

Hydrodynamic Performance and Self–Stabilizing Mechanism of an Axially Symmetric Bow in Severe Sea Conditions

Le Tuan Son*, Nguyen Dinh Hai, and Pham Trung Duc

Abstract: Deep bow immersion (bow–diving) during operation in severe cross seas ($\varphi \neq 180^\circ$) poses significant survivability risks to surface vessels due to asymmetric waterplane variations that generate hazardous transverse hydrodynamic forces and heeling moments. This paper investigates the hydrodynamic performance of a novel axially symmetric bow design comprising a cylindrical main hull and a coaxial cylindrical bulbous bow. Numerical simulations combining Smoothed Particle Hydrodynamics (δ –SPH) for severe wave impact and RANS–VOF (ANSYS Fluent) for calm–water towing resistance were conducted. Grid convergence was established using the Grid Convergence Index ($GCI \leq 1.23\%$), and the SPH formulation was independently validated against experimental benchmark data for dynamic water entry. Results under prescribed forced–immersion conditions ($z_{imm} = L/4$) demonstrate that the axially symmetric bow effectively suppresses asymmetric pressure distributions, reducing the maximum transverse force coefficient to $P_y \leq 0.016\Delta$ (a 5.5 – fold reduction compared to conventional V–shaped bows under head seas). Parametric analysis reveals a strong passive restoring moment (m_{zy}) increasing the bulb length l from $L/24$ to $3L/24$ enhances m_{zy} by over 400% (from 0.20 to 0.85 at $\theta = 60^\circ$), whereas doubling the bulb diameter d from $0.25D$ to $0.5D$ yields a moderate 30% increase. While calm–water resistance increases by 21% at high speeds (22 knots) the resistance penalty is negligible ($\leq 2\%$) at operating speeds below 13 knots. These findings confirm a favorable localized geometry–driven self–stabilizing tendency, providing physical insights for the design of specialized offshore and arctic service vessels.

Keywords: Axially symmetric bow, bow–diving, hull form optimization, hydrodynamic stability, smoothed particle hydrodynamics.

1. INTRODUCTION

Analysis of maritime casualties within the global fleet indicates that one of the primary causes of ship losses is adverse weather conditions, associated with the occurrence of significant wave–induced loads acting on the ship’s hull structure. Under the influence of these loads, phenomena such as capsizing and structural failure of the hull frequently occur [1–3]. The optimization of hull form, particularly in the bow region, aimed at improving seakeeping performance, enhancing motion stability in waves, and ensuring navigation safety, has been extensively investigated by numerous researchers both domestically and internationally [4–5].

In studies [6, 7] various structural concepts have been proposed to enhance ship performance under wave and wind conditions, through the adoption of specific

variable waterline section geometries, as well as the application of advanced engineering and technological solutions in the stem, bilge lines, and the above–water bow structure. In addition, several well–known bow design solutions have been developed including: Axe–Bow, Ax–Bow, LEADGE–Bow, SP–Bow, Beak–Bow, X–Bow, as well as the bow design concept developed by STX OSV Design [8, 9, 14, 15, 16, 17].

Newly developed hull forms are characterized by features such as inward flare (tumblehome) in the bow region and a reverse stem, enabling a wave–piercing bow configuration. These design characteristics allow the bow to penetrate waves more effectively, thereby significantly improving seakeeping performance in severe sea states [6, 18]. Given the wide range of naval architectural and structural solutions that have been proposed, it is not feasible to present each configuration in detail within the

Manuscript received April 22, 2026; received in revised form May 23, 2026; accepted September 3, 2026; available online September 30, 2026.

Le Tuan Son, Nguyen Dinh Hai, and Pham Trung Duc are with the Faculty of Navigation, Vietnam Maritime University, Haiphong, Vietnam (e-mails: sonlt.hh@vimaru.edu.vn, nguyendinhhai@vimaru.edu.vn, and phamtrungduc@vimaru.edu.vn).

* Corresponding author.

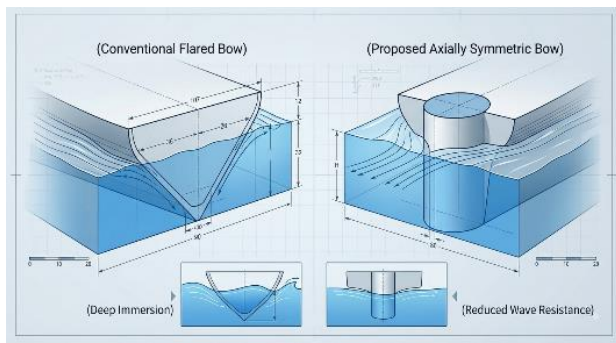


Fig. 1. Geometric cross-sectional comparison of a conventional flared bow and proposed axially symmetric bow during deep immersion.

Table 1. Scaling relationships based on Froude similarity ($\lambda=100$).

Parameters	Scaling ratio	Model value	Full-scale value
Length (L)	$\lambda=100$	1.68 m	168.0 m
Velocity (V)	$\lambda^{0.5} = 10$	0.5 m/s	5.0 m/s
Time (t)	$\lambda^{0.5} = 10$	t_m	$10 \times t_m$
Force (F)	$\lambda^3 = 10^6$	F_m	$10^6 \times F_m$
Moment (M)	$\lambda^4 = 10^8$	M_m	$10^8 \times M_m$
Froude Number (F_r)	1.0	0.123	0.123
Reynolds Number (R_e)	$\lambda^{1.5} = 1000$	8.41×10^5	8.41×10^8

scope of this paper.

However, while these standard advanced bows excel in moderate-to-heavy seakeeping performance, they introduce vulnerabilities during extreme "bow-diving" or "bow-burying" phenomena in severe storms. When an asymmetric flared or sharp bow is deeply submerged under green water at an oblique wave heading ($\varphi \neq 180^\circ$), the instantaneous submerged waterplane area becomes highly non-symmetrical. This asymmetry induces massive transverse hydrodynamic forces P_y and an associated twisting or overturning heeling moment (M_{zy}) which significantly destabilizes the ship's steering orientation and heightens the risk of broaching-to or sudden capsizing.

The axially symmetric bow configuration proposed in this study operates on a fundamentally different hydrodynamic principle aimed at ultimate survivability. Both the main forward hull section and the bulbous bow are engineered as bodies of revolution (cylindrical forms) symmetrical about the longitudinal axis.

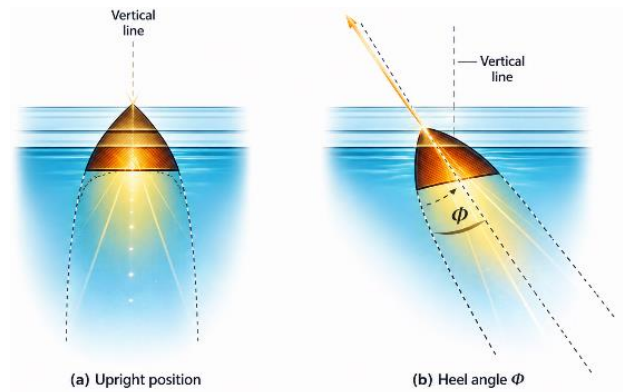


Fig. 2. Schematic of green water shipping (deck wetness): (a) head sea condition (bow facing the waves), (b) oblique sea condition (waves acting on the beam/quarter).

