

Underwater Acoustics: Paper ICA2016-35**Numerical and experimental prediction methods of cavitation noise radiated by underwater propellers****Taehyung Kim^(a), Jonghoon Jeon^(b), Sunghan Chu^(c), Sunghoon Kim^(d), Wonho Joo^(e)**^(a) Advanced Research Institute, Hyundai Heavy Industries, Republic of Korea, thk81@hhi.co.kr^(b) Advanced Research Institute, Hyundai Heavy Industries, Republic of Korea, jhjeon@hhi.co.kr^(c) Maritime Research Institute, Hyundai Heavy Industries, Republic of Korea, cgcrew@hhi.co.kr^(d) Advanced Research Institute, Hyundai Heavy Industries, Republic of Korea, shkim74@hhi.co.kr^(e) Advanced Research Institute, Hyundai Heavy Industries, Republic of Korea, whjoo@hhi.co.kr**Abstract**

Underwater propeller cavitation noise is composed of tonal blade rate noise and high frequency broadband noise. In this paper a numerical method is developed to predict propeller tonal noise while experimental approaches are performed to predict broadband noise. For prediction of the sheet cavitation which contributes to tonal noise characteristics, its area and volume on the propeller blades are calculated by finite volume method. Then propeller tonal noise is calculated using the acoustic analogy with consideration of cavitation volume variation on the blade surface in the time domain. This procedure was validated with the acoustic measurement test in the water tunnel. The experimental approaches for propeller broadband noise are composed of the development of semi-empirical formula through water tunnel test and the onboard measurement in the real ship. The semi-empirical formula for tip vortex cavitation noise is developed based on the aero-acoustic theory of tip vortex formation noise and then optimized through water tunnel test for nine kinds of model propellers. The transfer function is developed to acquire quasi-free field acoustic response using an underwater loudspeaker in the water tunnel and the towing tank. The propeller broadband noise is also predicted by sound transmission coefficient method using the relationship between sound transmission coefficients in the dry dock and structure-borne noise measurement at the sea trial. The proposed methods were validated by underwater radiated noise measurement during sea trials. From the results, it is expected that the proposed methods enable to predict propeller cavitation noise with ease and accuracy.

Keywords: Marine propeller, Cavitation noise, Blade passing frequency, Broadband noise, Sound transmission coefficient

Numerical and experimental prediction methods of cavitation noise radiated by underwater propellers

1 Introduction

Propeller cavitation noise has high adverse impacts on both ship and marine environment for wide frequency ranges [1]. Underwater propeller noise is directly related to survivability for military vessels and is closely connected with quality level and ship orders for commercial vessels. The report of IMO/MEPC (International Maritime Organization) stated that noise in the low frequency range of 10 Hz to 1 kHz has huge impacts on the marine ecosystem [2].

Numerical investigations of propeller cavitation noise have been mainly focused on the prediction of blade rate noise related to sheet cavitation. Acoustic prediction using the Ffowcs Williams and Hawkings equation [3] based on the Lighthill's acoustic analogy [4] was performed by Seol et al [5]. They modelled the sheet cavitation volume on the blade as a point source of thickness noise. Ianniello et al.[6] predicted the cavitation noise using RANS (Reynolds averaged Navier Stokes) simulation and acoustic analogy. Hsiao et al. [7] focused on the tip vortex cavitation inception noise based on the Rayleigh-Plesset equation of bubble dynamics.

Prediction of propeller broadband noise has been performed mainly by model and full scale measurements. Yamada et al.[8] and Oshima et al.[9] suggested the prediction methods for tip vortex cavitation noise by experimental deduction of tip vortex cavitation inception and its scaling exponent. Jeon et al. [10] suggested the indirect prediction method using sound transmission coefficient through hull transfer function measurement in the dry dock and subsequent onboard measurement of structure-borne noise at the sea trial.

In this paper, advanced numerical and experimental noise prediction methods are presented through the improvement of cavitation source modelling, the development of semi-empirical formula, and the validation of transmission coefficient method with full scale measurements. In particular, propeller tonal noise prediction with a distributed cavitation source model is described. Then the development of semi-empirical formula for propeller cavitation broadband noise is described. This formula is based on the theory of tip vortex formation noise and optimized by water tunnel tests with nine kinds of model propellers. Lastly, the sound transmission coefficient method is carried out to predict cavitation broadband noise for a full scale ship. This method is validated by the underwater radiated noise measurement in deep water. The test vessel is a commercial 14,200 TEU container vessel built by Hyundai Heavy Industries in 2015 with an eleven-cylinder two-stroke diesel engine and five-bladed propeller.

