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Design and Analysis of an AI-Class Solar Boat Race Hull for Performance, Stability & Competition Compliance A Simulation-Driven Approach with Experimental Validation

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Abstract

For many years, the UAntwerp Solar Boat Team has competed against universities worldwide in the Solar Class at the annual Monaco Energy Boat Challenge. Since 2025, this class has been replaced by the AI-Class, in which all vessels must operate autonomously across multiple disciplines. Insights gained during the team's first year in the AI-Class provided a solid foundation for a systematic design approach aimed at developing an optimized hull and improving current designs through extensive simulation, iterative testing, and validation making them capable to handle more power, have higher cruise speeds, and increase seaworthiness.

The analysis of existing designs provides insight in the potential and flaws, and suggest adaptations to improve performance stretching its lifespan. The new hull design learns from flaws in previous designs and accounts for both hydrodynamic and aerodynamic effects to maximize efficiency, maneuverability, and stability during competitive racing at sea. Emphasis was placed on achieving a modern high-performance, seaworthy configuration capable of safely reaching speeds of up to 30-50 knots on straight tracks. Lessons learned from previous designs and industry standards were applied to eliminate known shortcomings while maintaining a modular and easily manufacturable architecture with increased sponsor and media value. Furthermore, the new hull has been conceived with long-term adaptability in mind, ensuring future-proof usability for several competition cycles.

Abstract – NL

Al vele jaren neemt het UAntwerp Solar Boat Team het op tegen universiteiten van over de hele wereld in de Solar Class tijdens de jaarlijkse Monaco Energy Boat Challenge. Sinds 2025 is deze klasse vervangen door de AI-Class, waarin alle vaartuigen autonoom moeten opereren in meerdere disciplines. De inzichten die het team tijdens het eerste jaar in de AI-klasse heeft opgedaan, vormden een solide basis voor een systematische ontwerpbenadering gericht op het ontwikkelen van een geoptimaliseerde romp en het verbeteren van de huidige ontwerpen door middel van uitgebreide simulaties, iteratieve tests en validatie, waardoor ze meer vermogen aankunnen, hogere kruissnelheden halen en zeewaardiger zijn.

De analyse van bestaande ontwerpen biedt inzicht in het potentieel en de tekortkomingen en suggereert aanpassingen om de prestaties te verbeteren en de levensduur te verlengen. Het nieuwe rompontwerp leert van de tekortkomingen in eerdere ontwerpen en houdt rekening met zowel hydrodynamische als aerodynamische effecten om de efficiëntie, wendbaarheid en stabiliteit tijdens races op zee te maximaliseren. Bijzondere nadruk werd gelegd op het bereiken van een moderne, hoogwaardige, snelle en zeewaardige configuratie die veilig snelheden tot 30-50 knopen kan halen op rechte banen. Lessen uit eerdere ontwerpen en industriënormen werden zorgvuldig geanalyseerd om bekende tekortkomingen te elimineren, en een modulaire en gemakkelijk te vervaardigen architectuur te bereiken met een hogere sponsor- en mediawaarde. Bovendien is de romp ontworpen met het oog op aanpasbaarheid op lange termijn, waardoor hij toekomstbestendig is voor meerdere wedstrijdcycli.

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1. Introduction

1.1. Methodological Framework

The design and optimization of high-speed racing hulls remains a complex challenge at the intersection of hydrodynamics, structural engineering, and applied design practice. Despite extensive prior hydrodynamics and hull research and significant advances in both experimental and numerical methods across the industry, no single approach offers complete predictive accuracy across the full spectrum of operating conditions. Within this context, the present study adopts a structured and scientifically grounded methodology aimed at evaluating existing hull configurations of the UAntwerp Solar Boat Fleet, validating their performance through both experimental and computational techniques, and providing a technically justified design proposal for a new racing hull as well as adaptations to the current designs that addresses the evolving requirements of the AI-class race competition of the Monaco Energy Boat Challenge (MEBC). The basis of hull adaptations and the new design suggestion will be shaped by industry proven geometry, concepts and hull features to ensure predictability and effectiveness throughout every race run.

1.1.1. Rationale and Scope

The motivation for this study arises from the need to systematically compare three distinct hull concepts within the framework of performance racing: (1) an operational displacement hull (Torpedo Mk.II EVO), (2) a semi-displacement hull (Shark) that has been constructed (currently non-operational), and (3) a completely new design tailored to the specific demands of the AI-class races. Each of these designs represents a different stage of technological maturity and optimization philosophy. The comparison is intended not merely as a catalog of differences, but as a comprehensive evaluation that identifies the most influential parameters in terms of efficiency, stability, predictability, and overall performance.

The analysis recognizes that every hull type carries inherent advantages in specific operating regimes. The displacement hull provides a well-characterized baseline, offering insight into fundamental trade-offs in hydrodynamic resistance and efficiency. The semi-displacement hull represents a transitional form, intended to bridge displacement efficiency and speed but lacking operational validation. On the other hand, the new design is conceived to integrate advantageous features of both existing hulls, while incorporating proven concepts from the broader racing boat industry. By situating the study within this comparative framework, the methodology ensures that conclusions are not only relevant to the current competitive context, but also applicable to the future trajectory of hull development within the team.

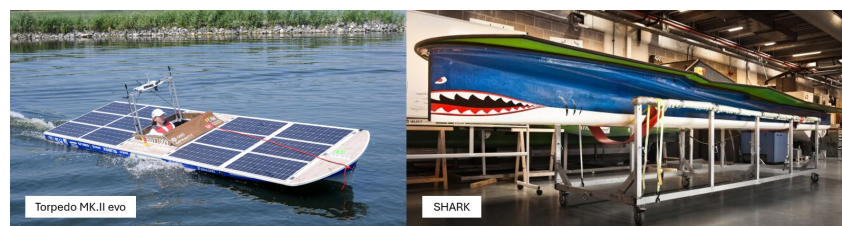


Figure 1.1.: Current hulls of the UAntwerp Solar Boat Team (UASBT)

1.1.2. Scientific Foundation and Validation

The methodological framework is grounded in fundamental engineering principles and supported by widely recognized scientific studies in naval architecture and hydrodynamics. All statements and conclusions are either validated experimentally, reviewed by industry experts, or substantiated by peer-reviewed literature. This approach ensures that the analysis remains both scientifically defensible and technically relevant.