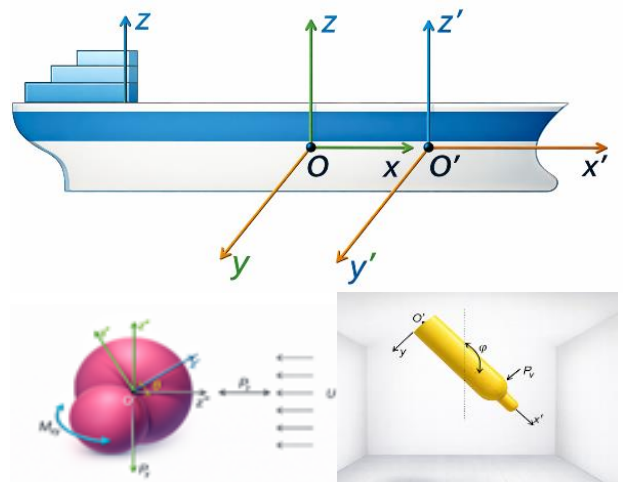


Fig. 3. Coordinate systems and directions of hydrodynamic forces: (a) Coordinate systems. (b) Bottom view of the model at $\phi=0$. (c) Front view.

The underlying hydrodynamic mechanism can be explained through pressure distribution integrated over the immersed geometry. Because the hull surface is perfectly circular in cross-section, the elemental hydrodynamic pressure forces acting normal to the submerged body during deep immersion always converge toward the central axis of symmetry. As a result, even when the bow is completely buried under a wave at an angle, the net horizontal destabilizing force is minimized, and the generation of a localized overturning heeling moment is theoretically eliminated. Furthermore, when the vessel experiences a coupled roll or heel angle (θ) during immersion, the symmetric flow redirection around the cylindrical bulbous bow creates a localized asymmetric upward buoyancy/dynamic lift distribution that naturally induces a positive hydrodynamic restoring moment. This geometry-driven restoring torque acts as a passive self-stabilizing mechanism, forcing the bow back toward the vertical center plane. Thus, the novelty of this design lies not in wave-slatting reduction at transit speeds, but in providing a passive aerodynamic–

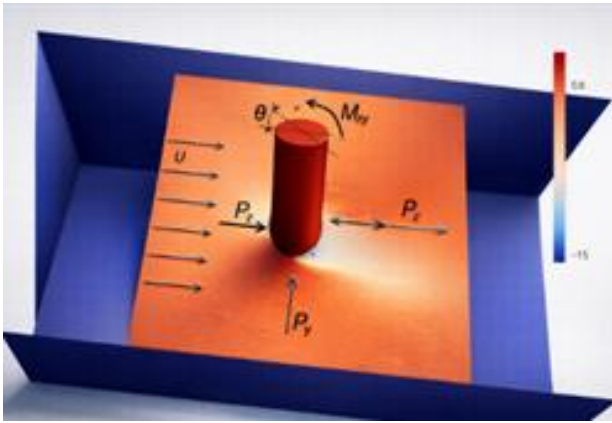


Fig. 4. Bow model for the experiment.

Table 2. Summary of SPH computational parameters utilized in the current study.

Parameters	Value/Formulation
Kernel function	Cubic spline
Initial particle spacing (Δx)	0.005m
Smoothing length (h)	0.0100m ($2\Delta x$)
Total number of particles	$\approx 1.25 \times 10^6$
Boundary treatment method	Modified dynamic boundary conditions (mDBC)
Turbulence model	Sub-particle scale (SPS) smagorinsky
Time-step control	Dynamic CFL condition $\Delta t_{min} \approx 10^{-5}s$
Artificial viscosity(α)	0.02

like stabilization during critical hull–submersion states in survival conditions.

However, comprehensive reviews and comparative analyses have been conducted by various authors and can be found in [6, 8]. It should be emphasized that the safety level of a vessel must be assessed based on its behavior under extreme operating conditions. One of the most critical scenarios is the phenomenon of bow diving (bow burying) under conditions of severe ship motions in heavy seas. In such cases, a complex system of hydrodynamic forces and moments arises in the bow region, which may cause the vessel to broach-to (turn beam-on to the waves) or even lead to capsizing. The force system acting on the bow during bow immersion has been investigated and described in detail in [5, 19, 20].

To fundamentally resolve this physical mechanism, this study proposes an axially symmetric bow, where in both the main fore body hull and the bulbous bow are designed as coaxial bodies of revolution (circular cylindrical geometry). As illustrated in Fig. 1, because the hydrodynamic pressure vectors acting on a circular cylinder act normal to the surface, they intersect at the longitudinal center axis. Under forced immersion, the

Table 3. Principal particulars and hydrostatic parameters of the prototype ship.

Parameters	Symbol	Value	Unit
Ship type	--	Product Tanker	--
Length between perpendiculars	L_{PP}	168.0	m
Moulded breadth	B	27.4	m
Characteristic bow diameter	D	12.0	m
Design draft	T	9.5	m
Mass displacement	Δ	31,200	tonnes
Vertical center of gravity	KG	8.65	m
Initial metacentric height	GM_o	1.45	m
Evaluated immersion depth	z_{imm}	$\frac{L_{PP}}{4} = 42.0$	m

Table 4. Grid convergence verification metrics for ANSYS Fluent resistance simulation C_T .

Grid metric	Grid 3 (Coarse)	Grid 2 (Medium)	Grid 1 (Fine)
Total cell elements	1.2×10^6	2.85×10^6	5.5×10^6
Refinement ratio r	Baseline	1.414	1.414
Target output $C_T \times 10^{-3}$	4.312	4.195	4.168
Relative error ϵ_{21}	--	--	0.648%
Calculated GCI_{fine}	--	--	0.81%

transverse force component is minimized, and the submerged body generates a passive restoring moment (m_{zy}) that acts to push the hull back toward the upright equilibrium state.

This article delivers three core contributions:

(1) *Geometric innovation*: Establishing the physical mechanism of an axially symmetric forebody in eliminating asymmetric immersion loads.

(2) *Quantitative parametric insights*: Quantifying the non-linear relationship between bulb length (l), bulb diameter (d), heel angle (θ), and the resulting restoring moment (m_{zy}).

(3) *Hydrodynamic evaluation*: Assessing both severe wave-immersion loads using validated δ -SPH and calm-water resistance penalties using RANS-VOF CFD.

2. NUMERICAL SIMULATION

To translate the numerical simulation results of the physical model to full-scale prototype predictions, Froude similarity law was applied, as gravity and inertia

Table 5. Particle resolution verification metrics for SPH bow-diving simulation P_y .

