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Review

Literature review on evaluation and prediction methods of inland vessel manoeuvrability

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ABSTRACT

Ship manoeuvrability plays a major role in navigation safety. In order to achieve certain manoeuvring requirements, ship configurations need to be specifically considered, which in turn influence the economic performance of shipping. Compared with seagoing ships, inland vessels sail in a more complex navigation environment. At the same time, inland vessels have to operate independently without additional assistance like tugs. This paper presents a review of the state-of-the-art of evaluation and prediction methods for inland vessel manoeuvrability. First of all, different aspects that influence manoeuvrability and the methods assessed for inland vessels and seagoing ships are compared. Accordingly, additional knowledge is required regarding the modelling methods and evaluation criteria of inland vessel manoeuvrability. Furthermore, a step-by-step review on manoeuvrability research relates various impact factors to manoeuvring performance. Finally, the gaps in knowledge of evaluation and prediction methods for inland vessel manoeuvrability in shallow/restricted waterways are described.

Contents

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Nomenclature

α	non-linear damping factor of the turning motions (–)
δ	rudder angle (rad)
ψ	heading angle (rad)
Fr	Froude number (–)
Re	Reynolds number (–)
J	propeller advance ratio (–)
V_a	advance speed (m/s)
n	propeller rate of revolution (1/s)
D	propeller diameter (m)
H	water depth (m)
K	turning ability index (–)
T	course keeping index (–)
T	ship draught (m)
x_0, y_0, z_0	coordinates in the earth-fixed frame of axis (m)
x_{0G}, y_{0G}	coordinates of the centre of gravity of the ship in the earth-fixed frame of axis (m)
x, y, z	coordinates in the body-fixed frame of axis (m)
X, Y	external forces acting on the ship in the direction of x, y axes on midship (N)
F_x, F_y	external forces acting on the ship in the direction of x_0, y_0 axes at ship centre of gravity (N)
M_z	yaw moment acting ship around centre of gravity (N m)
x_G, y_G, z_G	coordinate of centre of gravity in $o{-}xyz$ (m)
N	external moment acting on the ship around midship (N m)
I_{zG}	moment of inertia of the ship around centre of gravity (kg m^2)
I_z	moment of inertia of the ship around midship (kg m^2)
u_G, v_G	velocity in the direction of x axis and y axis at centre of gravity (m/s)
u, v	velocity in the direction of x axis and y axis on midship (m/s)
U	resultant velocity on midship $U=\sqrt{u^2+v^2}$ (m/s)
r	angular turning speed around midship z axis (rad/s)
m	mass of the ship (kg)
L	ship length between perpendiculars (m)

1. Introduction

Thus far, research in the field of ship manoeuvrability has primarily focused on seagoing ships (Central Commission for the Navigation of the Rhine, 2013). However, interest in inland shipping is growing (Rigo and Wolters, 2013). In general, inland vessels have more complex arrangements than seagoing ships. One of the common inland ship arrangements is shown in Fig. 1, which has a hull tunnel in front of ducted propellers with multiple rudders. At the same time, inland waterways, which are commonly limited in water depth and channel width, are also more complex for ships to sail in than the open sea. Accordingly, inland vessel manoeuvrability is even harder to tackle and thus expected to be more crucial than that of seagoing ships.

In the early stage of ship design, pragmatic rules of thumb are commonly used. These basic rules are not sufficient for the interaction effects of each ship component. In order to ensure a good performance in practice, accurate methods of prediction are needed to link the design parameters to manoeuvring performance. Additionally, relevant requirements on manoeuvrability should be established for designers and authorities. Meanwhile, because of the potential high cost of improving manoeuvrability, a wise compromise between good performance and economics has to be made (Biancardi, 1993).

Presently, the most common standards for ship manoeuvrability are issued by the International Maritime Organization (IMO). These standards should be applied to “ships of all rudder and propulsion types, of 100 m in length and over, and chemical tankers and gas carriers regardless of the length” (International Maritime Organization, 2002a,b). The type of ships (seagoing ships or inland vessels) is not clearly stated in the IMO standards. However, the applicable condition of the standards is described as deep, unconstrained water ($H/T>4$). It is rational to suppose that the IMO standards are intended for seagoing ships, but still valuable as guidance for inland vessels.

Regulations for inland vessels are normally proposed by regional authorities (Central Commission for the Navigation of the Rhine, 1995, 2012) and classification societies (Bureau Veritas, 2011). Compared to the IMO standards for seagoing ships, these regional requirements have fewer test manoeuvres and criteria. According to Gray et al. (2003), it is still doubtful if the standards lead to adequate manoeuvrability in shallow, restricted, and congested waterways. Therefore, in either case of inland vessels or seagoing ships, new test manoeuvres and procedures for

Fig. 1. An inland vessel with a hull tunnel, a ducted twin-propeller, and two rudders per propeller. Source: own photograph.

shallow water operations are demanded to predict and evaluate ship manoeuvrability (Landsburg et al., 2005; Liu et al., 2014).

This paper focuses on the lack of knowledge in current inland vessel manoeuvrability evaluation and prediction. State-of-the-art methods and procedures are presented. Differences between inland vessels and seagoing ships are highlighted. Currently, methods of manoeuvrability prediction are mainly seagoing ship oriented. To apply these models for inland vessels, crucial impact factors are identified. Further research on manoeuvring tests and force analyses is suggested for shallow-water manoeuvrability.

First, Section 2 gives a description of the internal and external impact factors on ship manoeuvrability. Section 3 shows existing criteria and manoeuvres for manoeuvrability evaluation. Section 4 describes mathematical models on ship motion as the theoretical basis for manoeuvring modelling. Section 5 presents two classic expressions of hydrodynamic forces. Section 6 introduce methods of hydrodynamic force analysis for full-scale tests, model-scale tests, and numerical methods. Section 7 describes validation and verification methods. Finally, Section 8 draws conclusions that inland vessel manoeuvrability deserves more attention in hydrodynamic force prediction and benchmark data for validation.

2. Impact factors on ship manoeuvrability

Ships sail under certain impacts which affect ship manoeuvring capacities. In this paper, these impacts are roughly characterised as two aspects: external environment and internal design. External factors (Section 2.1) depend on the environment in which the ship sails, while internal factors (Section 2.2) are determined by the design and operation profiles. Indirect impacts, for instance human behaviours (Hooft, 1974; Hetherington et al., 2006) are also important but not included in this paper.

2.1. External factors

The following paragraphs provide a review on the two main effects of the navigation environment on the manoeuvrability of inland vessels, namely shallow water and ship–ship/ship–bank interactions.

2.1.1. Shallow water effects

As a consequence of the scale enlargement in fleets (Del et al., 2000; Gray et al., 2003; Landsburg et al., 2005), the ship manoeuvring performance in constrained channels is no longer only a concern for inland vessels but also crucial for seagoing ships. Seagoing ships sail in the deep sea most of the time, thus performance in shallow water is often considered to be less important. On the contrary, for inland vessels which commonly sail in shallow water, shallow water effects are very important (Hofman and Kozarski, 2000).

The ratio of the water depth to the ship's draught (H/T) is typically larger than 4 for seagoing ships, but it is commonly smaller than 2.5 for inland vessels, especially in the dry period. The influence of water depth begins noticeable in medium deep water ($1.5 < H/T < 3.0$), becomes significant in shallow water ($1.2 < H/T < 1.5$), and dominates in very shallow water ($H/T < 1.2$) (Vantorre, 2003). Basically, shallow water has complex effects on the primary ship manoeuvring forces (X, Y) and moments (N) affecting the manoeuvring performance.