Special care is taken to acknowledge the inherent limitations of predictive tools. While advanced design software has demonstrated a high degree of accuracy in forecasting hull performance, its application is constrained by the fact that current team hulls were not originally developed within the same computational environment and configuration. For all hulls, performance estimates will rely on a combination of experimental data, computational fluid dynamics (CFD) in different sea state conditions, and analytical calculations. This hybrid methodology, while demanding, ensures that comparisons are as accurate as possible. Note the required estimation method and its accuracy highly depends on the specific design stage. Bias in data collection and interpretation will be actively mitigated through rigorous validation protocols, including scale-model testing and, where feasible, full-scale experimental verification.

1.1.3. Focus of the Study

Unlike previous hull studies within the team, which emphasized ultimate efficiency as required in endurance racing, the present investigation places greater emphasis on dynamics, stability, and physical benchmarking of performance parameters. This shift in focus reflects the specific conditions of the targeted competition, which requires high-speed cornering capability and predictable behavior in seagoing environments, rather than maximum endurance. Efficiency remains a relevant consideration, but it is evaluated alongside — and sometimes subordinated to — stability, responsiveness, robustness, and above all: pilot safety.

1.1.4. Design Evaluation and Comparison

The comparative methodology employed in this thesis is designed to reveal the significance of individual design parameters such as hull geometry, weight reduction, drivetrain optimization, and power increase. By systematically isolating and evaluating these elements across the three hull configurations, including upgraded versions, the study aims to provide clear evidence of which factors most directly influence competitive performance. This analysis forms the basis for technical recommendations on geometry modifications, which will guide the development of the new design proposal alongside the study on design improvements. An important objective of the new hull design is to achieve a high level of predictability in performance, thereby increasing confidence in both engineering forecasts and race outcomes.

1.1.5. Integration of Computational and Experimental Methods

The design methodology adopted in this thesis combines carefully selected computational fluid dynamics (CFD) tools with predictive empirical software dedicated to a niche hull type. This approach enables rapid yet physically grounded exploration of design parameters while maintaining practical relevance. The empirical prediction framework, developed specifically for high-speed tunnel hulls and refined through decades of competitive racing experience, provides reliable performance and stability trends across wide ranges of mass, power, and trim conditions. When complemented by CFD analyses and scaled experimental validation for the existing fleet, this hybrid workflow allows both existing and newly developed hull concepts to be assessed with scientific rigor while remaining constrained by real-world feasibility. Close collaboration with the Canadian software developer and experienced tunnel hull racing champion Jim Russell ensured that the resulting designs and conclusions remain not only theoretically sound, but also manufacturable, operationally robust, and directly applicable to competitive racing conditions.

1.1.6. Strategic and Organizational Context

The study goals, methods and design approach is fully supported by the race team and its sponsors, aligning technical evaluation with organizational objectives and competition goals. While the scope of the thesis does not extend to financial analysis, the technical insights generated here will serve as the foundation for strategic decision-making regarding the future direction of hull development. In practice, these findings will fuel sponsor discussions, internal planning, and resource allocation. Importantly, the thesis serves as the technical foundation for subsequent development work, with a designated team member responsible for leading the incorporation of design changes or initiating new hull production.

1.1.7. Contribution and Distinction

The distinct contribution of this study lies in its balanced approach. By integrating engineering fundamentals, industry practices, advanced computational tools and experimental validation, the methodology provides a robust framework for design evaluation. Unlike previous studies within the team narrowly focused on efficiency or on isolated design parameters, this investigation emphasizes system interconnection and the complexity of the field of hull design, recognizing that performance emerges from the interplay of geometry, dynamics, stability, and power train integration. The evaluation is neither purely theoretical nor purely empirical, but rather a synthesis of both approaches, designed to produce actionable insights.



Figure 1.2.: Project information sources

1.1.8. Conclusion

In summary, the methodological framework of this thesis is deliberately constructed to balance scientific rigor with practical applicability. Limitations are acknowledged openly, but every effort is made to ensure that conclusions are supported by validated data, expert review, or established scientific consensus. The result is a study that is not only technically credible but also strategically relevant, providing the race team with a trustworthy basis for decision-making in the design and optimization of high-speed hulls for future competitions.