Particle metric	Case 3 (Coarse)	Case 2 (Medium)	Case 1 (Fine)
Particle spacing Δx	0.0106 m	0.0075 m	0.0053m
Total fluid particles	0.44×10^6	1.25×10^6	2.91×10^6
Peak force P_y (N)	125.4	131.8	133.1
Relative error ϵ_{21}	--	--	0.986%
Calculated GCI_{fine}	--	--	1.23%

Table 6. Quantitative comparison between δ -SPH and EFD benchmark data [30] for wedge water entry 30° deadrise angle).

Hydrodynamic benchmark parameters	Benchmark EFD [30]	Present δ -SPH	Relative Error
Peak impact pressure coefficient $C_{p,max}$	4.85	4.71	2.88%
Dimensionless time to peak pressure $\tau = \frac{v_0 t}{d}$	0.128	0.132	3.125%
Max vertical dynamic impact force $F_{z,max}$ (N)	245.2	251.8	2.69%
Maximum spray root elevation $\frac{z_{spray}}{d}$	1.42	1.39	2.11%

forces dominate free-surface wave-structure interactions during severe bow-diving events. The geometric scale factor is defined as $\lambda = \frac{L_s}{L_m} = 100$ where subscripts s and m denote full-scale ship and model-scale, respectively. Based on Froude scaling laws, the conversion factors for kinematic and dynamic quantities are defined in Table 1.

The Froude number is preserved across scales ($F_{rm} = F_{rs} = 0.123$ at $V_m = 0.5 \frac{m}{s}$). However, a significant Reynolds number mismatch exists between the model scale ($R_{em} = 8.41 \times 10^5$ and full scale ($R_{es} = 8.41 \times 10^8$), spanning three orders of magnitude ($\frac{R_{es}}{R_{em}} = 10^3$). During severe bow-diving under forced immersion conditions, the dominant hydrodynamic actions—specifically the wave impact dynamic pressure,

Table 7. Quantitative validation metrics between present CFD and EFD data for the KCS hull.

Froude number (F_r)	Experimental $C_T \times 10^{-3}$ (EFD)	Present CFDC $_T \times 10^{-3}$	Relative error (%)
0.16	3.921	3.854	1.71%
0.20	4.015	3.942	1.82%
0.24	4.128	4.051	1.86%
0.26 (Design speed)	4.234	4.143	2.15%
0.28	4.385	4.276	2.48%

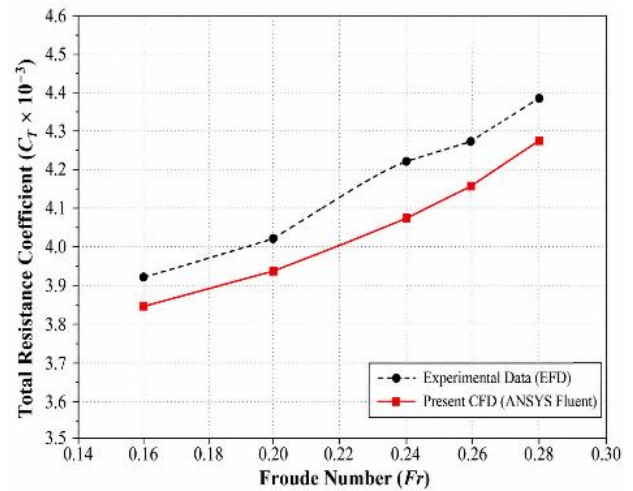


Fig. 5. A visual comparison between simulation and experimental results.

transverse dynamic forces (P_y) and the passive restoring moment (m_{xy})—are predominantly driven by localized normal pressure distributions over the circular geometry rather than skin-friction shear stress. Because inertial and gravitational wave forces strictly follow Froude scaling, Froude-based conversion provides reasonable physics-based estimates for dynamic restoring loads. Nevertheless, the substantial Reynolds number discrepancy inevitably introduces viscous scale effects at $R_{em} = 8.41 \times 10^5$, boundary layer flow over the cylindrical forebody may experience localized laminar separation, whereas full-scale flow $R_{es} = 8.41 \times 10^8$ is fully turbulent with delayed separation. Underestimation/Overestimation: Skin-friction resistance coefficients C_f do not follow Froude scaling and drop significantly at higher R_e (as described by ITTC-1957 friction lines). Consequently, calm-water resistance penalties predicted at model scale cannot be directly scaled without friction extrapolation (e.g., ITTC 1978 Performance Prediction Method). Therefore, while Froude scaling effectively isolates the localized geometric self-stabilizing mechanisms under dynamic water entry, full-scale predictions of absolute drag and viscous flow separation require caution and dedicated 6-DOF turbulent extrapolation in future studies.

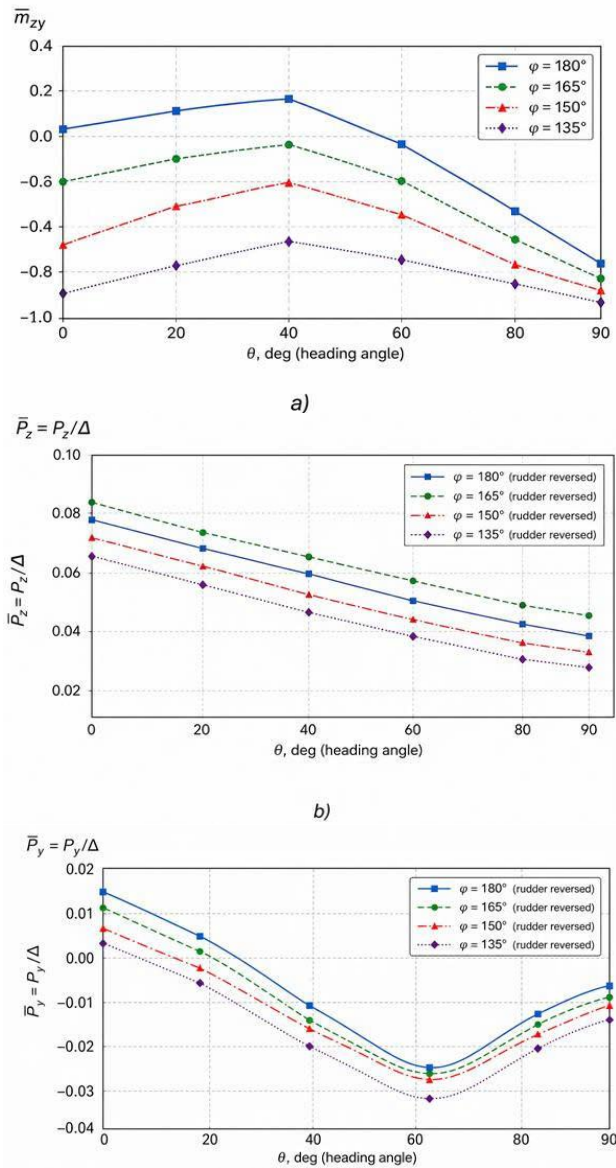


Fig. 6. Experimental results for a conventional bow with bulwark, with flare angle $\beta=45^\circ$: (a) Hydrodynamic moment. (b) Longitudinal force. (c) Transverse force.

