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The Roll Damping of High-Speed Craft in Waves

Oliviero CONSOLO[a], Lodewick HILLEGE[b] and Matteo BONCI[c,1],
[a] *Consulting Engineer, TURIN TECH, Italy*
[b] *Delft University of Technology, The Netherlands*
[c] *Project Manager Seakeeping, MARIN, The Netherlands*

Abstract. The main component of high-speed craft (HSC) roll damping is related to the hydrodynamic lift developed on the hull surface. This is very different from displacement type hull forms. However, the estimation of roll damping of HSC is often treated in the same manner as for larger and slower ships. Being able to model the roll of HSC correctly is of paramount importance in the prediction of the lateral component of acceleration of an impact at a roll angle in waves, or during a manoeuvre at high speed. These are phenomena that can have severe consequences on the comfort and safety of the crew on-board of HSC.
Three procedures meant to estimate the HSC roll damping were analyzed. The outcomes of these procedures were compared in terms of roll and lateral accelerations statistics of HSC sailing in irregular waves. The HSC motions were predicted by a 2D+t mathematical model. Differently from the majority of the state-of-art HSC seakeeping tools, which focuses only on the vertical impacts in head waves, in this work the roll was included in the simulations. The numerical results of the simulations were validated by means of free sailing model tests at beam and quartering irregular seas carried out at the Seakeeping and Manoeuvring Basin of MARIN.

Keywords. Roll damping, High Speed Craft, Seakeeping, FASTSHIP, Numerical simulations, Free sailing model tests

1. Introduction

Until the beginning of the 1960s, most of the research on planing craft was focused on the calm water behavior optimization, i.e. speed, resistance, trim and sinkage [1,2]. This area of focus led to the development of craft which were equipped with relatively low deadrise angles. The uncomfortable behavior of the craft experienced by the crew in form of excessive vertical motions and accelerations was until then not a limiting factor of the design. This attitude towards the seakeeping characteristics gradually changed throughout the late 1960s and beginning of the 1970s as fast transportation on water became more popular. The excessive motions and accelerations quickly appeared to be the most important limiting factor during the operations of these craft at sea and the interest in high-speed vessel seakeeping grew [3,4,5,6].

In 1976 [7] Martin and few years later Zarnick [8] presented a linear mathematical model to predict the heave and pitch accelerations of planing vessels in head seas. These

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[1] Corresponding Author; e-mail: m.bonci@marin.nl.

tools were based on the dropping wedge extending lamina analogy, firstly introduced in 1929 by Von Karman [12] and later extended by others such as Wagner [13] and Pierson [14]. Between 1985 and 1994 the section Ship Hydromechanics of the department of Naval Architecture of the Delft University of Technology (DUT) developed Zarnick's method into a fully non-linear, time domain strip theory computer code called FASTSHIP, for the calculation of the resistance, calm water running attitude and motions of fast planing monohulls in irregular head seas [9]. Using a 2D+t approach allows little computational effort simulations: as a result FASTSHIP can solve the motion equations faster than real-time. For this, it can be used in training simulators, design optimizations and proactive control systems development [10,11].

The high-speed vessel seakeeping design criteria has been restricted, until now, to situations where the vertical motions in head waves with the consequent violent slamming events are of main concern. For this reason, the mathematical models introduced previously were restricted only to the craft vertical motions in heave and pitch, neglecting roll. The prediction of the roll behavior of high-speed craft has proven to be rather difficult due to non-linear effects dictated by the roll exciting and damping forces. In order to evaluate the pressure distribution on a wedge after an impact at an oblique angle, a fully non-linear numerical approach can be followed [15,16]. Another way is a purely analytical method [17,18,20,21]. Consolo [19] in 2019 extended FASTSHIP for the transverse motion in roll using an analytical method. The model was validated for high deadrise hull sections, typically used for high-speed craft with good seakeeping qualities. FASTSHIP computations could provide a good match with the results of captive model tests on a 20 degrees deadrise wedge oscillating in roll, carried out at DUT. This new extended version of FASTSHIP is utilized in the present work to further investigate the prediction methods of high-speed craft roll damping, and their consequent effects on the motions in irregular waves.