2 Numerical Approach

2.1 Cavitation flow analysis

We construct a reduced computational domain with propeller, shaft, rudder and wake screen to improve numerical calculation efficiency and to compare numerical solution directly with water

tunnel test. This computational domain has a square cross section of 0.6m x 0.6m and is extended from the propeller rotation plane to the upstream and downstream location as 0.6m and 1.2m, respectively. Figure 1 shows the computational domain used for this analysis. The software is STAR-CCM+ 10.06 and we consider Spalart-Allmaras 1-equation turbulence model with sliding polyhedral mesh. The seed density and seed diameter for cavitation model are $1.0 \times 10^{12}/m^3$ and $1.0 \times 10^{-6}m$, respectively. The number of grid points for blade is 6.3×10^5 while the number of grid points for surrounded domain is 4.2×10^5 . The symmetry condition is imposed at all boundaries except the inlet and outlet boundaries. The velocity inlet boundary condition and the initial velocity condition are applied for the inlet boundary with the axial wake distribution as shown in Figure 2. The pressure outlet condition is imposed for the water tunnel outlet boundary.

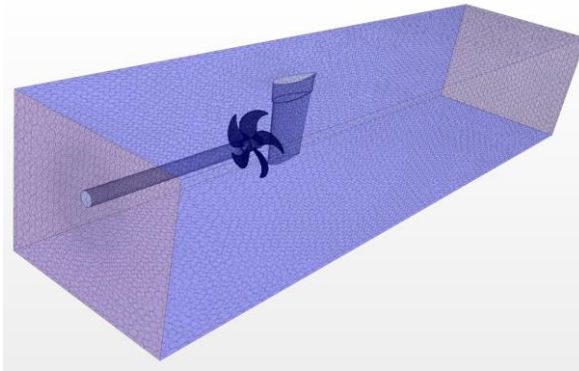


Figure 1: Computational domain

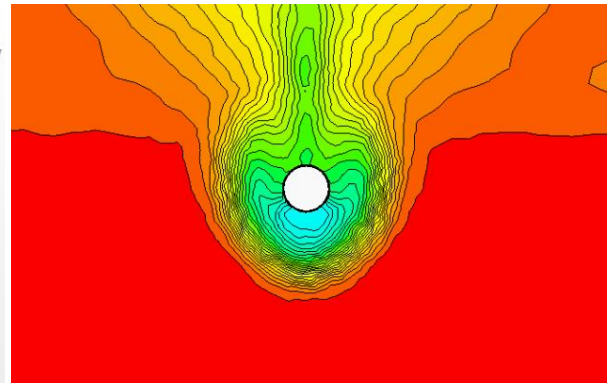


Figure 2: Inlet axial wake distribution

2.2 Acoustic prediction

Sound generation by turbulence and moving surfaces in arbitrary motion can be predicted by Ffowcs-Williams and Hawkings equation [3] which is based on Lighthill's acoustic analogy [4]. This is the rearranged Navier-Stokes equation which has the form of an inhomogeneous wave equation. The source terms are composed of monopole source, dipole source on the body surface and quadrupole source in the volume surrounding that body. The differential form of Ffowcs Williams-Hawkings (FW-H) equation can be described as:

$$\frac{1}{c^2} \frac{\partial^2}{\partial t^2} p'(\vec{x}, t) - \nabla^2 p'(\vec{x}, t) = \frac{\partial}{\partial t} [\rho_0 v_n \delta(f)] - \frac{\partial}{\partial x_i} [l_i \delta(f)] + \frac{\partial^2}{\partial x_i \partial x_j} [T_{ij} H(f)] \quad (1)$$

where v_n is a local velocity of the body normal to the body surface defined as $f = 0$. l_i is components of the local force on the body surface, and $\delta(f)$ and $H(f)$ are Dirac delta function and Heaviside function, respectively. The monopole source is caused by the displacement of fluid as the blade passes through it. The sheet cavitation volume on the blade surface is also considered as additional term of monopole source. The dipole source resulted from unsteady motions of force distribution on the blade surface is also contained to predict thickness noise.