To determine the hydrodynamic characteristics of the bow section, numerical simulations were conducted using the Smoothed Particle Hydrodynamics (SPH) method [21–24]. This method belongs to the class of mesh-free approaches, in which the fluid domain is discretized into a set of particles. The fluid motion is governed by the Navier–Stokes equations, assuming a viscous and weakly compressible fluid.

The fluid is modeled as weakly compressible, and the governing Navier–Stokes equations are solved using a Cubic Spline kernel function to calculate particle interactions. The smoothing length (h) is dynamically coupled to the initial particle spacing (Δx) with a ratio of $h/\Delta x = 2$. To mitigate unphysical pressure oscillations common in weakly compressible SPH formulations, a δ – SPH artificial diffusion term ($\delta = 0.1$) was

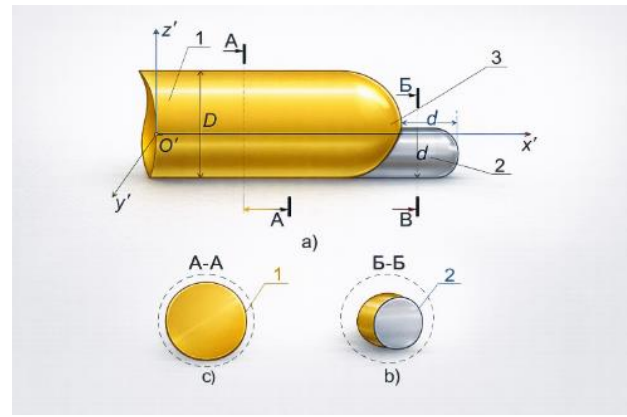


Fig. 7. Proposed new bow form: (a) General arrangement. (b) Cross-section of the main hull. (c) Cross-section of the bulbous bow.

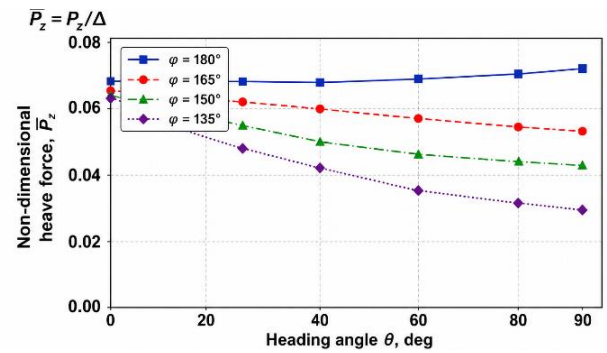


Fig. 8. Relationship between longitudinal hydrodynamic force and heel angle for different heading angles.

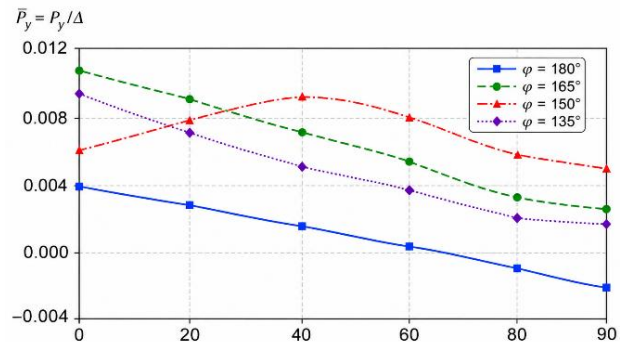


Fig. 9. Relationship between transverse force and heel angle for varying heading angles.

introduced into the continuity equation. Solid boundaries, including the complex 3D bow geometry and the virtual tank walls, are treated using the Modified Dynamic Boundary Conditions (mDBC), which enforces a zero-penetration condition while accurately estimating boundary pressure profiles. The high-Reynolds-number turbulent structures generated during the high-velocity bow-diving impact are resolved using a Sub-Particle Scale (SPS) turbulence model based on the Smagorinsky subgrid scale formulation.

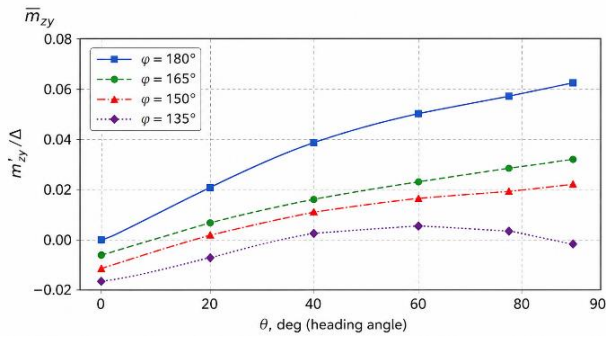


Fig. 10. Relationship between hydrodynamic moment and heel angle for varying heading angles.

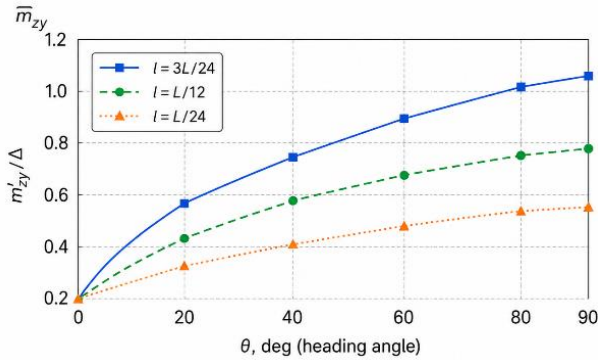


Fig. 11. Effect of bulbous bow length on the hydrodynamic restoring moment – Case 1.

Within the framework of this method, the hydrodynamic field is approximated by a set of discrete particles and all required physical properties $F(r)$ are determined based on the following condition

$$F(r) = \int F(r') W(r - r', h) dr' \quad (1)$$

where, h is the smoothing length (smoothing radius); $W(r - r', h)$ is the smoothing kernel (smoothing function)

In discrete form, the above equation can be expressed as follows

$$F(r_a) \approx \sum_b F(r_b) W(r_a - r_b, h) \Delta v_b \quad (2)$$

where, indices a and b correspond to individual particles, Δv_b is the volume of the neighboring particle.

Further details regarding the implementation of this method can be found in [23, 24].

During the simulations, a bow model (scale 1:100) was subjected to vertical immersion to various depths in a fluid flow with velocity v , in order to reproduce the bow diving (bow immersion) phenomenon (see Figures 2, 3, and 4).