2. The High-Speed Craft

The high-speed craft FRISC (Fast Raiding Interception and Special force Craft, Figure 1) was chosen as test case of this work. The FRISC is operated by the Royal Dutch Navy for fast patrol, rescue and interception operations; it was designed with a high average deadrise and large L/B ratio to be very manoeuvrable and seaworthy even in the highest sea states. High-speed craft as the FRISC must often operate in rough waves, where the manoeuvrability and the roll are important for the performance of the vessel and the safety of the crew. Low speed manoeuvrability in following breaking waves, sailing in the wake of bigger ships or turning at speed are just some of the situations in which the lateral dynamics plays a dominant role.

Table 1 reports the full scale characteristics of the FRISC. A 1:8 scale model of the FRISC was extensively tested in the Seakeeping and Manoeuvring Model Basin of MARIN in Wageningen. The model tests included free sailing tests in irregular waves and roll decay tests in calm water, at zero speed and in transit.

Figure 1. FRISC sailing in waves at a forward speed of 30 knots. Picture taken from https://magazines.defensie.nl.

Designation	Symbol	Value	Unit
Length between perpendiculars	L_{PP}	10.5	m
Breadth moulded on WL	B	2.35	m
Average draught	T_M	0.61	m
Average Deadrise	β_M	27	deg
Displacement mass in sea water	Δ	8.14	t
Transverse metacentric height	GMT	0.75	m
Mass radius of gyration around X-axis	K_{XX}	0.92	m
Natural period of roll	T_ϕ	2.46	s

Table 1. Main characteristics of the high-speed craft FRISC and stability data.

3. Mathematical Model

The mathematical model FASTSHIP used in this study is based on a time domain 2D strip theory and it is an extension of Zarnick [8] and Keuning's [9] work. The main assumption behind this model is that a planing hull can be seen as a series of wedges penetrating the water surface. Each section is simplified, being described by three points of intersection with the keel, the chine and the deck. This simplification greatly decreases the computational effort.

The mathematical model that is used to calculate the forces acting on each hull section is based on the concept of variation of added mass. The force acting on a section due to the variation of fluid momentum can be expressed as:

$$f_\xi = \frac{D}{Dt}(m_a V) \tag{1}$$

where m_a is the sectional added mass and V is the vertical velocity of the section. FASTSHIP makes use of a number of approximations and corrective coefficients in order to take into account several non-linear viscous effects, that are significant for high-speed craft. These include the vertical added mass approximation, bow wave generation, vertical cross-flow drag, flow separation at the transom, and buoyancy modification due to the flow separation at the chine. For a more detailed explanation of these model, the reader can refer to [8] and [9].

3.1. Roll implementation

One of the main restriction of the expanding lamina analogy is that added mass can be calculated only for wedges entering the water symmetrically, i.e. upright. The implementation of roll requires then another assumption. The adopted method is to divide the cross section into two parts: these two parts should be considered separately as two independent upright wedges having different deadrise angles, i.e. once adding and once subtracting the roll angle to the actual deadrise of the original wedge. This approach is visualized in Figure 2. Considering 1 the half hull on starboard side and 2 the port side, the equivalent deadrise angles are:

$$\beta_1 = \beta - \phi \text{ and } \beta_2 = \beta + \phi. \tag{2}$$

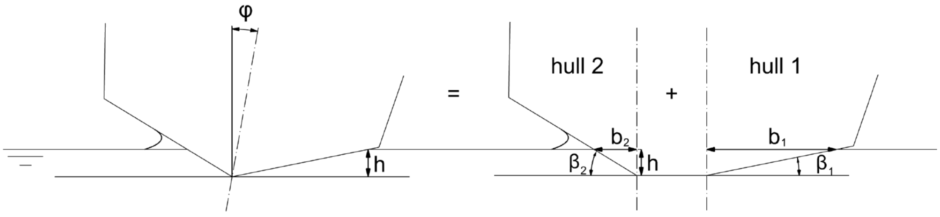

Figure 2. Equivalent deadrise assumption. The rotated wedge is split in two symmetrical wedges each having the corresponding equivalent deadrise. Forces and moments are computed separately for the two and combined with linear superposition.

