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EFFECT OF THE FSICR TO PROPELLER EFFICIENCY

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FOREWORD

In this report no 108, the Winter Navigation Research Board presents the results of the study on the effect of the FSICR to propeller efficiency and thus relevant correction factors for propulsion power into EEDI index for ice classed ships.

The study covers propellers for ice classes 1C, 1B, 1A and 1A Super for two vessel types and sizes sailing in the Baltic frequently also in winter time are selected. For both ship-types an open water propeller is designed. The performance data recorded allows clear and transparent comparison of performances between the ice classes.

The Winter Navigation Research Board warmly thanks Mr. Ilkka Saisto and Mr. Tuomas Turunen for this report.

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EFFECT OF THE FSICR TO PROPELLER EFFICIENCY

FOR

FINNISH-SWEDISH WINTER NAVIGATION RESEARCH BOARD

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Effect of the FSICR to propeller efficiency

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Summary:

The target of the study is to find out the effect of the FSICR to propeller efficiency and thus relevant correction factors for propulsion power into EEDI index for ice classed ships. The study covers propellers for ice classes IC, IB, IA and IAS for two vessel types and sizes sailing in the Baltic frequently also in winter time are selected.

For both ship-types an open water propeller is designed. The blade thickness of initial propeller design is increased to propellers that fulfils the strength requirements of IC, IB, IA or IAS.Finally, the open water curves of the propeller families are evaluated. The performance data calculated allows clear and transparent comparison of performances between different ice classes.

For the vessel 1, the single screw vessel of bulk carrier type, the relative delivered power increase, due propeller strength demands, at optimization point compared to open water propeller is 102.8% for IC and IB ice class, 103.3% for IA and 104.3 % for IAS ice class.

For the vessel 2, the twin-screw RoRo or ferry, the relative delivered power increase, due propeller strength demands, at optimization point compared to open water propeller is 100.2 % for IC and IB ice class, 100.9 % for IA and 101.4 % for IAS ice class.

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1 INTRODUCTION

The regulations on energy efficiency of ships in Chapter 4 of Annex VI to the MARPOL Convention (EEDI regulations) will affect the world shipping and especially the transport regime in the Baltic Sea area, where high ice classes are the basis of year-round regular transport. The ice strengthening of propellers required by the Finnish-Swedish Ice Class Rules (FSICR) affects the propeller efficiency of ice classed vessels adversely. It may be necessary to take this into account in the calculation of the EEDI number for ice classed vessels.

The target is to find out the effect of the FSICR to propeller efficiency and thus relevant correction factors for propulsion power into EEDI index for ice classed ships. The study covers propellers for ice classes IC, IB, IA and IAS.

The aim of the work is to provide information for the sessions of the Marine Environment Protection Committee (MEPC) of the International Maritime Organization (IMO) and future development of the EEDI criteria for the Baltic ice class tonnage.

The research aims to address the subject by comparing the open water performance of the ice-strengthened propellers to that of open water propellers. The comparison is based on modern Computational Fluid Dynamics (CFD) tools created for marine propeller design. The tools were developed in the joint project by TEVO Oy, Aker Arctic Technology Inc. and VTT during the years 2014 through 2016.

Two vessel types and sizes sailing in the Baltic frequently also in winter time are selected. From the propulsion point of view the vessels are different, the one is typical single screw bulk carrier or tanker and the other is twin shaft RoRo or ferry with higher service speed. These vessel types form a representative set of examples of traffic in the Baltic Sea and thus give a solid basis for future estimates of reasonable EEDI criteria.

For both ship types, an open water propeller is designed. The blade thickness of initial propeller design is increased to propellers that fulfils the strength requirements of IC, IB, IA or IAS.

The hydrodynamic design is based on a lifting line code in terms of initial optimization and a 3D Computational Fluid Dynamics code, which is used for final optimization and performance calculations.

The propellers for each vessel were evaluated against the 2017 FSICR to assess the strength of the blade. The FSICR assess the strength of a propeller through two methods: minimum ct^2 values for the inner portion of the blade and five load cases for the outer portion of the blade.

Finally, the open water curves of the propeller families are evaluated. The performance data calculated allows comparison of performances between different ice classes.

2 VESSEL CASES – INPUT DATA

Two vessel types and sizes sailing in the Baltic frequently also in wintertime are selected to this study. These vessel types form a representative set of examples of traffic in the Baltic Sea and thus give a solid basis for future estimates of reasonable EEDI criteria. From the propulsion point of view the vessels are different, the one is typical single screw bulk carrier or tanker, vessel 1, and the other is twin shaft RoRo or ferry with higher service speed, vessel 2.

In general, from propellers design point of view, the propellers should have enough bollard pull thrust depending on required vessel ice-going performance. The blade area ratio of the propellers shall be adequate to prevent deterioration effect of cavitation to bollard pull and free-running at maximum main engine (M.E.) power conditions. The propellers should also be free of cavitation erosion. As material of the propellers bronze is selected.

2.1 VESSEL 1.

Vessel 1 is a single screw bulk carrier equipped with fixed pitch propeller with diameter 5.1 m mechanically coupled to one 4-stroke main engine through reduction gear. Maximum continuous power of the main engine is assumed to be 8400 kW**,** attainable at the engine nominal rotation speed of 500 rpm. Shafting efficiency including gearbox is assumed to be, η_s = 0.95, with gearbox reduction rate ~3.7.

- Design draught of the ship 8.5 m
- V-type aft ship hull form with centre-line (CL) shaft gondola.
- Propeller diameter 5.1 m (molded fixed pitch propeller)

Table 1. Main data of the Vessel 1 propeller

Pitch			Fixed
Diameter	D	(mm)	5100
Hub diameter ratio	d/D	(approx.)	0.22
Number of blades	Z		4
Shaft immersion		(mm)	6000 (at propeller disk)
Tip clearance		(mm)	1450 (to hull surface (mld.))
Ice class			case depended
Propeller material			Bronze

The most important hydrodynamic performance characteristic of the propeller is a high open water efficiency at the ship design service speed of 14 knots.

The ship hull towing resistance at standard seawater condition (ρ=1026 kg/m^3) is presented in Table 2:

Table 2. Vessel 1 towing resistance at standard seawater condition

V_S [kn]	R_{TS} [kN]
4	29
7	79
11.5	211
14	345
16.5	597
18	820

The designed propeller should provide ship speed of 16.5 knots in trial conditions at max. 85 % of M.E. Maximum Continuous Power Rating (MCR) and propeller revolutions of 130-136 rpm. Constant values of propulsion interaction coefficients are used across speed range for free-running condition:

- thrust deduction factor t = 0.17
- nominal axial wake fraction coefficient w_{nx} = 0.325
- propulsion relative efficiency η_R = 1.03

2.2 VESSEL 2.

The vessel 2 is a twin shaft RoRo or ferry with higher service speed of 18 knots. The vessel has two-fixed pitch propeller mechanically coupled to two 4-stroke main engines through reduction gear. The maximum continuous power of both main engines is assumed to be 9600 kW, attainable at the engine nominal rotation speed of 500 rpm. The shafting efficiency including gearbox is assumed to be, η_s = 0.95, with gearbox reduction rate ~3.1.

- Design draught of the ship 6,9 m
- V-type aft ship hull form with CL shaft gondola.
- Propeller diameter 5.1 m (molded fixed pitch propeller)
- Turning direction is not specified.

Table 3. Main data of the Vessel 2 propeller

Pitch			Fixed
Diameter	D	(mm)	4500
Hub diameter ratio	d/D	(approx.)	0.22
Number of blades	Z		4
Shaft immersion		(mm)	4500 (at propeller disk)
Tip clearance		(mm)	1150 (to hull surface (mld.))
Ice class			case depended
Propeller material			Bronze

The most important hydrodynamic performance characteristic of the propeller is a high open water efficiency at the ship design service speed of 18 knots.

