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Comparison of Ship's Speed Loss in Waves in Some Numerical Simulation Methods

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ABSTRACT

This study evaluates the difference in speed loss, engine power, and fuel oil consumption of a bulk carrier from the viewpoint of weather routing. The evaluation of speed loss is important as speed loss influences accurate ship operation. The evaluation of speed loss is based on seakeeping and machinery theories. Speed loss due to added resistance is evaluated by different simulation methods, and the results are compared. Furthermore, to reproduce actual operations, three types of engine controls are considered. Numerical simulations for evaluating speed loss are carried out for a 20,000DWT-class bulk carrier in irregular wave patterns. No significant difference is observed in speed losses evaluated using the enhanced unified theory and new strip method when the mean wave period is longer. However, NSM underestimates speed loss if the mean wave period is relatively shorter. In addition, it is known that asymptotic formulae contribute accurate evaluations in a short period of waves. It is possible to implicitly evaluate speed loss under actual situations if these seakeeping models are combined with propulsion and engine control models. This study conducts basic research on this possibility to improve safe and efficient operation of vessels in rough seas.

Keywords: speed loss, added resistance, engine control, fuel oil consumption, weather routing

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1. INTRODUCTION

In rough seas, the various influences of waves and winds on ships are inevitable; thus, it is necessary to evaluate these influences. In addition to influencing the motions of ships, waves cause a complex speed loss with added resistance in ships. The EEDI (Energy Efficiency Design Index) requires a decrease in CO_2 gas emission, which depends on the fuel consumption of the main engine. This means that an accurate evaluation of speed loss and other related factors is necessary in rough sea states. Speed loss depends on various factors that are related to the performance of a ship. It is already known that an adequate combination is necessary for accurate evaluation. There are many studies on the validation of ship performance in model tests. However, there are not many studies that have validated the performance of ships in actual seas. In particular, added resistance of waves significantly contributes to speed loss.

There are also many unknown points of engine controls in rough seas. Here, the influence due to numerical accuracy of added resistance is compared among the EUT (Enhanced unified theory) (Kashiwagi, 1997), the NSM (New strip method) (Salvesen et al., 1970), and asymptotic formulae. There are many studies that have compared computed added resistance with the measured results of model tests. However, there are only a few studies that have validated speed loss by comparing with results measured under actual sea conditions. It is difficult to obtain a continuous database regarding the performance of ships in rough seas. This is a very important part to construct accurate weather routing with seakeeping, propulsion and machinery models.

The NSM is a practical method and is used to estimate relative ship motions and added resistance. However, various numerical methods have been proposed as computer performance has been improving. It is necessary to compare the evaluated accuracy of ship performance using each method. Here, the computed results of various parameters, such as added resistance, speed of the ship, and engine power of a 20,000DWT-class bulk carrier are evaluated using different methods, and the results are compared with each other. Finally, the difference among numerical models is revealed to simulate speed loss in rough seas.

2. Numerical Modeling

2.1 Seakeeping models

The implicit consideration of hydrodynamic forces with the forward speed of floating bodies is a complicated phenomenon. Three-dimensional models, such as the Rankine panel method or time domain models, are expected to accurately estimate hydrodynamic forces. However, these models are time consuming and therefore are not suitable for practical applications. The unified theory consists of matching the solutions of two- and three-dimensional problems of hydrodynamic relations, the inner and outer regions of slender ship body. EUT is one of the unified theories, and can partly consider three-dimensional wave effects in the outer region. The theoretical details of EUT are not elaborated here as they can be found in the reference (Kashiwagi, 1997).

