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THE DEVELOPMENT OF NEW BALTIC MACHINERY ICE CLASS RULES

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ABSTRACT

The purpose of this paper is to publicly present for the first time the draft of the new Finnish-Swedish machinery ice rules. The basic philosophy behind the ice class rules and the underlying ideas of the work to revamp the machinery ice rules are described in this paper. The joint Finnish-Canadian research project "JRPA#6" in the field of propeller-ice interaction that forms the technical background of the renewal work is presented briefly. The structure of the requirements as well as the main points in the draft of the new regulations are described. The results of some comparisons between the new proposal and the present regulations are shown.

SYMBOLS

c	m	chord length of blade section
C_{class}		ice class factor
D	m	propeller diameter
d	m	propeller hub diameter
D_{limit}	m	limit value for propeller diameter
EAR		expanded blade area ratio
F_b	kN	maximum backward blade force
F_{ex}	kN	ultimate blade load due to blade loss by plastic bending
F_f	kN	maximum forward blade force
I	kgm^2	equivalent mass moment of inertia for all the parts on the engine side of the component under consideration
I_t	kgm^2	equivalent mass moment of inertia of the propulsion system
n	1/s	propeller rotational speed
N_{class}		reference number of ice loads per propeller rotational speed per ice class
N_{ice}		total number of ice loads on propeller
$P_{0.7}$	m	pitch at 0.7R radius
Q	kNm	torque
Q_{emax}	kNm	maximum engine torque
Q_{max}	kNm	maximum ice torque of the propeller
Q_r	kNm	maximum response torque of shaft line component
R	m	propeller radius
r	m	blade section radius

t	m	maximum blade section thickness
$t_{0.7}$	m	maximum blade section thickness at 0.7R propeller radius
Z		number of propeller blades
$\sigma_{0.2}$	MPa	proof strength of blade material
σ_{exp}	MPa	experimentally determined mean fatigue strength of blade material at 10^8 cycles to failure in circulation sea water expressed in stress amplitude
σ_{fat}	MPa	equivalent fatigue ice load stress amplitude for $1x10^8$ stress cycles
σ_{fl}	MPa	characteristic fatigue strength for blade material
σ_{ref}	MPa	reference stress
σ_u	MPa	ultimate tensile strength of blade material
$(\sigma_{ice})_{bmax}$	MPa	stress due to maximum backward ice load
$(\sigma_{ice})_{fmax}$	MPa	stress due to maximum forward ice load

1. INTRODUCTION

1.1 The Finnish - Swedish Ice Class Philosophy

Sweden and Finland are highly industrialised countries, dependent on scheduled, year-round fast and reliable sea transport. The ice class regulations form a part of the year-round transportation strategy.

Most of our ports (and all Finnish ones) are icebound for some part of each year. For much of the year, ships sail through open water, and even in winter a part of the voyage takes place in open water. Thus, ships should first and foremost be efficient open water ships. Icebreaker assistance should then be provided when the situation so warrants. Merchant ships should be strengthened and otherwise modified for ice navigation only to the degree necessary.

Some adjustment of ships is necessary, however. The hull, propeller, shafts and gear have to withstand some ice load even when the ship is assisted by an icebreaker or is following a broken lead. Sufficient engine power is also required, so that ships only need to be towed in very difficult situations. With the amount of traffic that we have today, the demand for icebreakers would otherwise be excessive.

Depending on the ice situation, restrictions are imposed on ships receiving icebreaker assistance, based on their ice class and dead-weight. Ice class IA will entitle a ship of sufficient dead-weight (4000 tdw for the northernmost ports) to assistance through the winter. The lower ice classes may prolong the season at the beginning and end of the winter and may be sufficient for the southern ports. Ships with ice class IA Super will be treated as IA ships with regards to assistance, but they will benefit from lower fairway fees under the Finnish system.