The estimated ship hull towing resistance at standard seawater condition (ρ=1026 kg/m^3) is presented in Table 4:

Table 4. Vessel 2 towing resistance at standard seawater condition

Vs [knots]	Rt [MN]
8	0.108
9	0.135
10	0.165
11	0.199
12	0.236
13	0.277
14	0.324
15	0.376
16	0.436
17	0.508
18	0.588
19	0.680
20	0.790
21	0.918
22	1.056

The designed propeller should provide ship speed of 20.5 knots in trial conditions at max. 85 % of MCR and propeller revolutions of ~155 rpm.

Constant values of propulsion interaction coefficients are used across speed range for free-running condition:

- thrust deduction factor t = 0.14
- nominal axial wake fraction coefficient w_{nx} = 0.12
- propulsion relative efficiency η_R = 0.98

The propeller should also have a sufficient bollard pull thrust depending on required vessel ice-going performance.

The blade area ratio of the propeller shall be adequate in order to prevent deterioration effect of cavitation to bollard pull and free-running at maximum main engine power conditions.

3 PROPELLER DESIGN

The hydrodynamic design is based on a lifting line code in terms of initial optimization and a 3D Computational Fluid Dynamics code which is used for final optimization and performance calculations.

The lifting line code presents the propeller blade as a single line with circulation spanning from the propeller hub to blade tip. The code takes blade profiles and their hydrodynamic performance data as inputs. The main outputs of the code are pitch and camber distributions which give an optimal performance under given constraints (such as thrust or torque requirement) and the assumptions associated with the method. More information about the lifting line method can be found e.g. in Reference [7], Chapter 8.

The final performance calculation and the optimization of chord and camber distributions are performed based on CFD computations. CFD also allows to evaluate pressure distributions in more detail and make decisions based on those.

The CFD computations are steady-state analysis which have been proven to predict global forces such as propeller thrust and power with a good accuracy. In literature, this is referred to as the Moving Reference Frame (MRF) method which solves the Navier-Stokes equations governing the fluid flow not in the inertial coordinate system in which the propeller rotates but in a coordinate system that moves along with the propeller. In such a system the propeller appears to be in a steady state.

The flow code used is an Open Source Toolbox, OpenFOAM [1], which has been previously tailored for use in propeller computations. A more detailed description of the algorithm including the discrete equations and guiding parameters can be found in Ref [2]. More generally literature about the methods can be found in, for example, References [3] and [4].

The physical discretization of the fluid domain contains the propeller blade, hub and the surrounding region. The inlet and outlet boundaries are located 10 propeller diameters from the propeller plane and the outer (cylindrical) boundary is located four propeller diameters from the rotation axis. Only one blade is included in the computation and the rest of the blades are accounted for by periodic boundaries. This reduces the computational time by a factor equal to about the number of blades but not the accuracy since the blades are identical to each other and differences in time are approximated to be negligible.

Propeller surroundings were discretized with tetrahedral elements while farther parts of the mesh were extruded from triangles. Propeller tip, leading edge and trailing edge have quadrangular elements and the flatter parts of the blade is discretized with triangles. The propeller boundary layer discretization is an extrusion from the propeller blade with a total of 15 boundary layer elements (wedges and cubes).

The cell sizes in the direction normal to the blade is 0.9mm which was checked to result in a non-dimensional distance of $y+ = 25\ldots100$ on the blade surface which ensures a proper performance of the turbulence model mostly responsible for friction prediction. Otherwise the surface grid dimensions were decided to allow the grid to represent the propeller geometry, in particular the curvature in the geometry. The cell size grows by a factor of max 1,2 between adjacent cells when the distance from the surfaces grows. The method applied is the T-Rex method implemented by Pointwise, Inc. A description of the method is given in Reference [8].

Three figures of one representative computational grid is given below (Case1, FSICR-IC). All grids were created using the same algorithm with identical parameters, so they look and behave very similarly. The grids have also been checked against quality problems and seen to be appropriate for use.

The numerical error of the computations is estimated to be below other modelling errors such as errors due to wake prediction and simplification, physics modelling and manufacturing tolerances. A grid comparison study was not conducted for the propellers studied, however, the CFD method has been validated against measured model scale data and against full scale data in a Lloyds Register Workshop on CFD in Ship Scale Hydrodynamics in 2016. Based on this previous experience and the fact that the analysis procedure is automated and the same for every propeller, the method is used for the current study.

Since the predicted propeller performance only varies little between different ice classes the numerical errors are likely to play a role in the results. As shown in Tables 11 and 12, propeller efficiencies are within tenths of a percentage and thus fall within the numerical accuracy of the computations. The fact that all computations and discretizations are identical to each other reduces the effect of numerics when different geometries are compared to each other.

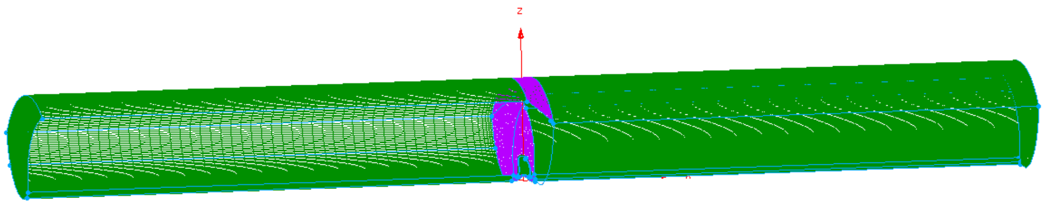


Figure 1. The overall grid set-up with inlet, outlet, outer and periodic boundaries

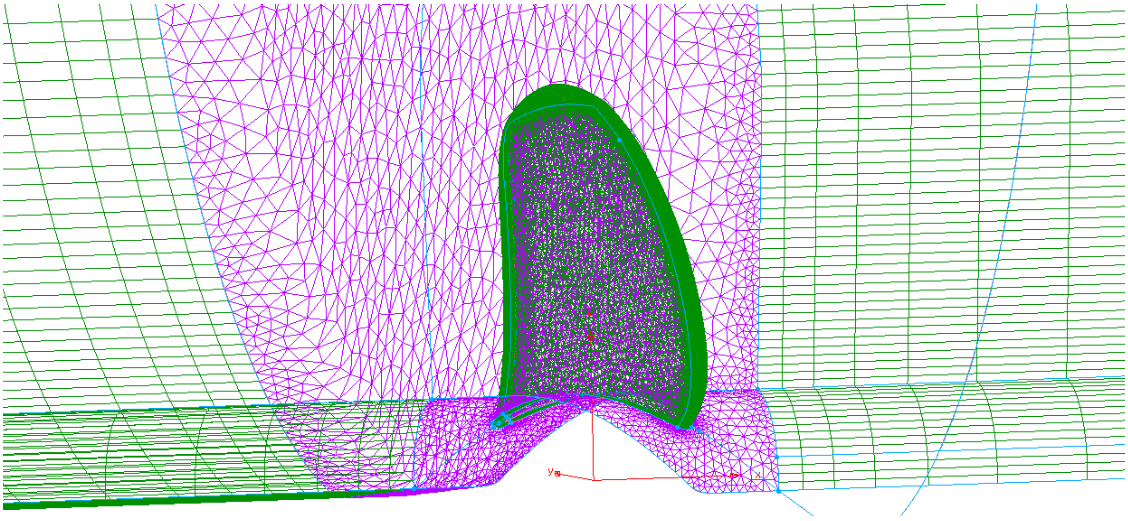


Figure 2. A closer view on the blade surroundings.