In the unified theory, the inner boundary conditions are defined using the following equations:

$$\frac{\partial^2 \phi_j^{(i)}}{\partial y^2} + \frac{\partial^2 \phi_j^{(i)}}{\partial z^2} = 0 \tag{1}$$

$$\frac{\partial \phi_j^{(i)}}{\partial z} - K\phi_j^{(i)} = 0 \quad \text{at } z=0 \tag{2}$$

$$\frac{\partial \phi_j^{(i)}}{\partial N} = N_j + \frac{U}{i\omega} M_j \quad (j=1,3,5) \quad \text{on } B(x) \tag{3}$$

where $\phi_j^{(i)}$ is the j-th mode velocity potential of the inner region, K is the wave number ($=\omega^2/g$), ω is the angular frequency, U is the ship speed, N_j is the j-th mode normal vector, M_j is the j-th mode m-term for the slender ship theory, and $B(x)$ is the shape of the ship. The velocity potential of the inner region can be expressed as

$$\phi_j^{(i)}(x,y,z) = \varphi_j(y,z) + \frac{U}{i\omega}\hat{\varphi}_j(y,z) + C_j(x)\varphi_j^H(y,z) \tag{4}$$

$$\varphi_j^H(x,y,z) = \varphi_3(y,z) - \hat{\varphi}_3(y,z) \tag{5}$$

where φ_j and $\hat{\varphi}_j$ are special solutions that satisfy the first and second terms on the right hand side of Eq. (3), and $C_j(x)$ is the j-th mode unknown coefficient of homogeneous solution. The velocity potential of the outer region is expressed as

$$\begin{aligned}\phi_j^{(o)}(x,y,z) \approx\; & Q_j(x) G_{2D}(y,z) \\ & - \frac{1}{\pi}(1+Kz)\int_{-\infty}^{\infty} Q_j(\xi) f(x-\xi)\, d\xi\end{aligned} \tag{6}$$

where $G_{2D}(y,z)$ is the two-dimensional Green function, $f(x-\xi)$ is the three-dimensional effect term, and $Q_j(x)$ is the j-th mode unknown coefficient. In a matching stage, unknown coefficients, $C_j(x)$ and $Q_j(x)$, are solved in the integration equation. The solutions for two-dimensional problems obtained using the unified theory are similar to those obtained using the NSM. However, the matching procedure might have caused differences in the solutions obtained for the three-dimensional problems. Added resistances are computed as follows:

$$\begin{aligned}\frac{R_{AW}}{\rho g \zeta^2} =\; & \frac{1}{4\pi k_0}\left[-\int_{-\infty}^{k_1} + \int_{k_2}^{k_3} + \int_{k_4}^{\infty}\right] \\ & \times \left\{|H_C(k)|^2 + |H_S(k)|^2\right\} \\ & \times \frac{\kappa(k)\{k - k_0\cos\chi\}}{\sqrt{\kappa^2(k) - k^2}}\, dk\end{aligned} \tag{7}$$

where R_{AW} is the added resistance, ρ is the water density, g is the acceleration due to gravity, ζ is the wave amplitude, k_0 is the wave number of the incident wave, and $H_C(k)$ and $H_S(k)$ are Kochin functions that correspond to symmetric and asymmetric modes, respectively. $\kappa(k)$ is expressed as

$$K = \frac{\omega^2}{g}, \tau = \frac{U\omega}{g}, K_0 = \frac{g}{U^2} \tag{8}$$

$$\kappa(k) = \frac{1}{g}(\omega + kU)^2 = K + 2k\tau + \frac{k^2}{K_0} \tag{9}$$

where ω is the circular frequency, τ is the Hanaoka parameter, and U is the speed of the ship. k_1, k_2, k_3, and k_4 are wave numbers of k_1 and k_2 wave systems and are expressed by the following equations:

$$\left.\begin{matrix} k_3 \\ k_4 \end{matrix}\right\} = \frac{K_0}{2}\left(1 - 2\tau \mp \sqrt{1-4\tau}\right)$$
$$\left.\begin{matrix} k_1 \\ k_2 \end{matrix}\right\} = -\frac{K_0}{2}\left(1 + 2\tau \pm \sqrt{1+4\tau}\right) \tag{10}$$

The two Kochin functions are obtained from the EUT and NSM. In the EUT, the Kochin function is obtained from source strength and the doublet in the outer problem.