Designing efficient hull forms are both an art and a science. Nicholas Barczak

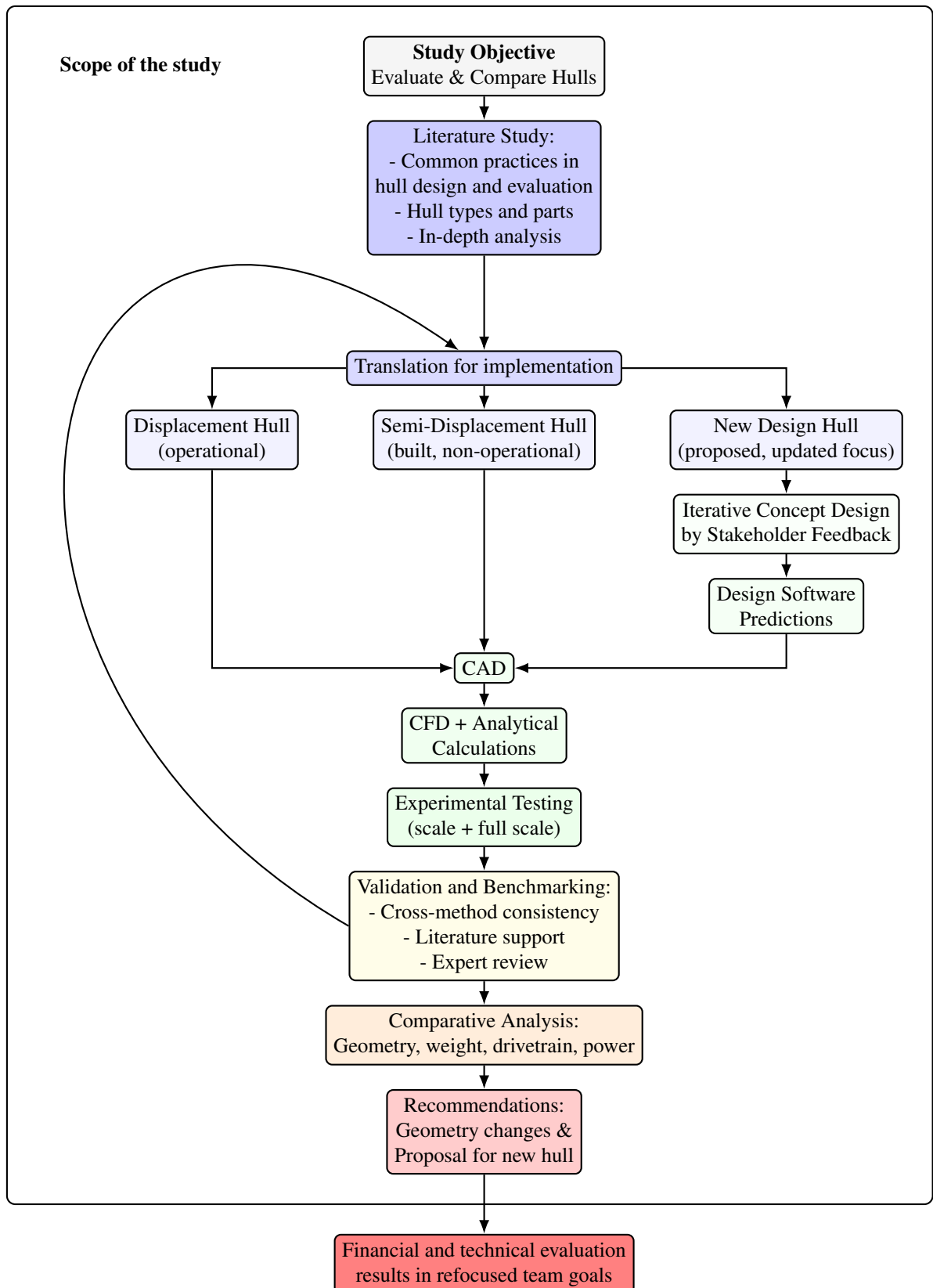


Figure 1.3.: Methodology flowchart: evaluation of displacement, semi-displacement, and new hull designs through experimental, computational, and software methods; validation and benchmarking lead to comparative analysis and recommendations.

1.2. Objectives

The true preliminary set objectives of the performance study is flexible as the analysis will point out the critical issues with the current hulls and indicate the potential of a new hull geometry. The final goals will be adapted to these findings and take shape along the investigation. In general, the improvements are about:

- Seakeeping: reduced greenwater, use of a deadrise and flare for softer entries, spray rails,...
- Stability: improved stability and acceleration curves, porpoising,...
- Performance (new hull): < 200 kg carbon sandwich hull, reduced wetted surface, optimized prismatic coefficient, high-speed stable planing (using proven technology and engineering).
- Integration of higher power with controlled trim.
- Structural robustness regarding to slamming for pilot safety and structural integrity. Study shape to reduce pressure. The structural design is out of the scope of this study.
- Validation via free surface CFD, scale testing and industry experts' review.
- Provide relevant documentation on the background and findings within naval architecture.

Main numerical acceptance criteria may include:

- Deck-wetness $< 5\%$ of time in target sea state.
- Slamming accelerations < 3 g peak, < 1.5 g 95th percentile.
- Porpoising damped within 3 cycles or similar criteria.
- Stability (CG, CL) within safe margins.
- Structural stresses (limited analysis) & reduced hull mass < 200 kg

1.3. Tasks

The tasks within this project were already indicated by Figure 1.3, the accompanying sections above and can be summarized more clearly by Figure 1.4. It is important to acknowledge the literature review's dual role: it will serve as both an introductory guide to hull design principles for this thesis and as a foundational reference — hull design 101 — for the team.