1.2 The Development of the Finnish-Swedish Machinery Ice Rules

The present Finnish-Swedish machinery rules are based on the ice torque principle that dates back to the 1960s. The shaft torque that is caused by ice has been measured in full scale tests and converted into an ice load that acts on the blade, using certain assumptions and simplifications. This philosophy has worked fairly well so far. New applications, nozzles, large propellers and highly skewed propellers have resulted in areas where extrapolation is difficult. The rules have not been changed since the 1971 version. Most of the existing propeller ice regulations are also based on the ice torque principle and many of them in fact repeat more or less directly the Finnish-Swedish ice class rules as such. The process of renewing the existing Baltic regulations started already in the 1980´s. However, the proposal for the renewed rules was never published since the Canadian Coast Guard and the Finnish Board of Navigation established a joint research project, "JRPA #6 (Joint Research Project Arrangement #6), Propeller Ice Interaction" in 1991, to develop a new propeller ice interaction model.

An international harmonisation process for polar navigation rules started while the joint Canadian-Finnish project was going on. This process originally covered only the structural rules, but the machinery rules were included in the process in 1996. Since then a great deal of effort has been expended within the IACS machinery working group on formulating a joint approach for the new IACS polar machinery rules. The international harmonisation process and the development of the Finnish-Swedish ice class rules for machinery have been going on in parallel. The objective has been to base both rules on the same technical background. At the moment it seems that this objective has been reached to a reasonable degree.

2. THE JOINT FINNISH CANADIAN PROPELLER ICE CONTACT RESEARCH PROJECT “JRPA # 6”

"JRPA #6” had as its aim the development of a new propeller ice interaction model (Jones et. al 1997). The final goal was to use the model in formulating new machinery regulations for Arctic and Baltic ice conditions. The new model was to be based on a theoretical analysis and experimental modelling of the physical phenomena involved, and of existing full-scale measurements.

The propeller-ice interaction loads consist of both actual contact loads due to propeller penetration into blocks of ice and non-contact loads, i.e. hydrodynamic disturbance loads generated by the presence of ice blocks in the vicinity of propellers. The dynamics between a block and a propeller affect the build up of the load. The ice block size, location and orientation in relation to the blade influence the dynamics. The exposure of the blade is affected by the operative conditions of the vessel and the design of the after body in relation to the propeller. The relative magnitude between the contact and non-contact loads is affected by the propulsion concept, i.e. an open propeller vs. a ducted design. The approach in the JRPA#6 project was to develop a load model for the direct contact and non-contact loads. The main responsibility for development of a contact load model was in Finland, whereas the main responsibility for development of a model for hydrodynamic loads was in Canada. This division reflected the areas of expertise that had been developed in both countries. The contact model was based on observations and measurements from a laboratory test series where the pressure distribution along a blade like tool was measured during its impact into an ice field

(Soininen et. al 1995 and Soininen 1998). The non-contact model was based on propeller model tests in an ice basin and on cavitation tests in blocked conditions (Newbury et. al 1993, 1994 and Newbury 1996). The resulting contact and non-contact models were combined in a simulation process where the propeller geometry, ice block geometry, pitch angle, ship speed and propeller speed give the contact conditions at any moment. The contact geometry, in turn, affects the magnitude and distribution of both contact and non-contact loads. The used non-contact load contributed up to tens of percents of the total load in some contact conditions. Its relative importance decreases in extreme cases, in contacts with massive ice blocks. The load is calculated for each section and integration of these loads along the blade radius gives the effective load for the blade. The force balance changes the velocity pattern of the ice block and accordingly the contact geometry (Koskinen et. al 1996). The developed simulation programme was applied to an extensive parameter study. The results of some of the parameter combinations where verified with existing full-scale measurements. Finally, the load dependencies of the parameters were expressed with a simplified regression type set of equations that form the basis for development of the load formulae for the ice class rules.

