The Specialist Committee on Manoeuvring in Waves

Final Report and Recommendations to the 29th ITTC

1. INTRODUCTION

1.1 Membership

The 29th ITTC Specialist Committee on Manoeuvring in Waves consisted of:

- Prof. Hironori Yasukawa (Chairman). Hiroshima University, Japan
- Dr. Manasés Tello Ruíz (Secretary). HSVA, Germany, formerly, Ghent University (UGent) and Flanders Research Hydraulics (FRH), Belgium
- Dr. Evgeni Milanov. BSHC, Bulgaria
- Dr. Young-Jae Sung. Hyundai Heavy Industries, Korea
- Dr. Yeongyu Kim. Korea Research Institute of Ships & Ocean Engineering (KIOST), Korea
- Dr. Xiechong Gu. Shanghai Jiao Tong University, China
- Prof. Wenyang Duan. Harbin Engineering University, China
- Dr. Marc Steinwand. SVA Potsdam, Germany (left in 2019)

1.2 Meetings

The committee met four times:

1. BSHC, Varna, Bulgaria, February 2018
2. Shanghai Jiao Tong University, China, October 2018
3. Ghent University and Flanders Research Hydraulics, Antwerp, Belgium, May 2019
4. Hiroshima University, Japan, January 2020

1.3 Tasks and Report Structure

The following lists the tasks given to the 29th ITTC, the Specialist Committee on Manoeuvring in Waves (SC-MW). Originally, we planned to add the results of the SIMMAN workshop, but we did not mention it because it was postponed due to the influence of the Corona-virus.

1. Define the overall framework for what manoeuvring in waves means. (*section 2*)
2. Present the state of the art based on a comprehensive literature review. (*section 3*)
3. Create a guideline for benchmark tests on manoeuvring in waves. Consideration should be given to the generation of data for the validation of numerical tools. (*Publication of the new guideline was postponed*)
4. Investigate the methodology needed to combine experimental tests and numerical tools. (*section 3*)
5. Investigate new manoeuvres to assess minimum power requirements (e.g. return to head waves). (*section 4*)
6. Address the issues brought about from IMO-MEPC71 and following meetings concerning the minimum power requirements, including issues on manoeuvrability under adverse weather. (*section 4*)
7. Validate the Level 2 – Simplified Assessment Method of the 2013 Interim Guidelines (MEPC.1/Circ.850). (*section 4*)
8. Liaise with IMO and/or IACS to address manoeuvring in waves. (*section 5*)
9. Liaise with the Manoeuvring Committee, the Seakeeping Committee and the Stability in Waves committee. (*section 5*)
10. Establish a mathematical model for manoeuvring in waves. (*section 3*)

2. GENERAL

2.1 Overview on the Ship Manoeuvring in Waves

Over the years the ship's manoeuvring qualities have been traditionally analysed, predicted and normalized for calm and deep water by means of 3DOF manoeuvres, assuming negligible influence of external sea conditions. However, the assumption of negligible external effects such as wind, current, shallow water and waves is not strict. Several studies have focused on the first three factors, because accounting for them fits well into the time-domain studies of ship manoeuvrability.

In real navigation conditions, the two problems overlap. When performing manoeuvre in waves the wave induced ship motions interacts with the ship's manoeuvring motion, thus waves may substantially influence the hydrodynamic forces and thereby change the manoeuvring behaviour, and vice versa. The need to evaluate the manoeuvring behaviour in such scenarios leads to a necessity to combine the knowledge gained in the two separate approaches. This is not an easy task as even in the study of the ship's controllability in calm water the mathematical model in calm water is still in the horizontal plane. Note that research on manoeuvrability and seakeeping intersects in two fields, on one hand the frequency dependency of hydrodynamic coefficients and ship motions and on the other the fluid memory effects.

Regarding the frequency dependency, a number of works have been devoted to justify the use (in the task of predicting ship manoeuvrability in still water the PMM) of “slow motion derivatives” under the assumption of quasi-steady flow. This approach is still applied as a standard in PMM data analysis. Such assumption, however, is rather questionable because in a conventional towing tank, the length of run is quite short and the frequency of captive motion is quite high. The effect of past history of the motion in view of “slow motion derivatives” is assumed negligible. Some approaches have been used for separation of fluid memory effects by adequate PMM data post-processing.

Since the introduction of the Energy Efficiency Design Index (EEDI), serious concerns regarding the manoeuvrability of ships in waves have been brought to the forefront. IMO Marine Environment Protection Committee issued determined a “Minimum propulsion power to maintain manoeuvrability of ships in adverse conditions” (MEPC 232(65), 2013). The definitions of adverse conditions and minimum power line requirements in the document were stated. However, the question what means to “maintaining manoeuvrability” remains open. Research into the problem has been initiated in a number of centres, using experimental, numerical and hybrid approaches.

In the frame of SHOPERA project development, added resistance and ship manoeuvrability have been studied experimentally and numerically focusing on second order wave forces (Shigunov and Papanikolaou, 2015). A subset of benchmark data was established relating to added resistance in regular an irregular waves; drift forces, turning and zig-zag manoeuvres in regular waves at 4 and 6 knots approach speed. Particular attention was paid to developments in the area of the problem of additional criteria for manoeuvrability in waves. In this regard, three critical scenarios were proposed for consideration, where the ship operation functional requirements, the practical criteria and the environmental conditions have been specified. As the ship may fall into each of the scenarios, it would be appropriate to identify the most critical one. In principle, the question of simplifying the set of requirements regarding the ship's manoeuvrability in waves is on the agenda.

In this context, SHOPERA and JASNAOE in 2016 submitted to IMO a coordinated proposal which contains unique description of adverse seas and one most critical scenario (“escape from coastal area”). During a meeting organized by ITTC Manoeuvring Committee workshop at LR in London, 2016 all aspects of ship manoeuvring in waves were discussed and opinion on the subject by participants have been presented. Due to the complexity of the problem there were more questions than answers. The corresponding panel discussion focused on five main topics: a) methods and procedures to work on; b) how to simplify and still be relevant; c) how to improve simulations; d) environmental or input conditions; e) manoeuvres to consider; f) general comments. Based on above, the discussion covers many issues of varying complexity and importance to considered research area.

On the basis of the so far considered requirements of the regulatory authorities and the results of previous studies, a general and preliminary definition of the sufficient ship manoeuvring in adverse sea conditions we can accept: "The ship has ability to maintain certain advance speed and change and keep the course in most unfavourable waves and wind conditions".

2.2 Indices Representing Manoeuvring in Waves

The manoeuvrability of a ship in waves can be classified, as shown in Table 1, based on that in calm water. The major difference from calm water is that ships sailing straight in waves at an average constant ship speed generally require a check helm, which leads to a hull drift angle and a ship speed drop. The condition of the ship moving in waves is called "steady sailing condition" here.

The turning circle in the presence of waves does not become a circular trajectory as in the case of the calm water. During turning, the ship drifts to a different direction from the incident waves. Two indices representing the drift characteristics during turning, the drifting distance H_D and the drifting direction μ_D, are used (Ueno et al., 2003). Here, the successive ship positions in ψ=90°, 450°, 810°, etc. are numbered as 1, 2, 3, and so on, as shown in Figure 1. Then, H_{D1} and H_{D2} are defined as the distances of ship drifting from 1 to 2 and 2 to 3, respectively. Similarly, μ_{D1} and μ_{D2} are defined as angles of the ship drifting from 1 to 2 and 2 to 3, respectively (Hasnan et al., 2020).

Table 1: Indices representing manoeuvring in waves

	straight moving	small/medium rudder angle	large rudder angle
calm water	propulsive peformance	10/10 or 20/20 zig-zag maneuvers	35deg turning
waves	steady sailing performace		
indices in waves	check helm, drift angle, speed drop, etc.	overshoot angles	advance, tactical dia., drifting distance, drifting direction

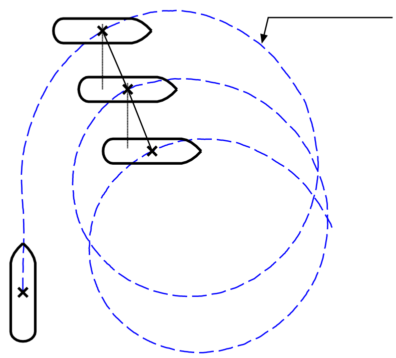

Figure 1: Definition of drifting distance H_D and drifting direction μ_D (Hasnan et al. 2020)

As manoeuvring in waves other than those shown in Table 1, it should be considered stopping in waves and crabbing in waves.

3. STATE-OF-THE ART OF PREDICTION METHODS OF SHIP MANOEUVRING IN WAVES

3.1 Experimental Methods

3.1.1 Free running tests in waves

General: Free running model tests are commonly used to investigate the manoeuvring of a ship directly. Their results can also be used as validation data for developing a computer simulation model. Traditionally, free running model tests are conducted in calm water condition, but those can also be conducted in waves.

Free running tests should be designed for the ship model to move autonomously. For measuring the position of the ship model, three methods are frequently used: a) a method of using an acoustic measurement equipment installed at the bottom of the tank (Hirano et al., 1980), (Ueno et al., 2003), b) a method in which a carriage in the tank automatically tracks and measures the position according to the movement of the ship (Yasukawa and Nakayama, 2009), and c) a method of measuring the ship position by an optical method (Yasukawa et al., 2015), (Kim et al., 2019a). Commonly, the inclination angle of the ship is measured using a gyro. Figure 2 shows a measurement system by an optical method (total station system) in KRISO, Ocean Engineering Basin.

