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Excitation Response Estimation of Polar Class Vessel Propulsion Shafting System

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ABSTRACT

The prospect of Arctic trade transportation opening on a year-round basis creates a vast opportunity of exploring untapped resources and shortened navigational routes. However, the environment's remoteness and lack of technical experiences remains a big challenge for the maritime industry. With this, engine designers and makers are continually investigating, specifically optimizing propulsion shafting system design, to meet the environmental and technical challenges of the region.

Further, classification societies recognize the need to upgrade the Unified Rules concerning elements to meet current Polar requirements. Hence in this paper, excitation torque calculation on Polar class vessels propulsion shafting system will be reviewed. The propeller – ice interaction load effect, which is a main consideration of excitation source of Polar Class propulsion shafting system, on shaft design calculation will be analyzed.

1. Introduction

The prospects of energy and mineral resources availability in the Arctic area have continuously dictated a need for vessels with ice navigation capability. Further, the opening of the Northwest Passage, which ironically a beneficial result of the greenhouse effect and may allow routine transit in its receding ice, shortens transportation distances between Europe and Asia. The growing importance of the region also attracted the attention of those nations who are geographically involved particularly the U.S. and Russia. Such is the importance that the US is looking to assert its influential presence and territorial interest in the area and Russia accounting a vast amount of gas and crude oil reserves from the region. On the other hand, crucial challenges remain and must be addressed. Merchant vessels aimed to operate in ice-covered waters are adhered to comply in accordance with the Ice Class requirements of Classification Society and National Regulations.[1~4]

Classification societies have recognized the fact that even though the Unified Rules on Polar Class vessels are in place, there is a need to address and meet the demands of current requirements through continuous updates.[5] Typical examples are merchant vessels navigating through ice of up to more than a meter thick thereby entailing reliability and integrity of the propulsion shafting system. On propulsion shafting system design, emphasis on strength, vibration, alignment, and sag & gap calculation are given during the design stage.[6] In addition, the interaction between the propeller and ice, i.e. ice milling and ice impact, results in higher load and exciting torque along the propulsion shafting system.[7] Hence, this paper estimates the design and excitation torque of Polar Class ships propulsion shafting system. In addition, the propeller – ice load interaction effect on the vibration characteristic of the propulsion shafting is analyzed.

2. Propulsion Shafting System for Polar Class Vessels [8~13]

The International Association of Classification Society (IACS) and classification societies i.e. as Den Norske Veritas (DNV), American Bureau of Shipping (ABS), and

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Lloyd's Register (LR) among others have laid the rules for Polar Class ships intended for Arctic navigation. In addition to this, state rules such as Finnish-Swedish ice class rules' Guidelines for the Application of the Finnish – Swedish Ice Class Rules and Canada's Arctic Shipping Pollution Prevention Regulations (C.R.C., c.353) are also in co-operation with the societies and have been integrating their regulations for the classification of ships.

Table 1 shows the main feature of IACS Polar Class Rule which is the range of ice classes. Operation in the most severe ice conditions is given in higher classes while lower classes are developed for light ice conditions only.[14]

Table 1 Polar Class Descriptions

Polar Class	Ice Description
PC 1	Year-round operation in all Polar waters
PC 2	Year-round operation in moderate multi-year ice conditions
PC 3	Year-round operation in second-year ice 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

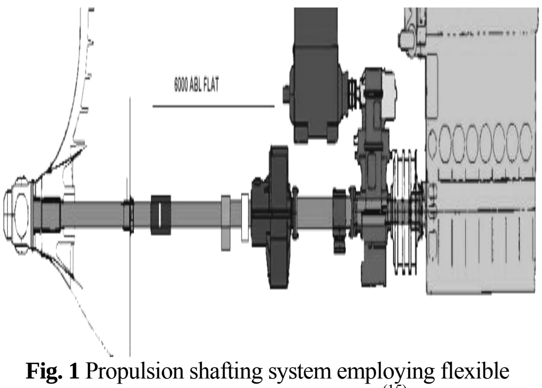

Fig. 1 Propulsion shafting system employing flexible coupling and reduction gear[15]

Figure 1 illustrates an example of propulsion shafting utilizing a marine reduction gear and flexible coupling in its drive train component. Design principle states that Polar class ships propulsion line strength should be designed for maximum load, plastic bending of a propeller blade shall not cause damages in other propulsion line components, and with sufficient strength. This principle dictates to the different propulsion systems such as diesel-electric, medium speed diesel with gear box and slow speed diesel, utilized in ice-going ships.

