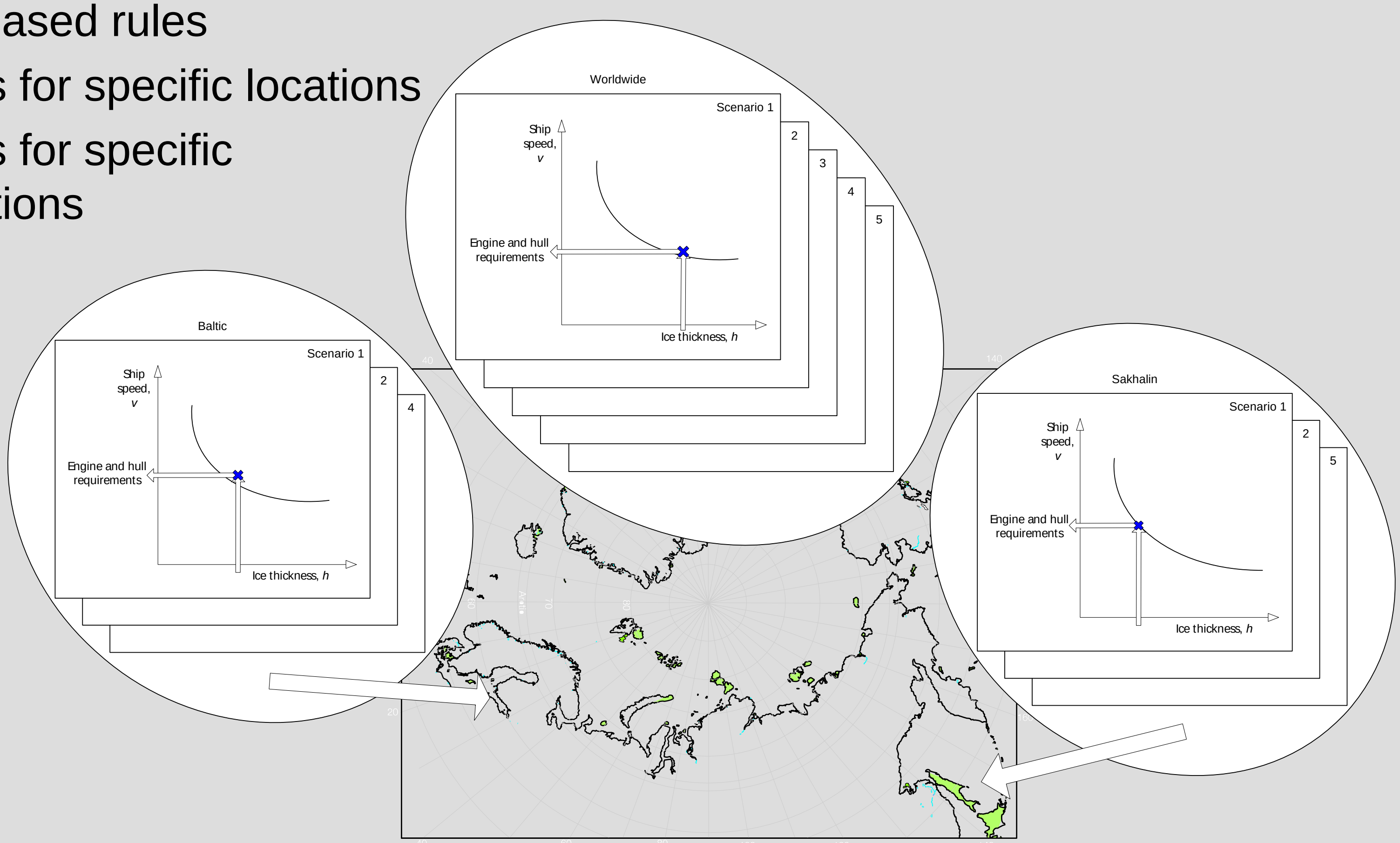
# The future of ice class?

- Speed/ice curves for ice classes



# The future of ice class?

- Scenario based rules
  - Caters for specific locations
  - Caters for specific operations



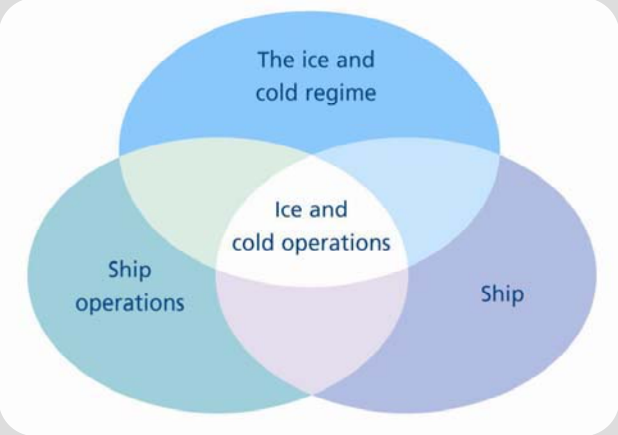
# Summary

Lloyd's Register is committed to:

- Involvement in ice related projects
- Developing and implementing ice class rules and requirements
- Providing guidance and advice in ice and cold operations

For future generations of ice classed ships to:

- Promote maritime safety
- Safeguard the marine environment




For more information, please contact:

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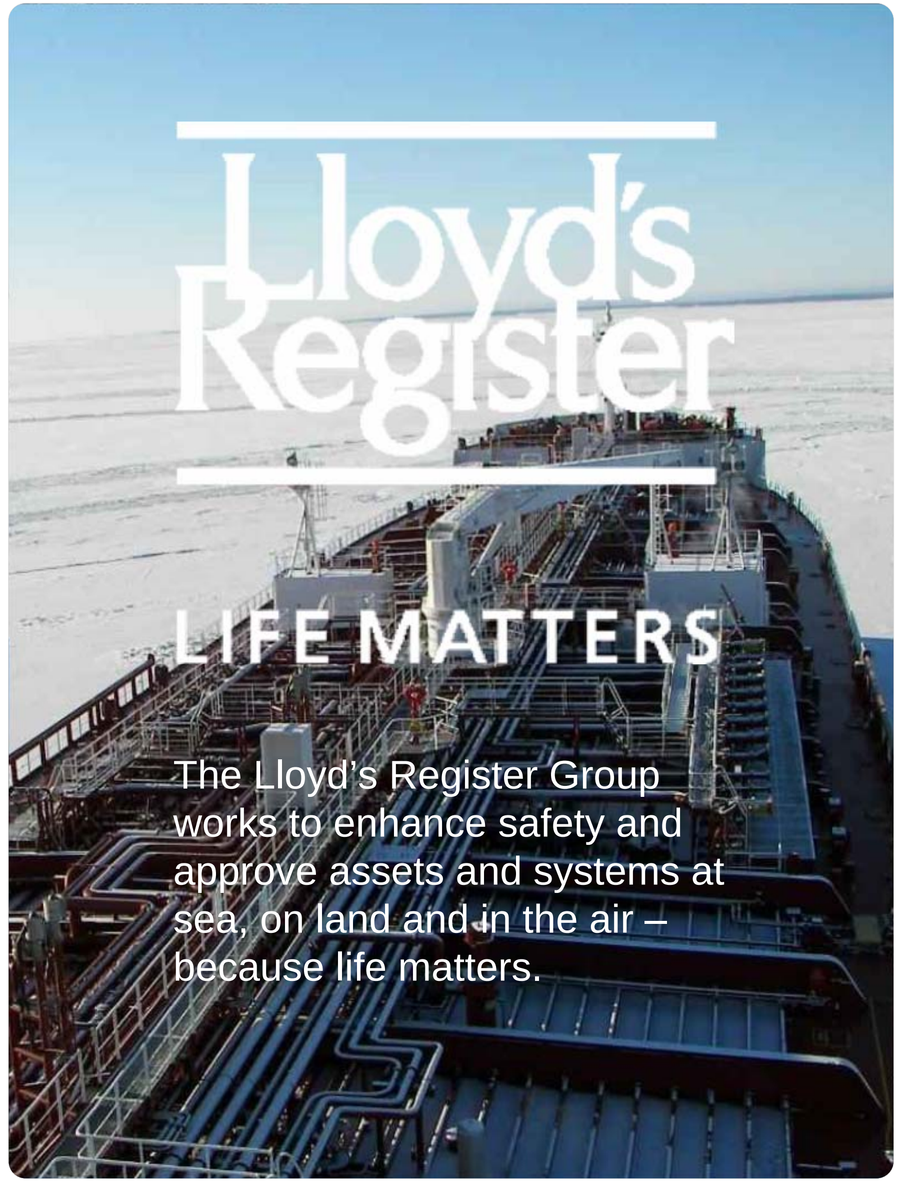
F +44 (0)20 7423 2061
E robert.bridges@lr.org

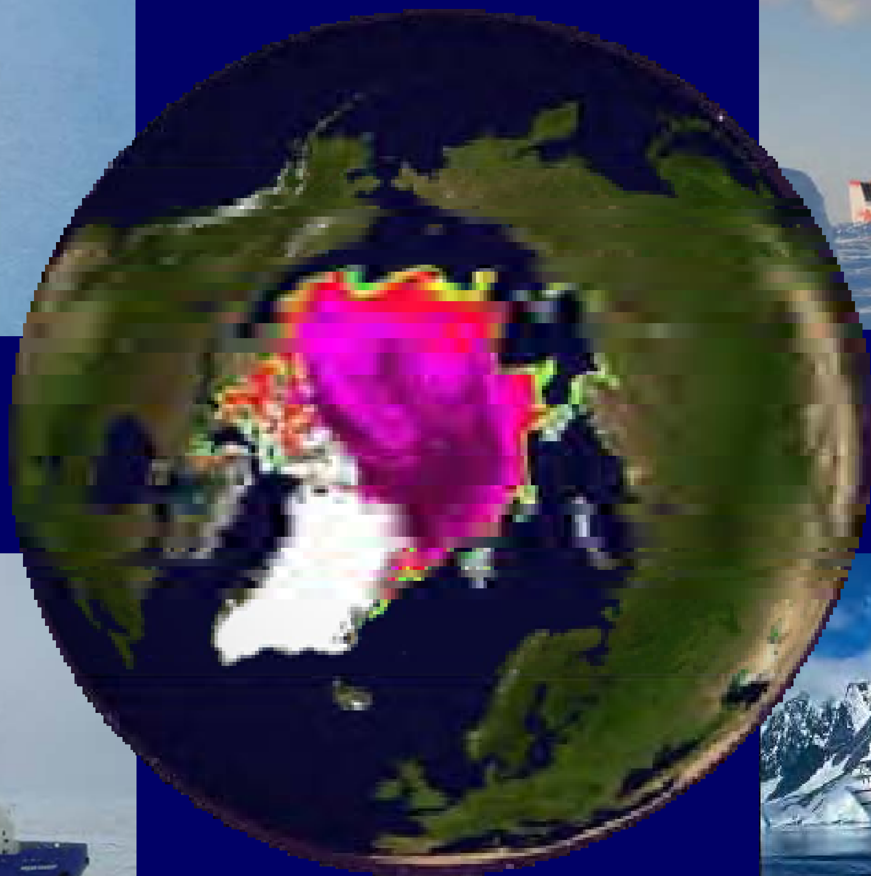




# Future Sea Ice Operating Conditions in the Arctic Ocean

## *Design & Construction of Vessels Operating in Low Temperature Environments*

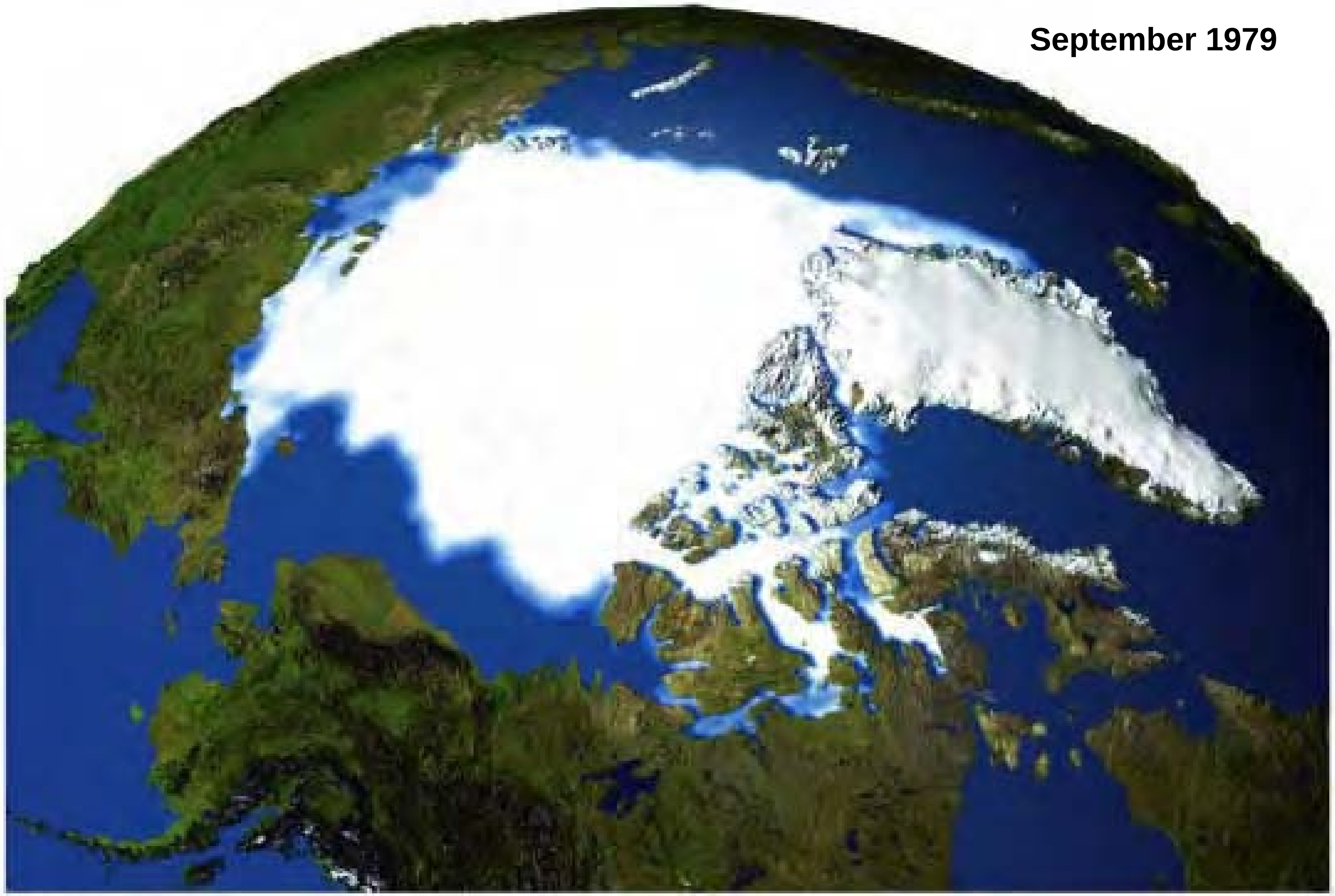
**RINA Headquarters, London**
**30-31 May 2007**

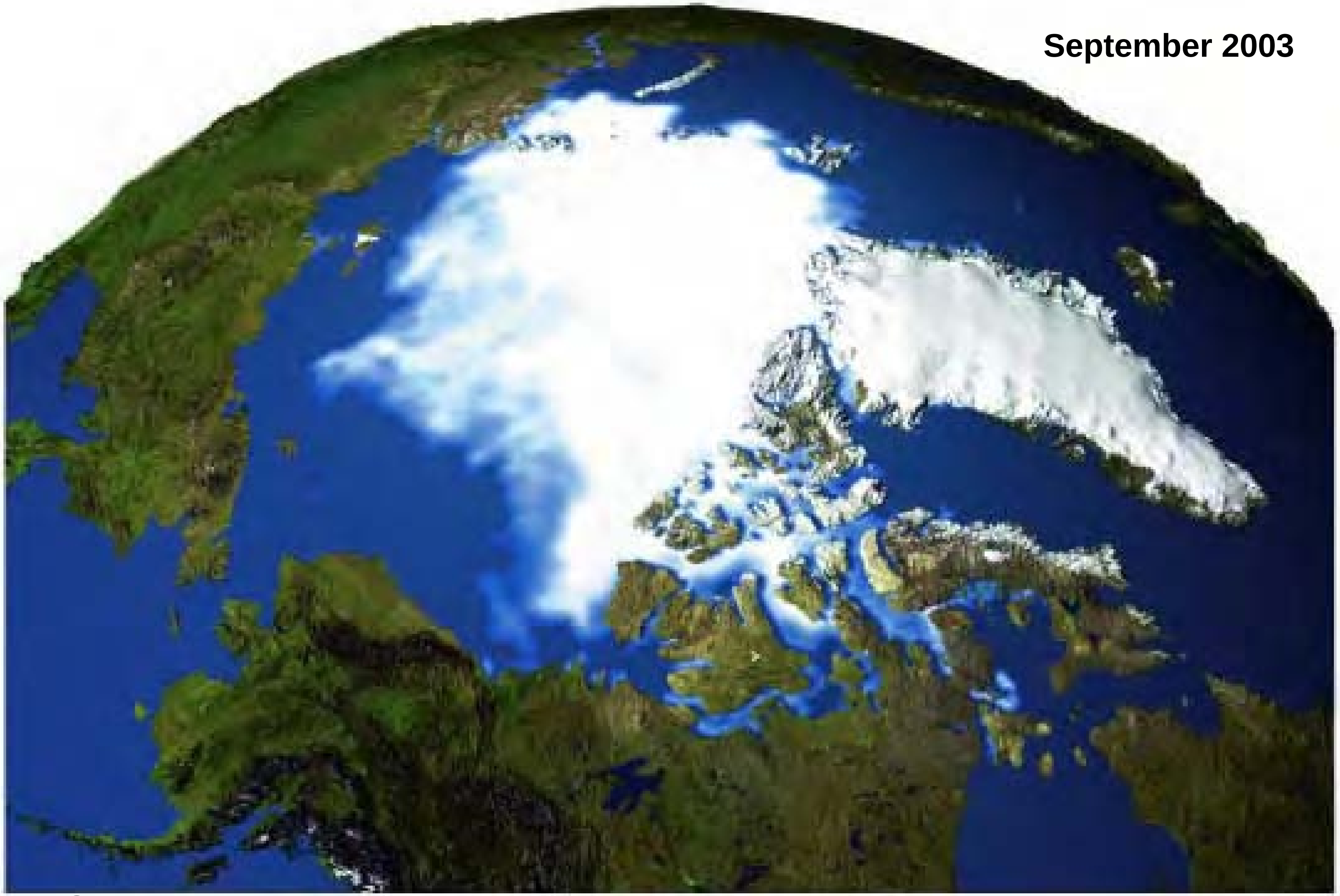

**Lawson W. Brigham, PhD**
**Vice Chair, PAME & Chair, AMSA**
**U.S. Arctic Research Commission ~ Anchorage**

# Presentation Outline

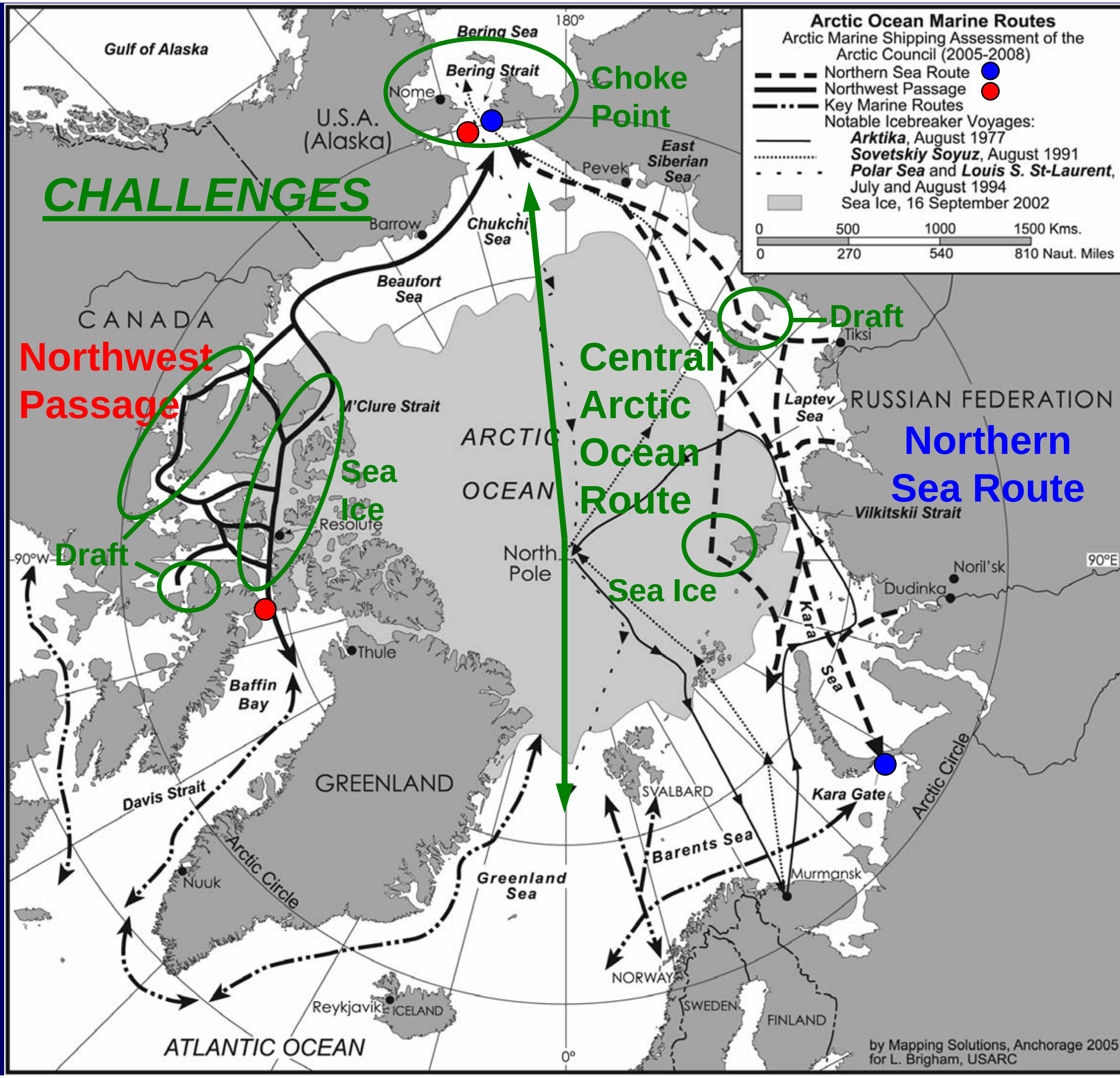
- Operational Challenges & Recent Sea Ice Coverage
- NSR & North Pole Voyages (1977-2006)
- Arctic Climate Impact Assessment
- Arctic Sea Ice Minimum Extents (2002-2006)
- Arctic Sea Ice Model Simulations
- Conclusions
- Future Arctic Ocean
- Arctic Marine Shipping Assessment of the Arctic Council



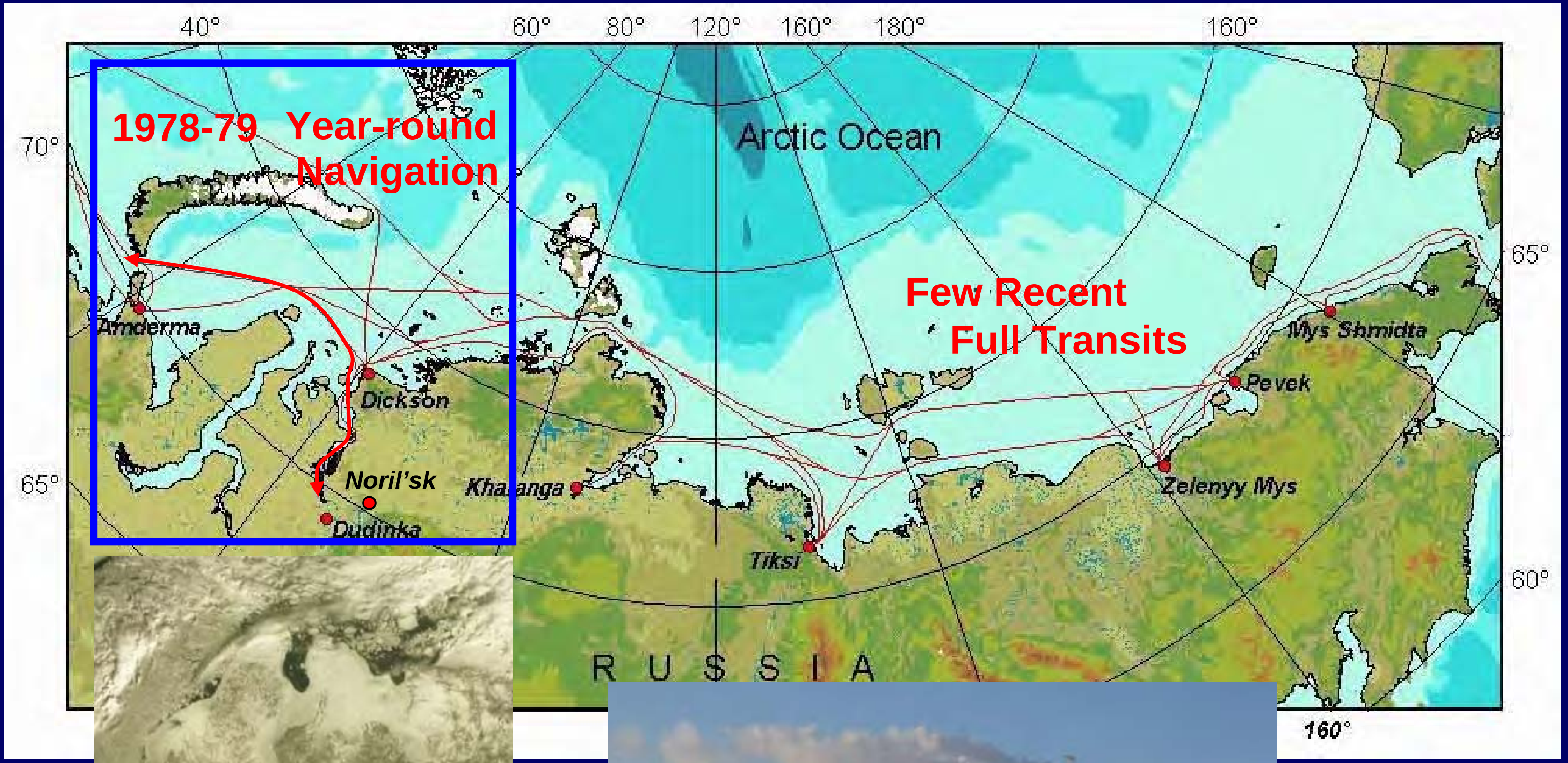

©NASA



NASA


CHALLENGES
Choke Point
Northwest Passage
Draft
Sea Ice
Central Arctic Ocean Route
Sea Ice
Draft
Northern Sea Route
Arctic Ocean Marine Routes
Arctic Marine Shipping Assessment of the Arctic Council (2005-2008)
Northern Sea Route
Northwest Passage
Key Marine Routes
Notable Icebreaker Voyages:
Arktika, August 1977
Sovetskiy Soyuz, August 1991
Polar Sea and Louis S. St-Laurent, July and August 1994
Sea Ice, 16 September 2002
0 500 1000 1500 Kms.
0 270 540 810 Naut. Miles
Gulf of Alaska
Bering Sea
Bering Strait
180°
Nome
U.S.A. (Alaska)
Pevek
East Siberian Sea
Barrow
Chukchi Sea
Beaufort Sea
CANADA
M'Clure Strait
ARCTIC OCEAN
North Pole
Resolute
Tiksi
Laptev Sea
RUSSIAN FEDERATION
Vilkitskii Strait
90°W
90°E
Noril'sk
Dudinka
Thule
Kara Sea
Baffin Bay
Davis Strait
GREENLAND
SVALBARD
Kara Gate
Arctic Circle
Barents Sea
Murmansk
Nuuk
Arctic Circle
Greenland Sea
NORWAY
SWEDEN
FINLAND
Reykjavik
ICELAND
ATLANTIC OCEAN
0°
by Mapping Solutions, Anchorage 2005 for L. Brigham, USARC

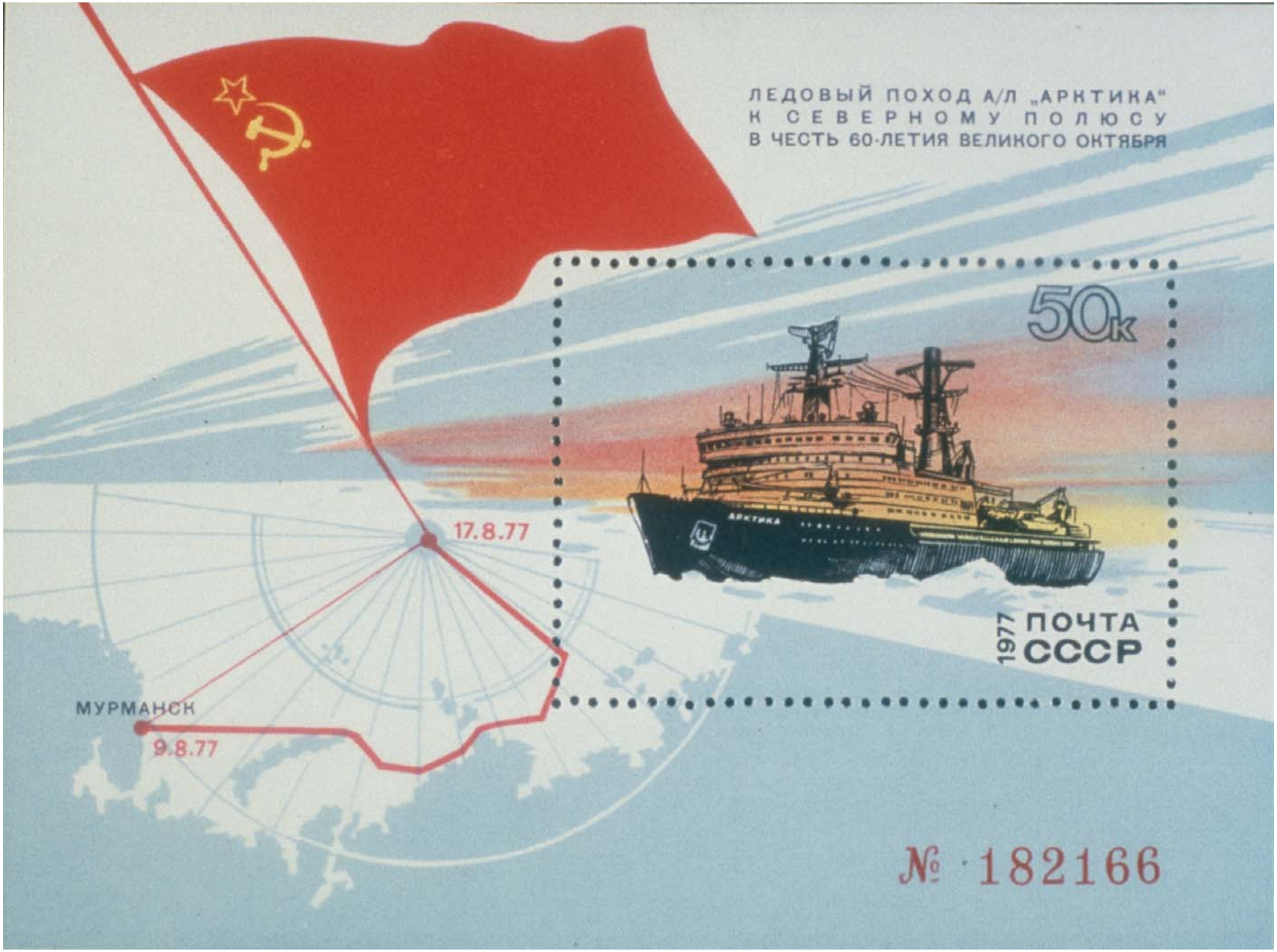
# The Northern Sea Route

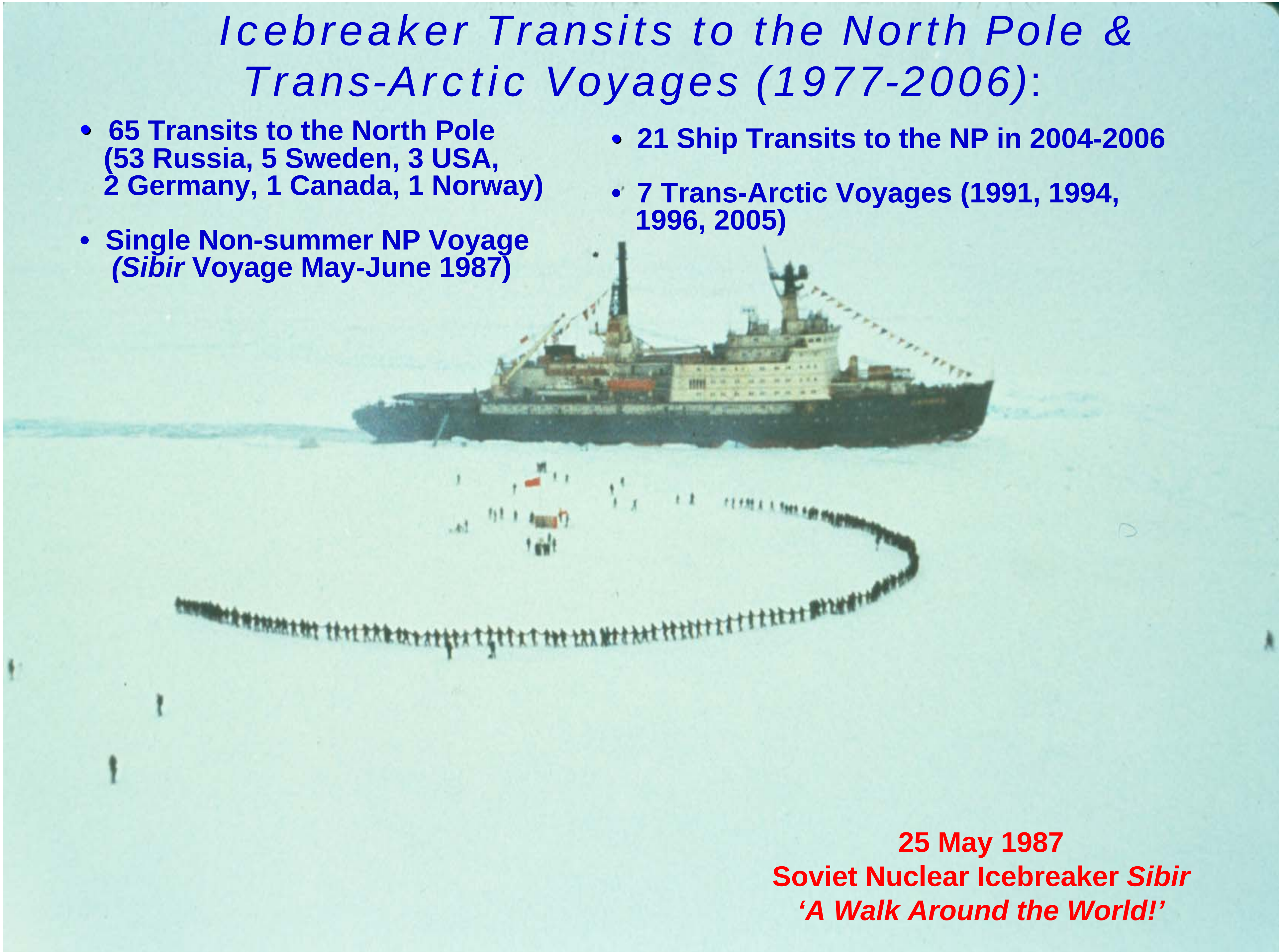


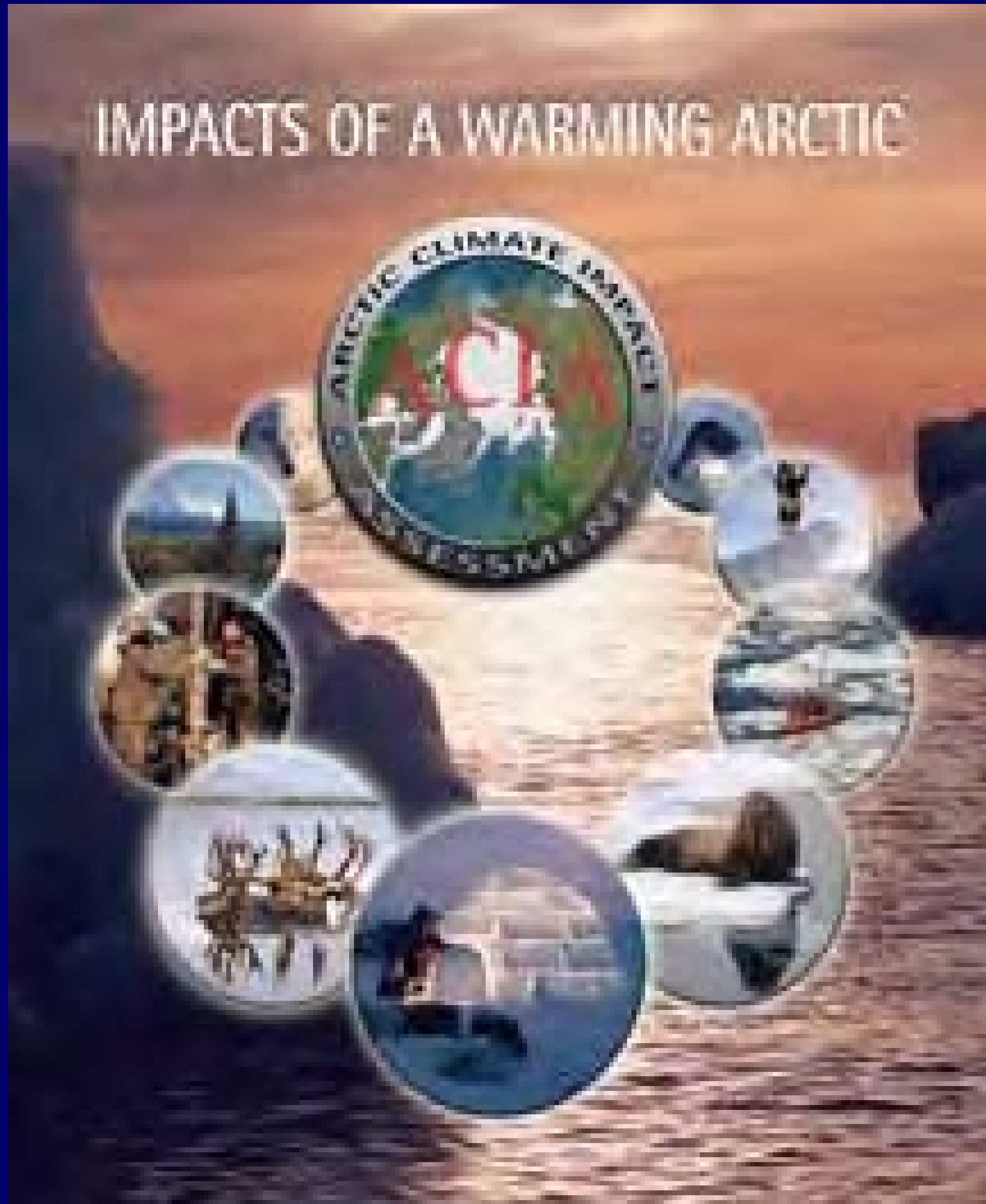



INSROP (1999)


Future Convoy Requirements?
Aker Arctic


ЛЕДОВЫЙ ПОХОД А/Л „АРКТИКА"
К СЕВЕРНОМУ ПОЛЮСУ
В ЧЕСТЬ 60-ЛЕТИЯ ВЕЛИКОГО ОКТЯБРЯ
50к
АРКТИКА
1977 ПОЧТА СССР
17.8.77
МУРМАНСК
9.8.77
№ 182166


Icebreaker Transits to the North Pole & Trans-Arctic Voyages (1977-2006):
• 65 Transits to the North Pole (53 Russia, 5 Sweden, 3 USA, 2 Germany, 1 Canada, 1 Norway)
• Single Non-summer NP Voyage (*Sibir* Voyage May-June 1987)
• 21 Ship Transits to the NP in 2004-2006
• 7 Trans-Arctic Voyages (1991, 1994, 1996, 2005)
25 May 1987
Soviet Nuclear Icebreaker *Sibir*
*'A Walk Around the World!'*


IMPACTS OF A WARMING ARCTIC
ARCTIC CLIMATE IMPACT ASSESSMENT
ACIA

www.amap.no


The Arctic is a Preview
of Earth's Future Climate
10 Years of Change in Arctic =
25 Years in Rest of the world.

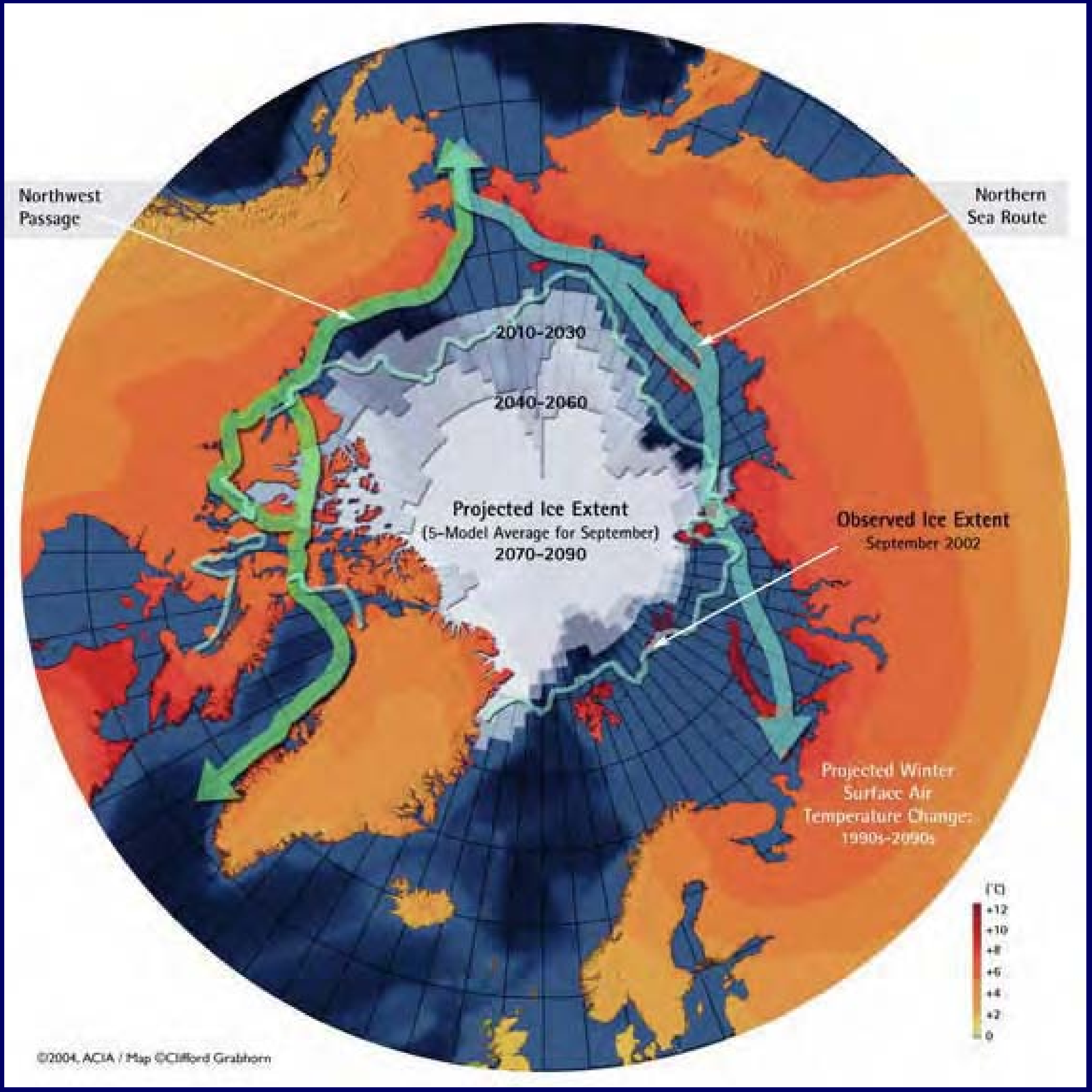
Arctic Climate Impact Assessment

# Arctic Sea Ice Transformations Significant to Marine Transport

- Extent: ~3% decrease per decade
- Multiyear Ice/Perennial Pack Ice: ~7% decrease per decade
- Thickness: 14 to 32% reductions reported
- General increase in the length of the ice melt season

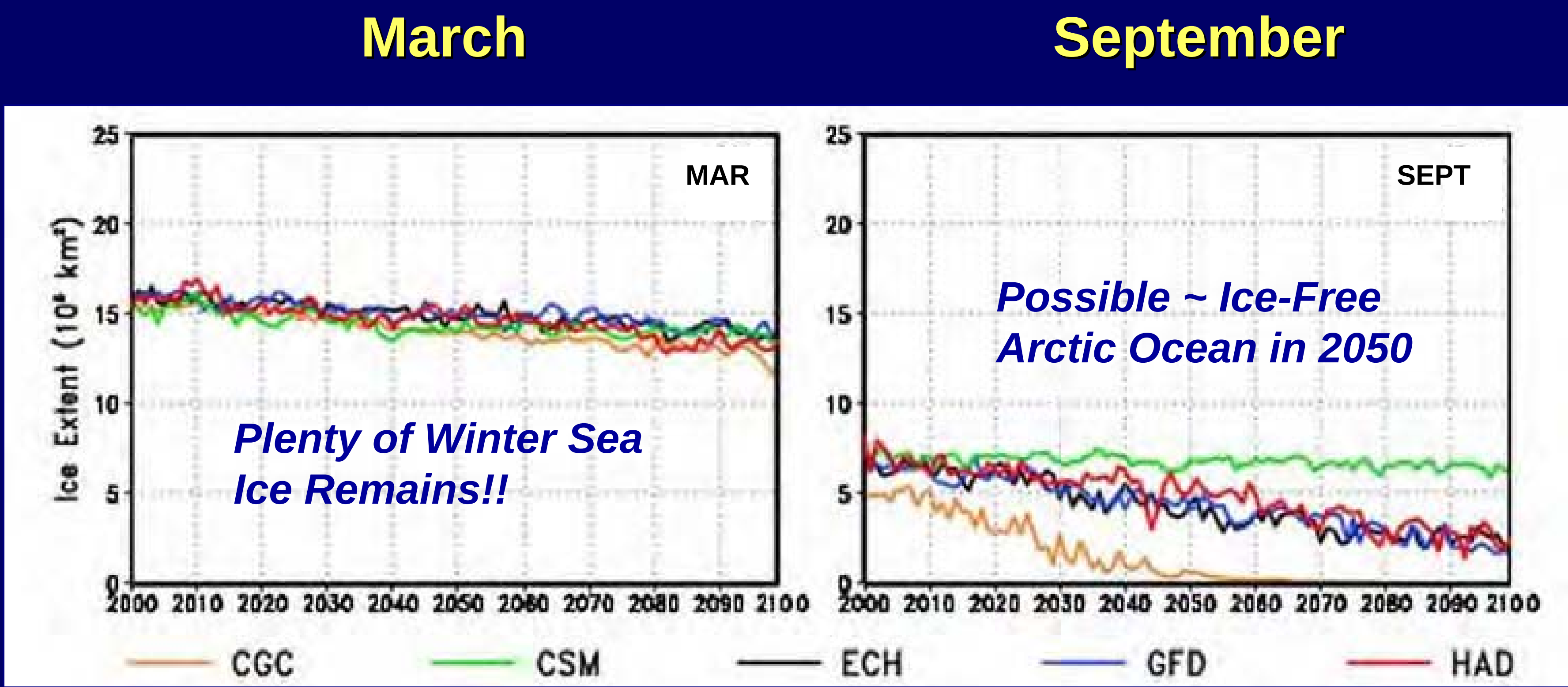
# Arctic Climate Impact Assessment

**Key Finding #6: "Reduced sea ice is very likely to increase marine transport and access to resources."**



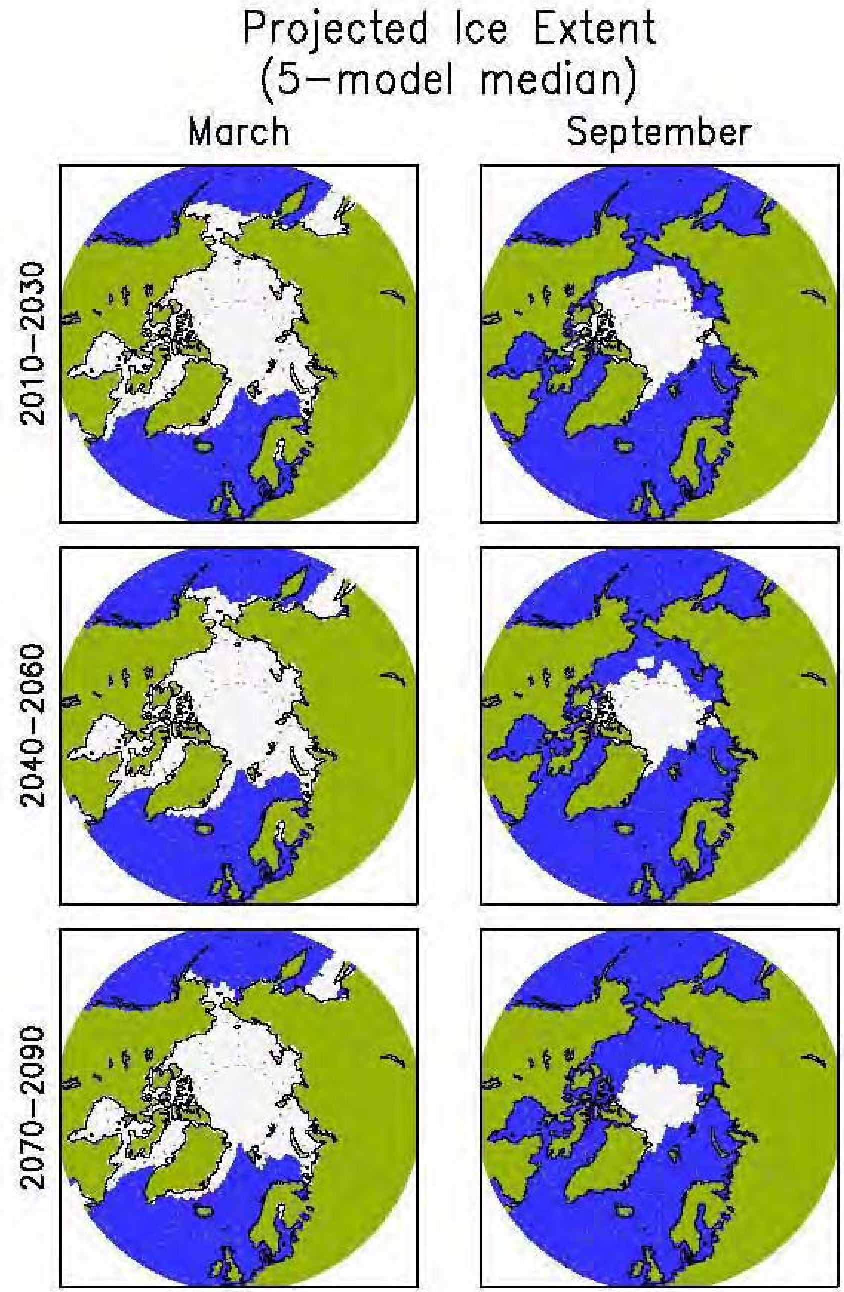
# Arctic Climate Impact Assessment

## Climate model projections of sea ice extent: 2000 - 2100




**B2 IPCC Moderate Global Scenario**

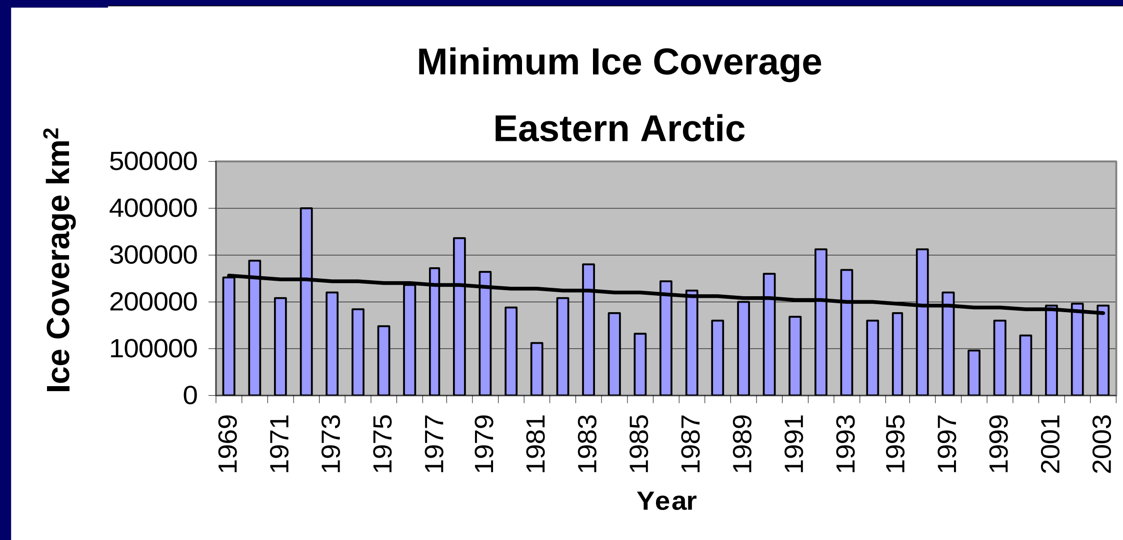
# Arctic Climate Impact Assessment



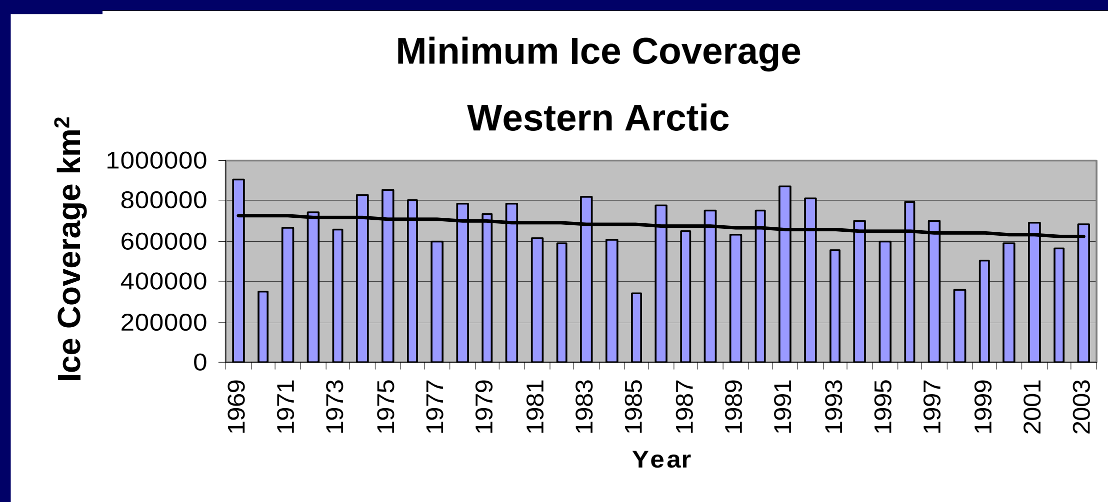
# ACIA and the Northwest Passage

- Loss of Sea Ice Coverage
- Large Inter-annual Variability

## *Regional Eastern Arctic*




## *Regional Western Arctic*




**Canadian Ice Service (2004)**

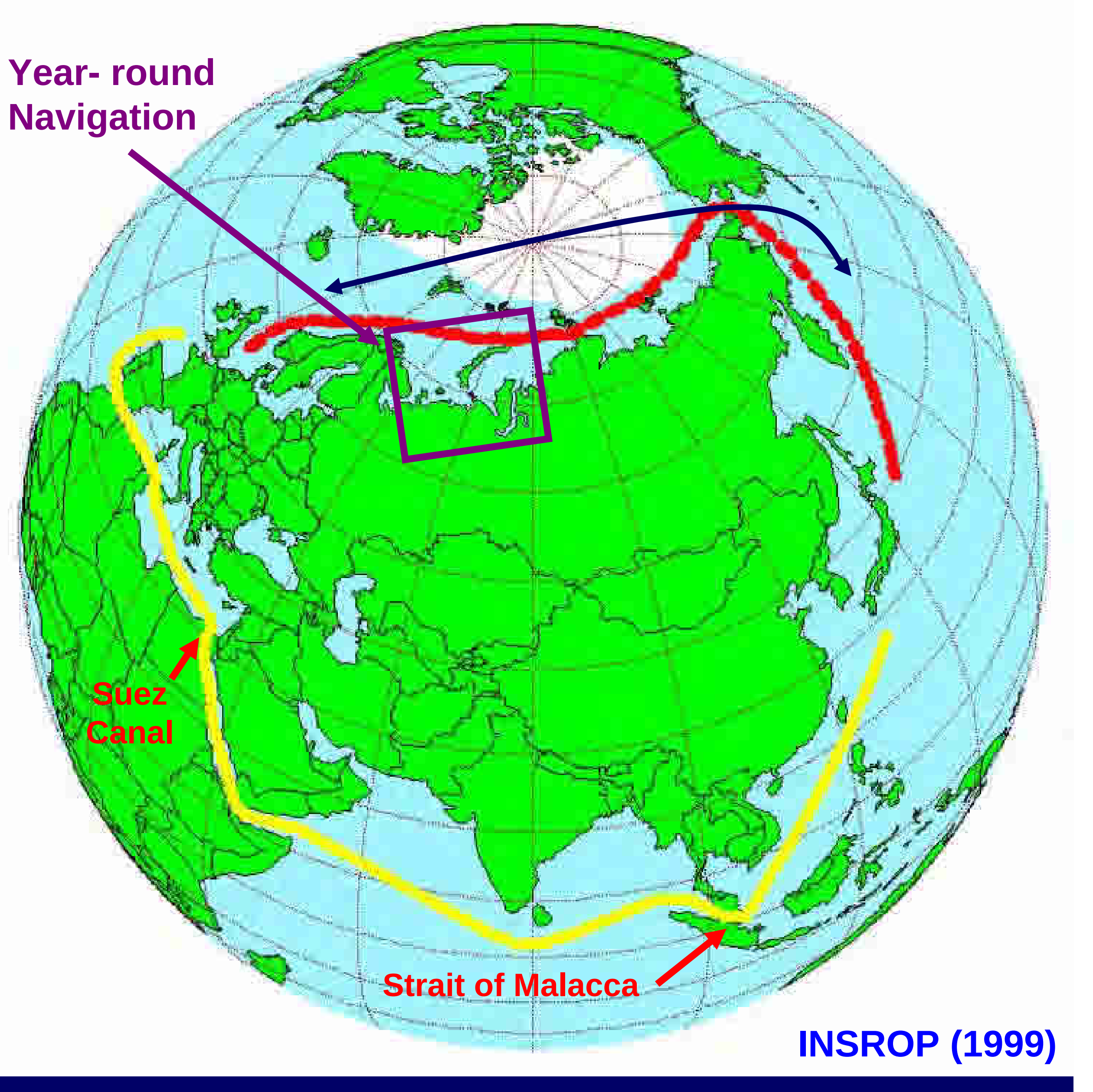
**Distance (Nautical Miles)**
**Hamburg to Yokohama**