3. THE BASIC IDEAS OF THE NEW PROPOSAL

The main principle has been to base the new proposal on the estimated ice loads on the propeller. Formulae to calculate the scantlings such as blade thickness and shaft diameters are not given. The strength of the propulsion line shall be designed according to the pyramid strength principle. Thus, the component that can be easily replaced and when damaged does not risk a ship's safety should be damaged first, preventing failures in parts that are more essential for ship safety. Therefore the requirements demand that in case bending leads to a break in the propeller blade, a yielding of the propeller shaft, damages in the thrust or shaft bearing or damages in the CP-mechanism should not be caused.

The regulations assume that good ice navigation seamanship will be followed in operating the ship and its propulsion machinery. Damages may be expected if a stopped or slowly rotating propeller or a CP-propeller at zero pitch is dragged through ice, e.g. if the ship is towed by an icebreaker. The regulations must not be assumed to cover such situations.

The out-of-plane force is the principal load component of the propeller blade. The blade bending moment, spindle torque and propeller ice thrust can be calculated on the basis of the out-of-plane force (Koskinen et. al 1996). The most dominant parameters that affect the blade loads are ice block size and angle of attack.

The scenario for determining the blade ice load is that the blade has to withstand the loads due to an extreme ice block (corresponding to the thickness of the consolidate layer of a ridge) when the propeller is operating under normal operating conditions. For this case, the estimated angle of attack is 4 degrees. The blade load equations that were originally developed within JRPA #6 (Koskinen et. al 1996) were simplified for the rules so that some parameter influences are included in a class coefficient in the formulae. Blade stresses are determined on the basis of the realistic location of the ice pressure area on the blade.

In addition, a fatigue design methodology is applied. The load distributions for the ships' service life are estimated as a first step for fatigue design of the propeller blades. On the basis of the load distribution and estimated S-N curve of the blade materials, a method was developed to calculate the equivalent fatigue stress that causes the same fatigue failure to the blade as the load distribution.

4. MAIN CONTENTS OF THE NEW PROPOSAL

A general remark is that some refinement work on the new proposal will still go on before it is issued. In other words, the contents of this chapter may experience some changes in the final version of the rules. It has to be noted that in the load equations each parameter has to be given in the correct units according to the symbol list in order to obtain the correct loads. Furthermore, it has not been possible to present in the limited space available more than just the blade force and shaft torque formulae. These formulae are simplified from the formulae presented in the reference (Koskinen et. al 1996). Different formulae are given for small and large propellers. The reason for this is that during the propeller milling of an ice block small propellers are fully immersed in the ice, but for large propellers contact length at the leading edge of the blade is limited by ice block size.

4.1 Propeller/Ice Interaction Loads

4.1.1 Extreme Blade Out-of-Plane Force

The principal load that is considered is the blade out-of plane force as expressed below.

$$F_b = K \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^2 \qquad (1)$$

where coefficient $K = 33.4$ for open propellers and $K = 9.5$ for ducted propellers. The term nD represents the tangential velocity of the profile. Together with the assumption of the angle of attack it describes the inertial effects of the contact. n is the nominal rotational speed for a CP propeller and 85% of nominal rotational speed for a FP propeller. The term *EAR/Z* together with an assumption of ice strength represents the load intensity, *EAR/Z* giving the nondimensional blade area. D represents the size factor to the non-dimensional contact length and height. Equation 1 is valid in the case of open propellers for diameters smaller than D_{limit}. For diameters larger than D_{limit}, the blade force Equation changes to:

$$F_b = 29.2 \cdot C_{class}{}^{1.4} \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D \qquad (2)$$

The ice class factors that are used in the load formulae are:

Ice class	Ice class factor C_{class}
1Asuper	1.75
1A	1.5
1B	1.2
1C	1.0

D_{limit} is defined as:

$$D_{limit} = 0.873 \cdot C_{class}{}^{1.4} \cdot 1[m] \qquad (3)$$

Figure 1 shows the maximum backward blade force for open propellers for different ice classes.