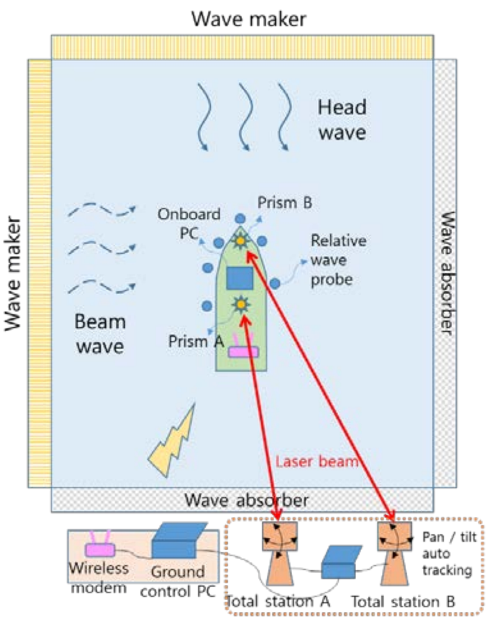

Figure 2: Measurement system for free-running tests in KRISO Ocean Engineering Basin (Kim et al., 2019a)

Most manoeuvring tests start from a straight course condition with as steady as possible values of heading, speed, rpm and rudder angle. Speed trial tests should be carried out in order to find the propeller rpm corresponding to the desired test speed. Methods for accelerating the ship model to the target speed are summarized as follows:

- A propeller revolution scheduling system which is usually used at acceleration phase to reduce acceleration time and distance. But relatively long distance is required. (Kim et al., 2019a)
- A catapult system (Yasukawa et al., 2015) (Hasnan et al., 2020)
- A carriage releasing system: the ship model is released after acceleration phase. (Yasukawa and Nakayama, 2009) (Sanada et al., 2013)

Free running tests in regular waves:

In 1980, Hirano et al. (1980) conducted a free running test in regular waves using a self-propelled Ro-Ro ship model to investigate the effects of waves on the turning trajectory. The drifting behaviour during turning in regular waves was studied. Ueno et al. (2003) performed free running tests for turning, zig-zag, and stopping manoeuvres in regular waves using a VLCC tanker model. It was shown that the drifting direction of a ship was different from the incoming wave direction. In addition, a large drift of the ship during turning was observed for shorter wavelengths.

Yasukawa (2006a), Yasukawa (2008), and Yasukawa and Nakayama (2009) conducted free running tests for turning, zig-zag, and stopping manoeuvres using the S-175 container ship model. Lee et al. (2009) conducted turning and zig-zag manoeuvre tests in regular waves using a VLCC model to capture the wave height effect. However, details such as wave-length were not revealed.

Sanada et al. (2013) performed turning tests for the ONR Tumblehome in calm water and regular waves and presented time histories of 6-

DOF motions during turning in waves. Moreover, Sanada et al. (2019) performed repeat tests of turning and zig-zag manoeuvres for the same ONR Tumblehome in regular waves and discussed the effect of ship speed and wave-length on manoeuvring with the measured accuracy.

Sprenger et al. (2017) performed turning and zig-zag manoeuvre tests for a DTC container ship and KVLCC2 tanker models in regular waves with variations in wave directions, wave-length, etc. The obtained data was mainly used to validate the calculation method for manoeuvring in waves.

Kim et al. (2019a) carried out the turning tests in regular waves using KVLCC2 model. Figure 3 shows the turning trajectories with variations of the ratio of wave-length of ship length (λ/L). The rudder angle was 35°. While turning, the ship model drifts in the direction near the steering point. The drifting distance becomes larger at shorter wave-lengths, and the angle drifted obliquely to the incident wave direction becomes smaller at shorter wave-lengths. Such characteristics are similar to the results of the S-175 model (Yasukawa and Nakayama, 2009).

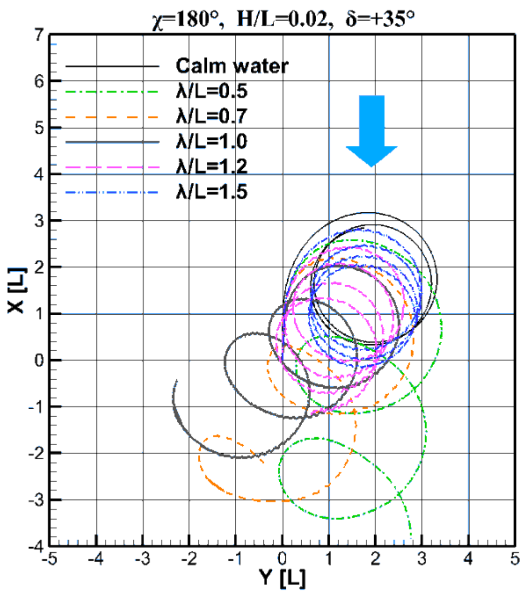

Figure 3: Turning trajectories in regular head waves with variations of λ/L for KVLCC2 (Kim et al., 2019a)

Free running test results in irregular waves:

Yasukawa et al. (2015) conducted free-running model tests using a KVLCC2 model in short-crested irregular waves. Turning tests, and 10/10 zig-zag manoeuvre tests were carried out to obtain the validation data of the manoeuvring simulation method in irregular waves.

Hasnan et al. (2020) conducted the turning tests in short-crested irregular waves using two ship models of KVLCC2 tanker and KCS container ship. The tests were performed in head waves at the time of approaching with the significant wave height 4.5 m for KVLCC2 and 3.0 m for KCS at full-scale. With a decrease in the approach speed of the ships sailing in the same wave condition, advance A_D decreases but tactical diameter D_T does not change significantly. With a decrease in the approach speed, both drifting distance H_D and drifting direction μ_D increase significantly, and the tendency of the ship drifting to the rudder execution point in space becomes remarkable.

Kim et al. (2019b) conducted turning tests with various seeds using KVLCC2 model in long-crested irregular waves for different propeller revolutions (n_P). Figure 4 shows comparison of turning trajectories in irregular waves. The trajectories at n_P=8.2 rps (corresponding to 7.0 kn in calm water) are significantly different with changing the seeds of the irregular waves. At n_P=4.0 rps (corresponding to 4.0 kn in calm water), the ship cannot turn in waves. It is necessary to study more for better understanding of the ship behaviour in irregular waves.

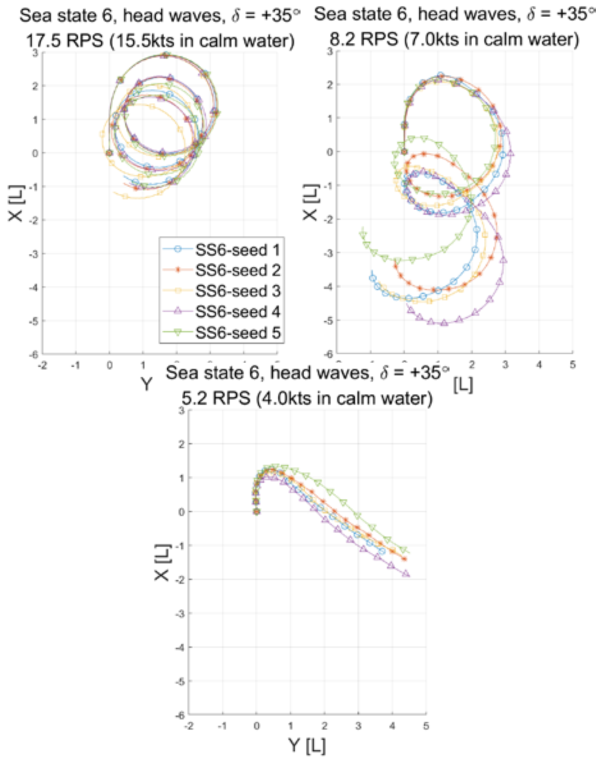

Figure 4: Comparison of turning trajectories in irregular waves for KVLCC2 model (Kim et al.,2019b)

<u>Free running test results in wind and waves:</u>

Fujiwara, T. et al. (2008) carried out free-running model test on a large container ship under heavy wind and regular wavs at the 400m towing tank, NMRI, Japan. Averaged navigation conditions and time fluctuations of the ship speed, hull drift angle, rudder angle, ship motions and propeller thrust etc. were captured in the experiments.

3.1.2 Captive model tests in waves

<u>General:</u> Captive model tests in waves are performed to verify the forces and moments induced by waves. Up to date, the mathematical model for interpreting manoeuvring performance in waves is largely divided into two problems, a) a mathematical model that interprets the force induced by waves by linear superposition on the calm water manoeuvring equations of motion, and b) a mathematical model that incorporates all elements of seakeeping and manoeuvring. In the former case, the method of analysis through model testing and other verification is well set-up for the calm water manoeuvring analysis model, but in the latter case, interpretation and verification methods through model testing are not well set-up yet.