**Northern Sea Route ~ 6,920**

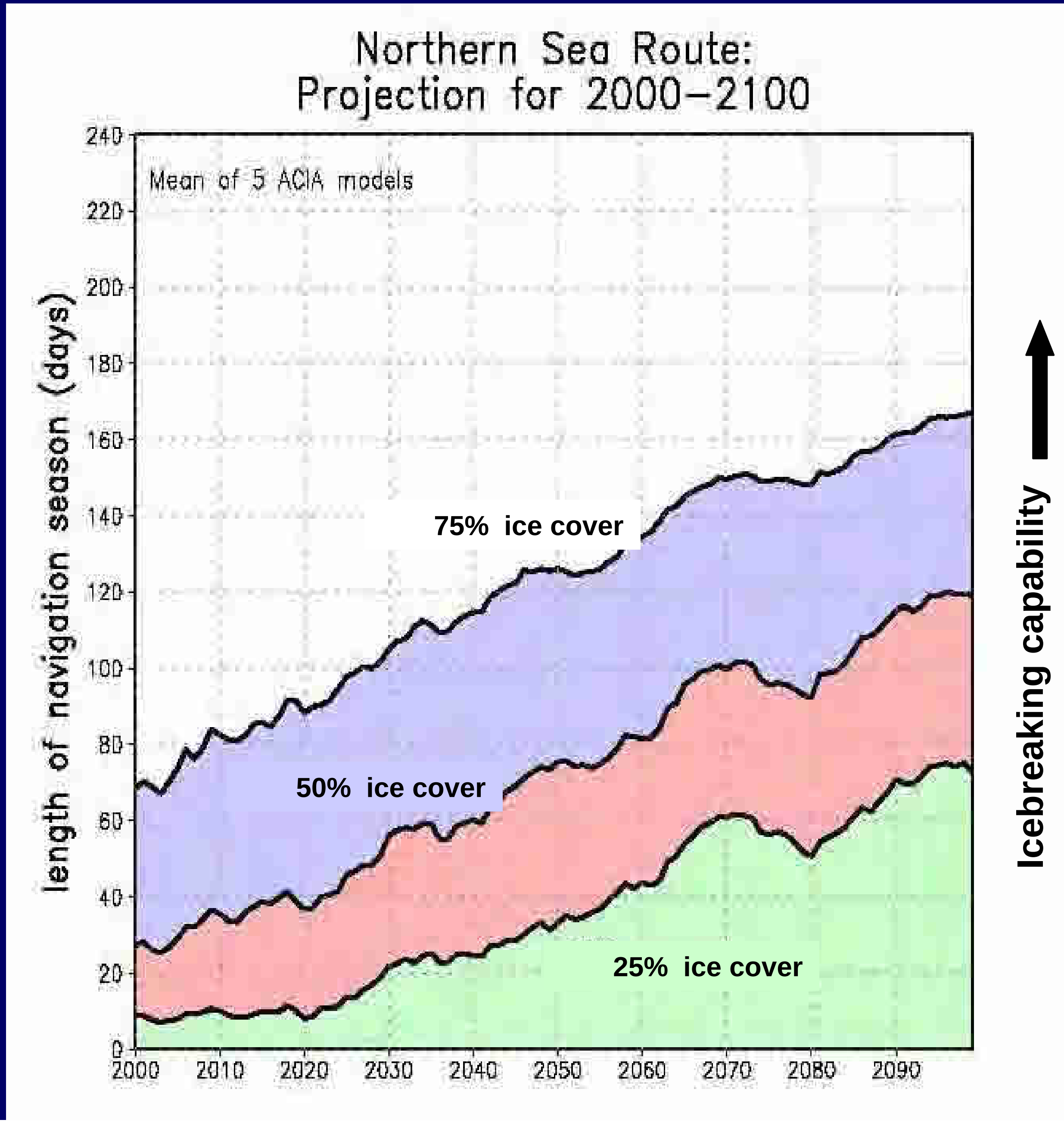
**Suez Canal ~ 11,073**

**Panama Canal ~ 12,420**

**Cape of Good Hope ~ 14,542**



# Arctic Climate Impact Assessment

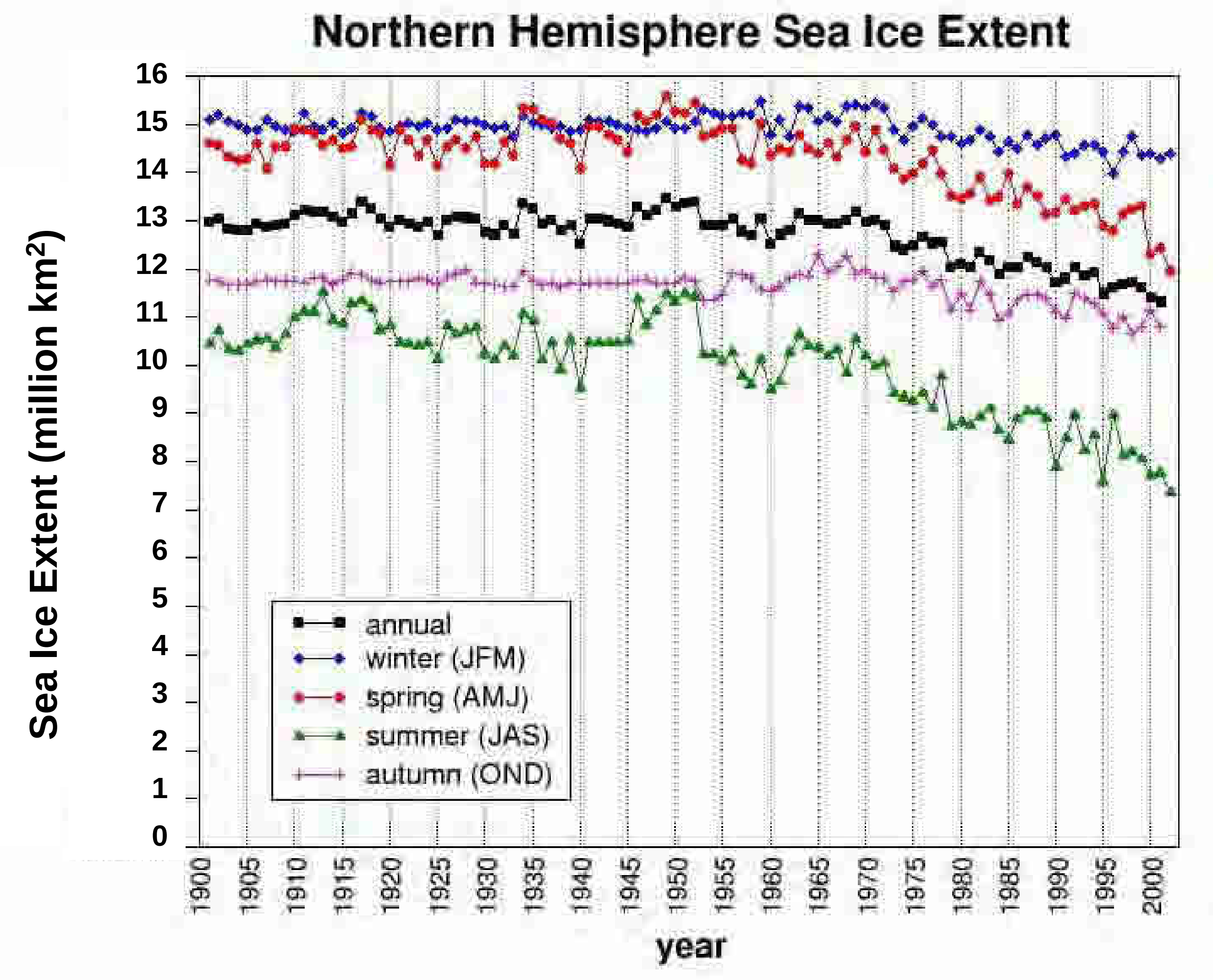


# Sea Ice

## Arctic Climate Impact Assessment

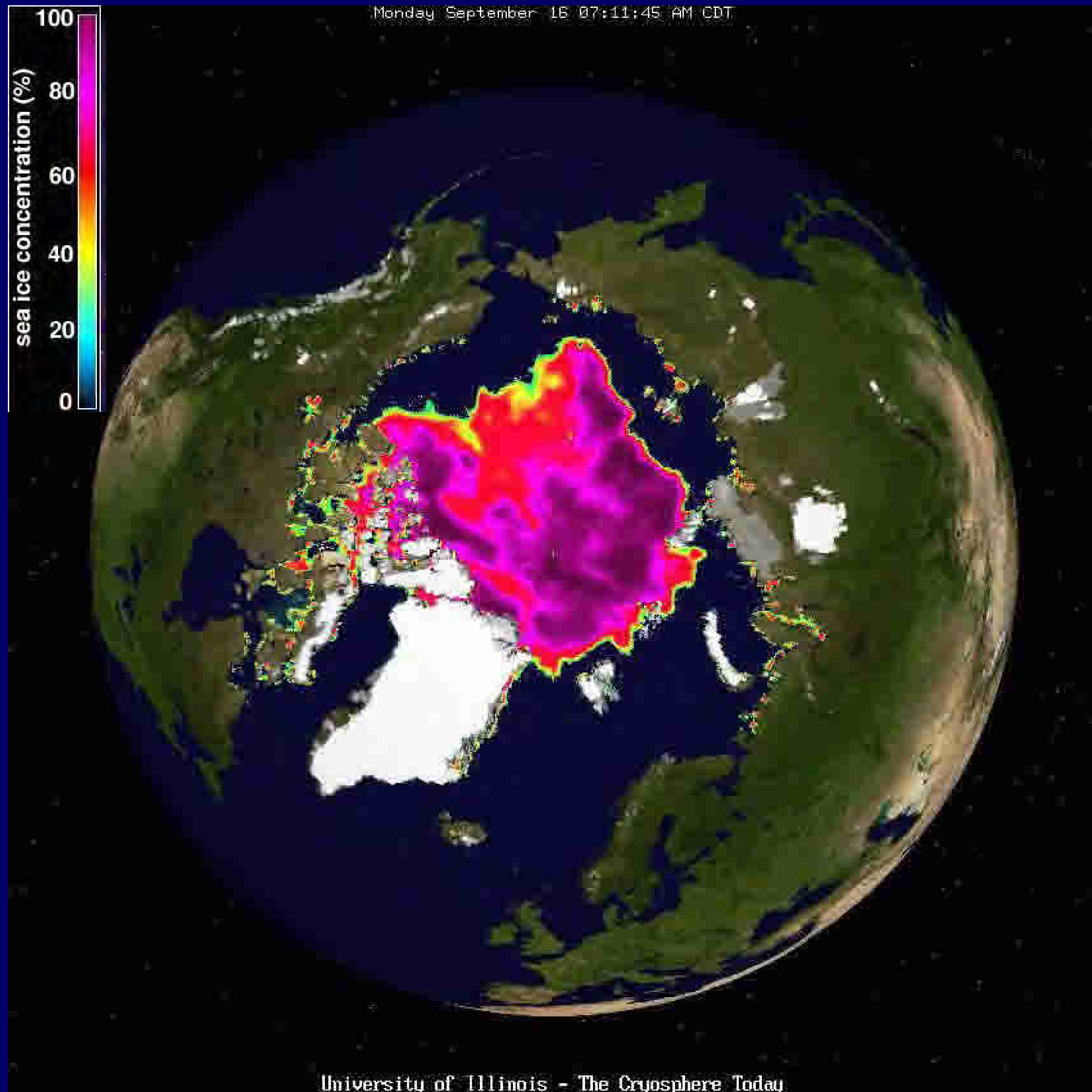
### Observational data show a decrease of coverage

- Extent decrease is largest in summer
- Extent decrease is largest since late 1980s
- Extent seasonal decreases since 1950s

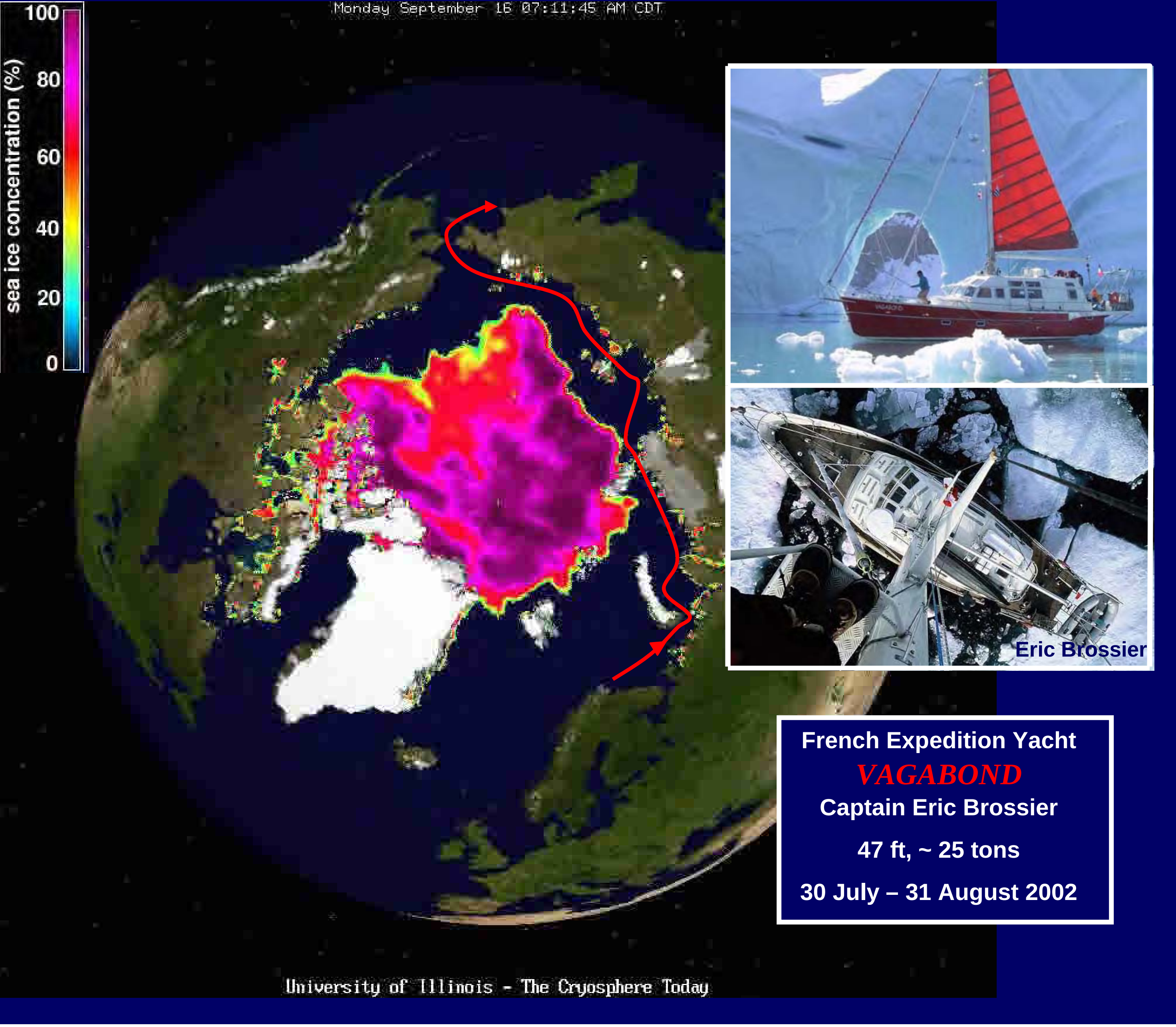



Chapman & Walsh (2003)

# 16 September 2002



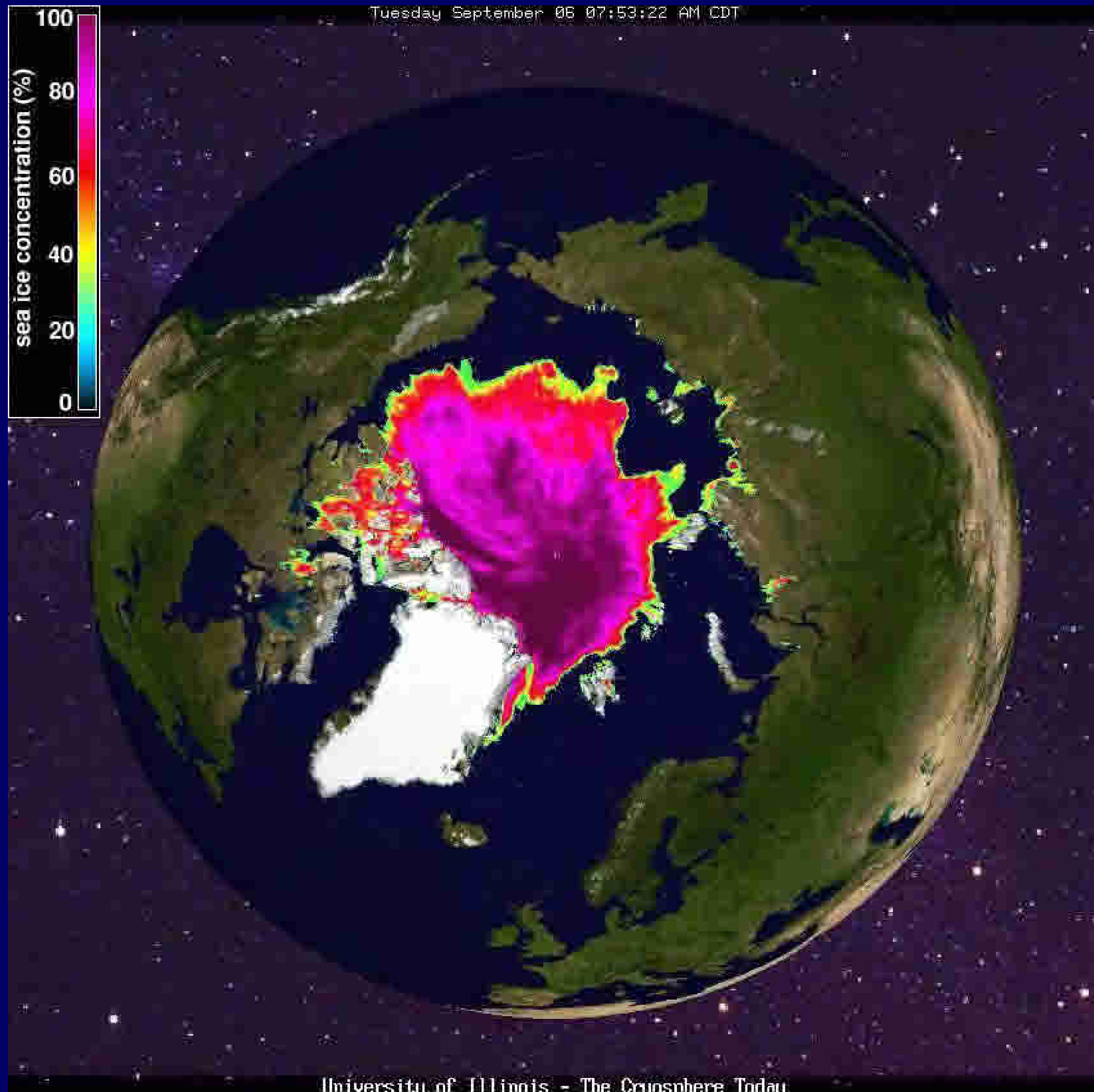
# 16 September 2002



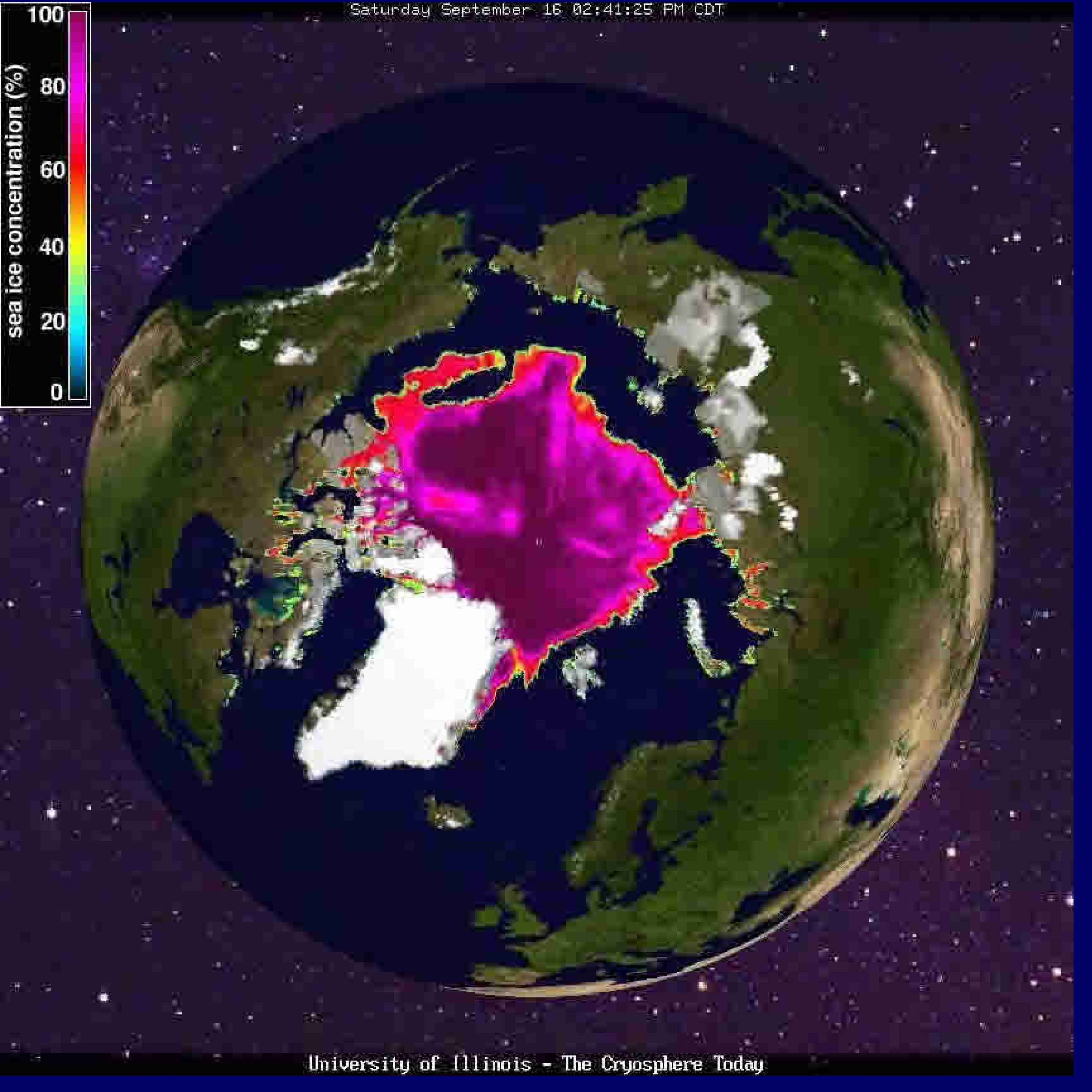

French Expedition Yacht
*VAGABOND*
Captain Eric Brossier
47 ft, ~ 25 tons
30 July – 31 August 2002

# 6 September 2005

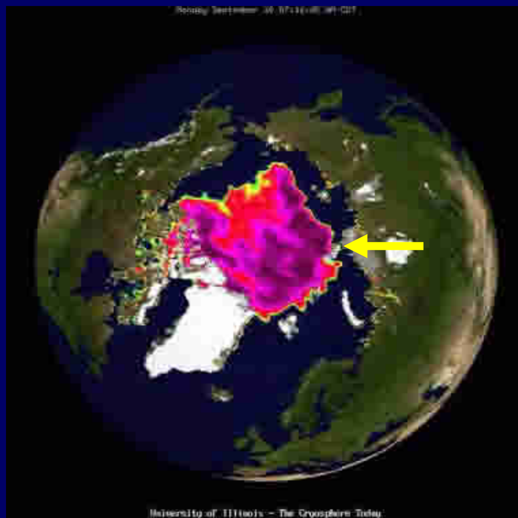
## Historic Minimum Arctic Sea Ice Extent



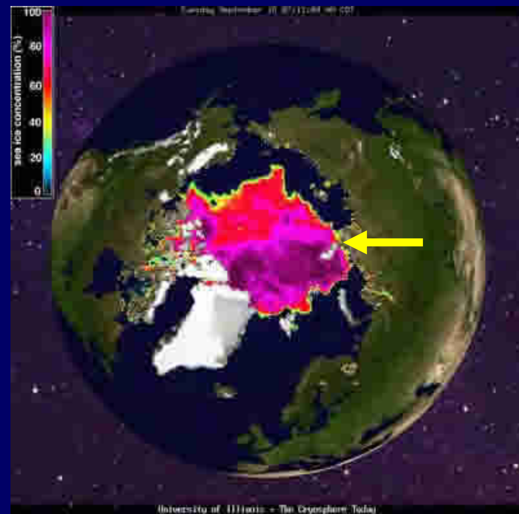
# 16 September 2006



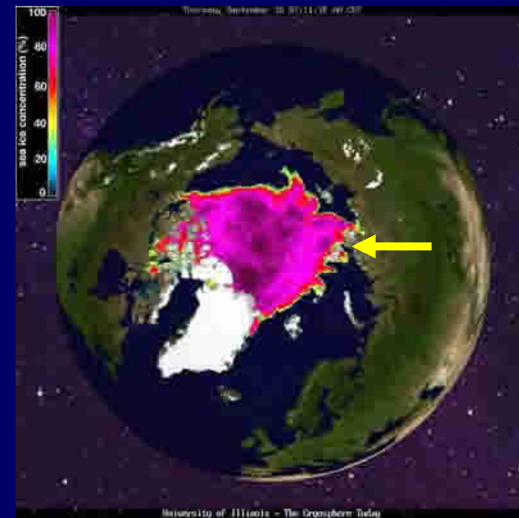
16 September 2002

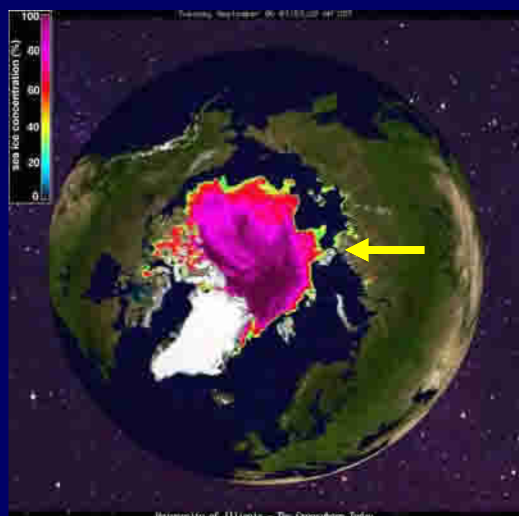


16 September 2003

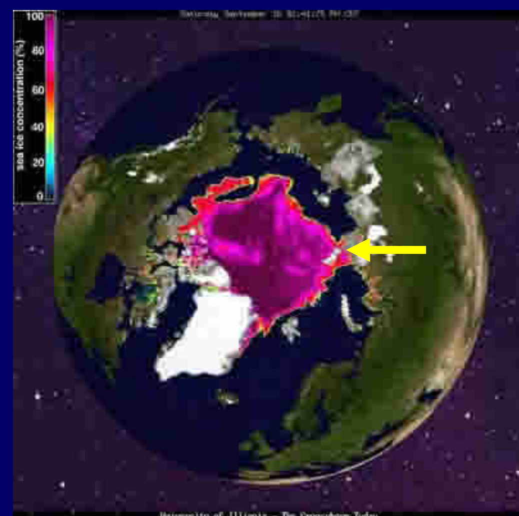


16 September 2004



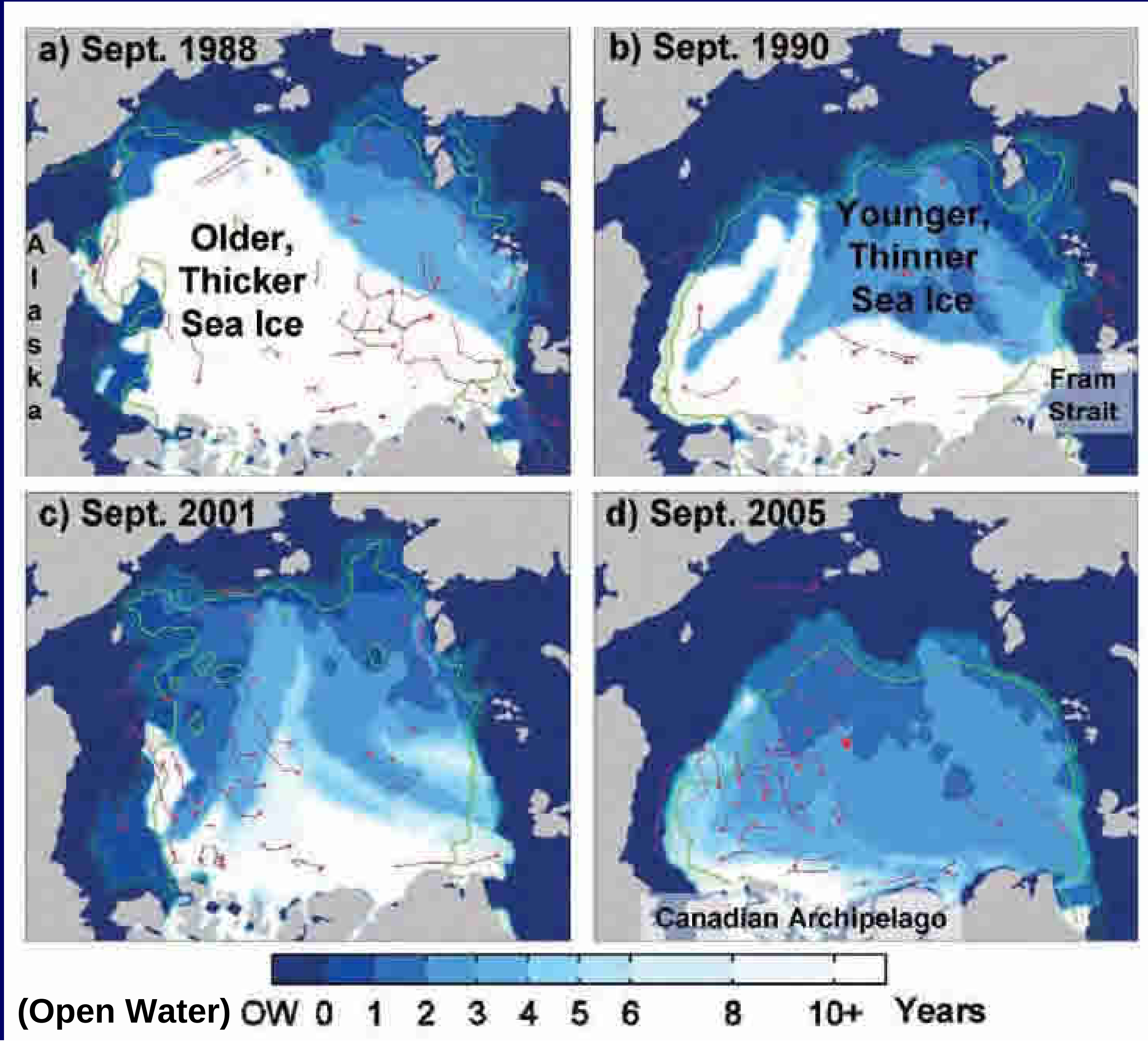


6 September 2005

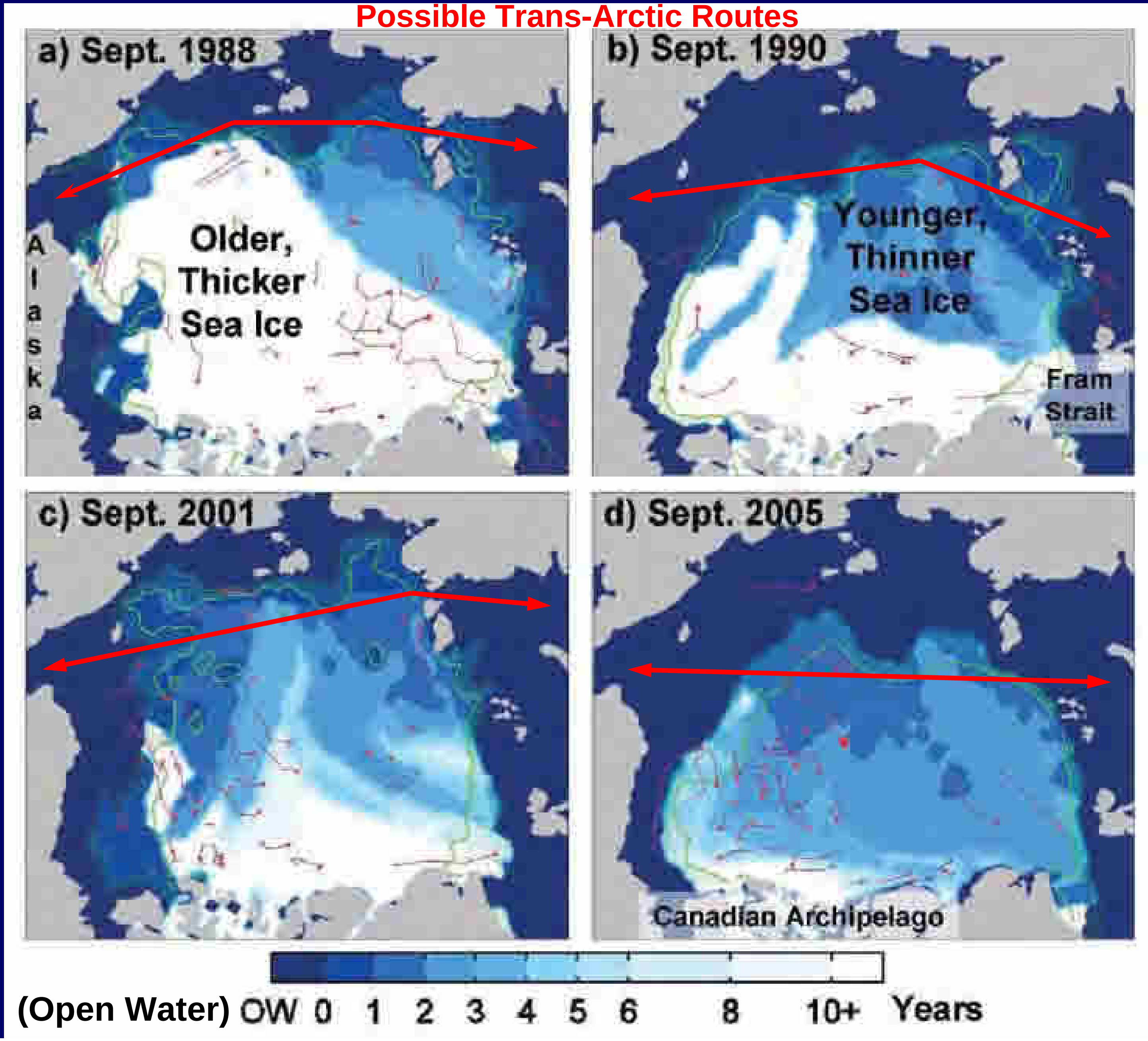


16 September 2006

# Changing Nature of Multi-year Arctic Sea Ice



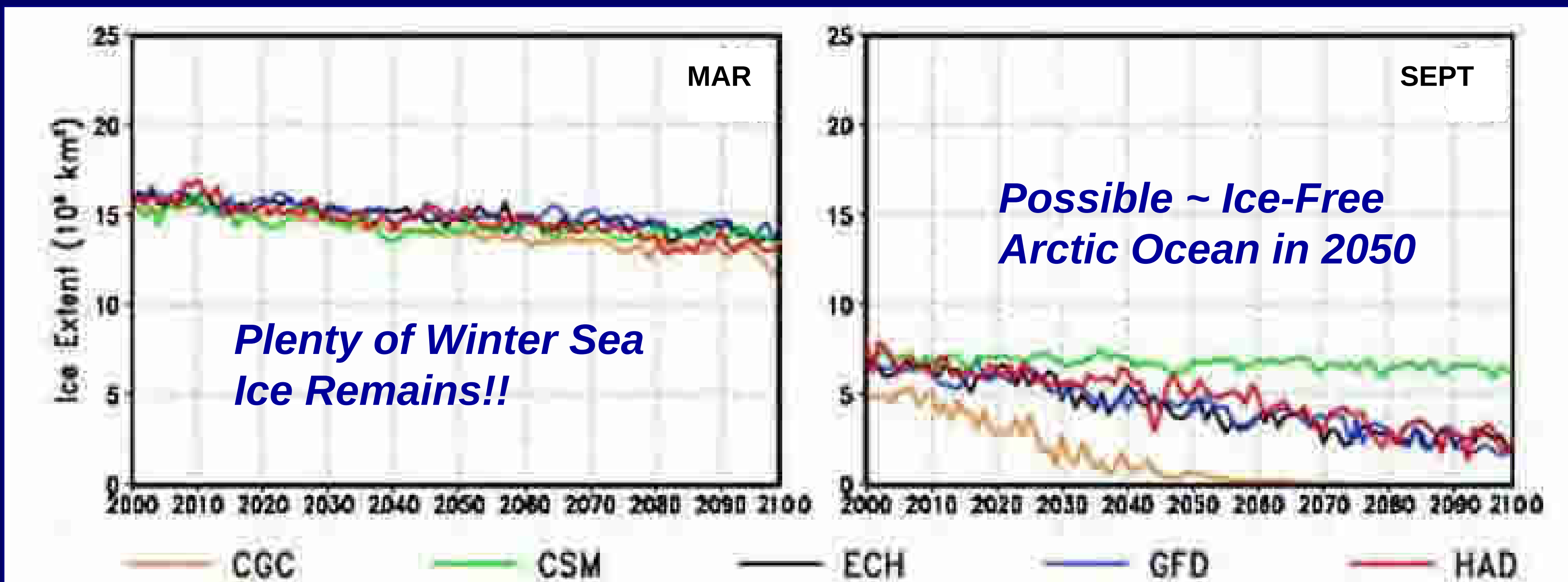
# Changing Nature of Multi-year Arctic Sea Ice



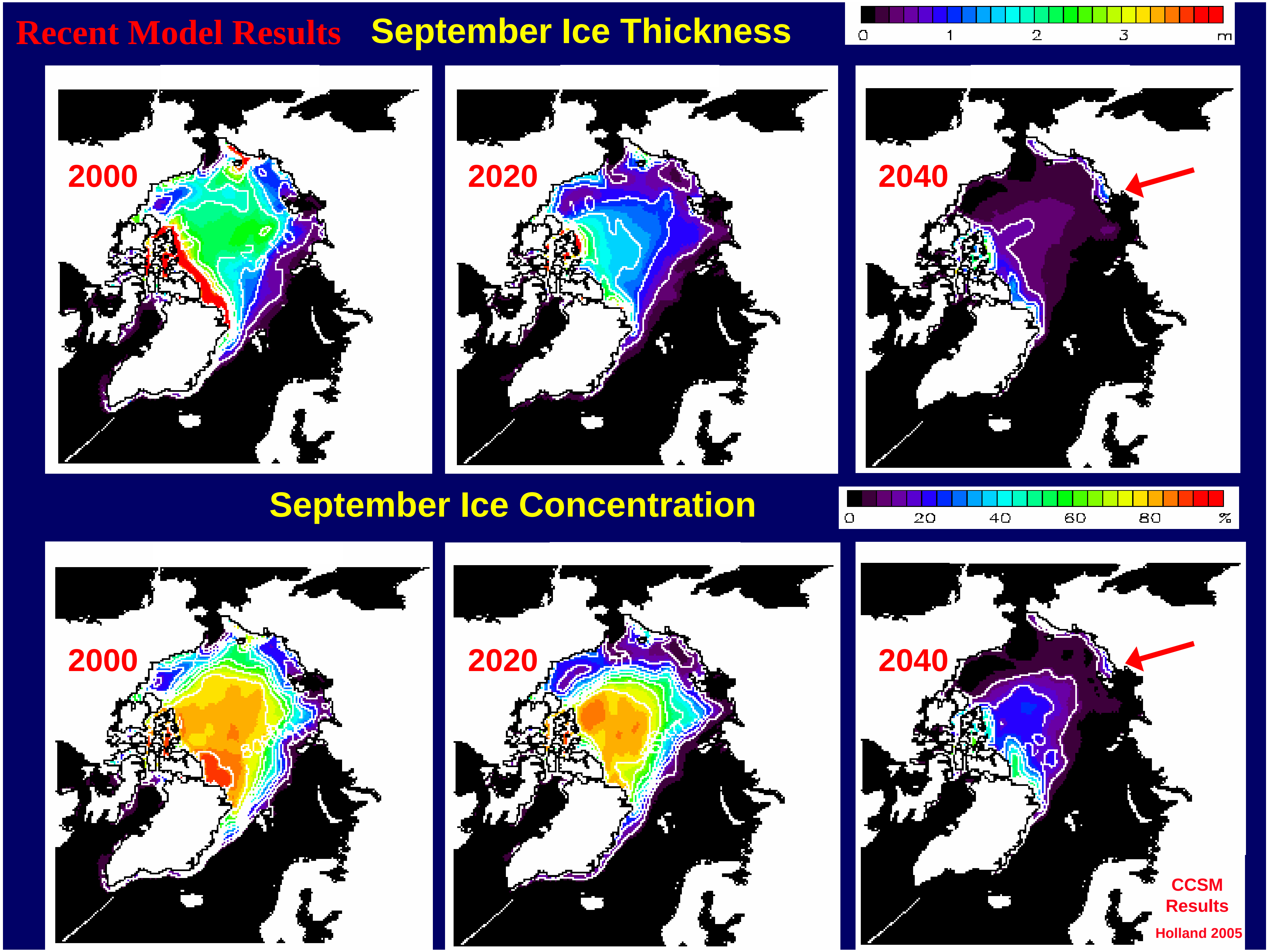
# Arctic Climate Impact Assessment

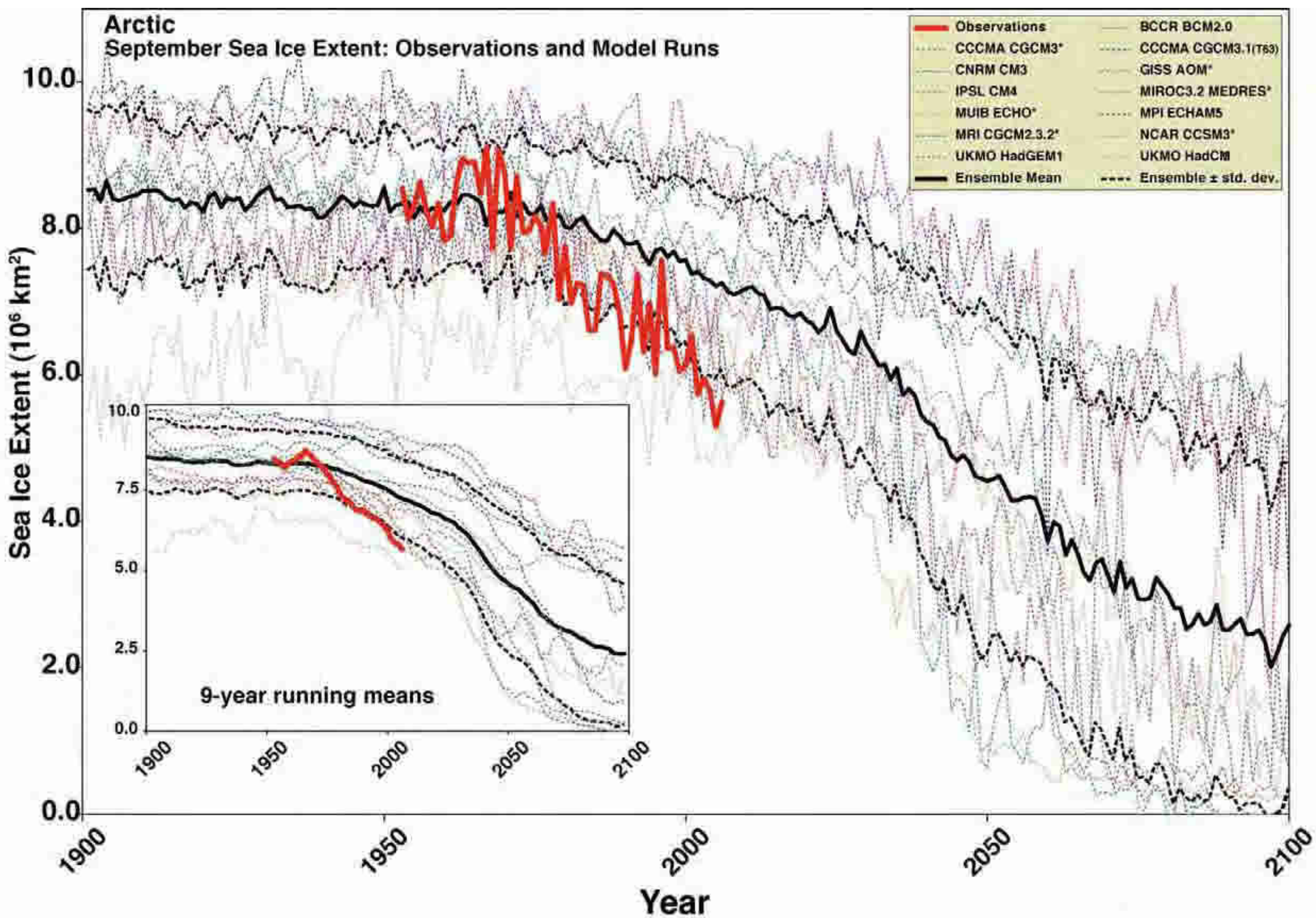
## Climate model projections of sea ice extent: 2000 - 2100

**March** **September**



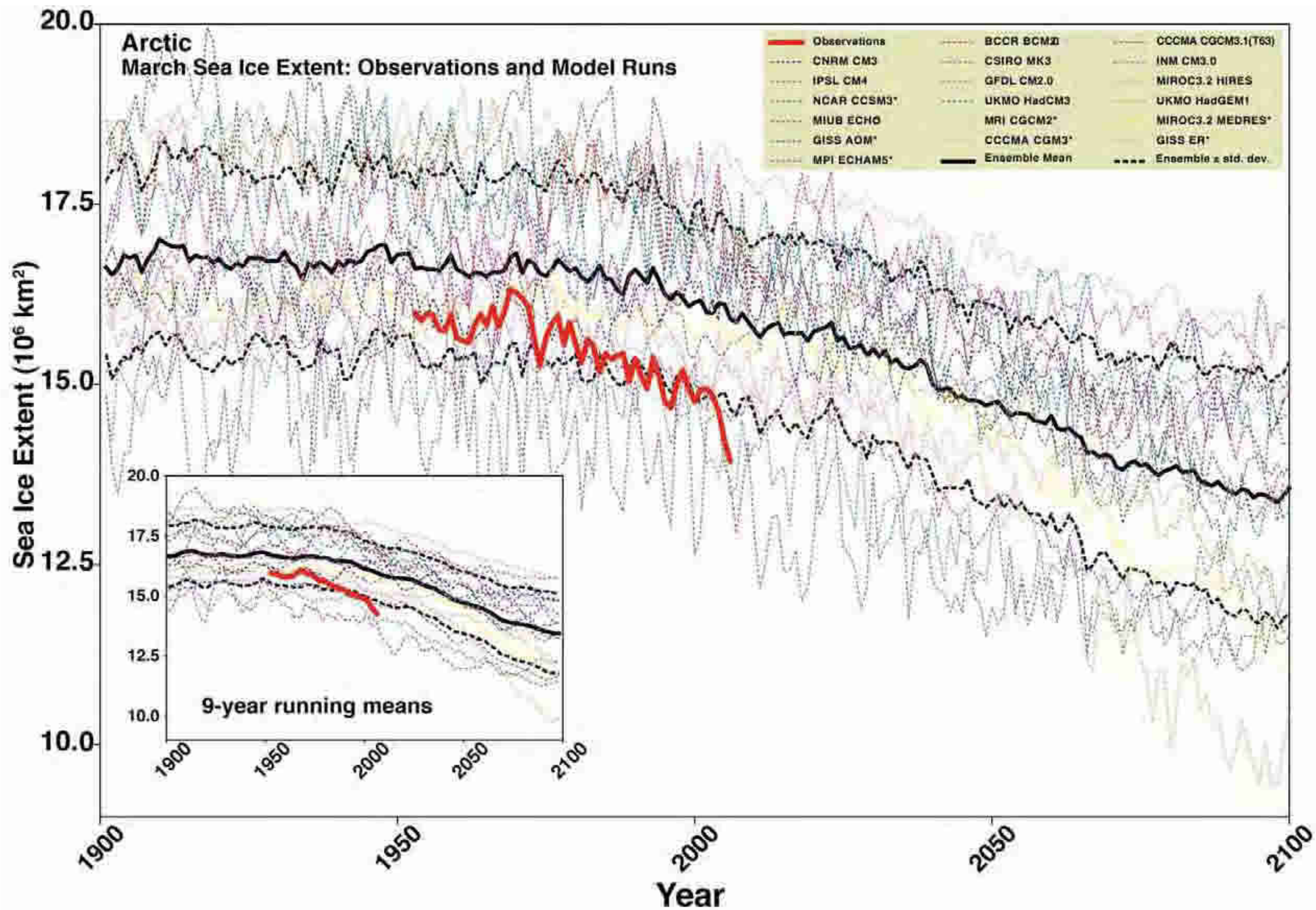

**B2 IPCC Moderate Global Scenario**


Recent Model Results
September Ice Thickness
0
1
2
3
m
2000
2020
2040
September Ice Concentration
0
20
40
60
80
%
2000
2020
2040
CCSM
Results
Holland 2005




**September Arctic Sea Ice Extent ~** Stroeve et al. Fig 1

***Arctic sea ice decline: Faster than forecast*** (GRL, 1 May 2007)




**March Arctic Sea Ice Extent ~** Stoeve et al., Fig 2

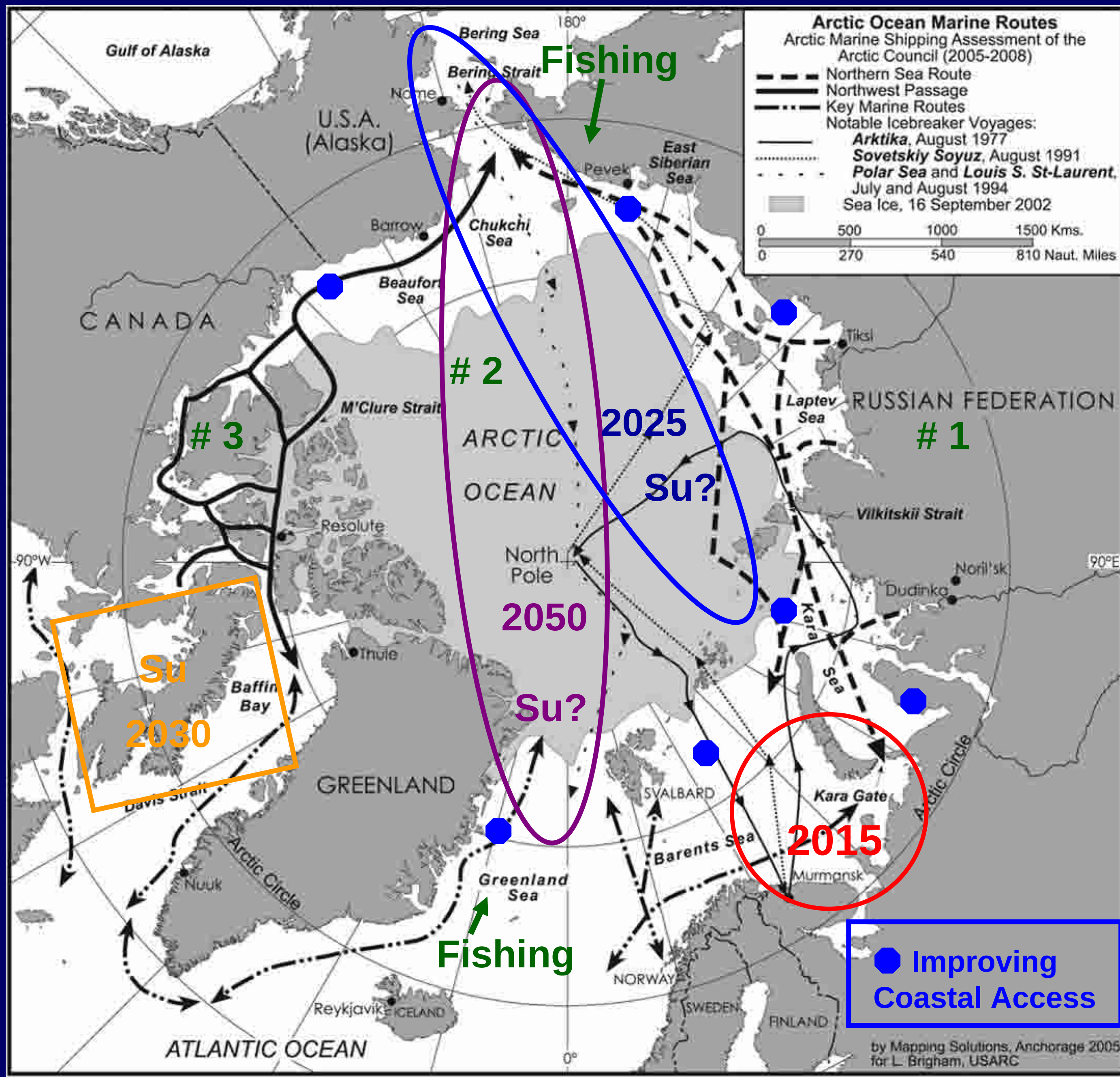
***Arctic sea ice decline: Faster than forecast*** (GRL, 1 May 2007)

# Conclusions

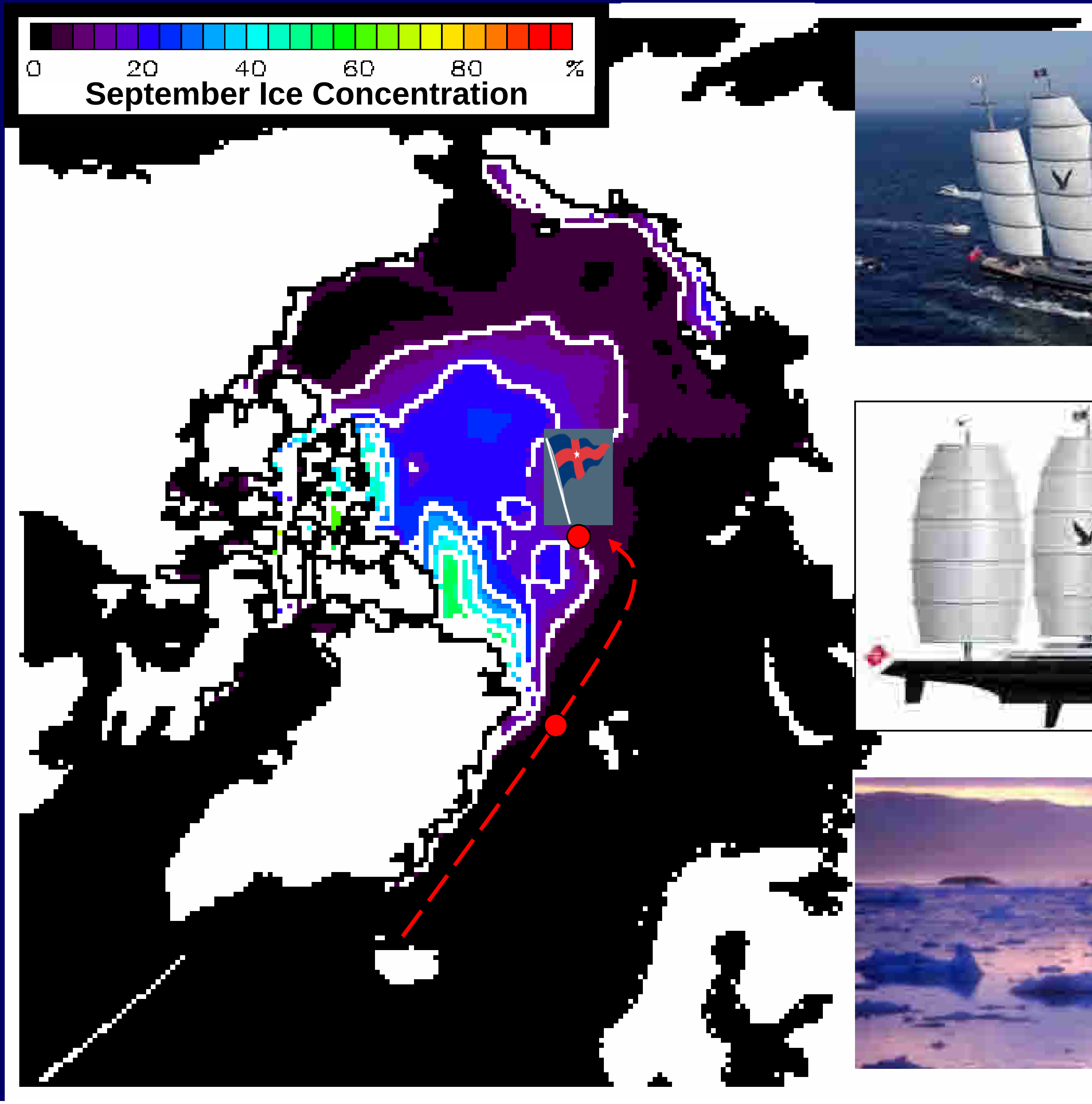
- **Diminishing Arctic Sea Ice ~ Extent & Thickness**
  - **Arctic coast regions increasingly ice-free (longer ice-free summers and autumns)**
  - **Continued winter sea ice extent**
- **GCM Simulations ~ Indicate Diminishing Arctic Sea Ice**
  - **IPCC AR4 models *cannot* replicate recent observed trends**
- **GCMs ~ Resolution Not Adequate for Sea Ice Simulations of the Arctic's Complex Geography**
- **Canadian Arctic ~ Large Inter-annual Sea Ice Variability in the Recent Observed Record**
- **Russian Arctic ~ Plausible Longer Seasons of Navigation**