Therefore the final form of acoustic prediction formula

$$p'(\vec{x}, t) = p'_T(\vec{x}, t) + p'_L(\vec{x}, t) + p'_C(\vec{x}, t) \quad (2)$$

Where

$$4\pi p'_T(\vec{x}, t) = \int_{f=0} \left[\frac{\rho_0 (\dot{v}_n + v_{in})}{r |1 - M_r|^2} \right]_{ret} dS + \int_{f=0} \left[\frac{\rho_0 v_n (r \dot{M}_r + c M_r - c M^2)}{r^2 |1 - M_r|^3} \right]_{ret} dS \quad (3)$$

$$4\pi p'_L(\vec{x}, t) = \frac{1}{c} \int_{f=0} \left[\frac{\dot{l}_r}{r |1 - M_r|^2} \right]_{ret} dS + \int_{f=0} \left[\frac{l_r - l_M}{r |1 - M_r|^2} \right]_{ret} dS + \frac{1}{c} \int_{f=0} \left[\frac{l_r (r \dot{M}_r + c M_r - c M^2)}{r^2 |1 - M_r|^3} \right]_{ret} dS \quad (4)$$

$$4\pi p'_C(\vec{x}, t) = \int_{f=0} \left[\frac{\rho_0 \dot{Q}(\tau^*)}{r |1 - M_r|^2} \right]_{ret} dS + \int_{f=0} \left[\frac{\rho_0 \dot{Q}(\tau^*) \dot{M}_i \hat{r}_i}{r |1 - M_r|^3} \right]_{ret} dS + \int_{f=0} \left[\frac{\rho_0 \dot{Q}(\tau^*) c_0 (M_r - M^2)}{r^2 |1 - M_r|^3} \right]_{ret} dS \quad (5)$$

Equation (3) represents the acoustic pressure fluctuation related to monopole source of thickness noise. Equation (4) is related to the dipole source contributes to loading noise. Equation (5) represents the acoustic pressure fluctuation at the observer time t and position \vec{x} by the sheet cavitation on the blade surface. This equation is suggested by Seol [5]. The integrands with $1/r$ are far-field terms and those with $1/r^2$ are near-field terms. A dot over a variable indicates the source time derivative of that variable. The subscript n , r and M refer to the dot products with the unit normal vector, the unit radiation vector, and the surface velocity vector normalized by the speed of sound, respectively. As the volume of sheet cavitation is changed according to the rotation of blades, the acoustic pressure fluctuation is stronger when the sheet cavitation moves closer to the receiver than when the sheet cavitation moves away from the receiver. This phenomenon is caused by the Doppler effects and is valid for the receiver fixed with the same distance from the centre of rotation.

The distributed source model is developed to predict the acoustic field emitted by complex pattern of sheet cavitation variation. The point- and distributed-source models are illustrated in Figure 3. This point-source model assumes that the monopole source related to sheet cavitation volume is concentrated on a specific point of blade surface. However, the resulted sound pressure level is different according to the choice of source position. The calculation error is also increased when the cavitation volume shape is complex. Therefore a distributed source model is suggested to describe the sheet cavitation volume more accurately and to reduce the prediction error for the blade passing frequency harmonics noise. The five representative cavitation volumes are assumed to be distributed sources along the blade span from $0.5 r/R$ to $0.99 r/R$.

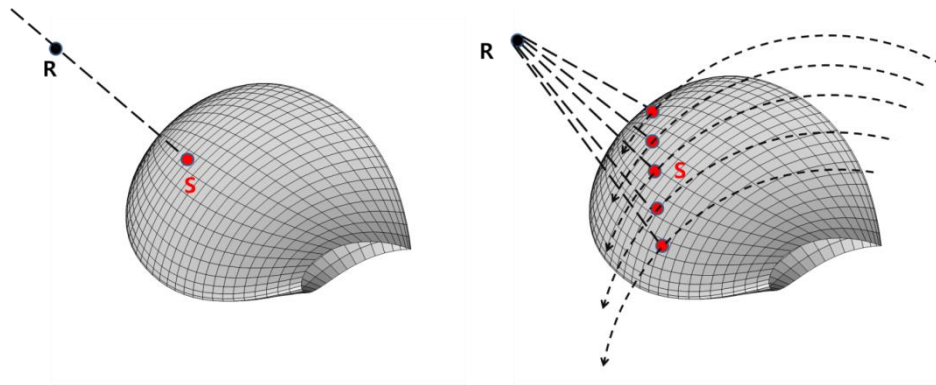


Figure 3: Schematics of point-source model (left) and distributed-source model (right)

3 Experimental Approach

3.1 Development of semi-empirical formula through water tunnel test

The nine kinds of model propeller noise tests were performed at the medium sized water tunnel to verify numerical solution. Table 1 summarizes the types of test propellers designed and produced by Hyundai Heavy Industries. The noise measurement was performed using hydrophone B&K8103 with a chamber type system. The acoustic distortion effects of the water tunnel walls are calibrated by measuring transfer function using the underwater loud speaker, Electro-Voice UW30, at both water tunnel and deep water towing tank (210 x 14 x 6 m) located in the Hyundai Heavy Industries.