In case of static straight or oblique tests, in which the incident direction of waves is fixed, both 1st and 2nd order wave forces can be obtained by fitting and averaging. But in case of dynamic tests such as PMM, wave incident direction is changed, so it is difficult to obtain 1st and 2nd order wave forces by fitting and averaging. Even at steady circular motion test (CMT), it is impossible to obtain test results with the same wave condition such as wave direction and wave encounter frequency, even more, those are not affected by the wave

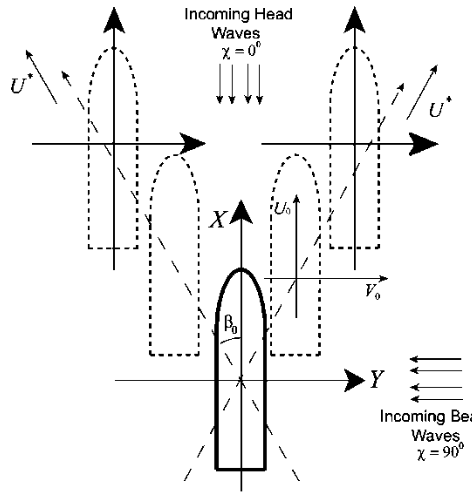

Figure 5: Schematic diagram of horizontal motions of ship in oblique tests in waves (Yasukawa and Adnun, 2006)

produced by the model ship. Therefore, only static straight or oblique tests are performed. In the oblique tests in waves, the wave encounter angle to the incident waves has to be kept with the drift angle as shown in Figure 5.

<u>Ship motions and measured forces in waves:</u>

Yasukawa and Adnan (2006), and Yasukawa et al. (2010) measured the ship motions in regular waves for an obliquely moving ship. The experiments were carried out for S-175 container ship in head waves and beam waves. Figure 6 shows the amplitude of the wave-induced motions (sway, roll, heave and pitch) in regular head waves. Due to the effect of the hull drift angle (β_0), the lateral motions such as sway, roll and yaw are induced even in pure heading waves. Their amplitudes become larger with increase of absolute value of the hull drift angle. On the other hands, the influence of the hull drift angle on the motions of surge, heave and pitch is not remarkable.

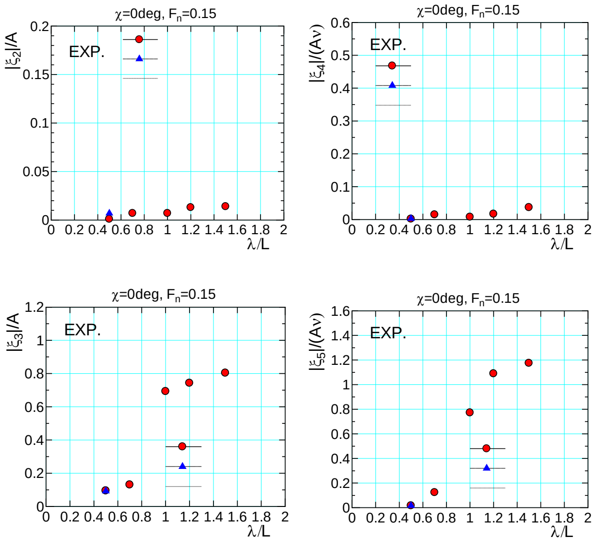

Figure 6: Amplitude of wave-induced motions (sway, roll, heave and pitch) for an obliquely moving ship in regular head waves (Yasukawa and Adnun, 2006)

Choi et al. (2019) presented test results of average value of lateral force and yaw moment acting on a KCS model obtained at oblique tests in regular waves. The ship model was fixed in the tests. Therefore, the measured forces represent the sum of hydrodynamic forces acting on the obliquely moving ship and mean wave drift forces (only diffraction component).

For a limited combination of wave amplitudes and wave lengths, drift angles and in shallow water conditions tests with a scaled model of ULCS have been investigated in Tello Ruíz et al. (2019). In Figure 7 a sample of their findings for the mean forces obtained for fully captive model tests are shown. The influence of waves at lower speeds were found to be significant important (see vertical offset of the square markers in Figure 7). For intermediate to larger speeds the wave influence was found to be less relevant for all tested wave lengths. Similar observation were found in following waves (see Figure 8)

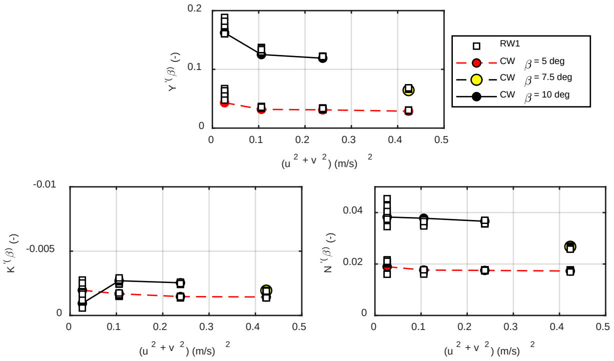

Figure 7: Mean sway force (top), roll moment (bottom left) and yaw moment (bottom right) in calm water (CW) and in head waves at T_M=13.1m, 50% UKC, ζ_a=1m (RW1) and at three drift angles, model fully captive. Results are plotted as function of V^2 only (from 0.2 to 0.8 $/Lpp$) (Tello Ruíz et al. 2019a,b)

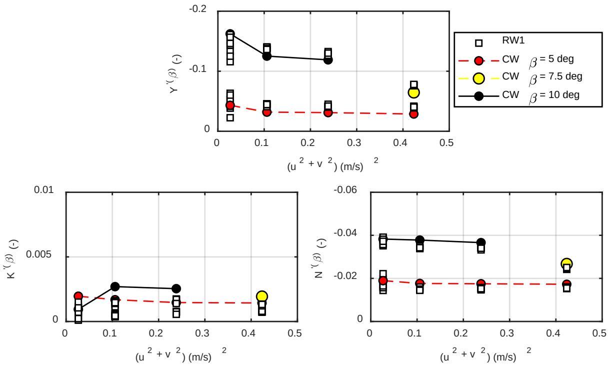

Figure 8: Mean sway force (top), roll moment (bottom left) and yaw moment (bottom right) in calm water (CW) and in following waves at T_M=13.1m, 50% UKC, ζ_a=1m (RW1) and at three drift angles, model fully captive. Results are plotted as function of V^2 only (from 0.2 to 0.8 $/Lpp$) (Tello Ruíz et al. 2019a,b)

<u>Mean wave drift forces:</u>

Ueno at al. (2001) measured mean wave drift forces and moment in turning motion. The wave encounter angle changes gradually during turning motion in the tests.

Yasukawa and Adnan (2006) measured the mean wave drift forces and moment in regular waves for an obliquely moving ship. Figure 9 shows added resistance, mean lateral force, and

were incorporated in the maneuvering equation of motion. Zhang and Zou (2016), Zhang et al. (2017), and Lee and Kim (2020) extended this approach by using double-body linearization, and the modeling for vortex flows that may occur at the end of the hull was introduced for the analysis of the double-body flow. Lee et al. (2020) extended the method to consider the weakly non-linear effect induced by hull geometry on the ship maneuvering in regular waves, but this effect was not significant.

Piro et al. (2020) proposed a hybrid method that combines a Reynolds-averaged Navier-Stokes (RANS) solver with a potential flow boundary-element method (BEM). The low-frequency manoeuvring problem mostly handled with the RANS method, and the relatively high-frequency seakeeping problem with the BEM. This method was applied for predicting the ship turning of KCS model in calm water and waves.

3.2.4 Unified methods

This method aims to propose a more general formulation for the ship’s hydrodynamic problem. Thus, disregarding the assumptions, partially or totally, taken for the independent analysis of manoeuvring in calm water and seakeeping. In this manner, avoiding the same computation of the same hydrodynamic problem twice, for instance, added masses in seakeeping codes and acceleration derivatives in manoeuvring models.

As the manoeuvring is a time domain problem, all phenomena is aimed to be represented in the time domain, thus avoiding (wherever possible) frequency domain computations. Up to date works using this method have mainly covered the body reaction forces (the radiation problem), e.g. Ankudinov (1983), Bailey et al. (1998). Other approaches attempt to model the entire problem directly in the time domain, e.g. Subramanian and Beck (2015).

All works considering the unified method differ in the selection of the mathematical model to account for the manoeuvring forces due to viscous, cross flow effects, and lift effects. But they all agree treating the potential contribution apart and expressed in the time domain. Note, however, that wave exciting forces and moments (first and second order) are mostly computed by the sum of the components obtained in the frequency domain and over the mean wetted surface. At most, because of its simplicity, only Froude-Krylov forces have been incorporated taking into account the real time variation of the wetted surface.

Ankudinov (1983) used this method to predict the ship response in irregular waves. In his work the radiation problem is model using memory terms proposed by Cummins (1962). Due to computational limits, the kernel functions were simplified by using higher order differential equation with constant coefficients. Exciting wave forces were also intended to be evaluated by convolution integrals for the first order and second order, but resulted in larger computing times, they were model instead by the sum of the frequency components.

In Bailey et al. (1998) the unification of the fluid phenomena was also extensively discussed for the body reaction forces. In addition, they introduced corrections to the kernel functions in order to account for viscous effects. First order wave forces were also computed by convolution integrals. They work, however, did not extend to incorporate second order wave forces, and only considered linear manoeuvring forces given by the velocities and acceleration derivatives.

Other approaches such as the works of McCreight (1986), Lee (2000), Nishimura and Hirayama (2003), Ayaz, et al. (2006), Sutulo and Guedes Soares (2006) , Yen et al. (2010), Araki et al. (2011), Subramanian and Beck (2015) and Tello Ruíz (2018) fall into the classification of unified methods. Most of the above works avoid to solve the convolution integral problem by directly computing the radiation each time step, increasing considerably computation times.

Works on this method have enjoyed less attention that the simplify method of considering second order wave forces only. Some results following the works on Ankudinov (1983) and Subramanian and Beck (2015) are presented in Figure 13 and Figure 14, respectively.