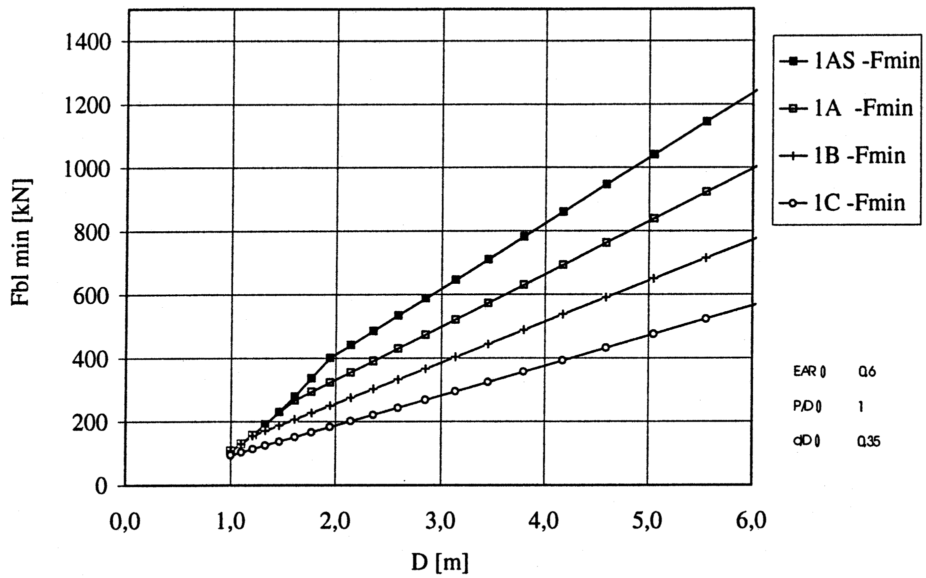

Figure 1. Maximum backward blade load for open propellers (EAR=0.6, d/D=0.36)

For nozzled propellers, Equation 1 is valid for propellers smaller than $0.25 \cdot D \leq C_{class} \cdot 1[m]$ and for bigger propellers the maximum backward bending blade load is:

$$F_b = 66.5 \cdot C_{class}{}^{1.4} \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^{0.6} \tag{4}$$

The available full scale results have been applied to define the maximum forward bending blade force for open propellers:

When $D<2/(1-d/D)C_{class} \cdot 1[m]$

$$F_f = 314 \cdot \left[\frac{EAR}{Z}\right] \cdot D^2 \tag{5}$$

When $D>2/(1-d/D)C_{class} \cdot 1[m]$

$$F_f = 628 \cdot C_{class} \cdot \left[\frac{1}{(1-d/D)}\right] \cdot \left[\frac{EAR}{Z}\right] \cdot D \tag{6}$$

Figure 2 shows the maximum forward blade load for open propellers of different ice classes.

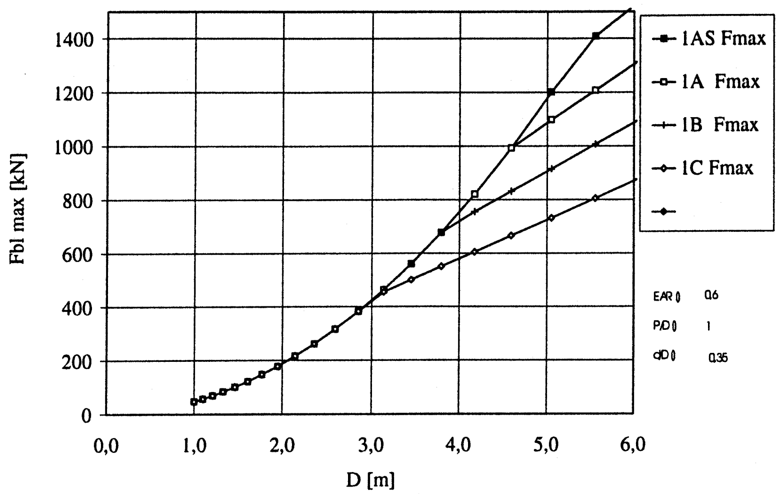

Figure 2. Maximum forward blade load for open propellers (EAR=0.6, d/D=0.36)

The maximum forward blade load for nozzle propellers is obtained with the open propeller formulae, but with a 10% reduction in the loads.