In case of asymptotic low wavelengths, the wave does not contribute to the added resistance due to ship motions. Most of the wave reflects from the ship bow as a diffraction component. This is sensible in blunt-type ships. Faltinsen et al. (1980) derived the estimation formula as

$$\frac{R_{AW}^A}{\rho g \zeta^2} = \frac{1}{2}\int_L \left\{\begin{matrix} \sin^2(\theta+\alpha) \\ + \dfrac{2\omega_0 U}{g}\left(1 - \cos\theta\cos(\theta+\alpha)\right) \end{matrix}\right\} n_1 d\ell \tag{11}$$

where α is the relative wave direction, θ is the angle between the longitudinal axis and tangential axis of the hull of the ship, and n1 is the normal vector in the longitudinal direction. Okusu (1986) developed Eq. (11) to the following formulae:

$$\frac{R_{AW}^A}{r g z^2} = \frac{1}{2}\left(1 + \frac{2 w_0 U}{g}\right)\int_{-B/2}^{B/2} \frac{K_{1n}}{k_0} n_1 \, dy \tag{12}$$

$$K_{1n} = \frac{\left(\omega_e - k_0 U \cos^2\theta\right)^2}{g}\sin(\phi - \theta) \tag{13}$$

where ω_0 is the circular frequency of the incident wave, and ω_e is the encounter frequency. The speed loss is evaluated using each formula and is compared in this paper.

2.2 Propulsion and machinery models

If the speed loss is numerically simulated, then modeling of propulsion and machinery models is important. There are many studies that consider hydrodynamic propulsion forces, and it is possible to accurately evaluate parameters such as ship resistance by using computational fluid dynamics (CFD). However, high-performance computers are necessary for CFD analysis. In this study, empirical formulae are combined with seakeeping models. The thrust and torque of propellers are approximated as polynomials, which are based on model tests of B-series propellers.

$$T(n,U) = r n^2 D^4 K_T \tag{14}$$

$$\begin{aligned} Q_B(n,U) &= r n^2 D^5 K_{QB} \\ &= r n^2 D^5 h_R K_Q \end{aligned} \tag{15}$$

where T is the thrust of the propeller, Q_B is the torque around the propeller, ρ is the water density, n is the speed of the propeller speed (rps), D is the diameter of the propeller, K_T is the thrust coefficient, K_Q is the torque coefficient, and K_{QB} is the torque coefficient around the propeller. K_T and K_Q are modeled as the summation of 39 and 47 terms of polynomials (Oosterveld, 1975), and they are approximated to second-order polynomials as follows:

$$\eta(t) = \sum_{i=1}^{5000} \sqrt{2S(\omega_i)\Delta\omega_i} \cos(\omega_i t - \varepsilon_i) \tag{16}$$

$$\begin{aligned} K_Q &= \sum_{i=1}^{47} CP_i J^{s_i} \left(\frac{P}{D}\right)^{t_i} \left(\frac{A_E}{A_D}\right)^{u_i} I^{v_i} \\ &= dJ^2 + eJ + f \end{aligned} \tag{17}$$

$$K_{QB} = d'J^2 + e'J + f' \tag{18}$$

where CP, s, t, u, and v are coefficients obtained from model experiments of B-series propeller, P is the pitch of the propeller, A_E/A_O is the ratio of propeller areas, I is the number of wings of the propeller, and J is an advanced constant that is expressed using the following equation:

$$J = \frac{(1- w)U}{nD} \tag{19}$$

where w is the wave fraction, U is the speed of the ship (m/s). The coefficients *a, b, c, d, e, f, d', e'*, and *f'* are obtained for each ship. The speed of the ship is determined using the relation between propulsion and resistance. The propulsion forces consist of deducted thrust of waves, and the resistance force is divided into wave resistance in still water and added resistance.