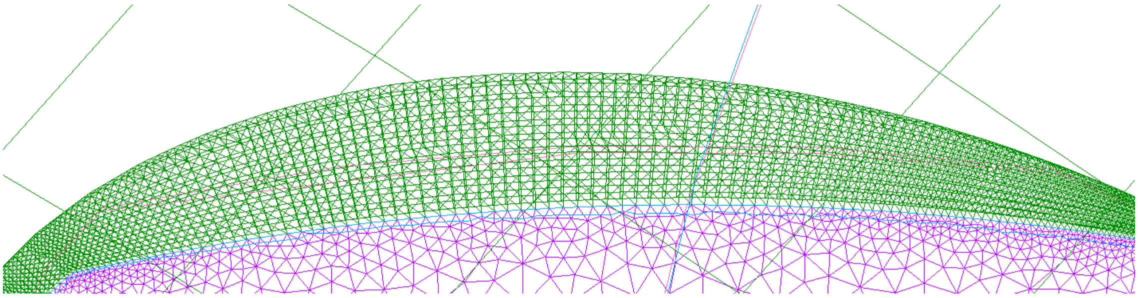


Figure 3. A close view on the blade tip. The tip along with the leading and trailing edges are discretized with quadrangles. The rest of the blades has triangles.

4 STRENGTH ANALYSIS

The propellers for each vessel were evaluated against the 2017 FSICR to assess the strength of the blade. The FSICR assess the strength of a propeller through two methods: minimum ct^2 values for the inner portion of the blade and five load cases for the outer portion of the blade. There are no requirements for the root fillet of the blade. Therefore, the analysis composed of a mathematical assessment of the geometry between 0.25R and 0.5R and a finite element analysis (FEA) from 0.5R to 1.0R, where R is the radius of the blade.

The analysis for the inner portion of the blade consisted of a simple comparison between the blades ct^2 values against the FSICR requirements. The ct^2 values for blade sections at 0.25R, 0.3R, 0.4R and 0.5R were assessed. Generally, the inner blade sections required the most significant changes to meet the FSICR.

The FEA consisted of modelling the original open water propellers and applying the load cases for IC ships to obtain a baseline for each design. The thicknesses of each blade section were varied systematically to determine the limiting blade thicknesses to meet the design criteria. The procedure was repeated for each ice class IC, IB, IA and IAS. All strength analysis was done using Abaqus.

The strength of the outer edge of a blade is analysed through five load cases when dimensioning propellers according to the FSICR. Load Case 1 and 2 analyse the strength of the propeller in reaction to forces applied to the suction side of the blade at the leading edge and the blade tip, respectively. Load Case 3 and 4 assess the strength of the propeller to forces applied to the pressure side at the leading edge and the blade tip respectively. Finally, Load Case 5 applied a load to the trailing edge of the pressure side.

4.1 VESSEL 1

The relevant input properties for the propeller of Vessel 1 are summarised in Table 5. Figure 4 shows the basic propeller model used in the strength analysis.

Table 5. Summary of FSICRs propeller strength calculation input for Vessel 1

Property	Value
Propeller Diameter	5100 mm
Hub Diameter	1122 mm
Propeller Type	Fixed Pitch
Propeller RPM	134.8
EAR	0.650
Number of Blades	4
Material	Bronze
Ultimate Stress of Blade Material	600 MPa
Yield Stress of Blade Material	240 MPa

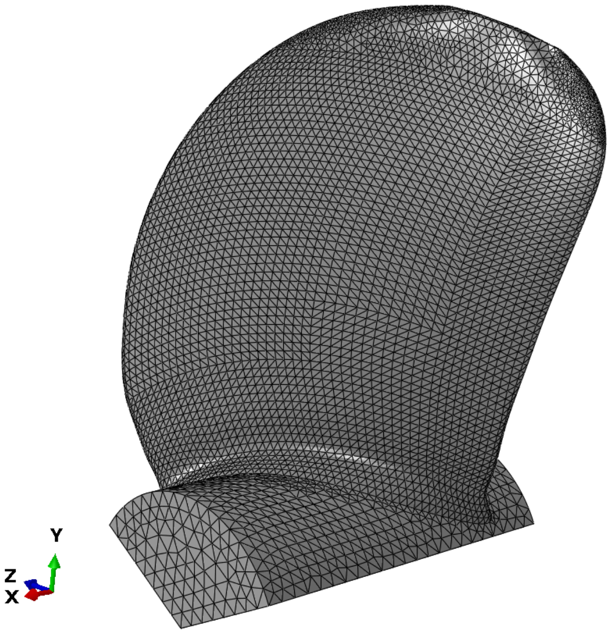


Figure 4. Original propeller model for Vessel 1

Load Case 5 was the limiting case for the propeller of Vessel 1 throughout the entire investigation. Table 6 presents the maximum pressure for each load case as the ice class increased. The maximum pressure is 295.4 MPa for all load cases and all ice classes.

The shape of the pressure distributions for each load case were observed to be constant as the ice class increased. Figure 5 shows the FEM results of each ice class for Load Case 5, the limiting load case. The peak pressure occurs at the trailing edge of the blade near 0.55R.

Table 6. Summary of maximum pressures for each ice class for Vessel 1

Load Case	**Maximum Pressure [MPa]**			
	IC	**IB**	**IA**	**IAS**
1	140.9	140.2	160.7	173.1
2	154.6	191.2	198.7	223.7
3	198.7	183.6	193.9	195.2
4	218.3	250.7	238.0	252.0
5	278.1	287.7	277.4	285.1

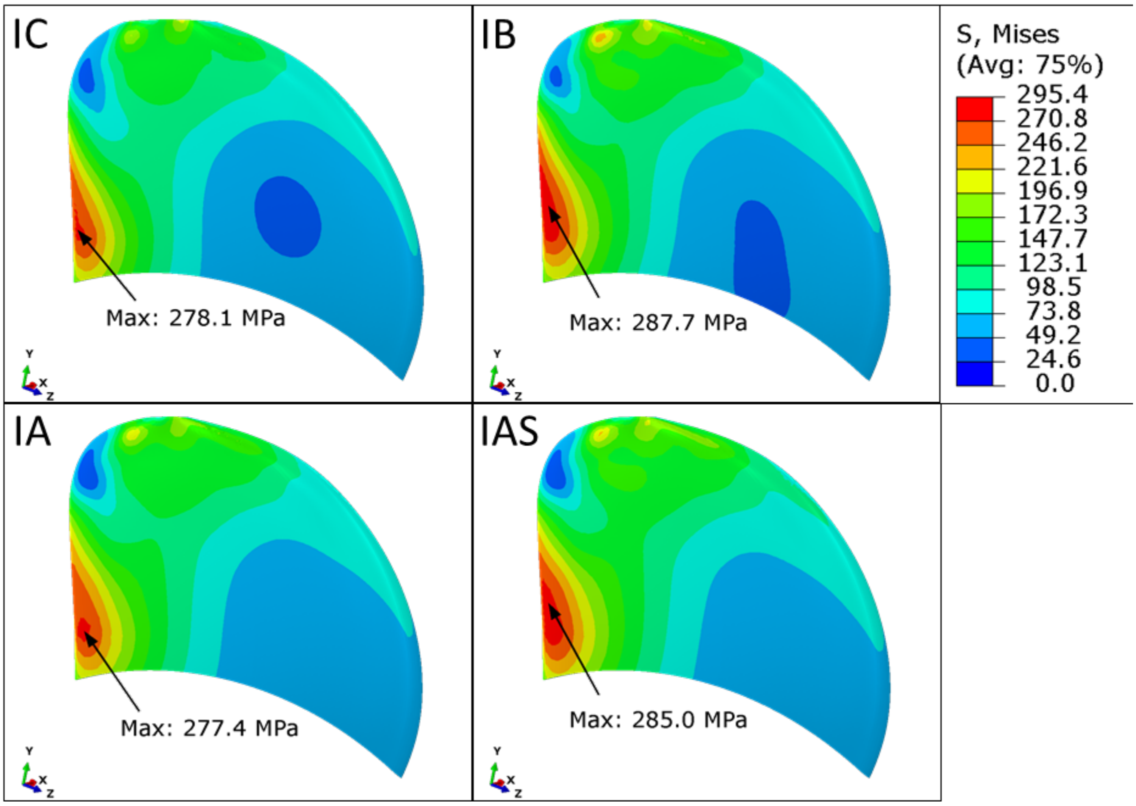


Figure 5. Vessel 1 results for the limiting load case for each ice class

The thickness increases required to be the FSICRs requirements, as percentages of the original blade thickness, can be found in Table 7. Generally, the largest increase in thickness occurred halfway between the root and tip of the blade. The lower portion of the blade required significant increases in thickness to meet the required ct^2. After applying the initial increase in the lower portion of the blade, the thickness addition was decreased incrementally towards the blade tip, Figure 6.