2.1 Propulsion Shafting System Design and Excitation Loads

Propulsion shafting system is a complex structure considering the different loads and torque it is made to withstand. Further, each propulsion shafting design will exhibit distinct characteristics. With the continual development of ice class vessels, not only is the hydrodynamic load is being considered but also the load brought about by the ice-propeller interaction wherein their variation is significant.[16] Under the LR Rules, it has identified the conditions and loading to achieve a satisfactory understanding of the characteristics of the shafting design.[17] As such, much importance is given on strength, vibration, alignment and sag & gap component of the propulsion shafting system during the design stage. On the other hand, torque, together with power and the rotational speed of the propulsion shaft, is a dominating load and can exhibit the dynamic characteristics of the engine.[18,19] This torque will either be static or dynamic in nature and this harmonic torque will manifest as torsional vibrations.[20] Furthermore, classification rules on shafting diameter, being a factor in torque load, was said to have had neglected the alternating load and should be treated as a guide only.[21] This alternating load can be associated to the contact (milling load or impact load) existing between the ice and the propeller.

2.2 Propeller Load, Forces and Excitation Torque $(Q_{(\varphi)})$[8b]

For the clarity of this paper, this focus will be entirely on the loading characteristics and the resulting torque of an open-type propeller assumed with a controllable pitch or fixed pitch blades during propeller-ice interaction.

When a propeller rotates in ice-covered water, it is subjected to contact loads, non-contact loads and generated forces. The non-contact loads refers to the hydrodynamic load on blade under open water condition.[16b] The contact loads, then again, are differentiated as ice milling- and as ice impact-type load. Sodhi described ice milling taking place when an ice is large or is trapped between the hull, wherein the ice is crushed resulting to higher loads. Ice impact is referred to smaller ice pieces which are passed through the propeller and resulting to moderate loads.[7b] JY Wang, et al. also hypothesized in their paper the ice loads acting on a propeller as separable hydrodynamic loads, inseparable hydrodynamic load (ice-impact load) and ice contact load. [22] A representation of total ice loads was made as the sum of ice milling load, separable hydrodynamic load and inseparable hydrodynamic load. Ice related loads refer to the sum of ice milling load and inseparable hydrodynamic load occurring during the milling period. Another characterization of these loads was made by H Soininen. Accordingly, he stated that definition for milling load and impact-type loads is not very clear. He assumed milling load is when ice made contact with the leading edge of the blade while impact load is when ice hits the back or face of the blade.[23]

IACS have defined the forces acting on the propeller blade during the propeller-ice interaction. The forces are namely, F_b, being the force bending the propeller blade backwards when the propeller mills an ice block while rotating ahead and, F_f, as the force bending a propeller blade forwards when a propeller interacts with an ice block while rotating ahead. Table 2 lists the design ice thickness and ice strength index to be used for the estimation of ice loads. H_{ice} refers to the ice thickness for machinery strength design, S_{ice} for the ice strength index for blade ice force and S_{qice} the ice strength index for blade ice torque.

Table 2 Design Ice Thickness and Ice Strength Index Value in Estimating Propeller Ice Loads.