$$\begin{aligned} h(n,U) &= Q_B(n,U) - Q_E(n_E) = 0 \\ g(n,U) &= R_T(U,\omega,\chi) - T_T(U,\omega,\chi) = 0 \end{aligned} \tag{20}$$

$$T_T(U,\omega,\chi)=\beta(U,\omega,\phi)(1-t)T(U) \tag{21}$$

where β is the thrust deduction factor caused by relative vertical motions in waves, t is the thrust deduction factor due to mechanical loss, R_{SW} is the resistance in still water, and R_{AW} is the added resistance. When a ship sails in rough waves, the engine is usually controlled to prevent overload conditions. There are four types of engine controls–constant speed control, constant revolution control, constant torque control, and constant power control. As constant speed control is not realistic in rough waves, it is not considered in this study. The rest of the engine controls are modeled as follows (Naito and Kan, 1984):

(Constant revolution control)

$$\begin{aligned} h(U,S_W)&=2\pi n Q_B(n,U)-(1+S_W)P_0(U)=0 \\ g(U,S_W)&=R_T(U,\omega,\chi)-T_T(U,\omega,\chi)=0 \end{aligned} \tag{22}$$

(Constant torque control)

$$\begin{aligned} h(n,U)&=Q_B(n,U)-Q_E(n_E)=0 \\ g(n,U)&=R_T(U,\omega,\chi)-T_T(U,\omega,\chi)=0 \end{aligned} \tag{23}$$

(Constant power control)

$$\begin{aligned} h(n,U)&=2\pi n Q_B(n,U)-P_E(n_E)=0 \\ g(n,U)&=R_T(U,\omega,\chi)-T_T(U,\omega,\chi)=0 \end{aligned} \tag{24}$$

where Q_E is the engine torque, n_E is the engine revolution, P_0 is the power of the ship in still water, and P_E is the power of the engine. Simultaneous equations (22), (23), and (24) must be satisfied if these engine controls are used at sea. Wave states are irregular in an actual sea, and speed loss under irregular sea states must be considered. The longitudinal motion (speed of the ship) can be expressed as follows (Prpic-Orsic and Faltinsen, 2012):

$$(M+m_{11}(\omega_i))\frac{dU_i}{dt}=T_T(U_i,\omega_i,\chi)-R_T(U_i,\omega_i,\chi) \tag{25}$$

where M is the mass of the ship, and $m_{11}(\omega)$ is the added mass in surge mode. Damping force is relatively small in the longitudinal direction and is neglected in this study. In an irregular sea state, Eq. (18) must be solved in the time domain. As shown in Fig. 1, a wave series has different amplitudes and periods for each component. Based on Hsu’s assumption, added resistance (wave drift force) in irregular waves can be approximated as “a series of regular waves with different amplitudes and periods.” Each regular wave is a combination of two neighboring waves of half wavelength. The zero-up cross method is used to analyze wave series. Eq. (25) is numerically solved using the fourth Runge–Kutta method. Eqs. (22)–(24) are considered in each step of integration. Once the speed of the ship, U, is obtained, the thrust, torque, and power of the ship power can be determined easily. Fuel oil consumption, *FOC*, is also an important parameter that must be determined to evaluate speed loss in an actual sea.

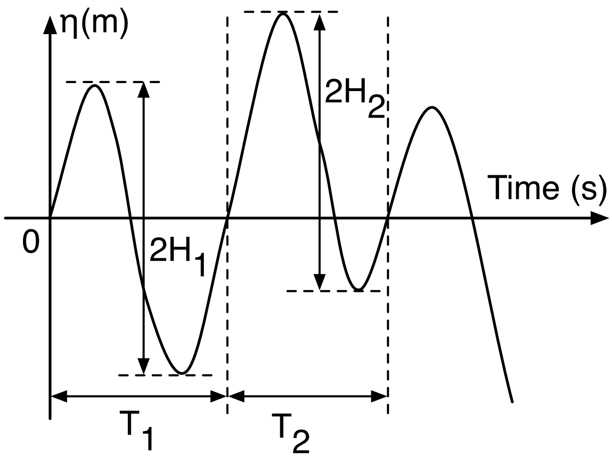

Fig.1 Approximation of wave drift force in irregular wave

FOC has a strong relation with engine power and can be estimated using the following equation:

$$FOC = q_1 P(n,U)^4 + q_2 P(n,U)^3 + q_3 P(n,U)^2 + q_4 P(n,U) \quad (26)$$

where P is the power of the ship power in waves, and q_1, q_2, q_3, and q_4 are coefficients of the polynomial obtained using the least square method. Thus, speed loss can be evaluated as the total performance of the ship in irregular waves.

3. RESULTS AND DISCUSSION

3.1 Condition of numerical simulation

In this study, speed loss is simulated for a 20,000DWT-class bulk carrier. The dimensions of the carrier are listed in Table 1.

The ship that is considered in this study has sailed all over the world over several years, including the Pacific Ocean, the Atlantic Ocean, the Indian Ocean, and the Southern Hemisphere.

Table 1 Dimensions of bulk carrier

Length, between perpendicular	160.4 m
Breadth	27.2 m
Draft (full loaded)	9.82 m
Metacenter height	3.11 m
Displacement	34,757 t
Navigation Speed	14 knots

The numerical model estimates several parameters such as the speed of a ship, shaft revolution (RPM), power of a ship, and fuel oil consumption. Added resistance for different patterns is computed using the following equations:

(a) Eq. (7) with EUT

(b) Eq. (7) with NSM

(c) Eq. (7) with NSM and Eq. (11) by Faltinsen

(d) Eq. (7) with NSM and Eq. (12) by Okusu

Eqs. (11) and (12) are additionally considered for high-frequency regions in patterns (c) and (d).

3.2 Condition of incident waves

Two types of irregular waves are considered to discuss speed loss. They are generated by the ITTC spectrum as

$$S(\omega)=\frac{A}{\omega^5}\exp\left(-\frac{B}{\omega^4}\right) \tag{27}$$

where *A* and *B* are coefficients and are expressed as follows:

$$A=173\frac{H_{1/3}^2}{T_{01}^4},\, B=\frac{691}{T_{01}^4} \tag{28}$$

where $H_{1/3}$ is the significant wave height, and T_{01} is the mean wave period. A wave series is generated using the following relation:

$$\eta(t)=\sum_{i=1}^{5000}\sqrt{S(\omega_i)\Delta\omega_i}\cos(\omega_i t-\varepsilon_i) \tag{29}$$

The time length of a wave series is set to 3 hours to maintain the stationary wave series state. Wave conditions (wave height, wave period, and relative wave direction) for numerical simulations are listed in Table 2. The wave direction is set as head sea.

Table 2 Conditions of irregular waves

Wave	$H_{1/3}$	T_{01}	Direction
1	5.0 m	11.2 s	180 deg.
2	3.0 m	6.0 s	180 deg.

3.3 Simulated result of ship speed

Table 3 enumerates the conditions assumed for the numerical simulation of the 20,000DWT-class bulk carrier. A fully loaded condition is commonly defined in this study. Fig. 2 shows the simulated time series of the speed of the ship for cases A-1, A-2, and A-3.

Table 3 Enumeration of numerical simulation

Case	Wave	Engine Control	Loaded Condition
A-1	1	Constant RPM	Fully loaded
A-2	1	Constant Torque	Fully loaded
A-3	1	Constant Power	Fully loaded
B-1	2	Constant RPM	Fully loaded
B-2	2	Constant Torque	Fully loaded
B-3	2	Constant Power	Fully loaded

As shown in the figure, a significant speed loss occurs at constant RPM, constant torque, and constant power controls. There is no significant difference between the simulated results obtained using the EUT, NSM, and NSM with asymptotic formulae. The reason behind the observation is that the mean wave period (11.2 seconds) is relatively longer than typical wind waves. This implicates that added resistance in short wavelengths does not contribute significantly to this wave condition.