Table 1: Types of test propellers

No.	Vessel	No.	Vessel
1	8,500 TEU Container Carrier	6	174,000 CBM LNG Carrier
2	13,800 TEU Container Carrier	7	105,000 DWT Product Tanker
3	14,000 TEU Container Carrier	8	181,000 DWT Bulk Carrier
4	82,000 CBM LPG Carrier	9	320,000 DWT Crude Oil Carrier
5	154,800 CBM LNG Carrier		

According to Baiter [11]’s works, the tip vortex cavitation shows dominant contribution to the continuous sound spectrum than other cavitation types. Therefore propeller cavitation broadband noise is assumed to be mainly generated by tip vortex cavitation formation. In this study, the proposed formula for tip vortex cavitation noise is developed by modification of aero-acoustic research performed by Brooks and Marcolini [12]. As a result of experimental test using nine model propellers, the semi-empirical formula for the tip vortex cavitation formation noise is optimized. The following equation is the final form of the developed formula. In this formula, the turbulent flow noise component is considered near the propeller tip in the local separating flow.

The contribution terms for the tip vortex core diameter, the blade boundary layer thickness, the Strouhal number and the Reynolds number are considered as following equations:

$$SPL_{TVC_contribution} = -1.5(\log(St) + 0.7)^2 + 31$$

$$St = f_{vs} \times \frac{D_{core}}{\sqrt{(2\pi n_m)^2 + (V_m)^2}}$$

$$D_{core} = \alpha \delta_{thick} \quad (6)$$

$$\delta_{thick} = 0.154cRe^{-1/7}$$

$$Re = l_{chord} \sqrt{(2\pi n_m)^2 + (V_m)^2} / \nu$$

where St is the Strouhal number and f_{vs} is vortex shedding frequency. D_{core} is the vortex core diameter, and α and δ_{thick} are the vortex core scaling factor and the boundary layer thickness at the trailing edge at $0.7r/R$ location in the propeller spanwise direction, respectively. The chord length l_{chord} is assumed to be the quarter of propeller diameter. The Re means the Reynolds number at the $0.7r/R$ location in the propeller spanwise direction. The other constants are optimized by the water tunnel model tests in this paper.

$$SPL_{TVC} = 10\log Z + 60\log D + 60\log N - 20\log f - 1.5(\log St + 0.7)^2 + 31 \quad (7)$$

Equation (7) represents the sound pressure level normalized to a distance of 1 m for cavitation propeller with consideration of tip vortex cavitation noise. This equation is derived by the combination of the measurement formula by Ross [13] and the developed tip vortex cavitation prediction formula. Here, Z is the number of blades, D and N are the propeller diameter and the rotation speed in RPS (revolutions per second), respectively. The scaling law used in this paper is based on the cavitation committee method of ITTC [14]. This law was derived from the assumption that the cavitating dynamics between model and full scale was identical. The continuous part of the sound spectrum can be expressed as follows:

$$\frac{f_s}{f_m} = \frac{n_s}{n_m}, \quad SPL_s = SPL_m + 40 \log_{10} \frac{n_s}{n_m} + 60 \log_{10} \frac{D_s}{D_m} \quad (8)$$

where f is the frequency, n is the revolutions per second, r is distance from the centre of propeller to the receiver, D is the propeller diameter and the p is the acoustic pressure. The suffixes m and s refer to model and ship scale respectively.

3.2 Validation of sound transmission coefficient method through sea trial

The structure-borne noise measurement was performed at the centre of steering gear room in the 14,200 TEU container vessel at the sea trial as shown in Figure 4. The measurement equipment was an accelerometer Dytran type 3148E which has a sensitivity of 100mV/g and frequency response of $\pm 5\%$ in the range of 0.5 to 5,000 Hz. The sound transmission coefficient

of hull structure was measured for the similar 13,800 TEU container vessel of the Hyundai Heavy Industries by Jeon et al [10]. Experiments in dry dock were carried out to measure sound transmission coefficient of each ship's stern. The structure-borne noise on each ship's outer spaces, surfaces, and onboard spaces were measured simultaneously while propeller and hull were excited outside by impact hammers and impulsive noise generators. The hull's structure-borne noise transmission coefficients were measured using the ratio of inside and outside signals.