The limited water depth influences the ship resistance (Tuck and Taylor, 1970), which is the main component of hull forces in the longitudinal direction (X_H). Meanwhile, in constrained waterways, the horizontal restrictions also have an influence on the resistance, which is the so-called blockage effect (Kim and Moss, 1963; Tamura, 1972). At the same advance speed, a decrease of H/T ratio results in an increase of squat and wave heights (Jachowski, 2008) leading to an increase of X_H. In order to reduce the wave resistance in shallow water, Saha et al. (2004) proposed a numerical optimization method for hull forms based on seagoing ships. For inland vessels, Rotteveel (2013) compared existing models for ship resistance, proposed shallow water corrections for manoeuvring, and pointed out that the existing estimation methods do not clearly present the shallow water effects.

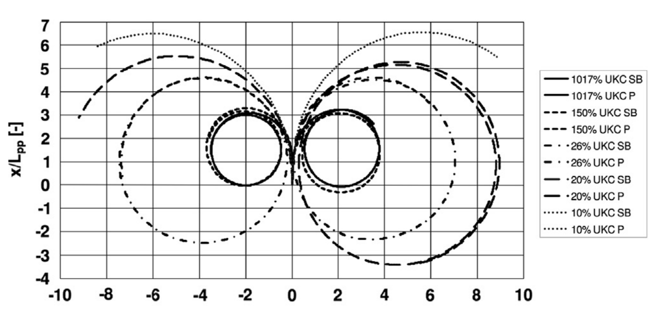
Considering the turning forces (Y) and moments (N), Eloot and Vantorre (2011) reported an increase of course keeping ability leading to a larger tactical diameter in shallow water than in deep water for a slender seagoing container ship as shown in Fig. 2. Due to an increase of the hull damping force in shallow water, tactical diameters commonly become larger, which is a typical shallow water effect that is widely known (ITTC Manoeuvring Committee, 2008).

However, Yoshimura and Sakurai (1989) and Yasukawa and Kobayashi (1995) discovered that the tactical diameter is smaller in shallow water than in deep water for twin-propeller wide-beam ships as shown in Fig. 3. This phenomenon is caused by the increase of rudder forces due to the high propeller load in shallow water.

Kijima and Nakiri, 2004 and Lee and Lee (2005) also showed that shallow water effects on the tactical diameters and advances vary with ship profiles. As shown in Fig. 4, the tactical diameter increases while the advance almost remains the same on the left, but both the tactical diameter and the advance increase on the right.

On the contrary to the above findings, Koh and Yasukawa, 2012 found that a pusher-barge system may have a smaller turning circle and a worse course keeping ability in shallow water than in deep water (Fig. 5).

Above all, research on shallow water effects is mainly done for large seagoing ships manoeuvring in harbours or entrance channels rather than for inland vessels. Those studies include Yoshimura and Sakurai (1989), Yasukawa and Kobayashi (1995),



UKC (%)	T/L_{PP} SB	T/L_{PP} P
150		
32	9.1	7.0
26	8.4	7.2
15	10.5	8.6
10	13.6	12.4

Fig. 2. Free-running turning circle model tests with a 6000 TEU container ship in deep and shallow water compared with the tactical diameter prediction (table) from captive model tests (Eloot and Vantorre, 2011).

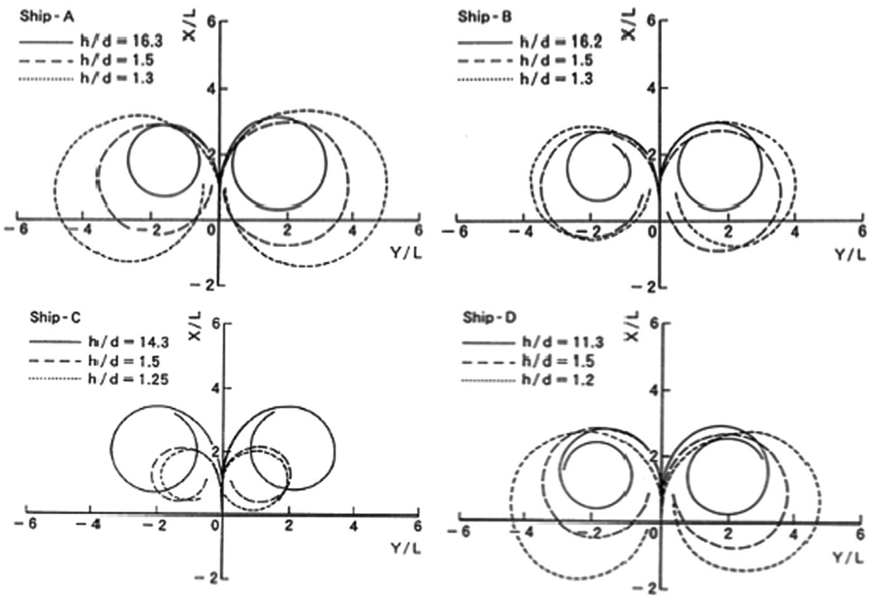


Fig. 3. Comparisons of ship trajectories in shallow water and deep water (Yasukawa and Kobayashi, 1995).

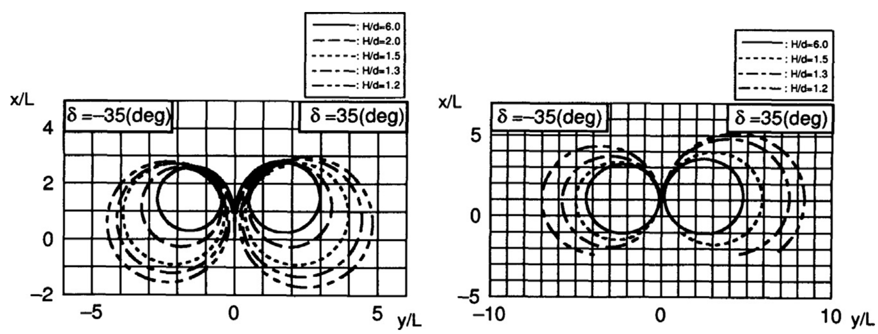


Fig. 4. Various shallow water effects on the tactical diameters and advances for different ship profiles (Lee and Lee, 2005).

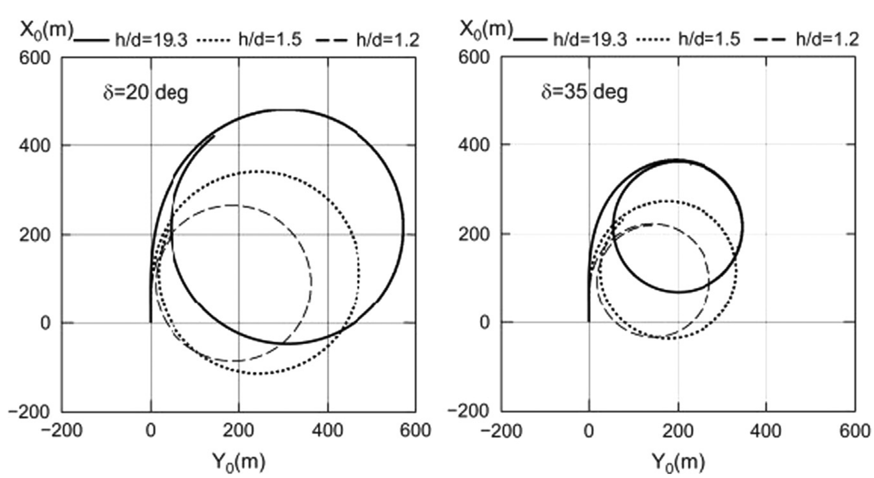


Fig. 5. Turning trajectories of a pusher-barge system at rudder angle 20° (left) and 35° (right) (Koh and Yasukawa, 2012).