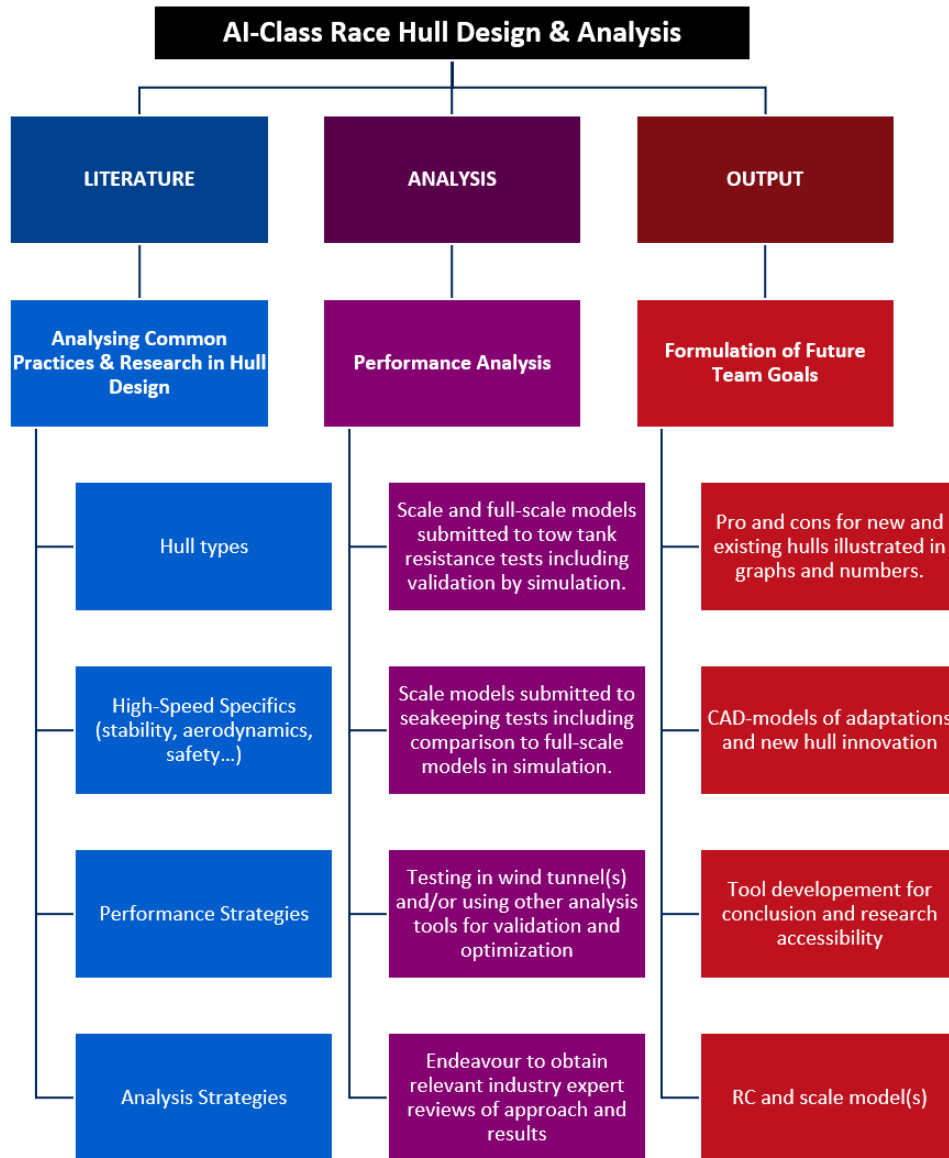


Figure 1.4.: Project tasks

2. Literature Review

The literature review focuses on establishing a comprehensive foundation for the study of hull design, resistance, and performance analysis in the context of the UAntwerp Solar Boat's transition toward a high-speed, fully autonomous racing platform. The review first explores fundamental principles of naval architecture, including an overview of hull typologies, geometric configurations, and bow, midship and stern concepts that form the basis of hydrodynamic performance. From there, attention is refined toward special considerations relevant to high-speed and autonomous craft, addressing their unique operational regimes, stability requirements, and design constraints. Further sections discuss established and emerging validation methodologies, encompassing experimental techniques such as towing-tank testing and wind-tunnel analysis, complemented by numerical and analytical approaches including Computational Fluid Dynamics (CFD) and theoretical formulations.

The overarching goal is to provide an extensive and structured reference for future hull-related research within the team, ensuring consistency and reproducibility in design evaluations. Note that Appendix C.1 contains extra background information and overviews covered by this literature study. The review also integrates the project's additional outcomes, such as the development of performance assessment tools, scaling procedures, and resistance translation software, consolidating them within the broader theoretical and experimental framework of naval hydrodynamics.

2.1. Hull Types and Geometries

This first section provides an introductory framework for readers who are less familiar with hull design or naval architecture. It outlines the fundamental principles required to understand hull analysis, beginning with the distinction between displacement and planing regimes, followed by an overview of common hull configurations, geometric characteristics, and general hull anatomy. The section concludes with a discussion of drag decomposition and a preliminary examination of design implications for autonomous marine applications, establishing the conceptual basis for the more detailed analyses presented in the subsequent chapters.

2.1.1. Overview of Hull Families

When reviewing hull families, a clear geometrical distinction can be made according to the cruising regime targeted during the design phase. This section presents an overview of typical hull types, outlining their common applications, classified and sorted based on their operational regimes. Subsequent sections examine the most relevant configurations in greater detail, focusing on designs applicable to small-scale, high-speed craft. While an exhaustive review of all existing hull types would fall beyond the scope of this

work, a concise overview provides essential context and highlights the diversity and complexity inherent to modern hull design.

Planing, Semi-Planing, and Displacement Modes

The distinction between displacement, semi-displacement, and planing regimes lies in the dominant physical mechanisms supporting the vessel's weight:

- **Displacement regime** — The vessel is supported almost entirely by hydrostatic buoyancy. Speeds are typically limited to $Fr \leq 0.4\text{--}0.5$ (see paragraph Hull Speed & Froude Introduction), beyond which wave-making resistance rises steeply.
- **Semi-displacement (or semi-planing) regime** — At intermediate speeds, hydrodynamic lift begins to contribute, but buoyancy still plays a major role. This regime is marked by the hump speed phenomenon, trim angle changes, and high resistance peaks.
- **Planing regime** — At sufficiently high speeds and favorable hull forms, hydrodynamic lift dominates. The vessel rides on top of the water, with reduced wetted surface area and wave-making resistance. Resistance in this regime scales more with dynamic pressure and wetted area than with wave-making.

This transition illustrates that hump speed is not a fixed physical barrier but rather a threshold between fundamentally different hydrodynamic support mechanisms. If not familiar with the resistance implications around hump speed or the classification of hulls based on the designed cruising speed, it is recommended to read Appendix C.1.1.

Hull Speed & Froude Introduction The concept of hull speed has historically been used to approximate the maximum efficient speed of displacement vessels. It is based on the physical phenomenon that, as a ship moves through water, it generates a system of transverse and divergent surface waves. The dominant transverse wave is characterized by a wavelength λ that grows with the square of the ship speed V , expressed as

$$\lambda = \frac{g}{2\pi} T^2 = \frac{2\pi v^2}{g} \implies v = \sqrt{\frac{g\lambda}{2\pi}} \quad (2.1)$$

where g is the gravitational acceleration and T the wave period [1]. When the wavelength of the primary transverse wave becomes approximately

equal to the vessel's waterline length L_{WL} , the vessel appears to be “trapped” between the bow and stern waves. This condition gives rise to the so-called hull speed:

$$V_H \approx 1.34\sqrt{L_{WL}}, \quad (2.2)$$

where V_H is in knots and L_{WL} is in feet (or equivalently $V = Fr\sqrt{gL}$, with $Fr = 0.4$, in SI units) [2]. Although useful as a heuristic, this expression does not represent a strict hydrodynamic limit, but rather the onset of a regime where resistance increases rapidly due to wave-making effects.

The hull speed condition can also be described in terms of the dimensionless Froude number which is defined in Equation 2.3.