Figure 4: Onboard structure-borne noise measurement in the steering gear room

The underwater radiated noise from the real ship at specific operating conditions was measured with the aid of KRISO (Korea Research Institute of Ships and Ocean Engineering). The test was performed using a floating type remote-transmitted hydrophone system in accordance with the ISO 17208 [15]. The number of total hydrophones was three and the deepest one was located 300 m in depth. Table 2 summarizes the specifications of the test vessel and conditions.

Table 2: Specifications of test vessel: 14200 TEU container carrier

Item	Value
Ship builder	Hyundai Heavy Industries
Site	East Sea of the Republic of Korea
RPM	65% MCR / 85% MCR
Water depth	400 meter

4 Results and Discussion

The RPM was constant at 1500 RPM for the water tunnel test. The similarity of cavitation number and thrust coefficient of the model propellers were matched with the real ship's conditions. The angular position of a key blade is measured from the vertically upward position in a counter clockwise direction. The cavitation patterns of the model test and the corresponding numerical result are shown for specific angular position in Figure 5. The calculated cavitation patterns show good agreement with the experiments for both 65% and 85% MCR conditions. The sheet cavitation is located in the spanwise range 0.6 to 0.99 r/R.

Figure 6 shows the comparison of cavitation tonal noise between numerical analysis and water tunnel measurement for the model propeller of 14,200 TEU container carrier. Both the point source model and distributed source model show good agreement with the measured sound pressure level up to the third blade passing frequency harmonics within 5 dB error bound. The prediction accuracy is improved up to 3 dB for the second blade passing frequency harmonics for the distributed source model rather than the point source model. The first blade passing frequency sound pressure level is mainly affected by the overall shape of acoustic wave while the small fluctuation of that contributes to the second and higher harmonics.

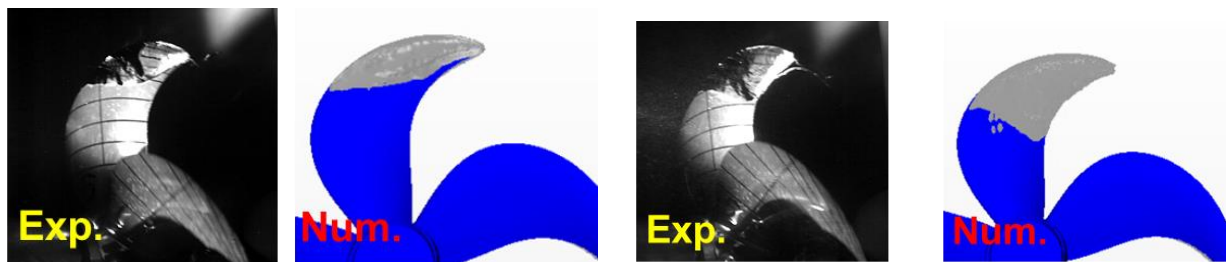


Figure 5: Photograph of the cavitation patterns and the results of numerical flow solver: Blade angle 6 degree, 65% MCR (left), 85% MCR (right)

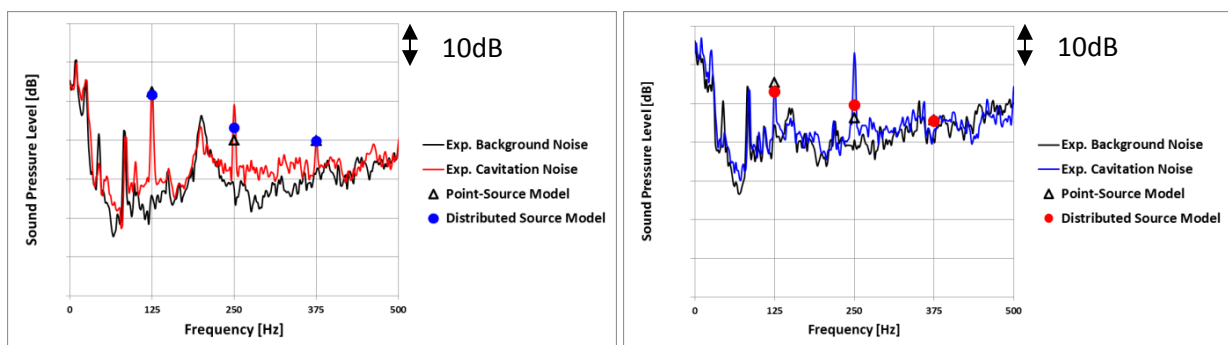


Figure 6: Spectrum of propeller cavitation tonal noise, 65% MCR (left), 85% MCR (right)