Kijima and Nakiri (2004), Lee and Lee (2005), and Eloot and Vantorre (2011). The shallow water effects on the inland vessel manoeuvrability are still unclear. It should be noted that inland vessels have similar characteristics to the twin-propeller wide-beam ship applied by Yoshimura and Sakurai (1989) and Yasukawa and Kobayashi (1995), which means shallow water effects on inland vessels could be different from seagoing ships. To further study the ship manoeuvrability in shallow water, Landsburg et al. (2005) suggested to apply more accurate full-scale trials and mathematical modelling techniques.

2.1.2. Ship–ship and ship–bank interactions

Inland waterways feature many artificial structures such as locks, terminals, and bridge pillars along or in the channel, which restrict the navigable area (Liu et al., 2012). Thus, knowledge of ship behaviour in horizontally and vertically restricted areas helps to reduce infrastructure and operation costs and enhance the navigation safety (Eloot and Vantorre, 2011). Inland vessels suffer far less from the strong wind and waves than seagoing ships. However, in natural waterways, currents may lead to very different sail conditions for the upstream and downstream directions. Last but not least, unlike seagoing ships, which are in many cases requested to use additional manoeuvring assistance like tugs in mandatory pilotage areas when entering confined waterways or during mooring operations, inland vessels have to manoeuvre independently all the time. Consequently, highly manoeuvrable inland vessels are required to guarantee the navigation safety.

Due to the high density of traffic in inland waterways, inland vessels have to pass and meet each other at close range much more frequently than seagoing ships. When a ship moves into the close proximity of other ships, lateral forces and yaw moments are induced due to the asymmetrical flow around the ship. For seagoing ships, both numerical (Chen et al., 2002; Lo, 2012) and experimental (Vantorre et al., 2002; Lataire et al., 2009, 2011; Eloot et al., 2012) methods were applied to analyse these interactions and their impacts. However, for inland vessels, no systematic research was found.

Ship handling is also greatly affected by the interactions between the ship and banks (Eloot et al., 2007; Lee and Lee, 2008). The minimum channel width to the ship's beam ratio may be as small as 4, 3, or 2 for double-lane, narrow-double lane, or single-lane channels respectively (Rijkswaterstaat, 2011, 2013). Kijima and Yasukawa (1985), Kijima (1987), and Kijima and Qing (1988) discussed the impacts and possibilities of induced collisions. Lee et al. (2007) researched the transverse distance and the maximum rudder angle for safe passing, while Vantorre et al. (2003) proposed an empirical formula to predict ship-bank interaction forces through model tests. de Koning Gans (2005), Zou and Larsson (2013), and Lo (2012) investigated the sinkage and trim caused by ship–ship and ship–bank interactions through CFD simulations. Similar to the research on shallow water effects, the scenarios here were primarily described for a large seagoing ship, for instance KVLCC2 (Zou and Larsson, 2013), in a narrow channel.

2.2. Internal factors

External impacts are determined by the characteristics of the waterways, thus ship designers cannot control them. What designers can do is to analyse the navigation environment and adapt ship particulars to compensate the negative influences of the external disturbances. Four main features of inland vessels which are different from seagoing ships, specifically slow speed, hull forms, propulsion, and rudders are addressed in the following paragraphs.

2.2.1. Slow speed

Inland vessels' cruising speed 8–28 km/h is often significantly slower than the speed of seagoing ships 18–36 km/h. To compare the velocity among ships, the Froude number is commonly evaluated, which is referred as $Fr = v/\sqrt{gL}$. Considering the range of ship dimensions and velocities, inland vessels commonly have a smaller Fr than seagoing ships, which affects the wake-making resistance. Meanwhile, relatively slow sailing speed also means a slow incidence velocity to the propeller and the rudder affecting their performance.

Even though slow-speed manoeuvres are not the central concern for the design of most seagoing ships, their crucial impacts on safe operations are getting more and more consideration as discussed by Hwang et al. (2003) and Dand (2003). Thus, inland vessels which consistently sail at even slower speed than seagoing ships should be optimised to improve the slow-speed performance. When the ship moves at even lower forward speed, for example proceeding to a berth, Yoshimura and Sakurai (1989) indicated that the hydrodynamic forces are more complicated as the sway and yaw motions may become larger than the forward speed.

ITTC Manoeuvring Committee (2008) raised the necessity of standards for the slow-speed manoeuvring. After that, ITTC Manoeuvring Committee (2011) presented an overview of the existing slow-speed manoeuvring models for seagoing ships. Oh and Hasegawa (2012) further compared the effectiveness of typical slow-speed manoeuvring models. In sum, Eloot and Vantorre (2004) concluded the opportunities and limitations of the slow-speed models emphasizing the differences between low speed and ordinary speed manoeuvring.

2.2.2. Hull forms

Inland vessels include motor vessels, pusher-barge systems, and towed-barge systems. Compared with the hull forms of seagoing ships, inland motor vessels commonly have larger block coefficients, much larger L/B ratios, and much larger B/T ratios due to the limits imposed on draught, length, and beam (Quadvlieg, 2013). These differences in hull forms will not only greatly influence the ship resistance but also affect other hull generated hydrodynamic forces and moments in manoeuvring. A significant amount of research is conducted on the hydrodynamics of pusher-barge systems, which are superior compared to motor vessels in terms of transport capacity in shallow water. Pusher-barge systems and towed-barge systems are widely used in inland waterways all around the world. Luo and Zhang (2007), Koh et al. (2008a,b,c), Maimun et al. (2011), and Koh and Yasukawa (2012) presented manoeuvring research on pusher-barge systems through numerical or experimental methods. Tabaczek et al. (2007) and Tabaczek (2010) analysed hull resistance and planar motion of a twin-screw inland vessel with different bow forms.

2.2.3. Propulsion

The propulsion of inland vessels is affected by the ship propulsors and appendages. Since propellers are not far from the free surface, inland motor vessels are commonly designed with a tunnel at the aft ship to improve the propeller inflow and prevent the ventilation. In the process of model tests for ships with appendages, scale effects were found by Clement (1957) and Gregory and Beach (1979) in determining the resistance. In order to relate test results to practical ships, Holtrop (2001) covered the extrapolation methods of ships with multiple appendages and complex propulsors.

Contrary to seagoing ships, inland vessels have an extensive use of azimuth thrusters, bow thrusters, and stern thrusters. As the propeller size of inland vessels is constrained by the water depth, multiple propellers, especially twin-propellers, are commonly installed. Kim et al. (2007) reported worse turning but better course keeping and course changing abilities of a twin-propeller ship compared to a single-propeller ship at sea. For inland vessels, further research is still in need for the impacts of thrusters on manoeuvring.