The sectional hydrodynamic roll moment due to lift is formulated as:

$$M_{XLIFT} = L_{zPS} \cdot y_{PS} - L_{zSB} \cdot y_{SB} \tag{3}$$

where the side lift is expressed as a half of the total lift of the section ($L_{zSB} = \frac{L_z(\beta_1)}{2}$, starboard side). The formulation of the transverse arms y_{SB} and y_{PS} is semi-empirical and is taken from Smiley [22]: $y = EC_{pu}\frac{b}{4}$. E is 0.8 if the chine is under the water surface, or 1 in the dry chine region. The contribution of lift is located at $b/4$ of the wetted breadth of the half section. The velocity of the side section is then:

$$V_{SB} = V + \frac{b_{SB}}{4}\phi\omega_\phi; \text{ and } V_{PS} = V + \frac{b_{PS}}{4}\phi\omega_\phi. \tag{4}$$

4. Roll damping estimation

The total hydrodynamic roll damping of ships can be written, in a linear fashion, as:

$$K_p = (K_{0p} + K_{up} \cdot u)p \tag{5}$$

where p and u are the roll and forward speeds, respectively. The roll damping at zero speed K_{0p} is separated from the additional roll damping at speed K_{up}. In this section, three different calculation methods of these terms are presented for the high-speed craft FRISC.

4.1. Method 1: experimental roll decay

From roll decay tests, it is possible to retrieve the roll damping coefficients [23], given as a percentage of the critical roll damping equal to:

$$b_{cr} = 2 \cdot \sqrt{I_x \cdot C_\phi} \tag{6}$$

where I_X is the roll inertia (including the added inertia) and C_ϕ is the static restoring term. This restoring moment does not take into account the hydrodynamic lift. Table 2 summarizes the experimental results of the roll decays carried out on the FRISC model, in terms of percentage of critical damping.

Forward speed [kn]	b_{cr} [Nms]	Average roll damping [%]
0	4.10E+04	4.5
10	4.10E+04	7.5
20	4.36E+04	19

Table 2. Experimental roll decay tests results of the FRISC model, expressed in full scale.

4.2. Method 2: Correction of the restoring moment

The total roll restoring moment of high-speed craft is composed by the hydrostatic and hydrodynamic components:

$$C^*_\phi = (\Delta g \cdot GM_T + K_\phi)\phi \tag{7}$$

According to this second method, the critical roll damping of Equation 6 is modified keeping into account the hydrodynamic component of the roll restoring moment: a different restoring moment could change the critical roll damping, and thus the roll damping of the vessel. The coefficient K_ϕ is estimated using FASTSHIP, by setting the FRISC at several constant heel angles and forward speeds.

4.3. Method 3: Numerical roll damping estimation

The roll damping can also be estimated using a fully numerical approach, i.e. using the mathematical model FASTSHIP described in Section 3. The FRISC is forced to oscillate in roll at certain roll periods and amplitudes, at a constant forward speed. The roll damping prediction of FASTSHIP was validated by means of oscillating captive model tests on 20 degrees deadrise wedge, carried out at DUT [19]. Figure 3 shows a picture

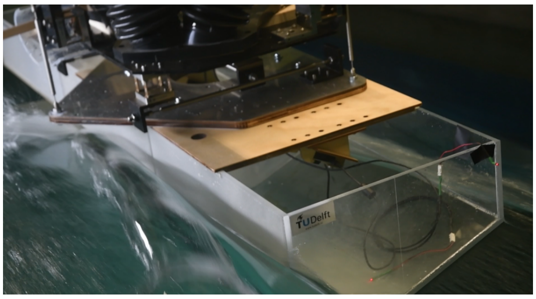

Figure 3. 20 deg deadrise wedge oscillating in roll. Model test run taken at Delft University of Technology, Ship Hydromechanics Laboratory (Towing Tank)

taken during the model test campaign. The resulting total roll damping coefficient K_p is calculated from the hydrodynamic roll moment M_X divided by the roll speed $\phi\omega_\phi$.