Generally, the IB, IA, and IAS blades followed a consistent trend both as the ice class increased and within each blade itself. The IC blade, however, did not follow the same trend.

Table 7. Minimum required blade thickness increases for Vessel 1

r/R	IC	IB	IA	IAS
0.25	125.0%	140.0%	153.0%	165.0%
0.30	175.0%	191.0%	215.0%	230.0%
0.40	171.0 %	190.0%	215.0%	230.0%
0.50	164.5%	190.0%	205.0%	220.0%
0.60	157.0%	175.0%	195.0%	210.0%
0.70	149.5%	160.0%	185.0%	195.0%
0.80	142.0%	145.0%	170.0%	180.0%
0.90	134.5%	130.0%	155.0%	165.0%
0.95	127.0%	122.5%	147.5%	157.5%
1.00	119.5%	115.0%	140.0%	150.0%

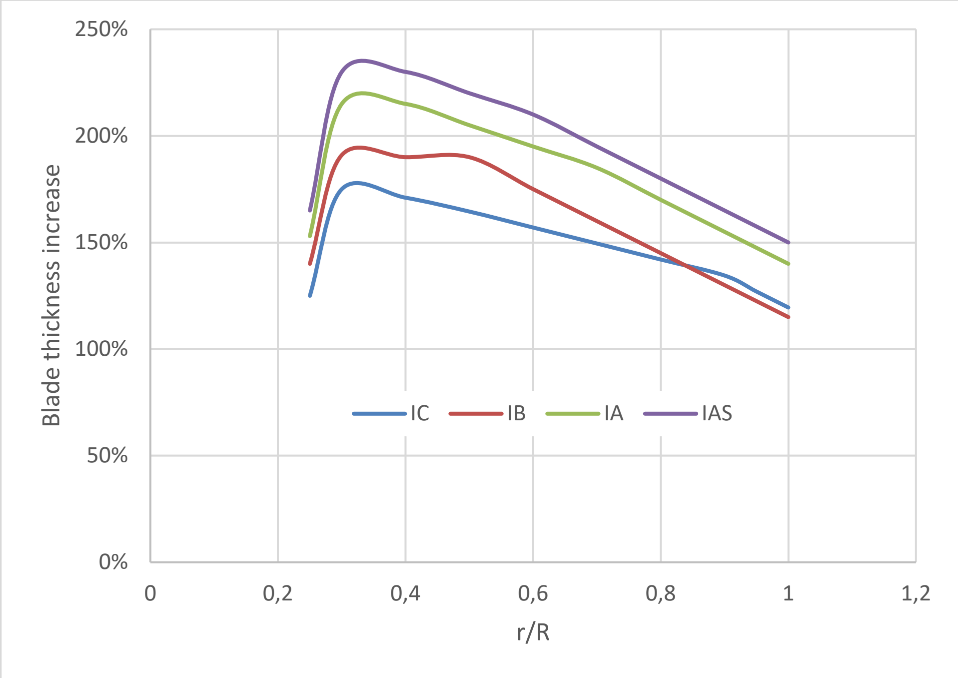


Figure 6. Minimum required blade thickness increases for Vessel 1

4.2 VESSEL 2

The summary of inputs used in the FSICRs calculations for the applied loads and required ct^2 values can be found in Table 8. The basic propeller model can be found in Figure 7.

Table 8. Summary of FSICRs propeller strength calculation input for Vessel 2

Property	Value
Propeller Diameter	4500 mm
Hub Diameter	990 mm
Propeller Type	Fixed Pitch
Propeller RPM	155
EAR	0.550
Number of Blades	4
Material	Bronze
Ultimate Stress of Blade Material	600 MPa
Yield Stress of Blade Material	240 MPa

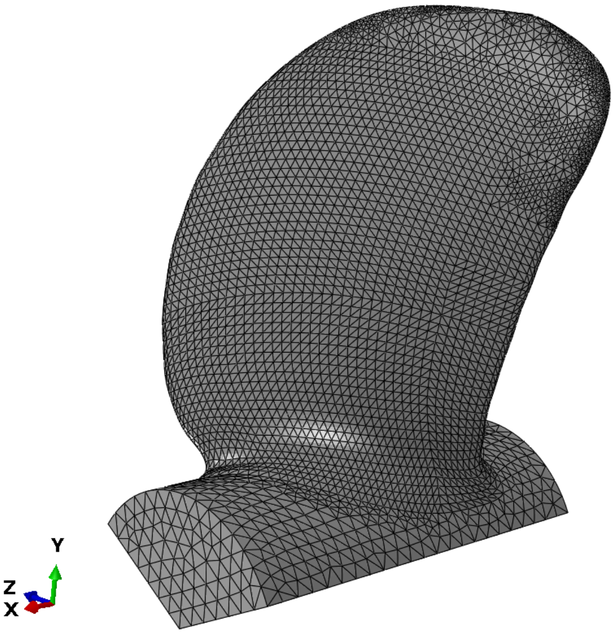


Figure 7. Original propeller model for Vessel 2

Table 9 presents the maximum pressures on each blade for the different load cases and the ice classes. Figure 8 shows the pressure distributions and the location of maximum stress for the limiting load cases for each ice class for Vessel 2. For Vessel 2, Load Case 4 was the limiting case for the strength of the blade for ice classes IA, IB and IC; however, for IAS ice class, Load Case 2 was the limiting case. In both Load Case 2 and 4, the load is applied to the leading edge of the blade on either the pressure side or the suction side, respectively.

Table 9. Summary of maximum pressures for each ice class for Vessel 2

Load Case	**Maximum Pressure [MPa]**			
	IC	**IB**	**IA**	**IAS**
1	138.5	146.1	165.6	173.9
2	237.9	246.3	277.0	288.3
3	171.4	167.7	173.9	172.3
4	291.0	281.3	289.3	282.7
5	287.7	273.2	274.1	264.6