4.1.2 Extreme Propeller Ice Torque

After assumptions for ice strength, the torque for an open propeller is obtained with the formula below. The terms (1-d/D), P/D and t/D are related to the loaded area on the blade. Term D^3 represents the scale and moment arm effects. The term *nD* represents the tangential velocity of the profile describing the inertial effects of the contact.

$$Q_{\max} = 105 \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot \left[\frac{t_{0.7}}{D}\right]^{0.6} \cdot (nD)^{0.17} \cdot D^3 \tag{7}$$

when $D < D_{limit}$, and:

$$Q_{\max} = 202 \cdot C_{class}^{\ 1.1} \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot \left[\frac{t_{0.7}}{D}\right]^{0.6} \cdot (nD)^{0.17} \cdot D^{1.9} \tag{8}$$

when $D > D_{limit}$,

Here:

$$D_{limit} = 1.82 \cdot C_{class} \cdot 1[m] \tag{9}$$

Due to nozzle protection, a propeller operating in a nozzle experiences lower maximum ice torque than an open propeller. On the base of simulation, the ice loads on propellers due to ice blocks that fit in the nozzle and on the basis of full scale experience it is reasonable to estimate that the maximum ice torque for nozzle propellers is 70% of the ice torque for open propellers.

4.1.3 Extreme Torque of Propulsion Shaft Line Components

If there is a blade order torsional resonance within the designed operating rotational speed range extended 10 percent above and below the maximum and minimum operating rotational speeds, the extreme torque (Q_r) of the studied shaft component shall be determined by means of dynamic analysis of the propulsion line.

If the resonant speed is outside the given range, the following estimation of maximum torque can be used.

$$Q_r = Q_{e\max} + Q_{\max} \cdot \frac{I}{I_t} \qquad (10)$$

where all the torques and the inertia moments shall be reduced to the rotation speed of the examined component.

The propeller ice torque excitation for shaft line dynamic analysis is to be described with a sequence of blade impacts which are of half sine shape and occur at the blade frequency or at twice the blade frequency, see Figure 3. The length of the milling sequence in propeller revolutions shall be obtained with the formula:

$$N_Q = 2 \cdot C_{class} \qquad (11)$$

The number of impacts is $Z \cdot N_Q$ for blade order excitation and $2 \cdot Z \cdot N_Q$ for twice the blade order excitation.