The relative inflow velocity around the bow was determined according to the approach presented in [20], based on the modeling of ship pitch motion under head-sea bow immersion conditions. In the numerical experiments, this velocity was varied across different scenarios; the results reported in this study correspond to a value 0.5 m/s . The 3D geometry of the bow model

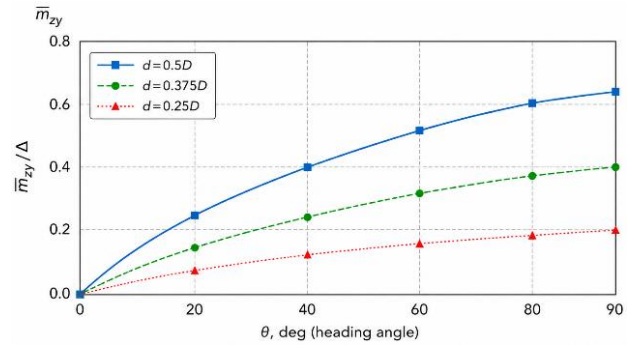


Fig. 12. Effect of bulbous bow diameter on the hydrodynamic restoring moment – Case 2.

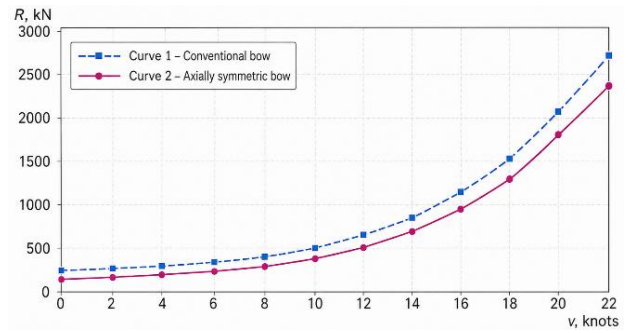


Fig. 13. Towing resistance comparison between the conventional bow and the proposed axially symmetric bow.

was developed using the NURBS (Non-Uniform Rational B-Splines) modeling software Rhinoceros 3D.

During the simulations, three coordinate systems were employed (Figure 2). The coordinate system $Oxyz$ is body-fixed to the ship, where: the Ox axis is directed along the ship's longitudinal axis (toward the bow), the Oy axis is directed toward the starboard side, and the Oz axis is oriented vertically upward from the keel to the deck. The origin O is located on the centerline (diametric plane), at a height equal to half the molded depth above the baseline, and at the midship section.

The coordinate system $O'x'y'z'$ has the same structure as $Oxyz$, but its origin is shifted forward toward the bow by a distance equal to $\frac{1}{6}L$, where L is

the ship length. The coordinate system $O'x'y''z''$ coincides with $O'x'y'z'$ under zero heel conditions. When the vessel experiences a heel angle, the angles between the axes $O'y'$ and $O'y''$, as well as between $O'z'$ and $O'z''$ are both equal to θ , where θ denotes the ship's heel angle.

During the experiments, the following parameters were prescribed: the heel angle θ , corresponding to roll motion about the longitudinal axis of the model, and the heading (drift) angle φ . The bow model used in the experiments is shown in Figure 3.

For fixed values of θ and φ , the following quantities were determined: P_z – longitudinal hydrodynamic force (along the flow direction) acting along the $O'z''$ axis; P_y – transverse hydrodynamic force acting along the $O'y''$ axis; M_{zy} – hydrodynamic moment. For simplicity, the double-prime notation of the coordinate system is omitted hereafter. The positive direction of the force P_z is defined as opposite to the direction of the $O'z''$ axis.

For convenience in analysis, these quantities are presented in dimensionless form according to the following expressions

$$\overline{P_z} = \frac{P_z}{\Delta} \quad (3)$$

$$\overline{P_y} = \frac{P_y}{\Delta} \quad (4)$$

$$\overline{m_{zy}} = \frac{M_{zy}}{M_{maxM}} \quad (5)$$

where, P_z is the longitudinal force acting on the model at a heel angle θ , P_y is the force perpendicular to the flow (transverse force), Δ is the displacement (weight displacement) of the model, M_{zy} is the hydrodynamic moment, and M_{maxM} is the maximum righting moment, determined in accordance with [21, 25].

According to the Rules of Classification Societies [25], within the general requirements for intact stability, the static stability curve (righting lever curve, GZ curve) must satisfy several criteria, including the required area under the positive portion of the curve and the maximum righting lever l_{max} as specified in Section 2.2.1 of [25]. The maximum righting lever is determined as follows

$$\frac{M_{max}}{\Delta_H} = l_{max} \quad (6)$$

where, Δ_H is the ship's displacement, and M_{max} is the maximum righting moment corresponding to the peak value of the static stability curve.

Taking into account the model scale, the maximum righting moment of the model, M_{max} satisfying the requirements of [25], is determined as follows

$$M_{maxM} = l_{maxM} \Delta \quad (7)$$

where, l_{maxM} is the maximum righting lever of the model, and Δ is the displacement of the model.

During the study, the following parameters were varied (parametric study): bulb length (bulbous bow length) and bulb diameter. Three bow configurations were investigated, corresponding to bulb diameter ratios of $0.5D$; $0.375D$; and $0.25D$; where D is the main hull diameter.

To rigorously assess and quantify the spatial discretization errors inherent in both the grid-based RANS (ANSYS Fluent) and particle-based SPH solvers, a formal verification procedure was performed using the GCI method based on Richardson extrapolation. Following the ITTC guidelines [26], three systematically refined computational setups were developed for each solver using a constant refinement ratio

$$r = \frac{h_{coarse}}{h_{fine}} = 1.414 \quad (8)$$

The numerical error estimator GCI_{fine} which quantifies the percentage by which the computed solution is away from the asymptotic numerical value, is mathematically formulated as

$$GCI_{fine} = \frac{F_s |\epsilon|}{r^p - 1} \quad (9)$$

where, F_s is the safety factor (set to 1.25 for three-grid comparisons), $\epsilon = (\phi_2 - \phi_1) / \phi_1$ is the relative error between consecutive grids p and represents the formal local order of accuracy.

For the ANSYS Fluent resistance simulations (Table 2), the target scalar variable selected for verification was the total ship resistance coefficient C_T at the maximum speed of 22 knots. The quantitative grid error verification metrics are summarized in Table 4.

Additionally, for the SPH bow-diving dynamic calculations, the peak dynamic transverse hydrodynamic force P_y during the deep-immersion phase was selected as the critical verification parameter across three particle spacing values Δx . The quantified numerical errors are presented in Table 5.

Since the primary conclusions regarding transverse force P_y suppression and restoring moment m_{zy} generation are derived from SPH simulations, the δ -SPH formulation was independently validated against the experimental water entry of a rigid wedge with a 30° deadrise angle [30]. The SPH initial particle spacing was set to match the fine resolution ratio $\Delta x = 0.0053$ m. The maximum relative percentage error Err_{rel} is defined as

$$Err_{rel} = \frac{\phi_{SPH} - \phi_{EFD}}{\phi_{EFD}} \quad (10)$$

where ϕ represents the target dynamic hydrodynamics scalar parameter (peak pressure, peak force coefficient, or non-dimensional peak time).