Hydrodynamic Speed Regimes and ITTC

Classification The International Towing Tank Conference or ITTC procedures and guidelines form an important framework for experimental hydrodynamics, offering standardized methods for scaling, uncertainty analysis, and validation of test results. In the context of this research, the ITTC recommendations are adopted as a primary source for both theoretical considerations and practical testing procedures. However, due to financial and time constraints, it is not feasible to comply with every detail of the regularly updated ITTC documentation. Instead, a pragmatic approach is taken in which the most relevant and impactful recommendations are followed, while deviations are well-considered and found to be non-significant with respect to the current research objectives. This allows for the achievement of accurate and usable results without requiring full compliance to every procedural aspect.

High Speed Marine Vehicles According to ITTC The ITTC (2017) provides a formal classification of High Speed Marine Vehicles (HSMVs) [3]. These are defined as vessels that operate under one or more of the following conditions:

- Froude number $Fr > 0.45$,

- or speed $V > 3.7\nabla^{1/6}$ (m/s), where ∇ is the vessel displacement in m^3 (see Appendix C.1.47),
- or vessels exhibiting high trim angles,
- or vessels that are dynamically or aerostatically supported (e.g., planing craft, hydrofoils, air-cushion vehicles).

Representative examples of HSMVs include

Small Waterplane Area Twin Hulls (SWATH), Surface Effect Ships (SES), and Air Cushion Vehicles (ACVs). For most of these vessels, hydrodynamic and aerodynamic lift effects dominate over pure buoyancy, and conventional displacement-based scaling laws are no longer sufficient. A list of popular hull families has been added to the Appendix C.1.5. In subsequent chapters, the role of the Froude number in scaling and performance prediction will be discussed in more detail.

2.1.2. Froude Defined

$$Fr = \frac{V}{\sqrt{gL}} \quad (2.3)$$

- Fr = Froude number (Fr or F_n in literature) [-]
 V = characteristic velocity of the vessel [m/s]
 g = gravitational acceleration [9,81 m/s^2]
 L = characteristic length [typically waterline length, m]

The Froude number [4] is, assuming some simplifications, a rather basic equation in which the object velocity is divided by $\sqrt{gL}$, which represents the wave velocity [5]. As this is a reference work, a more understandable explanation with the parallels drawn to the mach number, is provided in Appendix C.1.2. In the *Appendix*, note the relevance of the Kelvin wake with a fixed half-angle of about 19.47° below hump speed and decreasing above the hump.

Froude and Scale Modeling

Together with viscosity μ and the Reynolds number Re , the Froude number is a highly relevant parameter in dimensional analysis and therefore defines some of the relations between scale models and full scale designs [5]. To illustrate the application of Froude similarity in practice, consider the scaling of a small high-speed craft. When investigating hull dynamics experimentally, it is common to construct a reduced physical model and perform towing-tank tests at a velocity chosen to preserve the equality of prototype (full-scale) and model Froude numbers. This ensures that the dominant gravity-related forces, particularly wavemaking and free surface interactions, are correctly represented. Such tests are of particular relevance when assessing instabilities including porpoising and pounding, as these phenomena are strongly dependent on the balance between hydrodynamic lift, hull geometry, and gravitational scaling. A model calculation is provided in Appendix C.1.3. In the example, an 8 m hull is scaled geometrically to a 1 m model, and the corresponding model speed and hydrodynamic responses are derived from the Froude criterion.

Implications of testing at matched Froude number are listed in Appendix C.1.4 and some will be highlighted in subsequent sections of the *Literature Review*. In summary, while Froude-scaled models reliably capture the relative onset and progression of free-surface dominated instabilities, viscous (and aerodynamic) effects require either empirical correction or computational fluid dynamics validation at full scale. An alternative and direct comparison is suggested in Section 2.6.2.

2.1.3. Mono-Hull Designs

The mono-hull is the fundamental configuration in naval architecture and forms the conceptual basis from which all multi-hull and hybrid geometries have evolved. Its extensive variety in form, function, and hydrodynamic behavior makes it the most widely studied and documented hull family in marine engineering. Due to its broad complexity, this section does not attempt to present an exhaustive analysis of all mono-hull design variants. Instead, specific aspects of the mono-hull geometry — such as bow, midship, and stern shaping — are discussed in detail in subsequent chapters (see Section 2.1.8), while flow phenomena and theoretical considerations are addressed separately in Section 2.1.7. Nevertheless, deadrise design principles relevant to planing mono-hulls are introduced in Appendix C.1.6, as they provide essential context for the discussions that follow.

2.1.4. Multi-hull Designs

Multi-hulls may offer significant advantages compared to traditional mono-hulls. However, as with any engineering solution, there are certain drawbacks to consider. Why catamaran and trimaran designs earn most interest in this study, will be explained below.

Current design of most Solar Class boat designs are actually slender displacement hulls and only act as a “stable” trimaran while cornering (worse efficiency ratings). When increasing the load on the Torpedo’s trimaran hull, the surface and shape drag — including wavemaking — significantly increases when sponsons are (deeper) in the water (see Section 2.2). Also, displacement hull resistance is less affected by increased mass than planing hulls (as demonstrated in Boat Curve Dashboard or BCD), making them more profitable for supporting solar panels, controllers, and more batteries in the regime below 32 km/h for 6 m hulls. Due to limited power and maximum focus on straight line performance, flat surface efficiency in the 20 to 40 km/h range is still competitive for slender displacement concepts compared to planing designs of the same mass. In that case, intuitively but well considered, one could state a trimaran with non-touching sponsons (although adding mass and minor air friction) is the perfect

match to fulfill all requirements set in the Solar Class. The AI-Class on the other hand, demands more powerful designs operating above 35 km/h with increased cornering efficiency, control predictability, stability, seakeeping, and — if possible — more flexibility regarding sensor mounts to expand the autonomous system’s capabilities. Catamarans may have slightly more viscous resistance in exchange for more predictable cornering due to limited roll. Excessive roll motion can potentially blind some the installed sensors’ perception or transmission range. Another pro catamaran argument is weight independent stability. Because of the “flying” sponsons of the Torpedo Mk.II EVO, a slight change in load might significantly impact the stability behavior of the boat compared to similar twin-hull setups. Given that modularity and autonomy are significant design constraints for a potential new hull, and ultimate efficiency is not a primary concern, a catamaran clearly has a distinct advantage in this section.