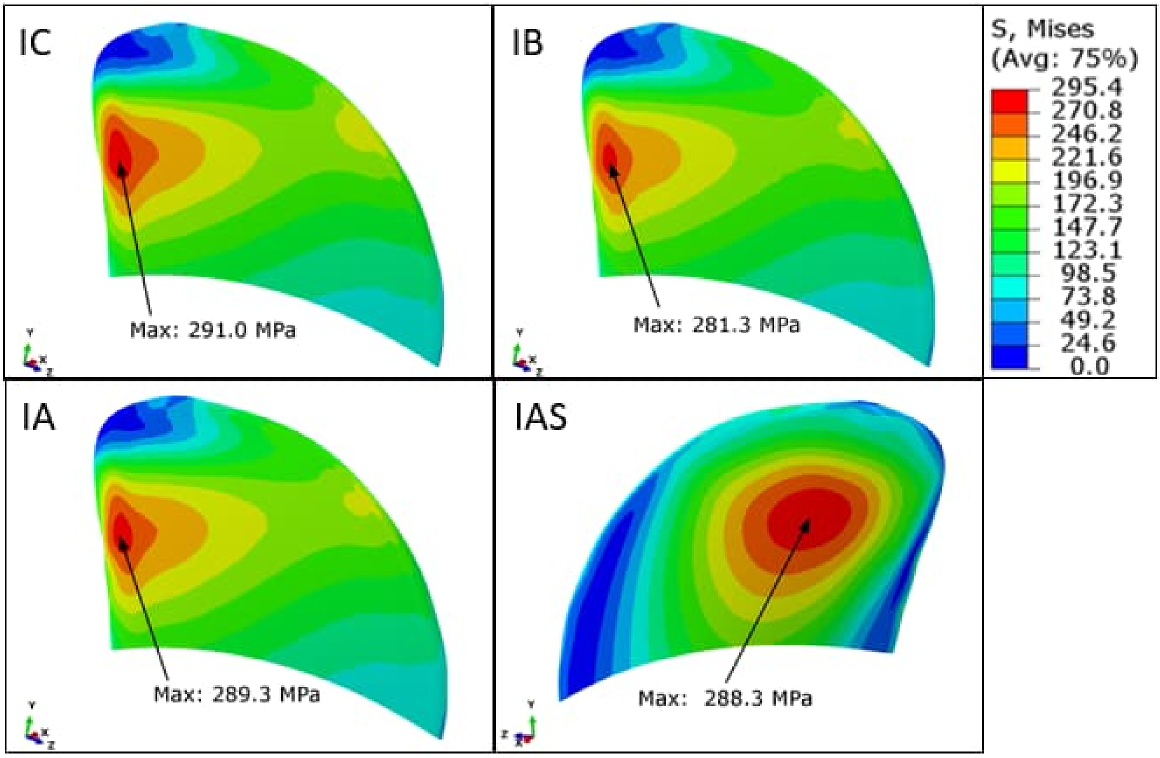


Figure 8. Vessel 2 results for the limiting load case for each ice class

Table 10 presents the required thickness additions for Vessel 2. Unlike the propeller for Vessel 1, the trends between each section of the individual blades and between each consecutive ice class remained consistent throughout all blades. The highest thickness occurred at the 0.3R section, similar to Vessel 1. The required thickness increment was then reduced for each consecutive section towards the blade tip, Figure 9.

Table 10. Minimum required blade thickness increases for Vessel 2

r/R	IC	IB	IA	IAS
0.22	120%	120%	120%	120%
0.25	140%	140%	150%	160%
0.30	210%	220%	240%	260%
0.35	200%	215%	235%	255%
0.40	190%	210%	230%	250%
0.45	185%	205%	225%	245%
0.50	180%	200%	220%	240%
0.55	180%	200%	220%	240%
0.60	175%	195%	215%	235%
0.65	170%	190%	210%	230%
0.70	165%	185%	205%	225%
0.75	160%	180%	200%	220%
0.80	155%	175%	195%	215%
0.85	150%	170%	190%	210%
0.90	145%	165%	185%	205%
0.95	140%	160%	180%	200%
1.00	135%	155%	175%	195%

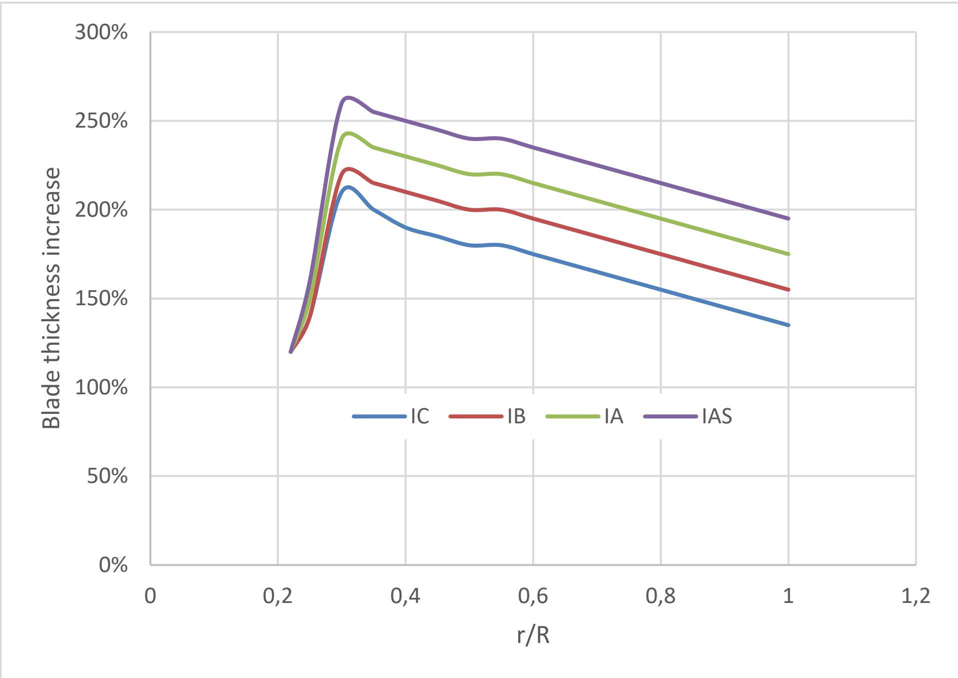


Figure 9. Minimum required blade thickness increases for Vessel 2

5 PERFORMANCE ANALYSIS

The hydrodynamic performance is calculated using a 3D Computational Fluid Dynamics code. The open water curves of each propeller are shown in figures below. The curves cover the extent required for the operational analysis required by the design input i.e. the optimization point and the required maximum speed.

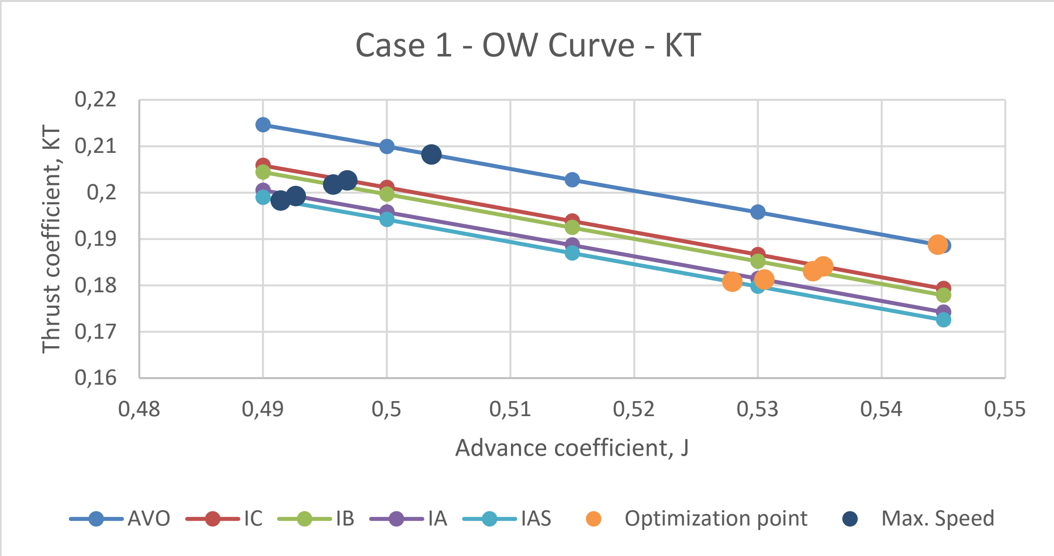
The higher the ice class the lower the advance coefficient corresponding to an optimal point of operation and the maximum operation speed is. This trend is shown in both cases Case 1 and Case 2.

There is a trend of a decreasing efficiency with an increasing ice class. However, differences are small (all within 1.4 %-units), and the trend is even distorted in Case 2 at a maximum speed. This distortion is probably due to numerical accuracy which also implies that no clear statements of absolute efficiencies can be made. The conclusion is that there is a trend of a decreasing efficiency and the differences are limited to about 1-2%-units, however differences of the order of 0.1%-unit cannot be reliably given.