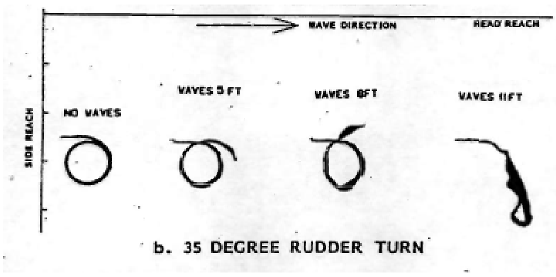

Figure 13: Turning circle trajectory of Mariner in calm water and in different irregular waves, Ankudinov (1983)

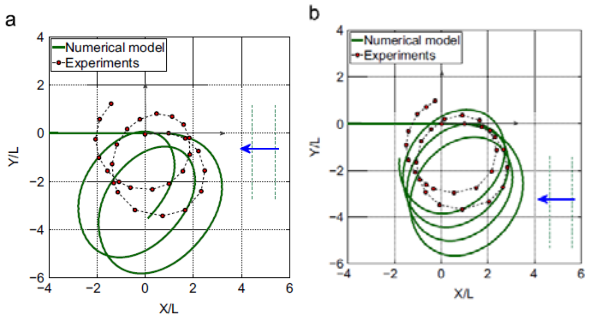

Figure 14: Turning circle trajectory of S-175 in waves. Rudder angle of -35 deg (starboard turns) (a) at λ/L of 1, in head seas and H/λ of 1/50. (b) at λ/L of 1:2, in head seas and H/λ of 1/60. (Subramanian and Beck, 2015)

3.2.5 CFD based direct simulation methods

Direct simulation using computational fluid dynamics (CFD) is the most promising method which can solve specific local flow details around the hull and its appendages and then provide a better understanding of the hydrodynamic problem of ship manoeuvring. Most of CFD studies on ship manoeuvre solves the Reynolds-Averaged-Navier-Stokes (RANS) equations for unsteady turbulent flows around complex geometries. For free running ship models, the propeller body force model or the sliding mesh method and most the overset grid method coupled with full 6-DoF motion are used. However, a precise simulation of ship manoeuvring in waves has to consider large-amplitude ship motions with more violent free surface deformation and more notable hull-propeller-rudder interactions, which is more difficult than the simulation in calm water.

Carrica et al. (2012) performed numerical simulations of ship manoeuvring in waves by using a simplified body force propeller model and applied overset grid to handle the ship motions and rudder steering. It is found that the main discrepancy between the CFD and experiments can possibly be tracked to the simplistic propeller model.

Shen and Korpus (2015) used dynamic overset grid technique and performed simulations of free running ship in head and quartering waves under course keeping control.

Wang et al. (2016, 2018a, 2018b) using naoe-FOAM-SJTU to simulate the free running course-keeping problem， zigzag manoeuvre and turning circle manoeuvre under various wave conditions for a fully appended twin-screw ship (ONR Tumblehome). The trajectory and main parameters agree well with the experiment, which show that the RANS dynamic overset grid is a reliable approach to directly simulate of such ship manoeuvre in waves. Figure 15 shows the local grid distribution for CFD simulations. Figures 16 and 17 shows free-surface elevation and vorticial structures around ship hull during turning in waves, respectively.

The above research shows the capability of CFD approach in directly simulating free running ship model in deep water regular waves. However, due to the high computational cost and even more long-time simulation requirement, direct CFD manoeuvring simulations in irregular waves, certainly for manoeuvring in shallow water waves are still a changeling problem.

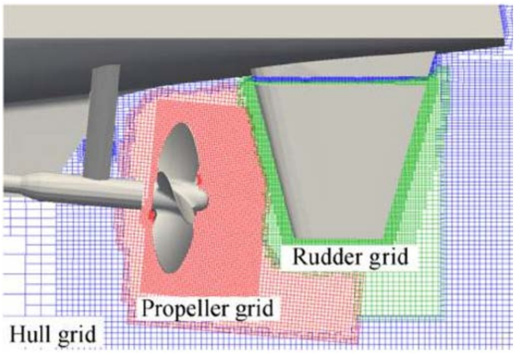

Figure 15: Local grid distribution for CFD simulations of ONR Tumblehome model ship (Wang et al. 2018b)

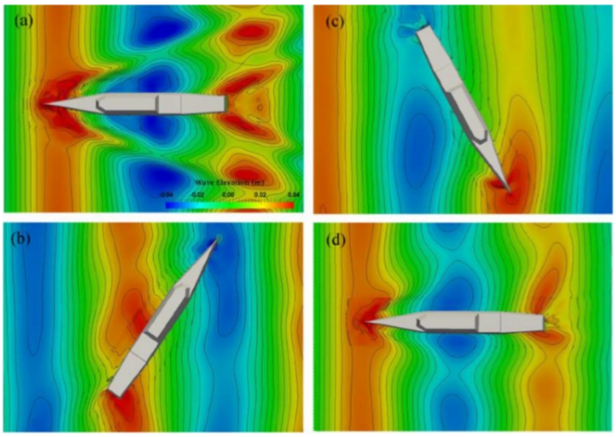

Figure 16: Free-surface elevation during turning in waves (a–d correspond to heading change of 0°, 120 °, 240 ° and 360 °, respectively) (Wang et al. 2018b)

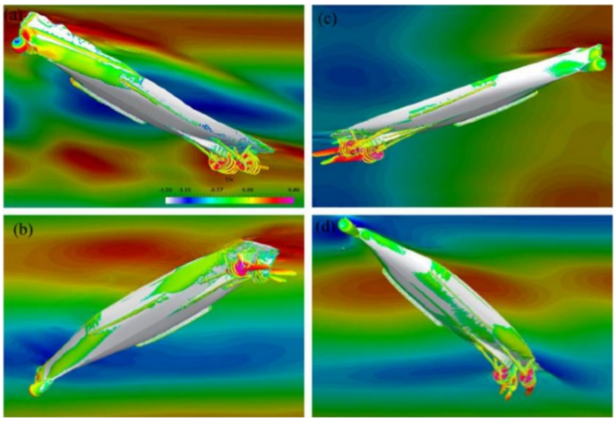

Figure 17: Vorticial structures around ship hull during turning in waves (a–d correspond to heading of 0 °, 120 °, 240 ° and 360 °, respectively) (Wang et al. 2018b)

3.3 Steady Sailing Performance and Manoeuvring Limit in Wind and Waves

For discussing the manoeuvring limit in adverse weather conditions, it is useful to evaluate the average steady sailing conditions (SSC), such as check helm, speed drop, hull drift angle, etc., of a ship moving straight in steady wind and waves. In addition, the dynamic stability, or course stability (CS), of the ship should be checked at the SSC. Both the SSC and the CS of ships under external disturbances are called the steady-sailing performance (SSP). For this analysis, the mean wave force methods mentioned in 3.2.2 are normally used.

The basic principle to conveniently obtain the SSP of the ships in steady wind and waves has already been presented by Eda (1968) and Ogawa (1969) as follows:

1. By setting acceleration, angular acceleration, and angular velocity to zero in the motion equations, the equilibria equations, that is, the balance with respect to forces and moments acting on the ship can be obtained. The check helm, speed drop, hull drift angle, and so on are obtained by solving the equilibria equations after setting the environmental condition.

2. The course stability of the ship under adverse conditions is adjudged by evaluating the eigenvalues of the linearized motion equations.

According to the aforementioned ideas, many studies have been performed on the SSP of ships under external disturbances. However, a remarkable difference can be observed in the existing studies.

The difference exists in the base model of the hydrodynamic forces acting on the manoeuvring ship, which can be classified as follows:

- Original MMG-model (Ogawa et al.,1977)

- Models expressing the hydrodynamic forces acting on the ship by the polynomial function with respect to ship motions and operation parameters such as rudder angle and propeller revolution (perturbation method), (Eda, 1968) (Ogawa, 1969)

- Simplified model based on the models mentioned above (Ishibashi, 1975) (Tanaka et al., 1980) (Martin, 1980)

In addition, the following points must be considered. There are two methods to solve the equilibria equations: one is an exact method (Hirano et al., 1984) (Kadomatsu et al., 1990), (Spyrou, 1995) (Naito and Takagishi, 1998) (Fujiwara et al., 2005) (Fujiwara et al., 2006) (Umeda et al., 2016), and the other is an approximate method. For solving the equilibria equations precisely, an iterative calculation is required, with the usage of a computer, since the equilibria equations are mathematically non-linear. In order to obtain the solution in a short time, it is useful to employ approximations, although the calculation accuracy becomes worse. In particular, the approximation that the ship speed is known has been often employed in several studies (Tanaka et al., 1980) (Asai, 1981) (Yasukawa et al., 2012).

Spyrou (1995) and Spyrou et al. (2007) presented a method to investigate the course stability of ships in steady wind by locally linearized stability analysis at the equilibria condition based on the Jacobean matrix expression that is obtained from the motion equations. This is a general method for solving the problem numerically regardless of the expression of the base hydrodynamic force model. Umeda et al. (2016) applied this method for investigating the manoeuvring limit of a full hull ship in wind and waves based on the low speed hydrodynamic force model presented by Yoshimura et al. (1988) and Yoshimura et al. (2009).