2.2.4. Rudders

The aspect ratio of an inland vessel's rudder is limited by the water depth to around 1 compared to a common value of 2 for seagoing ships (Kim et al., 2012). Meanwhile, rudder orders for course corrections are more frequently applied in the channels than in the sea. To obtain sufficient manoeuvring forces and moments, a configuration of multiple rudders per propeller is commonly used to increase the total rudder area. Additionally, inland vessels feature a wider range of rudder profiles for instance the Schilling rudder with additional plates on the root and tip. The rudder profile largely affects the rudder hydrodynamic characteristics and has further impacts on the manoeuvrability (Liu and Hekkenberg, 2015).

Nagarajan et al. (2008) demonstrated the superiority of a VLCC ship with Schilling rudders compared to mariner rudders at constant engine torque under various encounter angles of wind in course keeping ability. Vantorre (2001) carried out comprehensive model tests to determine the open water characteristics of several rudders in a shallow water towing tank. Last but not least, inland vessels may use a large rudder angle over 35° which is the common maximum rudder angle for seagoing ships, in hard manoeuvring situations. Rudder angles may even be 90° when side movement is needed at slow speed. Currently, most of the research on propeller and rudder performance is done for deep water (Quadvlieg, 2013). The necessity to adjust the existing propeller and rudder models for shallow water was proposed by Eloot and Vantorre (2004).

With all these impact factors, existing estimation methods for hull forces based on seagoing ships may not give proper results for inland vessels leading to a bad prediction of manoeuvrability. At the same time, due to the unique nautical operation limits, the interest in manoeuvres for seagoing ships and inland vessels is not always the same. For instance, in case of an imminent collision, a seagoing ship may either initiate a crash stopping or a turning manoeuvre, while inland vessels can only make a crash stopping actions due to the limits of constrained waterways (Bertram, 2012). This enhances the importance of the crash stopping ability for inland vessels. All these different impact factors and interest in the manoeuvring performance request further research on the mechanisms of ship motions for more accurate manoeuvrability prediction.

This section presented literature on the external and internal impacts to initialise inputs for equations of ship motion in Section 4. According to the different impacts, requirements of manoeuvrability for inland vessels and seagoing ships should be adjusted. To explore the state-of-the-art of manoeuvring capabilities, existing standards of ship manoeuvrability are listed and compared in the next section.

3. Manoeuvrability standards

In order to improve maritime safety and enhance marine environmental protection, standards for ship manoeuvrability should be used in ship design, construction, repair, and operation (Biancardi, 1993). Due to the lack of uniform manoeuvring standards, some ships had been built with very poor manoeuvring qualities, which may result in casualties and pollution (Nobukawa et al., 1990; Dijkhuis et al., 1993). Hence, elaborate and uniform criteria should be established for safety (Pérez and Clemente, 2007). At present, IMO has built uniform criteria for ships in the deep open sea (International Maritime Organization, 2002a,b). Actions are still needed to ensure new built ships meet those requirements.

ITTC Manoeuvring Committee (2014) gave an review of the criteria in use for inland vessels, fast ships, and dedicated low speed manoeuvres. More requirements are in need for other scenarios like constrained waterways and port areas. The necessity of more critical requirements for specific situations was discussed by Li et al. (2005). To solve a certain manoeuvring problem, three phases were advised by Del et al. (2000):

- Study all factors influencing the problem.
- Propose all possible solutions and choose those feasible and acceptable.
- Study emergency situations.

In order to identify the missing knowledge in these aspects, a comparison of existing standards is shown in Table. 1. From this table, it is concluded that more elaborate criteria for different navigational conditions, especially for inland waterways, should be issued to define the minimum performance. Furthermore,

Table 1
Overview of standards and criteria of ship manoeuvrability (International Maritime Organization, 2002a,b; American Bureau of Shipping, 2006; Central Commission for the Navigation of the Rhine, 2012; European Parliament, 2009a,b).

Category of manoeuvrability	Testing manoeuvres	IMO criteria	ABS criteria	CCNR criteria
Turning ability	Turning test with maximum	Advance $<4.5L$	Not rated	The turning capacity of vessels and convoys whose length (L)
	rudder angle ($-35°/35°$)	Tactical diameter $<5L$	Rtd $>=1$	
Initial turning ability	10°/10° zigzag test	Distance ship travelled $<=2.5L$ by the time the heading has changed by 10° from the original heading	Rti $>=1$	does not exceed 86 m and width (B) does not exceed 22,90 m shall be considered sufficient
		First overshoot angle:		Evasiver manoeuvres with a rudder angle of 20° and 45° to starboard and port shall be checked by yaw rate and
		$<10°(L/V<10s)$;		
		$<(5+0.5L/V)°(10s \le L/V<30s)$;	Rated $>=1$	
Yaw-checking and		$<20°(LL/V \ge 30s)$		
	10°/10° zigzag test	Second overshoot angle:		
course-keeping ability		$<25°(L/V<10s;)$		
		$<(17.560.7L/V)°(10s \ge L/V<30s)$;	Not rated	maximal period instead of overshoot angles for zigzag manoeuvres. Criteria vary for different ship dimensions and water depth
		$<40°(L/V \ge 30s)$		
	20°/20° zigzag test	First overshoot angle $\ge$ 25°	R$t\alpha20>=1$	water depth
stopping ability	Full astern stopping	Track reach $<15L$	not rated	$L<=110\ m$ within 305 $m, L>110$
		None for head reach	Rts $>=1$	m within 350 m

emergency situations, such as engine failure, strong wind, currents, and waves, should be examined to predict the worst manoeuvring cases.

Due to the differences in navigation conditions and ship particulars between inland vessels and seagoing ships as discussed in Section 2, the standards for manoeuvrability are expected to be “different” in the aspects of test manoeuvres and criteria. A summary of existing standards, namely International Maritime Organization (IMO), American Bureau of Shipping (ABS), Central Commission for the Navigation of the Rhine (CCNR), and Bureau Veritas (BV) is given in the succeeding paragraphs to find the gaps to improve inland vessel standards. After discussing the existing manoeuvrability standards, the required full-scale test conditions and contents are given out.

3.1. International Maritime Organization

The most widely accepted criteria for ship manoeuvrability are issued by IMO including turning ability, initial turning ability, yaw-checking and course-keeping abilities, and stopping ability (International Maritime Organization, 2002a,b). Daidola et al. (2002) described how these IMO standards were defined and improved. However, these standards are specified for ships longer than 100 m with traditional propulsion systems (propellers and rudders) in deep unconstrained waters. Shorter ships and vessels with unconventional propulsion systems, for instance azimuth thrusters (Toxopeus and Loeff, 2002; Ghassemi and Ghadimi, 2008), are not subject to the standards. Based on the opinion of the administration, current rules can be taken as reference for unconventional ships.

To comply with the requirements of authorities, the manoeuvrability criteria should be evaluated under specified test conditions and procedures (International Towing Tank, 2008b). For seagoing ships, the trial should be conducted in deep unconstrained sea water to eliminate the effects of the waterway bottom, banks, and other external objects. Deep water here means that the depth of water should be more than 4 times of the mean draught (International Maritime Organization, 2002a). The trial speed should be set to at least 90% of the ship's speed corresponding to 85% of the maximum engine output. The test ships should be loaded to the dead weight and even keel within 5% deviation.