- Plausible  ~ Ice-free Arctic Ocean in 2040 (or *Earlier*)
    - Possible ‘window for navigation’
    - End of MY Arctic sea ice
- Plausible ~ More Mobile Sea Ice (Increased Ridging?)
- Requirements:
    - Regional sea ice models verses GCMs
    - Enhanced real-time sea ice obs including ice thickness
- *Key*:  Uncertain Future Operating Conditions, Yet Greater Marine Access & Longer Seasons of Navigation
- *Key*:  Recent Arctic Sea Ice Trends Have Significant Implications for Design, Construction & Operational Standards

# The Maritime Arctic of the Future?



# Sailing Cruise to the North Pole in 2035-2040 ?





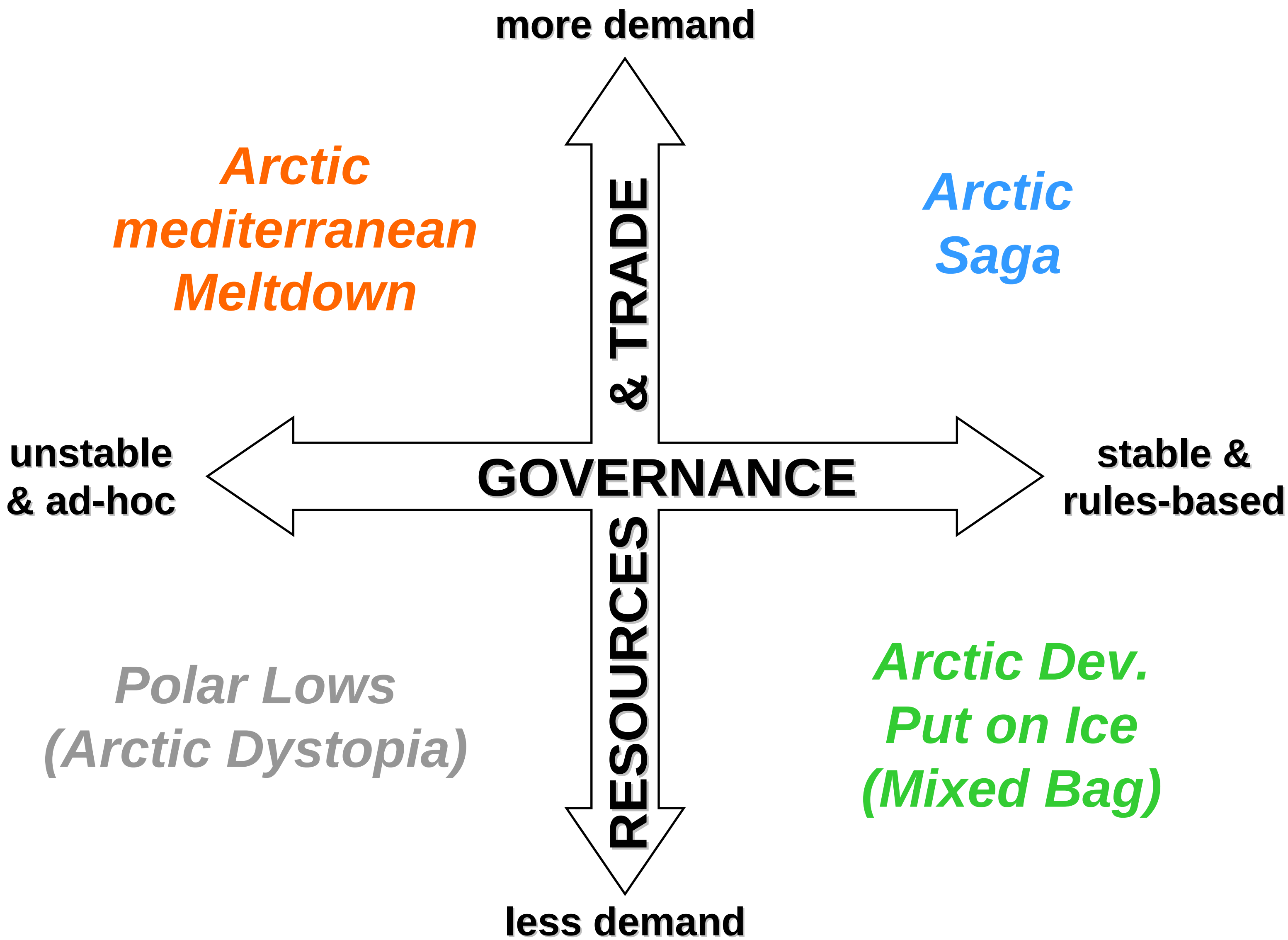

# Arctic Council, PAME-led Arctic Marine Shipping Assessment

- *Lead Countries*: Canada, Finland, and USA
- *Key Countries & Regions*: Norway & Russia (Norwegian-Barents-Kara seas), Iceland, Denmark-Greenland-Faroe Islands, Sweden
- *Timeline*: 2005 – 2009
- *Electronic Survey Questionnaire* ~ Sent to SAOs Jan 2006; Continuing 2004 Data Collection from the Arctic States
- *Inclusive Participation*: Member States, Permanent Participants, Council Working Groups; Council Observers; Shipping Industry; Ship Classification Societies; Research Organizations; Others ~ *Key Challenge: Many Non-Arctic Stakeholders*

# Arctic Marine Shipping Assessment

- *Task* - View of Today's Arctic Marine Shipping Situation (Data from Arctic Coastal States for 2004)
- *Task* - Review of Current Traditional / Indigenous Marine Use
- *Task* - Projections of Maritime Activity Based on ACIA ~ Regional Climate & Economic Scenarios (2020 & 2050)
- *Task* - Impacts (Social, Environmental, Economic) of Today's and Future Arctic Marine Activity
- *Task* - Risk Analyses, Accident Scenarios, Responses
-
- Findings of the Assessment
    - Arctic Council ~ PAME Recommendations for the Member States and the International Maritime Community

# *Scenarios on the Future of*
# ***Arctic Marine Navigation in 2050***



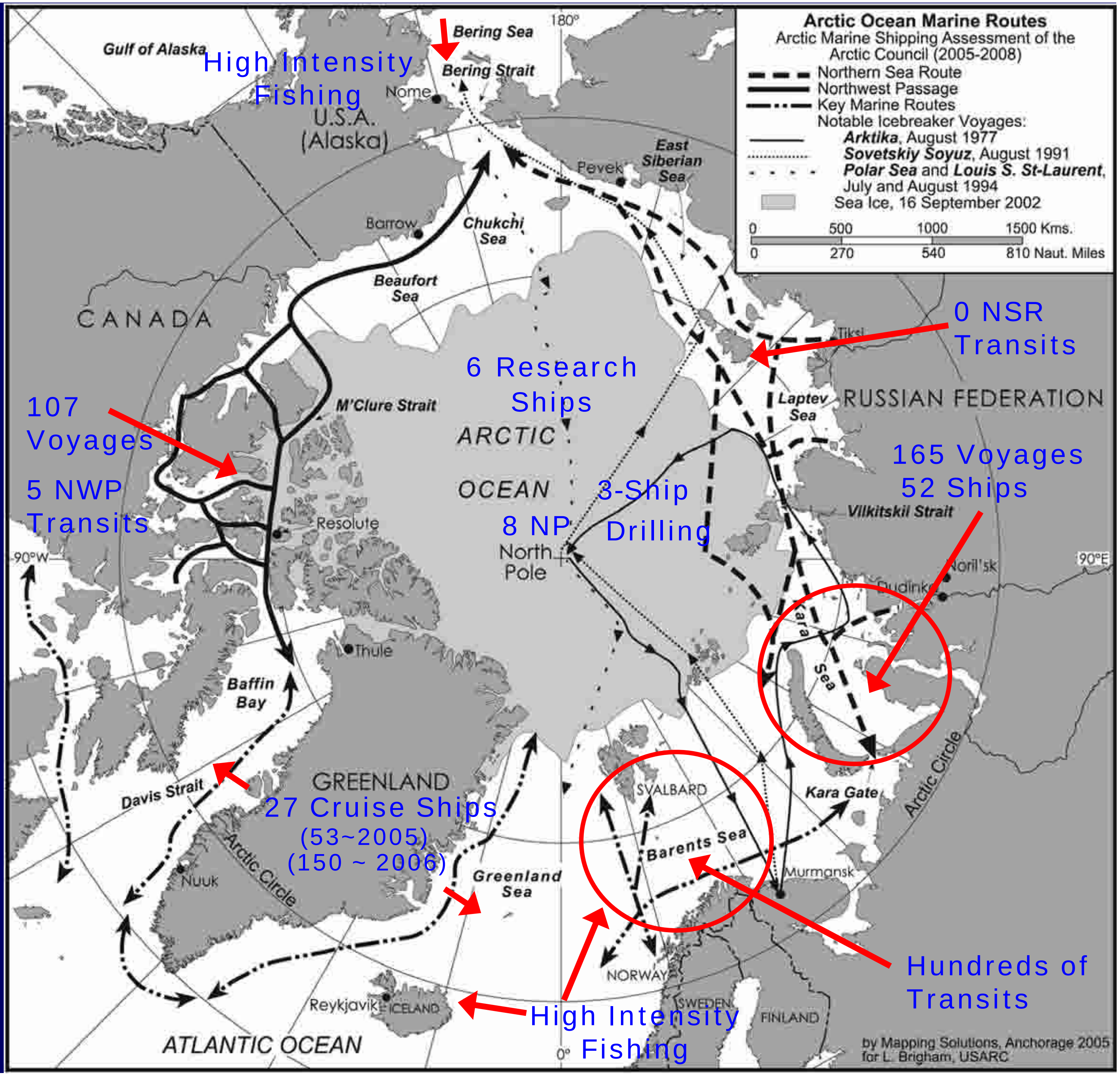
# The Maritime Arctic of Today

## How Many Ships?

## Snapshot of Summer 2004 Traffic

***Modes of Arctic*** 
***<u>Marine Transport</u>***

- **Destinational & Regional**
- **Trans-Arctic**
- **Trans-Arctic with Transshipment**
- **Intra-Arctic**

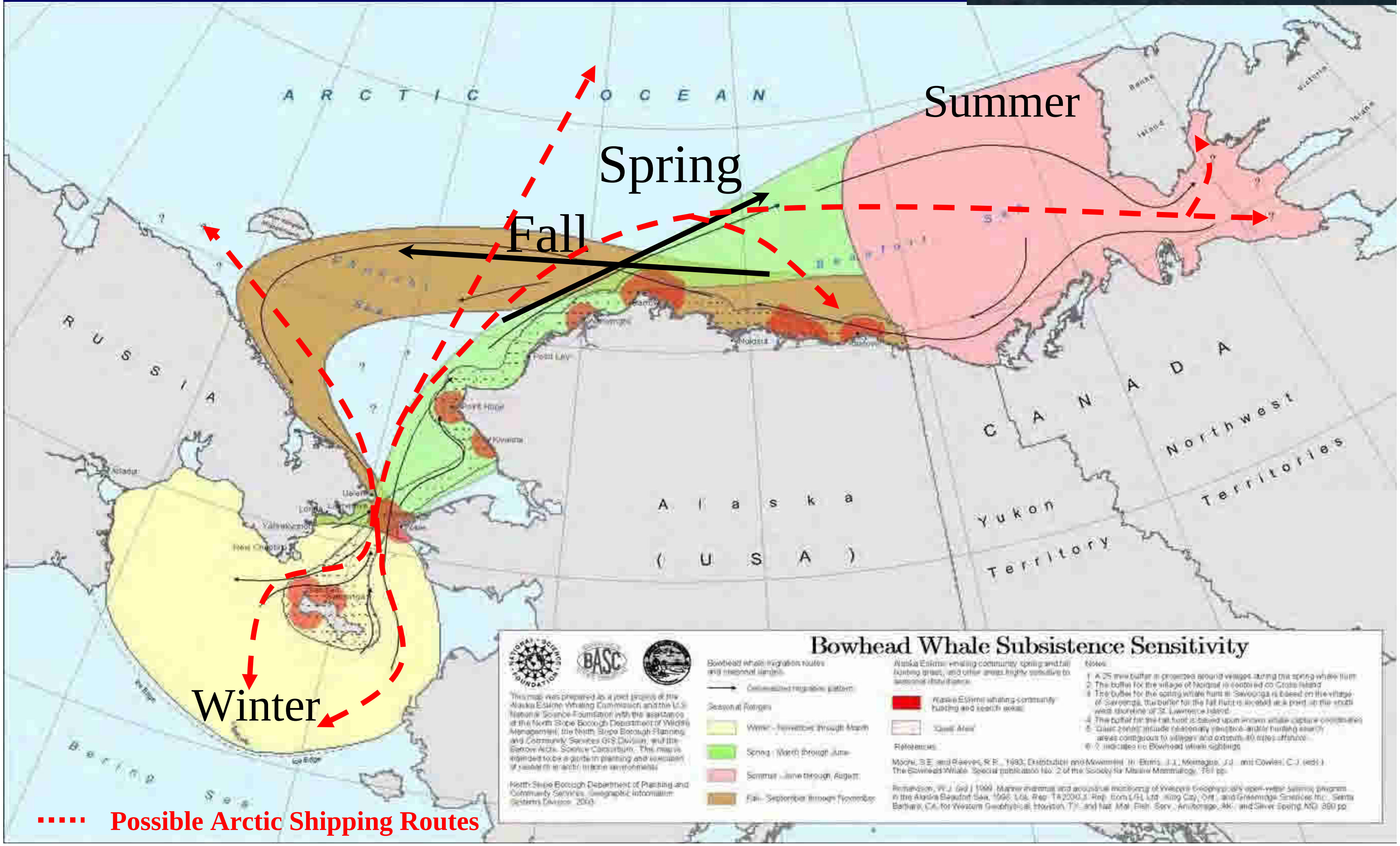



Timeless Arctic
Marine Transport
Arctic Marine Vessel Activity ~ AMSA Ship Types
Tankers ~ Bulk Carriers
Container Ships ~Tug-Barge Combinations
Fishing Vessels ~ Ferries ~ Passenger Vessels/Cruise Ships
Research Vessels ~ Offshore Supply Vessels
Icebreakers (Government & Commercial) ~ Others
PAME

# ‘Wild Card’ Issue 1 ~ Multiple Ocean Uses

**Bowhead Whale Migrations**

**& Arctic Marine Operations**



# 'Wild Card' Issue 2 ~ Arctic Ship Emissions

## New pathway to pollution in Arctic

ONE of the bonuses of global warming is the potential for new shipping routes to open up through the Arctic as ice retreats, shortening journeys by many thousands of miles. There is a downside, however. New northern passages could significantly boost levels of low-lying ozone as ship exhausts pump pollutants into the pristine environment.

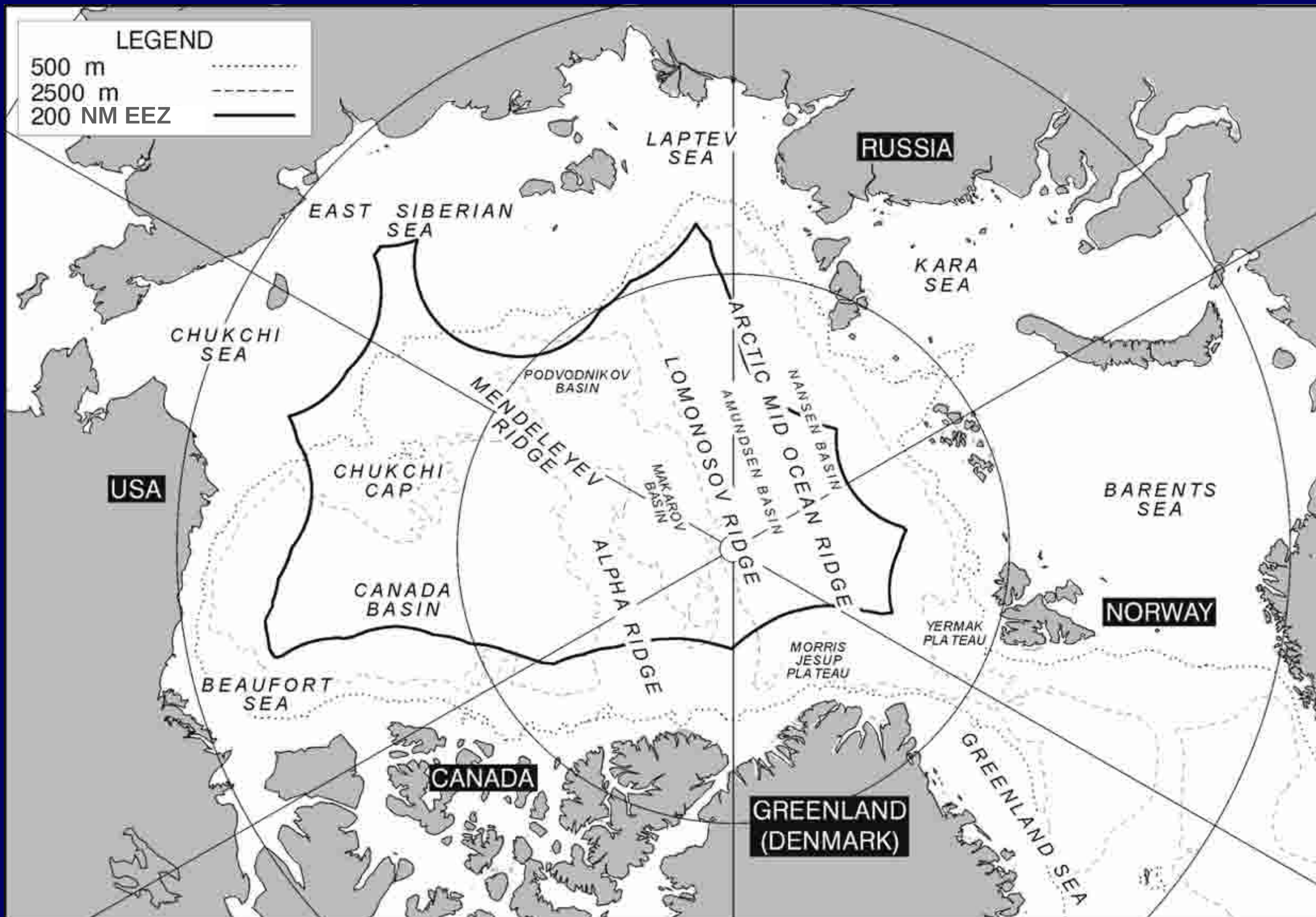
Climate models indicate that the northern passages – the north-east coast of Siberia, northern Alaska and around the Canadian archipelago – may be open to shipping during the summer months from around 2050 onwards. Claire Granier, from the University of Pierre and Marie Curie in Paris, France, and her colleagues calculated the likely ozone emissions associated with such a scenario, assuming that the routes would be accessible for six months of the year.

Emissions of nitrogen oxides and carbon monoxide from ships could triple ozone levels, making them comparable to those in industrialised regions today (*Geophysical Research Letters*, DOI: 10.1029/2006GL026180).

"The Arctic is a very sensitive region and these very high ozone levels are likely to have a serious impact on plant life," says Ulrike Niemeier, a co-author from the Max Plank Institute for Meteorology in Hamburg, Germany.

**New Scientist**
**22 July 2006**

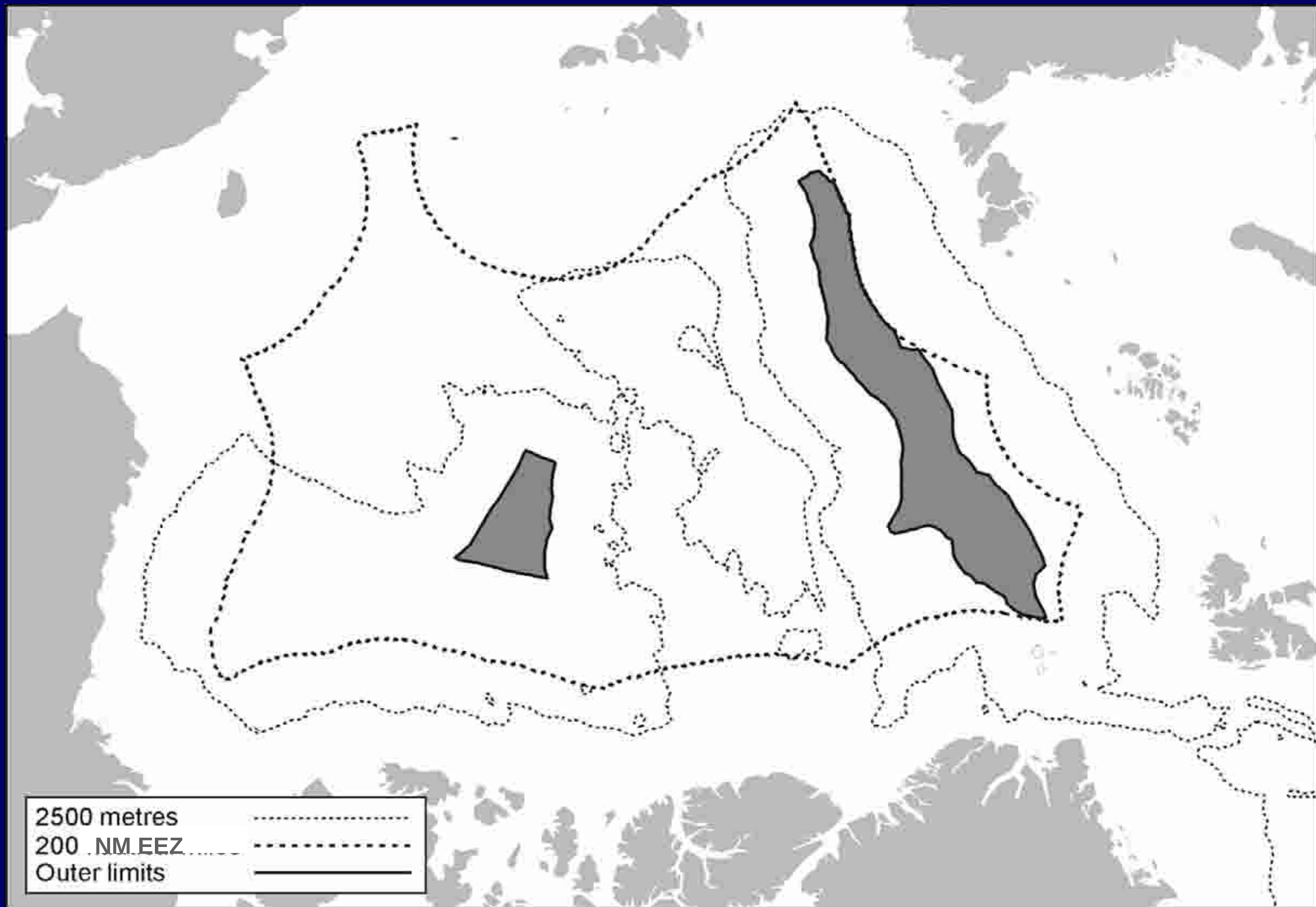
# Today's Maritime Arctic
## (200 NM Exclusive Economic Zone)




**'Wild Card' Issue 3A**

**(Macnab 2000)**

# Hypothetical - Future Maritime Arctic
## (After UNCLOS Article 76)




(Macnab 2000)

**'Wild Card' Issue 3B**

# Lifeboat operational performance in cold environments

**Royal Institution of Naval Architects**

**London, May 30-31, 2007**

# Lifeboat operational performance in cold environments

**António Simões Ré**
Institute for Ocean Technology
National Research Council
Canada

**Brian Veitch**
Ocean Engineering Research Centre
Memorial University of Newfoundland
Canada

# Lifeboat operational performance in cold environments

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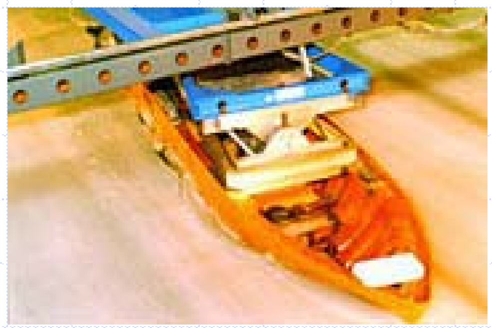
Ocean Engineering Research Centre
Memorial University of Newfoundland
Canada

# Lifeboat operational performance in cold environments

## Institute for Ocean Technology

IOT conducts ocean engineering research through modeling of ocean environments, predicting and improving the performance of marine systems, and developing innovative technologies

Ocean Engineering Basin



Ice Tank

Clear Water Tank

## Ocean Engineering Research Centre

OERC advances ocean engineering research, promotes interaction amongst researchers, fosters an innovative research milieu, and takes an active role in shaping policies relating to ocean technology

Tow Tank & other engineering labs

Research programs

## Offshore & Maritime Safety Research Program

- Process safety & major offshore hazards
- Emergency response
- Technical performance / environmental factors
- Human factors / performance
- Occupational safety
- Training elements & competencies

## Offshore & Maritime Safety Research Program

- Process safety & major offshore hazards
- Emergency response
- Technical performance / environmental factors
- Human factors / performance
- Occupational safety
- Training elements & competencies

# Lifeboat operational performance in cold environments

## Offshore & Maritime Safety Research Program

- Multi-disciplinary interconnected research projects & students
- Large experimental facilities for performance evaluation of evacuation systems
- Full mission ship bridge simulator with full motion articulation
- A marine base - facilities for launching of lifeboats & FRCs
- Cold water physiology lab & Biomechanics and ergonomics research lab

# Lifeboat operational performance in cold environments

## Offshore & Maritime Safety Research Program

- Technical performance / environmental factors
  - Davit launched lifeboats, various configurations
  - Free-fall lifeboats
  - Rafts, FRCs
  - waves & wind
  - ice
  - model & full scale
- Large experimental facilities for performance evaluation of evacuation systems
  - Ocean Engineering Basin
  - Clear Water Tow Tank
  - Ice Tank
  - Lifeboat and rafts for field trials

## Lifeboat operational performance in cold environments

# Offshore & Maritime Safety Research Program

- Technical performance / environmental factors
  - Davit launched lifeboats, various configurations
  - Free-fall lifeboats
  - Rafts, FRCs
  - waves & wind
  - ice
  - model & full scale

- Large experimental facilities for performance evaluation of evacuation systems
  - Ocean Engineering Basin
  - Clear Water Tow Tank
  - Ice Tank
  - Lifeboat and rafts for field trials

# Lifeboat operational performance in cold environments

## Offshore & Maritime Safety Research Program

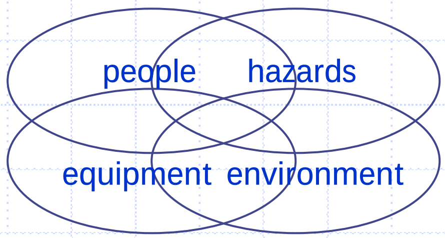
- Improve safety of personnel at sea


Ice Tank with model lifeboat

Instrumented lifeboat on field trials

# Lifeboat operational performance in cold environments

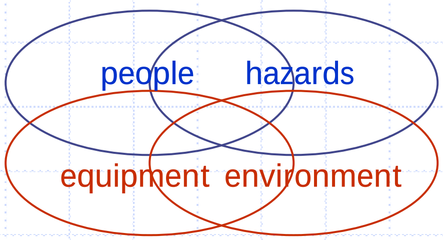
- Goals
- Methods
- Results




- In circumstances that necessitate an emergency escape-evacuation-rescue response,
  - personnel must have a reasonable expectation of avoiding harm in the environmental conditions that can reasonably be expected to prevail during operations.
    - (i) adequate means for personnel to protect themselves and for personnel to escape the potential harm posed by credible hazards;
    - (ii) adequate means for personnel, including injured personnel, to abandon the vessel in a controlled manner;
    - (iii) adequate means and support for the rescue and recovery of personnel to a place of safety where medical assistance is available

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results




- In circumstances that necessitate an eer response,
  - personnel must have a reasonable expectation of avoiding harm in the environmental conditions that can reasonably be expected to prevail during operations.
    - (i) adequate means for personnel to protect themselves and for personnel to escape the potential harm posed by credible hazards;
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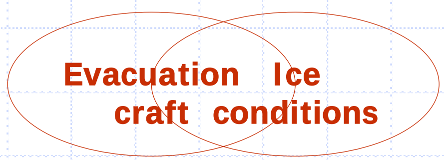
# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results




- To evaluate lifeboat performance capabilities in ice conditions
  - Investigate the operating limits imposed by ice
    - concentration, floe size, thickness, waves
    - power
    - hull form
  - Limits constitute boundary for complementary, innovative evacuation system

# Lifeboat operational performance in cold environments

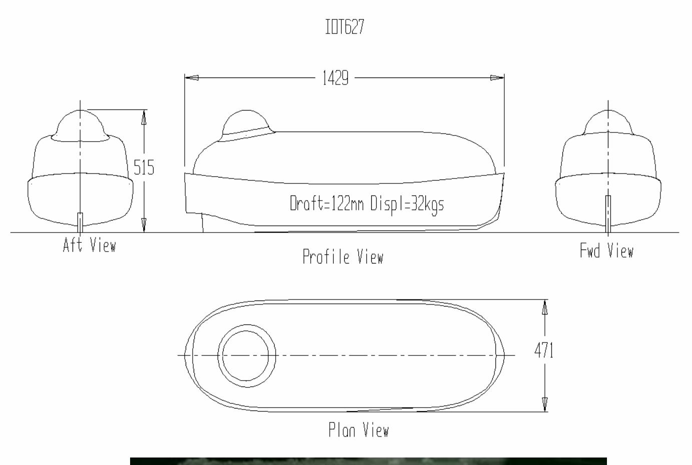
- Goals
- Methods
- Results

## Approach

- Large scale model experiments in an ice tank
- Range of pack ice conditions
- Three lifeboat hull forms
- Range of power level
- Additional tests in combined ice and waves

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results



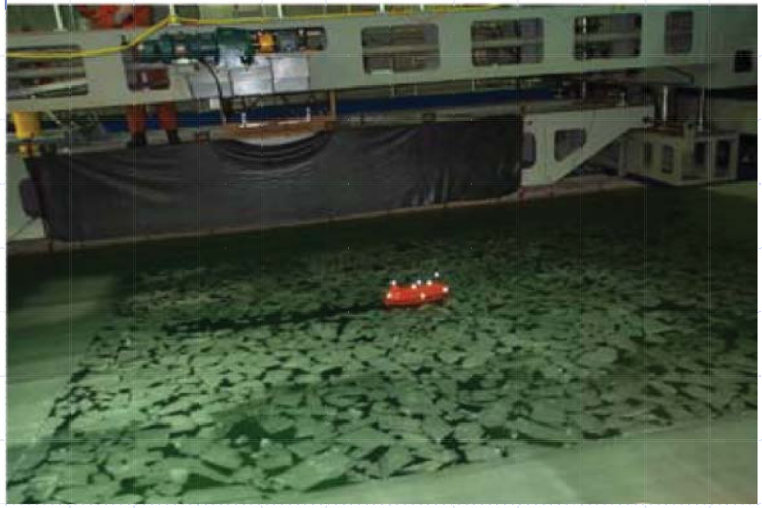

- Experimental setup - scope
  - Benchmark series
    - 1:13 model of a conventional TEMPSC lifeboat in the Ice Tank
    - Two models
      - Launch version: regular power (6 knots open water)
      - Power version: 4 power settings (~1×, 2×, 3×, 4×)
    - Ice conditions
      - Concentration: 4/10ths – 9/10ths
      - Thickness: thick (650 mm) and thin (325 mm)
      - Floe size: large (~6 - 7 m) and small (~2 - 4 m)

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

## Experimental setup – models

- Benchmark series
  - Model setup
    - Motor & battery pack
    - Propeller & steering nozzle
    - Camera in coxswain's position
    - RC & Wireless transmitter
    - Optical tracking markers
    - Data acquisition system
    - Davits & hook release (launch version)
    - Extra batteries (power version)



# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

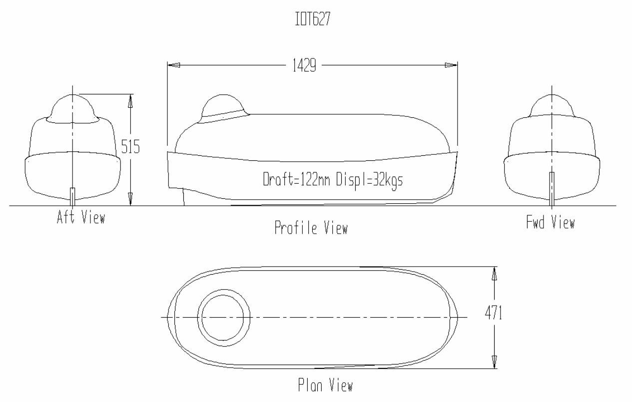
- Experimental setup – test plan
  - Benchmark series
    - Separate tests in thicker & thinner ice
    - For a given ice thickness, two pools were made
      - Large floes
      - Small floes
    - Concentration varied by changing pool dimensions incrementally
  - Test plan
    - Thin ice & small floes, 8/10$^{ths}$- 5/10$^{ths}$: 20 tests
    - Thin ice & large floes, 7/10$^{ths}$- 5/10$^{ths}$: 12 tests
    - Thick ice & small floes, 7/10$^{ths}$- 5/10$^{ths}$: 10 tests
    - Thick ice & large floes, 7/10$^{ths}$- 5/10$^{ths}$: 10 tests
    - Power varied significantly throughout
      - 52 tests total

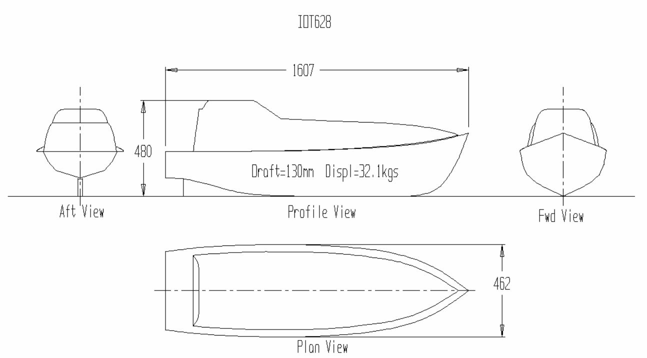
# Lifeboat operational performance in cold environments

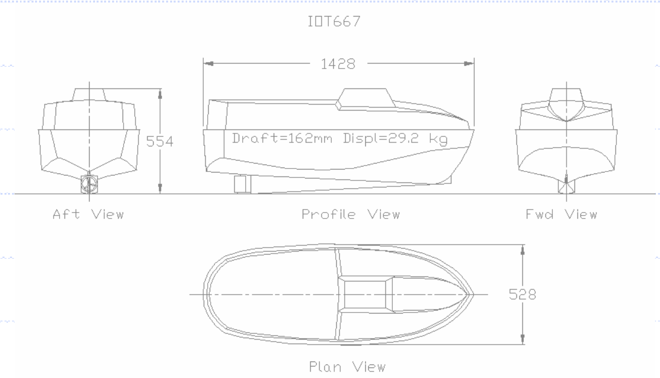
- Goals
- Methods
- Results

## Experimental setup – scope

- Secondary series
  - Three 1:7 models tested in the Ice Tank
    - Conventional TEMPSC lifeboat
    - Free-fall lifeboat
    - Hard-chine lifeboat
  - Ice conditions
    - Concentration: 5/10ths – 8/10ths
    - Thickness: thin (325 mm)
    - Floe size: large and small

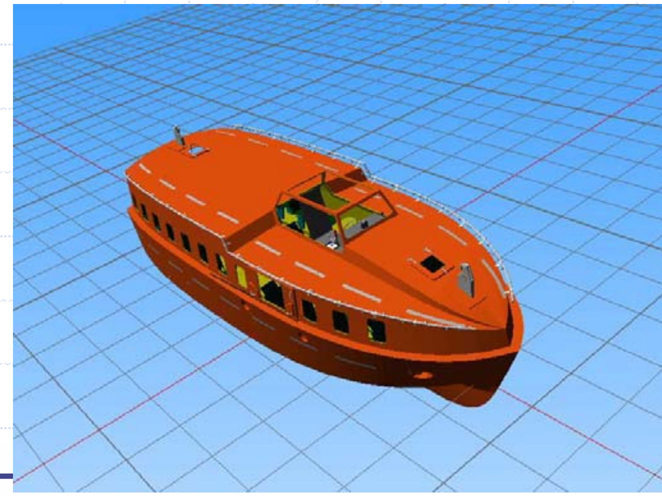


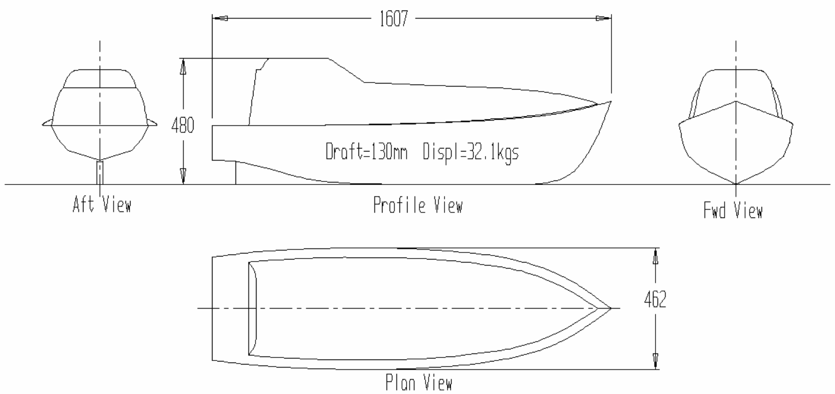



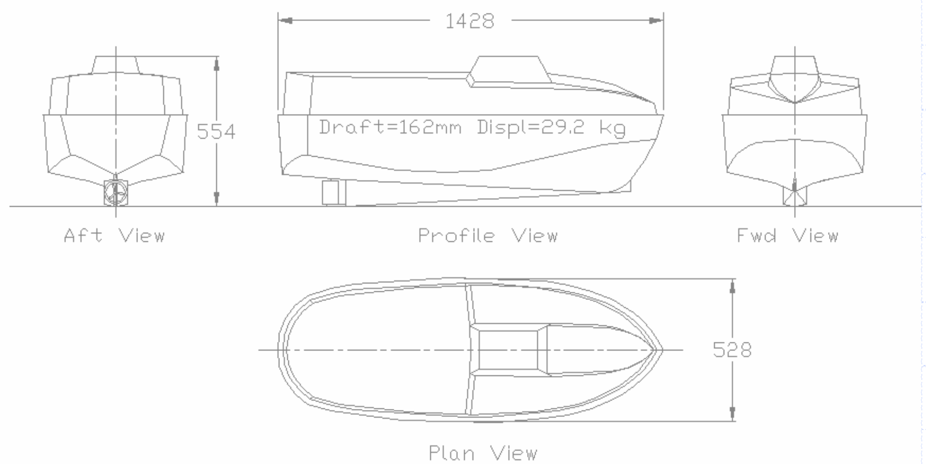



# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results





# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Experimental setup – test plan
  - Secondary series
    - Power
      - Setting for 6 knots in open water
      - Another setting + 10 - 25%
  - Test plan
    - Thin & small floes, 8/10$^{ths}$- 6/10$^{ths}$: 45 tests
    - Thin & large floes, 7/10$^{ths}$- 5/10$^{ths}$: 31 tests
    - Power variations (relatively minor)
      - 76 tests total

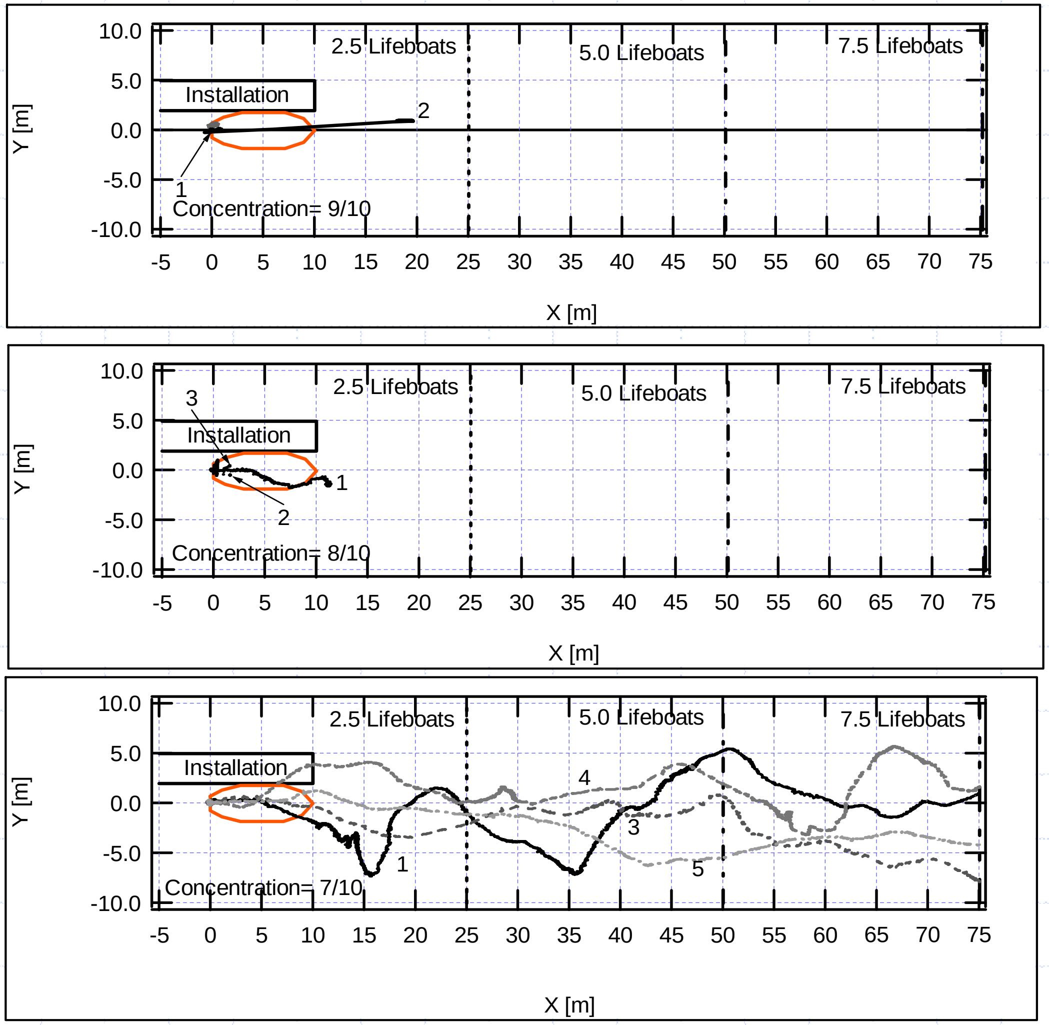
# Lifeboat operational performance in cold environments



- Performance measures
  - Ability of the vessel to make way through pack ice over a prescribed distance
    - Failure to do so earned a failing grade, and success earned a pass
  - A time limit was used as a secondary measure
  - Maneuvering in ice was evaluated using turning circle as a performance measure
- Scope and Limitations.
  - Simple pack ice in various concentration
  - No ridges, rubble, rafted ice
  - Bounded pack ice pools
  - Ice loads not considered

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results




Examples of transits made by the conventional TEMPSC in thin ice & small floes

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Results summary – benchmark tests
  - Ice condition limitations

| Nominal Ice concentration [10ths] | | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|
| nominal thickness | nominal floe size | grade [Pass or Fail] | | | | | |
| 25mm | small | 3P | 3P | 5P | 4P | 3F | 2F |
| 25mm | large | 3P | 3P | 3P | 3F | | |
| 50mm | small | 3P | 2P | 2P | 3F | | |
| 50mm | large | 3P | 2P | 2F | 3F | | |

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Results summary – benchmark tests
  - Ice condition limitations
  - Test results were consistent: repeated tests yielded the same result
  - Conditions became impassable at concentrations of between 6/10ths to 8/10ths, depending on the thickness and ice floe size
    - thicker ice and larger floes being more difficult to transit than thinner ice and smaller floes.

| Nominal Ice concentration [10ths] | | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|
| nominal thickness | nominal floe size | grade [Pass or Fail] | | | | | |
| 25mm | small | 3P | 3P | 5P | 4P | 3F | 2F |
| 25mm | large | 3P | 3P | 3P | 3F | | |
| 50mm | small | 3P | 2P | 2P | 3F | | |
| 50mm | large | 3P | 2P | 2F | 3F | | |

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

## Results summary – benchmark tests

- Power effects on ice condition limitations

| thickness [mm] | floe size [-] | ice concentration [10ths] 5 | | 6 | | 7 | | 8 | |
|---|---|---|---|---|---|---|---|---|---|
| 25 | small | | | | | | | 2F | |
| | | | | | 5P | 2P | 7P | | 3F |
| 25 | large | | | | | 2P | 3P | | |
| | | | | | 3P | 2P | 3P3F | | |
| 35 | small | | | | | 2P | 2P1F | | |
| | | | | | 2P | 1P1F | 3F | | |
| 35 | large | | | 2P | 1P1F | 3F | 3F | $T_4$ | $T_3$ |
| | | | 2P | 1P | 2P2F | | 3F | $T_4$ | $T_1$ |

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Results summary – benchmark tests
  - Power effects on ice condition limitations
  - Conventional TEMPSC - conditions became impassable at concentrations of between 6/10ths to 8/10ths, depending on the thickness and ice floe size
  - Addition of significantly more power (several times) extended the operability in ice only very marginally

| thickness | floe size | ice concentration [10ths] | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| [mm] | [-] | 5 | | 6 | | 7 | | 8 | |
| 25 | small | | | | | | | 2F | |
| | | | | | 5P | 2P | 7P | | 3F |
| 25 | large | | | | | 2P | 3P | | |
| | | | | | 3P | 2P | 3P3F | | |
| 35 | small | | | | | 2P | 2P1F | | |
| | | | | | 2P | 1P1F | 3F | | |
| 35 | large | | | 2P | 1P1F | 3F | 3F | | |
| | | | 2P | 1P | 2P2F | | 3F | | |

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Results summary – secondary tests
  - Hull form effects

| floe size | model | Ice concentration [10ths] | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 5 | | 6 | | 7 | | 8 | |
| small | TEMPSC (conventional) | | | 6P | 6P | 1F | | | |
| small | Free Fall | | | 7P | 7P | 1F | | 1F | |
| small | TEMPSC (new) | | | 6P | 5P 1F | 2F | | 2F | |
| large | TEMPSC (conventional) | | 2P | | 2P 1F | 5P 2F | | | |
| large | Free Fall | 3P | 4P | 3P | | 2F | | Power Legend | |
| large | TEMPSC (new) | | | 3P | 4P | | | T+ | T1 |

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Limiting ice conditions and power
  - The limiting ice concentration for the conventional lifeboat was typically $7/10^{ths}$.
  - The free fall lifeboat showed similar behavior as the conventional & hard chine boats in terms of limiting ice conditions.
  - No compelling evidence that one hull form performed better or worse than the others.
  - No significant improvement from adding more power.

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

- Maneuvering
  - Maneuvering in open water and in ice was evaluated using turning circle diameter.
  - All three lifeboats had larger turning circles in open water than in the ice conditions in which tests were done.
  - The open water turning circles for the lifeboats were different.
  - In ice, the turning circles for all the vessels were practically the same ...
  - ... suggesting that while the hull forms perform differently in open water, pack ice equalizes the performance.

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results


## Waves

- The presence of waves in combination with pack ice sometimes helped the vessel pick its way through the ice, even in relatively high ice concentrations that prevented progress in calm conditions
  - although this progress was often slow and often slower than the time benchmark permitted for a pass.
- The pass/fail grades for the conventional, free fall and hard chine models were similar.

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

## Conclusions

- In terms of ability to make progress through pack ice conditions & to maneuver through turning circles, hull form was found to have no significant effect.
- Further, adding more power to the vessel was found to yield no or only marginal change in performance limits.
- Ice in concentrations of about 6/10ths to 8/10ths was found to prevent the lifeboats from making progress in the calm water conditions tested.
- Larger floes & thicker ice were found to hinder performance slightly more than smaller floes and thinner ice.
- Waves marginally improved the vessels' progress through the ice, even in relatively high ice concentrations that prevented progress in calm conditions.

# Lifeboat operational performance in cold environments

- Goals
- Methods
- Results

## Conclusions

- The ice conditions that can reasonably be expected to prevent a lifeboat from making way, or to slow its progress drastically, are quite modest.
- Displacement type lifeboats of the sort tested are not suitable means of evacuation in pack ice conditions that approach the performance limits delineated here.
- In areas with such environmental conditions, another means of evacuation is required, whether in place of the conventional displacement type lifeboat, or to complement it.
- We have begun a field trial program to examine lifeboat operability in ice.

## Acknowledgements

The financial support of Natural Resources Canada's Program of Energy Research and Development (PERD) is acknowledged with gratitude.

# Thank you

for your kind attention.