A detailed discussion on trimaran versus catamaran selection is provided in Appendix C.1.8 and C.1.9. For completeness, the stability discussion on the ingenious Small Waterplane Area Twin Hull (SWATH) design is covered in Appendix C.1.7, but was rejected for use of high-speed vessel development in this study.

2.1.5. New Hull Selection

Designing or selecting an appropriate hull form cannot be reduced to a simple algorithm or purely analytical process. Nor is it a matter of uninformed experimentation. Instead, it relies on the informed application of experience, empirical coefficients, and simplified physical models to evaluate the suitability of different configurations. No single hull type is universally optimal. Each involves trade-offs that depend on the context. An engineer therefore should approach the problem holistically — combining rational analysis with mission objectives, operational constraints, and a degree of creative judgment — to succeed with a final design that is both technically sound and purposefully tailored to its intended function.

As mentioned in previous studies within the team [6], the deadweight coefficient is one way of making a preliminary selection between different hull shapes. Figure 2.1 shows that different zones have a preference for different hull shapes. For example, the zones of the trimaran and catamaran overlap — These two are the most interesting shapes due to their stability characteristics. In the preceding Section 2.1.4, the catamaran is given precedence based on the collective knowledge concerning the designs of both the trimaran and the catamaran. This is continued throughout the deadweight analysis. It must be noted that an interpretation error occurred in the previous analysis of the team as will be explained in the following paragraph. The deadweight coefficient or deadweight-to-displacement ratio is defined as [7]:

$$C_{dw} = \frac{W_S - W_L}{W_S} \quad (2.4)$$

C_{dw} = deadweight coefficient
 W_S = ship mass (loaded) [kg]
 W_L = light ship mass (unloaded) [kg]

Based on mass measurements of the UASBT fleet in previous studies [8], the deadweight coefficient was calculated 0.283 for the Torpedo and 0.389 for the Shark [9]. The used W_L was based on the bare hull, including structural reinforcements. Unfortunately, this seems incorrect within the analysis of C_{dw} . The value of the deadweight coefficient lies in its estimation of the impact of a variable load and how much a load takes up from that accumulated maximum mass. As the Torpedo Mk.II EVO and Shark both serve racing purposes with near invariable loading conditions (trim, resistance, and stability remain consistent), the deadweight coefficient is less valuable and near zero (allow a

small C_{dw} for future upgrades, different pilots, and minor added ballast). In that case, Figure 2.1 contradicts the previously stated load invariability for catamaran stability (a catamaran does not suffer as much from the change in load as the trimaran does). Self-contradictory, the trimaran has a higher average C_{dw} in Figure 2.1. However, the C_{dw} ensures that the hull geometry achieves an appropriate balance between carrying efficiency, hydrodynamic performance and stability across its loading conditions, rather than indicating stability in isolation. Unfortunately, the use of the deadweight coefficient for hull selection is loosely defined in literature and is usually only a gross way to compare and choose between different hull types. In essence, there are alternate ways of interpreting its value, hence the confusion in preliminary studies [9].

A second, valid way of framing the deadweight coefficient — as it more often used for selection of medium and large scale (steel) vessels — is that it indicates an overall required weight/displacement due to cargo density. For example, low-density cargo such as ferry passengers implies low displacement and large deck area, highlighting the advantages of multi-hull configurations for vessels with low deadweight coefficients. In general, one could argue that light ships — those requiring limited displacement, such as LNG carriers — tend to exhibit low C_{dw} values. In contrast, a heavy, large displacement ship (e.g., bulk sand transporter) will show high C_{dw} , indicating a mono-hull as most optimal design.

As globally stated in this thesis, a trimaran is a great choice in the low and medium speed range ($Fr \lesssim 1.27$ or $v \approx 35$ km/h for a $L_{WL} = 6$ m), and mainly excels during straight line races. However, at higher speeds ($Fr \gtrsim 1.45$ or $v \approx 40$ km/h

for a $L_{WL} = 6$ m), stability and planing efficiency are most important. That's why most ultra high-speed racing craft adopt the stability of a catamaran, optimized specifically for planing and wave dampening (air cushion, deadrise).

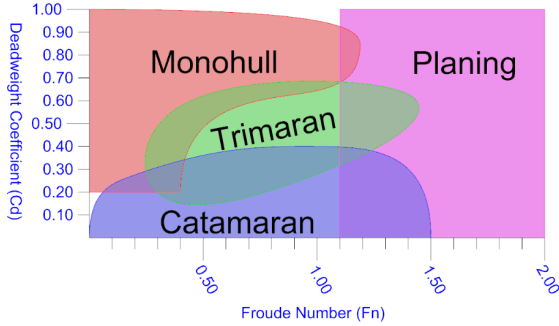


Figure 2.1.: Deadweight coefficient in hull selection (source: [7])

Higher power and speed combined with a higher average sea state and the potential to further increase power in the future strongly indicate that a new hull should be true planing ($Fn > 1.5$) and accounting for minor wave impact (near-port racing). It is clear from all listed racing craft that this is the way to go. Autonomy demands that a stable craft is desired for good perception sensor readings and seakeeping as well as predictability for path-planning and navigational purposes.

For future reference, summarized overview of example (semi-) planing performance characteristics is presented as a reference in Table 2.1, appended below. Common racing hull types considered for selection are listed in Appendix C.1.10. Hull anatomy is illustrated in Appendix C.1.11.

Name	Mass [kg]	Power [kW]	Top speed [km/h]	Remarks
Torpedo Mk.II EVO	300	6	22.2	Semi-displacement
Racebird	850	150	92.6	Passive foil configuration
UGent Sailing	320	10	37-46	Planing, concept phase
Sunflare	164	7	40	Slender semi-displacement
Molabo F4	-	50	91.5	F4-E, 48V

Table 2.1.: Examples of basic hull characteristics

2.1.6. Forces Acting on a High-Speed Hull

The dynamics of a high-speed hull are determined by the balance of forces and moments acting on it. These forces include hydrodynamic and aerodynamic drag, lift contributions from various mechanisms, and restoring forces linked to stability. A full description should consider the vessel's six degrees of freedom.