The symbols used in Chapter 5 are as follows:

Abbreviation	Quantity	Unit	Equation	Origin
KT	Thrust coefficient	-	$T / (\rho n^2 D^4)$	calculated
KQ	Torque coefficient	-	$Q / (\rho n^2 D^4)$	calculated
eta	Effeciency	-	TV / P	calculated
J	Advance coefficient	-	V / (nD)	calculated
V	Velocity	m/s		input
N	Rotational speed	1/min		input
n	Rotational speed	1/s		input
T	Thrust	N		output
Q	Torque	Nm		output
P	Power	W	$2\pi nQ$	calculated
P_{OW}	Power of open water design	W		calculated

5.1 VESSEL 1



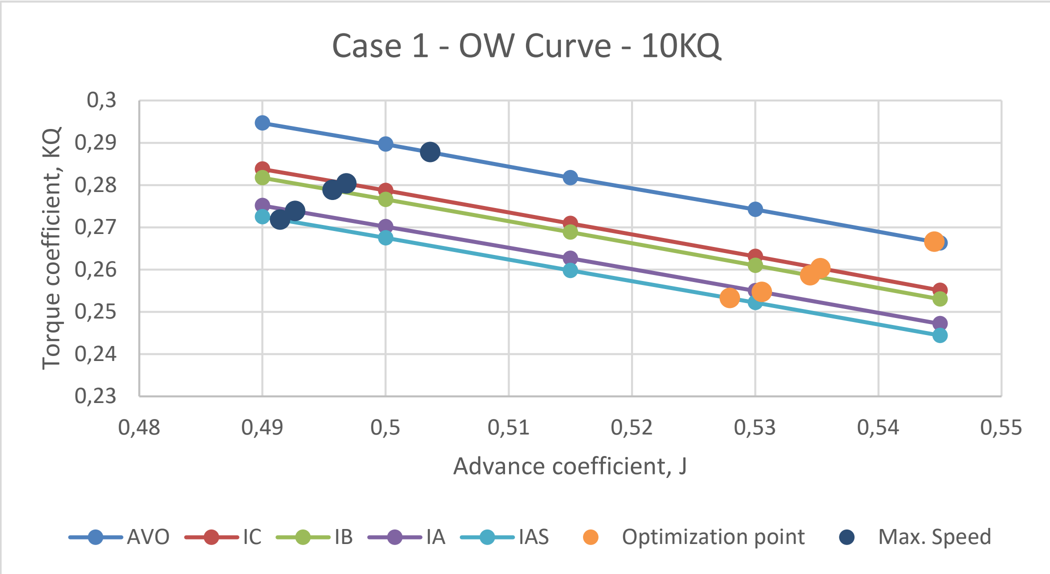


Figure 10. KT and 10KQ open water curves of different ice class propellers for vessel 1.

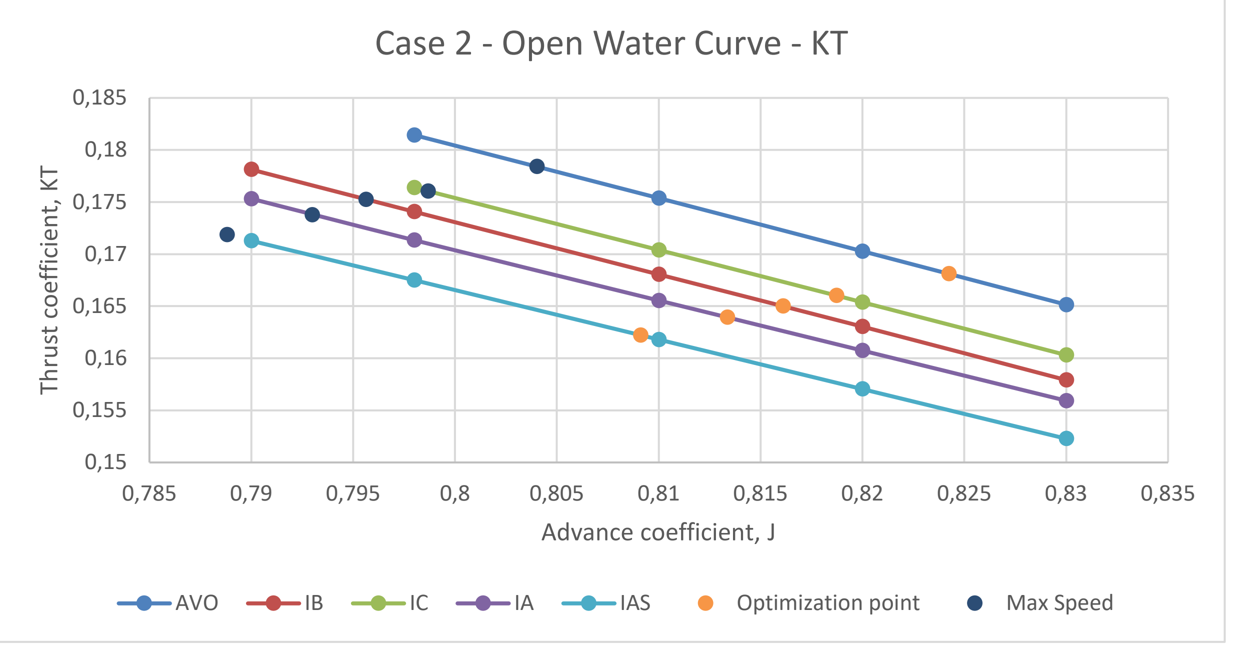
Table 11. Optimization Point – Vessel 1

Ice class	V_{ship} (kn)	N (RPM)	T (kN)	Q (kNm)	P (kW)	Efficiency	P/P_{OW}
Open	14.0	105	401	289.2	3180	61.4 %	100.00 %
IC	14.0	107	405	292.2	3270	60.2 %	102.83 %
IB	14.0	107	404	291.2	3263	60.2 %	102.61 %
IA	14.0	108	406	291.1	3286	60.1 %	103.33 %
IAS	14.0	108	409	292.3	3316	60.0 %	104.28 %

Table 12. Maximum Speed – Vessel 1

Ice class	V_{ship} (kn)	N (RPM)	T (kN)	Q (kNm)	P (kW)	Efficiency	P/P_{OW}
Open	16.5	134	719	506.9	7105	58.0 %	100.00 %
IC	16.5	136	719	507.5	7211	57.1 %	101.49 %
IB	16.5	136	719	507.0	7220	57.1 %	101.62 %
IA	16.5	137	719	504.1	7223	57.1 %	101.66 %
IAS	16.5	137	719	502.8	7223	57.1 %	101.66 %

5.2 VESSEL 2



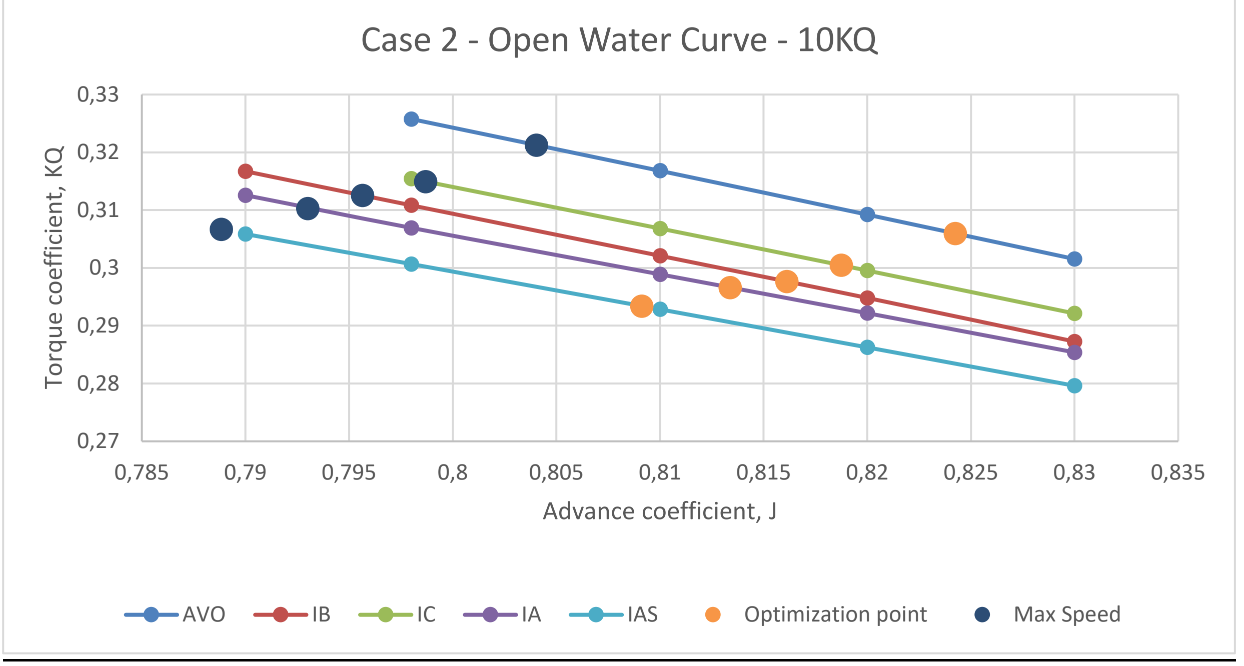


Figure 11. KT and 10KQ open water curves of different ice class propellers for vessel 2.