Yasukawa and Sakuno (2020) presented a method for conveniently obtaining the SSP under external disturbances in deep and shallow water based on `4D MMG method' (Yasukawa et al., 2019). Yasukawa (2020) extended the method to the SSP problem for a ship moving in a shallow channel. Figure 18 shows the results of the SSCs, including the longitudinal ship velocity component u_0, the check helm δ_0 and the hull drift angle β_0 at the average wave period T_P=10s for a pure car carrier (PCC) with a ship length of 180 m as calculation examples. The horizontal axis represents the absolute wind direction θ_W (wave direction χ is the same).

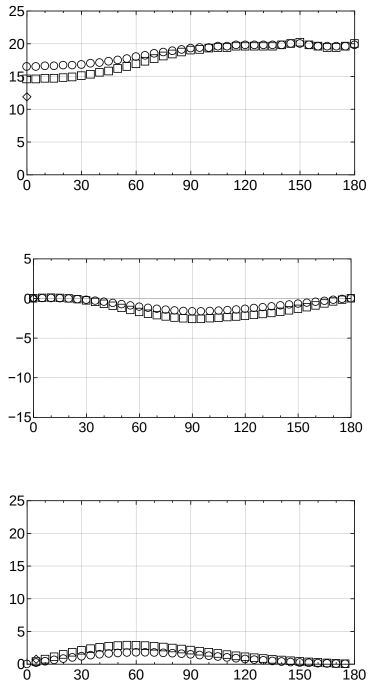

Figure 18: Results of the SSCs, including the longitudinal ship velocity component u_0, the check helm δ_0 and the hull drift angle β_0 at the average wave period T_P=10s for a pure car carrier (PCC) (Yasukawa et al., 2019)

Table 2 shows the conditions of wind and waves in the predictions of the SSP. The conditions are classified by the Beaufort (BF) Scale. The u_0 drops significantly at the head waves (wind) direction with an increase of the BF scale. The absolute value of δ_0 reaches the maximum at about 100^o in θ_W (χ). However, the maximum value is almost 10^o in BF10, and there is a safety margin for maximum rudder angle 35^o. β_0 is over 15^o in BF10 with the region of 15^o to 60^o of θ_W. In addition, it was shown that the studied ship had no problem in maintaining the course stability in adverse weather conditions.

$$V_{mean} = \sqrt{\left(1 - \frac{\Delta\overline{p}}{0.5\rho U^2}\right)} \cdot U \tag{1}$$

where,

$$\Delta\overline{p} \sim -\frac{\rho}{4}\omega_e^2|\eta_5|^2x^2 \tag{2}$$

Ueno et al. (2013) modelled the fluctuating velocity ($V_{fluctuating}$) due to the regular waves induced particle motion and the surge motion of a ship as follows:

$$V_{fluctuating}$$

$$= (1 - w_P)\{U - \omega_e\xi_a \sin(\omega_e t - \zeta_\xi)\} \tag{3}$$

$$+ \alpha\omega h_a e^{-kz_P} \cos\chi \cos(\omega_e t - kx_P \cos\chi)$$

Where w_P is wake fraction, ω_e is encounter wave frequency, ξ_a is surge amplitude, ζ_ξ is phase shift of the surge motion, h_a is incident wave amplitude, k is wave number of incident waves, χ is wave direction, x_P is propeller longitudinal coordinate, z_P is propeller immersion depth. α is a correction factor defined as

$$\alpha = \begin{cases} 0.2\left(\frac{\lambda}{L|\cos\chi|}\right) + 0.5, & \text{for } \frac{\lambda}{L|\cos\chi|} \leq 2.5 \\ 1, & \text{for } \frac{\lambda}{L|\cos\chi|} > 2.5 \end{cases} \tag{4}$$

Based on Eqs. (1) and (3), Taskar et al. (2016) modelled the time varying total velocity (V_{total}) in waves as:

$$V_{total} = \sqrt{\left(1 - \frac{\Delta\overline{p}}{0.5\rho U^2}\right)} \cdot V_{fluctuating} \tag{5}$$

Using this formula, it is possible to simulate the propeller thrust and torque of the ship manoeuvring in waves.

By RANS simulations, Guo et al. (2012) observed significant changes in wake filed in the presence of waves and ship motions. Similar results were observed by using PIV by Sadat-Hosseini et al. (2013).

3.5.2 Effect of ventilation in propeller performance

Vertical motions of a vessel and waves bring the thruster closer to the surface and make more susceptible to ventilation. Kempf (1934) was one of the pioneers on the study of ventilation effects on propellers. He studied the torque and thrust loss due to ventilation using similar propellers of different diameters as well as different immersion ratios and rate of revolutions. Shiba (1953) discussed the influence of different propeller design parameters e.g. expanded area ratio, contour of blade, radial variation of pitch, skewback, effect of rudder, turbulence of inflow on ventilation. Gutsche (1967) presented the test results of partially submerged propellers and suggested a procedure for calculating the out-of-water effect on average thrust. Fleischer (1973) presented average thrust and torque measurements that demonstrated interaction between propeller and hull when the propeller is partially submerged.

The effect of ventilation on average thrust and torque of propellers operating in waves has been discussed by Faltinsen et al. (1981) and Minsaas et al. (1983). Kaushan (2006) performed extensive model tests on an azimuth thruster with 6 DOF measurements of forces on one of the four blades. Based on the experiments, Kozlowska et al. (2009) observed three different types of ventilation inception mechanism and investigated influence of several factors on ventilation and thrust loss.

Thrust and torque loss factors, β_T and β_Q are defined as follows,

$$\beta_T = \frac{T_t}{T_n} \quad \text{where } T_n = K_{Tn} \cdot \rho n^2 D^4 \tag{6}$$

$$\beta_Q = \frac{Q_t}{Q_n} \quad \text{where } Q_n = K_{Qn} \cdot \rho n^2 D^5 \tag{7}$$

where T_t and Q_t are propeller thrust and torque including the ventilation effect, respectively. T_n and Q_n are propeller thrust and torque in open water, respectively. The n is propeller revolution, and D is the propeller diameter. K_{Tn} and K_{Qn} are open water characteristics of propeller thrust and torque, respectively. Here we introduce a model of the thrust loss factor by Minsaas et al. (1983). β_T was modelled as:

$$\beta_T = \beta \cdot \beta_V \tag{8}$$

where β is the thrust loss factor due to loss of propeller disc area, the Wagner effect and wave making, except the effects of ventilation, was approximated as follows:

$$\beta = 1 - 0.657 \cdot [1 - 0.0769(h/R)]^{1.258} \quad \text{for } h/R < 1.3 \tag{9}$$

where h is the propeller submergence from the shaft centre to the free surface and R is the propeller radius.

β_V is the thrust loss for a fully ventilated propeller, and approximated as follows:

$$\beta_V = \frac{1.5 \cdot \mathrm{EAR}}{K_{Tn}} \cdot \left(\frac{\pi}{2} \cdot \alpha + \frac{2gh}{V_\infty}\right) \tag{10}$$

where α is angle of attack of a propeller blade and V_∞ is velocity of propeller blade at $0.7R$. EAR is expanded area ratio of the propeller.

The result of these empirical relations was compared with measurements by Kozlowska et al. (2009) as shown in Figure 20. These formulas can roughly capture the thrust loss factors β_T.

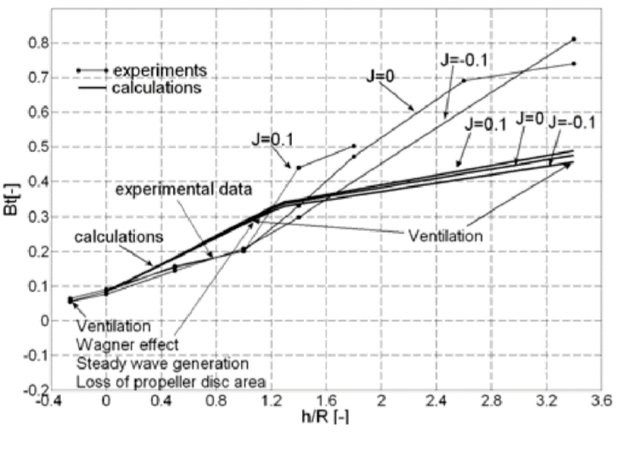

Figure 20: Comparison between calculated and experimental thrust loss factors at different advance ratios (Kozlowska et al., 2009)

3.5.3 Engine dynamics for simulation

Fluctuations of propeller loads also affect the engine performance due to shaft speed variations. Variable loads on the propeller in waves can cause mechanical failure (Amini, 2011). Livanos et al. (2006) and Theotokatos and Tzelepis (2013) studied coupled dynamics for a vessel-propeller-diesel engine system. Tanizawa et al. (2012) developed a methodology to include realistic engine response in the self-propulsion test to emulate real condition and get accurate estimates of fuel consumption in waves. Taskar et al. (2016) and Yum et al. (2017) studied unsteady interaction between engine and propeller caused by the waves from different directions by using the propeller inflow model of Eq. (5).

A generic equation of torque balance applied on propeller shaft has been described as

$$2\pi I_E \cdot \frac{dn_E}{dt} = Q_E - Q_f - Q_n \tag{11}$$

where, I_E is mass moment of inertia of the total propulsion system including main engine crank shaft, a main shaft and a propeller. n_E is rotating speed of engine. In case of no reduction gear, number of propeller rotation is identical to that of engine rotation. Q_E and Q_f stand for the engine torque and frictional torque of shaft bearing, respectively. Q_n is propeller torque. Engine torque could be given by the following non-dimensional form (Tanizawa et al., 2012):

$$\begin{cases} \bar{Q}_E = 0.5 \cdot \bar{h}_p^{\frac{2}{3}} + 1.5 \cdot \bar{h}_p^{\frac{1}{3}} \cdot \bar{n} + \bar{n}^2 \\ \bar{Q}_E = \dfrac{Q_E}{Q_E\,MCR},\ \bar{h}_p = \dfrac{h_p}{h_p\,MCR},\ \bar{n} = \dfrac{n}{n\,MCR} \end{cases} \tag{12}$$

where, Q_EMCR, h_PMCR, $nMCR$ are engine torque, stroke of fuel pump rack and rotating speed at the Maximum Continuous Rating, respectively.

