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Transient Torsional Vibration Response due to Ice Impact Torque Excitation on Marine Diesel Engine Propulsion Shafting

선박용 디젤엔진 추진축에서 빙 충격 토크 기진에 의한 과도 비틀림 진동 응답

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Key Words : Diesel Engine(디젤 엔진), Ice Impact Torque(빙 충격 토크), Marine Propulsion Shafting(선박 추진 축계), Transient Torsional Vibration(과도 비틀림 진동)

ABSTRACT

In recent years, there has been an increasing demand to apply the new IACS(International Association of Classification Societies) standards for ice and polar-classed ships. For ice-class vessel propulsion system, the ice impact torque design criterion is defined as a periodic harmonic function in relation to the number of the propeller blades. However, irregular or transient ice impact torque is assumed to occur likely in actual circumstances rather than these periodic loadings. In this paper, the reliability and torsional vibration characteristics of a comparatively large six-cylinder marine diesel engine for propulsion shafting system was examined and reviewed in accordance with current regulations. In this particular, the transient ice impact torque and excessive vibratory torque originating from diesel engine were interpreted and the resonant points identified through theoretical analysis. Several floating ice impacts were carried out to evaluate torque responses using the calculation method of classification rule requirement. The Newmark method was used for the transient response analysis of the whole system.

요 약

최근 극지 선박의 수요가 늘어나고 있고 IACS(국제선급연합)에서는 대빙 선박에 대한 새로운 기준이 적용되고 있다. 이 선박에서는 추진시스템에 대한 대빙 설계 기준으로 빙 충격 토크는 프로펠러 날개 수를 중심으로 한 조화 함수로 규정되어 있다. 그러나 실 상황에서는 이러한 주기적인 기진 토크보다는 불규칙한 빙의 충격 토크가 발생할 수 있는 확률이 오히려 크다. 이 논문에서는 비틀림진동이 비교적 큰 6개의 실린더를 갖는 디젤엔진을 주 기관으로 한 추진시스템의 안정성을 검토하고자 한다. 특히 불규칙한 빙 충격 토크와 디젤엔진에서 발생하는 진동토크를 동시에

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고려하여 비틀림진동의 공진점을 통과할 과도 비틀림 진동 응답을 이론적으로 해석하였다. 여기서 빙 충격토크는 빙이 프로펠러에 부딪칠 때를 여러 유형별로 가상하여 선급에서 규정된 방법에 의해서 구하였다. 전체적인 시스템의 과도응답 해석은 직접적분방법의 하나인 뉴마크(Newmark) 법을 이용하였다.

Nomenclature

$c_{0.7}$: Propeller blade chord length(m) at 0.7R
D : Propeller diameter(m)
d : Propeller hub diameter(m)
EAR : Expanded propeller blade area ratio
F_b : Maximum backward blade force(kN)
F_f : Maximum forward blade force(kN)
H_{ice} : Thickness for machinery strength design(m)
K_{Aice} : Application factor due to ice shock
lim : Limit
n : Rotational speed at bollard condition(r/s)
$P_{0.7}$: Propeller pitch at 0.7R(m)
Q : Torque(kNm)
R : Propeller radius
S : Strength index
$t_{0.7}$: Maximum thickness at 0.7R
T : Torque
T_0 : Maximum continuous torque(kN·m)
Z : Number of propeller blades

1. Introduction

Other than hull design, propulsion plant design is demanded to ensure reliable and efficient transport solutions for growing trades in ice-bound waters[(1)]. Propellers, shafts, transmission systems and prime movers are the main components of ice-classed vessels. For the directly coupled propulsion system, its main advantage over other configuration is the reduction in the power transmission loss[(2)]. Likewise, numerous researches have been focused on propeller-ice interaction. Garma, in his paper, concluded the importance of propeller geometry design in considering ice loads under extreme and transient operating conditions[(3)]. Baik, on the other hand, presented a propeller boss slippage accident brought by impact load on propeller at sub-zero sea water condition[(4)]. Yet, ice class challenges are still present and it is stated that not all aspects of design for cold climates are accounted for by Ice Rules[(5,6)].