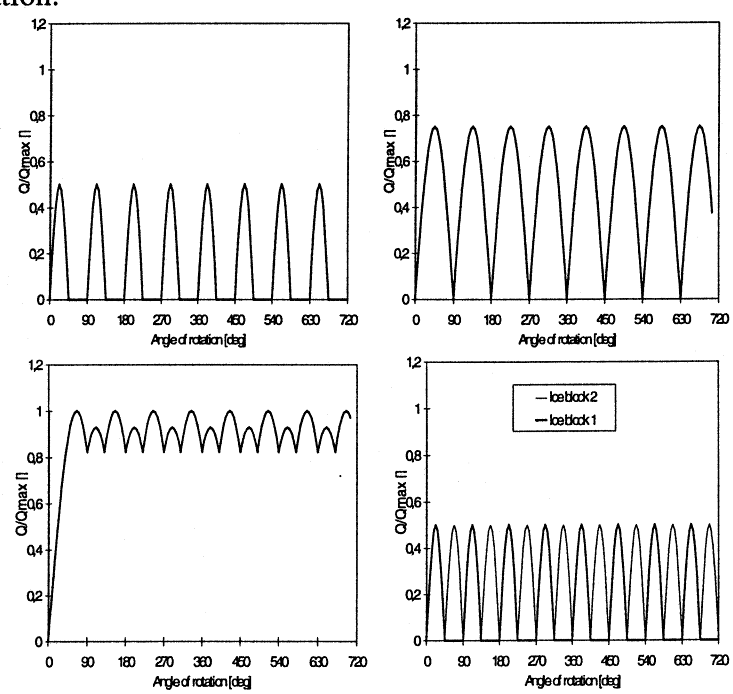

Figure 3. The shape of the propeller ice torque excitation for 45, 90, 130 degree single blade impact sequences and a 45 degree double blade impact sequence(two ice pieces).

4.2 Component Design Based on Loads

4.2.1 Extreme Strength of Propeller Blade

The blade stresses have to be calculated for the extreme loads given in section 4.1. A FE-analysis has to be performed. Both open and nozzled propellers are studied for leading edge loading. In addition, open propellers are studied for tip loading. For the leading edge load case, F_b and F_f are to be applied to the area of the blade radially from 0.6R to the tip and in the direction of the chord from the blade leading edge to 0.1 chord length, see Figure 4. For the tip loading case of an open propeller the load magnitude is 50% of F_b, however the magnitude is still subject to verification at the moment. The tip load case has been included mainly to cover the loads for skewed propellers. The load should be applied on the area which is outside the 0.9 radius, see Figure 5.

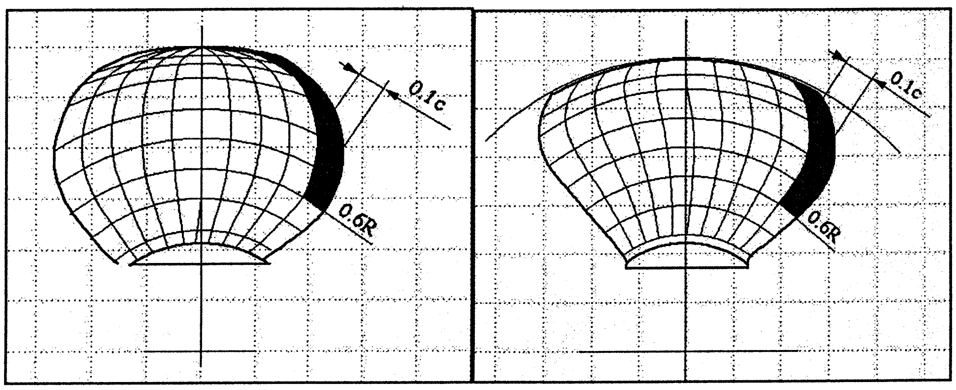

Figure 4. Loaded areas of the blade surface, leading edge contact. a) open propeller, b) nozzled propeller.

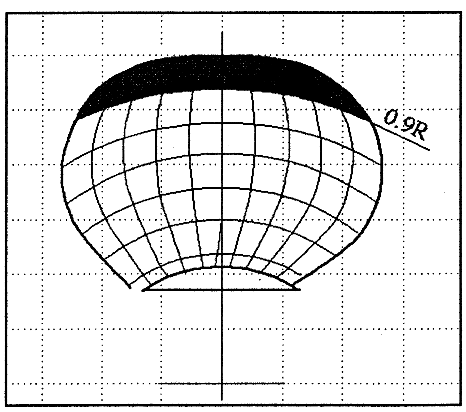

Figure 5. Loaded area of the blade surface in case of a tip contact

4.2.2 Fatigue Design of Propeller Blade

The fatigue design of the propeller blades is based on an estimation of the load distribution for the ship's service life and the S-N curve of the blade material. A Weibull distribution for the blade stresses due to ice loads has been applied. The ice class rules will include formulae to estimate the number of load cycles as well as formulae for the equivalent fatigue stress that gives the same fatigue damage as the stress distribution.