Table 3 summarizes the speed and the roll amplitudes and periods of the FRISC roll oscillation simulations carried out using FASTSHIP. The roll frequencies were chosen close to the roll natural frequency of the FRISC, i.e. in the frequency range where the roll motion is higher. The roll damping estimated by FASTSHIP is less sensitive to the variation of the roll amplitude than to the variation of roll period.

Roll oscillation characteristics	Value	Unit
Forward speed	10, 15, 20, 25, 30	kn
Period	2.21, 2.46, 2.95	s
Amplitude	8, 12, 16	deg

Table 3. Conditions simulated for the estimation of the FRISC roll damping by means of numerical roll oscillation tests.

4.4. Results of the roll damping estimation

In Figure 4 the roll damping coefficients estimated with the three different methods are plotted as function of Froude number of the FRISC. The first two methods give very similar results. Although the total hydrodynamic roll restoring moment can be rather different than the hydrostatic one estimated at zero speed, this has little effect on the estimation of roll damping. This is also due to the fact that the roll damping of the FRISC is only a little percentage of the critical damping: between 4% at zero speed to 20% at 20 kn, as reported in Table 2. The roll damping estimated using FASTSHIP is instead rather different than the experimental one, especially at high speed (Froude 1.0 and higher).

Figure 5 instead shows the comparison between the roll decay results in calm water carried out at the SMB and predicted by FASTSHIP. The roll oscillation decay $\delta\phi_i = \phi_{i+1}/\phi_i$ is plotted as function of the roll amplitudes estimated on a 10 knots run for each i-th oscillation. The agreement between experiments and FASTSHIP is satisfactorily.

5. Free sailing craft simulations

The mathematical model described in Section 3 is used to simulate the motions of the FRISC freely sailing in irregular waves. In these free sailing FRISC simulations, the roll

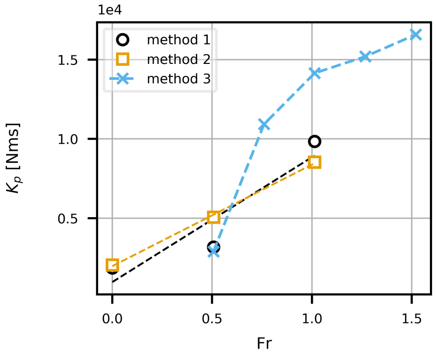

Figure 4. Total roll damping coefficients of the FRISC estimated from experimental roll decay tests (method 1), from experimental decays but correcting the roll restoring moment keeping into account the hull lift component (method 2), and from numerical FASTSHIP computations (method 3).

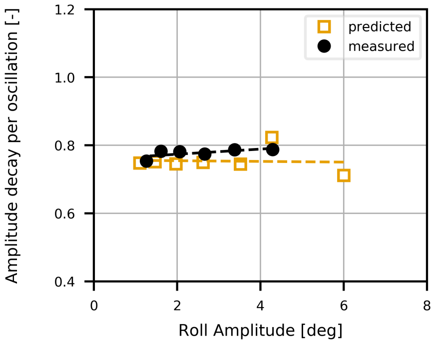

Figure 5. Comparison of experimental and predicted by FASTSHIP roll decay tests in terms of roll amplitude decay per oscillation.

damping moment is not directly computed using the formulation of Section 3.1, but it is implemented in a parametric fashion, employing the roll damping coefficients estimated in Section 4 (see Equation 5), improving computational efficiency.