This paper obtained theoretically the transient torsional vibration response of a low-speed two-stroke diesel engine due to the irregular ice impact torque in order to evaluate the design and reliability of a directly-coupled propulsion system for an ice-class vessel. Classification regulations on ice-propeller interaction impact torque are applied[(7)] whereas the Newmark numerical method was used to analyze the transient torsional vibration overall response of the propulsion system[(8)].

2. Ice and Propeller Impact Torque Calculation

Additional provisions on regulations for Polar vessels sailing the Baltic Sea includes the Finnish - Swedish Ice Class Rules and the Canadian Arctic Shipping Pollution Prevention Regulations which must be applied. Polar classed vessels are divided into seven (7) classifications from PC1 ~ PC7. This Polar ship classification is based on different estimation factors for the calculation of design ice loads of the propeller, i.e., the ice thickness (H_{ice}) and the ice strength indexes for the blade ice force (S_{ice}) and the propeller ice torque applied (S_{qice}). For the calculation of the maximum backward blade force (F_b) and the maximum forward blade force (F_f), this paper applied the Eqs. (1) to (4) respectively in reference of propeller diameter (D).

The backward blade force formula when $D<D_{lim}$ is

$$F_b = -27 \cdot S_{ice} \cdot [nD]^{0.7} \cdot \left[\frac{EAR}{Z}\right] \cdot [D]^2 \quad (1)$$

whereas when $D \geq D_{lim}$,

$$F_b = -23 \cdot S_{ice} \cdot [nD]^{0.7} \cdot \left[\frac{EAR}{Z}\right] \cdot [H_{ice}]^{1.4} \cdot [D]^2 \quad (2)$$

The diameter limit is given as D_{lim}=0.85·[H_{ice}]$^{1.4}$. Accordingly the forward blade force formula when $D<D_{lim}$ is

$$F_f = 250 \cdot \left[\frac{EAR}{Z}\right] \cdot [D]^2 \quad (3)$$

and when $D \geq D_{lim}$,

$$F_f = 500 \cdot \left[\frac{1}{\left(1-\frac{d}{D}\right)}\right] \cdot H_{ice} \cdot \left[\frac{EAR}{Z}\right] \cdot [D] \quad (4)$$

The D_{lim} is given as $\left[\frac{2}{\left(1-\frac{d}{D}\right)}\right] \cdot H_{ice}$. The calculation for blade spindle torque is given in Eq. (5).

$$Q_{smax} = 0.25 \cdot F \cdot c_{0.7} \quad (5)$$

The calculations for the maximum propeller ice torque applied on open propeller are shown in Eqs. (6) and (7). If $D<D_{lim}$,

$$Q_{max} = 105 \cdot \left[1-\frac{d}{D}\right] \cdot S_{qice} \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot \left[\frac{t_{0.7}}{D}\right]^{0.6} \cdot (nD)^{0.17} \cdot D^3 \quad (6)$$

and if $D \geq D_{lim}$,

$$Q_{max} = 202 \cdot \left[1-\frac{d}{D}\right] \cdot S_{qice} \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot \left[\frac{t_{0.7}}{D}\right]^{0.6} \cdot (nD)^{0.17} \cdot D^{1.9} \quad (7)$$

The diameter limit for the calculation of the maximum propeller ice torque is defined as

$$D_{\lim} = 1.81 \cdot H_{ice}.$$

Shaft line dynamic analysis of the propeller torque excitation is described by the sequence of blade impacts being half sine shape and occurring at the blade. The torque due to a single blade impact as a function of the propeller rotation angle is represented in Eqs. (8) and (9).

$$Q_\phi = C_q \cdot Q_{max} \cdot \sin\left(\frac{180}{\alpha_i}\phi\right), \text{ when } \phi = 0 ... \alpha_i \quad (8)$$

$$Q_\phi = 0, \text{ when } \phi = \alpha_i ... 360 \quad (9)$$

In Table 1, the parameters C_q and α_i are shown. By taking into account the phase shift 360°/Z, the total ice torque is attained by summing the torque of single blades. Equation (10) is the formula for obtaining the number of propeller revolutions during a milling sequence.

$$N_Q = 2 \cdot H_{ice} \quad (10)$$

The number of impact is implied as $Z \cdot N_Q$.