NRC · CNRC

*From Discovery to Innovation...*

# *Model Experiments to Support the Design of Large Icebreaking Tankers*

**David Molyneux**
Institute for Ocean Technology,
National Research Council Canada

**Hyun-Soo Kim**
Marine Research Institute,
Samsung Heavy Industries, Korea

# *Introduction*

- SHI identified growing market for ice capable ships to transport oil and gas products.
- Optimize design to balance power consumption in open water and ice.
- Focus on Arctic Ocean & Baltic Sea.
- Ice types to be considered:
  - Level, first year ice
  - Pack ice (70-99% cover)
  - Ridges and rubble
  - Brash ice (not all designs)

# Ice Properties Affecting Resistance & Propulsion

- Ice type and coverage
- Ice thickness
- Flexural strength
- Compressive strength
- Density
- Snow cover
- Degree of re-freezing and consolidation

# Modeling Icebreaking Performance

- Performance prediction for design optimization
  - Hull
  - Propellers
  - Rudders and appendages
- Scaled ship
  - Scale approx.1:35
  - Hull-ice friction coefficient

    0.05 (new ship)
- Scaled material

  Refrigerated ice
  - EGADS
  - EGADS/CD

NRC · CNRC

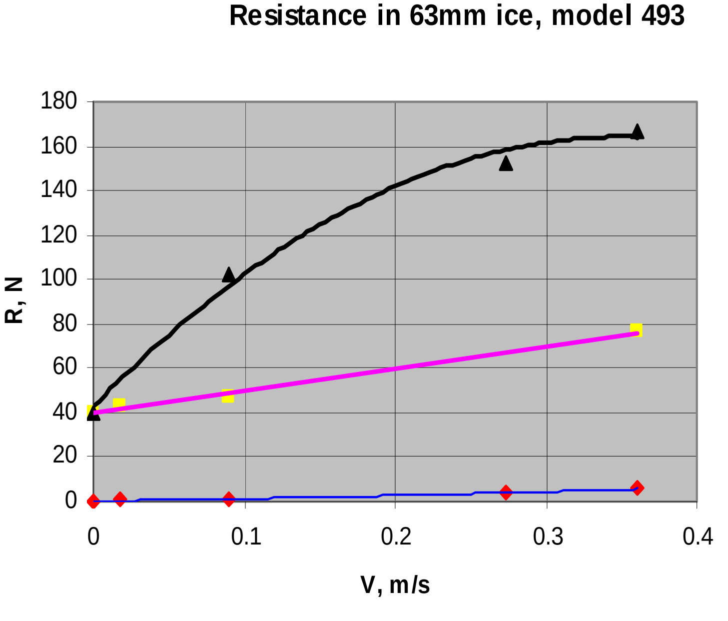
# Modeling Ice Properties- Level Ice

- Thickness
- Strength
  - compressive strength
  - flexural strength
  - role of pre-sawn test
- Ice density
- Hull-ice friction
  - Roughness
  - Snow cover

# Force Components in Icebreaking

- Breaking
- Submergence (buoyancy)
- Clearing
- Hydrodynamic

$$R_{tot} = R_{br} + R_b + R_c + R_{ow}$$




# Force Coefficients

$$R_t = R_{br} + R_c + R_b + R_{ow}$$

$$C_{br} = \frac{R_{br}}{\rho_i B h_i V_M{}^2}$$

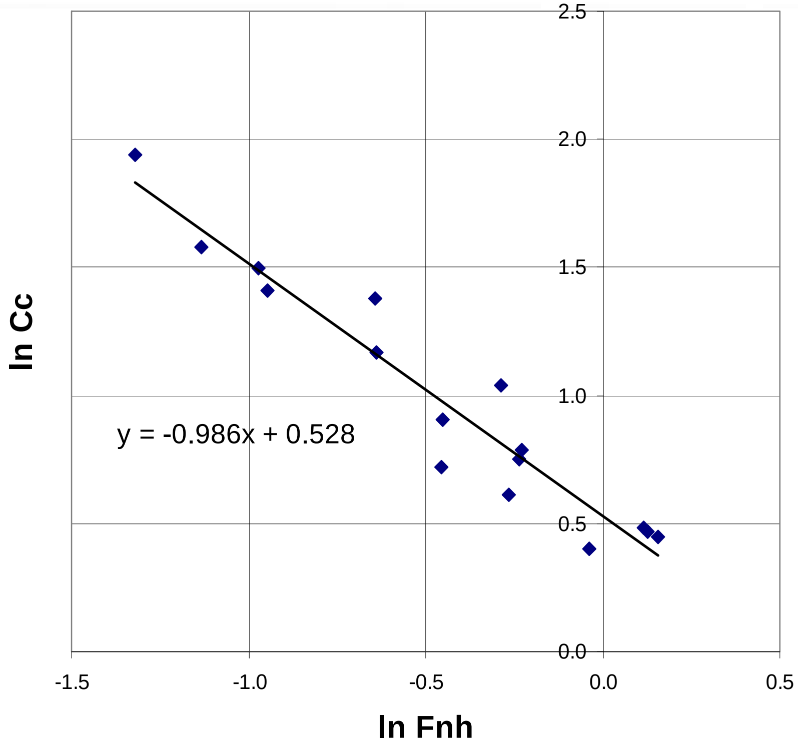
$$C_c = \frac{R_c}{\rho_i B h_i V_M^2}$$

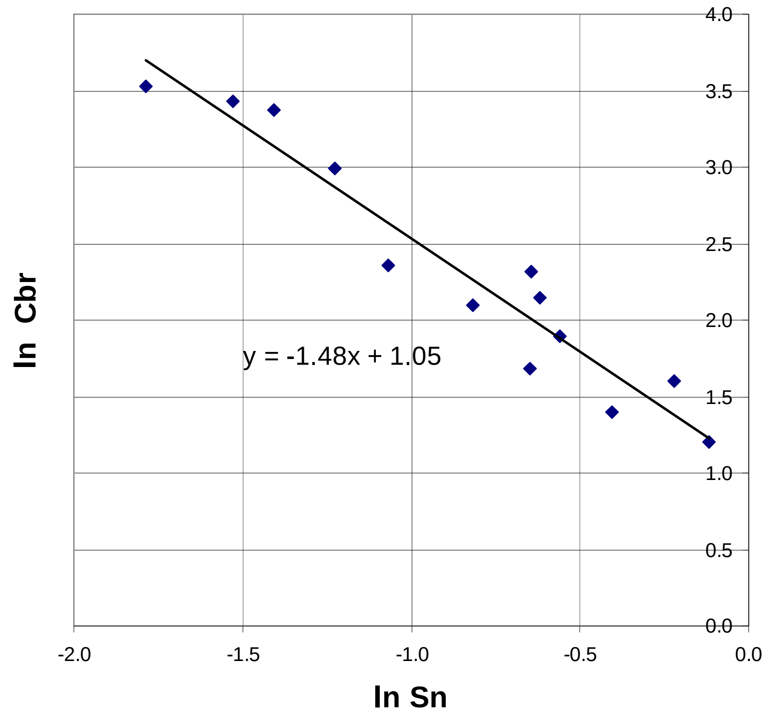
$$C_b = \frac{R_b}{\Delta\rho_i g B h_i T}$$

$$S_n = \left[\frac{\rho_i B V_M^2}{\sigma_f h_i}\right]^{1/2} \qquad Fn_h = \frac{V}{\sqrt{g h_i}}$$

NRC · CNRC

# Ice Clearing & Breaking Components



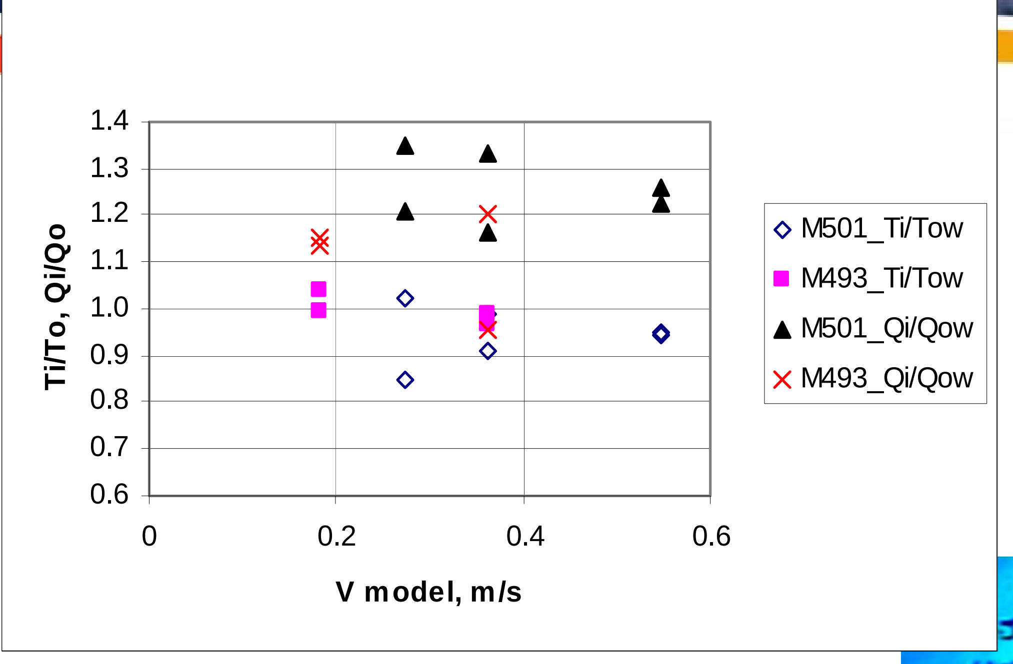

NRC · CNRC

# Ship Propulsion in Ice

- Delivered power is better measure of ship performance than resistance

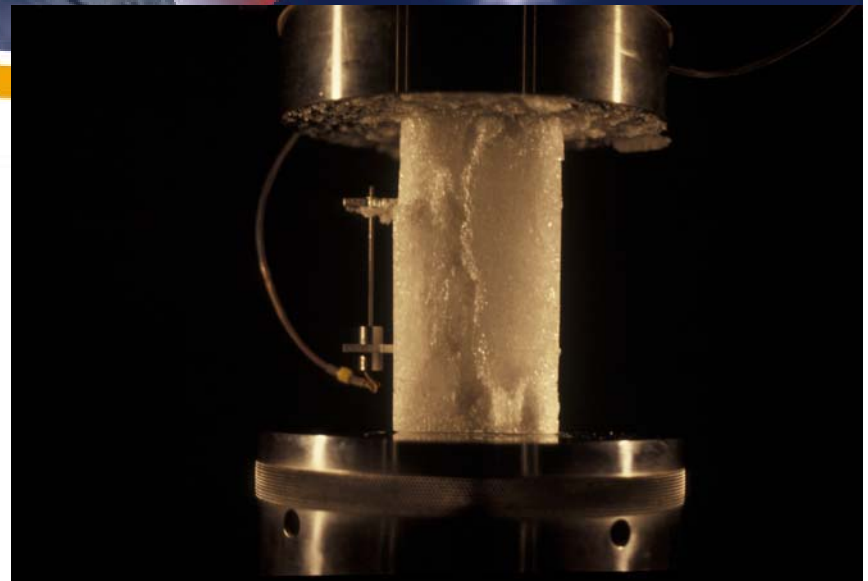
- Develop methods for predicting engine power
  – Range of shaft rotation rates to match delivered power from self-propulsion point to full power
  – Interpolate at point where restraining force=ice resistance
  – Obtain thrust and torque in open water
  – Correct torque for propeller-ice interaction

# Propeller-Ice Interaction




# *Model-Full Scale Correlation*

- Ship performance
  - Speed
  - Power
  - Thrust
  - RPM
- Ice properties
  - Thickness
  - Strength
    - Field or laboratory
    - Direct measurement
    - Temperature/salinity
  - Snow cover
  - Hull-ice friction



# Application of Modeling to Design

- Tankers for Samsung Heavy Industries
  - Aframax for Arctic ice conditions
  - Suezmax for Arctic ice conditions
  - Suezmax for Baltic ice conditions
  - Conventional tanker (pack ice & open water only)

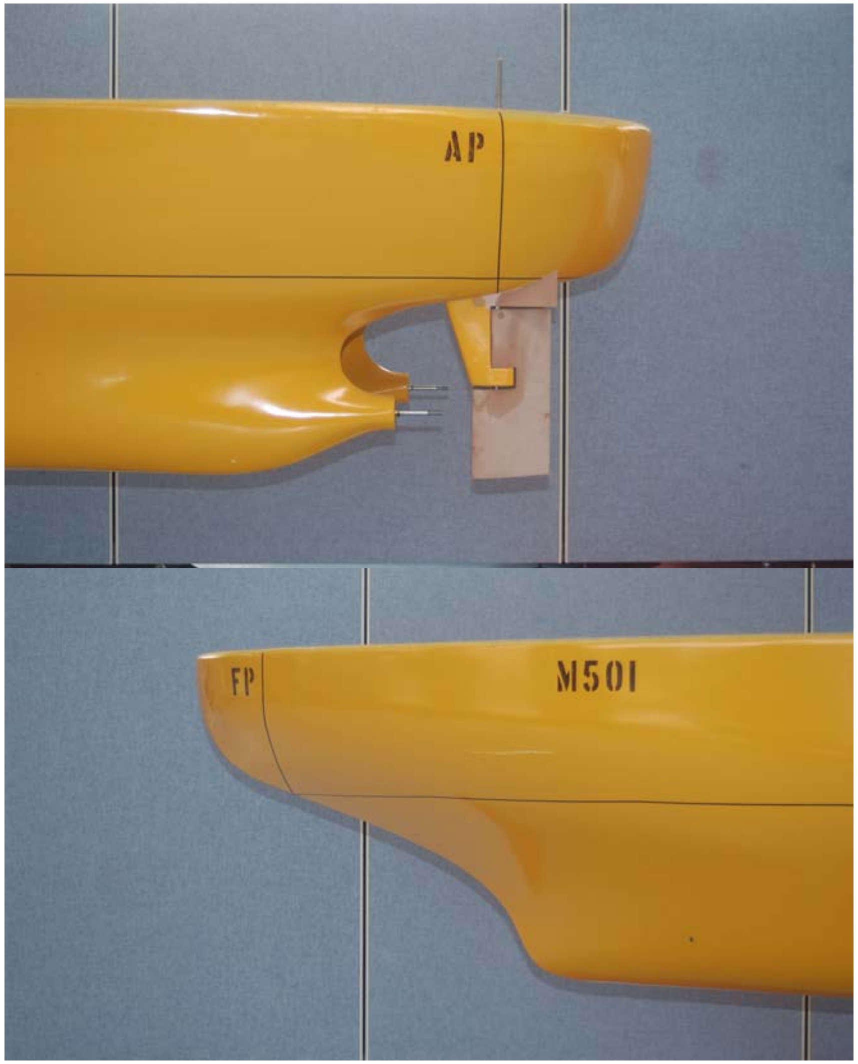
- Performance predictions in level ice, pack ice, rubble & open water

- Cost important factor for commercial shipping

NRC · CNRC

# Aframax Tanker for Arctic Ice

- Shallow draft
- Need for ice protection of propellers
- Twin gondola stern
- Two variations on R-class type bow
- Twin rudders

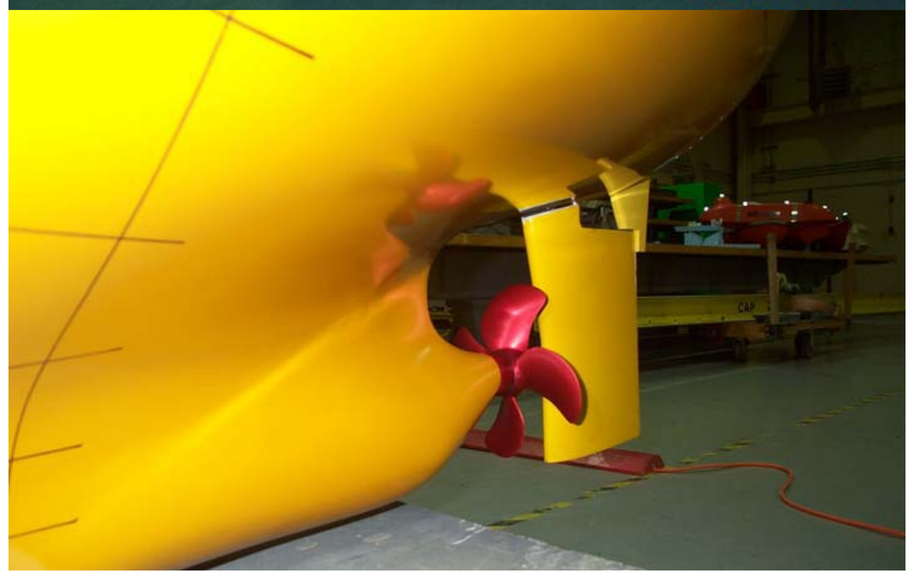


# Suezmax Tanker for Arctic Ice

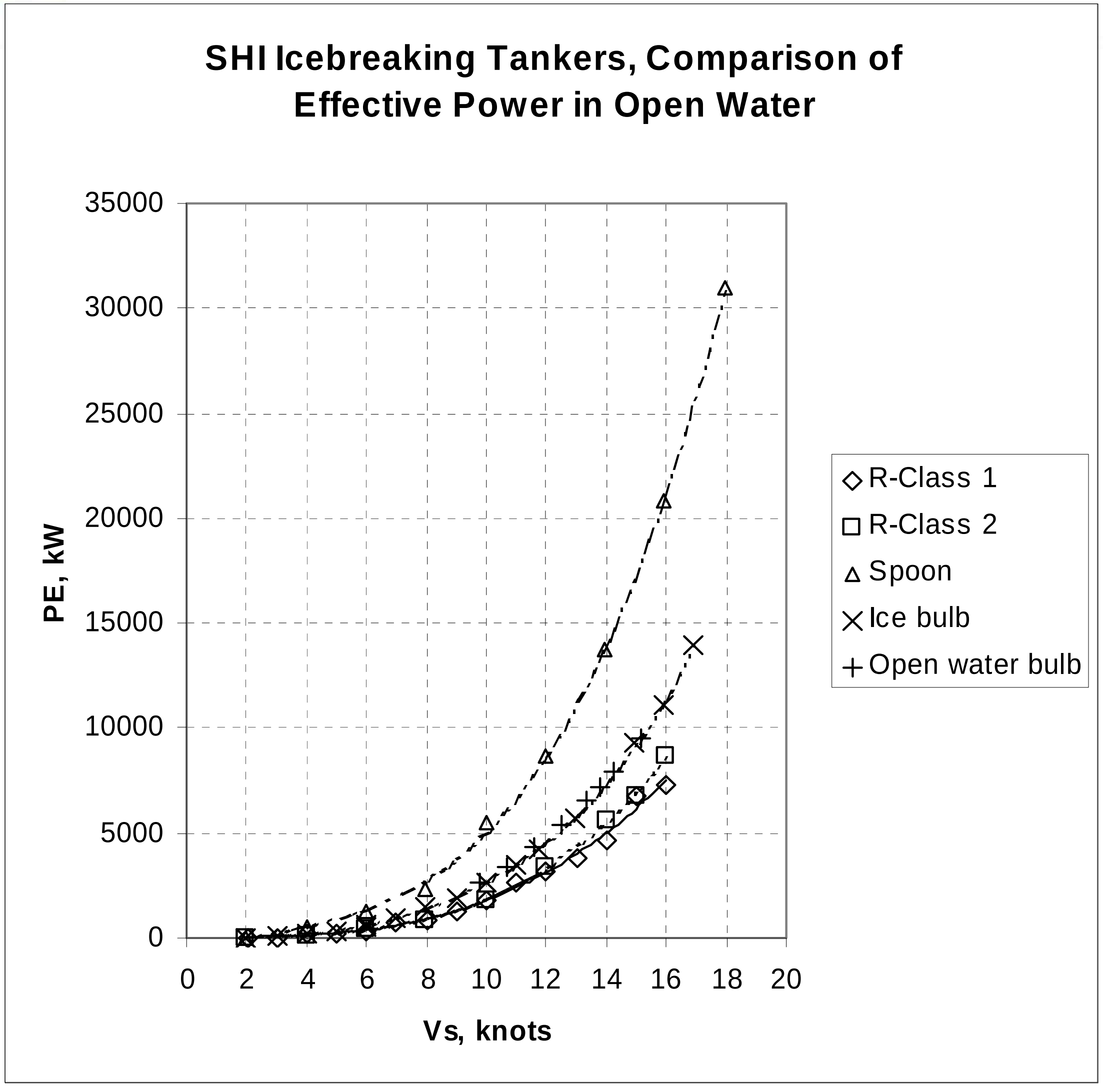
- Deeper draft, less need for propeller protection
- Twin screw stern, open shafts
- Spoon bow for reduced power in ice
- Twin rudders


# Suezmax Tanker for Baltic Ice

- Designed for performance in broken ice
- Single screw
- Single rudder
- Bulbous bow for open water performance

# Open Water Resistance

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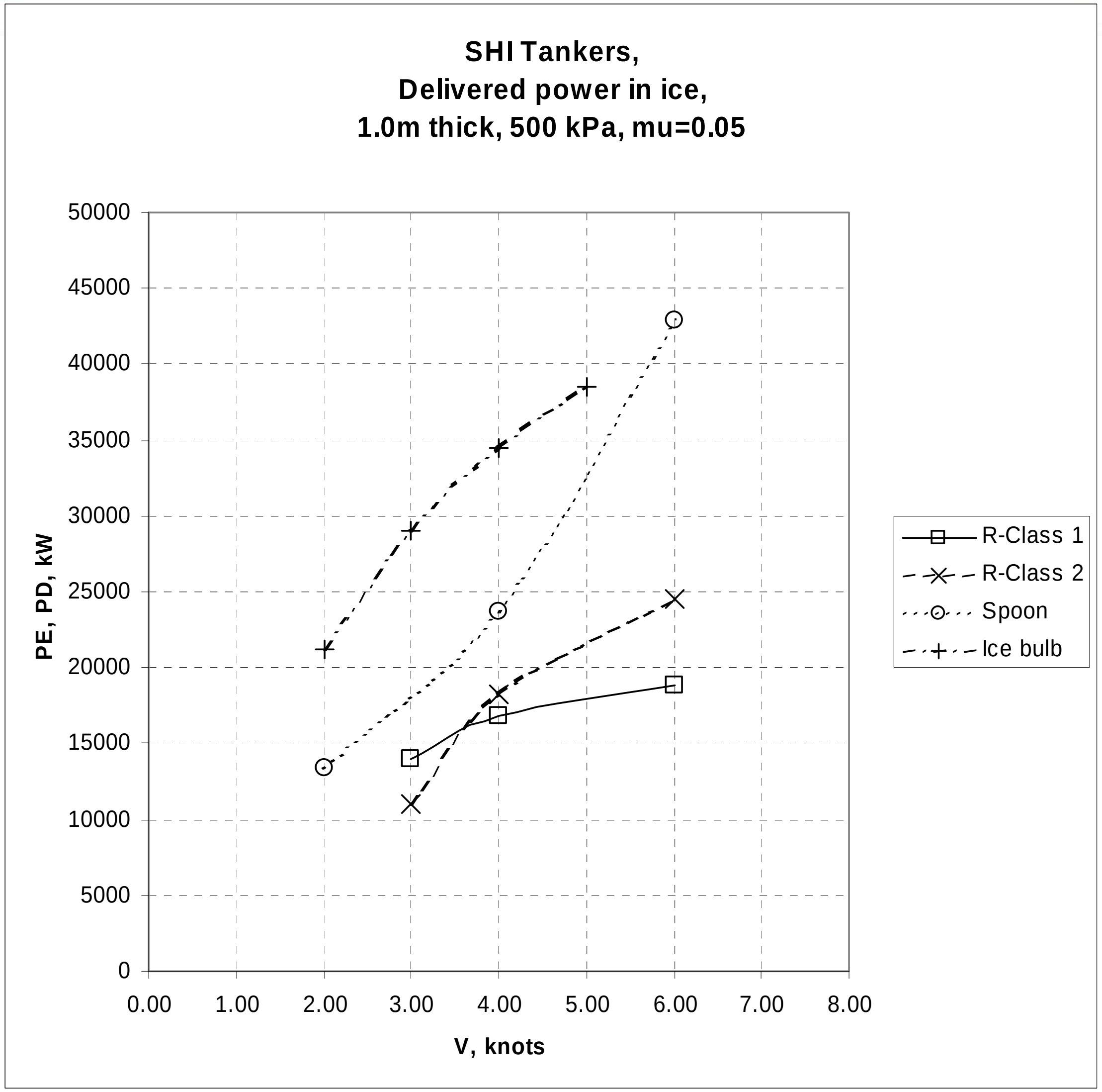
# Delivered Power in Level Ice

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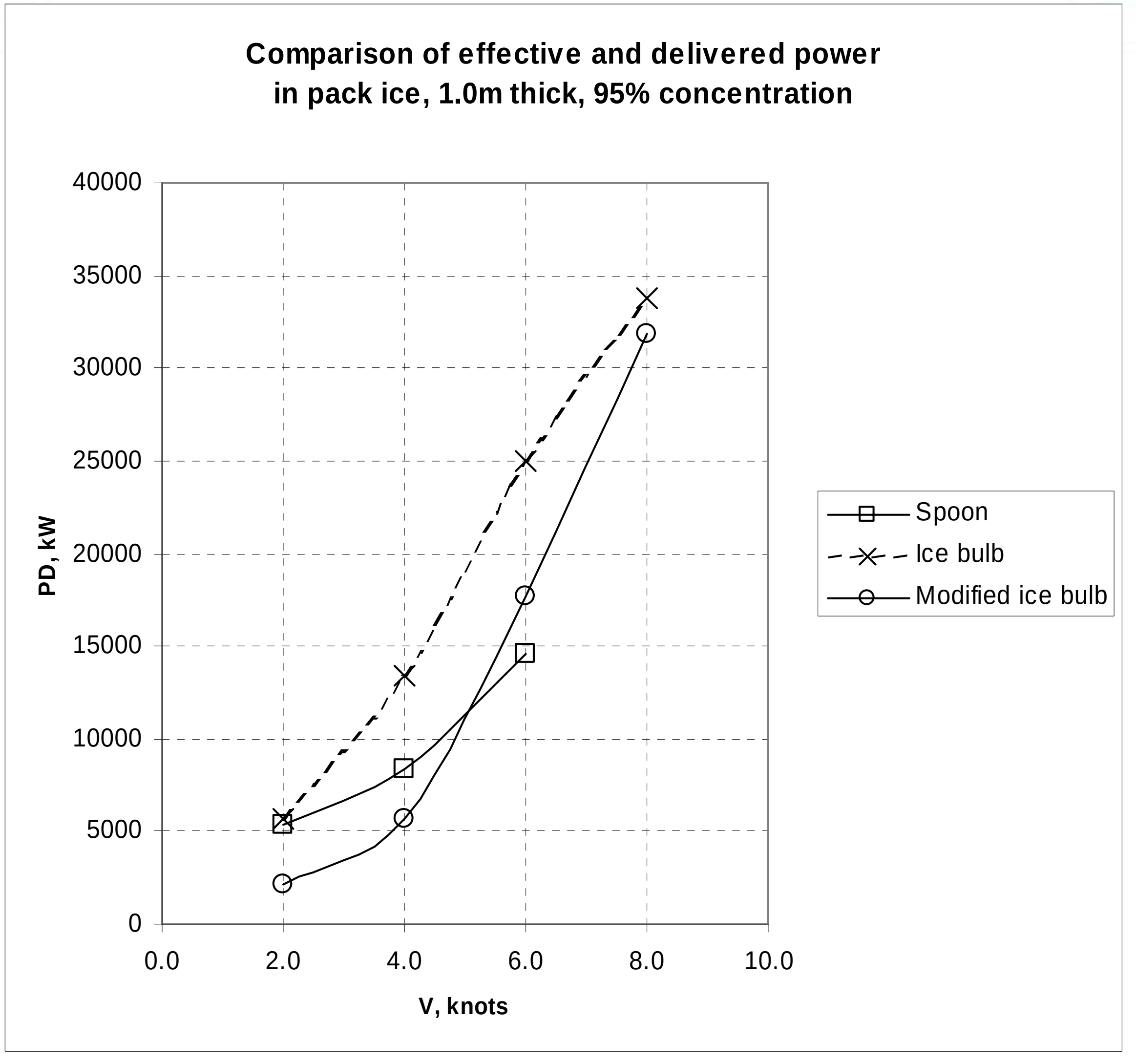
# Delivered Power in Pack Ice

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# Conclusions

- Oil, gas & minerals can be economically transported by ships through ice covered waters.
- Role of model testing becomes important when optimization is required, not just function.
- Optimum design will vary with voyage profile.

# Acknowledgments

Thanks to Bruce Colbourne, Stephen Jones, Bruce Parsons, Don Spencer, Mary Williams, Brian Hill, and Craig Kirby of the Institute for Ocean Technology.

Captain John Van Theil of Canadian Coast Guard for providing some of the pictures we have used.

NRC · CNRC

# Modern RS requirements and methods
# ensuring operating strength of icebreaking propulsion complex

(Requirements to CPP, FPP, Azimuth Thruster strength for ice going vessels and icebreakers)

Main approaches:

## 1. Requirements to propeller blade scantlings proceeding from fatigue and static strength

1.1 Design ice loads for ice milling conditions, (Extreme ice loads, fatigue parameters)
1.2 Stress conditions caused by ice loads
1.3 Permissible stress proceeding from fatigue and static strength
1.4 Assigning of propeller blade scantlings

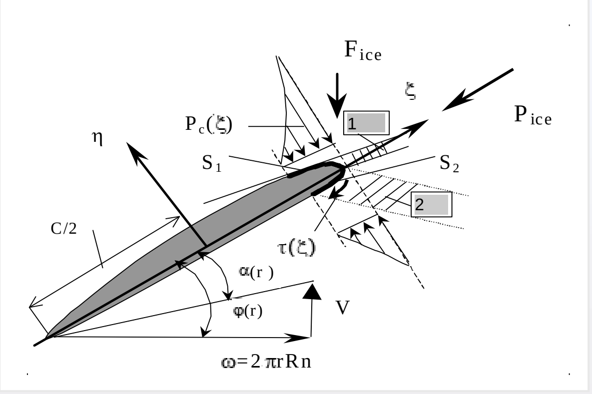
## 2. Requirements to CPP, FPP, Azimuth Thruster elements proceeding from pyramidal and fatigue strength

2.1 Design loads proceeding from fatigue and pyramidal strength
2.1.1 Fatigue: ice loads for ice milling conditions
2.1.2 Pyramidal strength: design ultimate loads caused by blade damage under off- design operating conditions
2.2 Stress conditions for CPM elements
2.3 Permissible strength conditions
- proceeding from fatigue strength;
- proceeding from pyramidal strength
2.2 Assigning and verification of the strength sizes

# 1. Designed ice loads to assign propeller blade scantlings for ice going vessels and icebreakers based on static and fatigue strength

Main approaches and results

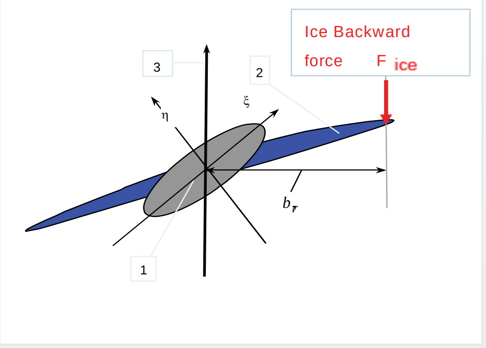
## Ice milling regime is taken as design regime to assign design ice loads.




Principal scheme of ice cutting by propeller blade under milling conditions in the plane of radial section at relative radius r (as per Dr. Belyashov, Dr. B. Veitch, Dr. H. Soininen, Dr. Andryushin)

V - axial velocity of ice block; n - propeller revolution; R - propeller radius; α(r) - attack angle; φ(r) - pitch angle; c - section width; 1 - ice crushing zone; 2 - spalling element

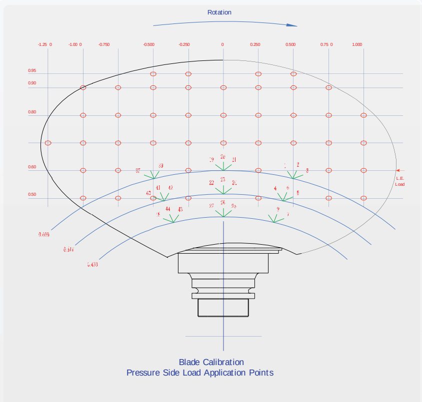
## Consideration of blade bending moment and blade spindle torque is required to assign blade scantlings. For conventional icebreaking propellers ice backward force acting on the blade is taken as main designed load to determine blade bending moment and blade spindle torque.

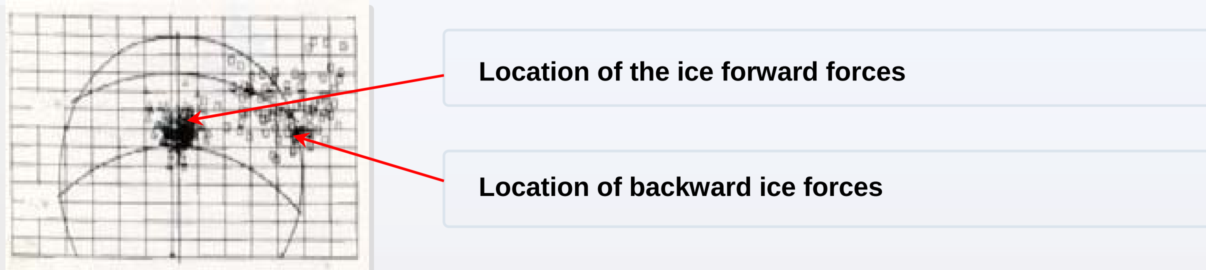



**The scheme of applying of designed ice force**
1. designed root section
2. bade section on relative radius ***r* = 0.8**
3. direction of ships motion

**Propeller blade scantlings are to be assigned proceeding from fatigue and static strength.**
**Operating ice propeller loads are random, therefore maximum ice loads and their distributions are to be determined.**
**In addition the number of the ice load impacts is to be estimated for the fatigue strength estimation.**

## 1. Full scale data (icebreaker "Polar star", ice going vessels "Gudingen", icebreaker "Arctica")

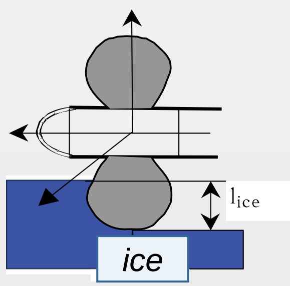


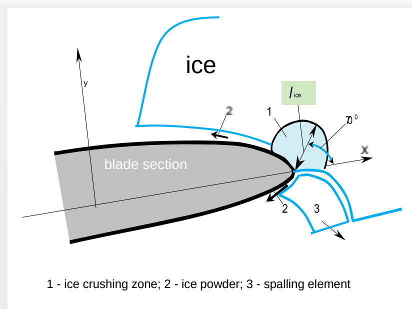

According to VTT full scale results (Koskinen P., Jussila M)

Fig. 5.4 Location of the maximum forward and backwards bending forces on the blade

## 2. Model Test in ice tank




## 3. Design model



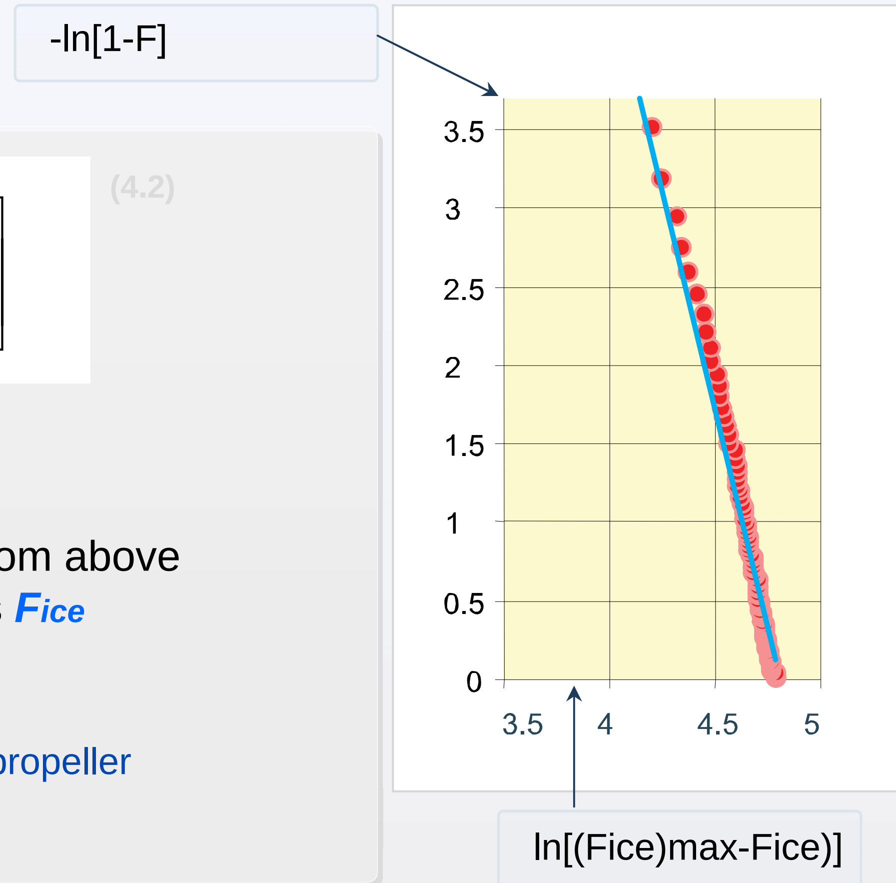
## Distribution of ice operating loads acting on the blade

**Distribution function:**

$$F(F_{ice}) = \exp\left[-\left(\frac{(F_{ice})_{\max} - F_{ice}}{S}\right)^{\alpha}\right] \quad (4.2)$$

Maximum ice force ***(Fice)max*** restricts from above the range of random operating ice loads ***Fice***
S, α - parameters

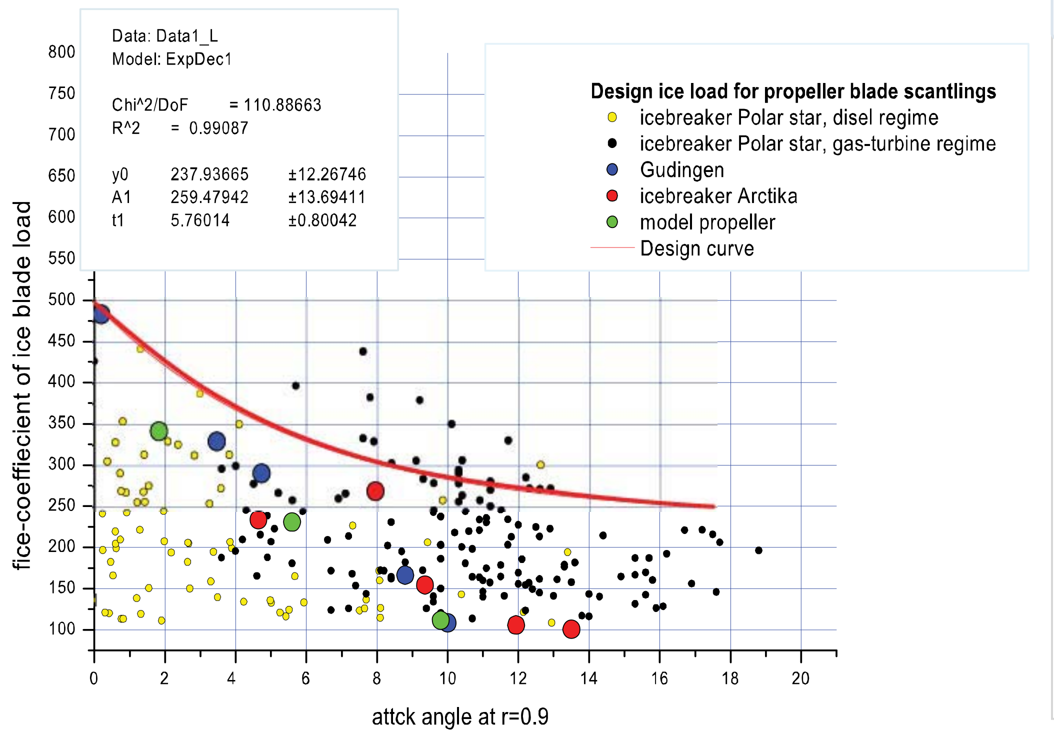
The distribution of ice designed loads on the propeller for «Arctic» icebreaker propeller



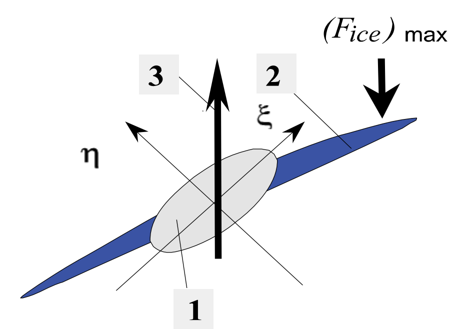
## Maximum backward ice force $(F_{ice})_{max}$ acting on the blade- main designed load to assign blade scantlings, N

$$(F_{ice})_{max} = 10^3 \cdot \left[ 22 + 24 \cdot e^{-0.17 \cdot \alpha(\bar{r}=0.9)} \right] \cdot D^{1,6}\, c_{mean}\, \sigma_{compr}(\bar{r} = 0,8) \qquad (4.2)$$

where ***α(r = 0.9)*** - propeller attack angle at a relative radius ***r = 0.9*** [deg]; Propeller Diameter [m]; ***cmean*** - mean dimensionless blade width at the depth of propeller penetration in ice; ***δcompr(r=0.8)*** - ice compressive strength [MPA], which corresponds to the blade penetration in ice at a relative radius ***r = 0.8***




### Maximum backward ice force on the blade




1. Root section at designed radius
2 Blade section at ***r = 0.8***
3. Direction of ship motion

## The consideration of the blade attack angle for modern propeller strength requirements is required.

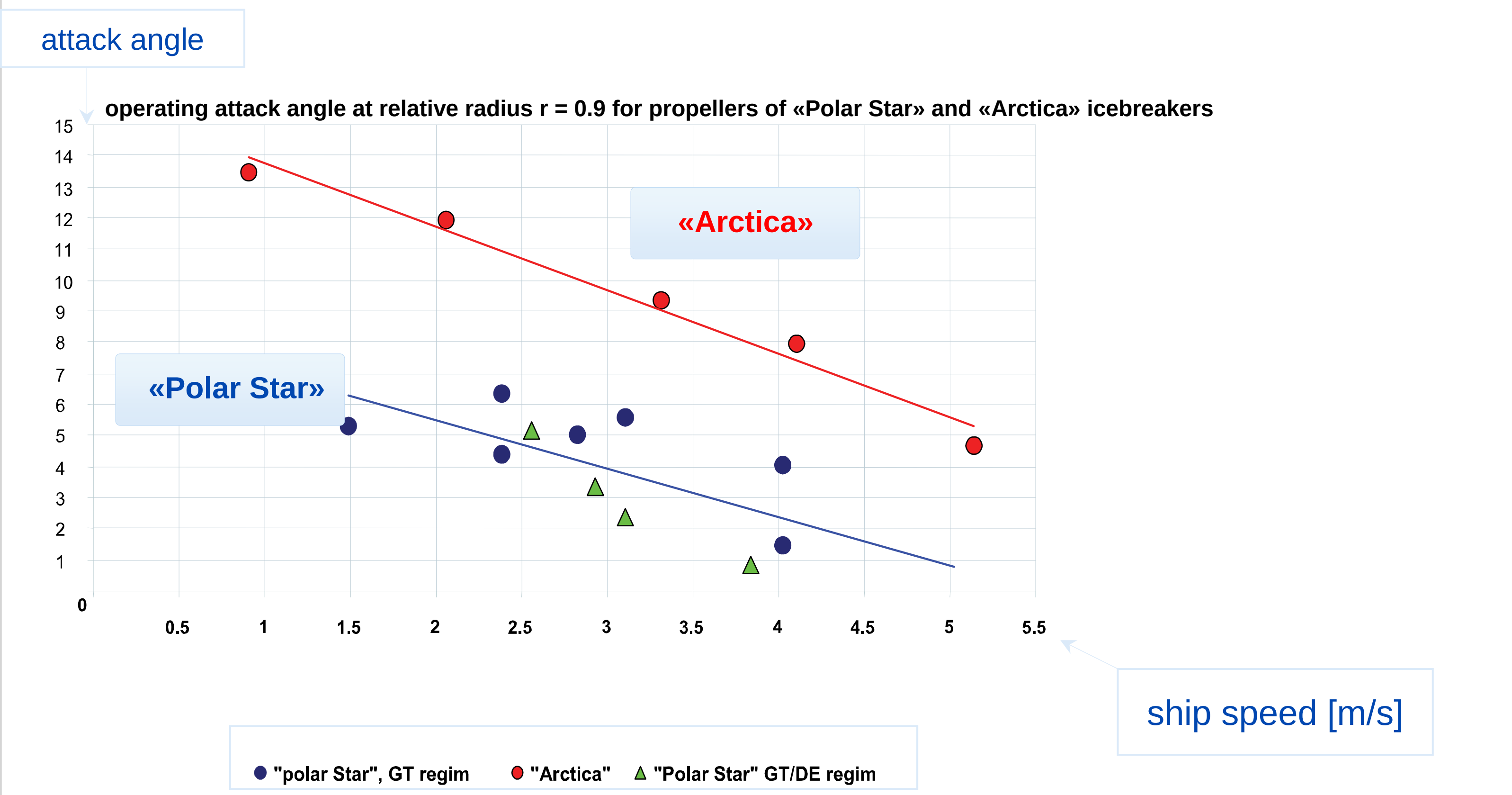



Fig. 5.1 Operating attack angles of the side propellers for «Polar Star» and «Arctica» icebreakers under sever ice conditions

The designed loads according to developed methods are confirmed by the full scale tests of «POLAR STAR» icebreaker, when ice blade stresses were measured under severe ice conditions.

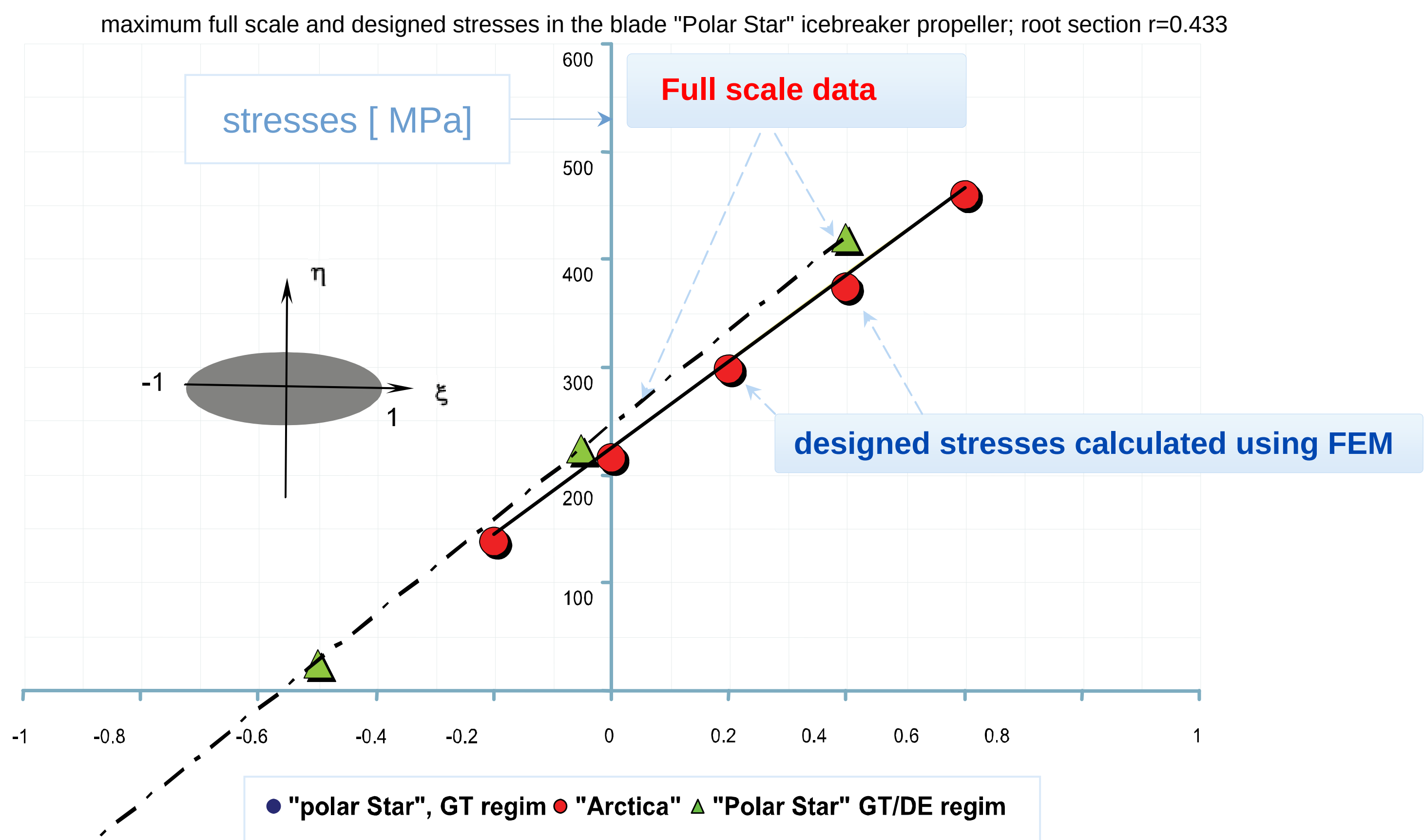



Fig. 5.2 The maximum full scale and designed stresses in the propeller blade of «Polar Star» icebreaker

## 2. Assigning and verification of propeller blade scantlings

Main approaches and results

### 2.1 Blade scantlings of the root sections

**Typical static and fatigue failure of design root section**

Fig. 6.1 **Failure of blade root section of steel icebreaker propeller under single ice load**

**Initiation of a fatigue crack**
**Main design point at root section**

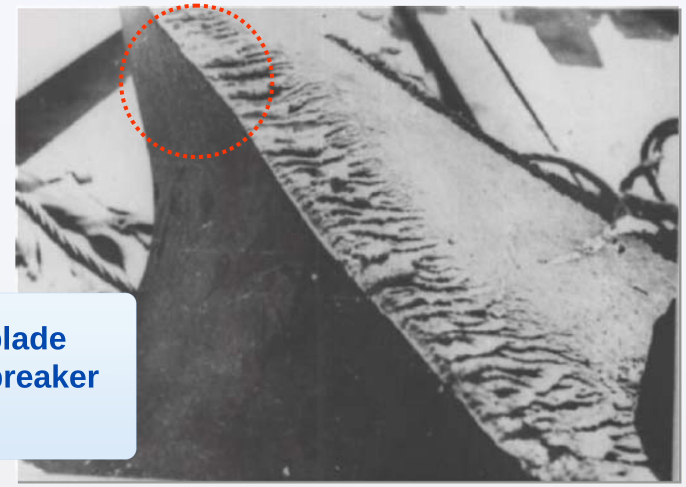


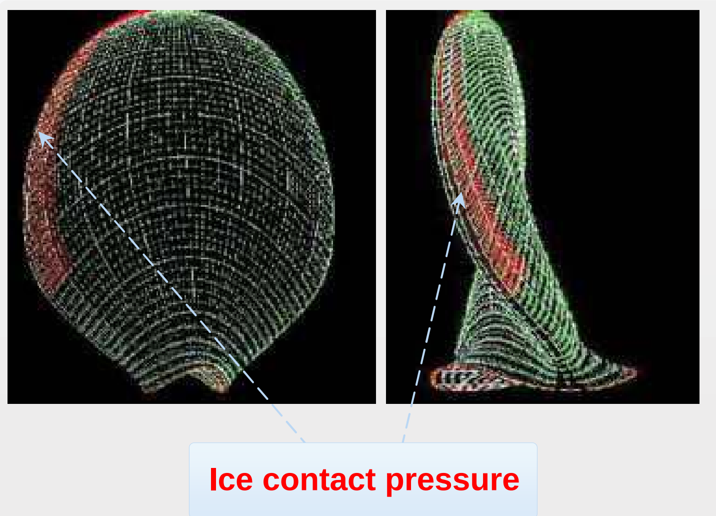
Fig. 6.2 **Fatigue failure of blade root section of steel icebreaker propeller**

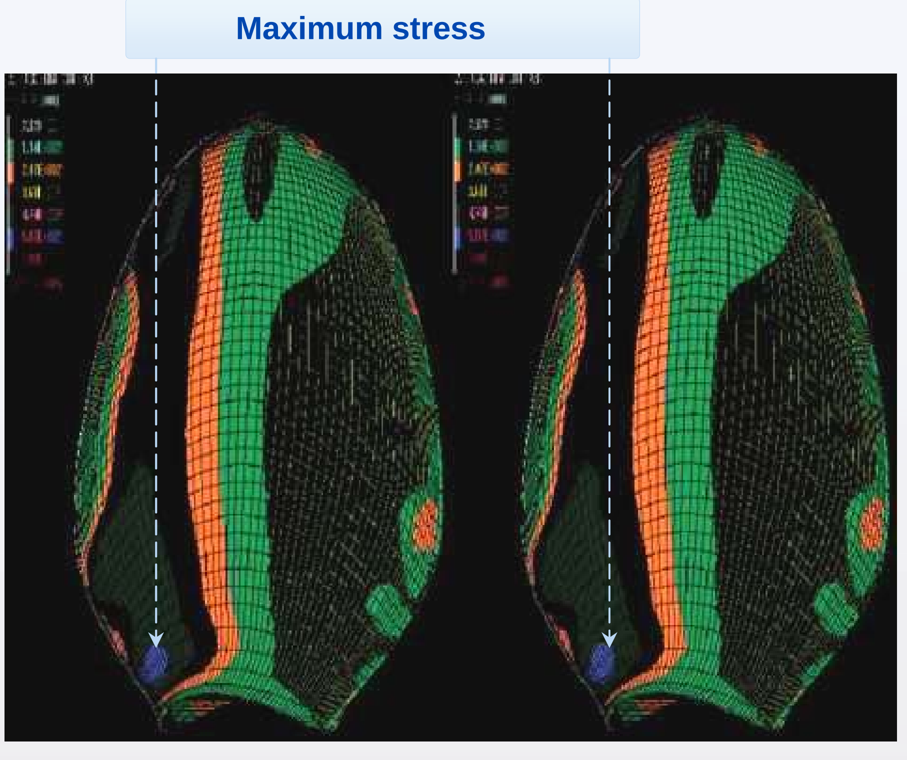
**The most typical blade failure is the breakage of root section, where the fillet surface adjoins to the blade. This section is taken as main design section**

**The blade failures can correspond to failure due to single load and to fatigue failure also. Thus, the blade scantlings are to be determined proceeding from both fatigue and static strength.**

# Typical stress condition of the propeller blade under action of ice backward load applied as ice contact pressure on the leading edge of the suction side (as per Unified Requirement project)

**Macro-finite element model for propeller blade of «Arctica» icebreaker under the action of ice edge load at the suction side.**
**The number of finite element is 8700.Type of element – solid element**



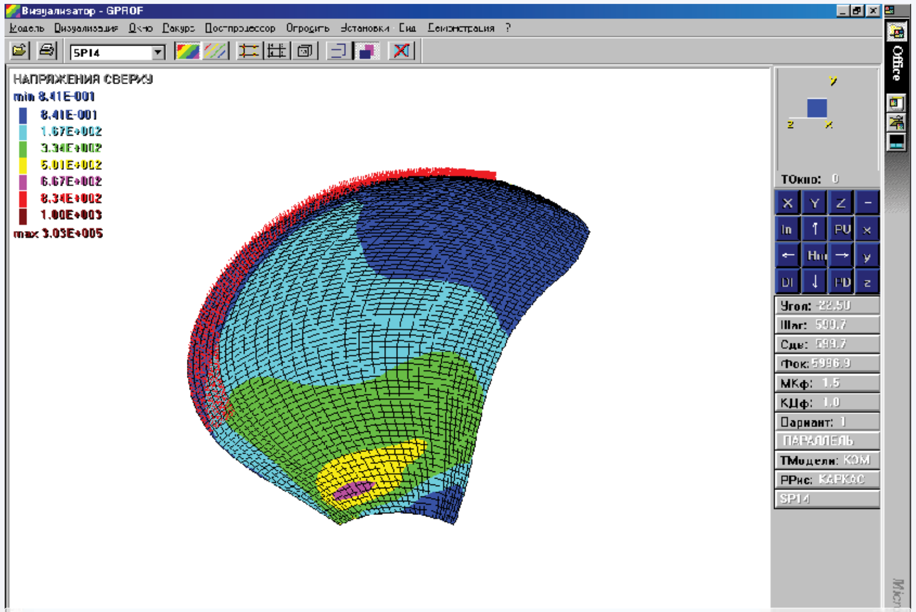
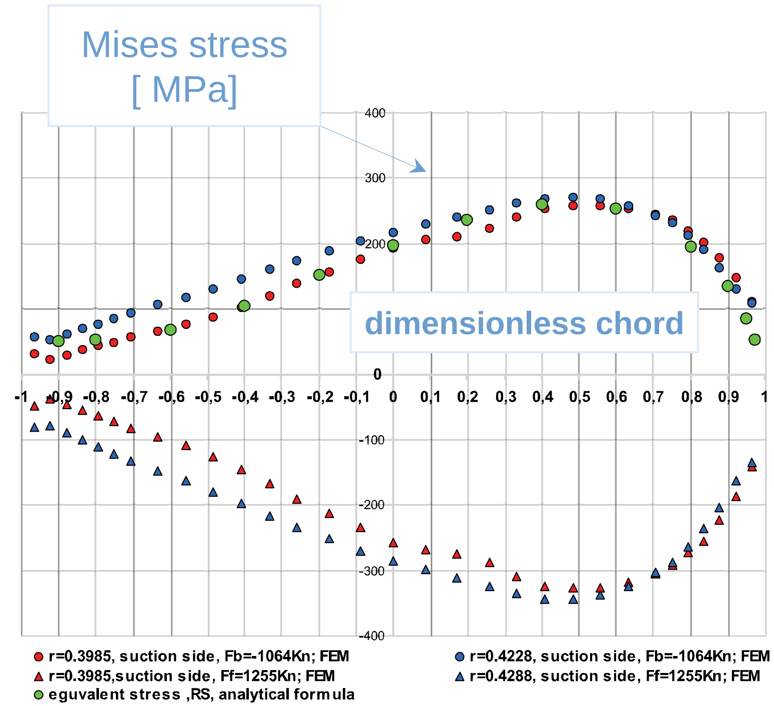


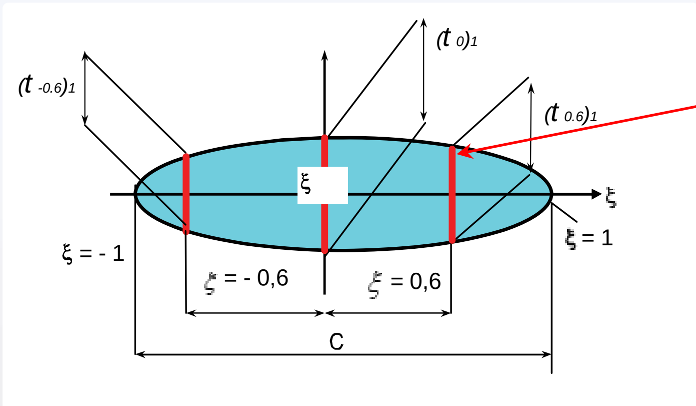

**Fig. 8.1**
**Equivalent stress on suction side of propeller blade caused by ice backward load**

**Fig. 8.2**
**Stress for propeller blade root section under action of ice backward load. Ice going tanker**



**Based on restricted twisting, beam theory and FEM results analytical formulas for maximum stress in the main design point and for blade scantlings of design root section have been developed. The twisting rigidity of a root blade section should correspond to a twisting rigidity of an elliptical section.**

**The scheme of design root section. Main scantlings of design root section.**

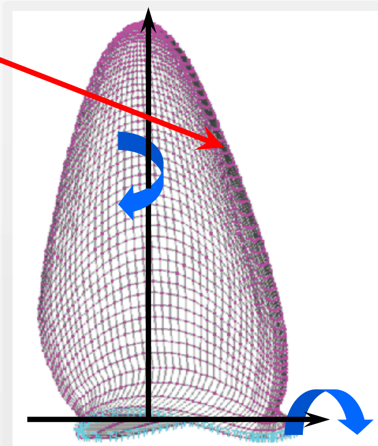


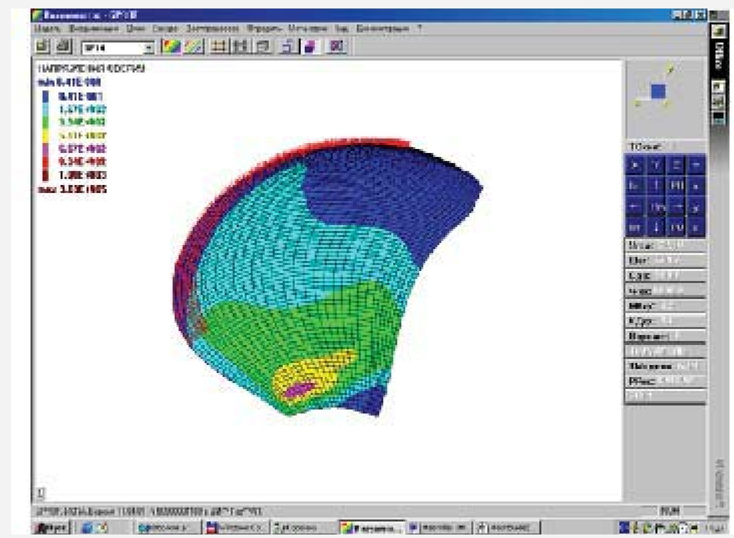

**Main design point (suction side)**

**Stress condition of designed root section under action of ice edge load**

**(Under action of ice bending moment and spindle torque)**

**Ice load**

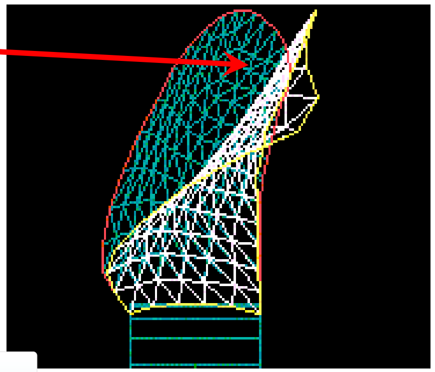




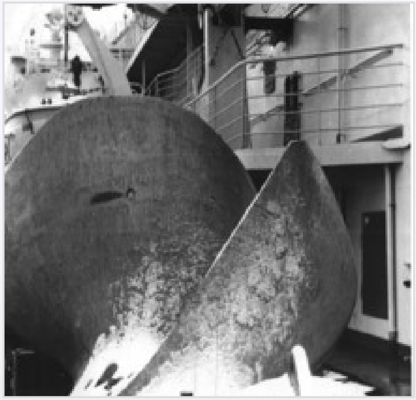
**The blade scantlings of peripheral sections are determined proceeding from a hypothesis of the blade edge bending under action of the ice load.**
**The maximum thickness of blade section at relative radius r=0.6 is taken as a main design strength size of peripheral sections.**

Deformation of peripheral blade section caused by ice load

Ice load

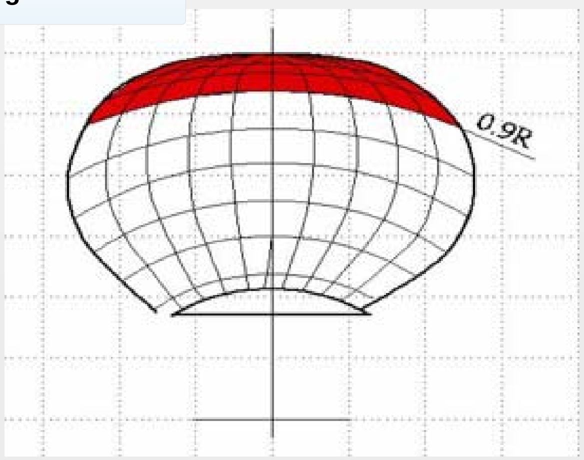


Design model 1



Icebreaker propeller blade failure along "skewed" peripheral blade section

Design model 2




- The analytical formulas for the maximum thickness at relative radius r = 0.6 have been developed.
- Blade scantlings for blade edges have been developed proceeding from operating experience and FEM results.
- Edge blade scantlings is regulated at tip radius r = 1 and at relative radius r = 0.8.
- The thickness of blade edges at relative radius r = 0.8 on a distance 5% of chord length from leading edge should not be less than 50% of a maximum blade thickness on the given radius.