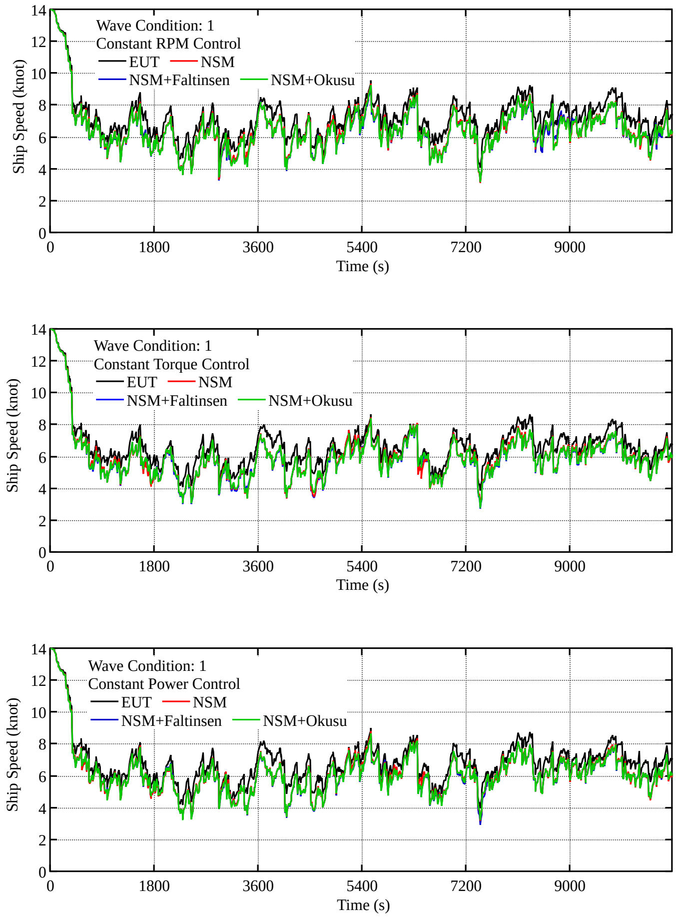

Fig. 2 Simulated time series of the speed of the ship (Case A-1, A-2, and A-3)

Fig. 3 shows the simulated time series of the speed of the ship for cases B-1, B-2, and B-3.

As shown in Fig. 3, it is obvious that NSM underestimates the magnitude of speed loss for wave condition 2. Speed loss can be evaluated if asymptotic formulae are considered. Furthermore, these results show that the EUT can evaluate speed loss in shorter wave periods of about 6 seconds.