The conditions investigated are reported in Table 4. The results of the simulations are compared with the model tests carried out at MARIN. the results are presented in terms of the standard deviation of roll and lateral acceleration measured at the wheel of the FRISC. Only method 1 and method 3 are compared with the experiments, since method 2 estimated a very similar roll damping than method 1. Figure 6 shows the standard deviation of roll and lateral acceleration measured at the wheel as function of the wave heading, for the three different sea states investigated.

The two methods of roll damping estimation gave very similar results with only marginal discrepancies above 20 kn, due to the larger difference between the experimental roll decay tests and FASTSHIP. However, for each conditions, both the numerical roll and the lateral accelerations are under-predicted.

Heading [deg]	Sea State	Significant wave height, Hs [m]	Wave modal period, Tp [s]	U [kn]
60, 90, 135 (stern-quartering, beam, bow-quartering)	3	1.00	6.0	32
	4	2.00	6.5	15
	5	3.25	7.0	10

Table 4. Conditions investigated for the free-sailing FRISC in irregular waves.

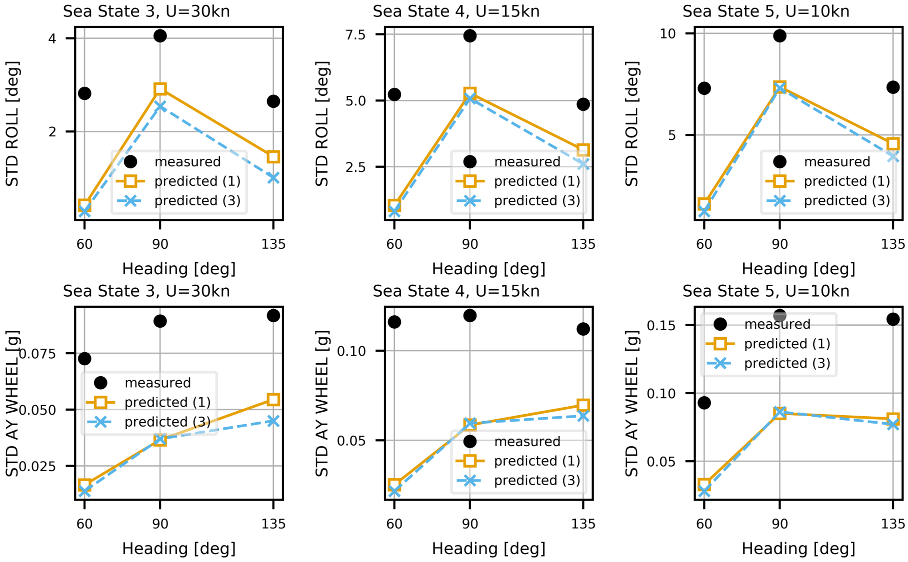

Figure 6. Standard deviation of roll (top) and lateral accelerations (bottom) for all the conditions investigated, as function of wave heading. Comparison between experiments and predictions of FASTSHIP.

6. Conclusions

The roll damping of the high-speed craft FRISC was implemented in the numerical simulations in terms of the calm water roll damping coefficients. The predicted roll and lateral acceleration statistics were compared with the free sailing model tests carried out at MARIN. The results showed that the 2D strip theory can be used to substitute in a reliable way the experimental roll decay tests.

However, both the roll and the lateral accelerations were under-predicted by the numerical simulations. The roll damping was estimated in calm water: the calm water damping can be significantly different from the one in waves, especially for small craft. A possible successive step of the research will be the direct computation of the roll damping without the use of linearized coefficient. In that case, the hydrodynamic roll damping should be evaluated at each section of the vessel with the actual submergence in waves. Moreover, a wrong prediction of the radiation and diffraction damping in waves could also influence the roll of the high-speed craft. In waves, the vessel can be excited at many different frequencies: a complete characterization of the frequency behaviour (retardation function) might be estimated using FASTSHIP, and then be applied in free sailing ship simulations.

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