The calculated torque variation in relation to the 6th order torsional vibration resonance of PL1 ice class subject vessel propulsion shafting at 52 RPM. is shown in Fig. 1.

Propeller assembly by shrink fitting using hydraulic equipment (fixed-pitch propeller blade) or bolts (variable pitch propeller blade) multiplied

Table 1 C_q and α_i parameters

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	Single ice block with 45 degree phase in rotation angle	0.5	45

by safety factor should be greater than any torque fluctuations.

Therefore, the torque required for assembly in the full speed range (T_{c1}) must satisfy Eq. (11) in the friction force 35 ℃. The minimum value for T_{c1} is $2.8 \cdot T_0$.

$$T_{c1} = 2.0 \cdot T_0 + 1.8 \cdot (K_{Aice} - 1) \cdot T_0 \tag{11}$$

Below is the application of ice factor due to impact load, as a minimum requirement of IACS (International Association of Classification Society) UR (Unified Requirements)_K, on shaft transmitted torque at maximum continuous rating.

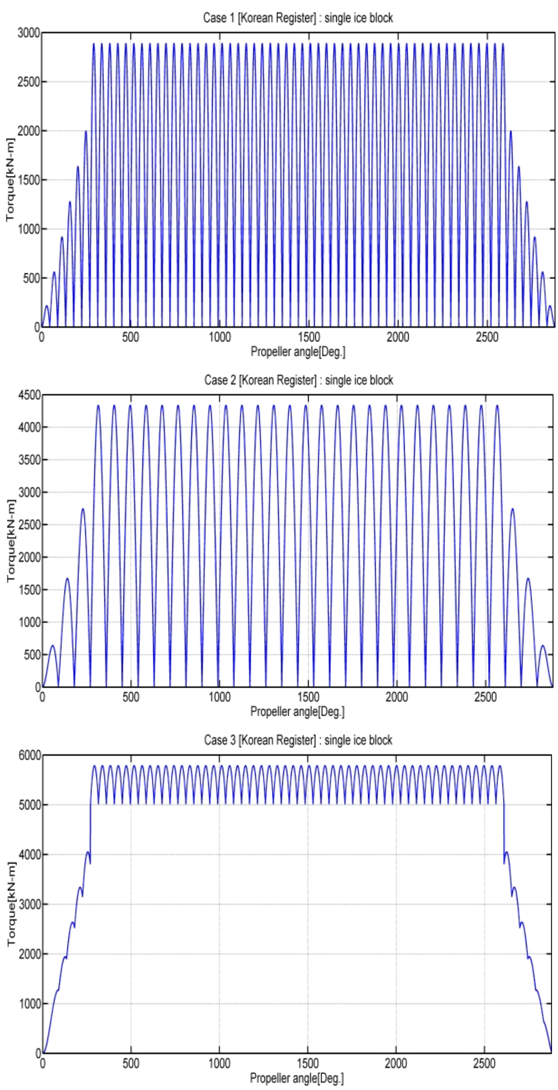

Fig. 1 Three cases of total ice impact torque for 4 blade propeller and 52 RPM of PC1 ice class