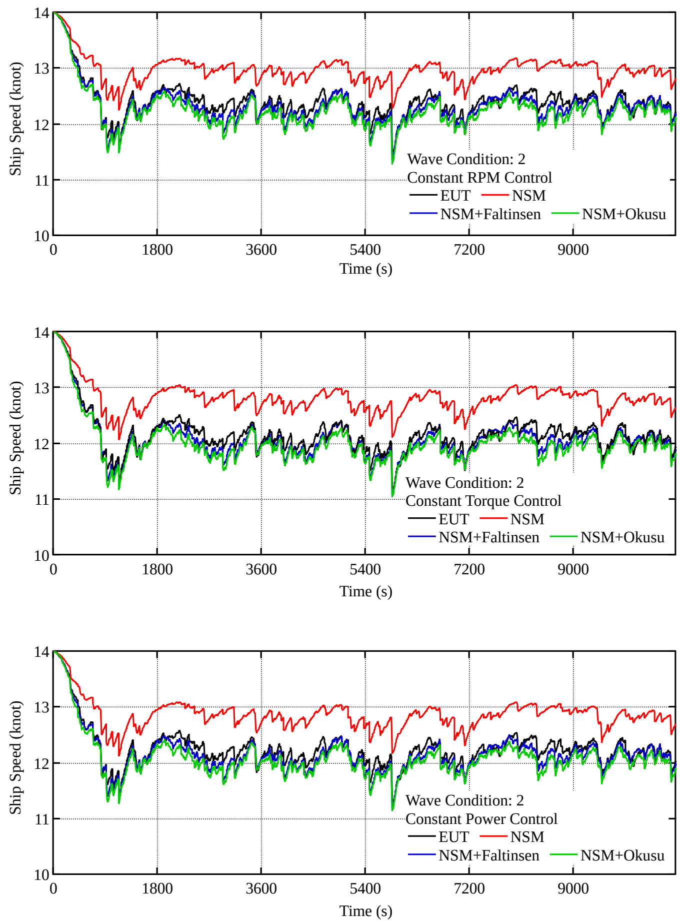

Fig. 3 Simulated time series of the speed of the ship (Case B-1, B-2, and B-3)

EUT evaluation of speed loss in all frequencies is validated. In contrast, asymptotic formulae must be considered if NSM is used. Fig. 4 shows the averaged values of the speed of the ship, revolution (RPM), power, and estimated fuel oil consumption per hour for each engine control and added resistance formula in cases A-1, A-2, and A-3.

The averaged peed of 0.5 knots obtained using the EUT is higher than those obtained using other simulation methods. However, there are few differences among methods in the revolution, power and fuel oil consumption. These four factors have the minimum values under constant torque control. It is numerated that constant RPM control results in higher power and fuel oil consumption, although the speed of the ship is only 0.5 knots. The fuel oil consumption is 100–200 liters per hour under constant torque or constant power controls. Fig. 5 shows the averaged values of the speed of the ship, revolution (RPM), power, and estimated fuel oil consumption per hour for each engine control and added resistance formula for cases B-1, B-2, and B-3.

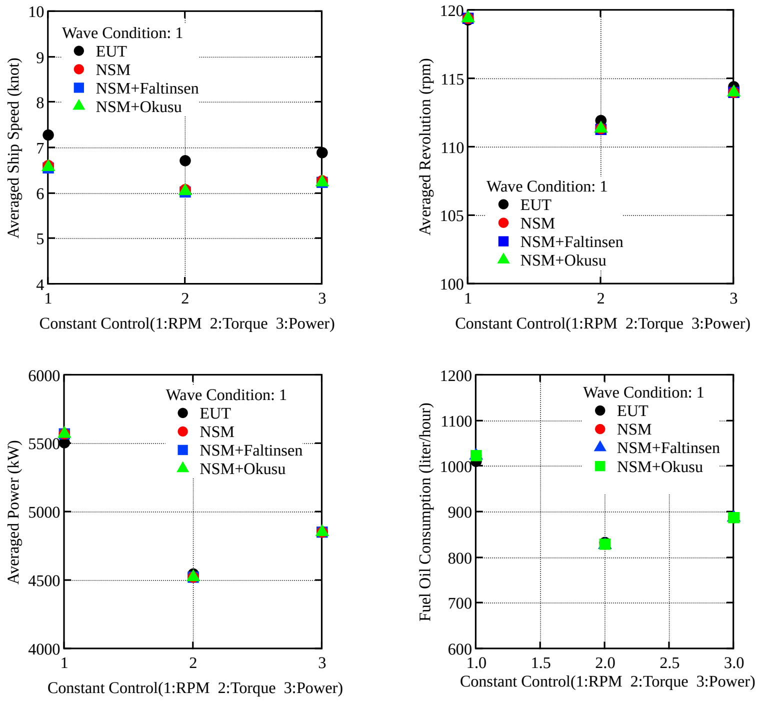

Fig. 4 Averaged values of speed, revolution, power, and estimated fuel oil consumption per hour (Case A-1, A-2, and A-3)