Details on fuel flow, parameters of engine speed control system and characteristics of air and exhaust gas were described by Bondarenko et al. (2009) and Yum et al. (2017).

In the future, it is necessary to complete a simulation method that couples the equation of motion for manoeuvring in waves with the equation of motion for propeller speed considering engine characteristics, Eq. (11).

4. MINIMUM ENGINE POWER REQUIREMENT

4.1 General

To reduce the shipping's green house gases emissions via improved ship design and operation, the International Maritime Organization (IMO) adopted two mandatory mechanisms as energy efficiency standards for ships: Energy Efficiency Design Index (EEDI) for new ships and Ship Energy Efficiency Management Plan (SEEMP) for all ships.

The EEDI is an index that indicates the energy efficiency of a ship in terms of gCO_2 (generated) / tonne.mile (cargo carried); calculated for a specific reference ship operational condition. The intention is that, by imposing limits on this index, IMO will be able to drive ship technologies to more energy efficient ones over time. EEDI is thus a goal-based technical standard that is applicable to new ships. Ship designers and builders are free to choose the technologies to satisfy the EEDI requirements in a specific ship design.

There was a concern that one of the most effective ways of reducing a ship's EEDI is simply by choosing a smaller main engine or main propulsion motor for the ship, thus consequently reducing the ship's design speed. Within IMO a debate took place on how far speed reduction could be used to attain low levels of EEDI? As a result, it was decided to limit the use of this method of EEDI reduction so that it does not lead to unsafe and underpowered ships that may lose manoeuvring capability under adverse weather condition. These guidelines effectively define a methodology for estimating the minimum propulsion power for each ship for safe manoeuvring, thus ensuring that choice of the main propulsion engines/motors that satisfies these minimum requirements.

Accordingly, the purpose of the guidelines is to assist administrations in verifying that ships, complying with EEDI, have sufficient installed propulsion power to maintain the manoeuvrability in adverse weather conditions (Resolution MEPC.232(65), as amended by resolutions MEPC.255(67) and MEPC.262(68)). The guidelines currently apply to tankers, bulk carriers and combination carriers.

4.2 Assessment

The guidelines proposed for estimating the minimum power are based on two assessment levels or methods;

<u>Assessment Level 1, Minimum power lines assessment</u>: This is a simple approach and involves calculation of the minimum power from a specific line as a function of ship deadweight. For this purpose, the verifier should check if the ship has an installed power not less than the minimum power defined by the line represented by the following equation:

$$\text{Minimum Power Line Value [MCR, kW]} = a \times (\mathrm{DWT}) + b \qquad (13)$$

where "a" and "b" are constants and vary with ship type. There had been some discussion on the determination of these parameters (Table 3), and present values were decided at the 68th MEPC meeting (MEPC.262(68)).

The effects of these parameters can be reviewed by applying the minimum power lines to the recently built bulk carriers and tankers. From the IHS Sea web database (https://maritime.ihs.com), 1,517 bulk carriers and 874 tankers, which were built after 2000, were selected, and the minimum power lines are applied as shown in Figure 21. MCR power of the recently built ships (red circles) are a little bit smaller than the ships built before 2014. Most of the ships are compatible with the previous criteria. But some ships cannot satisfy the strengthened present criteria.

Table 3: Parameter a and b for determination of the minimum power line values for the different ship types

Reference	Ship Type	a	b
MEPC 64/4/13 (IACS et al.)	BC (DWT<275,825 ton) BC (DWT≥275,825 ton)	0.0606 0.0273	4195.2 13366.0
	Tankers	0.0603	5495.5
MEPC 64/4/42 (Japan & ROK)	BC (DWT<275,825 ton) BC (DWT≥275,825 ton)	0.0606 0.0273	2648.0 11818.8
	Tankers	0.0603	3294.0
Resolution MEPC.232(65)	BC	0.0687	2924.4
	Tankers	0.0689	3253.0
Resolution MEPC.262(68)	BC (DWT<145,000 ton) BC (DWT≥145,000 ton)	0.0763 0.0490	3374.3 7329.0
	Tankers	0.0652	5960.2

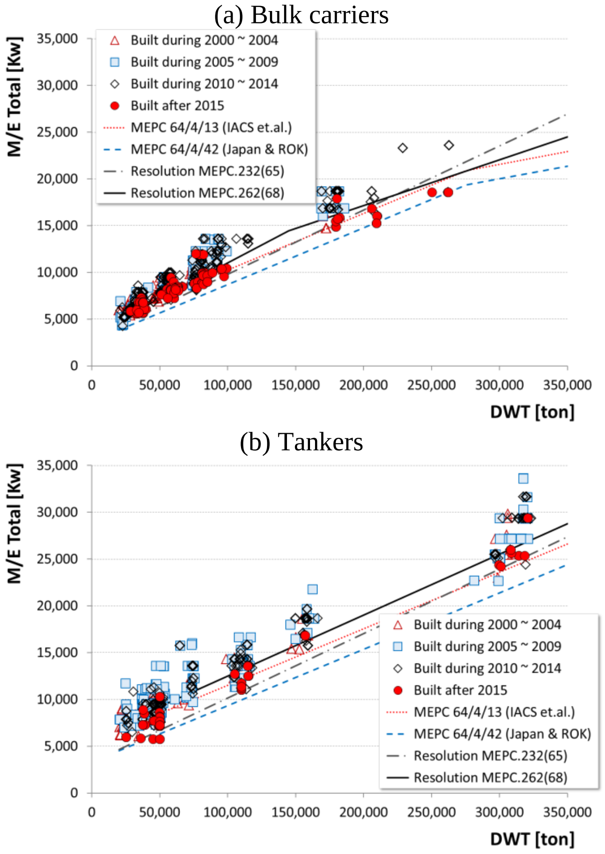

Figure 21: Application of minimum power lines.

<u>Assessment Level 2, Simplified assessment</u>: This is a more mathematically involved method. The assessment procedure consists of two steps:

Step 1: Definition of the required advance speed in head wind and waves, ensuring course-keeping in all wave and wind directions.

Step 2: Assessment whether the installed power is sufficient to achieve the above required advance speed.

Details of the assessment methods are given in the 2013 Interim Guidelines (MEPC.262(68)).

4.3 Subsequent Discussions on the Assessment

(1) <u>Discussion in IMO MEPC71</u>

At MEPC 71, two issues were discussed; China (MEPC 71/5/8, 2017) proposed amendments in light of the thrust deduction factor and the added resistance in wave. Although numerical and experimental results on the four tankers were submitted, it was not sufficient to draw support for the amendment. The second one was related to providing information on the progress and present status of the work of developing a draft revision of 2013 Interim Guidelines based on the research projects of SHOPERA and JASNAOE (MEPC 71/5/13, 2017, MEPC 71/INF.28, 2017). The project proposed the amendments shown in Table 4. Note that more severe adverse weather conditions were proposed than 2013 Interim Guidelines and more relaxed ship propulsion ability was proposed. For the latter, from previous 4 knots to 2 knots under the scenario of weather-vanning in coastal area under strong gale (see Table 5.)

Table 4: Proposed amendments on the adverse weather conditions (MEPC 71/5/13)

	Existing Guidelines	Draft revised Guidelines
Beaufort number	BF7 for L_{PP} < 200m BF8 for L_{PP} ≥ 250m	BF8 for L_{PP} < 200m BF9 for L_{PP} ≥ 250m
Wind speed	15.7m/s for L_{PP} < 200m 19.0m/s for L_{PP} ≥ 250m	19.0m/s for L_{PP} < 200m 22.6m/s for L_{PP} ≥ 250m
Hs	4m for L_{PP} < 200m 5.5m for L_{PP} ≥ 250m	4.5m for L_{PP} < 200m 6.0m for L_{PP} ≥ 250m

Table 5: Proposed amendments on the scenario for the evaluation of the sufficiency of ship's propulsion power to maintain the manoeuvrability in the adverse condition (MEPC 71/5/13)

Area	Coastal area
Weather condition	BF8 (gale) for L_{PP} < 200m BF9 (strong gale) for $L_{PP} \geq 250$m, linear over L_{PP} between 200m to 250m
Encountered wave and wind angle	Head seas to 30 degrees off-bow for a situation of weather-vanning
Propulsion ability	Speed through water at least 2 knots
Steering ability	Ability to keep heading into head seas to 30 degrees off-bow

However, considering that there were still different views on the adverse environmental conditions, it was further proposed that finalizing the draft revised guidelines at MEPC 71 would be premature and the Committee continue the discussion in parallel with the discussion of the EEDI review for phase 3 EEDI requirements. The Committee decided to consider the issue further at MEPC 72 and to extend the applicability of the 2013 Interim Guidelines to phase 2 EEDI requirements as an interim solution (MEPC 71/17, 2017).