Degrees of Freedom

A rigid hull in free motion has six degrees of freedom (DOF): three translational and three rotational. The translational motions are surge (longitudinal), sway (lateral), and heave (vertical), while the rotational motions are roll (about the longitudinal axis), pitch (about the transverse axis), and yaw (about the vertical axis). In high-speed craft, surge is dominated by the balance between thrust and drag (see Section 2.1.9); heave and pitch are strongly coupled to lift generation and dynamic trim; roll and yaw affect

lateral stability, particularly interesting in multi-hull and tunnel configurations. The interaction of these DOF with external forces defines both the efficiency and controllability of the vessel.

Main Lift Contributions

Lift can be generated through multiple mechanisms in high-speed hulls:

- *Body lift*: the geometric wing-like shape generates lift proportional to the wing sur-

face area, velocity squared, and angle of attack.

$$L_{body} = \frac{1}{2} \rho_{air} V^2 S C_L \quad (2.5)$$

with C_L dependent on geometry and angle of attack.

- *Tunnel lift*: catamaran tunnel geometries entrap air, creating an aerodynamic cushion that reduces wetted area and augments lift.
- *Planing lift*: planing surfaces generate dynamic lift proportional to wetted area, velocity squared, and angle of attack.
- *Hydrofoil*: submerged foils provide lift according to

$$L_{foil} = \frac{1}{2} \rho_{water} V^2 S C_L \quad (2.6)$$

with C_L dependent on geometry and angle of attack.

- *Hydrodynamic shaping*: reverse chine, cambered and stepped hulls can modify pressure distributions to enhance lift or control trim.

Ground Effect

When foils or tunnel hulls operate close to the free surface, the water surface acts as a mirror plane, effectively increasing wing aspect ratio and reducing downwash. This “ground effect” reduces induced drag and improves lift-to-drag ratio, an effect exploited by tunnel hulls and other low-clearance vessels.

Stability

To fully understand why stability and seakeeping is related yet not the same, a clear fundamental question is raised.



How to guarantee ship safety on an uncertain ocean?

Analyzing stability is a great and often used tool to provide information on the behavior of a hull on rough seas. However, there is an infinite amount of sea conditions possible and results are often also time dependent (e.g., near-resonance oscillations). Directional changes, wave height, ship inertia, thrust influence, wind,... are all very important parameters to learn about seakeeping. However, no model or simulation is capable of providing results to fit all situations. To address that shortcoming, naval architects often decide to limit the magnitude (seakeeping analysis, specified wave system) and dampening time (stability analysis, safety) of motion effects. Using these tools is great for rational decision making. However, as no solution fits all situations, there is never a guarantee to ship safety [10].

Definition — Seakeeping is the ability of a vessel to remain at sea and safely carry out its mission in various weather conditions, especially important in rough seas.

Stability in high-speed craft emerges from the interaction of forces across multiple degrees of freedom (DOF). The six DOF consist of the translational motions surge, sway, and heave, and the rotational motions roll, pitch, and yaw. Surge is governed primarily by thrust and drag balance, while the other DOF are strongly coupled to lift distribution and restoring forces. Static stability refers to the ability of the vessel to return to equilibrium after a perturbation, whereas dynamic stability concerns the damping of oscillatory motions in response to external disturbances.

For conventional planing hulls without foils, stability is dominated by hydrodynamic forces acting on the wetted surface. Roll stability is gov-

erned by hull beam and chine geometry, pitch stability by dynamic trim and center of pressure location, and yaw stability by lateral area dis-

tribution. Instabilities typical in the regime of a racing hulls include porpoising (pitch oscillations due to alternating lift and loss of contact), chine walking (lateral roll oscillations in catamarans), and directional instability in yaw. In these configurations, stability must be ensured by geometric design, weight distribution, and damping through wetted area and spray. At low speed, one could state the stability of a good hull is governed by the natural buoyancy pressure on the hull.

If foils are used, additional DOF interactions appear. In foil-assisted configurations, foils reduce the load on the hull and wetted area but introduce sensitivity in heave and pitch, as lift varies strongly with immersion depth and angle of attack. However, the vessel still benefits from the natural buoyancy stability. Passive foil systems, where the foil geometry and incidence are fixed, provide efficient lift but require precise hull-foil integration to avoid ventilation and ensure stability in roll and pitch. To eliminate the drag inducing water piercing foil configuration passive foils rely on, active foils might be a solution. These systems add controllability by adjusting foil incidence in real time, which allows feedback stabilization of heave, pitch, and roll while foils are fully submerged. As highlighted in hydrofoil control practice, maintaining stable flight depends on continuous regulation of angle of attack and immersion; disturbances such as wave impact or asymmetric ventilation can otherwise rapidly amplify and lead to loss of control. An active foil design has therefore six degrees-of-freedom.

With the aim of creating a useful reference work for the team that also covers possible roll stabilization techniques, a discussion has been added to Appendix C.1.13 that covers the shape of the hull, stabilization fins, tank and gyroscopic stabilizers, as well as reaction wheels. This is combined with an overview and comparison from which the following conclusion can be drawn: the multi-hull and (active) stabilizer fins suggestions have most potential for easy and economically feasible integration in the (existing) hull(s). A gyroscopic system is only a third-choice option due to the added energy, maintenance, complexity and cost.

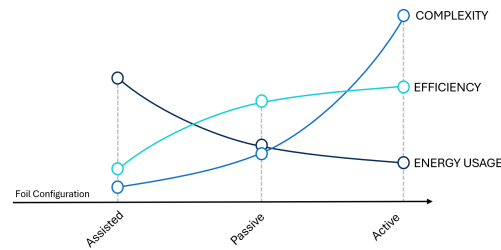


Figure 2.2.: Influence of foil configurations

The coupling of DOF differs across these options. In hull-only operation, heave, pitch, and roll are stabilized by the hydrodynamic pressure distribution over the underside. In foil-assisted craft, heave and pitch are highly sensitive, with roll stability depending on symmetric foil lift. In fully passive foil craft, stability margins are reduced and rely on design alone, whereas active foils allow real-time control over three DOF (heave, pitch, roll), often at the expense of added complexity and energy consumption. Across all cases, yaw stability remains critical for steering control, requiring appendages or differential lift to counteract perturbations. A comparison of stability characteristics across hull-only, foil-assisted, passive, and active foil configurations in relation to degrees of freedom is provided in Appendix C.1.12.