3. Theoretical Analysis and Result

The diesel engine and propulsion system particulars of a 154 k tanker as the subject vessel of this paper are given in Table 2. The torsional vibration measurement was carried out and the transient torsional vibration was analyzed in time domain as represented by the mass-spring system consisting of twelve (12) masses in Fig. 2. The propulsion shafting system excitation analysis was divided in two parts: the ice-propeller impact torque and the diesel engine excitation torque attributable to the cylinder gas pressure and reciprocating mass force of piston. Likewise, the analysis method was done utilizing a specialized software developed by the authors(9). The excitation torque curve at critical speed 52 RPM of 6^{th} order torsional vibration resonance, as shown in Fig. 3, of a six-cylinder engine is relatively higher when compared to Fig. 1 case 2 ice impact torque. Figure 4 illustrates the vibratory torque due to engine excitation at 52 RPM whereas Fig. 5 is the vibratory torque due ice-propeller impact torque. The total vibratory torque due to ice-propeller impact torque and engine excitation of PC1 ice class vessel at 52 RPM critical is shown in Fig. 6. In this case, the engine total vibration torque was analyzed in the time domain. However, the results have shown no significant differences with the results using the frequency domain order analysis. As such, actual vibration measurement was done during sea trial and the result confirms the vibratory resonance to be slightly higher than the theoretical analysis (Fig. 7)(10). The torsional vibration result confirms the full compliance of the design and assembly in accordance with the classification rule. Engine and propeller blade excitation torques are relatively low and higher resonance does not occur. However, the maximum and minimum excitation torque acting simultaneously in Fig. 6 is significantly higher compared to Fig. 4. This analysis applied the Classification Society rules on vibration response of the system during the initial phase but it should be considered in the subsequent phase that the occurrence of the ice impact torque will likely increase the actual vibration value. Hence, the result of this theoretical analysis considers simultaneously the torsional vibratory torque of the engine and the ice impact torque.

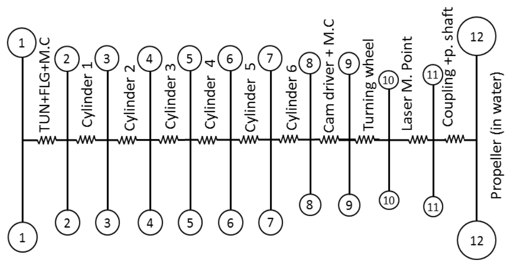

Fig. 2 Mass-spring system for torsional vibration analysis

Table 2 Specification for ship and propulsion shafting

Description	Major dimension	
Ship	L.B.P	264 m
	Depth(moulded)	23.1 m
	Scantling draft	17.0 m
	Ship speed(design) for ICE IA FS	18.0 kt
	Ship speed(service)	15.3 kt
Diesel engine	Type	6S70MC-C
	Max. continuous power	18,617 kW
	Max. continuous speed	91 RPM
	Cylinder bore	700 mm
	Stroke	2,800 mm
	Cylinder No.	6
	Transfer torque at m.c.r.	1,954 kNm
Propeller and shafting	Type	Fixed pitch propeller
	Blade No.	4
	Diameter	8.2 m
	Pitch	5.7 m
	M.O.I in water	144,210 kgm^2
	Intermediate shaft	ϕ610×7,815 mml
	Propeller shaft	ϕ770×8,178 mml

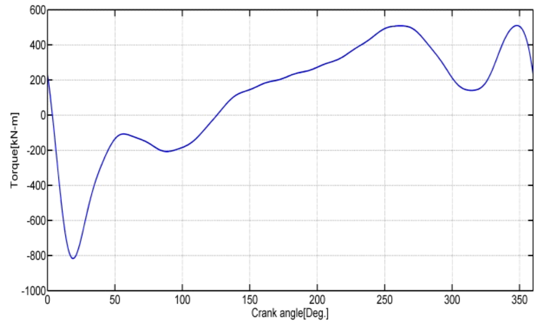

Fig. 3 Excitation torque due to gas pressure of cylinder and reciprocation mass of piston at the critical speed of engine cylinder number (1 node 6th order and 52 RPM)