Three manoeuvres are needed for sea trials: the turning circle manoeuvre (turning and initial turning ability), the zigzag manoeuvre (yaw-checking and course-keeping abilities), and the stopping test (stopping ability) (International Maritime Organization, 2002a,b). For example, the zigzag test, which is especially developed for towing tank tests but also popular for full-scale tests (Bertram, 2012), can show the manoeuvring capacities of initial turning and yaw checking ability. Brix (1993) carried out series of model tests yielding the typical values of ship zigzag performance. Whereas for inland vessels in the Rhine, only the evasive manoeuvre (similar to zigzag test but checked by yaw rates instead of heading angles) and the stopping test are required (Central Commission for the Navigation of the Rhine, 2012).

Since the test environment described by the IMO standards is open deep water, there is a need to consider ship's manoeuvring capacities in shallow constrained areas, such as harbour entrance channels and ports (Gray et al., 2003; Hwang et al., 2003; Landsburg et al., 2005). The existing manoeuvres may also lead to misunderstanding of the actual manoeuvring performance. Hooft (1974) found that the first overshoot angle is only influenced by the current distribution. Yoshimura et al. (2000) pointed out that an evaluation based on the second overshoot angle of 10° zigzag test and the first overshoot angle of 20° zigzag test may regard poor manoeuvrability as good.

At present, the hydrodynamics (constrained water), meteorological (wind, wave, and current), and navigational (other ships, artificial constructions) impacts are not covered in the existing IMO standards. Quadvlieg and van Coevorden (2003) proposed more elaborate criteria for shallow water, constrained waterways, and slow-speed manoeuvring. The need to formulate criteria for off service speed (slow speed) and water depth (shallow water) was recognised by Hwang et al. (2003), Dand (2003), Gray et al. (2003), and Landsburg et al. (2005).

Except from the standards specified for ship manoeuvrability, there are other requirements of IMO for navigation safety, which indirectly affect the ship manoeuvring performance. For example, International Maritime Organization (2012) states, “The main steering gear and rudder stock should be capable of putting the rudder over from 35° on one side to 35° on the other side with the ship at its deepest seagoing draught and running ahead at maximum ahead service speed and, under the same conditions, from 35° on either side to 30° on the other side in not more than 28 s”. This rule requires a minimal rudder turning rate, which affects the turning related manoeuvring abilities. Thus, the rudder torque and steering torque should be considered to achieve this.

3.2. American Bureau of Shipping

In most of the cases, designers only want to meet the minimum requirements of the authorities (Quadvlieg and van Coevorden, 2003). In order to enhance the seagoing ship safety instead of just meeting the minimum criteria, American Bureau of Shipping (2006) built a rating system to evaluate the overall manoeuvring capacity, which provides information of implementation and application of the IMO standards. Biancardi (1993) developed a set of performance indexes based on full-scale trial results. Spyrou (1994) applied a rating procedure based on a synthesised manoeuvrability index. Belenky and Falzarano (2006) compared the IMO requirements (International Maritime Organization, 2002a,b), the rating system established by Barr et al. (1981), and ABS Guide for Vessel Manoeuvrability (American Bureau of Shipping, 2006).

3.3. Central Commission for the Navigation of the Rhine

Even though there are no universal criteria such as IMO standards for inland vessels, requirements have been set up as regional regulations. To maintain the safe and smooth flow of traffic on the Rhine, manoeuvring criteria and assessment approaches were proposed by Dijkhuis et al. (1993). Central Commission for the Navigation of the Rhine (1995) stated the required inland vessel manoeuvrability in terms of forward speed, stopping capacity, manoeuvrability while going astern, capacity to take evasive actions, and turning capacity. However, as seen in Table 1, these criteria are not as elaborate as the IMO standards. On the other hand, inland vessels, which commonly sail in more complex situations than seagoing ships, may need more manoeuvres and criteria in detail to ensure safety.

Considering the differences between the open sea and inland waterways (Section 2), inland trials should be carried out in representative inland waterways. For ships in the Rhine, tests should be performed in areas of the Rhine or other inland waterways with similar conditions (Central Commission for the Navigation of the Rhine, 2012). The area should be straight not less than 2 km and sufficiently wide, in flowing or still water. The under keel clearance should be at least 20% of the water depth and not less than 0.5 m (Central Commission for the Navigation of the Rhine, 1995). The test load condition should be 70–100% of full load and even keel. The ship velocity relative to the water is at least 13 km/h.

Vessels and convoys proceeding downstream should be able to stop 'in good time' while remaining sufficiently manoeuvrable. Turning capacity should be demonstrated by upstream turning manoeuvres. Details of stopping capacity and evasive actions can be found in Central Commission for the Navigation of the Rhine (2012). Furthermore, standards for inland vessel manoeuvrability developed by the Central Commission for the Navigation of the Rhine have been adopted as European Commission Directives (European Parliament 2009a,b).

3.4. Bureau Veritas

Bureau Veritas (2011) put requirements on stopping capacity, astern trials, capacity of taking evasive actions, and turning capacity of inland vessels. The ships classified by Bureau Veritas should be checked by all navigation tests in areas designated by the ship classification society. The stopping capacity test for ships not longer than 86 m and not wider than 22.9 m can be replaced by turning manoeuvres (Bureau Veritas, 2011).

After the review on various standards, it is clear that the expected differences between regulations for inland vessels and seagoing ships exist. But the requirements have not always been specified in detail. Thus, Liu et al. (2014) proposed standard manoeuvres and parameters as benchmarks to efficiently evaluate the inland vessel manoeuvring performance. On the other hand, the existing methods of the full-scale test are also insufficient for ships in shallow/constrained water. In order to help naval architects to have further insight into the manoeuvrability of inland vessels, more elaborate criteria are needed.

4. Mathematical models on ship motion

After analysing the impact factors (Section 2), ship particulars and environmental parameters are determined as inputs for modelling. Mathematical models are used to investigate the ship manoeuvrability through computer simulations after which manoeuvring performance can be evaluated with the standards discussed in Section 3. In this section, these mathematical models and their required inputs are discussed as a basis for the evaluation of those inputs in Section 5. Initially, coordinate systems are defined to denote the ship trajectory and express the forces and moments (Section 4.1). Newton's laws of motion can be applied to relate ship movements to the acting forces and moments in the predefined domain (Section 4.2). Afterwards, Section 5 discussed expressions of forces and moments applied to solve the ship motion equations.

4.1. Coordinate systems

To formulate the mathematical description of the ship motion, the coordinate system should be established first. Fig. 6 shows the coordinate systems used in the presented article. Two right-handed frames of axes are applied in a bidirectional coordinate system: the earth-fixed coordinate system $o_0-x_0y_0z_0$ and the body-fixed coordinate system $o-xyz$. The origin of the earth-fixed coordinate system is commonly located at the start point of the manoeuvring simulation, while the origin of the body-fixed coordinate system is set on midship or at the centre of gravity. As highlighted by Yasukawa and Yoshimura (2014), midship is more convenient than the centre of gravity considering ship load conditions.

x_0 axis is placed in the direction of the original course of the ship, while x axis points towards the bow of the ship. The angle between x_0 and x is called the heading angle ψ. y_0 and y axes are clockwise normal to the x_0 and x axes respectively. z_0 and z axes