(2) Discussion in IMO MEPC72

At MEPC 72, China proposed that thrust deduction factor can be conservatively defined as 0.1 and wake fraction can be defined as 0.15, based on the model test results of wake fraction and thrust deduction at low speeds of a ship (MEPC 72/5/9, 2018).

China also provided information on an alternative numerical method for calculating quadratic transfer function of the added resistance in regular waves applied in the 2013 Interim Guidelines (MEPC 72/INF.16, 2018).

However, it was discussed that more background data should be provided to validate the proposed method. So, a further submission was requested for MEPC 73 (MEPC 72/17, 2018).

(3) Discussion in IMO MEPC73

At MEPC 73, two issues were proposed. One is allowing for a shaft power limitation in order to resolve potential conflict between EEDI requirement and minimum required propulsion power (MEPC 73/5/1, 2018). The other is providing information on the work done on the minimum power requirements for ships in adverse conditions in the Netherlands (MEPC 73/INF.13, 2018).

Germany et al. proposed to limit the ship's shaft power for normal operation to meet the EEDI target whilst reserving extra power for adverse weather conditions (MEPC 73/5/1, 2018). Whilst there was general support, concerns were also expressed on the proposed idea on actual implementation mechanism and, especially when the use of reserve power is appropriate and allowed, and further consideration on how to certify NOx EIAPP scheme under the regulation 13 of MARPOL Annex VI if the reserved power for an engine is allowed. To improve the idea and for further discussion, it was agreed to keep consideration at next session (MEPC 73/19, 2018).

(4) Discussion in IMO MEPC74

China provided further validation of the numerical method for calculating the quadratic transfer function of the added resistance in regular waves (MEPC 74/INF.38)

Denmark introduced a concept to increase engine torque at low engine loads called the "adverse weather condition" function, by which an engine could ensure sufficient power to the ship in adverse condition as shown in Figure 22. It was concluded that the load diagram extension offers a potential solution that will enable fulfilment of the required minimum propulsion power at adverse weather conditions without negative impacts on emissions and within the current regulatory framework (MEPC 74/5/17, 2019).

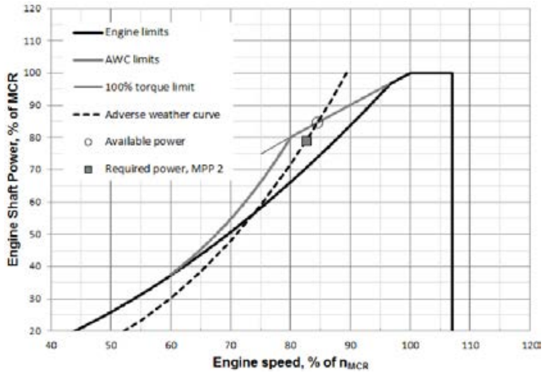

Figure 22: Extension of engine load limit by "Adverse Weather Condition" functionality (Denmark, MEPC 74/5/17, 2019)

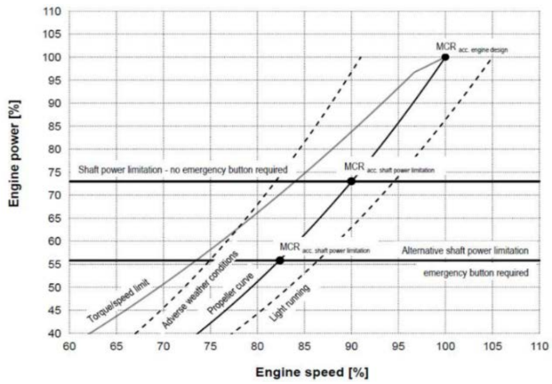

Figure 23: Concept of Shaft/Engine Power Limitation (France et al., MEPC 74/5/5, 2019)

France et al. proposed a refined proposal for Shaft Power Limitation ("ShaPoLi") related to the minimum propulsion as shown in Figure 23. (MEPC 74/5/5, 2019). The use of power reserve can be proceeded as follows:

1. In case of emergency (e.g. manoeuvrability in adverse conditions) the master can press / release an "emergency button" to use the power reserve (full installed engine power or torque reserve whatever the technical details of the power reserve provided);
2. In case of pressing the "emergency button", some defined conditions of the ship and of the engine will be automatically recorded in a tamper proof system which is part of the Shaft / Engine Power Limitation – device. Afterwards, the condition can be checked by the Administration or by a port State inspector;
3. Thereby, the installed engine power will remain as high as needed to maintain a ship's manoeuvrability in adverse condition, but for normal operation the power will be limited to the level set by the EEDI requirements; and
4. For calculation of attained EEDI for new ships, P_{ME} with the concept of power limitation would be on 75% of $MCR_{limited}$, and minimum propulsion power would be provided with some margins for reserved power.

Meanwhile, some objections and comments against the "ShaPoLi" was presented as (1) The proposal on "ShaPoLi" should not be agreed until the draft minimum propulsion power guidelines have been finalized and agreed by the Committee (MEPC 74/5/26, 2019), (2) The proposal on "ShaPoLi" should not be accepted as such a change to the power definition would undermine the intended goals of EEDI and would not result in improved energy efficiency for ships and (3) The shaft power limitation should be set with 15% sea margin (i.e. P_{ME} = 0.75 × 0.85 MCR ~ 0.64 MCR), so as to be in line with the recent shipbuilding practice (MEPC 74/5/31, 2019).

There were many supports on the application of "ShaPoLi" in resolving the improvement in energy efficiency with concerns over minimum power especially for large bulk carriers and oil tankers. However, there were still significant technical barriers to be addressed including which engine power should be used for NOx certification of marine diesel engine, etc; and there were concerns that "ShaPoLi" concept could discourage technical innovation as the same engine would have a lower EEDI, also there would be challenges for port State control. Hence, the Committee decided to further consideration at next session with concrete proposals on the shaft power.

(5) <u>Discussion in IMO MEPC75</u>

France et al. proposed an updated proposal for shaft power limitation (MEPC 75/6/6). For further discussion to improve the concept, it was agreed to consider this matter at a future session. It was also agreed to proceed with the revision work for the finalization of the Interim Guidelines.

For the finalization at MEPC 76, Corresponding Group was established. Definition of the “Adverse conditions” and assessment procedure (Deletion/Retention of Appendix 2, assessment Level 2) are being discussed.

4.4 Investigation on the Effects of Other Factors for the Assessment

The issues brought from MEPC 71 and following meetings can be categorized into four items. The first is the definition of adverse weather condition, the second is the calculation of added resistance in wave, the third is the determination of self-propulsion factors, and the last is the selection of the engine operation limits. The first three items are related with the Level 2, Simplified assessment. The effects of these items are reviewed by applying the assessment to KVLCC2. KVLCC2 is the second variant of the KRISO tanker which has been used as a benchmark test vessel for manoeuvrability study.

For the Simplified assessment, some parameters, such as the windage areas, dead weight, MCR power and RPM, are necessary. These parameters are assumed as Table 6. (Deadweight and MCR power are the averaged values of the VLCC built between 2000~2004, windage area are estimated from the similar ships). Under these assumptions, KVLCC2 complies with the minimum power line assessment criteria.

Table 6: Assumed parameters of KVLCC2 for the application of simplified assessment

Windage area		Deadweight / MCR	
Frontal, A_{FW} [m^2] Lateral, A_{LW} [m^2]	920 3,300	Deadweight [ton] Power $_{MCR}$ [kW] RPM $_{MCR}$ [-]	302,273 26,341 81

4.4.1 Effect of adverse weather conditions

There have been four suggestions on the definition of adverse weather conditions. For ships whose length is larger than 250m, it can be summarized in Table 7.

Table 7: Adverse weather conditions for ships with L_{PP} > 250 m ($H_{1/3}$: Significant wave height, T_P: Peak period, V_W: Mean wind speed)

References	$H_{1/3}$ [m]	T_P [s]	V_W [m/s]
Resolution MEPC.262(68)	5.5	7.0~15.0	19.0
MEPC 64/4/13 (IACS)	6.0	8.0~15.0	19.0
MSC 93/21/5 (Greece)	7.0	-	23.0
MEPC 71/5/13 (Denmark)	6.0	8.8~12.2	22.6

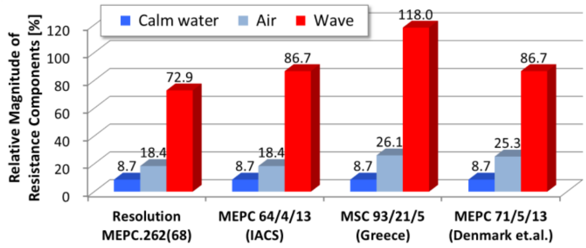

Figure 24: Relative magnitude of resistance components under different adverse weather conditions (100% means the total resistance by the Resolution MEPC.262(68))

Figure 24 shows the relative magnitudes of each resistance components with respect to the total resistance calculated by the present Interim Guidelines, Resolution MEPC.262(68). For this comparison, the wave added resistance were estimated by i-STAP. i-STAP is an ISO 15016 based speed trial analysis program develop by KRISO (Shin et al., 2016). The ratios of wind and wave added resistances are increased as the wind speed and the significant wave height are increased. In all cases, the wave added resistance amounts to more than 70% of the total resistance. This means that the accurate estimation of low speed wave added resistance can be one of the decisive factors for the simplified assessment.

The ratio of required power over the available power is shown as Figure 25. If this ratio is large than 100%, it means that the vessel is not compatible with the simplified assessment. Except the worst weather condition, MSC

93/21/5, the vessel satisfy the simplified assessment criteria with 6~12% power margin.