As shown in Table 6, the maximum discrepancy across all evaluated dynamic impact metrics remains bounded below 3.13%, confirming the mathematical fidelity of the SPH formulation in capturing violent free-surface splash-up, transient pressure spikes, and localized wave impact forces during deep bow-immersion events.

As demonstrated by the quantitative Verification and Validation (V&V) results, the GCI_{fine} for both distinct solvers remains well below the strict 2% threshold. The small values of the numerical uncertainty indices confirm that the computed total resistance and dynamic lateral forces have achieved spatial asymptotic convergence. Consequently, the numerical errors stemming from grid and particle discretization are rigorously quantified and minimized, establishing the mathematical reliability of the computational findings reported in this study.

After verifying the grid and quantifying spatial discretization errors through the GCI method detailed above, a formal validation study must be executed to demonstrate that the selected mathematical formulations

(RANS equations coupled with the SST $k-\omega$ model and VOF method) can accurately replicate real-world ship hydrodynamic resistance phenomena. To achieve this, the modern benchmark hull form of the KRISO Container Ship (KCS) was selected for validation [27,29]. The KCS model was specifically chosen because it serves as an internationally recognized ITTC benchmark hull operating at a representative Froude number ($F_r = 0.26$) featuring complex bulbous bow wave-making mechanisms and free-surface flow interactions that closely mirror the hydrodynamic challenges of the proposed design. The numerical domain, boundary conditions, cell-clustering strategies near the free surface, and non-dimensional wall parameters ($y^+ \approx 1$) were mapped exactly from the operational configuration used for the proposed axially symmetric bow geometry model. The computed total resistance coefficients C_T derived from the Fluent simulations were validated against high-quality Experimental Fluid Dynamics (EFD) towing tank data established by KRISO and endorsed by the International Towing Tank Conference (ITTC) [27,28].

The validation was conducted across a broad operational speed range, corresponding to Froude numbers F_r from 0.16 to 0.28. The total resistance coefficient is expressed as

$$C_T = \frac{2R_T}{\rho S V^2} \quad (11)$$

where, R_T is the total resistance, ρ is the fluid density, S is the wetted surface area, and V is the ship velocity.

As presented quantitatively in Table 7 and plotted in the comparative curves of Figure 5, the RANS simulation demonstrates an exceptionally high level of correlation with the physical tank test results. The relative error remains bounded below a maximum discrepancy of 2.48% across the entire speed spectrum, with a 2.15% error at the nominal design speed ($F_r = 0.26$).

The solver accurately captures both the frictional components within the boundary layer and the wave-making resistance components reflected in the free-surface Kelvin wave profiles. This formal validation establishes that the numerical settings, mesh strategy, and turbulence modeling applied in this study are physically realistic and highly reliable for predicting the resistance characteristics of the proposed bow form.

3. AXIALLY SYMMETRIC BOW FORM

The results of the hydrodynamic interaction analysis between a conventional bow and waves under bow immersion (bow burying) conditions are presented in dimensionless form in Figure 5, where θ is the heel angle (rotation about the longitudinal axis) and φ is the heading (course) angle. The results shown in Figure 6 correspond to the case where the bow is submerged to a length of $L/4$, where L is the ship length. A detailed analysis of the hydrodynamic characteristics of a conventional bow under these conditions is provided in

[21, 22, 25].

It can be observed that, for the conventional bow configuration, the acting moment is predominantly a heeling moment in most cases, which has an adverse effect on ship safety. In addition, significant transverse forces develop on the conventional bow (Figure 6c), tending to turn the vessel beam-on to the waves (broaching-to), which may ultimately lead to capsizing. Meanwhile, the vertical forces P_z acting on the bow (Figure 6b) significantly reduce the transverse metacentric height h_0 (GM), and may even reduce it to zero [5, 19, 20].

Under such conditions, considering that the point of application of the hydrodynamic resultant force is offset from the ship's centerline (diametric plane), even a relatively small force P_z can generate a dangerous heeling moment, potentially resulting in capsizing.

To improve ship safety in rough sea conditions, the authors propose a new bow form capable of reducing hydrodynamic forces and moments during bow immersion [27]. This configuration is illustrated in Figure 7, with the following notation: 1 – main hull; 2 – bulbous bow; 3 – transition section between the main hull and the bulb; D – main hull diameter; d – bulb diameter; l – bulb length.

In the proposed design, both the main hull and the bulbous bow are configured as circular cylinders, arranged such that the lower part of the bulb is located on the baseline. Compared to a conventional bow (under the same projected area normal to the flow), this design exhibits significantly reduced hydrodynamic loads due to its streamlined geometry. In particular, the use of a circular cylindrical form in the bow region helps to minimize the generation of capsizing moments, as the resultant hydrodynamic pressure force remains nearly aligned with the longitudinal axis of symmetry. In other words, dangerous heeling moments are almost eliminated, even when the bow is deeply immersed in waves.

For the proposed bow configuration, the component of hydrodynamic force that tends to induce broaching is significantly reduced. This is attributed to the axially symmetric bow form, which fundamentally differs from conventional bow geometries. Furthermore, in the proposed structural solution, when vertical flow develops around the bulbous bow, an additional hydrodynamic force is generated. Under heel conditions, this force tends to reduce the heel angle, it induces a localized hydrodynamic moment m_{xy} about the longitudinal axis.

This geometry-driven force redirection provides a localized hydrodynamic restoring tendency under the prescribed forced-immersion condition.

4. RESULTS AND DISCUSSION

Typical numerical results are presented in Figures 8–12. These relationships were obtained under conditions where the bow is immersed to a length of $L/4$, where L is the ship length. Figure 6 shows the deck impact pressure (green water load) as a function of the heading angle φ and the heel angle θ . The results indicate that, in

head seas ($\varphi=180^\circ$) the force value $\overline{P_z}=0.08$ is lower than the “small load” criterion commonly adopted in practical design. For other combinations of heading and heel angles, this force decreases by more than a factor of two. Notably, in the proposed design, the resultant hydrodynamic pressure force always acts along the ship’s centerline (diametral plane), and therefore does not generate additional heeling moments, in contrast to conventional bow configurations [5, 21, 22].

Figure 9 presents the transverse hydrodynamic force acting on the bow as a function of the heading angle φ and heel angle θ . The results show that the maximum value occurs at $\varphi=135^\circ$; however, it remains nearly two times smaller than that of a conventional bow. Specifically, in head seas $\varphi=180^\circ$, $\overline{P_y}=0.005$, whereas for a conventional bow $\overline{P_y}=0.027$. This reduction in transverse force minimizes the localized hydrodynamic yaw disturbance during deep immersion, thereby reducing the local hydrodynamic drivers that typically contribute to course deviation or broaching tendencies.