Table 13. Optimization Point – Vessel 2

Ice Class	V_{ship} (kn)	N (RPM)	T (kN)	Q (kNm)	P (kW)	Efficiency	P/P_{OW}
Open	18	132	341	279.6	3860	72.1 %	100.00 %
IC	18	133	342	277.5	3867	72.0 %	100.18 %
IB	18	133	342	278.3	3867	72.0 %	100.18 %
IA	18	134	342	278.4	3894	71.5 %	100.88 %
IAS	18	134	342	278.2	3913	71.2 %	101.37 %

Table 14. Maximum Speed – Vessel 2

Ice Class	V_{ship} (kn)	N (RPM)	T (kN)	Q (kNm)	P (kW)	Efficiency	P/P_{OW}
Open	20.5	154	494	400.2	6450	71.1 %	100.00 %
IC	20.5	156	495	397.6	6475	71.0 %	100.39 %
IB	20.5	155	494	397.6	6450	71.1 %	100.00 %
IA	20.5	156	494	397.3	6493	70.7 %	100.67 %
IAS	20.5	157	494	396.9	6519	70.4 %	101.07 %

6 RESULTS

Two vessel types and sizes sailing in the Baltic frequently also in wintertime are selected. From the propulsion point of view the vessels are different, the one is typical single screw tanker and the other is twin shaft RoRo or ferry with higher service speed. These vessel types form a representative set of examples of traffic in the Baltic Sea and thus give a solid basis for future estimates of reasonable EEDI criteria.

For both ship-types an open water propeller is designed. The blade thickness of initial propeller design is increased to propellers that fulfils the strength requirements of IC, IB, IA or IAS.

The hydrodynamic design is based on a lifting line code in terms of initial optimization and a 3D Computational Fluid Dynamics code which is used for final optimization and performance calculations.

The propellers for each vessel were evaluated against the 2017 FSICRs to assess the strength of the blade. The FSICRs assess the strength of a propeller through two methods: minimum ct^2 values for the inner portion of the blade and five load cases for the outer portion of the blade.

Finally, the open water curves of the propeller families are evaluated. The performance data calculated allows clear and transparent comparison of performances between different ice classes.

For the vessel 1, the single screw vessel, the propulsive efficiency dropped at optimization point (14 knots) due ice class from 61.4 % of the open water propeller to 60.2 % for IC propeller, to 60.2 % for IB, to 60.1 % for IA and to 60.0 % for IAS propeller. Correspondingly, the relative delivered power increase, due propeller strength demands, compared to open water propeller is 102.8% for IC and IB ice class, 103.3% for IA and 104.3 % for IAS ice class.

For the vessel 2, the twin-screw vessel, the propulsive efficiency dropped at optimization point (16.5 knots) due ice class from 72.1 % of the open water propeller to 72.0 % for IC propeller, to 72.0 % for IB, to 71.5 % for IA and to 71.2% for IAS propeller. Correspondingly, the relative delivered power increase, due propeller strength demands, compared to open water propeller is 100.2 % for IC and IB ice class, 100.9 % for IA and 101.4 % for IAS ice class.

The results between different ice classes are likely to fall within numerical uncertainties which makes an accurate comparison between different ice classes difficult. What can be said is that the difference between the performance of propellers designed according to difference ice classes is small and seems to fit between 1-2 percentage in both vessel types. The open water design is in both cases by about 1-4 percentage more efficient than those with an ice class.

The effect of propulsion efficiency decrease due ice class is studied for reference vessel 1. Table 15 below present the required power to fulfill EEDI phase 3. The power reduction used when calculating EEDI is presented step by step for each ice class and correction factor. The detailed calculation is presented in Appendix 1. The first row shows the required power calculated with FSICR formulas to fulfill each ice class. In the second row this required power is multiplied by the EEDI power correction factor f_j and the third row divided by EEDI correction factor for capacity f_i. The factors are based on current Resolution MEPC.308 (73) (adopted on 26 October 2018) [5]. In the fourth row the power is divided by proposed additional factor 1.05 for ships with ice classes IA and IAS or equivalent. As a result, we get corresponding open water power. We can notice that using these factors the vessel 1 cannot fulfil both FSICR IAS, based on FSICR formula, and EEDI. However, it may be possible to improve the hull form to decrease the required engine power by FSICR or to improve the energy efficiency of the ship in open water in order to meet the required EEDI. In the fifth row, it is presented how the efficiency reduction due ice strength requirements, found in this study, reduces the maximum open water power.

Table 15. The average engine power for tankers of different ice classes

		Maximum Power in open water (to fulfill EEDI phase 3)		FSICR IC		FSICR IB		FSICR IA		FSICR IAS	
		Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]
1	Required Power by FSICR formula				2490		3850		5500		7490
2	Power correction factor f_j; Corrected power	1	4020	0,943	2348	0,895	3444	0,719	3954	0,592	4432
3	Capacity correction facto f_i; Corrected power	1	4020	1,008	2329	1,011	3407	1,016	3890	1,031	4301
4	* Additional ice correction factor for FSICR IA and IAS	1	4020	1,000	2329	1,000	3407	1,050	3705	1,050	4096
5	** Propeller efficiency factor correction factor f_p	1	4020	1,028	2266	1,028	3314	1,033	3586	1,043	3927
	Required EEDI Phase 3		8,6963								

* Proposed additional factor 1.05 (f_m) for ships with ice classes IA and IAS or equivalent (see [6] resolution MEPC.322(74)).

** Additional propulsion efficiency factor from this study (Tankers)

7 CONCUSIONS

The study indicates the lowered propeller efficiency for ice classed ships, especially for slow single screw vessels like bulk carriers and tankers, compared to ships designed for open water only. The increased required power due propeller ice class to achieve the same service speed as for pure open water ships should be considered when calculating the attained EEDI values for ships with an ice class. This could be done with an additional correction factor for the lower propulsion efficiency, it can be included to the EEDI power correction factor f_j or, with regard to ships with ice class IA or IAS, it can be considered to be included in the new ice class correction factor f_m.

The study includes only two ship types in one size and one speed range. For following studies, it can be recommended to include at least general cargo ships and container vessel and to expand the study to different ship sizes and service speeds.

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APPENDIX 1 CALCULATING EEDI FOR TANKERS

The reference ship is supposed to be a tanker

L= 132 m
B= 21.7 m
T= 8.7 m
Displacement = 20310 ton
C_b = 0.795
DWT=14700 ton

The required power according to FSIR formulae

Ice Class	Required power for ice class
	[kW]
IC	2490
IB	3850
IA	5500
IAS	7490

Calculating EEDI for Tankers

In this section, the attained EEDI is calculated for the selected ships types using the same simplified form as in EE-WG 2/2/9:

$$EEDI = \frac{C_{F,ME} \cdot SFC_{ME} \cdot f_j \cdot P_{ME} + C_{F,AE} \cdot SFC_{AE} \cdot P_{AE}}{f_i \cdot capacity \cdot v} \frac{C_F \cdot (SFC_{ME} \cdot f_j \cdot P_{ME} + SFC_{AE} \cdot P_{AE})}{f_i \cdot capacity \cdot v}$$

This parameter depends on the type of engine: for 2-stroke engines, the *SFC* typically ranges from 160 to 170 g/kWh and for 4-stroke engines it ranges from 180 to 190 g/kWh.