The number of load cycles depends on the ice classes and ship parameters according to the formula:

$$N_{ice} = k_1 k_2 k_3 k_4 N_{class} n \tag{12}$$

N_{class} is the class dependent reference number of impacts per propeller rotation speed and factors k_1, k_2, k_3 and k_4 take into account the effect of propeller location (centre or wing), propeller type (open, nozzled, fixed, azimuthing) and the submersion of the propeller on the exposure of the propeller to ice contacts.

The equivalent fatigue stress for 100 million stress cycles corresponding to the given stress distribution and S-N curves is:

$$\sigma_{fat} = \rho \cdot \left(\sigma_{ice}\right)_{max} \tag{13}$$

where

$$\left(\sigma_{ice}\right)_{\max} = 0.5 \cdot \left(\left(\sigma_{ice}\right)_{f\max} + \left(\sigma_{ice}\right)_{b\max}\right)$$

is the mean value of the normal stress amplitudes due to extreme forward and backward blade forces at the studied location. ρ is a parameter relating the maximum ice load to the distribution of ice loads according to the formulae

$$\rho = C_1 \cdot \left(\sigma_{ice}\right)_{\max}^{C_2} \cdot \sigma_{fl}^{C_3} \cdot \log(N_{ice})^{C_4} \tag{14}$$

where

$$\sigma_{fl} = \cdot \gamma_{\varepsilon} \cdot \gamma_{v} \cdot \gamma_{m} \cdot \sigma_{\exp} \tag{15}$$

$\sigma_{\exp}$ is the experimentally determined mean strength at $1 \text{x} 10^8$ cycles to failure in circulation sea water expressed in stress amplitude. Reduction factors γ_{ε}, γ_{v}, and γ_{m} take into account the effect of scatter and test specimen size, the effect of variable amplitude loading as well as the effect of mean stress on the fatigue strength of the blade material. Coefficients C_1, C_2, C_3 and C_4 are regression parameters.

4.2.3 Shafts and Shafting Components

The shafts and shafting components, such as thrust and stern tube bearings, couplings, flanges and sealings, shall be designed to withstand the propeller-ice interaction loads as well as the extreme load due to loss of the blade by plastic bending. The loading shall consist of combined bending and torsion wherever it is significant.

The blade breaking load acts on the blade at the 0.8R radius in the weakest direction of the blade.

$$F_{ex} = \frac{0.3 \cdot c \cdot t^2 \cdot \sigma_{ref}}{0.8 \cdot D - 2 \cdot r} \tag{16}$$

where

$$\sigma_{ref} = 0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u \tag{17}$$

c, t and *r* are respectively the length, thickness and radius of the cylindrical root section of the blade

4.2.4 Reduction Gears

Reduction gears are to be designed with regards to surface durability, tooth strength and scuffing durability for maximum torque (Q_r) as determined according to Section 4.1.3 and equivalent oscillating loads due to ice as expressed in torque.

$$Q_N = 0.5 \cdot Q_{e\max} + 0.5 \cdot Q_r \tag{18}$$

All the torques and the inertia moments shall be reduced to the rotation speed of the examined component. Q_N is the equivalent torque at reduction gear at a number of load cycles calculated as $N_{ice}/100$

4.2.5 Azimuth Thrusters

In addition to the above requirements, the steering mechanism, the fitting of the unit and the body of the thruster shall be designed to withstand the loss of a blade without damage. The loss of a blade shall be considered in the propeller blade orientation that causes maximum loading on the studied component. Azimuth thrusters shall also be designed for estimated loads due to thruster body/ice interaction. However, these loads are not given in the machinery ice class rules.