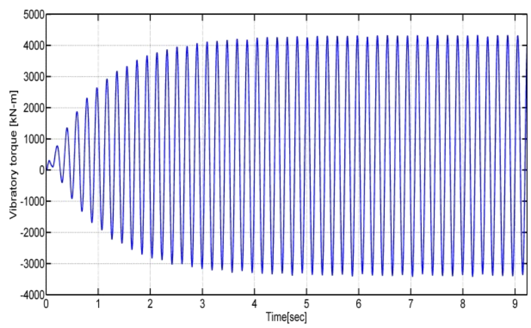

Fig. 4 Vibratory torque of intermediate shaft due to engine excitation at the critical speed of engine cylinder number (1 node 6th order and 52 RPM)

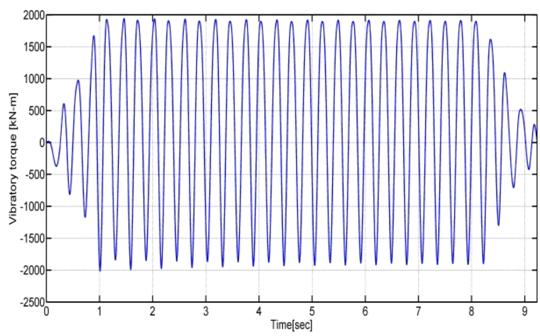

Fig. 5 Vibratory torque of intermediate shaft due to ice impact torque of PC1 class at the critical speed of engine cylinder number (1 node 6th order and 52 RPM)

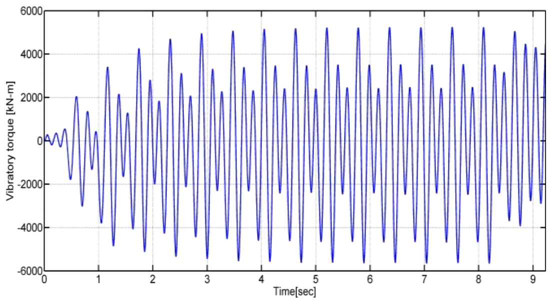

Fig. 6 Vibratory torque of intermediate shaft due to engine excitation and ice impact torque of PC1 ice class at the critical speed of engine cylinder number (1 node 6^{th} order and 52 RPM)

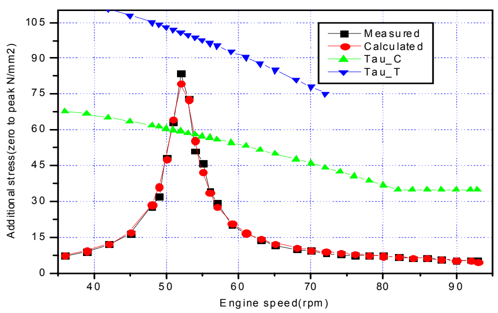

Fig. 7 Measured torsional vibration stress at intermediate shaft (scale : N/mm^2 = 44.57 kN-m)

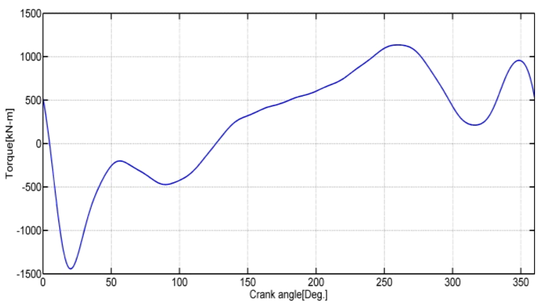

Fig. 8 Excitation torque due to gas pressure of cylinder and reciprocation mass of piston at the critical speed of propeller blade (1 node 4th order and 78.4 RPM)

In Fig. 8 above, the excitation torque curve of the propeller blades at critical speed 78.4 RPM is shown and the value was twice as that in comparison with Fig. 3. Figure 9 shows the vibratory torque mainly due to the engine excitation to be lower