Ice Class	H_{ice} [m]	S_{ice} [-]	S_{qice} [-]
PC 1	4.0	1.2	1.15
PC 2	3.5	1.1	1.15
PC 3	3.0	1.1	1.15
PC 4	2.5	1.1	1.15
PC 5	2.0	1.1	1.15
PC 6	1.75	1.0	1.0
PC 7	1.50	1.0	1.0

Equations 1 and 2 are the given formula for maximum backward blade forces while Equations 3 and 4 refer to the maximum forward blade force.

$$F_b = 27\, S_{ice} [nD]^{0.7} \left[\frac{EAR}{Z}\right]^{0.3} [D]^2 \quad \text{when } D \le D_{limit} \qquad \text{(Eq.1)}$$

$$F_b = -23\, S_{ice} [nD]^{0.7} \left[\frac{EAR}{Z}\right]^{0.3} [H\_ice]^{1.4} [D] \quad \text{when } D > D_{limit} \qquad \text{(Eq.2)}$$

$$F_f = 250 \left[\frac{EAR}{Z}\right] [D]^2 \quad \text{when } D \le D_{limit} \qquad \text{(Eq.3)}$$

$$F_f = 500 \left(\frac{1}{1-\frac{d}{D}}\right) H_{ice} \left[\frac{EAR}{Z}\right] [D] \quad \text{when } D > D_{limit} \qquad \text{(Eq.4)}$$

wherein:

$$D_{limit} = \left(\frac{2}{1-\frac{d}{D}}\right) H_{ice}$$

d = propeller hub diameter

D = propeller diameter

EAR = expanded blade area ratio

Z = number of propeller blades

Wang, in his paper, calculated the backward and forward forces (Eq. 1~4) and torque corroborated that ice loads per blade are much higher than hydrodynamic load of the whole propeller.[16c] The response torque at any shaft component of Polar Class PSS should consider the following: the excitation torque ($Q_{(\varphi)}$) at the propeller, the actual engine torque (Q_e) and the mass elastic system. IACS 13.4.3.4 established the maximum propeller ice torque applied to the propeller as:

$$Q_{max} = 105 \times (1 - d/D) \times S_{qice} \times (P_{0.7}/D)^{0.16} \times (t_{0.7}/D)^{0.6} \times (nD)^{0.17} \times D^3$$

when D < D_{limit} (Eq.5)

$$Q_{max} = 202 \times (1 - d/D) \times S_{qice} \times H_{ice}^{1.1} \times (P_{0.7}/D)^{0.16} \times (t_{0.7}/D)^{0.6} \times (nD)^{0.17} \times D^{1.9}$$

when D ≥ D_{limit} (Eq.6)

wherein:

D_{limit} = 1.81 H_{ice}

S_{qice} = ice strength index for blade ice torque

$P_{0.7}$ = propeller pitch at 0.7 R

$t_{0.7}$ = maximum thickness at 0.7 radius

IACS UR 13.4.6 set the design loads on propulsion shafting system for Polar class vessels. Stating, the propeller ice torque excitation for shaft line dynamic analysis is described by a series of blade impacts which are half sine shape and occur at the blade. Equations 7 and 8 indicate the torque due to single blade ice impact as a function of the propeller rotation angle. Table 3 details the C_q and α_i parameters for the propeller ice torque excitation in shaft line analysis. The total ice torque is achieved by summing the torque individual blades taking into account the phase shift 360°/Z. The number of propeller revolutions during a milling sequence is obtained according to Equation 11. The number of impacts is $Z \cdot N_Q$.

$$Q(\varphi) = C_q * Q_{max} * \sin(\varphi(180/\alpha_i)) \quad \text{when } \varphi = 0 \ldots \alpha_i \quad \text{(Eq.7)}$$

$$Q(\varphi) = 0 \quad \text{when } \varphi = \alpha_i \ldots 360 \quad \text{(Eq.8)}$$

$$N_Q = 2 \cdot H_{ice} \quad \text{(Eq.9)}$$

wherein:

$Q(\varphi)$ = torque due to single blade ice impact

Q_{max} = maximum propeller ice torque

φ = single blade phase angle impact

N_Q = number of propeller revolutions

H_{ice} = ice thickness for machinery strength design

Z = number of propeller blades

Table 3 C_q and α_i Parameters of Propeller Ice Torque Excitation for Shaft Line Dynamic Analysis