The propeller cavitation broadband noise is predicted using the developed semi-empirical formula and then converted to the full scale source level by the ITTC law [14]. The propeller cavitation noise is also predicted using the sound transmission coefficient method. The semi-empirical formula and the sound transmission coefficient method were used to estimate source levels of 14,200 TEU container carrier for 1/3-octave band (Figure 7). The developed semi-empirical formula shows good agreement with the experimental data within 5 dB for the frequency range of 200 Hz to 3 kHz. However, the level of difference is increased for the frequency range of 100 Hz to 200 Hz. The difference is caused by the lack of consideration of other cavitation noise sources such as hub vortex cavitation and cloud cavitation. The simultaneous consideration of tip vortex cavitation and other cavitation types is difficult to be

embodied in the water tunnel test because the range of cavitation number is different from each other. The semi-empirical formula presented in this study provides tools to estimate propeller cavitating broadband noise levels especially for tip vortex cavitation noise. The indirect sound transmission coefficient method also shows good trends with underwater radiated noise measurement. The limitation of this method is that the sound transfer function needs to be found according to various ship types by dry dock test.

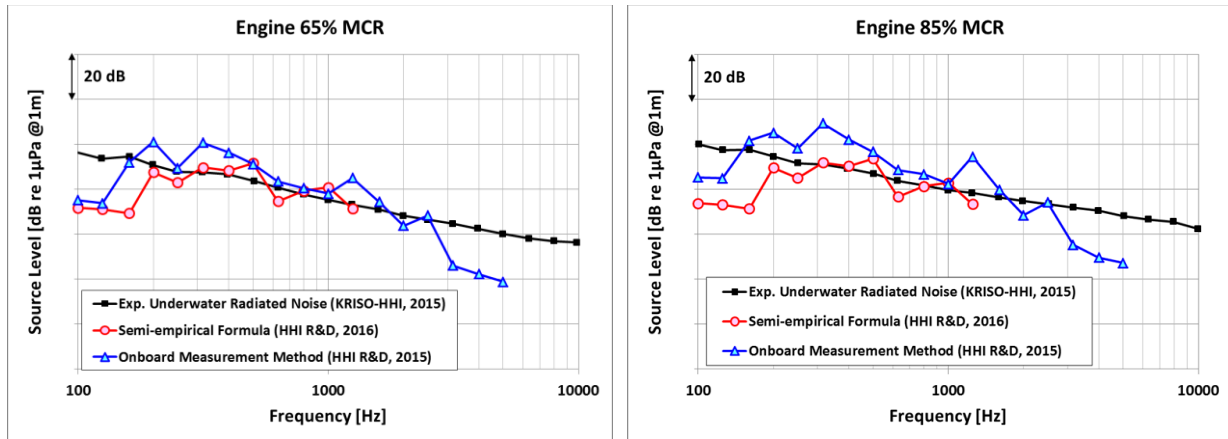


Figure 7: Spectra of propeller cavitating broadband noise; 65% MCR (left), 85% MCR (right)

5 Conclusions

This study described and compared acoustic levels and spectral characteristics from the numerical method for propeller cavitation tonal noise and an empirical method for broadband noise. The propeller tonal noise is predicted by the distributed source model based on the acoustic analogy and is verified with the water tunnel experiments. As a result, the proposed method shows good agreement with the measurement within 3 dB error bound.

The semi-empirical formula is developed based on the tip vortex formation theory and is optimized by water tunnel test for various propeller types. The underwater loudspeaker was used to measure transfer function between the tunnel wall and towing tank. The tank was assumed as quasi-free field for underwater sound radiation. The propeller cavitation noise is also estimated by the sound transmission coefficient method through the dry dock test of similar sized container ship and onboard structure-borne noise measurement. The proposed semi-empirical formula and the sound transmission coefficient method demonstrate good agreement in the frequency range of 200 Hz to 3 kHz with the underwater radiated noise measurement for the real container ship. Further studies, with multiple measurements of various ship types, will build on the robust methods of underwater radiated noise from ship characteristics and operating parameters.

Acknowledgments

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