## Principal scheme of assignment and verification of the blade scantlings

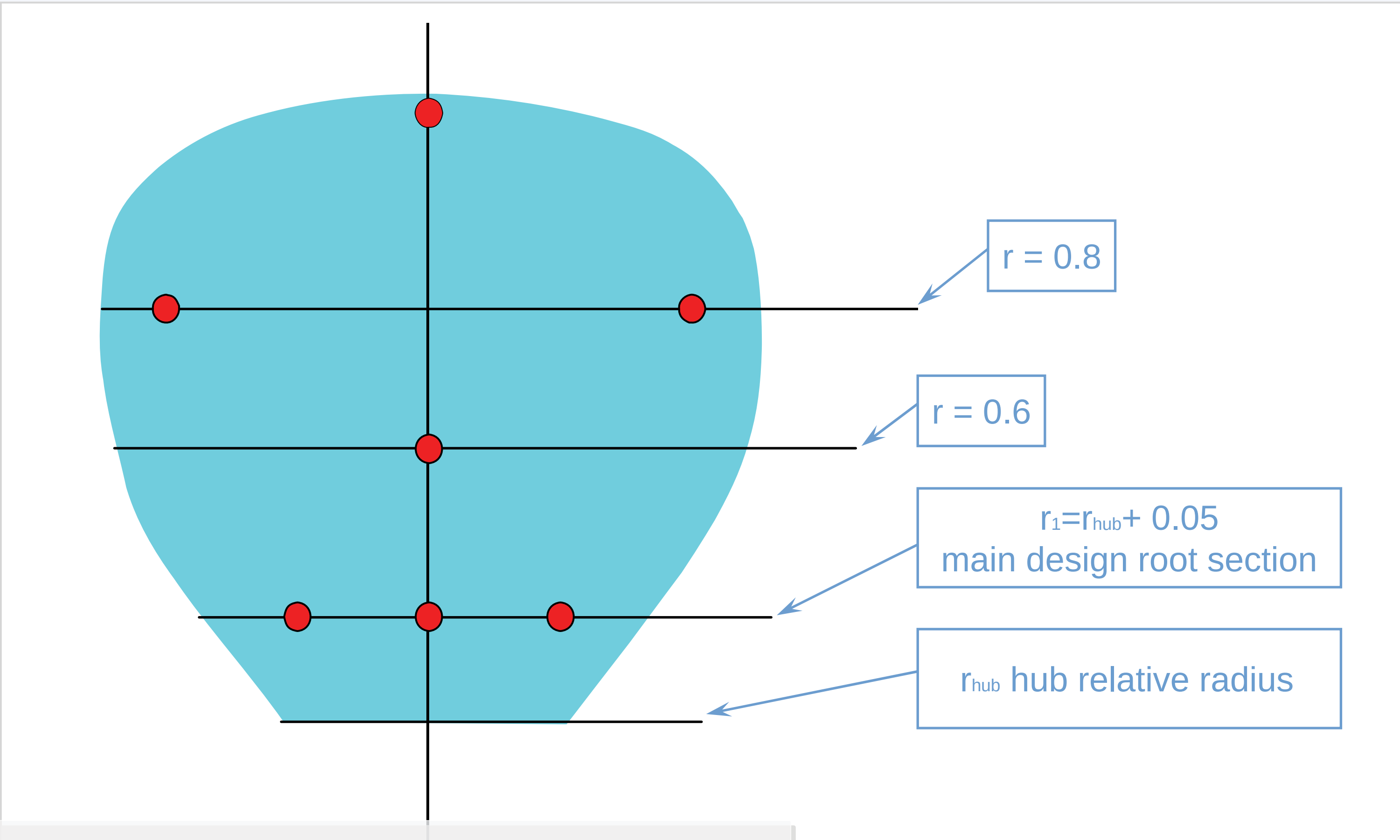



**Fig. 11.1**
**Principal scheme of assignment and verification of the blade scantlings**

## Determination of permissible stress to assign blade strength scantlings

**The consideration of residual stresses and manufactured defects in the blade casts is required.**

**The development of the requirements for permissible defect size is important problem to ensure propeller reliability. The permissible size of the defect is determined proceeding from a condition of non-propagation of defect as macro-crack.**

### Designed permissible stress to assign blade scantlings

$$\sigma_{perm} = \min\left[(\sigma_{perm})_S, (\sigma_{perm})_f\right]$$

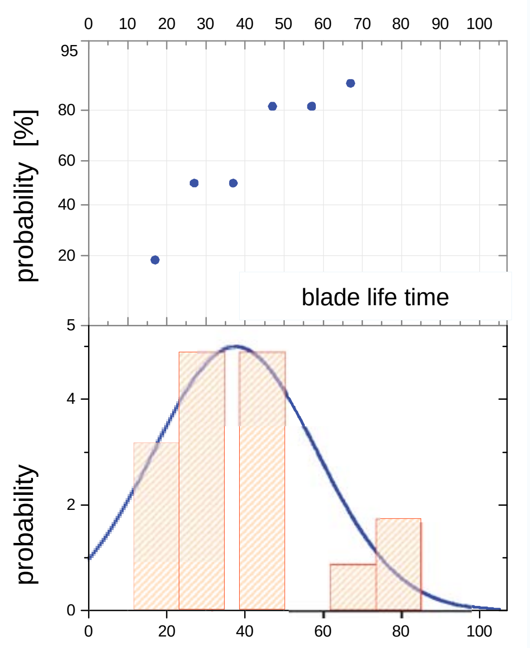
$(\sigma_{perm})_S$ - static permissible stress

$(\sigma_{perm})_f$ - fatigue permissible stress

**Fatigue crack**

### Permissible stresses proceeding from the fatigue strength

Fatigue permissible stresses are assigned proceeding from operating ship life time




Blade life time distribution Side propeller. Steel. Hardening

# Requirements to CPM, FPP and Azimuth Thruster elements strength for ice going vessels and icebreakers

## 1. Requirements to elements strength proceeding from pyramidal strength

Assurance of the pyramidal strength of propulsion complex (PC) elements means that if the blade has broken due to off-design ice load the elements located in the flow of lines of force are to remain intact and ensure PC operation within given technical conditions. The breakage of the blade means bending of the blade due to plastic deformation or splitting the blade into separate parts.

- Requirements to ultimate blade damage load
- Requirements to stress conditions for CPM, FPP and Azimuth Thruster elements (including stress concentration zones) under action of ultimate blade damage load
- Requirements to permissible stress conditions
- Requirements to permissible (critical) plastic deformation which realized in stress concentration zones of elements located in the flow of lines of force

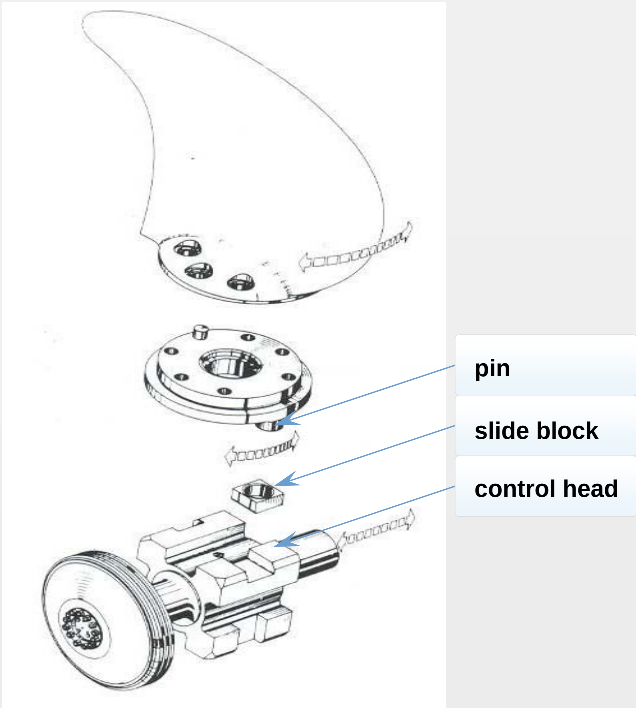
## 2. Requirements to elements strength proceeding from fatigue strength

- Requirements to blade ice load for ice milling conditions
- Requirements to fatigue permissible stresses

## 3. Requirements to materials of elements

## Main elements located in the flow of lines of force

**CPM:**



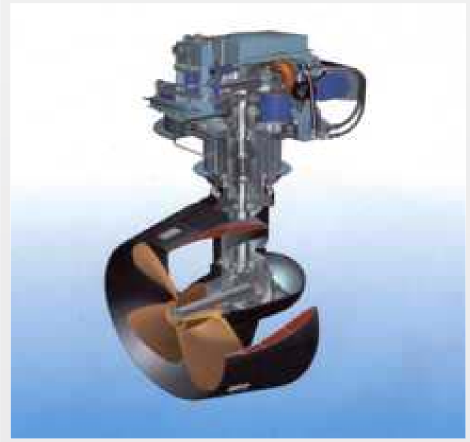

Type construction of the CPM

1 - control head
2 - slide block
3 – pin

**Azimuth Thrusters:**

**“Azipod”**

**“Aquamaster”**



Blade bolt; propeller shaft; bearing; shafting; gear; flange connections…

## Requirements to the ultimate blade damage load

**(main approaches)**

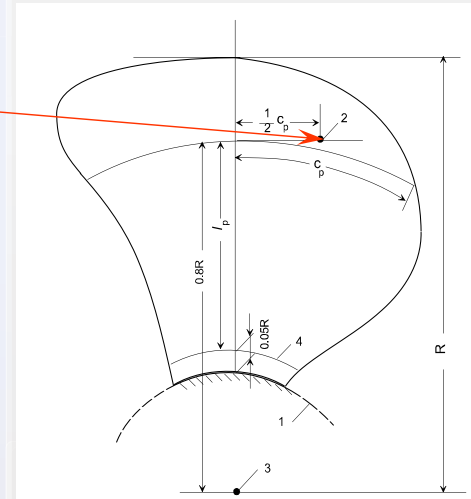
1. Location of ultimate blade damage load

2. The breakage of the blade means
   the plastic deformation or splitting the blade into separate parts
   for plastic hinge

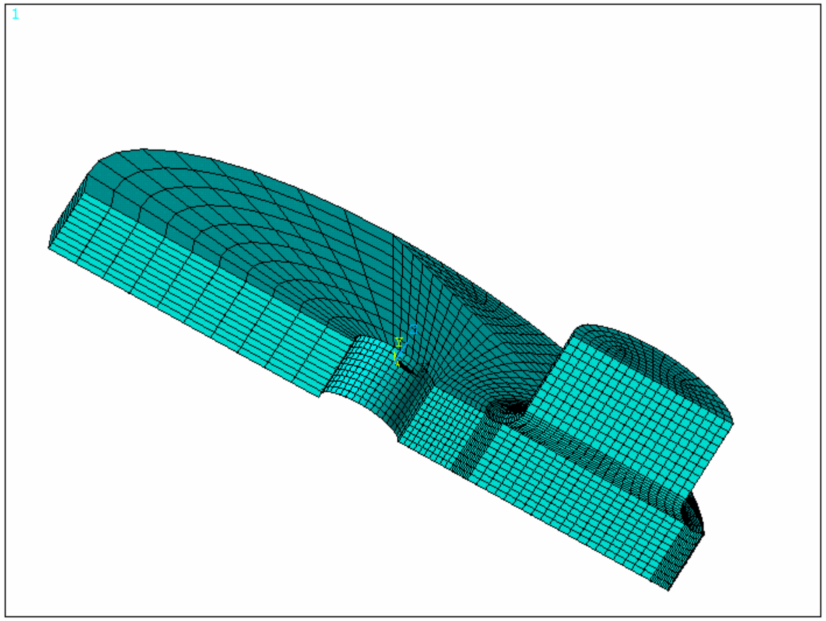
3. Design root section at $r_1 = r_{hub} + 0.05$

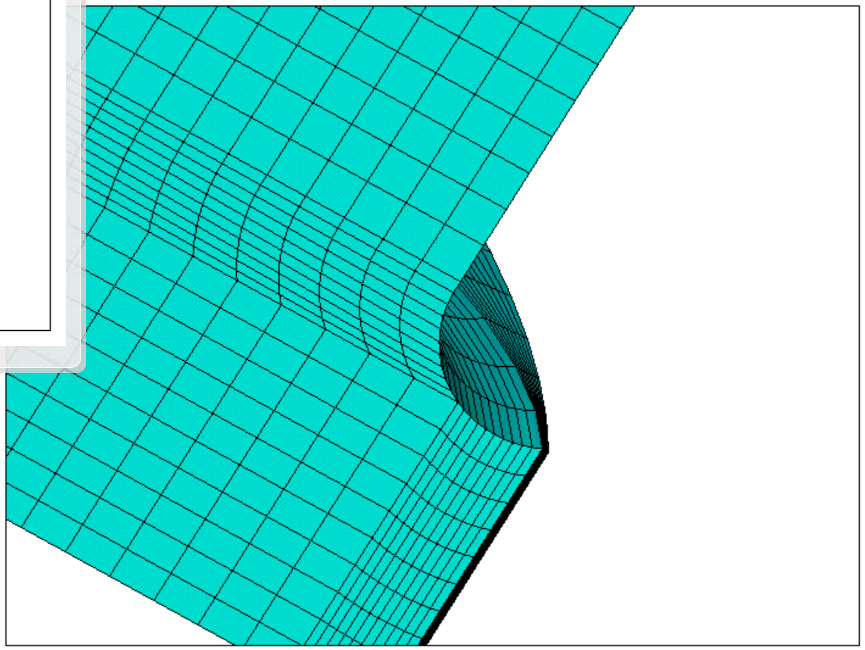
4. Ultimate blade damage
   load caused by bending and spindle torque

5. Damage criteria – critical plastic macro deformation $\varepsilon_{cr}$



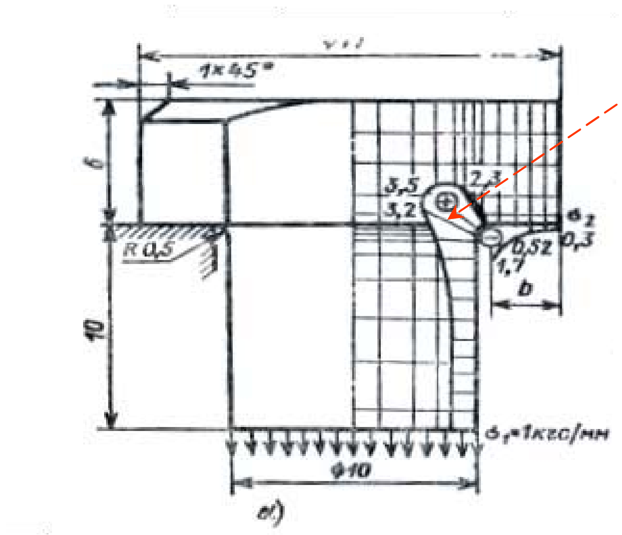

## FEM MODEL for CPM element





## Requirements to plastic deformation in stress concentration zones of elements, located in the flow of lines of force

**Concentration zone**




$$\varepsilon \le k_{safety} \cdot \varepsilon_{cr}$$

**martensitic steels**

$$\varepsilon_{cr} \cong 6.4 \frac{\sqrt{KV}}{(\sigma_{0,2})_{\min}} \frac{1}{t^{0,5}} \ln(1+\delta_{\min}) \text{ at } (\sigma_{0,2})_{\min} \ge 0,6(\sigma_u)_{\min}$$

**austenitic steels**

$$\varepsilon_{cr} \cong 1,3 \left( \frac{(KV)^{0,2}}{(\sigma_{0,2})_{\min}^{\ 0,4}} \right) \frac{1}{t^{0,2}} \ln(1+\delta_{\min}) \text{ at } (\sigma_{0,2})_{\min} \le 0,6(\sigma_u)_{\min}$$

where $(\sigma_{0,2})_{\min}$ - minimum yield strength (or minimum 0,2 per cent proof stress) guaranteed by the delivery specifications, MPa;
$(\sigma_u)_{\min}$ - minimum value of the ultimate tensile strength guaranteed by the delivery specifications, MPa;
**KV** - minimal required impact energy for Charpy V-notch test, J;
$\delta_{\min}$ - minimum relative elongation of the standard sample required by the technical specifications;
**t** - attributed thickness of the element, m;
$k_{safety}$ - safety factor, caused by probability statistical spread of the physical and mechanical characteristics of material of detail casts.


ARCTIC PRINCESS
PRACTICAL DESIGN OF LNG CARRIERS IN LOW TEMPERATURE ENVIRONMENTS
GRAND ELENA
May 2007

# Contents

# *Introduction*

## What is the optimum specifications for Winterisation ?

Environment
Voyage profile
Owner's policy

- Environmental impact
- Crew burden
- Damage Risk

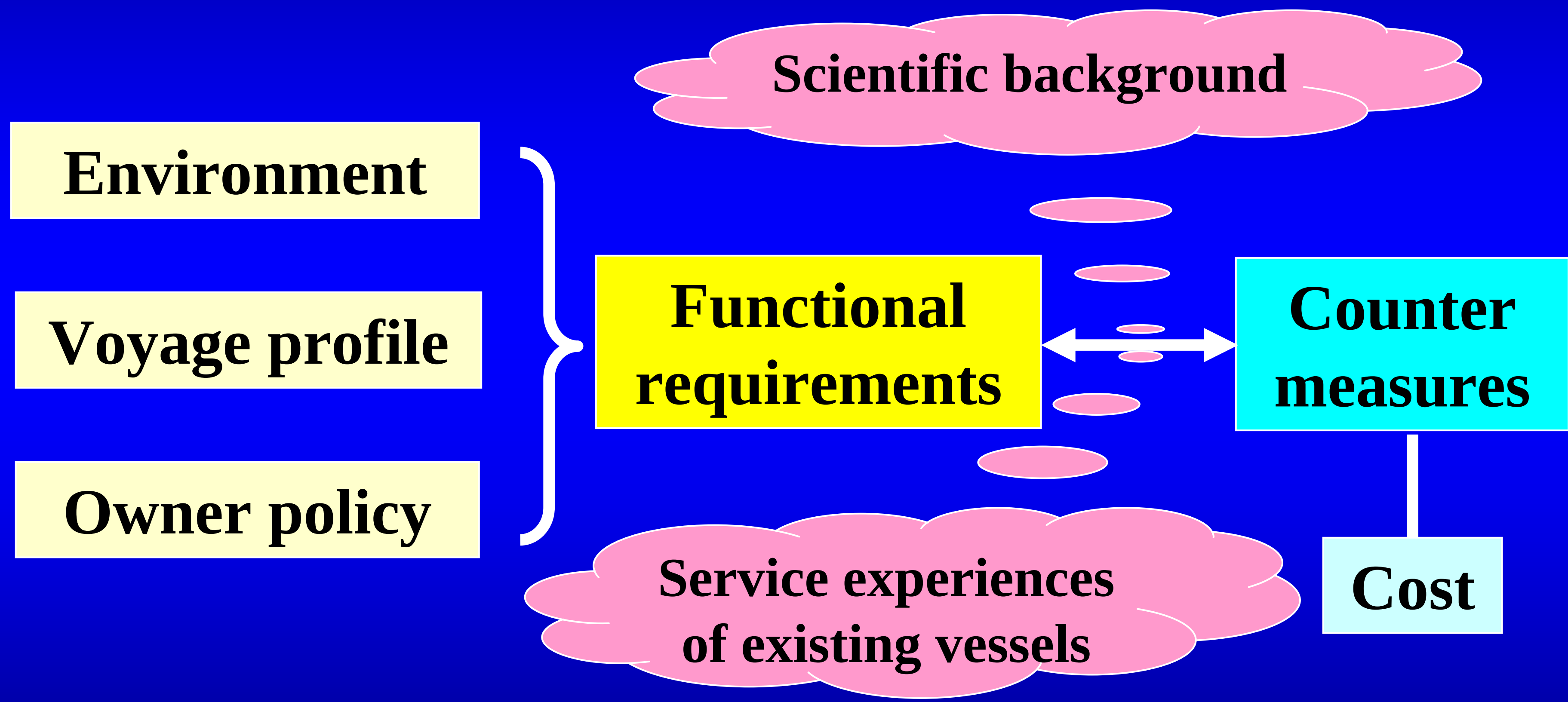
**Balanced**

Winterisation
Specifications

- Maintenance cost
- Initial cost

# *Introduction*

## What is the optimum specifications for Winterisation ?




**Deep understanding of each item is necessary to decide Winterisation Specifications**

MITSUBISHI HEAVY INDUSTRIES, LTD.

# *Classification of winterisation*

## Varieties of Environment and voyage profile

**Air Temp.**

Lowest Mean Daily Average temperature (Extreme)

| | | | |
|---|---|---|---|
| 0℃～-10℃ (-20℃～-30℃) | -10℃～-20℃ (-30℃～-40℃) | -20℃～-30℃ (-40℃～-50℃) | Under -30℃ (Under -50℃) |

**Sea ice**

| | | |
|---|---|---|
| Open water (no sea ice) | Thin First Year Ice (Less than 70cm) | Medium/Thick First Year Ice<br>Multi Year Ice |

Risk Up

**Encounter frequency**

| | | |
|---|---|---|
| Very rare to call on terminal in cold climate | Cold climate terminal surrounding only | All navigating route is cold climate |

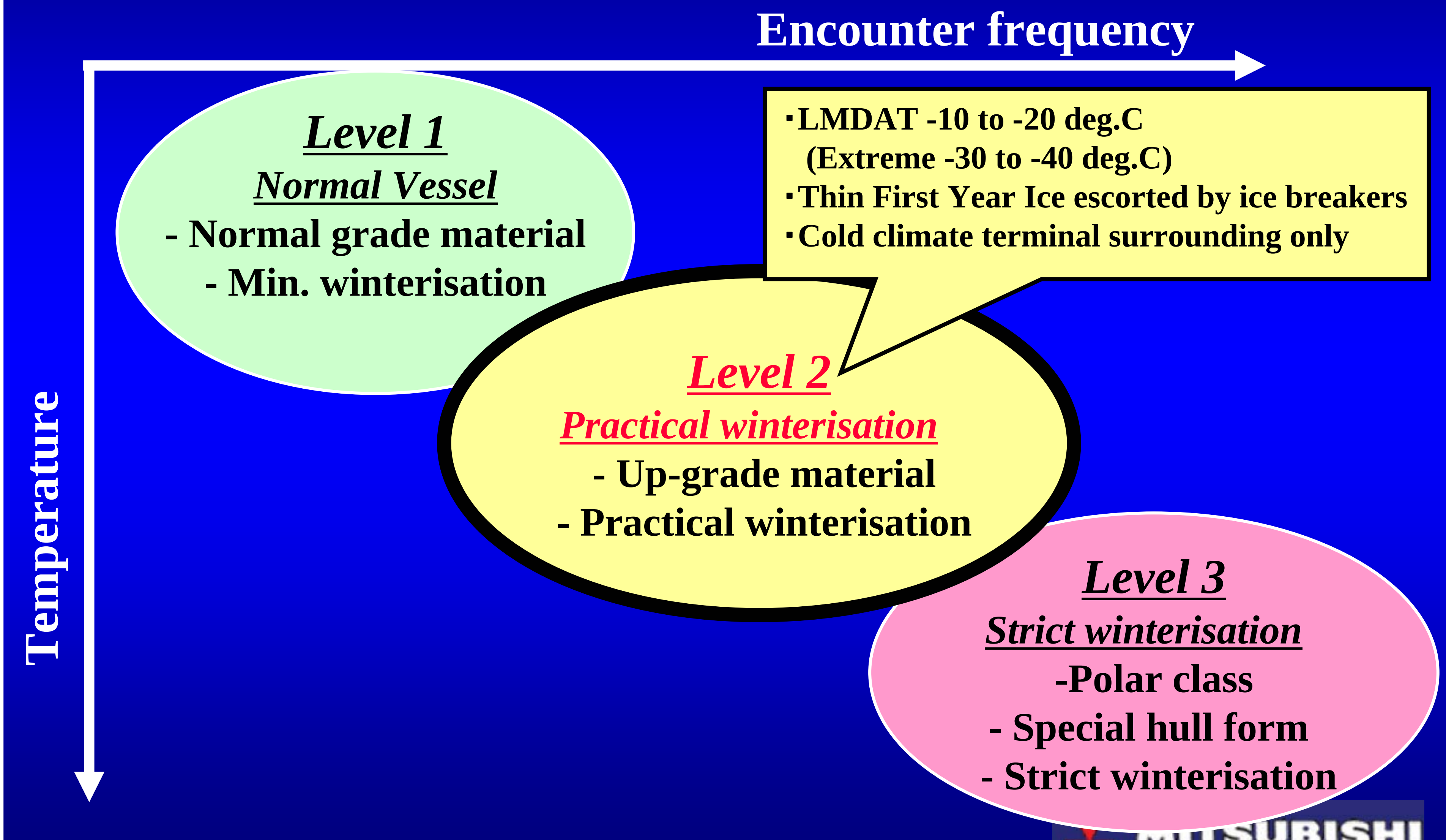
Risk Up

Big challenge
Purpose built

| Level 1 | Level 2 | Level 3 |
|---|---|---|

# *Classification of winterisation*

## Portfolio of Winterisation

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# Classification of winterisation

## Definition of air temperature

### Classification society

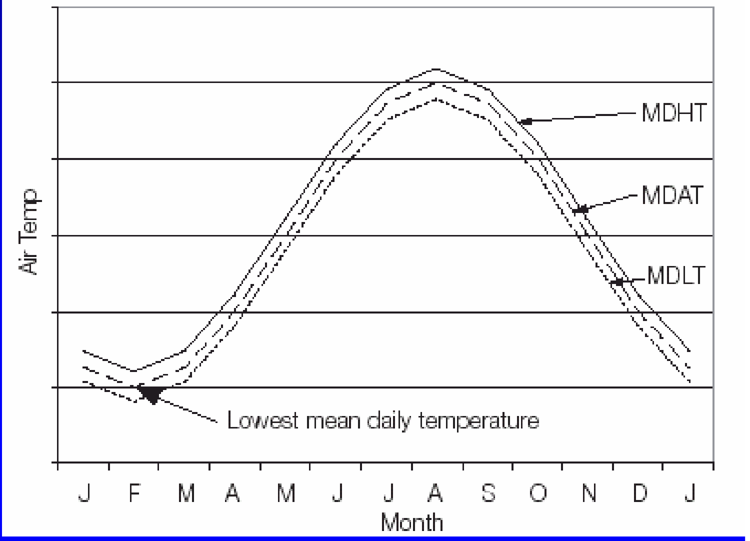
**MDAT: Mean Daily Average Temperature**

**Mean: Statistical mean over a minimum 20 years**

**Average: average during one day and one night**

**LMDAT: Lowest MDAT ⇒ DAT for Hull steel**

**Lowest: Lowest during the year**




### Russian standard

(Structural standards and Regulations 2.01.01-82, Moscow, 1997)

**The coldest day with probability of not exceeding 0.98**

**The coldest five day period with probability of not exceeding 0.98**

- **Sampling of air temperature of the coldest day and of the coldest five day period for 30-50years during 1925 - 1975.**
- **Values of a given probability taken from the air temperature distribution integral curves.**

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# *Classification of winterisation*

**Environment conditions and voyage profile selected for Level 2 "Practical Winterisation" can cover the trade to the existing or planning terminals in cold climate.**

- **LMDAT -10 to -20 deg.C (Extreme -30 to -40 deg.C)**
- **Thin First year ice escorted by ice breakers**
- **Cold climate terminal surrounding only**

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# *Hull Structure design*

## Key points for low temperature environment

1. **Ice Class selection**
2. **Safe Speed analysis considering hull form (lines) and optimization of ice reinforcement**
3. **Fatigue design considering dedicated trading route**

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# Hull Structure design

## Strength against ice pressure

**Ice class concept (ex. Finland/Sweden)**

**Designed based on ice class rule.**
**Ice class is assigned to vessel.**
**Operation is restricted by only Ice Class and DWT.**

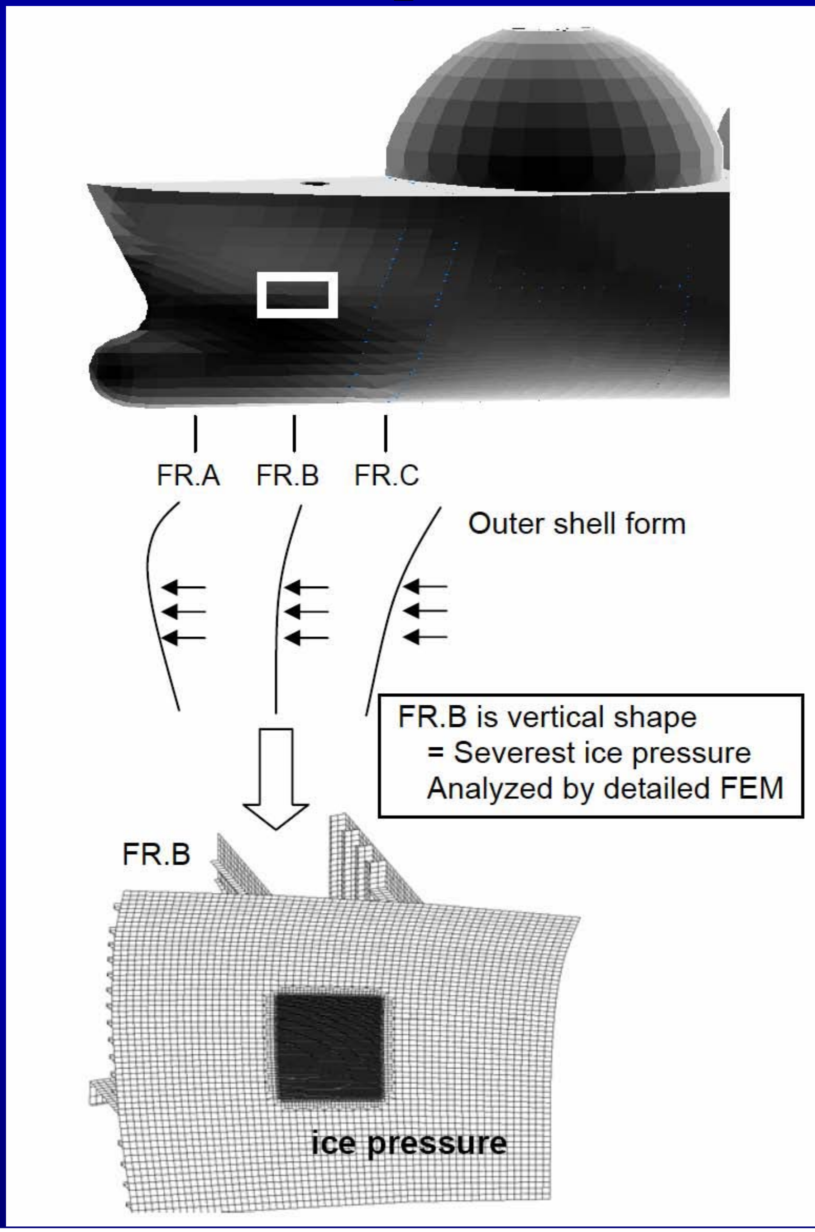
**Ice passport concept (Russia)**

**Operation is restricted by Ice passport.**
**(Safe speed, attainable speed, admissible speed)**

**Safe speed : Maximum speed when the hull/ice interaction in the channel does not result in the hull damage.**
**Attainable speed: Maximum speed the vessel can develop and maintain using the full power of main engines.**
**Admissible speed: Maximum speed corresponding to either safe or attainable speed whichever is lower.**

# Hull Structure design

## Optimization of ice reinforcement




Maximum Ice load around **shoulder area**

Structural capability by **non-liner FEM**

↓

**Safe speed**

↓

Specific area limiting safe speed

↓

Reinforcement around the area

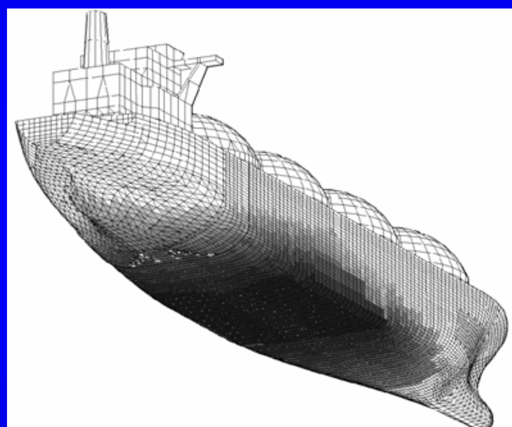
↓

**Better "as built" strength**

# *Hull Structure design*

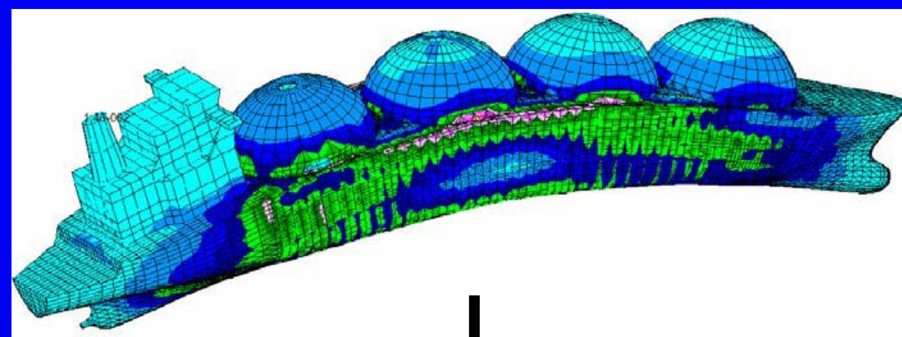
## Fatigue Assessment

**LNG Carriers in cold regions often sail in relatively severe wave environments, therefore fatigue design should be based on the dedicated sailing route of the ship**

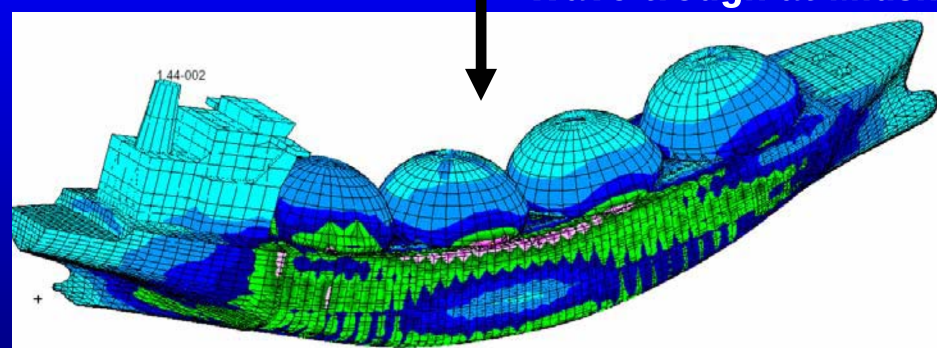


Wave Pressure distribution

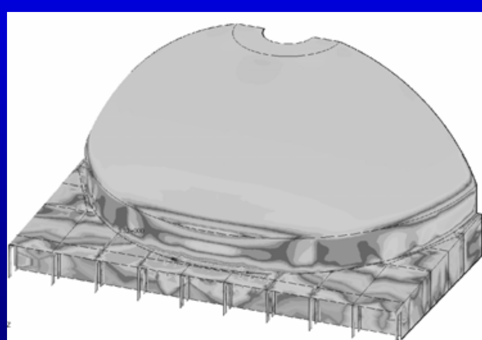
Wave crest at Midship



Wave trough at Midship



Whole ship F.E.Analysis
(Deformation non-scaled)



Assessment of structural detail

# Outfitting design

1. Practicable design of Anti-icing measures for Ballast tank and Hydrant
2. Major winterisations applied to LNG carrier
3. Material selection for deck equipment

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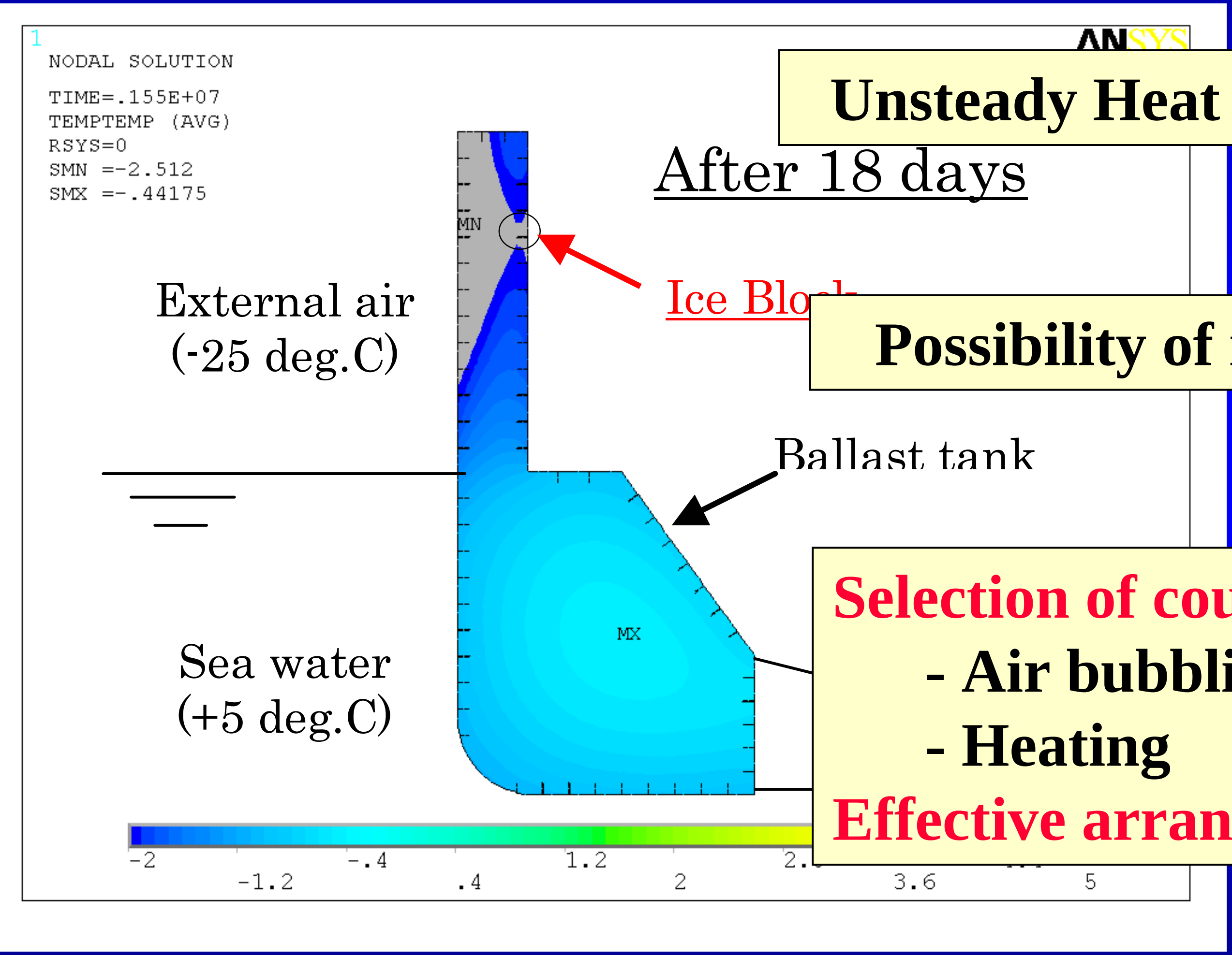
# *Outfitting design*

## Anti icing functions (Ballast tank)




**Unsteady Heat Transfer Analysis**

**Possibility of ice formation**

**Selection of counter measure**
- **Air bubbling/Recirculation**
- **Heating**

**Effective arrangement**

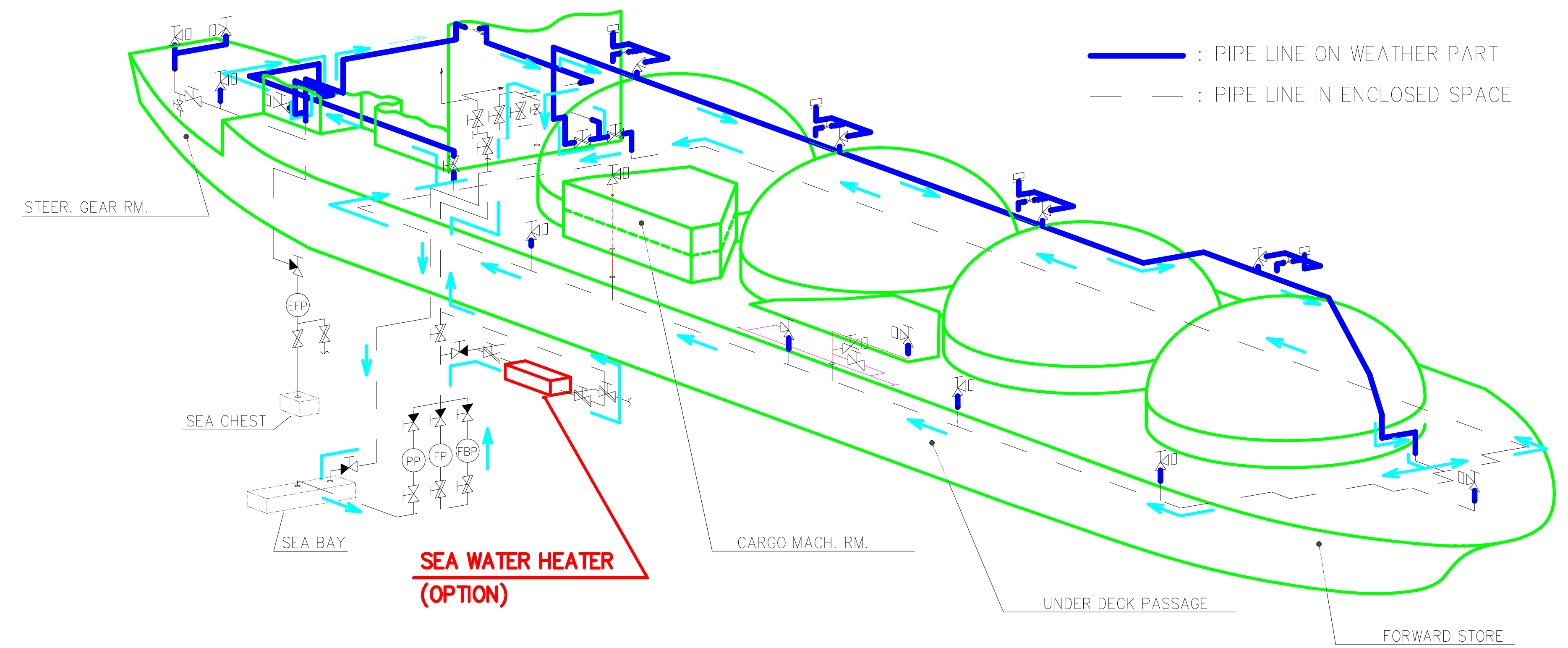
# Outfitting design

## Anti icing functions (Hydrant)

**Functional requirement**
- Hydrant lines pressurized
- Water curtain not freeze

Total Heat balance cal.

Counter measures



# *Outfitting design*

## Material selection for deck equipments

- There is no clear rule/guidance for material selection.
- Practical design method should be established.
- Consistency with service experience, practicability in production should be considered. (Strict requirement may be easy but may not be available?)

Proposal

Example of Anchor material

***Calculation of equivalent toughness by fracture mechanics***

**Base Temperature**

**(1) Service experience**

**"Grade U2"**
**Charpy energy 27 J/cm2**
**at test temperature 0 deg.C**

→

**Design temperature**

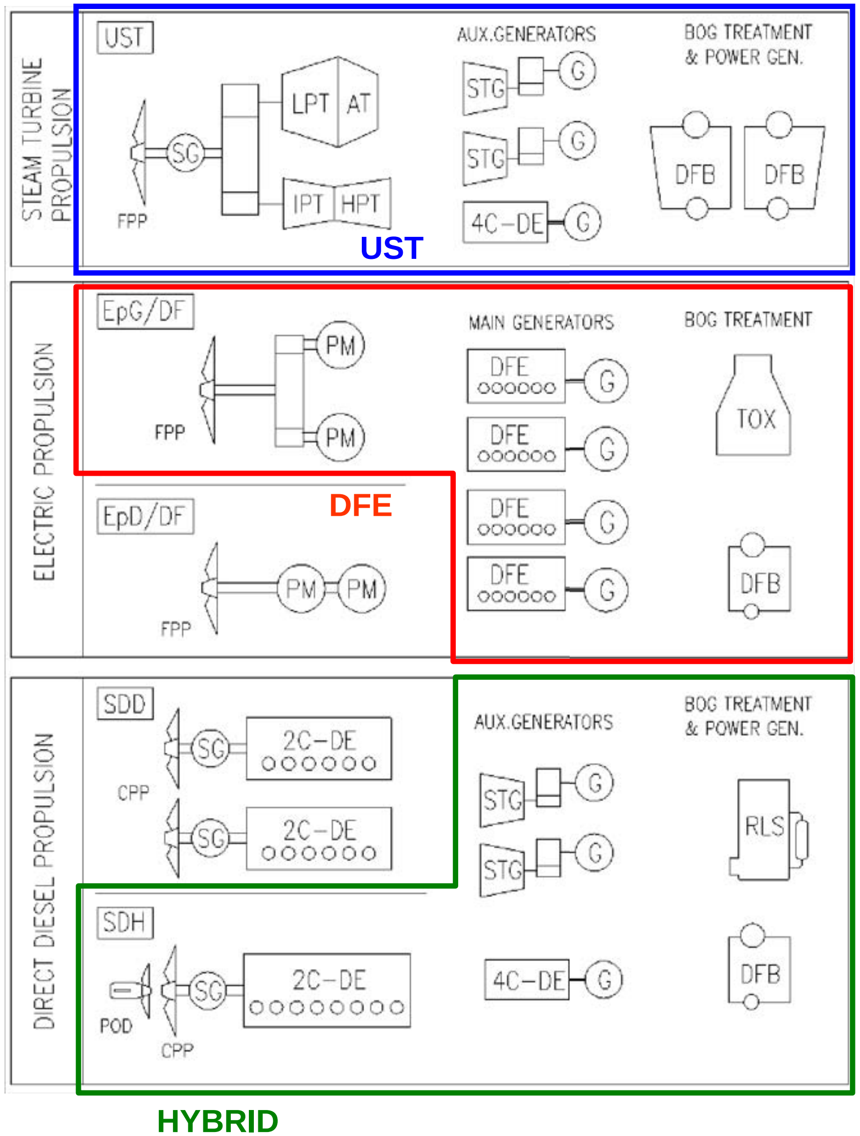
**(2) Required toughness**

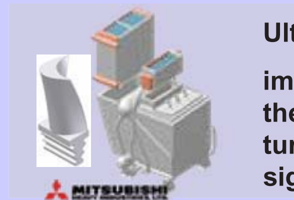
**Charpy energy 50 J/cm2**
**at test temperature 0 deg.C**

→

**Grade U3**

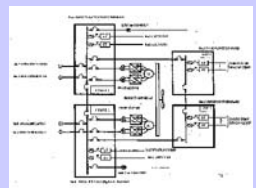
# Propulsion Plant line-up



**Ultra Steam turbine …**

**improves fuel consumption <u>more than 15%</u>, by the application of re-heater, high efficiency turbine and improved steam condition, without significant changes from the conventional steam turbine plant.**



**Electric propulsion …**

★ **Less maintenance**

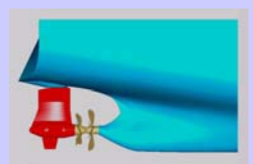
★ **High redundancy**

**Power plant …**

★ **High efficiency (improvement more than 20%)**

★ **High redundancy (Multi-engine plant)**

★ **Drastic fuel cost reduction by using LNG**

**Diesel propulsion**

★ **Conventional two stroke diesel direct**

★ **High efficiency (Improvement more than 30%)**

★ **High redundancy**

**BOG treatment system**

★ **No boil off loss of cargo**

★ **High efficiency by heat recovery**

★ **UMS operation**

**Key factor : Navigation in ice channel, Fuel consumption, impact on environment**

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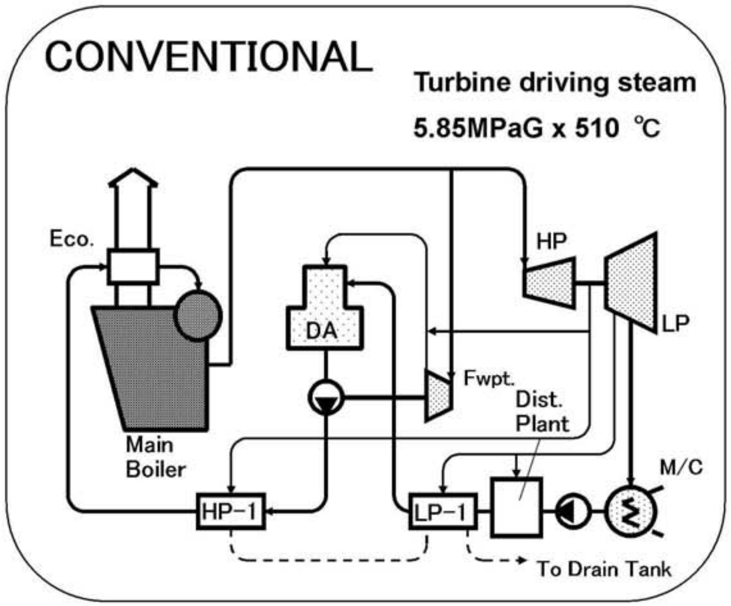
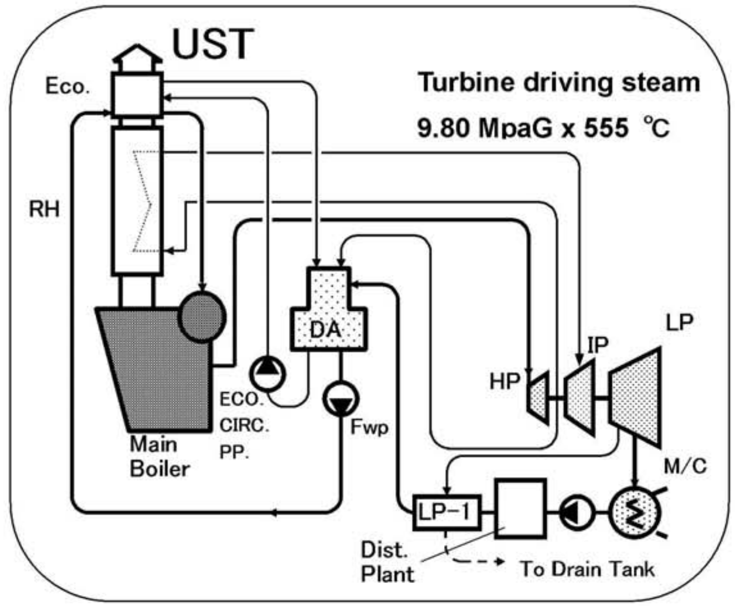
# ULTRA STEAM TURBINE PLANT

| | CONVENTIONAL PLANT | UST |
|---|---|---|
| BOILER STEAM CONDITION (at superheater outlet) | 6.0 MPaG X 515 ℃ | 10.0 MPaG X 560 ℃ |
| STEAM FLOW | BOILER→HP TURB→LP TURB | BOILER→HP TURB→REHTR →IP TURB→LP TURB |
| FLANGE RATING | ANSI 900 LB | ANSI 2500 LB |

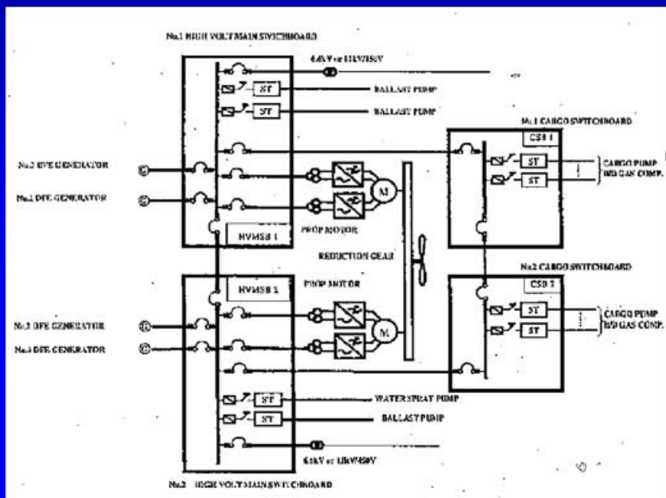


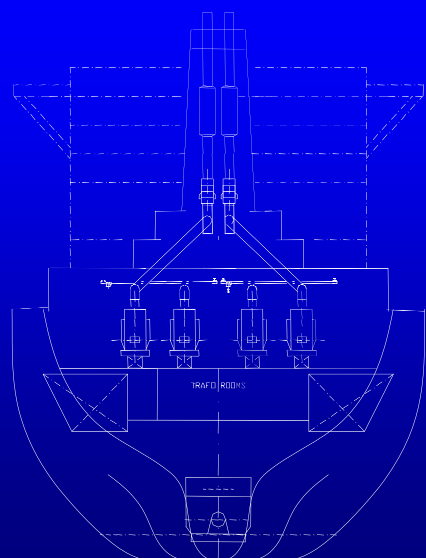

**FOC 15% Less**



# Alternative Propulsion

## Electric propulsion with DF engine plant : DFE




### Electric propulsion

- ★ Less maintenance
- ★ High redundancy

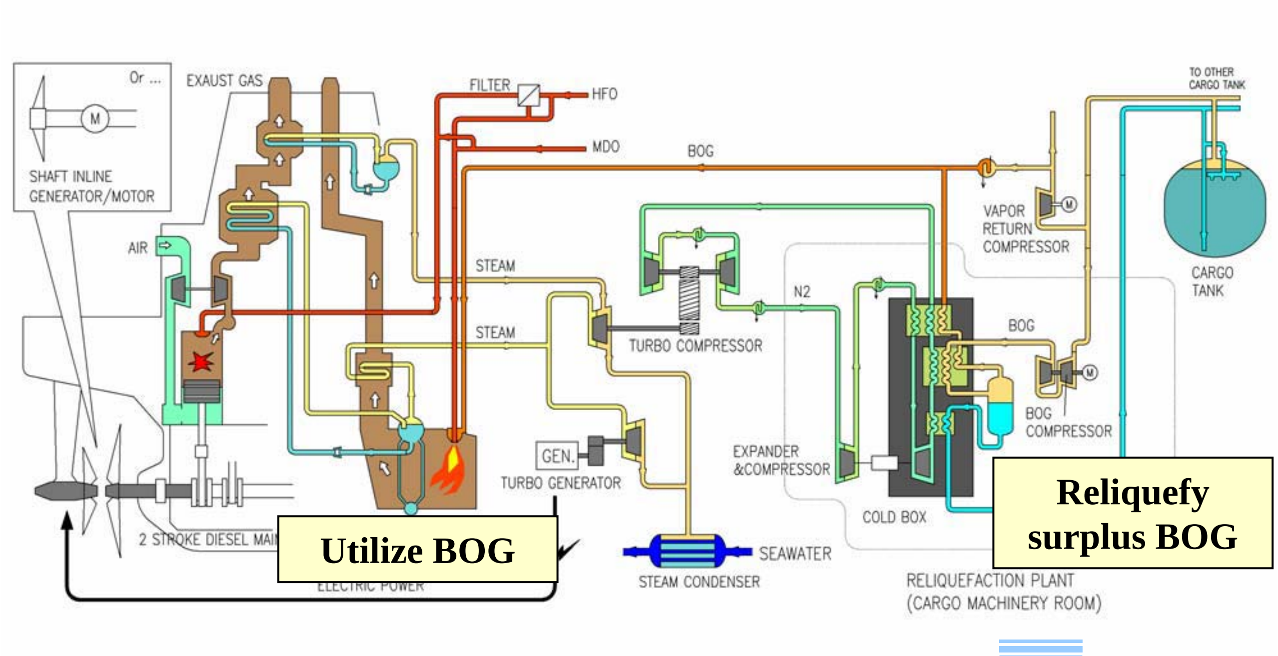
### Power plant

- ★ High efficiency
  (Thermo-eff. 20% more reduction)
- ★ High redundancy (Multi-engine plant)
- ★ Drastic fuel cost reduction by using LNG

Wartsila V50DF

# Alternative Propulsion

## Hybrid propulsion system with RL plant : HYBRID (DRL)



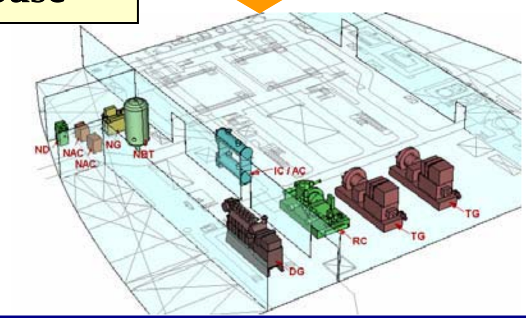

Flexible propelling

**CRP POD**

**POD enough power for slow steaming**

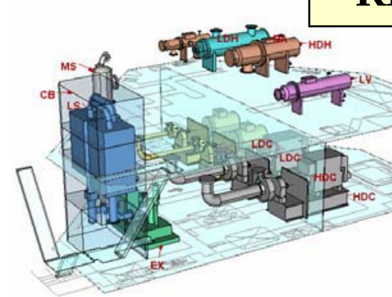
**T/G base**

Power plant in E/R



Compact re-liquefaction plant

**RL plant**



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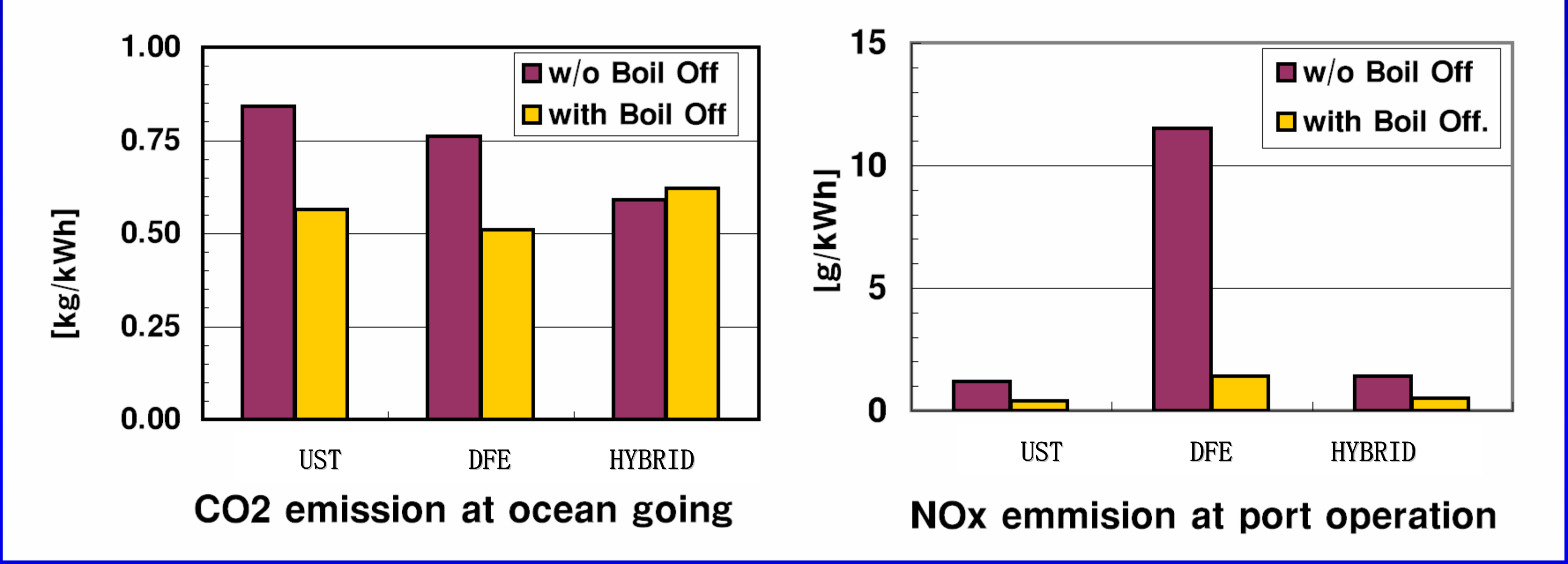
# Machinery design

## Emission comparison




*CO2 emission: No significant difference in case of BOG use.*
*NOx emission: Considerable NOx for DFE without BOG.*

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# Conclusion

1. **Level 2 "Practical Winterisation" can cover the trade to the existing or planning terminals in cold climate.**
   - **LMDAT -10 to -20 deg.C (Extreme -30 to -40 deg.C)**
   - **Thin First year ice escorted by ice breakers**
   - **Cold climate terminal surrounding only**
2. **Ice passport concept can be used for Hull structure optimization.**
3. **Reasonable setting of Design temperature for outfitting base on meteorological statistics, otherwise big cost impact.**
4. **Counter measures for outfitting should be selected based on scientific background and consistency with service experiences.**
5. **UST is an attractive plant among existing technology. HYBRID can be an attractive plant in future.**

**Detailed discussions between owners and builders would lead to beneficial solutions.**

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# Thank you for your attention

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# Structural Design of High Ice Class LNG Tankers

**Claude Daley**[1]
**Andrew Kendrick**[2]
**Han Yu**[3]
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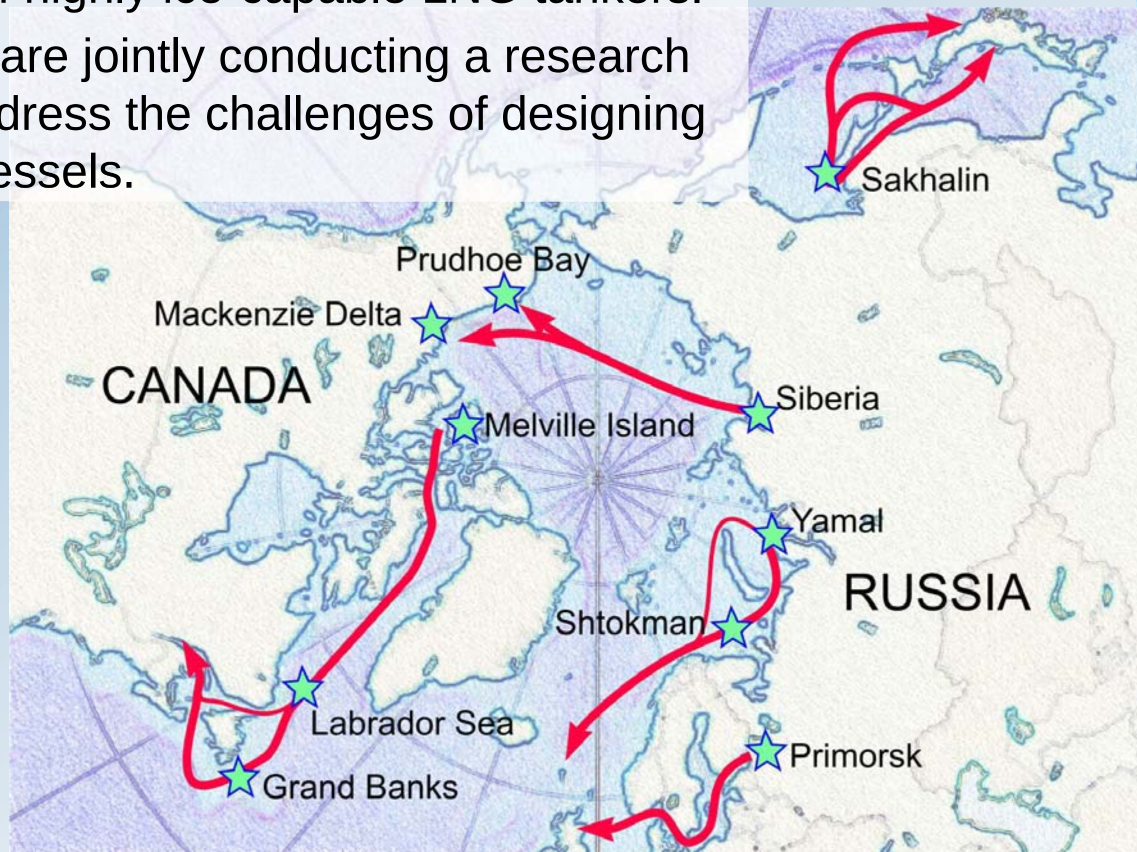
1 MEMORIAL UNIVERSITY Faculty of Engineering and Applied Science
3 ABS
2 BMT Fleet Technology
4 HYUNDAI

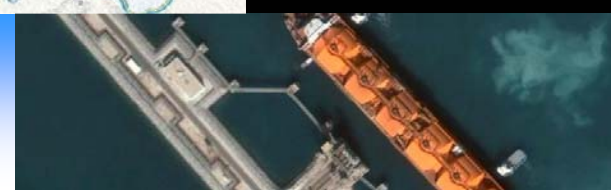
## Overview

Extensive gas reserves have been discovered in the Canadian and Russian Arctic

Development these reserves will require a new generation of highly ice-capable LNG tankers.