Torque Excitation	Propeller-ice Interaction	C_q	α_i
Case 1	Single Ice Block	0.5	45
Case 2	Single Ice Block	0.75	90
Case 3	Single Ice Block	1.0	135
Case 4	Two ice blocks with 45 degree phase in rotation angle	0.5	45

3. Case Study Presentation

On the supposition that during propeller-ice interaction, dominant torque stresses occur on the propeller and is transmitted towards the propulsion shafting. In this aspect, forces and torque response are represented by different functions and all propeller-ice loads. This transmitted excitation loads can be dissipated either by vibration dampers or flexible couplings. Hence, the damping coefficient performs an important role for the above-mentioned function. Equation 10 is the representation of force in relation to damping coefficient and angular velocity. The force in this equation can be presumed as the force stated in Equations 1~4 or the forces during the ice-propeller interaction at different load cases.

$$F = -cD\omega \quad \text{(Eq. 10)}$$

wherein:

F = force

c = damping coefficient

D = propeller diameter

ω = propeller angular velocity

Inasmuch as optimization of damping can be done during the design stage according to propulsion shafting requirement, the excitation loads on ice class propulsion shafting pose a challenge due to unpredictable responses in some stage of propeller-ice interaction. These dynamic responses can be due to relationships of ice thickness and strength, ship speed or shaft revolution to mention a few. Hence, variation of actual and calculated response is expected. Expanding Equation 10 considering the relationships of factors influencing the magnitude of loading in the propulsion shafting system, it is assumed to take the form of Equation 11.

$$F = -c \cdot C_i \cdot C_m \cdot C_s \cdot C_c \cdot D^a \cdot w \qquad \text{(Eq.11)}$$

wherein:

C_i = ice load factor
C_m = machinery factor
C_s = slippage factor
C_c = coupling factor
a = constant

This paper introduces the time response and slip factor on the propulsion shafting system that may affect the actual dynamic response during the propeller-ice interaction. In addition, an assumption of propulsion shafting configuration is made, i.e. rigid propulsion shafting, with flexible coupling and with Voith coupling.

3.1 Coupling Time Response Factor

Transient and continuous loads occur during propeller-ice interaction or every time a propeller blade hits an ice block. This results to stresses and must be dissipated by the shafting system. For rigid system, an optimized shafting design will be required to withstand the excitation load. This shafting configuration is assumed to be prone to material fatigue due to the cumulative stresses. On the case of flexible and Voith coupling, dissipation of stresses is expected. When excitation occurs on the blade, resulting stresses will emanate through the shaft and to the coupling itself. In this case, it is supposed that there will be time lag between the excitation occurrence and excitation dissipation brought about by the function of shafting length.

3.2 Coupling Slip Factor

The design of propulsion plant strength designates the propeller blade as the "weakest link" thereby preventing damages to other propulsion shafting component. Equation 5 and 6 above set the maximum propeller ice torque applied to the propeller. It is assumed then that extreme values exceeding the load limit will result damages to the propeller, such as propeller bending and in worst cases, loss of propulsion. However, these damages to the propeller are unwanted options.

With this theory, exploiting couplings potential of "slipping" during extreme excitation loading can prevent any major damage to the propeller.

4. Conclusion

This paper presented the existing Rules for Polar Class Vessel propulsion shafting system design according to classification societies and concerned states. Likewise, forces and excitation loads occurring during the propeller-ice interaction were known.

A case study is presented in this paper, summarizing:

1) This will be the spring board for future experimental and investigative studies. A concept, the time response factor and slip factor for flexible coupling, is presented in this paper. It is anticipated that the result of this study will form part in the guidance of propulsion shafting design of ice-class vessels.

2) Although different loads acting on the propeller can be defined as the total propeller-ice loads, different assumptions of researchers on ice milling load and ice impact load state may yield different experimental result. Hence, it is then suggested that loads existing during the propeller-ice interaction be defined and can be adopted by Rule bodies as guidance thereby creating a uniformity in experimental approach.

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