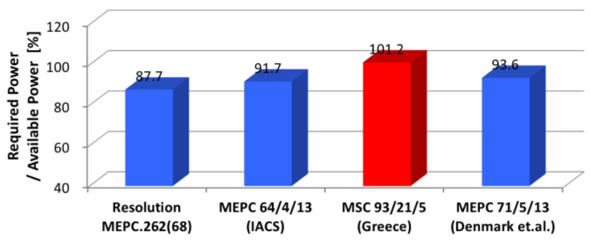

Figure 25: Ratio of the required power over the available power under different adverse weather conditions

4.4.2 Effect of wave added resistance estimation

According to the 2013 interim guidelines for determining minimum propulsion power, the added resistance in waves can be calculated by the quadratic transfer function. This function can be obtained from the added resistance test in regular waves at the required ship advance speed as per ITTC procedures 7.5-02 07-02.1 and 7.5-02 07-02.2 or from equivalent method verified by the Administration.

The required ship speed for the minimum power assessment usually ranges between 4 and 6 knots. Hence, due to the reflected waves, it is quite difficult to perform the model tests in the conventional towing and the square basin tests have been preferred. For the KVLCC2, Sprenger et al. (2017) performed the model test at F_n = 0.055 (corresponding to 6 knots in full scale) as a part of the SHOPERA project. This kind of model tests are possible but may not practicable for routine ship design purposes, as few such facilities exist. Therefore, the empirical formulae or the potential based calculations have been used as a practical alternative. Table 8 shows some estimation methods for the added resistance in waves for comparison study.

Table 8: Estimation methods for the added resistance in waves for comparison

Types	Name	Motion induced	Reflection correction
Empirical	STAWAVE2 (Boom et al., 2013)	Jenkine's method	Experimental data
	MEPC 70/INF.33		
2D strip	SLE	Maruo method	Faltinsen
	i-STAP (Shin et al., 2016)		NMRI emprical
	PrimeShip (Class NK)		
3D panel	WISH (Park et al., 2014)	Pressure integration	

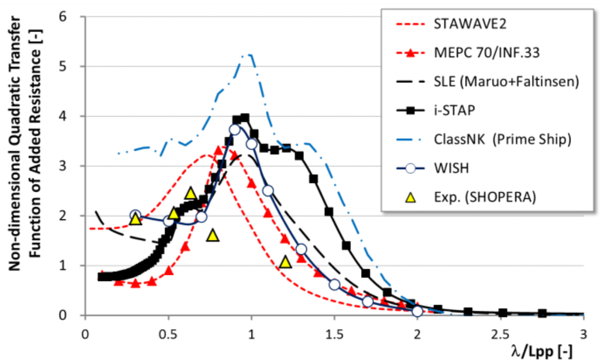

Figure 25: Non-dimensional quadratic transfer function of the wave added resistance for KVLCC2 at F_n = 0.055

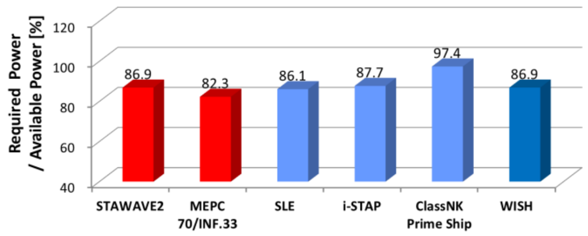

Figure 26: Ratios of the required power over the available power by the various different wave added resistance estimates

The non-dimensional quadratic transfer functions are compared with the results of empirical formulae or potential based calculations in Figure 25. Appreciable variances between the estimation methods could be found. The ratios of required power over the available power are shown in Figure 26. All the estimation methods comply with the simplified assessment criteria in the assumed KVLCC2 case. The MEPC 70/INF.33, which is an empirical formulae based on SHOPERA project, satisfies the criteria with a relatively large margin, while the PrimeShip gives the most conservative result.

4.4.3 Effect of self-propulsion factors

According to the Interim Guidelines, self-propulsion factors (wake fraction, *w*, and thrust deduction factor, *t*) can be obtained either from model tests or empirical formula. The recommended conservative estimates are given in the Interim Guidelines.

China noted that the values of thrust deduction factor and wake fraction obtained from the model test are fairly lower than those obtained from the Interim Guidelines. China was of the view that the value of thrust deduction factor should be obtained from required ship advance speed. That is it should be higher than the value in bollard pull state (about 0.04 for single screw ships), and lower than the value in calm water condition with design speed (about 0.2 for single screw ships). China proposed that the thrust deduction factor can be set to 0.1 (MEPC 71/5/8, 2017), and wake fraction can be conservatively defined as 0.15 (MEPC 72/5/9, 2018).

Table 9 shows four sets of the wake fraction and thrust deduction factors for KVLCC2. As was noted by China, the estimates for *t* and *w* by the Interim Guidelines are similar to those values at calm sea design speed. Considering that the required ship advance speed is about 4~6 knots and the added resistances are about ten times larger than the calm water resistance, the estimates by the Interim Guidelines do not seem to be realistic ones.

Table 9: Estimates on wake fraction and thrust deduction factors for KVLCC2

References	Wake fraction, *w*	Thrust deduction factor, *t*
Resolution MEPC.262(68)	0.350	0.245
Calm sea at F_n=0.141	0.347	0.233
MEPC 71/5/8 (China)	0.350	0.100
MEPC 72/5/9 (China)	0.150	0.100

Figure 27 shows the available power (dashed line) and the required powers for the different self-propulsion factors of Table 9. As the estimated values of the Interim Guidelines (△) and the calm sea design speed (◆) are similar, the required powers are almost the same. When only the small thrust deduction factor (□) is used, the required RPM and power are lower than the Interim Guidelines, but the available power margin is almost the same. When the both self-propulsion factors are changed (●), the required RPM and power are higher than the Interim Guidelines and the power margin is smaller than the Guidelines. This shows that the simplified assessment result can be affected by the estimates of self-propulsion factors and the estimates of the Guidelines are not the most conservative case. Hence, the more realistic estimates on these factors may results in the more reliable simplified assessment.

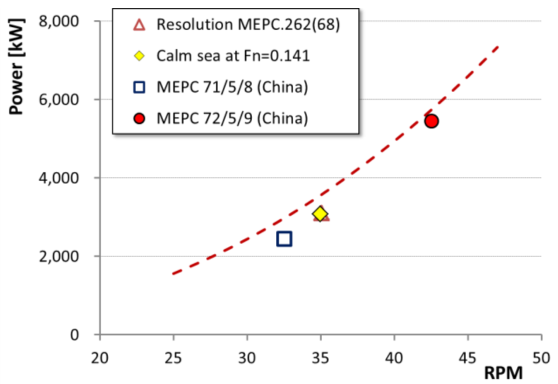

Figure 27: Effects of wake fraction and thrust deduction factors for KVLCC2 (Dashed line is a power limit curve under the assumed MCR condition)

5. CONCLUSIONS

5.1 Prediction Methods of Ship Manoeuvring in Waves

A large number of works on manoeuvring in waves methods have been published during in this period. Experimental research remains valuable and is being used complementary to numerical research. Due to technological developments in progress, tests in irregular waves with large wave height are becoming more feasible.

Direct CFD simulations of ship maneuvering in waves were presented by several authors. Using CFD simulations of self-propulsion and turning motions as well as zig-zag maneuvers of a free running ship model in regular waves can be conducted. However, due to the high computational cost and even longer time simulation requirement, direct CFD maneuvering simulations in irregular waves are still a changeling problems.

Until now, the problem of manoeuvring in deep water waves has been mainly treated, but the problem has been extended to shallow water area.

As an application example of the calculation of manoeuvring in waves, there are many studies on the manoeuvring limit of ships by analysing the steady sailing performance.

5.2 Benchmark data

The SIMMAN research project has facilitated new data for the KCS and the ONRT in regular waves. These data is quite valuable to support the validation and certification of numerical simulation method.

5.3 Minimum Engine Power Requirement

The issues brought about from IMO-MEPC71 and following meetings were addressed concerning the minimum power requirements. The accurate estimations of the wave added resistance and the self-propulsion factors in higher propeller load condition are a decisive factor for the simplified assessment.

6. RECOMMENDATIOS

Update the following guidelines:

- Free Running Model Tests in Waves
- Captive Model Tests for Measuring Forces in Waves

To improve the numerical method for manoeuvring in waves, the following actions are needed:

- Validate the numerical methods for mean wave drift forces, especially steady lateral force and steady yaw moment acting on an advancing ship in cooperation with the seakeeping committee.
- Provide the captive test data on the hydrodynamic forces acting on the ship in waves, such as
    - ✓ Oblique towing test data in waves
    - ✓ Circular motion test data in waves
    - ✓ PMM test data in waves
    - ✓ Rudder force data in waves when ship is straight moving.

for validation of CFD in cooperation with the manoeuvring committee.

- Investigate the effect of wave height on the propeller performance and the coupling with the main engine in cooperation with the propulsion committee.

Validate the Level 2 – Simplified Assessment Method of the 2013 Interim Guidelines (MEPC.232(65)) by enhanced and comprehensive methods.

Investigate the concept of “Shaft Power Limitation” (ShaPoLi) introduced for the first time at MEPC 73 (MEPC 73/5/1) and deliberated at following sessions (MEPC 74/5/5, MEPC 75/6/6), as a measure to overcome intrinsic conflict between safety and environmental regulatory requirements.

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