The EEDI index value for the reference vessel to fulfil Phase 3 is 8.6963

By knowing the index value, the allowed power can be calculated for the reference ship. The SFC_{ME} will then be 4020 kW. As the specific fuel consumption 190 g/kWh has been used.

The service speed is then 14.9 knots at that power level.

Correction factors for power

The basic form of the present correction factor for power f_j is as follows:

$$f_{j0} = \frac{P_{OW,ave}(dwt)}{actual\ power\ MCR\ 100\ \%}.$$

The minimum value of the ice class correction factor for power, f_j, is limited by the fact that the correction should not result in an engine power greater than the minimum power required by the ice class. This is expressed by the lower limit of $f_{j,min}$, which can be defined as follows:

$$f_{j,min} = \frac{P_{OW,ave}(dwt)}{P_{ice\ class}(dwt)}.$$

The ice class correction factors for power for tankers and ice class are presented in the table below. It should be noted that in the formula for f_{j0}, *MCR* is used as engine power, whereas in the existing 2014 Guidelines P_{ME} (=0.75 *MCR*) was used.

Ship Type	f_{j0}	$f_{j,min}$ Ice-class dependent			
		IAS	IA	IB	IC
Tankers	$\frac{17.444 \cdot dwt^{0.5766}}{\sum_{i=1}^{nME} MCR_{ME(i)}}$	$0.2488 * dwt^{0.0903}$	$0.4541 * dwt^{0.0524}$	$0.7783 * dwt^{0.0145}$	$0.8741 * dwt^{0.0079}$

The power correction factors f_{j0} and $f_{j,min}$ are then:

Ice Class	$f_{j,min}$	f_{j0}
IC	0,943	0,522
IB	0,895	0,522
IA	0,719	0,522
IAS	0,592	0,522

By using the power correction factors for each ice class the achieved power is:

		Maximum Ow Power (to fulfill EEDI)	IC power		IB power		IA power		IAS power	
	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]
Required Power by FSIR formula				2490		3850		5500		7490
Power correction factor Fj ; Corrected power	1	4020	0,943	2348	0,895	3444	0,719	3954	0,592	4432

Decrease of DWT due to ice strengthening

To determine the ice class correction factors for capacity resulting from a decrease in DWT due to hull ice strengthening, the procedure given in section 2.11.2 of the 2014 Guidelines, i.e. the use of a ship-specific voluntary structural enhancement correction factor, f_{iVSE} was applied:

$$f_{iVSE} = \frac{DWT_{reference\ design}}{DWT_{enhanced\ design}},$$

where $DWT_{reference\ design}$ is the deadweight of a ship designed for sailing in open water and $DWT_{enhanced\ design}$ is the deadweight of a ship having an ice class. Both ships were assumed to have the same displacement and the same steel grade as required in the 2014 Guidelines.

In addition of the five ship types for which the ice class correction factors for capacity have been given in the 2014 Guidelines, the additional steel weight as a result of ice strengthening was also calculated for ro-ro cargo ships. The calculations were done for four ship sizes, i.e. all ship types having the ice class IC, IB, IA and IA Super. Other aspects of ice strengthening, like ice strengthening of the propulsion machinery, may also decrease the DWT, but in order to simplify the analysis only additional steel weight in the hull as a result of ice strengthening was calculated.

The table below presents the Ice class correction factor for capacity due to hull ice strengthening.

Ice class	Ice class correction factor for capacity due to hull ice strengthening
Ice class IC	$f_{i(IC)} = 1.0041 + 58.5/DWT$
Ice class IB	$f_{i(IB)} = 1.0067 + 62.7/DWT$
Ice class IA	$f_{i(IA)} = 1.0099 + 95.1/DWT$
Ice class IAS	$f_{i(IAS)} = 1.0151 + 228.7/DWT$

Decrease of DWT due to improved ice-going capability

The ice-going capability of an ice-going ship can be improved by making the bow form slenderer compared to a ship designed for sailing in open water. This will result in a smaller C_b and, consequently, a smaller DWT compared to an open water ship with the same main dimensions.

The following method should be used to determine the effect of the decrease in the block coefficient on the ice class correction factor for capacity:

$$f_{iC_b} = \frac{C_{b\ reference\ design}}{C_{b\ enhanced\ design}},$$

where $C_{b\ reference\ design}$ is the average block coefficient for a given ship type and $C_{b\ enhanced\ design}$ is the actual C_b of a ship with an ice class.

The capacity correction factor, f_i, for ice-classed ships having DWT as the measure of capacity should be calculated as follows:

$$f_i = f_{i(ice\ class)} \cdot f_{iC_b},$$

where $f_{i(ice\ class)}$ is the capacity correction factor for ice strengthening of the ship, which can be obtained from table in previous chapter , and f_{iC_b} is the capacity correction factor for improved ice-going capability, which should not be less than 1.0 and which should be calculated as follows:

$$f_{iC_b} = \frac{C_{b\ reference\ design}}{C_b},$$

where $C_{b\ reference\ design}$ is the average block coefficient for the ship type, which can be obtained for tankers from the table below.

	$C_{b\ reference\ design}$				
Ship type	Small (< 10 000 DWT)	Handysize (10 000 DWT – 25 000 DWT)	Handymax (25 000 DWT – 55 000 DWT)	Panamax (55 000 DWT – 75 000 DWT)	Aframax (75 000 DWT – 120 000 DWT)
Tanker	0.78	0.78	0.80	0.83	0.83

Due to higher C_b than for the C_b reference, f_{iC_b} is 1

The capacity correction factors f_{j0} and $f_{j,min}$ are then:

Ice Class	$f_{i(ice\ class)} \cdot$	f_{iC_b}	$f_i = f_{i(ice\ class)} \cdot f_{iC_b},$
IC	1,016	1	1,016
IB	1,011	1	1,011
IA	1,016	1	1,016
IAS	1,031	1	1,031

Calculating the engine power using the capacity correction factors, the achieved power is:

		Maximum Ow Power (to fulfill EEDI)	IC power		IB power		IA power		IAS power	
	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]
Required Power by FSIR formula				2490		3850		5500		7490
Power correction factor Fj ; Corrected power	1	4020	0,943	2348	0,895	3444	0,719	3954	0,592	4432
Capacity correction facto Fi; Corrected power	1	4020	1,008	2329	1,011	3407	1,016	3890	1,031	4301

Additional correction factor for ice classes

Proposed additional factor 1.05 for ships with ice classes IAS and IAS or equivalent (MEPC 74 Annex 1).

The additional correction factor for IA and IAS gives.

		Maximum Ow Power (to fulfill EEDI)	IC power		IB power		IA power		IAS power	
	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]
Required Power by FSIR formula				2490		3850		5500		7490
Power correction factor Fj ; Corrected power	1	4020	0,943	2348	0,895	3444	0,719	3954	0,592	4432
Capacity correction facto Fi; Corrected power	1	4020	1,008	2329	1,011	3407	1,016	3890	1,031	4301
Additional ice correction factor for FSIR IA and IAS	1	4020	1,000	2329	1,000	3407	1,050	3705	1,050	4096

Propulsion efficiency factors

The proposed propulsion correction factors for each ice class is

Ice Class	$f_{prop(ice\ class)}$.
IC	1,028
IB	1,028
IA	1,033
IAS	1,043

The calculated power levels using the all of the correction factors for each FSIR ice class are:

		Maximum Ow Power (to fulfill EEDI)	IC power		IB power		IA power		IAS power	
	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]	Correction factors	[kW]
Required Power by FSIR formula				2490		3850		5500		7490
Power correction factor Fj ; Corrected power	1	4020	0,943	2348	0,895	3444	0,719	3954	0,592	4432
Capacity correction facto Fi; Corrected power	1	4020	1,008	2329	1,011	3407	1,016	3890	1,031	4301
Additional ice correction factor for FSIR IA and IAS	1	4020	1,000	2329	1,000	3407	1,050	3705	1,050	4096
Propeller efficiency factor correction factor Fprop	1	4020	1,028	2266	1,028	3314	1,033	3586	1,043	3927