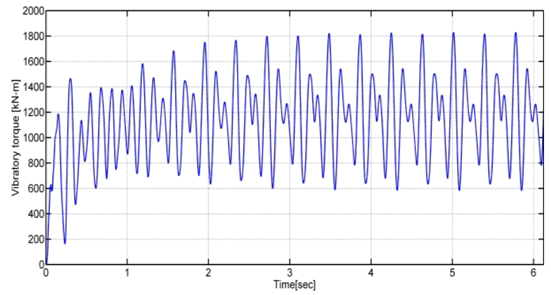

Fig. 9 Vibratory torque due to engine excitation at the critical speed of propeller blade number (1 node 4^{th} order and 78.4 RPM)

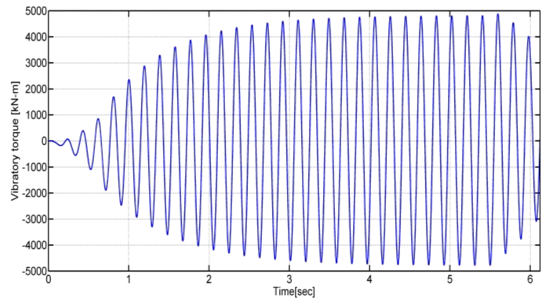

Fig. 10 Vibratory torque due to ice impact torque of polar class PC7 at the critical speed of propeller blade number (1 node 4^{th} order and 78.4 RPM)

than the resonance limit. The lowest e PC7 ice class grade was applied on the calculation of the ice impact vibratory torque of the propeller as seen in Fig. 10. The vibratory torque was seen to significantly increase compared to Fig. 5. Thus, the collected ice impact torque data information operating at prohibited range should be set, with the scope on the safety of the whole system and not only of the propulsion shaft. Figure 11 illustrates the increased total vibratory torque reflecting both the engine torque and the ice impact torque in comparison with Fig. 11. Hence, if the results of these ice vessels ice impact torque will affect the strength of the propeller, the propulsion and ice impact torque being far from the prohibited range should be taken into account for propeller shrink fitting assembly and should be considered both in the design and assembly stage.

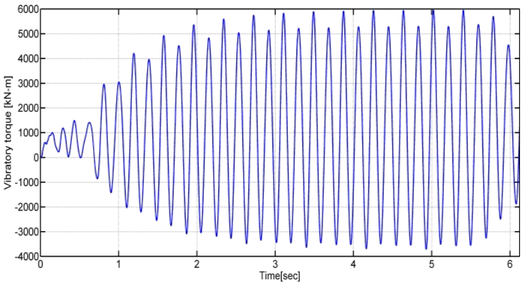

Fig. 11 Vibratory torque due to engine excitation and ice impact torque of PC7 ice class at the critical speed of propeller blade number (1 node 4^{th} order and 78.4 RPM)

3. Conclusion

In this paper, a theoretical analysis was performed to obtain the transient torsional vibration response due to ice impact torque in accordance with the Classification Rules. This was carried out in order to review the safety of propulsion systems directly coupled to main engine. The results are as follows:

(1) For low-speed two-stroke diesel engine having seven (7) cylinders or less, engine operation at barred speed range (critical speed) is prohibited due to torsional vibration limits. This vibratory torque increases further if the ice impact torque will act simultaneously on the propulsion shafting. The engine and propulsion excitation torque characteristics should be considered at the same time. In particular, the vibratory torque increased significantly when the engine is passing in the reverse rotation owing to the backward force on the propeller compared to the forward force. This force is the opposite action

(2) The new operation guide on ice impact torque is needed to protect ice-classed vessels. Even applying the lowest ice class grade factor, increased torsional vibratory torque on the resonance range acting on the propeller shaft and engine was confirmed.

Considering the IACS regulation and the Korean Register, it has been recommended to avoid resonance and propeller blades resonance should be within ±20 % range of the maximum continuous operating speed.

(3) For ice class propulsion systems directly coupled to a two-stroke low-speed marine engine, the resonance due to engine's number of cylinders should be considered alongside with the number of propeller blades. A smaller 8-cylinder engine can be employed in view of lower natural frequency of torsional vibration and considered advantageous to the main propulsion design and safety.

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