Figure 9 illustrates the hydrodynamic moments as functions of the heading angle φ , and heel angle θ . As previously noted, a key feature of the proposed design is that even under a very small heel angle, the hydrodynamic pressure acting on the bulbous bow generates a restoring effect, generating a positive hydrodynamic moment m_{xy} (localized passive restoring tendency). The results indicate that nearly the entire domain of the diagrams lies within the region of restoring moments, except for a limited range corresponding to oblique wave conditions. These negative heeling moments arise due to asymmetric pressure distribution on the hull, particularly in the emergence (water exit) region. However, as the heel angle θ increases, the restoring moment generated by hydrodynamic pressure on the bulbous bow also increases. Specifically, the greater the heel angle, the larger the restoring moment, which eventually compensates for and neutralizes the heeling moment.

The magnitude of the hydrodynamic pressure force acting on the bulbous bow can be increased by either increasing the bulb length l or the bulb diameter d . These relationships are illustrated in Figures 11 and 12.

Figure 11 shows the variation of the restoring moment generated by the bulbous bow as a function of bulb length. For example, at $\theta=60^\circ$ and $l=L/24$, the value is $\overline{m_{xy}}=0.2$; whereas for $l=3L/24$; $\overline{m_{xy}}\approx 0.85$; indicating that the restoring moment increases by more than four times. In practice, increasing the bulb length is often constrained by structural limitations, operational requirements, or arrangement constraints. In such cases, the restoring moment can be enhanced by adjusting the bulb diameter. The corresponding relationships between restoring moment and bulb diameter are presented in Figure 12.

The results indicate that, in this case, the restoring moment increases with bulb diameter, but not as significantly as when varying the bulb length. Specifically, at $\theta=60^\circ$, for $d=0.25D$ (where D is

the main hull diameter in the bow region), $\overline{m_{xy}}=0.4$. When the diameter is doubled to $d=0.5D$; the restoring moment increases to $\overline{m_{xy}}=0.52$; corresponding to an increase of only about 30%.

Due to the significant differences between the proposed bow geometry and conventional designs, the study also evaluated the impact of structural modifications on ship performance and resistance. The towing resistance was calculated using CFD simulations in ANSYS Fluent for a tanker ship (Table 2) comparing the conventional bow (curve 1) and the proposed bow (curve 2). The numerical framework in ANSYS Fluent has been updated with the following specifications:

(i) *Computational domain and boundary conditions*: The domain size was established based on the ship length (L). The inlet is located $1L$ upstream of the bow, the outlet is $2.5L$ downstream of the stern, and the side and bottom boundaries are placed $1.5L$ away from the hull to avoid blockage effects. Boundary conditions were set as Velocity Inlet, Pressure Outlet, Symmetry (for the centerline plane to save computational time), and No-slip Wall for the hull surface.

(ii) *Mesh strategy and independence study*: A hexahedral-dominant unstructured mesh with dynamic local refinement around the free surface and the bow geometry was utilized. A strict mesh independence study was performed using three mesh densities: Coarse (1.2×10^6 cells), Medium (2.8×10^6 cells), and Fine (5.5×10^6 cells). The Grid Convergence Index (GCI) was calculated, and the Medium mesh was selected as the optimal choice since the resistance discrepancy between the Medium and Fine meshes was under 1.2%.

(iii) *Turbulence and free surface modeling*: The Shear Stress Transport (SST) $k-\omega$ turbulence model was employed to accurately capture boundary layer separation around the cylindrical bow. The Volume of Fluid (VOF) method with open channel flow specifications was used for free-surface capturing.

The results indicate that, at speeds up to 13 knots, the difference in resistance between the two configurations is $\leq 2\%$, and therefore has a negligible impact on operational efficiency. As the speed increases, the resistance of the proposed design gradually becomes higher than that of the conventional design. At 22 knots, the resistance increases by approximately 21%.

Technical remark: The proposed design demonstrates favorable localized hydrodynamic restoring characteristics under deep immersion, but leads to an increased towing resistance penalty at higher transit speeds. This represents a typical trade-off in ship design between localized survivability/self-stabilization during extreme bow immersion and calm-water hydrodynamic efficiency at transit speeds. For practical application in shipbuilding, further investigations are required on the following aspects: ship motions (seakeeping performance), vibration characteristics, and global structural strength. These topics fall within the scope of dedicated, in-depth studies. The proposed design is particularly suitable for vessels operating in harsh sea conditions (e.g., offshore vessels and Arctic ships), while it is less suitable for high-speed vessels or commercial

ships optimized for fuel efficiency.

It is worth noting that the observed reduction in transverse forces and the generation of the restoring moment are localized hydrodynamic phenomena, evaluated under specific forced immersion depths and relative flow angles. While these static and quasi-static numerical configurations do not capture the fully coupled six-degree-of-freedom (6-DOF) rigid body dynamics of a ship in real irregular seaways, they effectively isolate the geometric advantages of the axially symmetric bow. The pressure distribution over the cylindrical surface ensures that the resultant dynamic force vector passes close to the longitudinal centerline, thereby minimizing the local overturning leverage that often exacerbates broaching-to or capsizing in conventional bow forms.

5. CONCLUSION

Based on the comprehensive analysis of the hydrodynamic characteristics of the proposed bow form in comparison with a conventional bow, the following conclusions can be drawn:

(1) *Vertical hydrodynamic forces* (bow diving condition): For a conventional bow, the maximum vertical force reaches approximately $(0.075 \div 0.082)\Delta$ under zero heel conditions. In contrast, for the proposed bow, this value does not exceed 0.08Δ over the entire range of heel angles, despite the fuller bow geometry. Notably, the resultant hydrodynamic pressure force always acts along the ship's centerline, thereby avoiding the generation of additional heeling moments—this is a key advantage over conventional designs.

(2) *Transverse hydrodynamic forces*: In the proposed design, transverse forces are limited to $\leq 0.016\Delta$ compared to approximately $\approx 0.029\Delta$ for the conventional bow. In head seas $\varphi=180^\circ$, the transverse force of the new design is about 5.5 times smaller, thereby mitigating the localized transverse destabilizing force that typically triggers course deviation under deep immersion. Moreover, no heeling moments are generated under head-sea conditions. In oblique seas, small heeling moments may occur. However, they are rapidly neutralized due to the increase in restoring moment generated by the bulbous bow.

(3) *Hydrodynamic restoring moment*: The restoring moment produced by the bulbous bow is proportional to its size and can be tuned by adjusting the bulb length or bulb diameter. Both parameters contribute to increasing the restoring moment, with bulb length having a more significant influence.

(4) *Towing resistance*: At speeds up to 13 knots, the difference in resistance between the proposed and conventional bows is $\leq 2\%$, indicating negligible impact on operational performance. However, at higher speeds (22 knots), the proposed design exhibits approximately 21% higher resistance compared to the conventional bow.

However, the authors acknowledge several limitations in the current stage of this research. The numerical experiments were conducted using an idealized, forced-immersion approach with fixed relative velocities, which simplifies the highly complex, coupled 6-DOF

seakeeping behavior of a vessel navigating through realistic, irregular sea states. Consequently, while the local geometric efficiency of the new bow form has been verified, the overall seakeeping performance and full-scale operational safety of the entire vessel cannot be definitively generalized from these local tendencies alone. Therefore, the present findings should be interpreted strictly as a localized geometry-driven restoring mechanism under prescribed forced-immersion conditions, rather than a definitive proof of overall vessel stability.