The authors are jointly conducting a research project to address the challenges of designing these new vessels.





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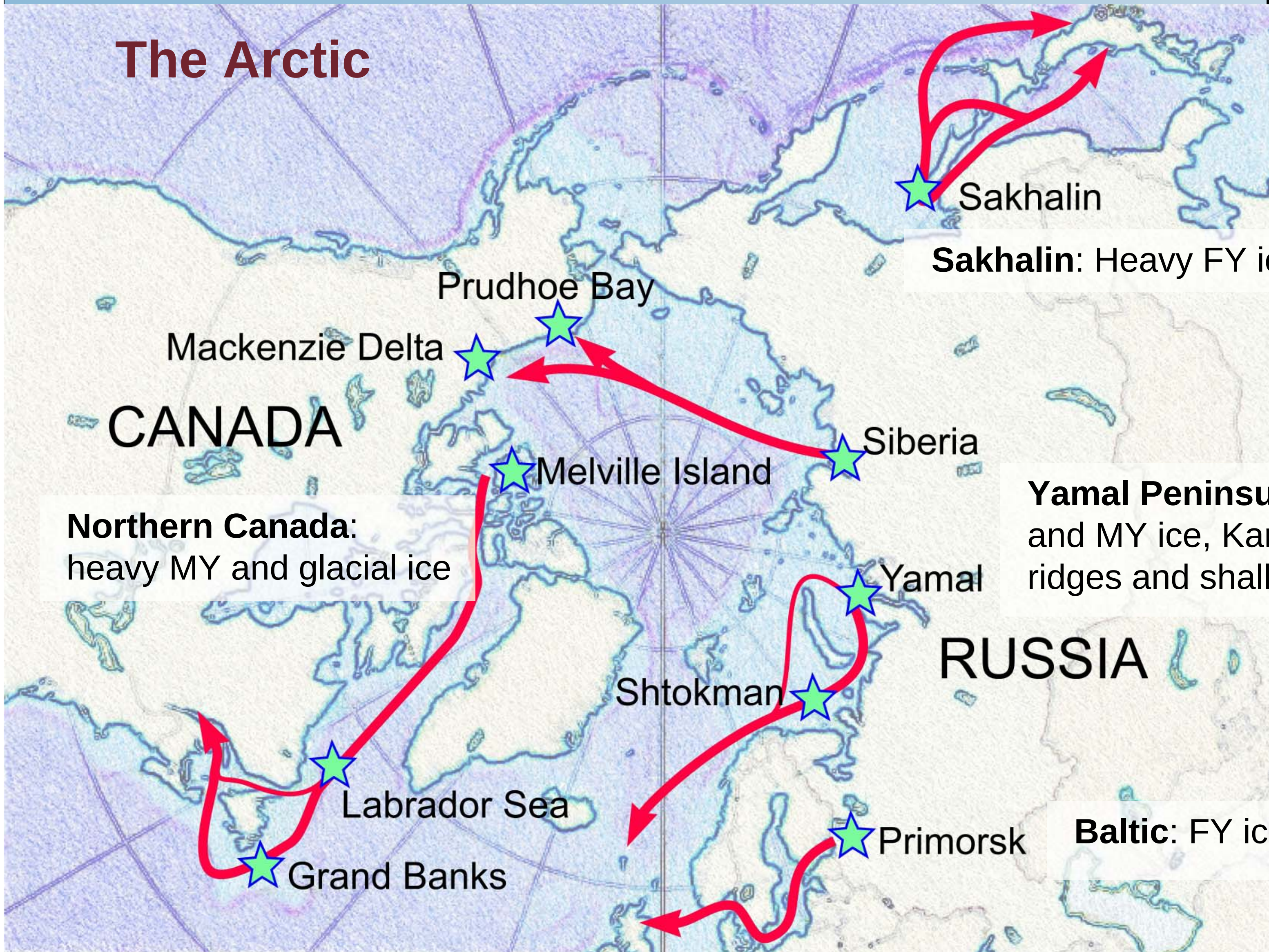
# The Arctic



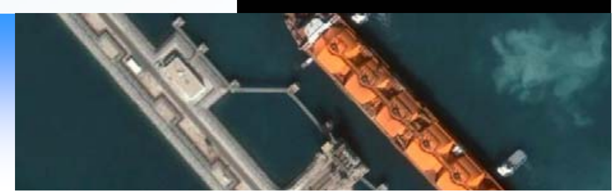

**Sakhalin**: Heavy FY ice

**Northern Canada**: heavy MY and glacial ice

**Yamal Peninsula**: FY and MY ice, Kara Gate – ridges and shallow water

**Baltic**: FY ice, ridges



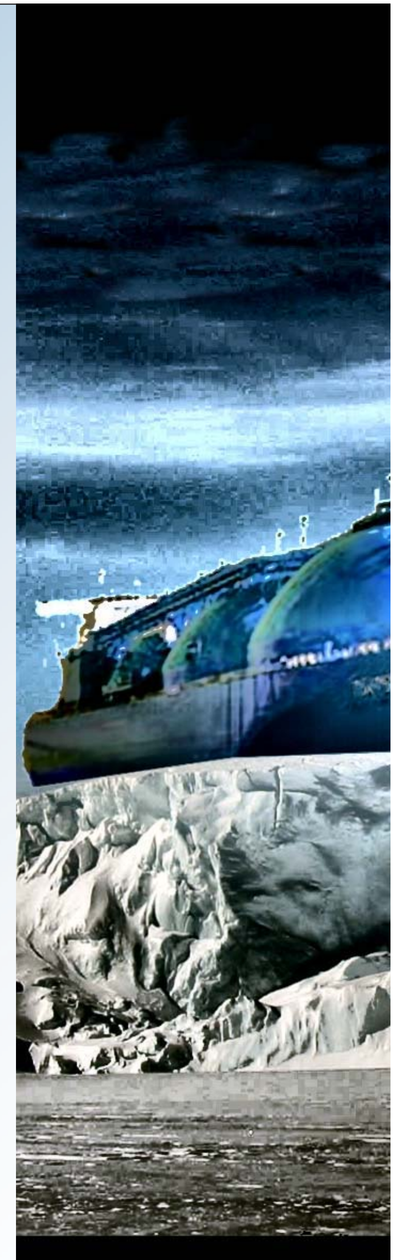


# Overview

Marine LNG transportation is a cost effective way to deliver the gas to market

Ice-going LNG ships present several unique challenges;

- size – LNG vessels are much larger than most ice going ships
- speed – LNG ships need to maintain high throughput and so may operate at relatively high speeds in ice
- season – LNG shipment must be a year-round operation, even through the coldest and darkest times
- LNG – the containment system presents a unique challenge





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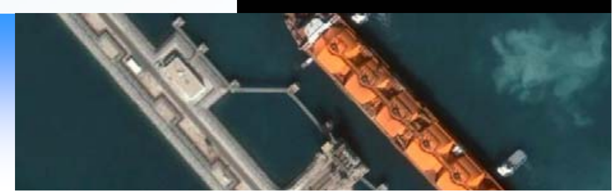
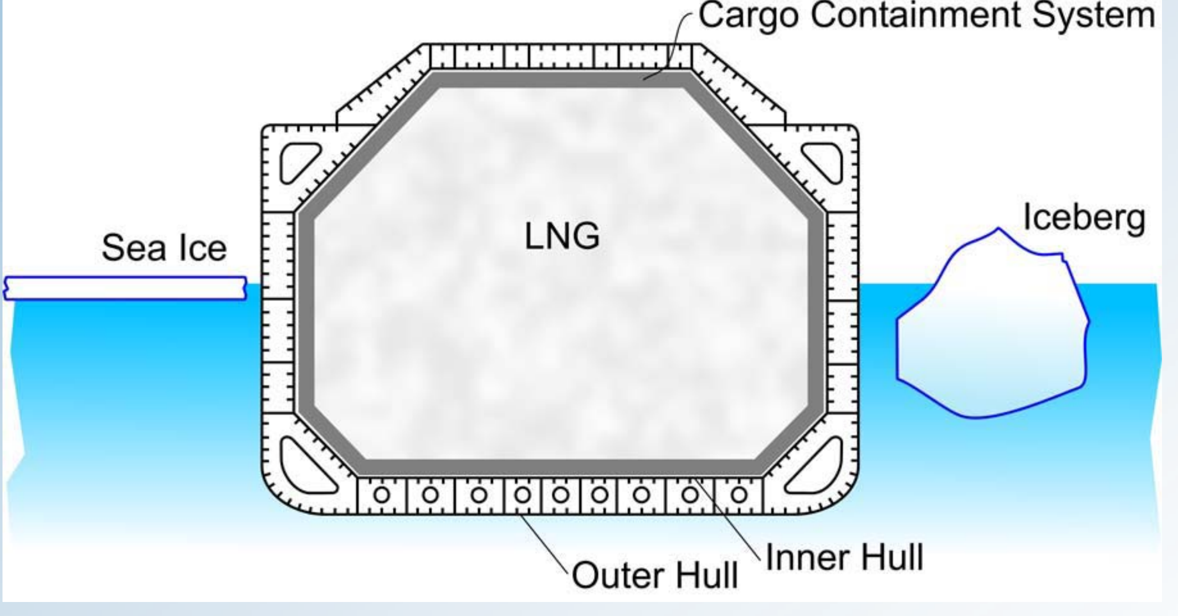
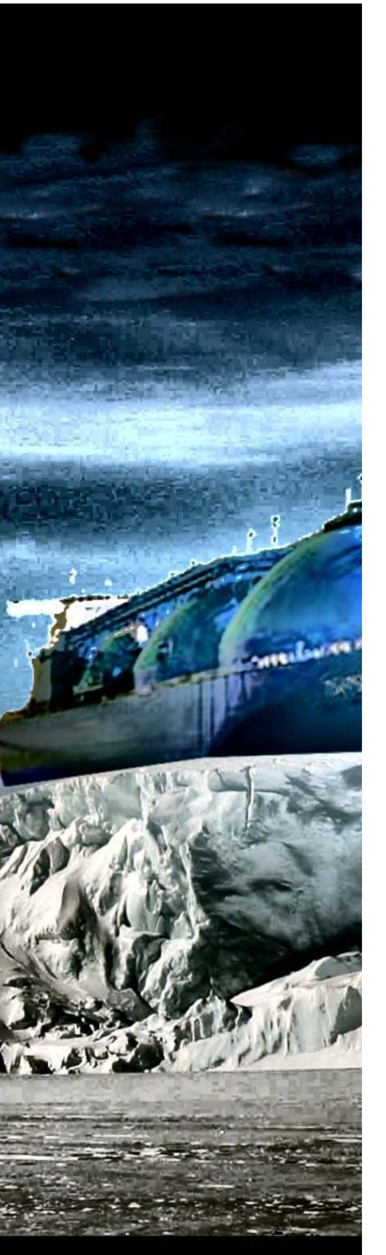
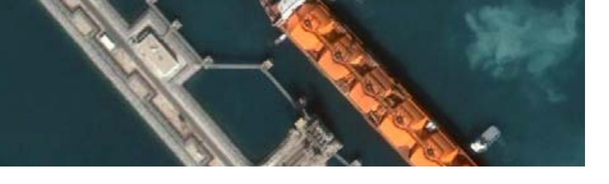


## Overview

- LNG ships have multiple barriers. The ship itself has both an outer and inner hull
- The membrane type LNG containment system has several layers for cargo containment
- The hull structure and cargo containment system must be designed for ice loads and ice load effects







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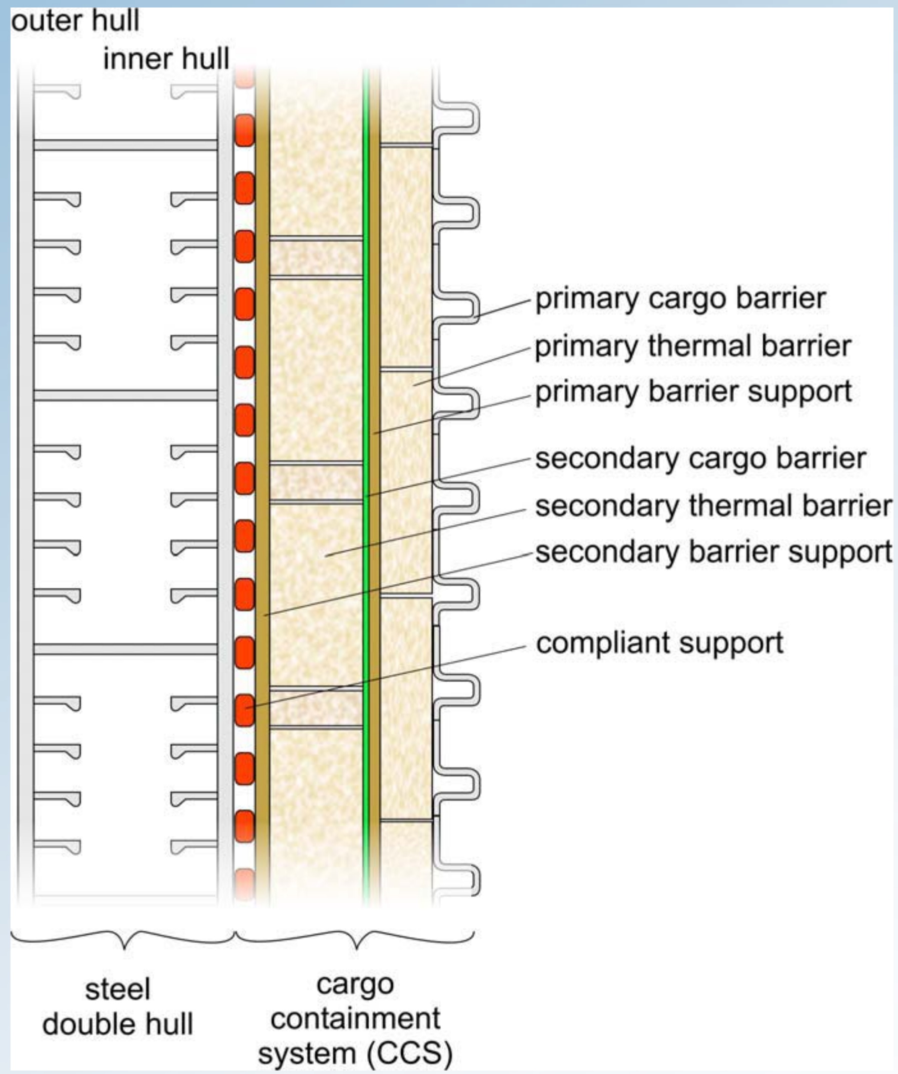
# Overview

A close-up sketch of the CCS shows the many barriers;

- liquid barriers
- thermal barriers
- deformation barriers
- strength barrier

All must be ice-load capable.




# Background

As of 2006, IACS introduced new Unified Requirements for Polar Ships

UR I1. Polar Class Description and. Application.

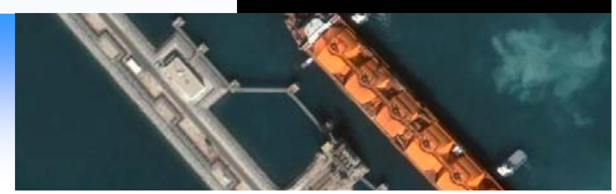
UR I2. Structural Requirements.

UR I3. Machinery requirements

The Polar Class rules have several new features;

- scenario-based design, with the load derived from collision mechanics, which links loads with ice conditions and operations
- limit-state structural assessment, with plastic capacity being the focus





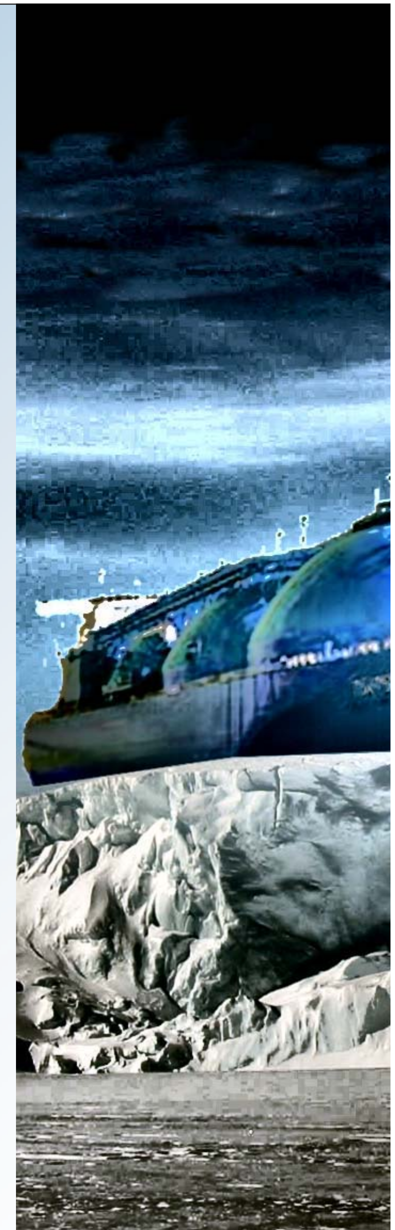
# Joint Research Project

Starting in 2006, ABS, HHI and BMT agreed to work towards developing the design tools needed to create a high ice class LNG tanker.

It was agreed that these LNG vessels would need to use the latest available knowledge, as they represent a unique class of ship.

This paper focuses on the structural requirements.

- We have developed ice load and structural assessment tools that follow the ideas in the new IACS UR for Polar Ships, but expand the range of load scenarios used
- With this expanded set of loads, the structure will be assessed with linear and non-linear finite element analysis





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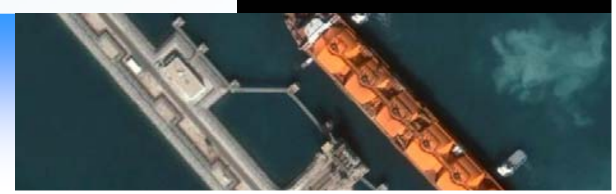
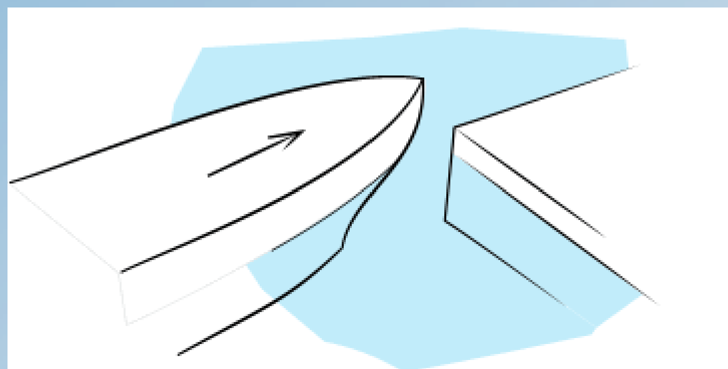
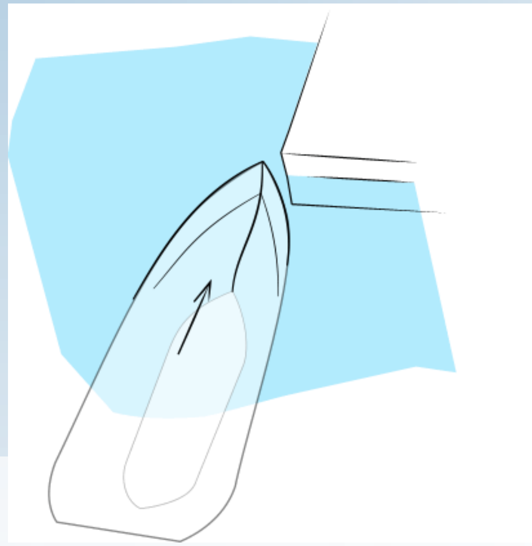


# IACS UR I2 Load Scenario

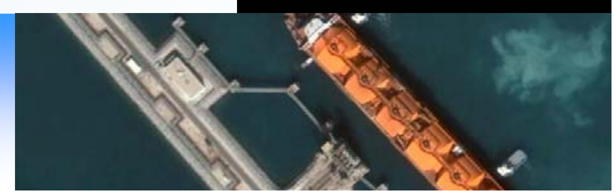
The basic load scenario in the Polar Rules is an oblique collision on the bow.





Ice load depends on

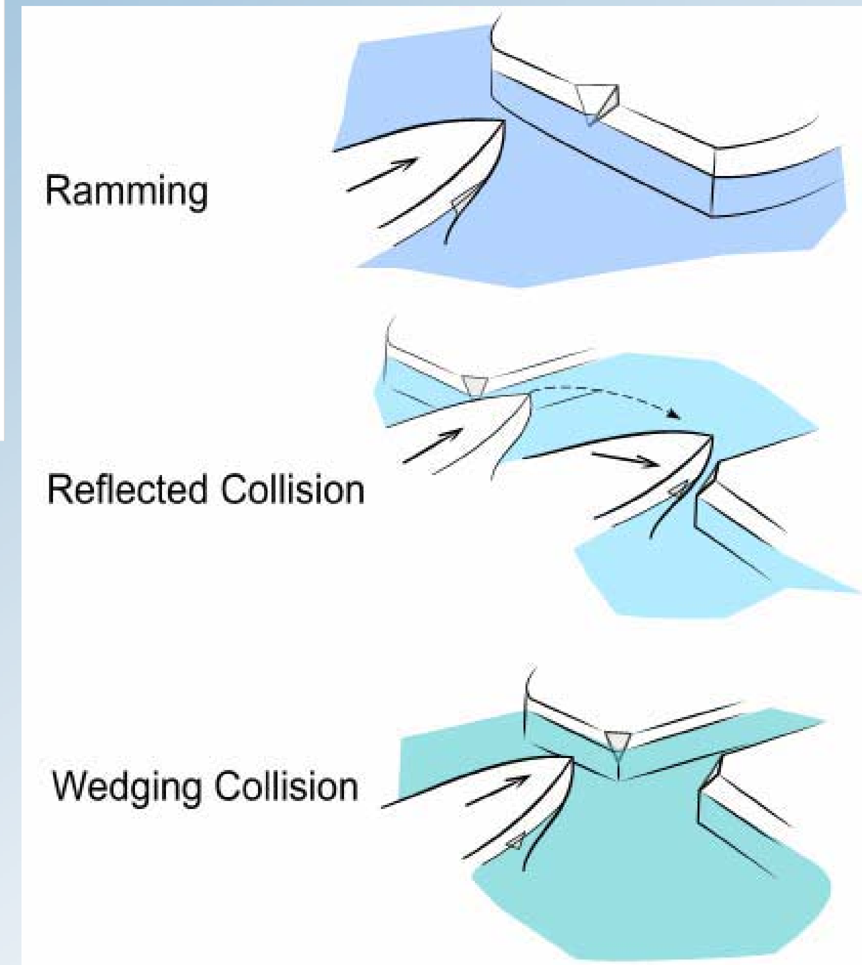
- ice shape (fixed)
- pressure-area terms (class dependent)
- ice thickness and flexural strength (class dependent)
- collision modelled using ‘Popov’ assumptions

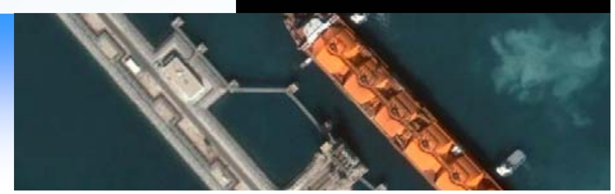


# LNG Load Scenarios

We've extended the scenario in the Polar Rules with 20+ additional ice interaction cases.

Here are 3 collision cases:





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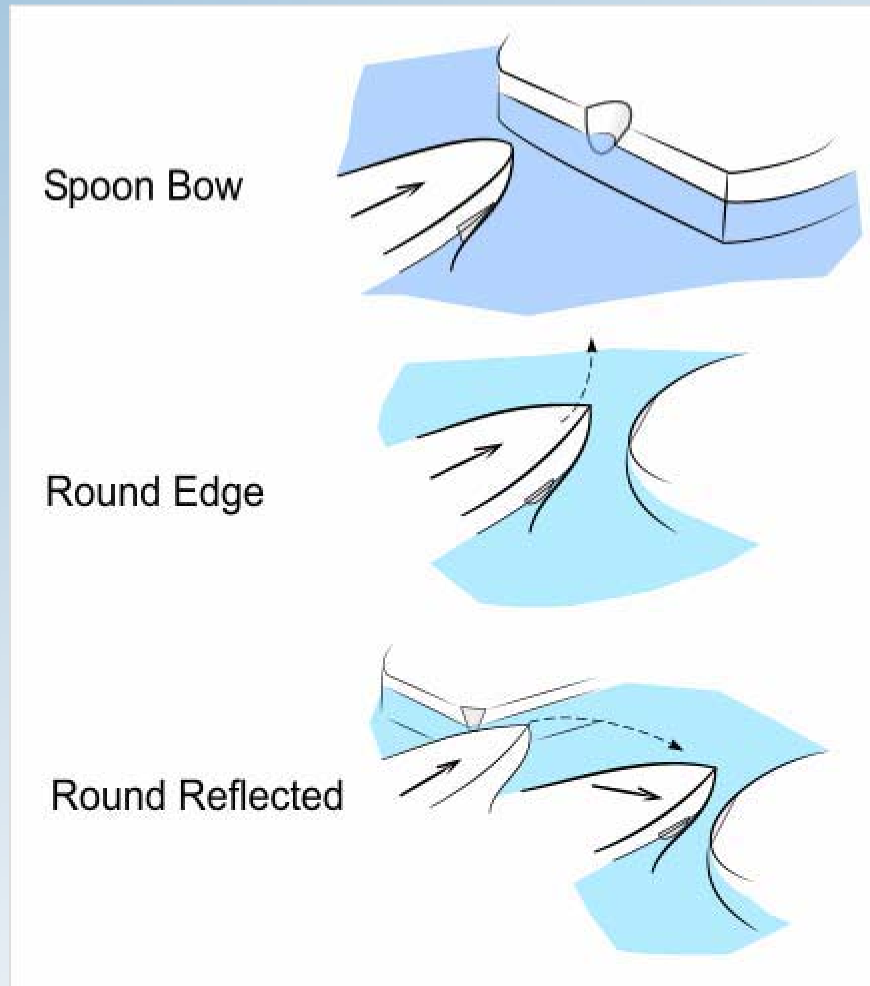
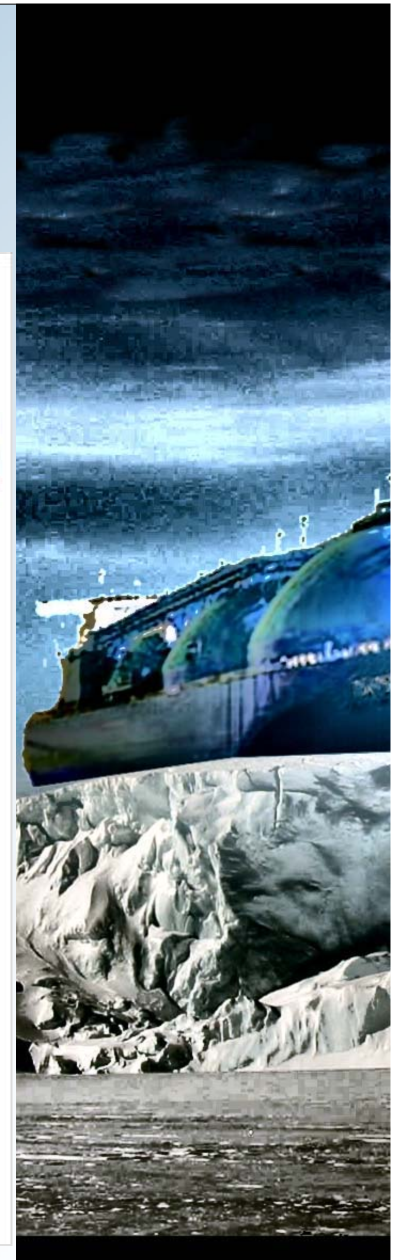
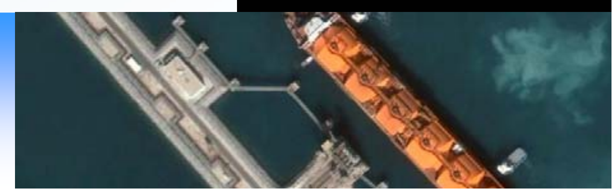
# LNG Load Scenarios

And 3 with alternate contact geometries:

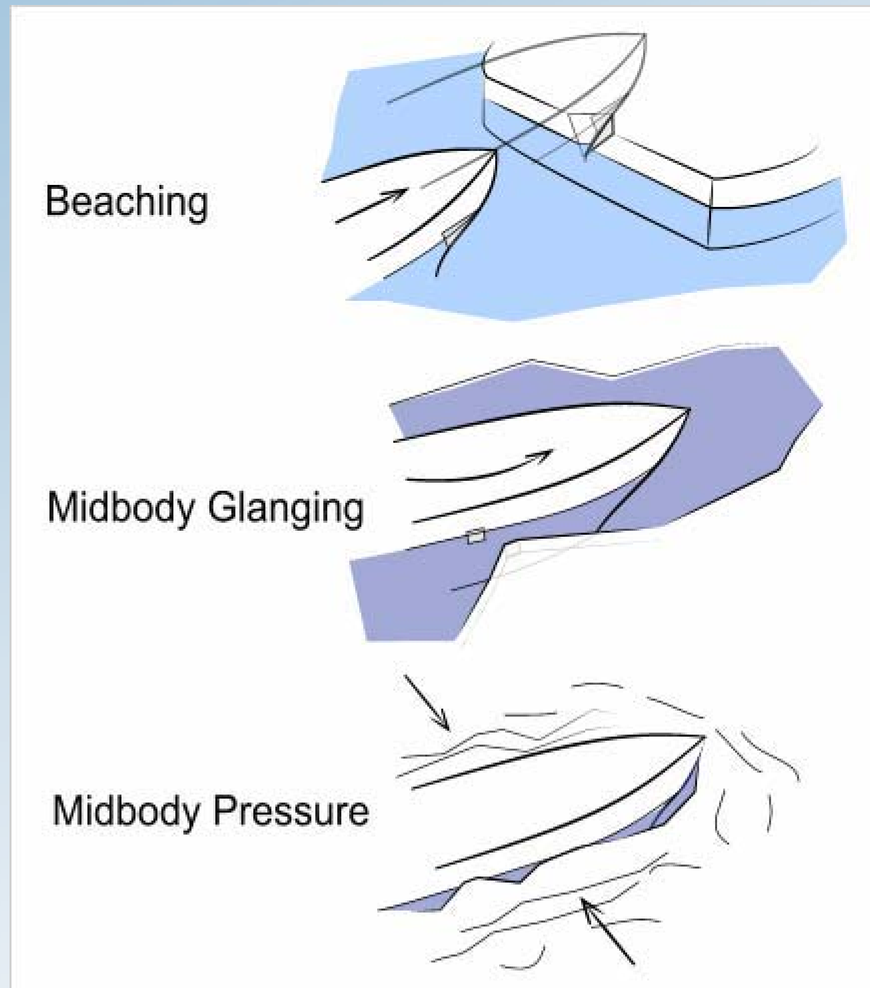


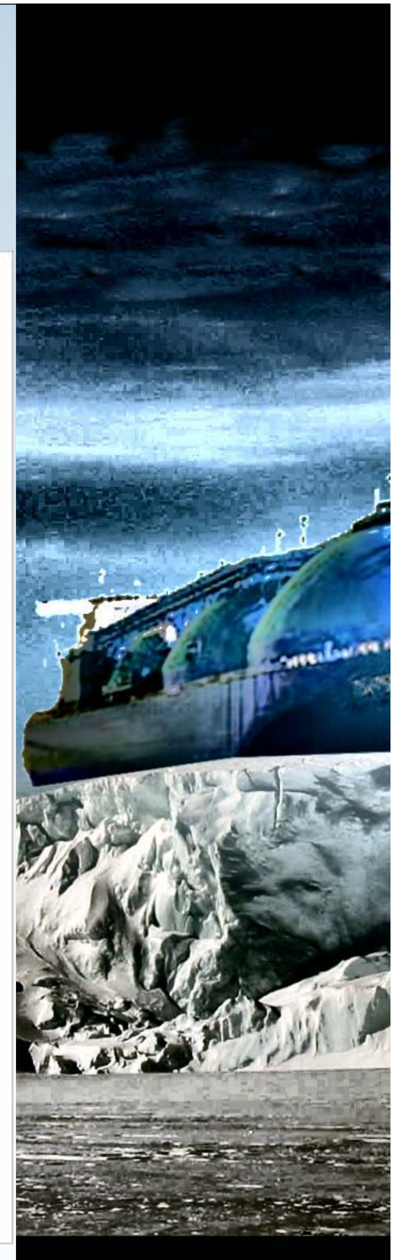





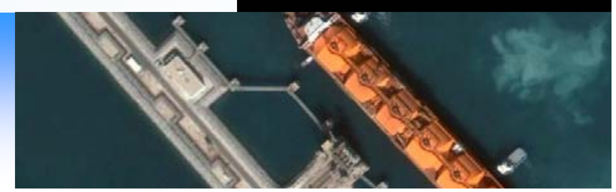
# LNG Load Scenarios

And 3 more load scenarios:



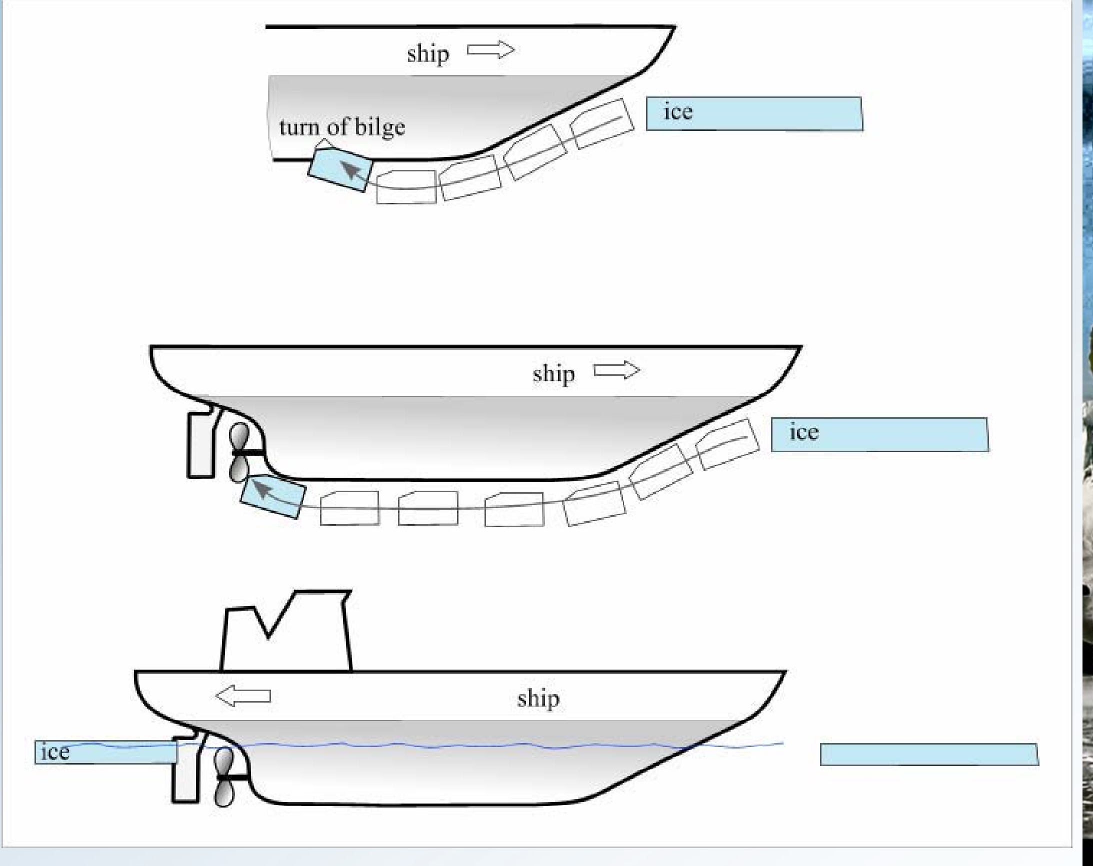



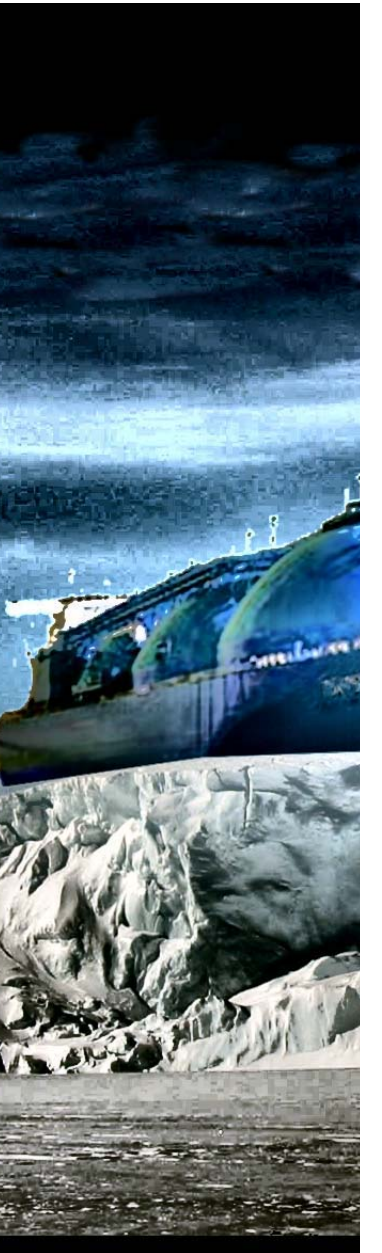




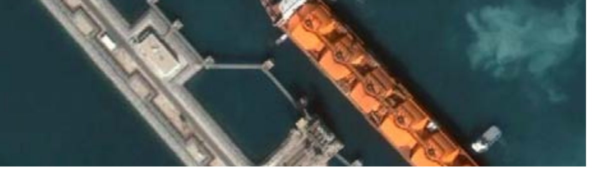
# LNG Load Scenarios

And 3 more load scenarios:



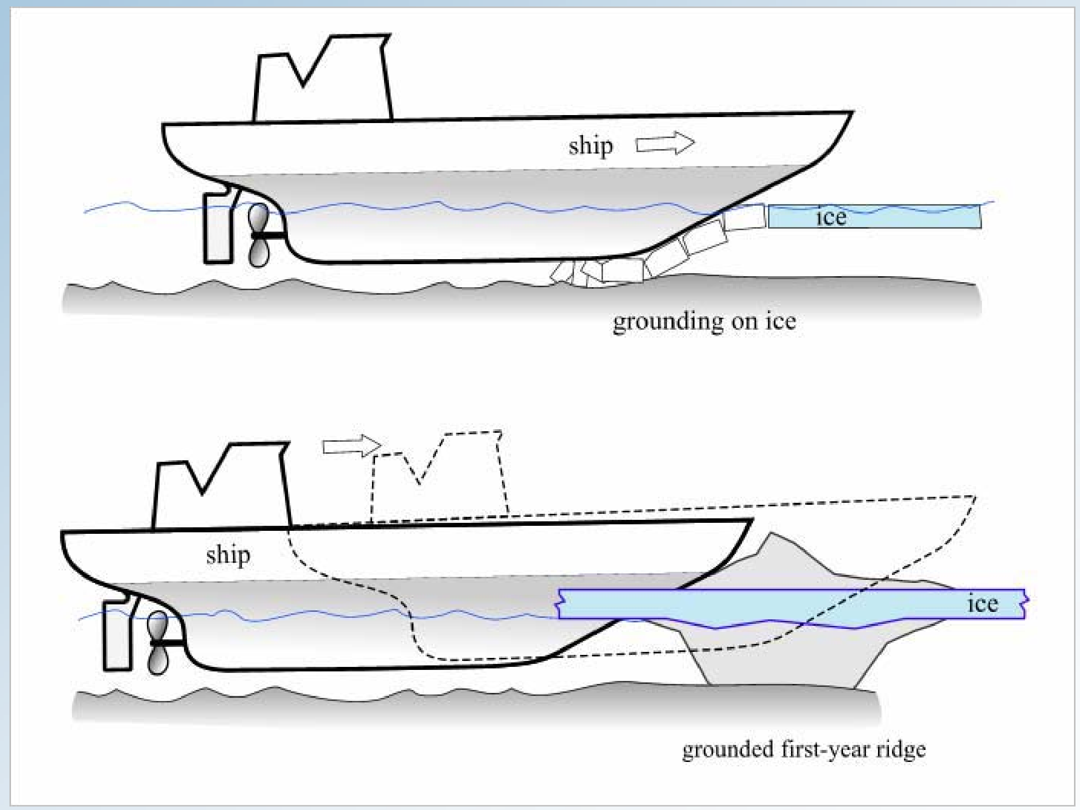



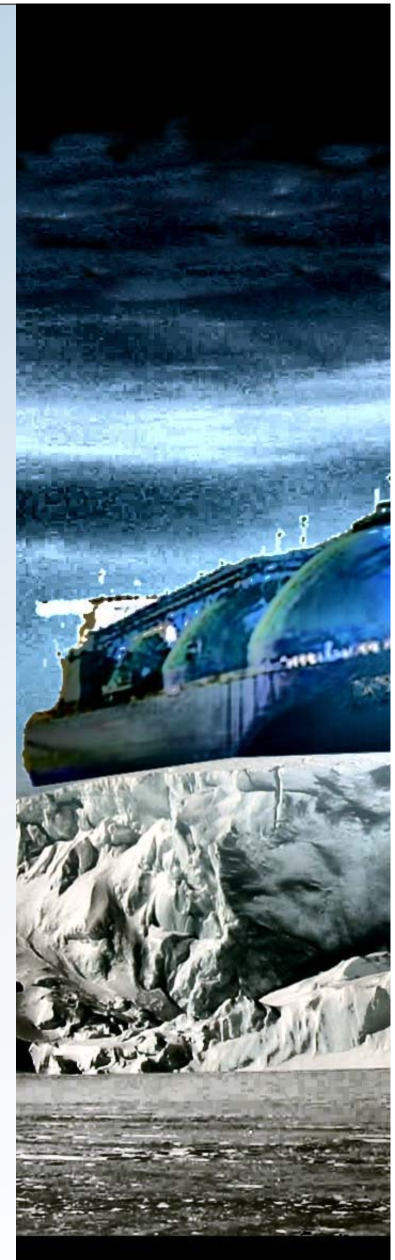


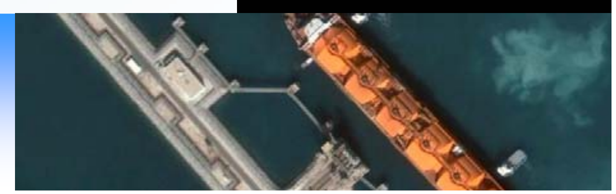


## LNG Load Scenarios

And 2 more load scenarios:





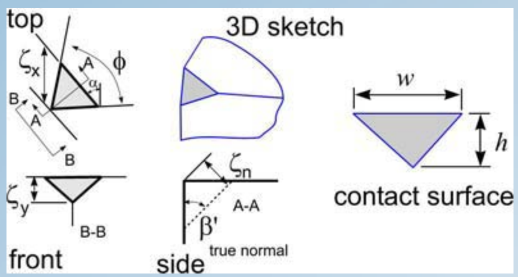
# Solving for Ice Loads

For the various load scenarios, load patches are derived, generally as follows:

The ice crushing energy is the integral of the product of force and crushing depth:

$$IE = \int_0^{\zeta} F_n \cdot d\zeta_c$$

The contact is the overlap between hull and ice:




contact force depends on pressure and area:

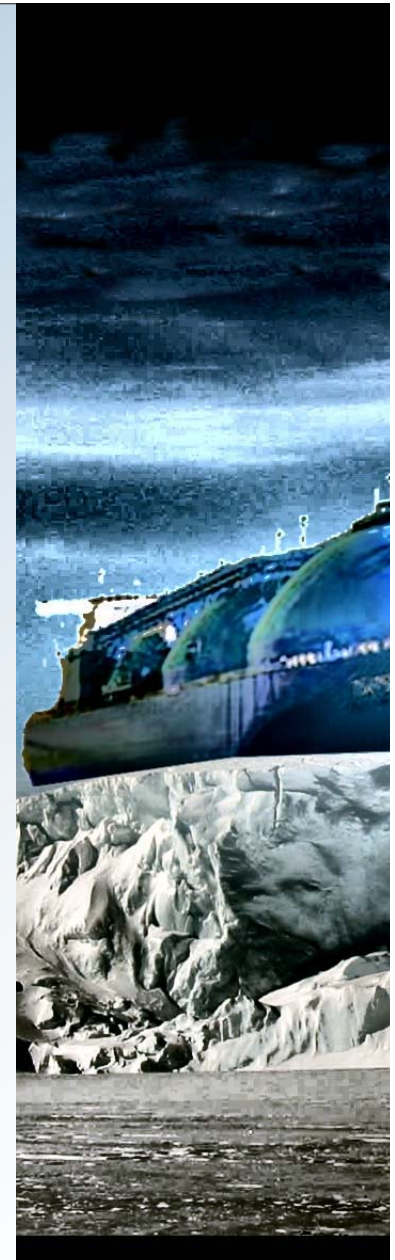
Pressure- Area expression:

$$P_{av} = P_0 \cdot A^{ex}$$

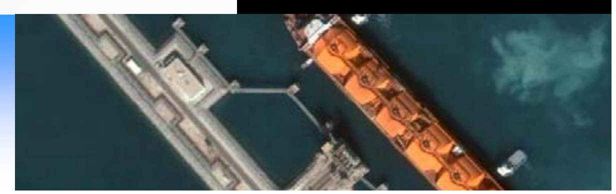
$$F_n = P_{av} \cdot A_n = P_0 \cdot A_n^{1+ex}$$

For this geometry case, the integration gives:

$$F_n = p_o \left( \frac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)} \right)^{1+ex} \cdot \zeta_n^{2+2ex} \Rightarrow IE = \frac{p_o}{(3+2ex)} \left( \frac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)} \right)^{1+ex} \cdot \zeta_n^{3+2ex}$$







# Solving for Ice Loads

Once we have the ice crushing energy as a function of penetration depth, we can solve for the penetration depth.