4.3 Alternative Design

As an alternative to the above design requirements, a comprehensive design study may be submitted. It has to include both fatigue and extreme load calculations and fulfil the pyramid strength principle of the shafting components. The loads of the propeller blade and propulsion system should be based on an acceptable estimation of hydrodynamic and ice forces. Vibration analysis is to be carried out and is to indicate that the complete dynamic system is free from harmful resonances due to propeller-ice interaction.

5. COMPARISON OF THE PRESENT RULES AND THE NEW RULE PROPOSAL

In the present rules the factors affecting the blade scantlings are: ice class, propeller diameter, shaft nominal torque (power/revolutions), pitch diameter ratio and ultimate strength of the blade material. The rules base on the assumption that the blade stresses are obtained with beam theory. In the new proposal the propeller scantlings are not dependent on the pitch diameter ratio or on the shaft nominal torque. The effects of ice class and propeller diameter are included as well as the effect of expanded blade area ratio. The real propeller geometry is included in calculation of the blade stresses.

A set of existing propellers were calculated in order to compare the present rules and the new proposal, see Table 1. Calculations were conducted for a few propellers that have experienced damages and for some others that have not. The comparison has to be taken as preliminary because FE-calculations of blade stresses were not carried out. Thus, blade stresses were estimated by multiplying the stresses obtained with beam theory equations by a factor of 1.6. This factor has been estimated on the base of FE-analysis of a few blades. However, depending on the blade shape the factor may vary aproximately ± 20 %. The loads were obtained with the rule formulae and the load was located at 0.7R radius. Extreme and fatigue criteria values presented in the Table 1 describe the load carrying capacity of the existing blade with respect to new requirement. If the value is below 1.0 the strength of the propeller should be increased according to the new rule proposal. Extreme loads dominate the blade scantlings for bronze propellers. For steel propellers the dominating design criteria is either extreme or fatigue criteria depending on the material strength values.

Table 1. Comparison of the propeller blades for existing ships

Propeller type	Propeller material	Ice class	D [m]	EAR	Extreme strength criteria	Fatigue strength criteria	Notes
Open CP	NiAlBr	1A	2	0.69	0.93	1.60	Minor plastic deformations
Open FP	NiAlBr	1C	3	0.49	0.81	2.75	Blade failures
Open CP	NiAlBr	1A	3.8	0.53	0.66	1.11	Blade failures
Open CP	Steel	1A	4.2	0.58	0.98	NA*	
Open CP	Steel	1As	5.45	0.52	1.15	1.17	
Open CP	Steel	1As	5.6	0.60	1.02	2.16	
Nozzle FP	NiAlBr	1B	2.4	0.55	0.92	1.18	
Nozzle CP	NiAlBr	1A	3.2	0.55	0.99	1.25	
Open CP	NiAlBr	1A	3.35	0.85	0.81	1.23	Highly Skewed
Open CP	NiAlBr	1A	4.1	0.72	0.76	1.30	Highly Skewed
Open CP	NiAlBr	1A	4.6	0.74	0.52	1.02	Blade failures; Highly Skewed

* fatigue strength of the blade material is not available

6. CONCLUSIONS

The proposal for a new rule is based on extensive work, where the physical phenomena involved in the propeller-ice interaction are interpreted in the text of the rule. It can be said that JRPA #6 and other works that form the basis of the new rules have brought the understanding of the contact process to a new level. A whole new range of phenomena are now reflected in the rule proposal compared to the approach of the old rules. In spite of this the differences of the required blade dimensions do not differ a lot from the existing requirements in the normal range of propeller sizes and power.

Some increased knowledge of the nature of the forward bending loads, as well as nozzled propeller loads, would be needed to be able to handle the whole problem physically more accurately in the future. More work is also needed in the case of azimuthing and highly skewed propellers as well as to estimate the effect of different parameters on the exposure of the propeller to ice contacts. When this knowledge is available the new rules should be modified in these respects accordingly.

7. ACKNOWLEDGEMENTS

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