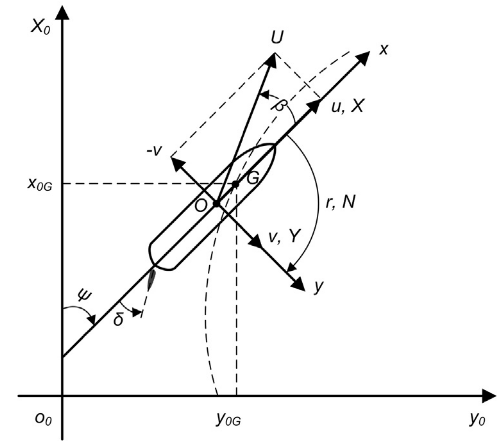


Fig. 6. Earth-fixed coordinate system and body-fixed coordinate system with origin on midship.

are defined downwards vertically. The ship's position is determined by the ship centre of gravity in the earth-fixed coordinate x_{0G} and y_{0G} in the orientation of heading angle ψ. The centre of gravity of the ship is situated at (x_G, y_G, z_G) in the body-fixed coordinate. x_G is the longitudinal coordinate of the centre of gravity in $o-xyz$. y_G depends on the ship configuration, which is mostly zero but can be different, for instance in a push-barge system. z_G relates to the ship loading. With the origin of the body-fixed coordinate system lying at the centre of gravity, transformation from body-fixed coordinate system to earth-fixed coordinate system is shown in the following equation:

$$\left.\begin{aligned} x_0 &= x_{0G} + x\cos\psi - y\sin\psi \\ y_0 &= y_{0G} + x\sin\psi + y\cos\psi \\ z_0 &= z. \end{aligned}\right\} \tag{1}$$

Assuming that the ship is manoeuvring at contemporary speed U on midship, the speed is decomposed into an advance velocity u in the x-axis and a transverse v in the y-axis. The velocities at centre of gravity can be expressed as $u_G = u$, $v_G = v + x_G r$. Drift angle β is the angle between the direction of actual velocity U and the x-axis achieving the relation $u = U\cos\beta$ and $v = -U\sin\beta$. Under the rudder angle δ, the ship is turning at an angular velocity $r = \dot{\psi}$ with respect to the midship.

4.2. Ship motion models

A lot of effort has been put into the ship manoeuvring models, but current work is primarily evolutionary rather than revolutionary. Ships are free to surge, sway, heave in the direction of axes and roll, pitch, yaw with respect to axes respectively. As it may seriously affect the speed and accuracy in simulations, the required number of degrees of freedom should be seriously considered according to the navigation conditions and ship particulars. For inland vessels which rarely encounter severe waves and swell, a model with three degrees of freedom including surge, sway, and yaw motion can give sufficient results.

Extra dimensions should be added considering the ship particulars to improve the prediction accuracy. For example, it is critical to use a 4 DOF (degree of freedom) model including roll motion for ships with low metacentric height and high speeds. Stern and Agdrup (2008) clearly demonstrated the large difference in the prediction of manoeuvring performance between 4 DOF and 3 DOF. Considering the impacts of sinkage and trim, a 6 DOF model can be more suitable for ships in constrained waterways and or in shallow water.

A system of non-linear three dimensional (surge, sway, and yaw) equations is discussed here as an example. The relation of motion and forces can be derived according to Newton's second law in the body-fixed coordinate system with the origin lying at the ship's centre of gravity as

$$\left.\begin{aligned} F_x &= m(\dot{u}_G - v_G r) \\ F_y &= m(\dot{v}_G + u_G r) \\ M_z &= I_{zG}\dot{r}, \end{aligned}\right\} \tag{2}$$

where F_x, F_y, M_z are external forces and moments acting at the centre of gravity in the direction of x-axis, y-axis, and around z-axis, m is the ship's mass, u_G, v_G are velocities at the centre of gravity of the ship in the direction of x-axis and y-axis respectively, r is the yaw rate, $\dot{u}_G, \dot{v}_G, \dot{r}$ are derivatives of the quantities, and I_{zG} is the moment of inertia of the ship around the centre of gravity.

In practice, the origin of the body-fixed coordinate system is normally located on the midship point instead of at the centre of gravity of the ship as shown in Fig. 6. Assuming that the ship is symmetrical about its longitudinal plane, Eq. (2) can be adapted into Eq. (3) with the origin of the body-fixed coordinate system on midship rather than at the centre of gravity

$$\left.\begin{aligned} X &= m(\dot{u} - vr - x_G r^2) \\ Y &= m(\dot{v} + ur + x_G \dot{r}) \\ N &= I_z\dot{r} + m x_G(\dot{v} + ur), \end{aligned}\right\} \tag{3}$$

where X, Y, N are external forces and moment acting on midship in the direction of x-axis, y-axis, and around z-axis, u, v are velocities on midship in the direction of x-axis and y-axis respectively, and I_z is the moment of inertia of ship around midship.

4.3. Ship manoeuvring response models

To simplify the description of ship motions, Nomoto (1957) derived a first order response model from the equations of ship motion through the Laplace transformation as

$$T\dot{r} + r = -K\delta, \tag{4}$$

where K is the turning ability index, T is the course keeping index.

However, this simplified equation neglects the influences of transverse speed, longitudinal speed, and heel, resulting in an inaccurate prediction for most practical situations (Bertram, 2012). Norrbin (1970, 1978) introduced the non-linear damping factor into the response model as

$$T\dot{r} + r + \alpha r^3 = -K\delta, \tag{5}$$

where α is a non-linear damping factor of the turning motions.

However, this Norrbin model still does not include non-linear terms, for instance the asymmetrical force induced by the single-screw propeller. Compared to the equations of ship motion, the response model is far less time consuming. Therefore, response models are superior to study the course control problems or autopilot design (van Amerongen and Udink ten Cate, 1975; van Amerongen, 1984).

In this section, the coordinate system of manoeuvring simulations is discussed. The coordinates are commonly used as a combination of the earth-fixed coordinate system and the body-fixed coordinate system. Attention should be paid to the origin of the body-fixed coordinate system and the directions of axes and angles for comparing prediction results from different research. Two manoeuvring models, i.e. the equations of ship motion (4.2) and the ship manoeuvring response model (4.3), are compared to better express the external impacts. The equations of ship motion are superior in combining multiple components of forces and moments, which is thus proposed to be applied for inland shipping manoeuvrability prediction. The ship manoeuvring response model is suggested to be used in autopilots because of the lower computation time.

5. Expressions of hydrodynamic forces

In order to solve the equations of motion equations (2) and (3) in Section 4, expressions are needed for the components of external forces (X and Y) and moment (N) acting on the ship. These forces and moments are caused by the ship hydrodynamics and environmental disturbances due to wind, waves, current, banks, and shallow water. Owing to the different manners of parametrizations for the hydrodynamic forces and moments, the mathematical models are generally divided in two types: the whole ship model (Abkowitz, 1964) and the modular model (Norrbin, 1970).

According to the typical manoeuvres involved at each speed, Quadvlieg (2013) proposed requirements of mathematical models for speed in the range of low speed to service speed. Based on these requirements, Quadvlieg (2013) indicated that a modular model is desired to describe the manoeuvring behaviours of inland vessels. The modular model is superior in the physical representation of force components, such as hull, propeller, and rudder forces. These forces are functions of hydrodynamic coefficients which normally come from databases, captive model tests, approximation formulas, numerical methods, and system identification techniques. With the fast development of computer science, Computational Fluid Dynamics (CFD) techniques are increasingly popular in manoeuvrability research for hydrodynamic coefficients and force prediction.