We do this by equating the effective collision energy with the crushing energy:

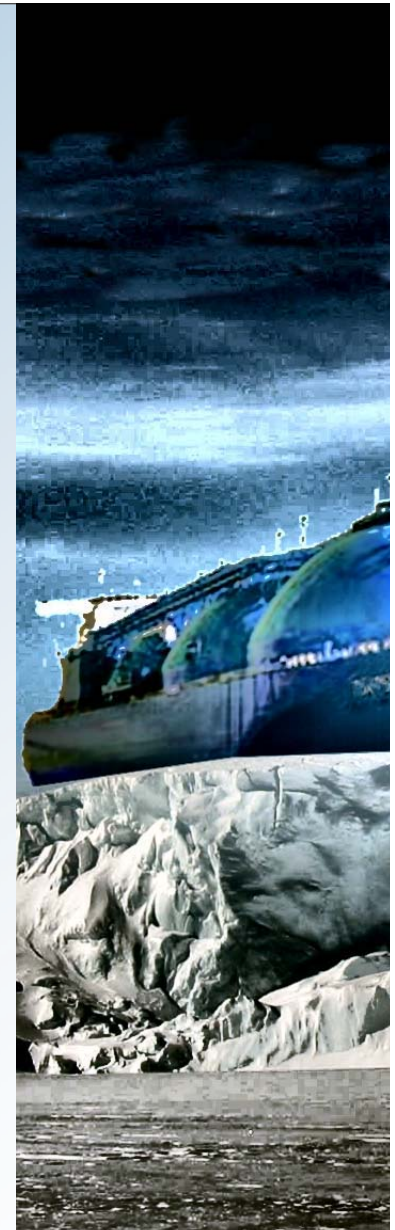
$$KE_e = IE$$

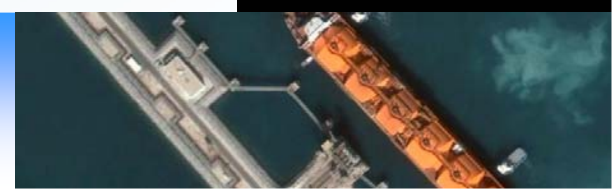
In general the force is: $F_n = P_0 \cdot (f_A)^{1+ex} \cdot (\zeta_n)^{d(1+ex)}$

Which lets us write: $IE = \int F_n d\zeta_n = P_0 \cdot (f_A)^{1+ex} \cdot (\zeta_n)^{d(1+ex)+1}$

And: $KE_e = P_0 \cdot (f_A)^{1+ex} \cdot (\zeta_n)^{d(1+ex)+1}$

We pull all this together to get: $F_n = p_o \cdot f_A^{\,1+ex} \cdot \left( \dfrac{KE_e}{p_o \cdot f_A^{\,1+ex}} \right)^{\frac{d(1+ex)}{d(1+ex)+1}}$

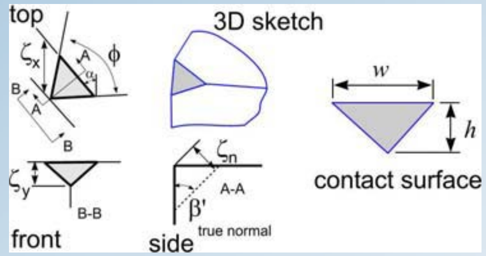
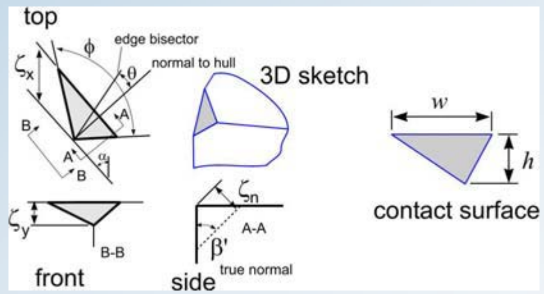
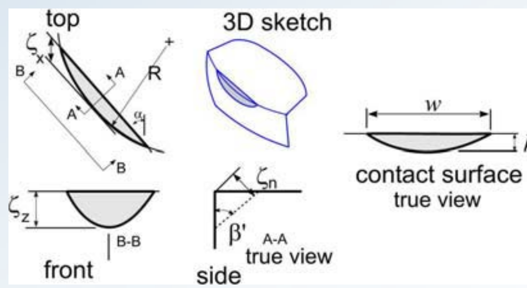


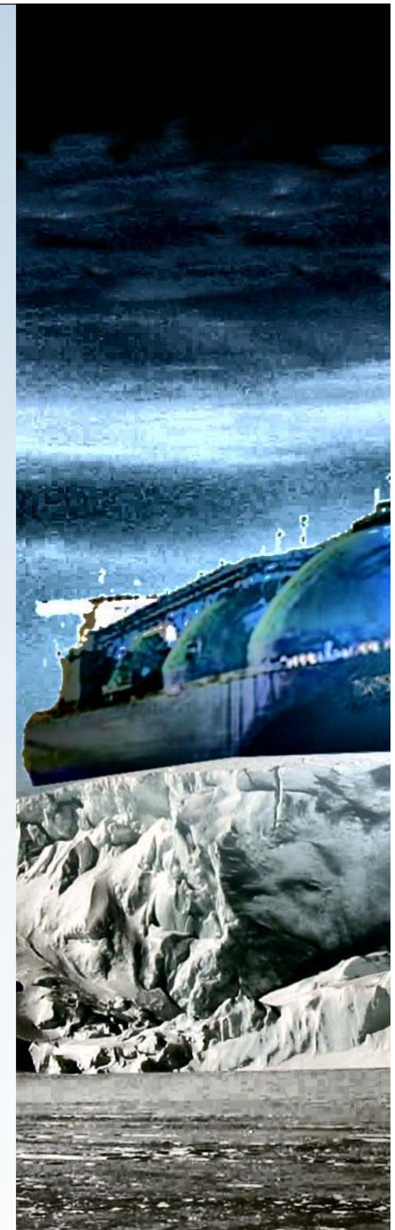


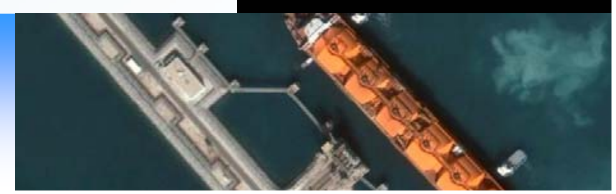
# Solving for Ice Loads –

## examples of geometry functions

### Table 5:

| | | |
|---|---|---|
| Case 1 : General Wedge (Normal to hull) | $f_A = \dfrac{\tan(\phi/2)}{\sin(\beta`)\cos^2(\beta`)}$ <br> $d = 2$ |  |
| Case 2 : General Wedge | $f_A = \dfrac{(\tan(\phi/2-\theta)+\tan(\phi/2+\theta))}{2\cdot\sin(\beta`)\cos^2(\beta`)}$ <br> $d = 2$ |  |
| Case 3 : General Round Edge | $fa = \dfrac{4}{3\cdot\cos^{1.5}(\beta`)\sin(\beta`)}\sqrt{2R}$ <br> $d = 1.5$ |  |







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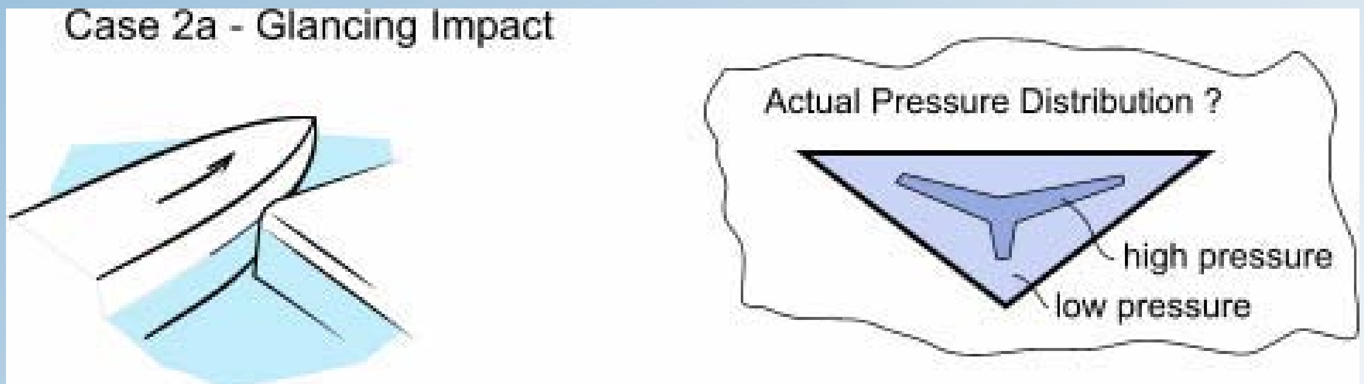
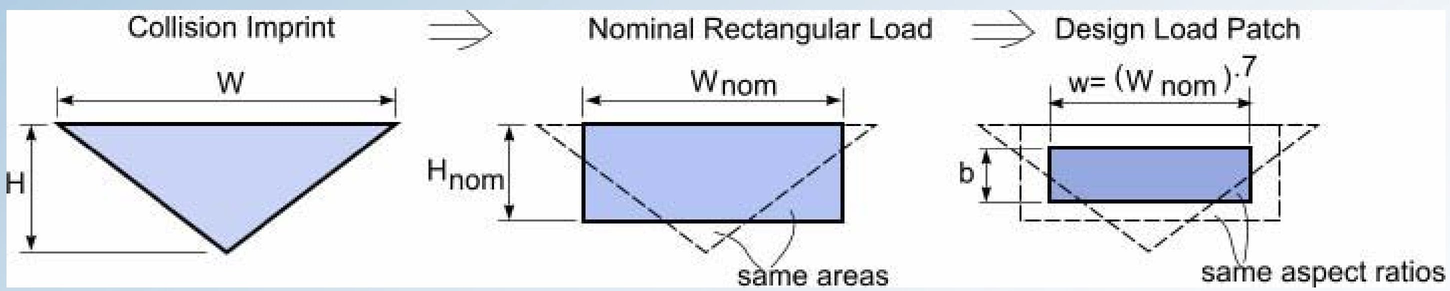
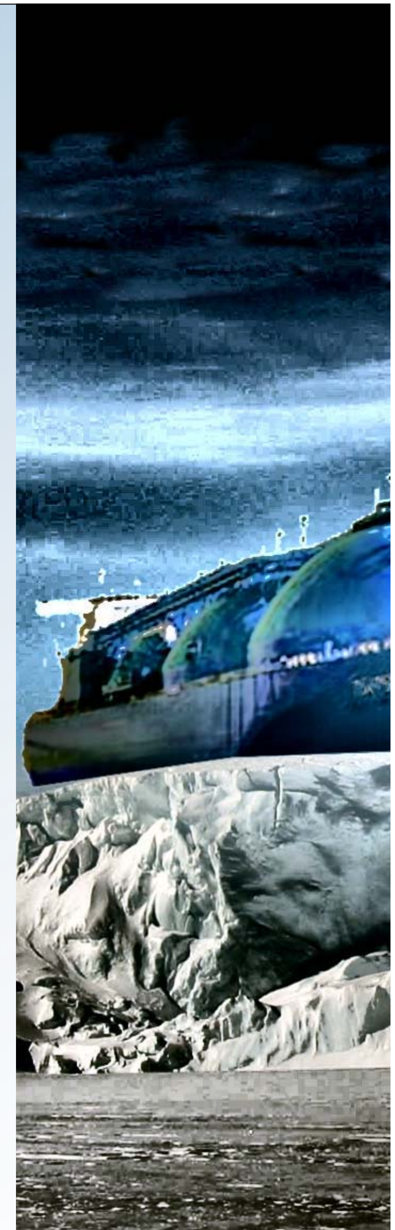
# Analyzing Structure-

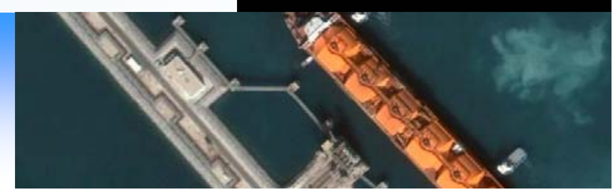
Once we have the various collisions modelled, we can develop a load patch to apply to a finite element model.

At this point we follow a process, just as was used to develop the design load in the We need to account for load peaks. We also would like to express the load as a rectangle for practicality.







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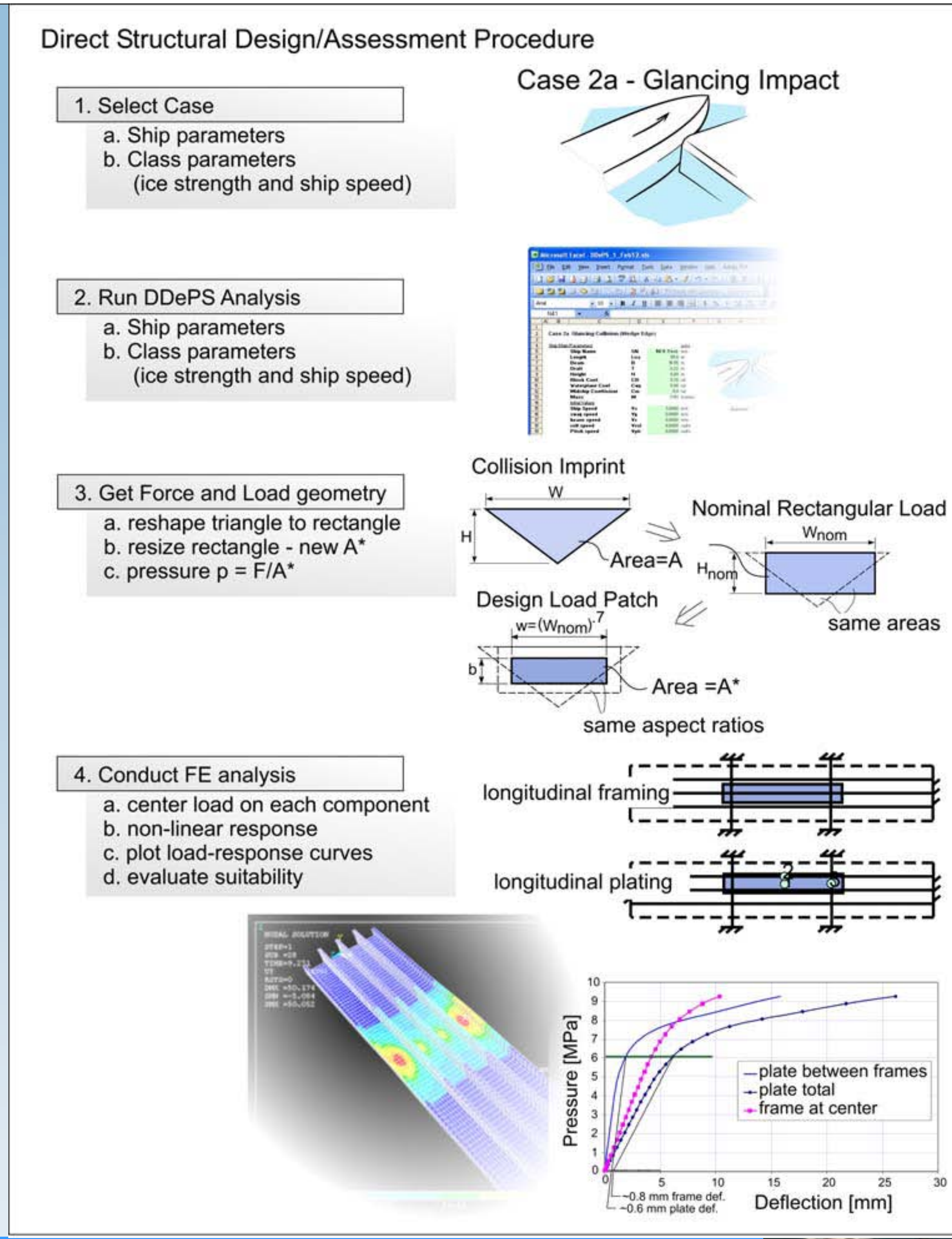
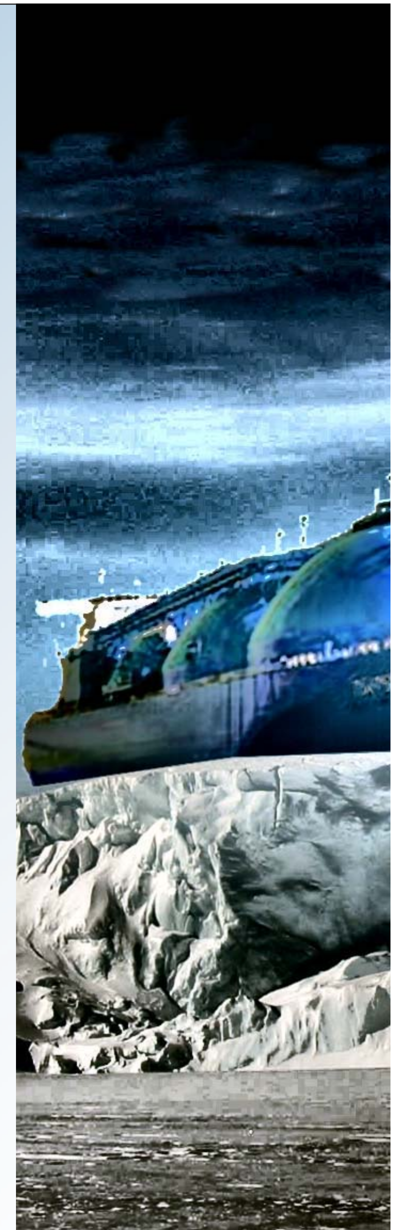
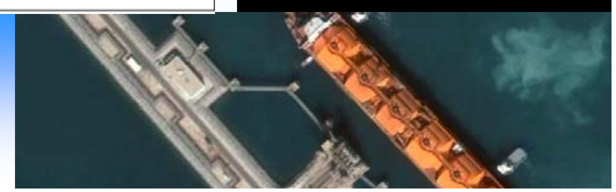
# Analyzing Structure-

## The whole process looks like:







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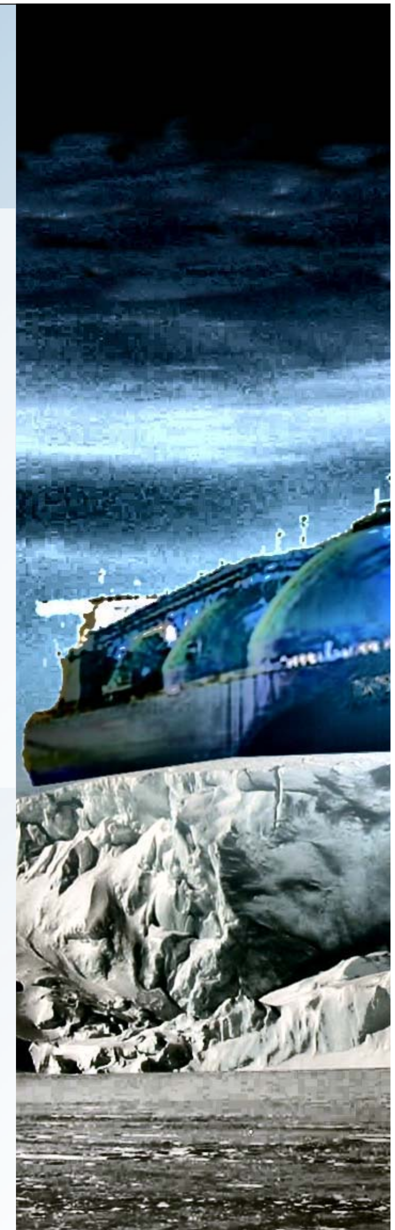
## Concluding Comments-

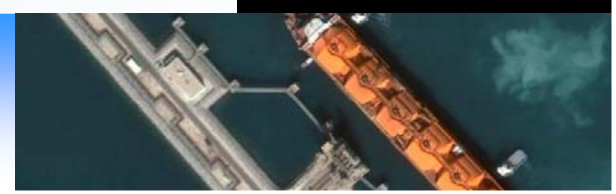


We have an approach that lets us

- Calculate loads for various scenarios
- Find load patches in a manner comparable to PC ice classes
- Check the hull response with linear and non-linear FE analysis
- Check the ice load effects on the CCS

This is a new and comprehensive system, that builds on the new IACS Polar Rules, reflecting the state-of-the-art in our knowledge of ice loads and structural strength.

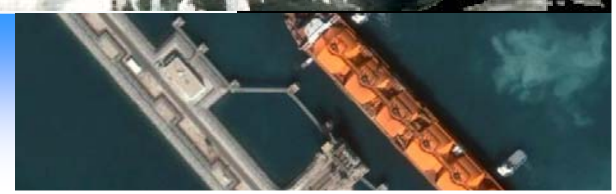




Thank you.

Comments? Questions?





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# Technical development of LNG Carriers for harsh ice conditions

Reko-Antti Suojanen
Manager, Arctic Consulting

Preferred for Innovation

aker yards.

part of the Aker group

# CONTENTS



- Design target
- Concept development
- Model testing
- Simulations
- Ice strenghtening
- Future work

# The LNG opportunity and challenge

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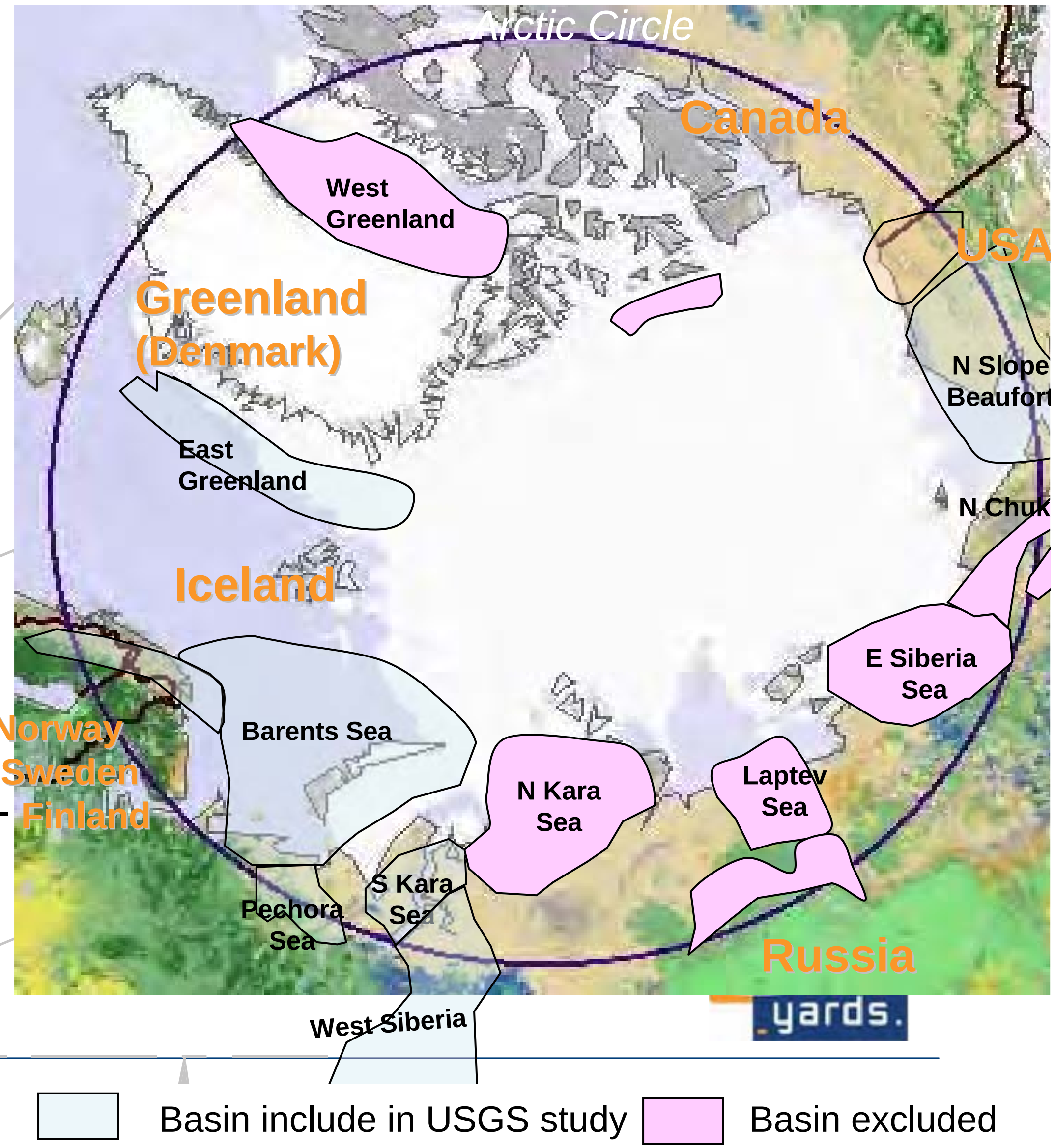
- IEA (International Energy Agency) forecast the wold energy demand increaces 60% to year 2030
- Oil, gas and coal (fossil energy) cover now 80% of energy production
- Share of fossil energy is predicted to furher increase
- Today oil is major energy source (35%)
- Gas is only third largest source after coal
- IEA predicts use of gas increases most, doubles to 2030
- New production focuses to Middle-east and Russia



# Where and how the gas goes

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For every energy source applies:

Production - transport - refining - transport - consumption

Transport of gas is most critical:

Energy content is small in NTP conditions.

| GAS in high pressure Pipelines | Liquefied by cold LNG | GAS to Liquid GTL |
|---|---|---|
| High investment<br>94% of transport | Higher investment<br>Energy for LNG<br>6% of transport | Easy use and transport<br>0 % of transport<br>Prospective |

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# Where and how the gas goes



Shipping solution, LNG

- Worldwide transport
- Flexibility of import
- LNG plant investment high
- Regasification
- Energy demand for liquefaction

# Aker solution for LNG transport from Yamal

# Aker solution for LNG transport from Yamal


Consept development started allready 2004.

First main solution  direct transport vs. shuttle

Direct

+ no realoading terminal

+ less boil-off

+ less risks

+ more constant delivery

-high number of special Carriers

- no ice bow possible

Shuttle

+ less special vessels

+ ice maximized Carrier

+ standard Carriers across NA

- higher investment cost

- less ice capable vessels

- unreliability in deliveries

# Aker solution for LNG transport from Yamal


Concept development started 2004.

DAT concept proven to be most attractive.

DEVELOPMENT PROGRAM

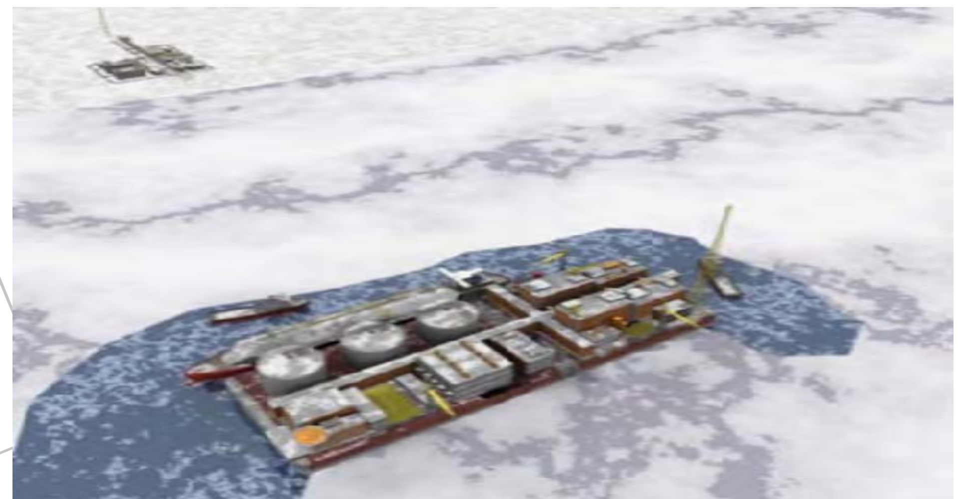
1. Icebreaking capability, simulations, model tests
2. Propulsion system selection
3. Cold environment challenges
4. Large size icebreaking vessel
5. Tank cover for Moss-tanks

# Arctic LNG Carrier

Icebreaking capability, simulations, model tests

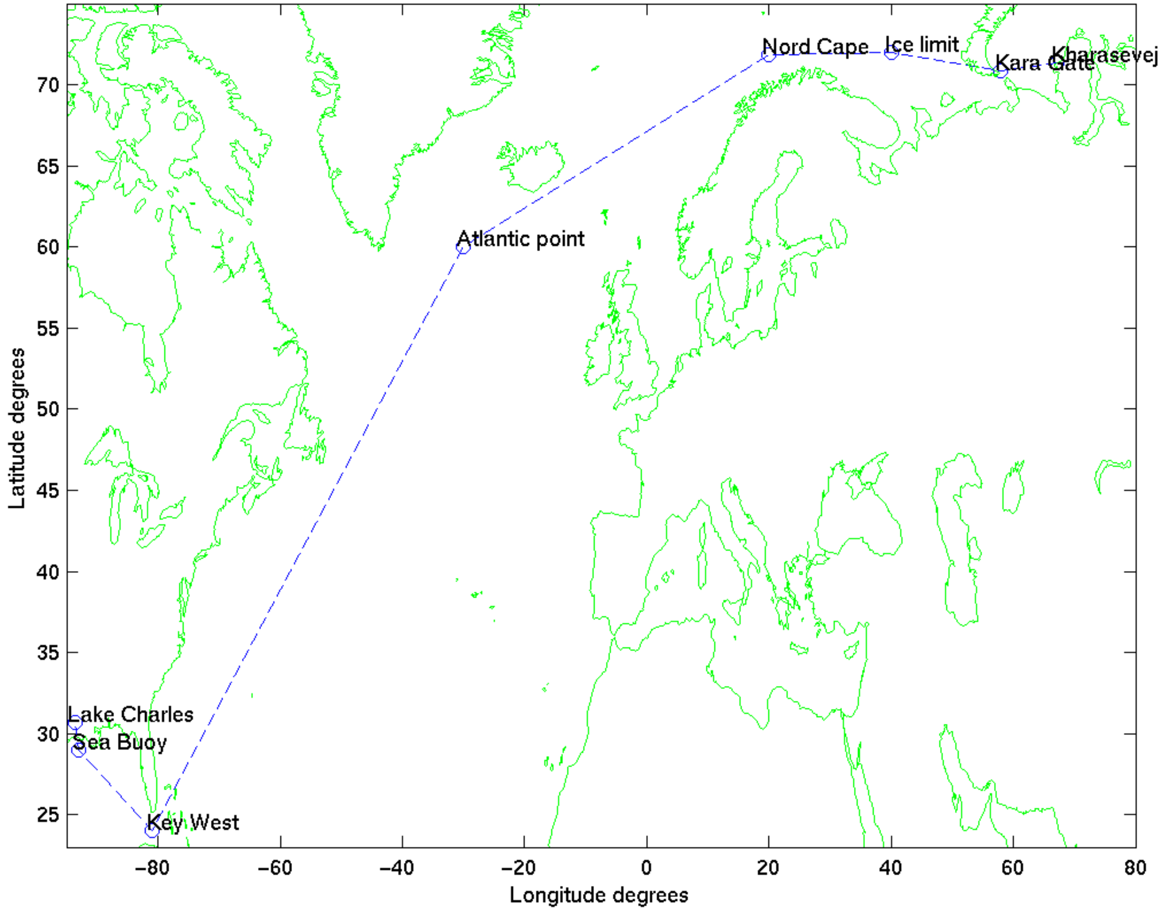
MAIN DESIGN CRITERIA

Arctic LNG Carrier for operation in Kara Sea.
Russian ice class level LU7.
Cargo carrying capacity about 200 000m3.
Year around traffic
North Atlantic crossing
Offshore loading capability
LNG tanks type A or B



# North Atlantic LNG from Yamal

Aker Arctic




Special challenges

N-A sea conditions.

Barents and Kara Sea ice.

Export terminal in Yamal coast line.

Purpose built LNG Carriers required.

# North Atlantic LNG from Yamal
# Sea State

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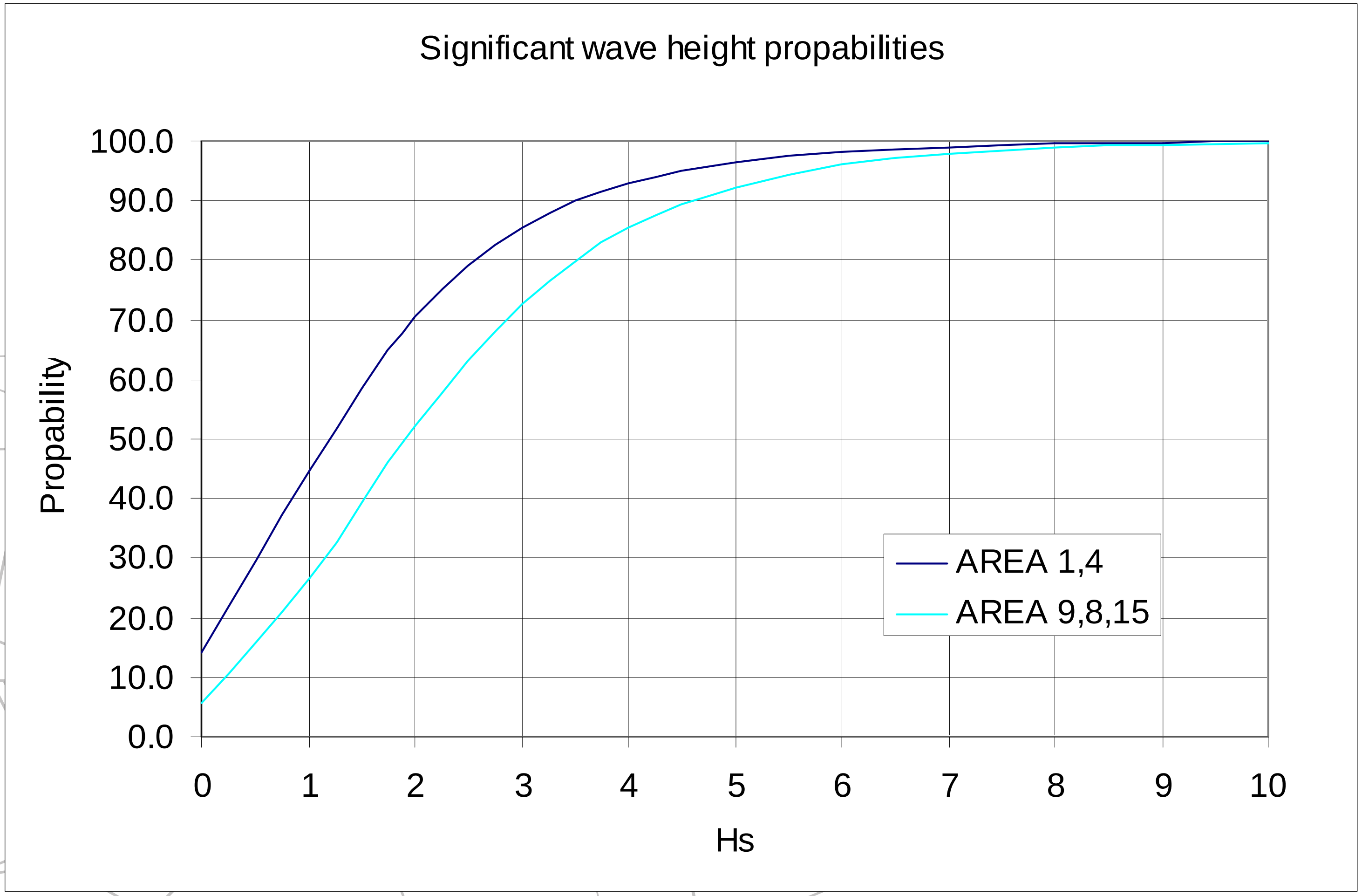


# Sea ice extent, average max

# Kara Sea ice massif

## Ice thickness at Kara Sea

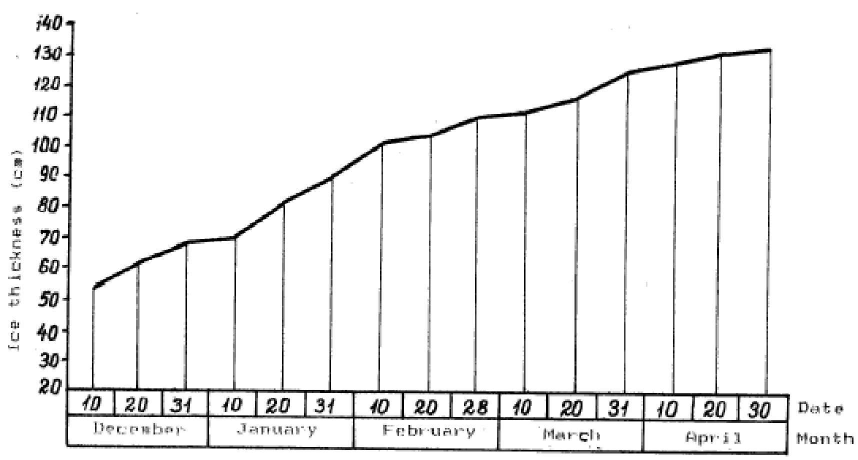
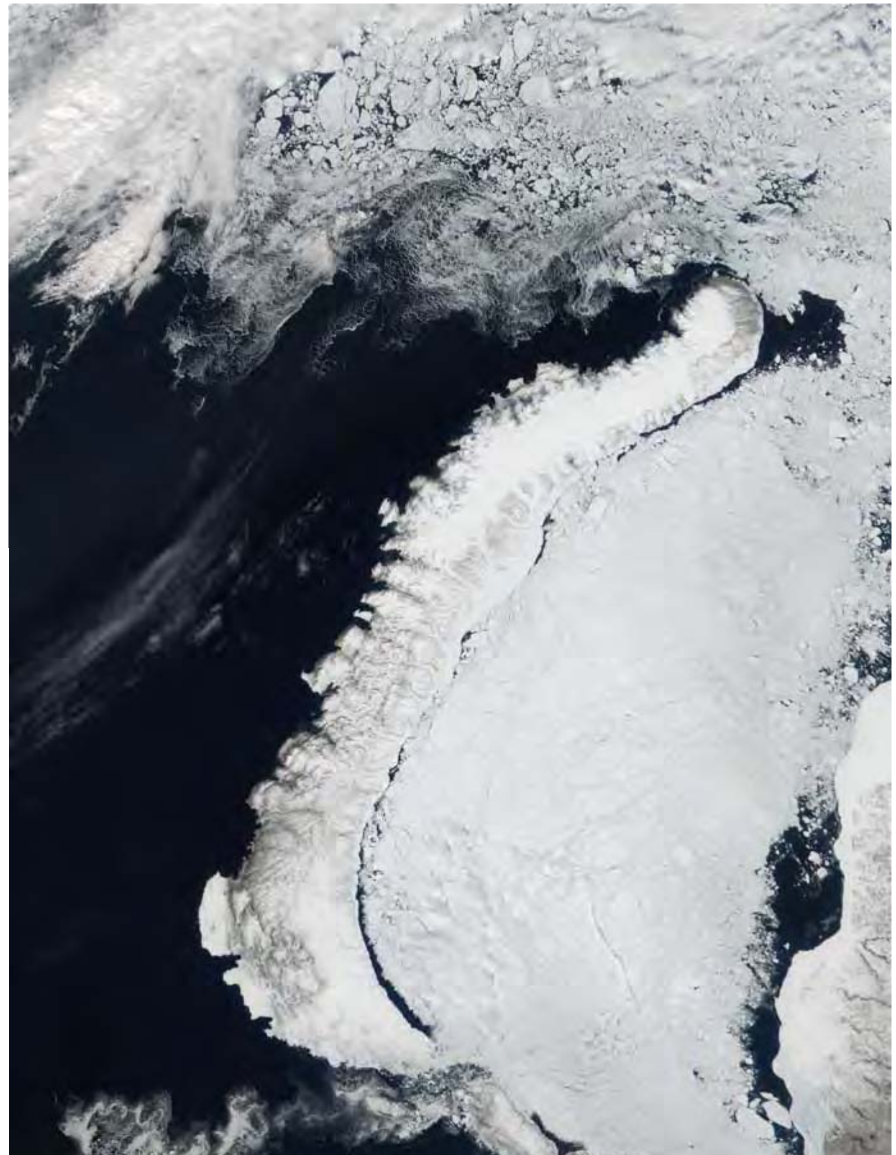



Fig. 3.1. Ice thickness growth near the polar station Kharasavey in 1976 – 1977



# Moving ice causes heavy ridging. Winter time reality at Karan Sea

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**And sometimes grow into stamukhas (Varandei)**

# Performance criteria

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Vessel must comply North-Atlantic conditions.

Service speed 19.5 knots.

Sea margin according to open sea conditions.

Speed through ice must be sufficient. Requires icebreaking capability 1.5 metres at 5 knots and capability to pass ridged ice fields at minimum 1 knot speed.

Vessels must operate with high level of safety criteria.

With this main criteria the transport will be cost effective for development of the LNG production.

# Icebreaking Transport Solution

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### Ice operating Mode:

The vessel will operate stern ahead with pulling propellers.

Very good ice breaking capability can be achieved with normal open water propulsion power.

### Open water Mode:

The vessel will operate in the open water conditions as the normal tankers.

Podded propulsion provides good open water efficiency and easy arranagement of general spaces.

# Aker Arctic LNG Solution

Aker Arctic

<u>Open water Mode:</u>

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Podded propulsion provides good open water efficiency and easy arranagement of general spaces.

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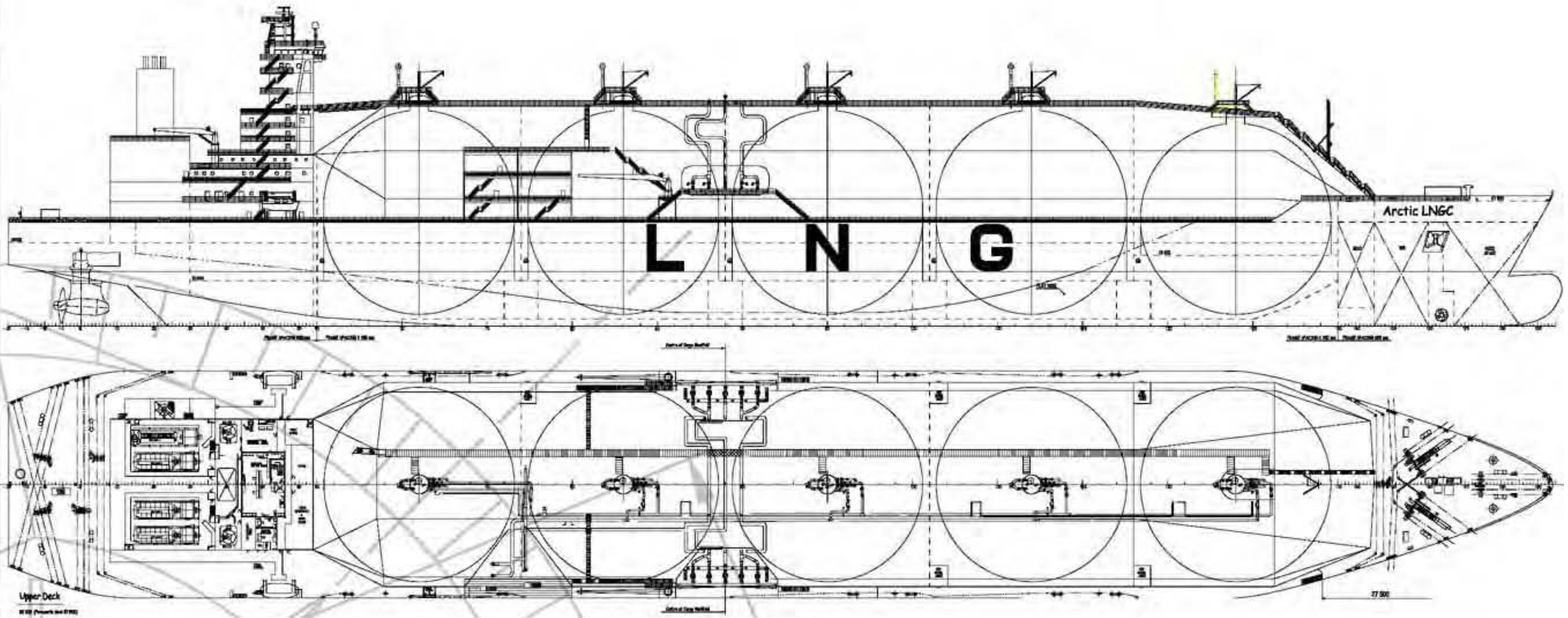
The vessel will operate stern ahead with pulling propellers.

Very good ice breaking capability can be achieved with normal open water propulsion power.

# Double Acting Arctic LNG Carrier

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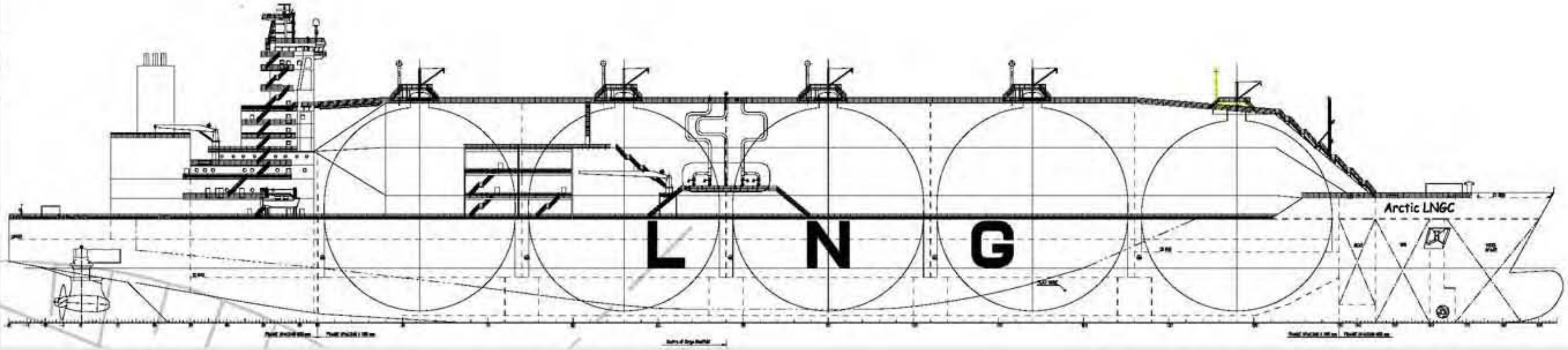
# Double Acting Arctic LNG Carrier




**Main Dimensions**

| | |
|---|---|
| Loa | 340 m |
| Lpp | 324.90 m |
| Breadth, moulded | 50.00 m |
| Depth | 22.90 m |
| Design draught | 12.00 m |
| Deadweight design | abt. 92,650 t |
| Scantling draught | 12.7 m |
| Deadweight scantling | abt. 95,800 t |
| Gross tonnage | abt. 133,000 |
| Cargo capacity | 206,000 m3 |
| Speed design draught | 19.5 kn |

Ice breaking performance @ 5 knots astern:1.5m / ahead:70cm

Radius of action abt.13,000 nm

**Class**

LR+100A1, Liquefied Gas Carrier, Ship Type 2G (Methane in independent tanks, Type B, Max. pressure 0.25 kg/cm2, Min. temperature -163°C) *IWS, +LMC, UMS, NAV1, IBS, SCM, LI, Ship Right (SDA, FDA, CM), ICC, TCM

Deep well pumps tanks 155 m.l.c 10 x 1,600 m3/h

Boil off rate 0.15%

Combined cargo heater/ vaporizer

Inert gas/venting plant

Bow thruster abt. 2,000 kW

**Machinery**

diesel electric propulsion,

2 x Azipod: Output 2 x 20,000 kW

Diesel generator aggregates dual fuel total abt. 46,000 kW

Fuel consuption 7.3t/h

Integrated automation system

Voltages for main consumers 11kV/6.6kV, 450/230V, 60Hz

Water/CO2/powder fire fighting system

Accommodation

Crew cabins incl. Pilot 49 pers.

# Double Acting Arctic LNG Carrier

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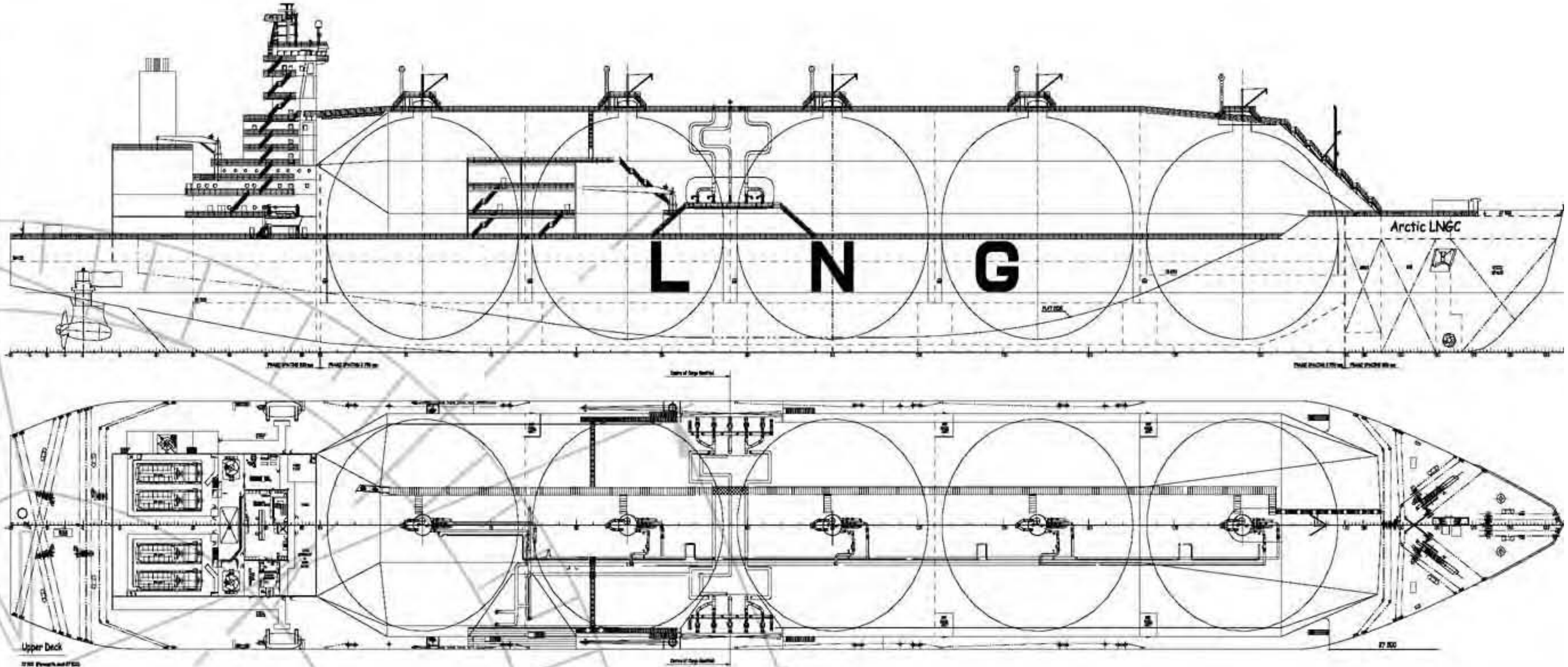


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# Model test program

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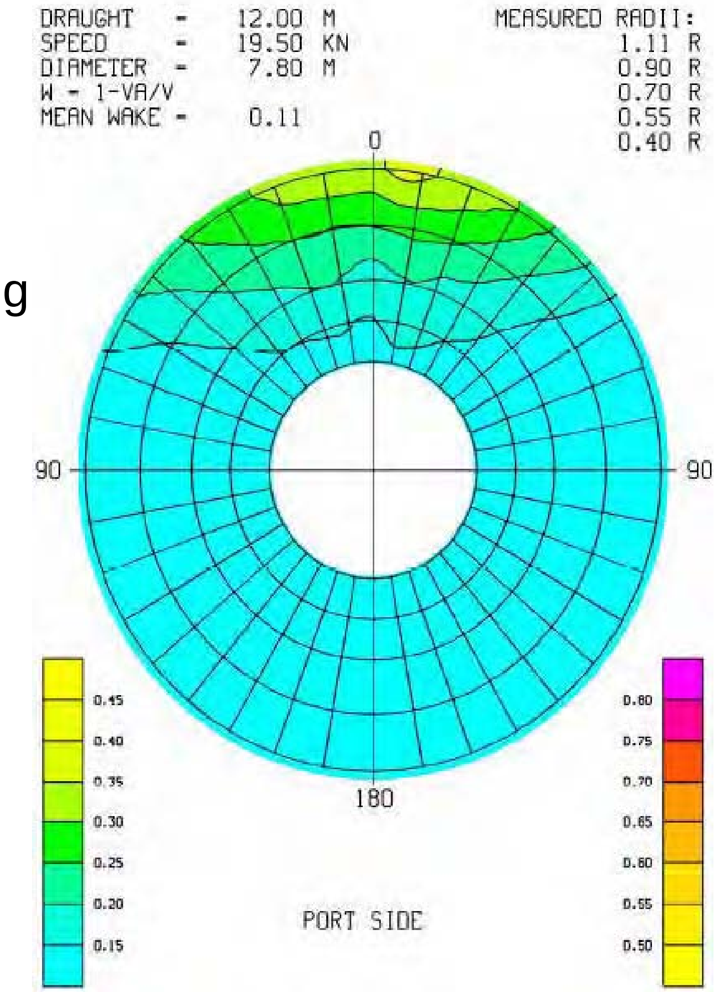
Model A

•DAT Icebreaking LNG Carrier with bulbous bow.

•Model B

•DAT Icebreaking LNG Carrier with sharp icebreaking bow.

•Testing was carried out at VTT model test basin in Espoo, Finland.




Resistance

Wake

Tuft tests

Propulsion tests

# Model test program

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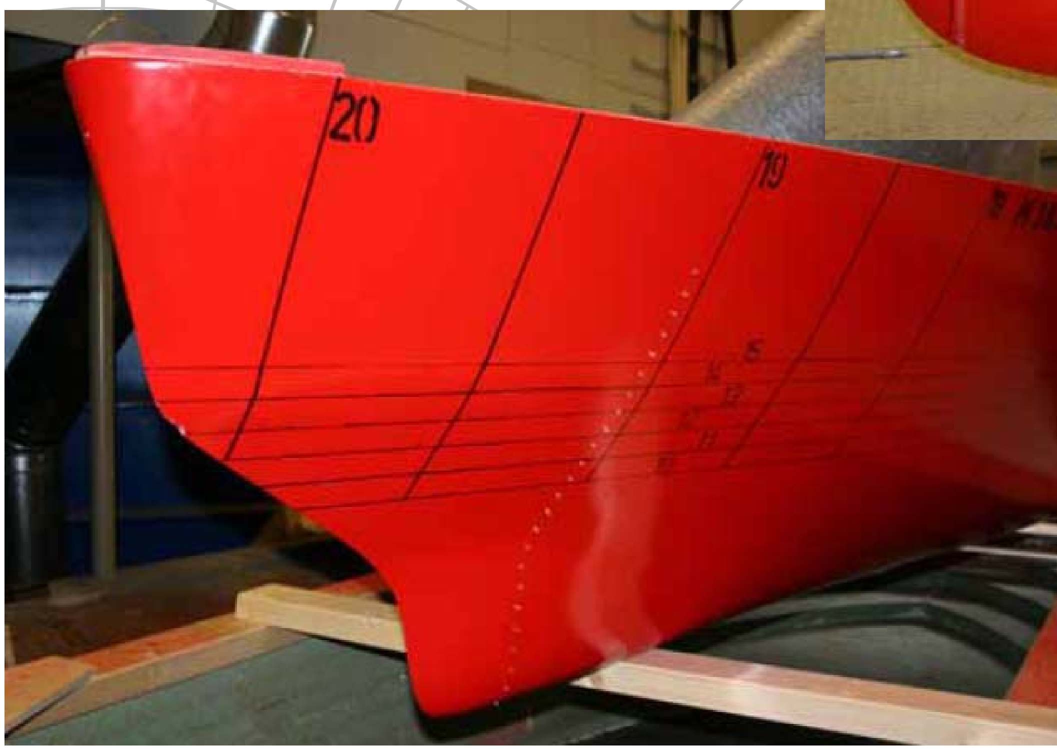


Model B


Model A

# Wave patterns

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# Wave patterns



# Model testing Results

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Propulsion

- model is characterized by low wake

- thrust deduction is also low (0.10 – 0.14)

- hull efficiency lower due to ice breaking effects

- rotation direction evaluated

# Model testing Results

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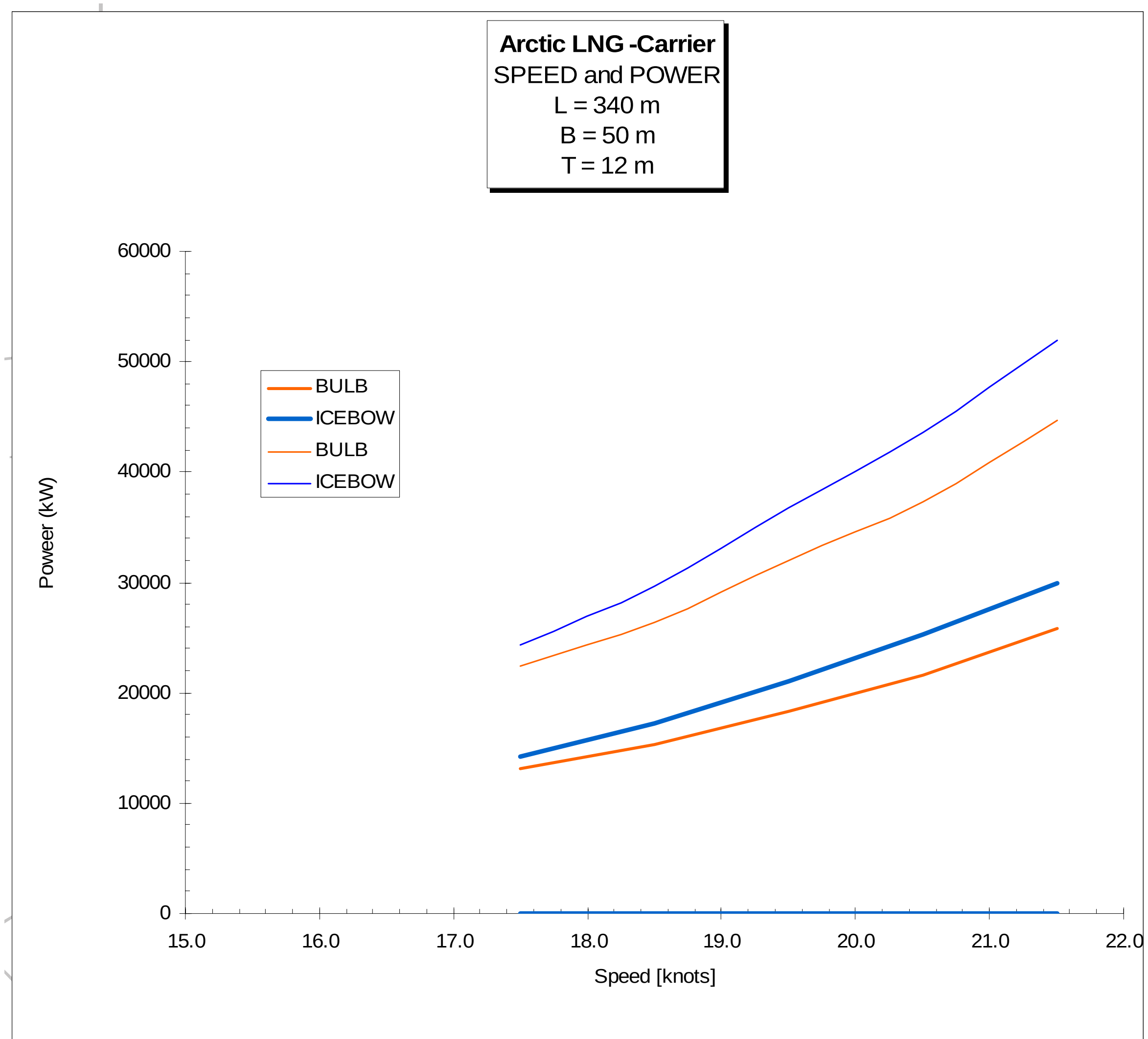
## Resistance and power




aker yards.