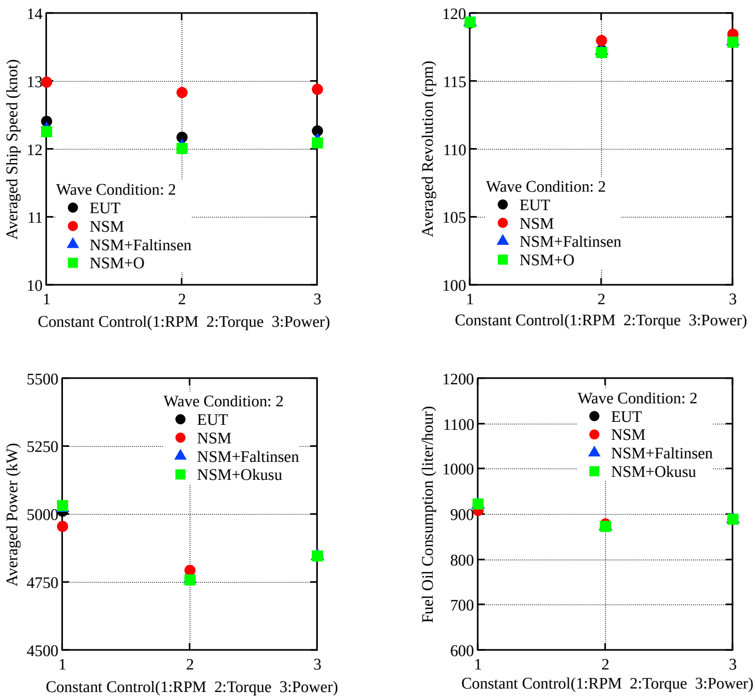

Fig. 5 Averaged values of speed, revolution, power, and estimated fuel oil consumption per hour (Case B-1, B-2, and B-3)

As shown in Fig. 3, if NSM is used, the speed of the ship is 1 knot for all engine controls. The speed loss is approximately 1–2 knots, which is smaller than that under wave condition 1. The difference among different engine controls is less than that under wave condition 1. In the constant RPM control, the fuel oil consumption is approximately 900 liters/hour, which is 100 liters less than that under wave condition 1. The fuel oil consumption under constant power control is approximately 900 liters per hour in both wave conditions 1 and 2. As fuel oil consumption depends on engine power, fuel oil consumption seems to be similar even if wave conditions are varied. It is necessary to compare the fuel oil consumption during a journey across oceans through weather routing simulations in the future. Machinery models must be improved to accurately consider the effects of fuel injection and combustion.

4. CONCLUSIONS

In this study, difference in speed loss in a seaway is evaluated using different simulation methods. Speed loss is simulated for a 20,000DWT-class bulk carrier, and the conclusions are summarized as follows:

(1) The difference in the speed of the ship is not significant among methods (a)–(d) if the mean wave period is longer (11.2 seconds). These waves might cause radiation forces beside diffraction forces. Thus, the influence of asymptotic added resistance in swells is very small.

(2) Speed loss is underestimated by the NSM (method (b)) if mean wave periods are shorter (6 seconds). The speed of the ship is less for 1 knot in NSM than that by other methods. The speed loss can be estimated accurately if asymptotic formulae are included.

(3) There is little difference in the values of revolution, power, and fuel oil consumption in each seakeeping model. These parameters are mostly influenced by engine control methods.

(4) This study verifies that the constant RPM control requires higher power and fuel oil consumption, although the speed of the ship cannot be maintained. This study makes it possible to determine the optimal operation conditions in rough seas.

(5) Authors have been accumulating an onboard database regarding the performance of the 20,000DWT-class bulk carrier for years. A detailed validation will be carried out for these data by using the simulation models.

(6) Current machinery models cannot consider the influence of injection and combustion of fuel oil. This point will be improved soon. In addition, the actual control patterns of marine engines must be investigated for a more accurate model.

(7) Voluntary speed loss is not considered in this study. However, it is necessary to include this point for optimal ship routing.

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