5.1. The whole ship model

Abkowitz (1964) expressed the hydrodynamic forces and moments as functions of the kinematic parameters and the rudder angle as Eq. (6), which can be further expressed in Taylor-series

$$\left.\begin{aligned} X &= X(u, v, r, \dot{u}, \dot{v}, \dot{r}, \delta) \\ Y &= Y(u, v, r, \dot{u}, \dot{v}, \dot{r}, \delta) \\ N &= N(u, v, r, \dot{u}, \dot{v}, \dot{r}, \delta) \end{aligned}\right\} \tag{6}$$

Substituting Eq. (6) in Eq. (3), the equations for the whole ship (Abkowitz model) are achieved. This whole ship model treats the hull–water interaction as a black box and proved to be successful for arbitrary simulations (Strom Tejsen and Chislett, 1966; Crane, 1973).

5.2. The modular model

Represented by Ogawa and Kasai (1978), the Mathematical Model Group of the Society of Naval Architects of Japan proposed a modular model, which is the so-called MMG model, as the following:

$$\left.\begin{aligned} X &= X_H + X_P + X_R \\ Y &= Y_H + Y_P + Y_R \\ N &= N_H + N_P + N_R. \end{aligned}\right\} \tag{7}$$

In the modular model, the hydrodynamic forces and moments are decomposed into three parts according to different origins, ship hull, propeller, and rudder, which are commonly denoted by the subscripts "H", "P", and "R". Consequently, the hull–propeller–rudder interactions and the performance of each component can be analysed.

In order to improve the accuracy and adaptability in different conditions, more followed work was done for ship manoeuvring capacity in shallow water and constrained waterways. Fujino and

Ishiguro (1984) showed a remarkable dependence of the rudder effectiveness on the water depth. Kijima and Yasukawa (1985), Kijima et al. (1990a,b), and Kijima et al. (1992) proposed a modular prediction method and approximation formulas for hydrodynamic coefficients of hull forces in shallow and deep water. Kijima and Yasukawa (1985), Kijima and Qing (1988), and Kijima and Nakiri (2004) also applied the MMG model to analyse the interaction forces generated by other ships and banks.

Details about the MMG model were discussed in a series of papers written by Yoshimura (1986, 1988, 2005) and Yasukawa and Yoshimura (2014). Oltmann and Sharma (1984) presented an approximation method of the interactions of hull–propeller–rudder in a combined engine and rudder model. Clarke (2003) commented that the whole ship model gives a smooth representation of the forces, but has no physical meaning, while the second order modulus expansions can well represent the hydrodynamic forces at angles of incidence. Comparing the advantages and disadvantages, the whole ship model is more suitable to obtain the overall performance of a manoeuvring ship as a system through free running tests or system identification methods, while Modulus expansions can better express the effects and interactions of each component, i.e. hull, propeller, and rudder.

6. Hydrodynamic force analysis

The purpose of ship manoeuvring tests is twofold. One objective is to directly obtain the overall performance in prescribed conditions, for instance the turning ability at a constant rudder angle. Full-scale tests or free running model tests are included in this category. The other goal is to gather hydrodynamic coefficients which are used to solve the mathematical equations discussed in Sections 4 and 5. Model tests (Section 6.1) are much less expensive and more controllable than the full-scale tests. Thus, model tests are widely used at the initial design stage to obtain parameters while full-scale tests are taken as the final manoeuvrability check.

6.1. Model tests

6.1.1. Captive model tests

The Planar Motion Mechanism (PMM) and the Computerized Planar Motion Carriage (CPMC) are used to determine the hydrodynamic coefficients (van Leeuwen and Journée, 1970). The ship model (bare hull or equipped with the propeller, the rudder, and the electrical motor for propulsion) is attached to the towing carriage. Hydrodynamic coefficients are then achieved through testing in straight lines and harmonic tests . With these coefficients, other tests which are not suitable for the size of the towing basin, such as pull-out manoeuvres and spiral tests, can be numerically simulated.

Another kind of captive model test, the so-called rotating-arm test, is designed to obtain stationary turning coefficients. The ship model is attached to a rotating arm which is set at the centre of the basin turning at a constant velocity. Due to the large disturbance of turbulent water generated by the moving model, the accuracy of this sort of test is relatively low except in the case of a sufficiently large basin, which is, however, much more expensive.

6.1.2. Free running tests

Static towing tests are done at constant drift angles and rudder angles. This means the motion is decoupled in the horizontal plane, so additional corrections have to be applied for the decoupling effects (van Leeuwen and Journée, 1970). To predict manoeuvring characteristics in a direct way, free running tests are applied to solve the uncoupled problems as it can generate a series of coefficients associated with positions (Im and Seo, 2009). International Towing Tank (2008a) presented the standard procedure of free running tests. System identification techniques are commonly associated with free running model tests for hydrodynamic force coefficients (Grochowalski, 1989; Yoon et al., 2007). Oltmann (1996) gave an example on how to create a mathematical model from a series of zigzag tests. However, free running tests are constrained by the dimension of the towing tanks. Thus, these tests are carried out in rather small scale with high scaling error.

6.1.3. Open water tests

In addition to tests for manoeuvrability, open water tests for propeller thrust, torque, and efficiency are usually performed in towing tanks or cavitation tunnels (International Towing Tank, 2008b). On the other hand, open water tests are also applicable for rudder open water characteristics (Vantorre, 2001). Contrary to the highly non-uniform ship wake in reality, these tests are carried out in relatively uniform inflow. Similar to the Froude number Fn for ships, the advance ratio J should be kept identical between the model and prototype for the geometrical and Froude similarities as the following:

$$J = \left(\frac{V_a}{n \cdot D}\right)_p = \left(\frac{V_a}{n \cdot D}\right)_m. \tag{8}$$

The local Reynolds number (Re) is commonly calculated at 0.7 of the propeller radius. Thus, the actual local velocity V_R is shown as the following:

$$V_R = \sqrt{V_a^2 + (0.7n\pi D)^2}, \tag{9}$$

where $0.7 \cdot n \cdot \pi \cdot D$ is the local tangential speed. As it is not possible to keep Reynolds and Froude similarity at the same time, corrections from the model scale to the full scale have to be done through extrapolation methods (International Towing Tank, 2011a).

Considering the size of towing tanks, test models are commonly built in range between 2 m and 9 m in length (International Towing Tank, 2011a). Due to the differences of vortex shedding and flow separation, errors exist in scaling model test results to full scale (Bertram, 2012). Thus, Holtrop (2001) and Oyan (2012) developed extrapolation methods to transfer model test results to full-scale applications (International Towing Tank, 2011a).

6.2. Numerical methods of hydrodynamic force modelling

Even though the ship model test (Section 6.1) is regarded as the most reliable solution (Vantorre, 1992; Vantorre and Eloot, 1998), numerical methods are also widely applied in practice, such as regression formulas and CFD methods. In practice, ship designers need simple prediction tools with reasonable accuracy. To design a new ship, coefficients may be obtained from one parent ship or series of hull forms through empirical constants or regression formulas. Extensive tests are needed to build up these empirical data bases. Todd (1953, 1963), Toda et al. (1992), and Longo et al. (1993) described the procedure tests on Series 60 (Gertler, 1954) and presented the results in different perspectives, such as wave profiles, wave elevations, mean velocities, pressure field distributions, and scale effects.