To bridge this gap, future research will focus on time-domain seakeeping simulations utilizing fully coupled 6-DOF solvers to investigate the dynamic response of the ship under irregular wave conditions. Furthermore, physical towing tank experiments with free-running models will be planned to validate the long-term stability, pitching motions, and actual broaching resistance of the proposed design.

REFERENCES

- [1] International Maritime Organization, Second Generation Intact Stability Criteria, MSC.1/Circ.1627, London, UK, 2020.
- [2] A. A. Belenky et al., "Stability and roll dynamics of surface ships in extreme waves," *Ocean Engineering*, vol. 220, pp. 108420, 2021.
- [3] C. J. Xiao et al., "Numerical investigation of bow slamming and deck wetness during bow-diving events," *Applied Ocean Research*, vol. 115, pp. 102831, 2021.
- [4] J. M. Falzarano et al., "Parametric roll and capsizing prevention in oblique seas," *Journal of Ship Research*, vol. 65, no. 3, pp. 145–158, 2021.
- [5] S. J. Lee and M. S. Kim, "SPH simulation of violent wave-structure interaction and bow impact pressures," *Ocean Engineering*, vol. 245, pp. 110412, 2022.
- [6] H. C. Raven et al., "Hydrodynamic optimization of hull forms for seakeeping and resistance," *Transactions of SNAME*, vol. 128, pp. 89–112, 2020.
- [7] T. I. Park et al., "Experimental and numerical study on green water loads on bow structures," *Marine Structures*, vol. 81, pp. 103105, 2022.
- [8] U. B. Ulvestad et al., "Inverted bow performance in rough sea states: Operational benefits and hydrodynamic loads," *Ships and Offshore Structures*, vol. 16, no. 4, pp. 388–402, 2021.
- [9] R. A. Slavounos, "Nonlinear wave forces and moments on surface ships in extreme waves," *Journal of Fluid Mechanics*, vol. 912, no. A15, 2021.
- [10] V. L. Khrushin, *Hydrodynamics of Axially Symmetric Hull Forms*, Marine Intellectual Technologies Publishing, St. Petersburg, 2021.
- [11] G. X. Wu, "Transient hydrodynamic loads during body entry into water," *Journal of Fluids and Structures*, vol. 102, pp. 103241, 2021.
- [12] V. N. Khrushin, "Ship, stable in stormy navigation," Russian Patent 2487043, IPC B63B 1/32, B63B 1/00, filed Jul. 12, 2011, published Jan. 20, 2013, Bull. No. 27.
- [13] V. N. Khrushin, "Bow end of a high-speed surface ship or a relatively slow-speed civilian vessel with increased storm seaworthiness and ice-crossing capability in autonomous navigation," Russian Patent 2607136, IPC B63B 1/06, B63B 1/32, B63B 1/00, filed Mar. 2, 2015, published Jan. 10, 2017, Bull. No. 1.
- [14] E. Y. Kamsvog, "Design of the forward part of a displacement-type vessel," Russian Patent 2374120, IPC

- B63B 1/06, filed Feb. 23, 2006, published Nov. 27, 2009, Bull. No. 33.
- [15] M. Tvete and H. Borgen, "Ship's forebody form," U.S. Patent Appl. 20120048164, IPC B63B 1/06, filed Jan. 27, 2010, published Mar. 1, 2012.
 - [16] Japan Marine United Corporation, "Hydrodynamic engineering," 2023. [Online]. Available: <https://www.jmuc.co.jp/en/rd/hydrodynamics/> (Accessed: Dec. 17, 2023).
 - [17] K. Matsumoto, "Ax-Bow: A new energy-saving bow shape at sea," NKK Technical Review, no. 86, pp. 46–47, 2002.
 - [18] G. Karafiath, "Stabilized tumblehome hull form," U.S. Patent 6601529, IPC B63B 3/00, filed Jun. 21, 2001, published Aug. 5, 2003.
 - [19] E. P. Burakovsky and P. E. Burakovsky, "On the scenario of ship losses during a storm due to wave capture of their bow," Marine Intellectual Technologies, vol. 2, no. 4, pp. 27–33, 201.
 - [20] E. P. Burakovsky and P. E. Burakovsky, "On the issue of determining the load acting on the vessel's deck at the bow end during its ditching in head seas," Marine Intellectual Technologies, vol. 3, no. 4, pp. 19–25, 2018.
 - [21] E. P. Burakovsky, P. E. Burakovsky, and V. M. Yusyp, "Study of hydrodynamic forces acting on the bow end during vessel ditching in waves while moving obliquely using the smoothed particle hydrodynamics (SPH) method," Marine Intellectual Technologies s, vol. 3, no. 4, pp. 27–32, 2022.
 - [22] E. P. Burakovsky, P. E. Burakovsky, and V. M. Yusyp, "Using SPH technology in design tasks to ensure navigation safety," in Proc. of the XXII International Conference on Computational Mechanics and Modern Applied Software Systems (VMSPPS'2021), Alushta, Sep. 4–13, 2021, pp. 24–26.
 - [23] J. J. Monaghan, "Smoothed particle hydrodynamics," Annual Reviews of Astronomy & Astrophysics, no. 30, pp. 543–574, 1992.
 - [24] A. J. C. Crespo et al., "DualSPHysics: An open-source parallel CFD solver on smoothed particle hydrodynamics (SPH)," Computer Physics Communications, vol. 187, pp. 204–216, 2015.
 - [25] Russian Maritime Register of Shipping, Rules for the Classification and Construction of Sea-Going Ships. Part IV. Stability. St. Petersburg: RMRS, 2018, pp. 82.
 - [26] ITTC Quality System Manual, Practical Guidelines for Ship CFD Applications. ITTC Recommended Procedures and Guidelines, Procedure 7.5–03–01–01, 28th International Towing Tank Conference, Wuxi, China, 2017.
 - [27] W. J. Kim et al., "Measurement of flows around modern commercial ship models for CFD validation. Journal of Marine Science and Technology, vol. 6, no. 1, pp. 14–37, 2001.
 - [28] Larsson, L., Stern, F., & Visonneau, M. (Eds.), Proceedings of the Gothenburg 2010 Workshop on Numerical Ship Hydrodynamics. Gothenburg, Sweden: Chalmers University of Technology, 2010.
 - [29] ITTC, Uncertainty Analysis in CFD Verification and Validation. ITTC Recommended Procedures and Guidelines, Procedure 7.5–03–01–02, 29th International Towing Tank Conference, 2021.
 - [30] M. Greenhow and W. Lin, Nonlinear free surface effects: Water entry and exit problems. Department of Ocean Engineering, Massachusetts Institute of Technology, Report No. 87–4, 1987.