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# Icebreaking tests, LNG Carrier

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Arctic icebreaking LNG Carriers

- not iceclass design

-icebreaking capability required

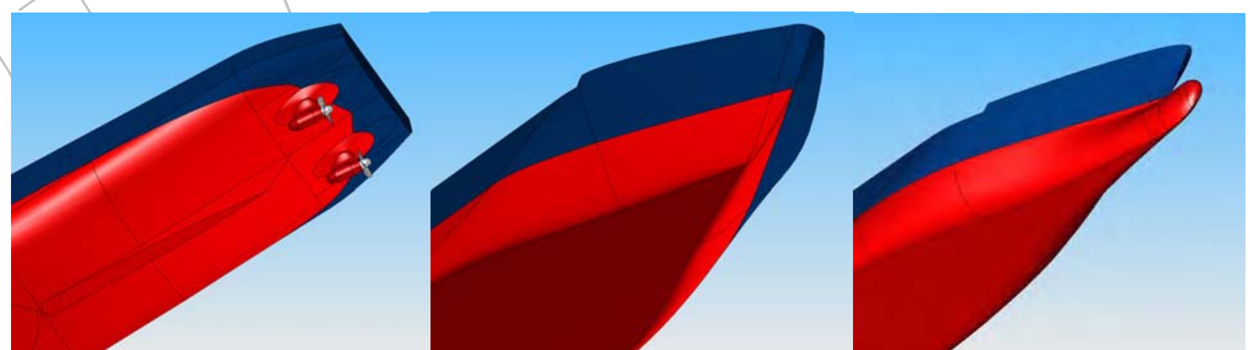
- design and strenghtening according to environmental requiremets

Three modes/versions developed and tested

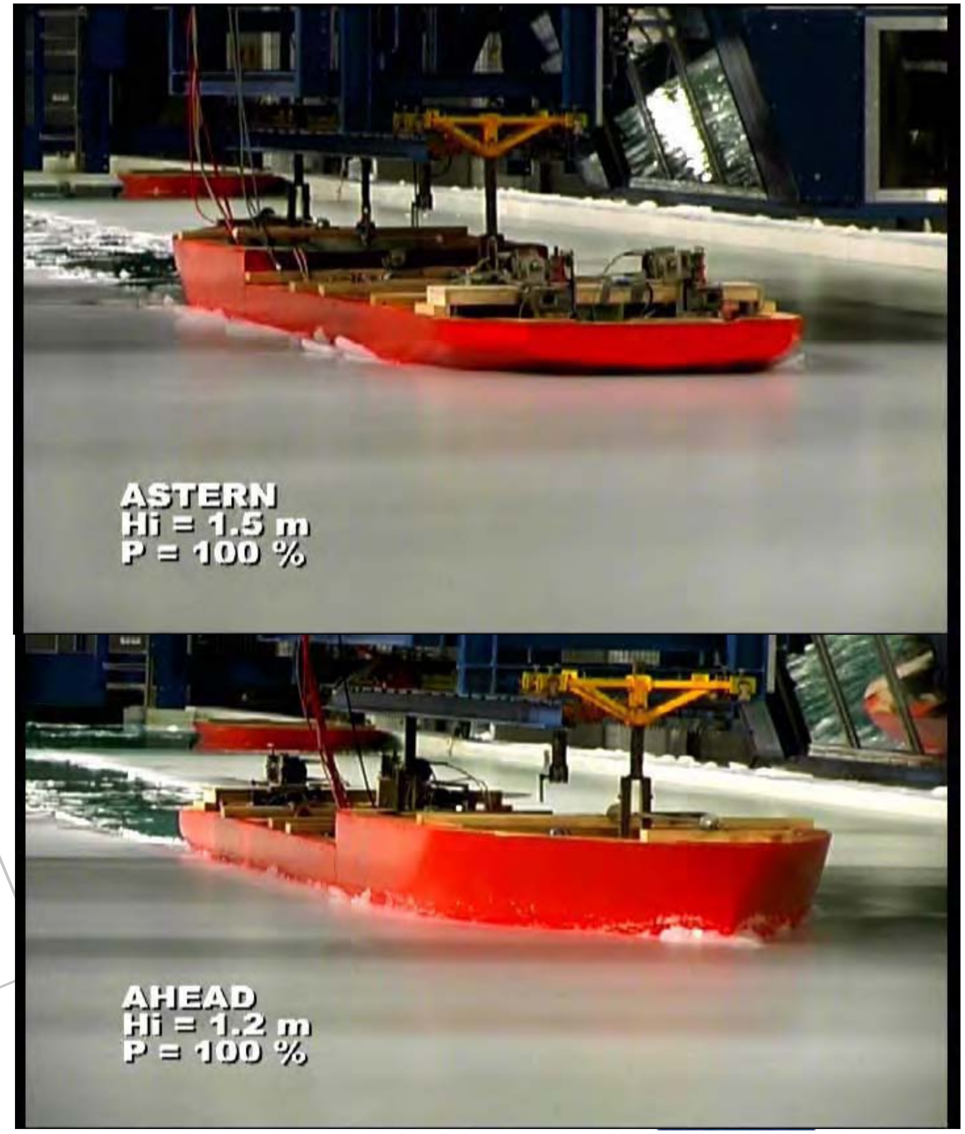
DAT stern

Icebow bow

Bulbous bow



# Arctic LNG model testing




## Test results astern

| | |
|---|---|
| 170cm | 4.9 knots |
| 150cm | 5.4 knots |
| 120cm | 7.0 knots |

## Test results ahead

| | |
|---|---|
| 150cm | 0 knots |
| 120cm | 1.0 knots |

# Arctic LNG model testing

## Test results astern

| | |
|---|---|
| 170cm | 4.9 knots |
| 150cm | 5.4 knots |
| 120cm | 7.0 knots |

## Test results ahead

| | |
|---|---|
| 150cm | 0 knots |
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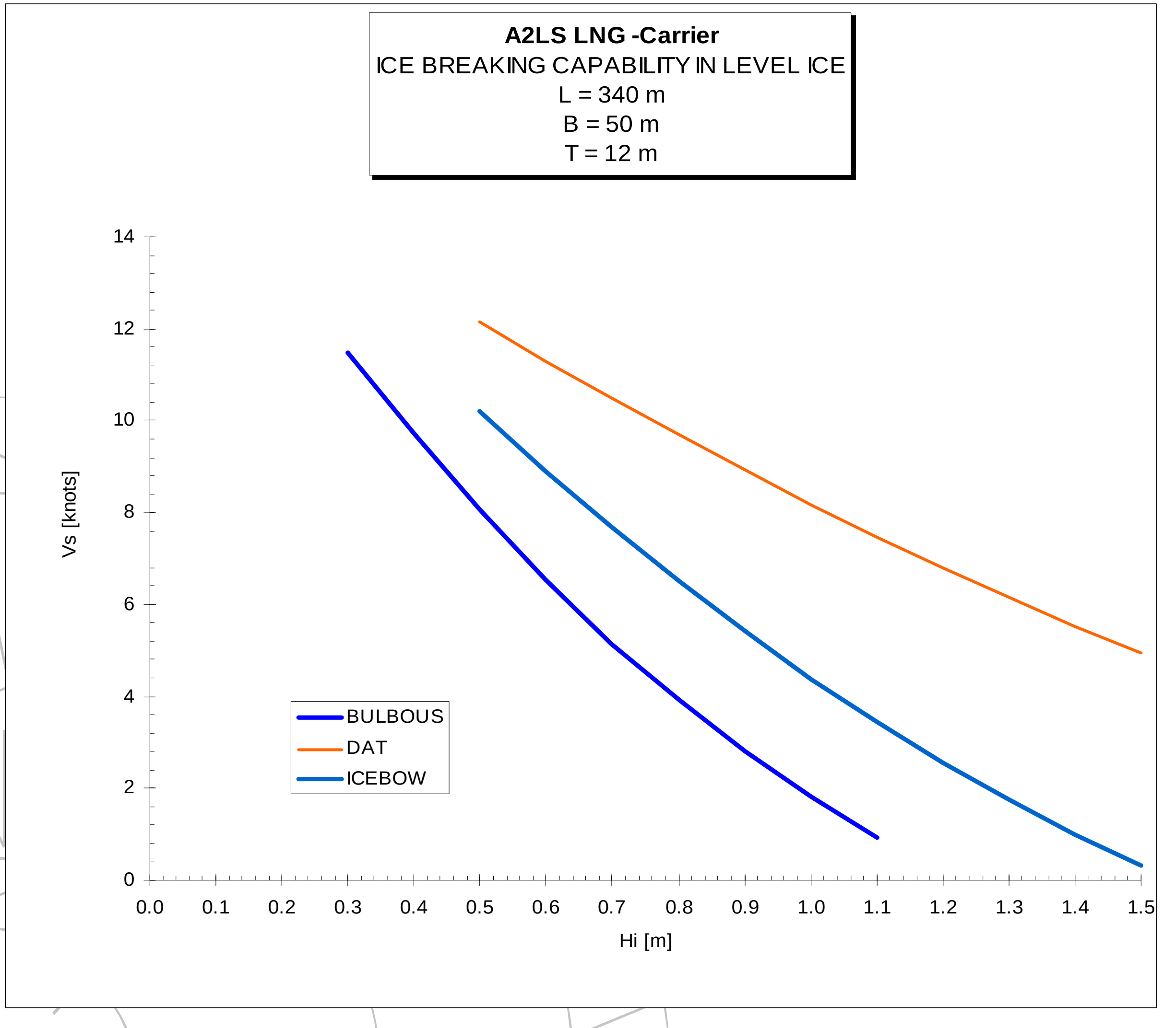
# Icebreaking LNG Carrier

Aker Arctic




Icebreaking capability

DAT stern

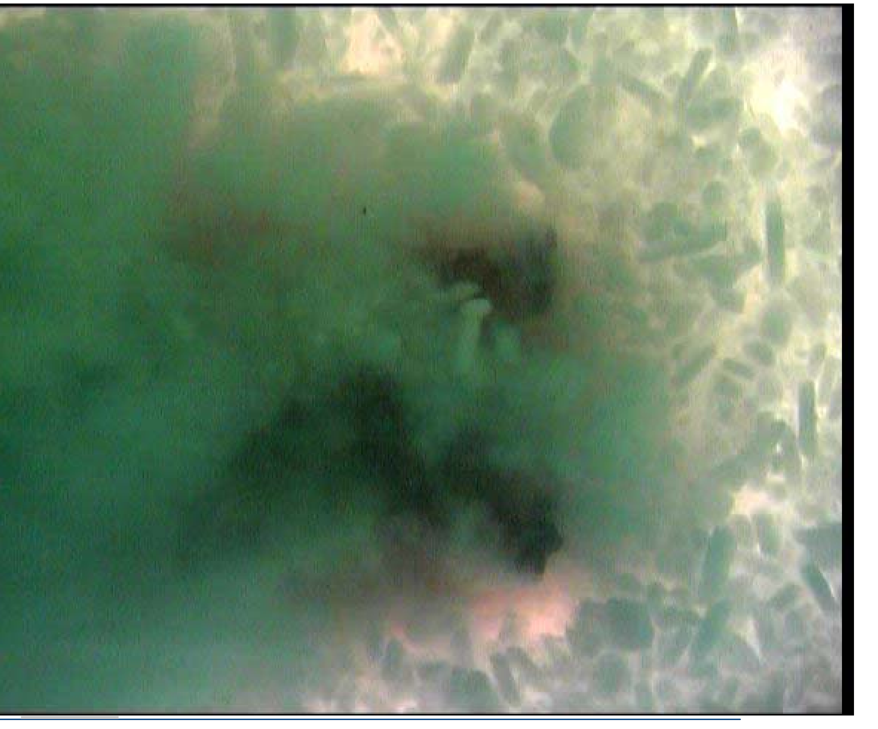
Icebow

Bulbous bow

# Arctic LNG model tests

## Ridged ice astern

11-14m , constant motion 0.5 knots

Arctic LNG model tests

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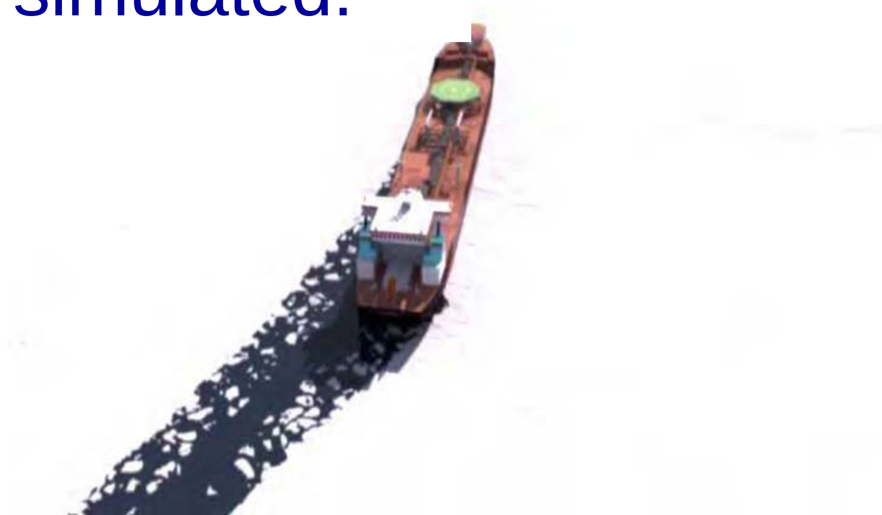
## Ridged ice ahead (icebow)

11 – 13 m, 2 rams required

Maximum deceleration 0.075 m/s2

# What ice conditions can be simulated:



- Level ice
- Channel ice
- Snow thickness
- Ice concentration
- Average floe size
- Ridge density
- Ridge average height and distridution
- Consolidation (ridge and channel)
- Ice compression

# Arctic LNG model tests
# Simulation

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# Arctic LNG model tests
# Simulation

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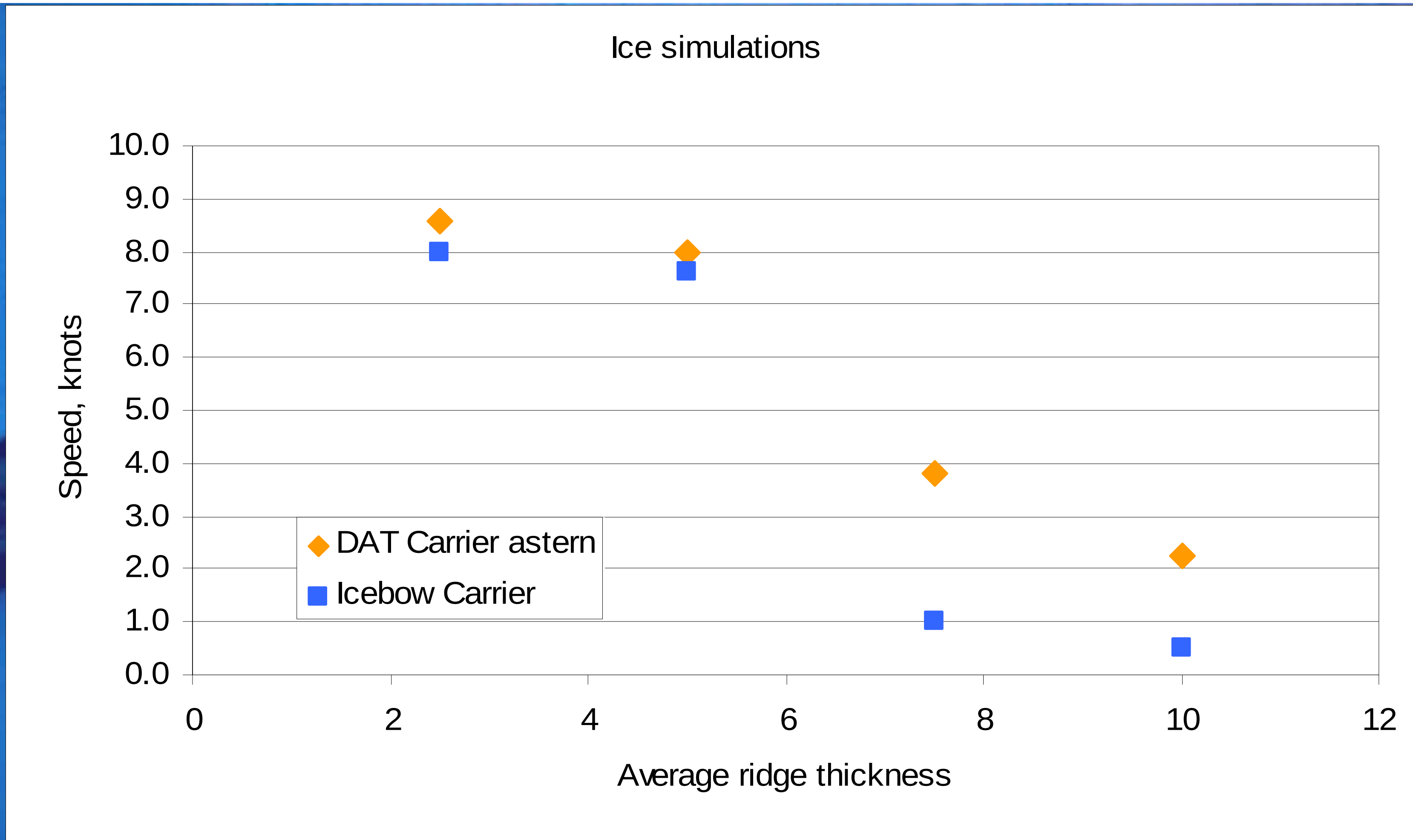

# Simulation

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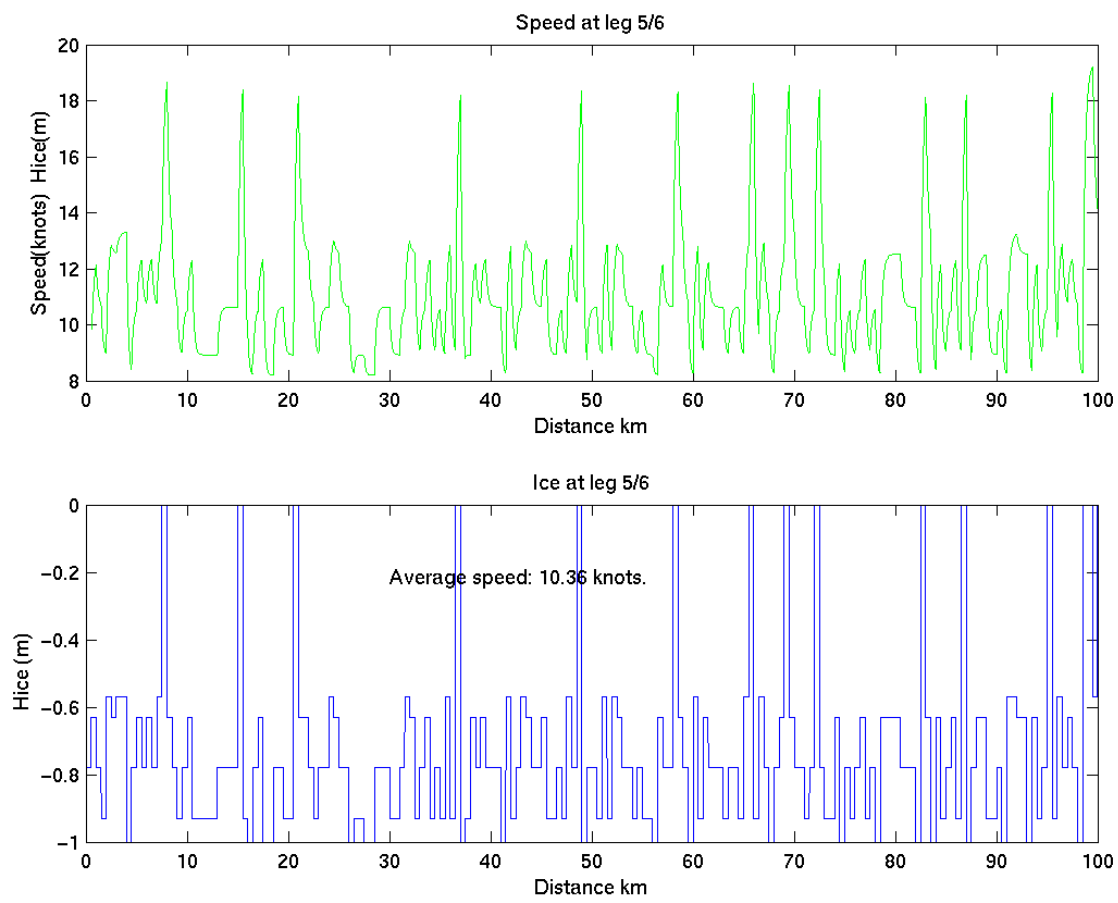
## Results

## Speed profile

## Ice profile



# Arctic LNG model tests

MAIN CONCLUSIONS FROM MODEL TESTING

Large vessels can have excellent ice breaking capability.

Also bulb bow can be designed for heavy icebreaking.

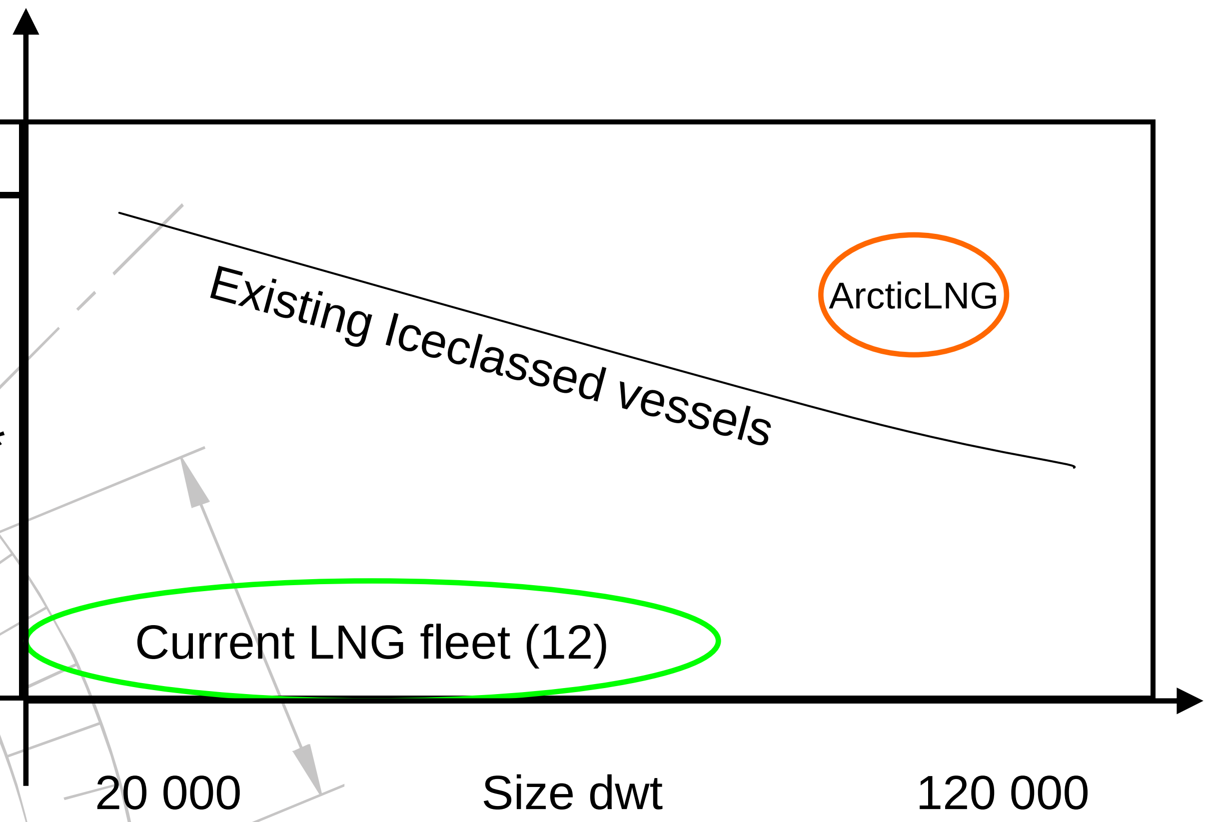
Penetration of ridges more easy than predicted.

Simulations based on test values show good average speeds in Kara Sea crossing!

With optimisation open water resistance is expected improve.

# Arctic Iceclass vessels

| Baltic | RMRS | DNV |
|---|---|---|
| | LU8 | Ice-15 |
| | LU7 | Ice-10 |
| | LU6 | Ice-05 |
| 1ASupe | LU5 | Ice-1A* |
| 1A | LU4 | Ice-1A |
| 1B | LU3 | Ice-1B |
| 1C | LU2 | Ice-1C |



# Arctic Iceclass vessels, combining the experience

Aker Arctic

170m Arctic DAS Container vessel

LU7 ice class 1 x 13MW

2 winters of self icebreaking operation

252m Arctic DAS tanker 106 tdwt

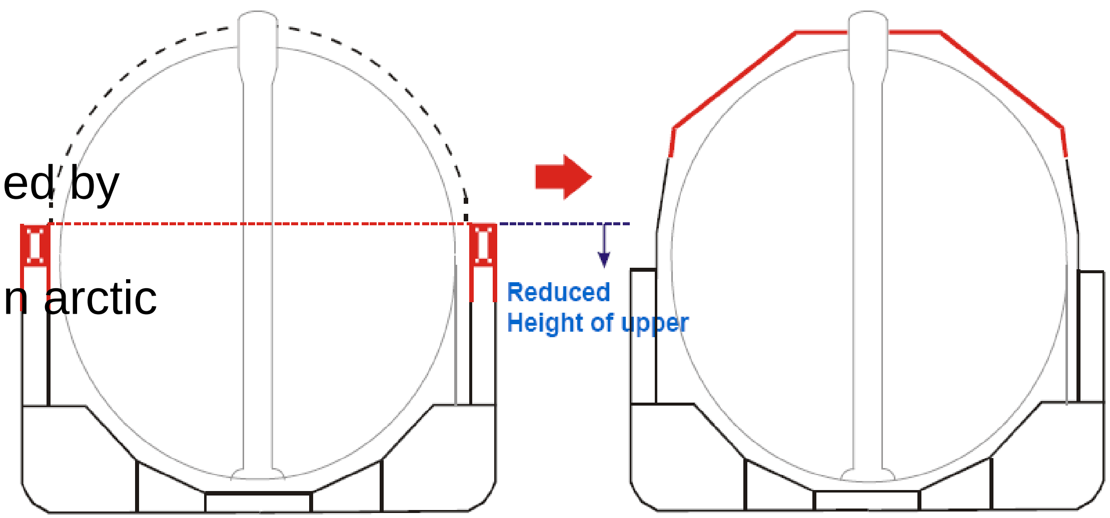
1ASuper ice class 1 x 16MW

5 winters of self icebreaking operation


# Arctic Iceclass vessels, combining the experience

## Aker IHS-concept

- Elimination of the negative consequences from large deck openings
- Increased stiffness of hull
- Reduced steel weight, up to -10 %
- Improved maintenance aspects
- Reduced construction cost.
- Lower freeboard
- Longitudinal strength well controlled by the design, distributed on wide
- Less structures exposed to icing in arctic waters

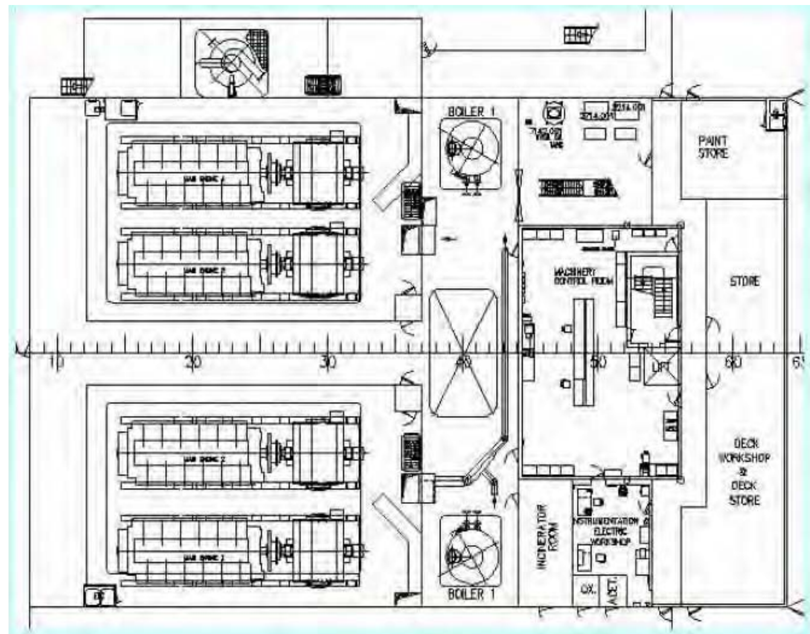


# Machinery selection

Aker Arctic

- Gas-diesel-electric propulsion
- Dual-fuel diesel engines running generators
- Engines will burn the natural boil-off gas and forced vaporized cargo gas with a small quantity of liquid fuel for ignition.
- The engines will mainly run on fuel gas with liquid MDO fuel as back-up or as alternative fuel, and can be switched over automatically as the need arises.

4 x Wärtsilä 12V50DF (45.6 MW)



## Propulsion system

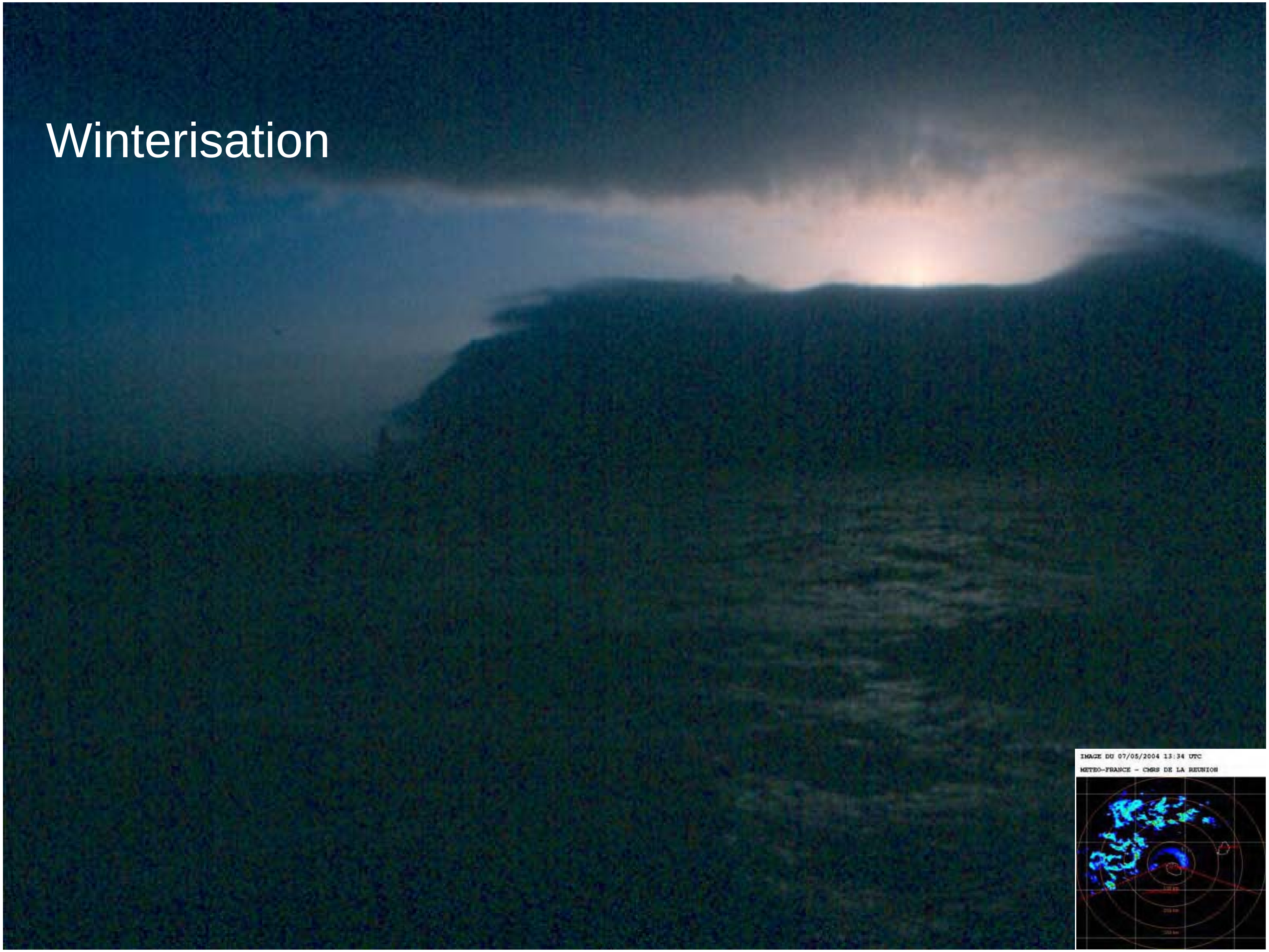
Aker Arctic

- Design has 2 x 20MW Azipod propulsion units.
- Units are dimensioned to meet the icebreaking loads.
- Current largest units are 13MW for heavy ice class.

Requires new size of pod unit.

Propeller dia. 7.8m.
Nominal rpm 120.
Max. torque 2800 Nm

# Winterisation




Winterisation

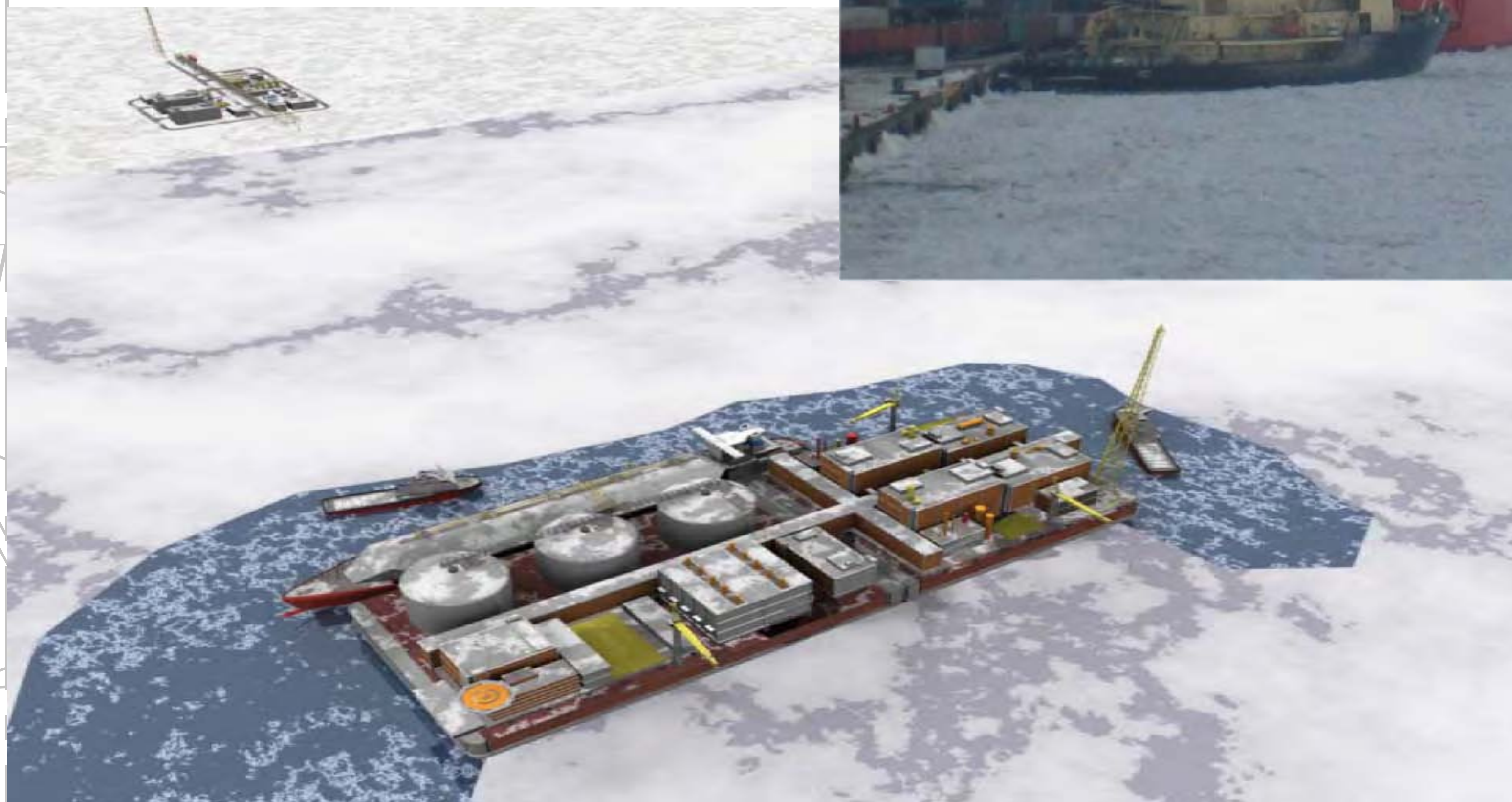
Experience allready existing for decades.

- spray, deck equipment
- heating, insulation
- machinery room arrangements
- navigation systems
- life saving

LNG specific issues.

- icing (IHS protection)
- cold environment
- over cooling
- leak calculations

# Loading
# Ice management



Preferred for Innovation

# MAIN CONCLUSIONS

Aker Arctic

Icebreaking DAT LNG Carrier is feasible solution for Arctic trades.

Wintertime speed reduction can be kept on acceptable level.

Ice indused drop of speed or down time in loading can be handled with reasonable number of LNG Carriers.

Offshore loading can be operated with the efficient ice management fleet.

# FUTURE WORK



- Detailed design of propulsion system (2 x 20 MW)
- Optimisation of hull form
- Follow-up work with reference vessels
- Handling of the load variations in power plant and propulsion drive system, sensitivity of DF engines
- Ice indused vibration effects
- Ice deflections in the hull <> tank system
- Offshore loading in Arctic environment

# Arctic LNG – is a today’s possibility

Aker Arctic

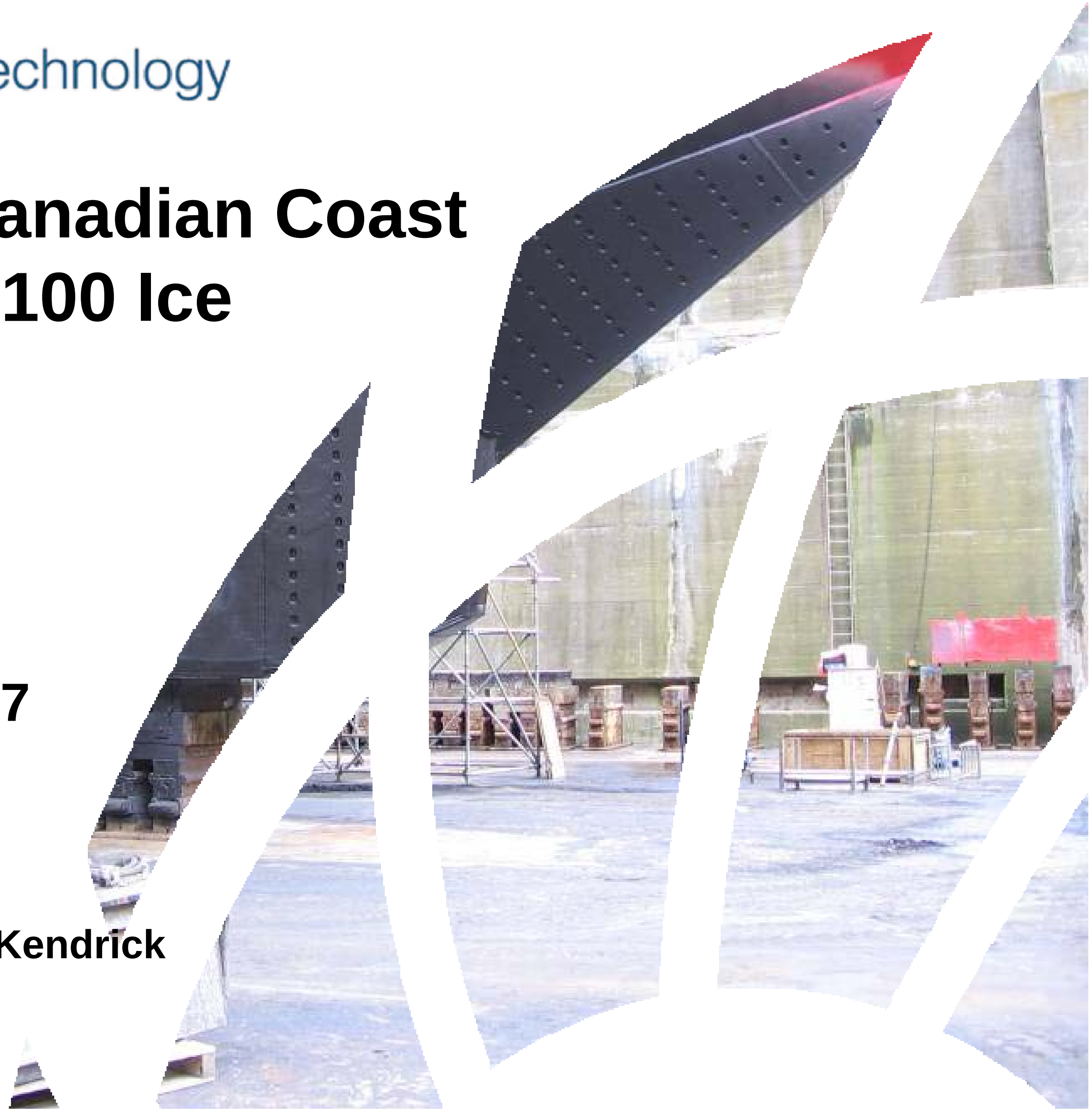

**Thank you!**

Preferred for Innovation

# Copyright

BMT Fleet Technology

# Upgrade of Canadian Coast Guard Type 1100 Ice Capability



**RINA**
**London June 2007**

**David T Stocks**
**Lindsay Fyfe**
**Presented by Andrew Kendrick**

# CCGS  Sir Wilfrid Laurier

Named after Canada's 8th prime minister 1896 – 1911

Built in 1986 by Canadian Shipbuilding Collingwood  Ontario

Multi Purpose vessel primary role Buoy tender.

| | |
|---|---|
| Displ: | 3,727 tons |
| Length: | 83 m |
| Beam: | 16.2 m |
| Draught: | 6 m |
| Propulsion | Diesel Electric |
| Speed: | 15.5 knots |
| Range: | 6,500 nautical miles |
| Ice Class: | CAC 2 |

# Ice breaking performance issues

Operates annually from Pacific coast ports to Western Arctic in summer.

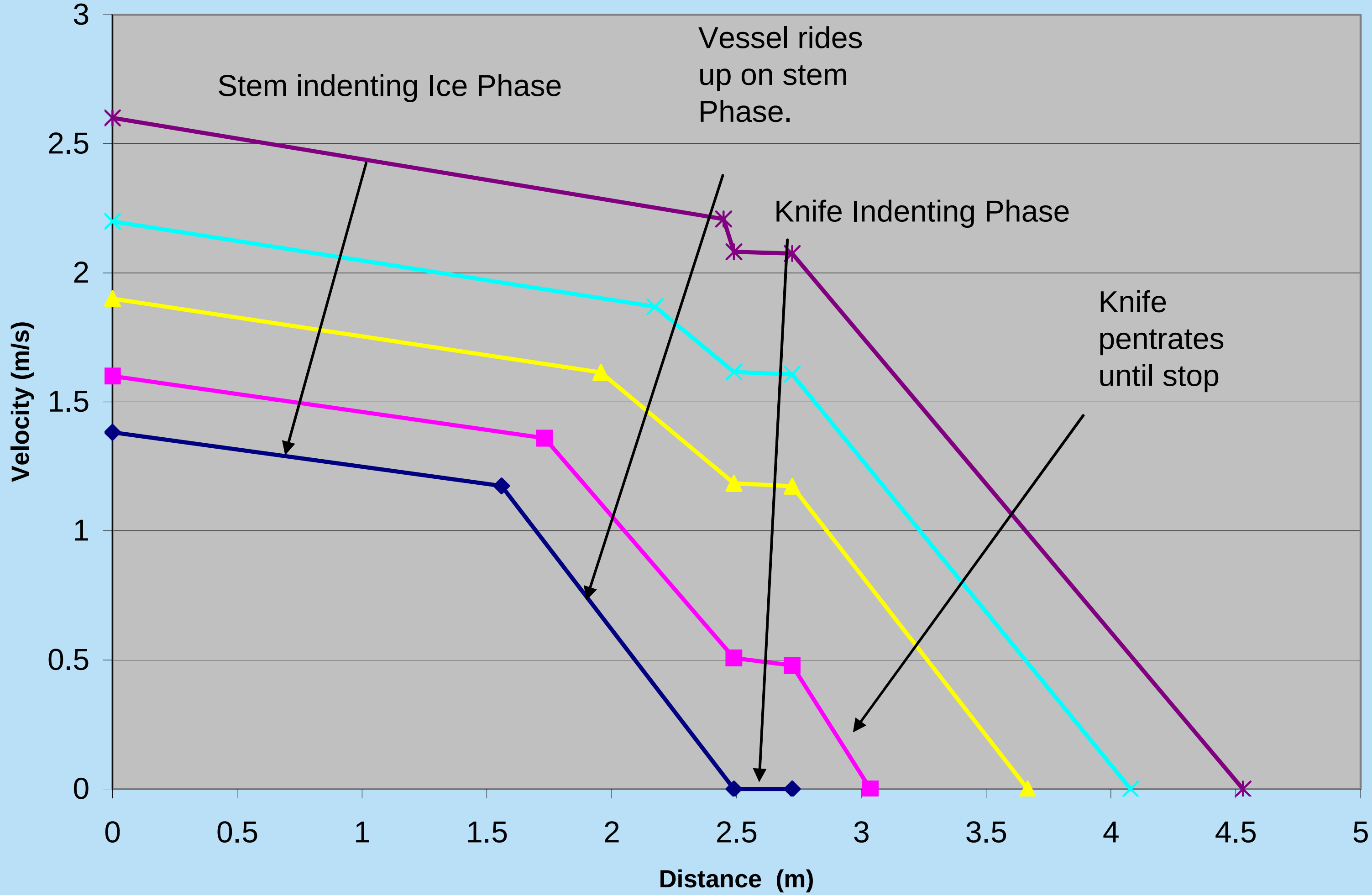
- Experiences rapid deceleration when operating in Ramming mode in heavy ice.
- Tends to push ice ahead of vessel


More significant in Light draft condition.

Opportunity as first of Class to undergo vessel life extension programme.

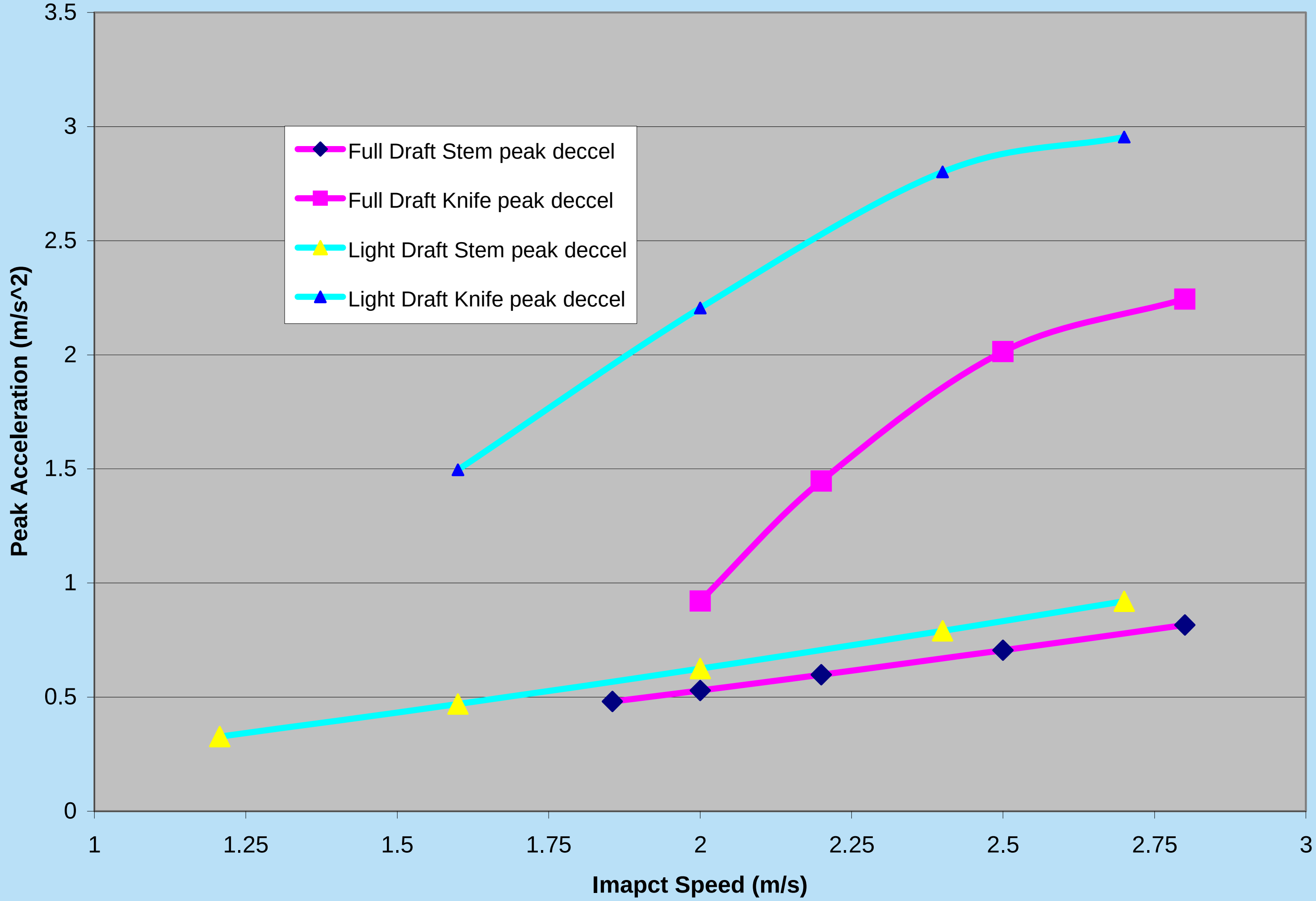
BMT Fleet Technology

# Ice Penetration –v- initial impact speed



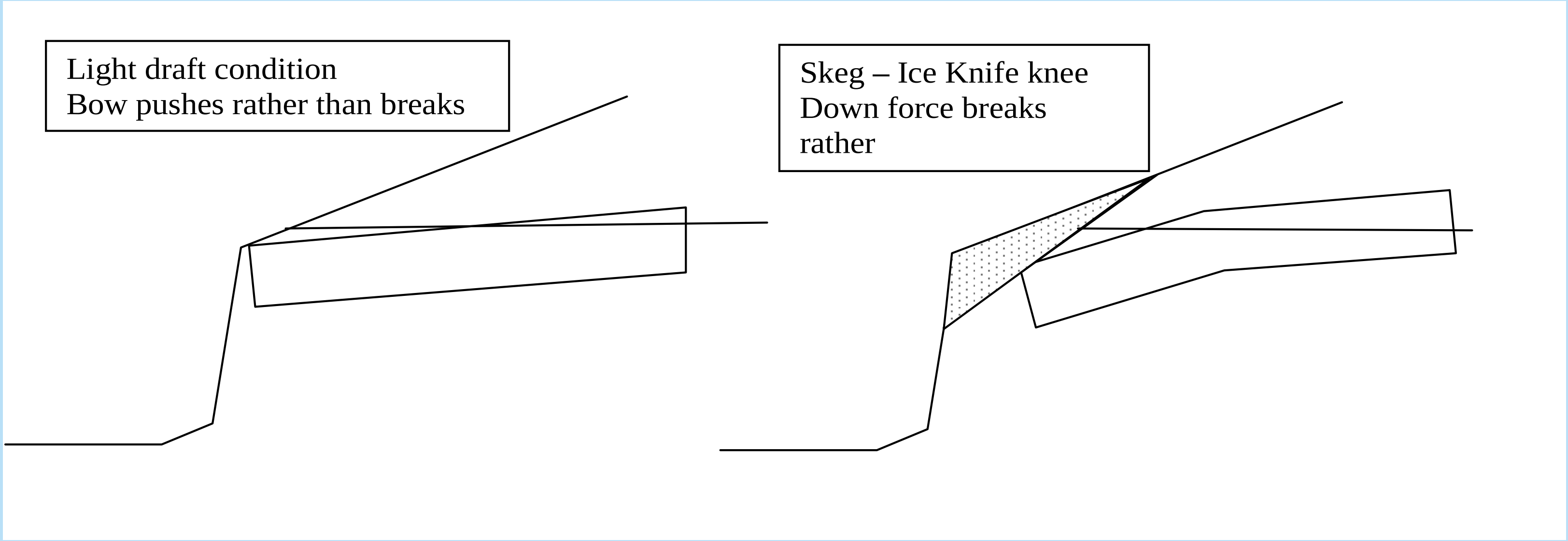

BMT Fleet Technology

# Peak decelerations at first impact and Ice Knife (skeg) penetration



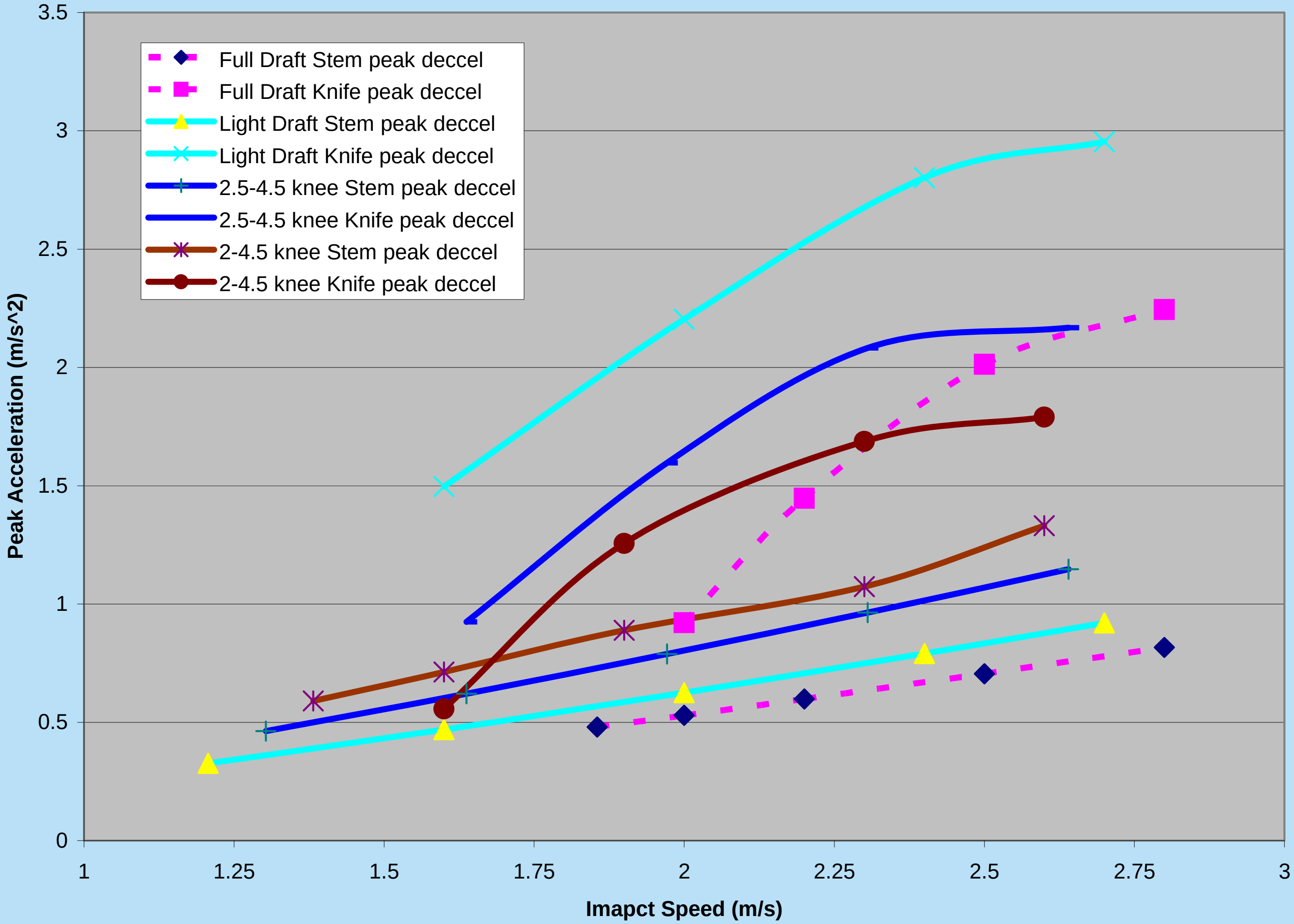

BMT Fleet Technology

# Redistributing Energy




BMT Fleet Technology

# Redistributing deceleration




BMT Fleet Technology

# Other structural issues

T 1100 design basis was 1972 ASPPR: strong bow but weaknesses in other hull areas

Prone to damage when encountering multi-year ice

Refit provided opportunity to upgrade damaged plate and restore coatings

# Fitting the Knee – vertical plate & flg stiff's

# Fitting the side plates

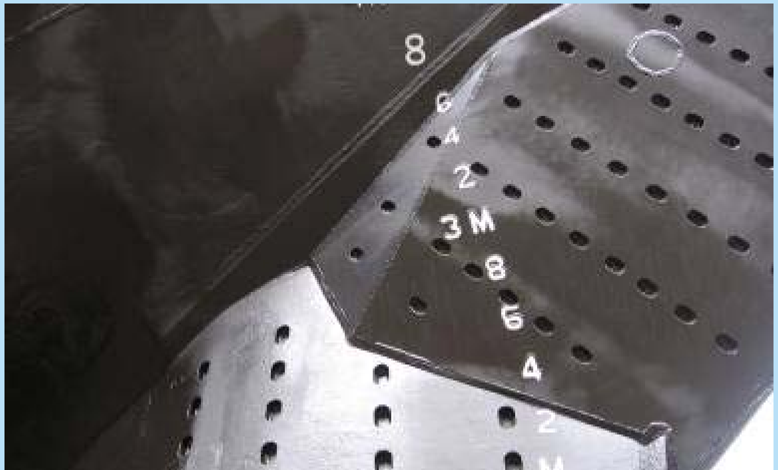
BMT Fleet Technology

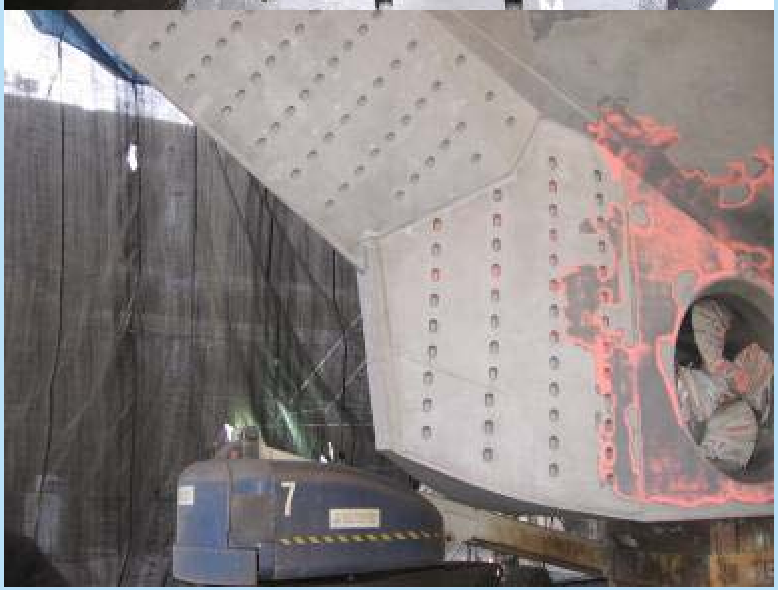
# Issues in the dock

Sniped
plate

Docking
plug

# Finished bow ready to float.




# Future

**2007 summer will be first foray into heavy ice;
we await infield experience.**


**Thank you**