Regression formulas based on model test data are useful for similar ship forms. Holtrop (1977, 1978, 1984) and Holtrop and Mennen (1982) formulated resistance prediction methods through regression analysis of model and full-scale tests, which are also suitable for hull-related force prediction in manoeuvring research. However, regression analysis is not applicable for innovative hull types or ships sailing out of test conditions. In order to compensate for the time consuming CFD calculation and high cost model/full-scale tests, system identification methods were applied by Abkowitz (1980), Yoon and Rhee (2003), Zhang and Zou (2011), and Araki et al. (2012).

As the computation power increases dramatically, numerical calculation through CFD methods is becoming more and more popular (Tyagi and Sen, 2006). Morgan and Lin (1998) gave an introduction to the historical development of the hydrodynamic prediction. However, CFD results are still not reliable for complex configurations requiring more work on verification and validation (Stern et al., 1999, 2001; Wilson et al., 2001; Rotteveel, 2013).

Manoeuvrability prediction methods can be divided into two aspects: free running model tests and computer simulations based on mathematical models. Due to the high cost and uncontrollable test conditions, full-scale tests are rarely used in prediction. Hence, model tests are still considered the most reliable measurement, which are commonly used as a final check before the ship construction. However, model tests are also expensive and not reusable for successive modifications. Providing that computer simulations based on mathematical models are more flexible at the initial design stage.

7. Validation and verification

As more and more manoeuvring capability predictions are done with computer modelling, the validation and the verification for assessing the validity and quality of simulations are becoming more and more important. The subjective of verification is to ensure the model implementation and results are correct, while the goal of validation is to show that if the model is applicable under requirements with acceptable accuracy (Garret, 1974). Sargent (2005) presented the approaches to study the validity of simulation models, which were also adoptable for marine simulations. Stern et al. (2001) and Wilson et al. (2001) presented the methodology and application of validation and verification of CFD simulations for container ships.

Manoeuvring simulation models can be used for manoeuvrability prediction and ship manoeuvring simulators (International Towing Tank, 2011b). Each component of the model simulation, such as ship particulars and environment data (Section 2), mathematical modelling (Section 4), hydrodynamic force prediction models (Section 6), should be validated step by step. Comparable structures about how to document the parameters, methodology, and results were recommended by Hwang (2004) and Endo (2005).

The validity and the quality of the simulation results can be clearly presented by comparing with full-scale test results. Trials of the tanker Esso Osaka had been widely cited since Crane (1979) published the data. Simonsen and Stern (2003, 2005) validated a RANS model of rudder effects on manoeuvring based on the Esso Osaka data. Other optional benchmark data for CFD validation were given by International Towing Tank (1999). However, tests for Esso Osaka (Crane, 1979) were carried out on an old ship design using old equipment. Therefore, this set of data was not recommended by International Towing Tank (2011b) as validation material.

As previously discussed by ITTC Manoeuvring Committee (1999), full-scale benchmark data are expensive and it is hard to control the external disturbances. Thus, free running and captive model tests are the favourable alternatives. Despite the scaling errors, model tests are superior in geometry control and test conditions. Four new benchmark models (two very large tankers KVLCC1 and KVLCC2, a container ship KCS, and a naval combatant DTMB 5415) were adopted by ITTC Manoeuvring Committee (2005). Though none of these models were built in full-scale, a series of model tests were done and compared by different institutions in the Workshop on Verification and Validation of Ship manoeuvring Simulation Models (Stern and Agdrup, 2008; Stern et al., 2009, 2011). These model parameters and test results are open to researchers and useful for various purposes of validation and verification.

Benchmark data mentioned above were applied to computational methods of seagoing ships in the deep open sea. Little information about experimental or numerical tests for inland vessels in deep or shallow water is publicly available. However, benchmark data for seagoing ships are still meaningful to develop prototype manoeuvring model for inland vessels. Benchmark data of model-scale and full-scale tests for inland vessels are proposed to be established for validation and verification of manoeuvrability prediction methods.

8. Conclusions

Ship manoeuvrability deserves more attention in ship design compared to the previous economic point of view (Quadvlieg and van Coevorden, 2003), not only for seagoing ships but also for inland vessels. In order to identify and achieve a highly manoeuvrable ship, the main challenges are how to accurately calculate the forces and moments acting on the ship in different conditions, for instance, in shallow water.

Moreover, the methods employed in the estimation process should be acceptable in time and cost while having adequate accuracy. Estimating the hydrodynamic forces on inland vessels with complex features is also a challenge. After all, with these challenges in mind, research should be carried out on standard manoeuvres and criteria from the administration point of view and further give insight into the theoretical problems.

To obtain additional insight into inland vessel manoeuvrability evaluation and prediction methods, this paper has presented a step-by-step review of the manoeuvrability research procedures, namely impact analysis, mathematical models, force expressions, manoeuvring simulations, and validation and verification. Through these steps, a general procedure of research on inland ship manoeuvrability is summarised:

Step 1 : Regarding the main impact factors discussed in Section 2, collect data on both external conditions of the test and internal properties of the considered inland vessel as inputs for manoeuvring simulations.

Step 2 : According to the estimated amplitudes of the ship motion in six dimensions, determine the number of degrees of freedom. Choose the position of the origin of the body-fixed coordinate system (at the centre of gravity or on midship). Apply the equations of ship motion studied in Section 4.

Step 3 : Express the forces in the equations of motion to build up the mathematical model as discussed in Section 5.

Step 4 : Determine the hydrodynamic forces according to the mathematical model chosen or developed in Step 3 through model tests or numerical methods described in Section 6.

Step 5 : Based on the requirements in Section 3, evaluate the ship manoeuvring performance.

In this paper, clear differences has been found in the external environment factors and internal design factors on ship manoeuvrability between inland vessels and seagoing ships. To have an accurate prediction of inland vessel manoeuvring performance, estimation methods of forces and moments have to be adapted to the characteristics of inland waterways and inland vessels, such as shallow water conditions, ship–ship and ship–bank interactions, and multiple rudders per propeller configurations.

One of the most significant impacts is shallow water, which affects hull related forces (resistance, turning forces, and turning moments) and performance of propellers and rudders, as discussed in Section 2. However, research on shallow water effects was mainly done for seagoing ships in restricted waterways. Gaps need to be filled for inland vessels at slow speed in shallow water considering the complex configurations, such as the hull tunnel and multiple rudders.

Since navigation conditions and ship particulars are different for inland vessels and seagoing ships, evaluation manoeuvres and related criteria should be adjusted. The existing manoeuvrability standards for inland vessels are less elaborate than those for seagoing ships. It is proposed that new manoeuvres and standards should be developed for shallow water and constrained waterways.

Inland vessels commonly have complex navigation conditions and design features. As the modular model can better present various manoeuvring related forces and moments including interactions among each component than the whole ship model, it is proposed to apply the modular model for inland vessel manoeuvrability prediction. While the whole ship model is suggested to be used with free running tests for ship controller development. No benchmark data has been publicly available as validation and verification materials for inland vessel manoeuvrability evaluation methods. Thus, benchmark test cases of model tests for inland vessels are suggested to be established.

As previously discussed, the research on ship manoeuvrability is increasing, but still mainly focusing on seagoing ships rather than inland vessels. Main challenges for inland vessel manoeuvrability analysis are how to estimate the hydrodynamic forces with given ship particulars in a specified waterway and how to evaluate their performance according to manoeuvrability criteria required by the navigation environment of that waterway. Therefore, studying hydrodynamic characteristics of different rudder configurations and innovation in inland vessel manoeuvrability criteria will be next research topics for the authors.

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