Considering the main stability characteristics for tunnel hull boats, stability comes down on two design criteria: (1) the accumulation of all forces must net out zero and (2) the center of gravity (CG) must be set as close to the aerodynamic center (AC) as possible [11]. A fully airborne vehicle will therefore not generate a pitching moment destabilizing the craft. It is important to note that, according to the *Secrets of Tunnel Boat Design* book [11], the achievement of the second criterion is practically unattainable. Typically, the static weight vector is aft of the aerodynamic lift, which traverses the operating range.

Summary on Force Balance

In summary, the forces acting on a high-speed hull consist of complex hydrodynamic and aerodynamic drag contributions, multiple sources of lift, and ground-effect phenomena. These forces act through six degrees of freedom, coupling resistance and stability. While drag reduction and lift maximization drive efficiency and speed, stability considerations place constraints on design, ensuring safe and predictable performance in the low-sea-state conditions typical for the expected racing environment.

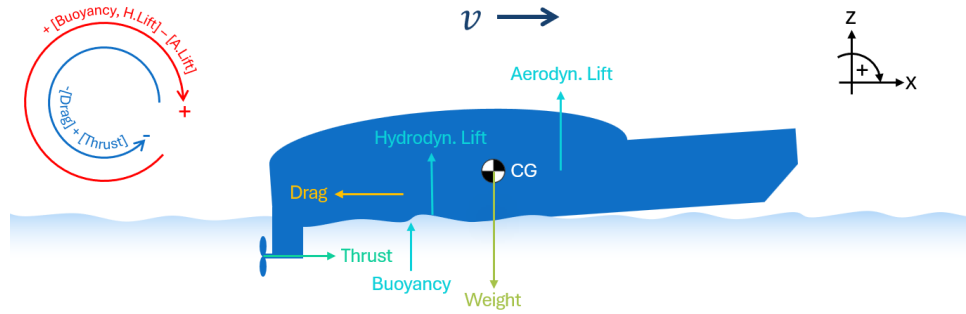


Figure 2.3.: Main forces acting on a hull

2.1.7. Basic Flow Theory

The development of new hull forms always builds upon existing geometries and the fundamental theory of fluid dynamics, as emphasized by naval architects [12]. At the core lies Bernoulli's principle, which in its simplified steady-flow form expresses the conservation of energy along a streamline [13]:

$$p + \frac{1}{2} \rho V^2 + \rho g z = \text{constant} \quad (2.7)$$

- p = static pressure [Pa]
- ρ = mass density of fluid [kg/m^3]
- V = flow velocity [m/s]
- g = gravitational acceleration [m/s^2]
- z = height relative to datum [m]

Equation 2.7 identifies three contributions to the total pressure: static (compression), dynamic (velocity-based), and hydrostatic. The sum of these terms remains constant along a streamline, such that changes in velocity or elevation are balanced by changes in pressure. This relationship is directly observed on a hull: increased flow velocity around slender sections corresponds to reduced static pressure, while stagnation regions at the bow or stern lead to elevated pressures acting on the structure. In most naval applications, pressure forces are interpreted as resistance or upward hydrostatic support. However, under specific conditions near the stern, these pressure distributions can positively contribute to thrust recovery, as the local pressure force acts normal to the hull surface and partially in the longitudinal direction.

How the pressure is decomposed through a moving point along the hull is extensively discussed in Appendix C.1.14, based on the explanation given by Datawave Marine Solutions [12] and Lars Larsson and Hoyte C. Raven [14]. The flow theory background will synergetically form a fundamental base for the displacement focused analysis of existing and design of the new hull.

Influential Naval Contributors

For reference, it is worth mentioning some of the important names related to this science domain. Some of the most influential contributors to the domain of hull design and flow theory are listed, since their work underpins the models and empirical formulations still in use today. Savitsky, Clement, and Navier-Stokes significantly contributed to the field of naval engineering advancing analytical predictive methods for planing hulls, geometry modifications, and mathematical (CFD) analysis respectively. A more detailed explanation is given in Appendix C.1.15. Froude's relation to hull design was introduced more in detail in Section 2.1.2. Crouch's equation is evaluated in the next section and Bernoulli was already evaluated in Section 2.1.7.

Crouch

In contrast to displacement-based scaling laws such as the Froude number, planing hulls exhibit a performance regime that depends more strongly on weight and available propulsive power than on length at the waterline (LWL). A widely used empirical relation in high-speed craft design is Crouch's formula, originally proposed by Crouch [15] and subsequently disseminated in naval architecture practice. It provides an estimate of the maximum attainable speed for a planing vessel:

$$V = C \sqrt{\frac{HP}{DISP}} \quad (2.8)$$

- V = vessel speed [knots]
- HP = shaft horsepower at the propeller [hp]
- $DISP$ = vessel displacement [lb]
- C = empirical hull constant

The formula highlights the weight-dependence of planing hulls: speed increases with installed power but decreases with displacement. Unlike the Froude number approach, the formula does not explicitly depend on waterline length, reflecting the dominance of hydrodynamic lift over wave-making for high-speed craft. Instead, all geometric and hydrodynamic influences are condensed into the coefficient C , which is determined empirically for each hull type.

To facilitate comparison in computational tools such as the BCD software developed in this study, the formula can be expressed in SI units:

$$V = C_{SI} \sqrt{\frac{P}{m}} \quad (2.9)$$

- V = vessel speed [m/s]
- P = delivered power at propeller [W]
- m = vessel mass [kg]
- C_{SI} = empirical hull constant in SI formulation [–]

Here C_{SI} differs numerically from C in the imperial formulation but carries the same physical meaning. Values for C or C_{SI} must be derived from sea trials, model test regression, or accumulated empirical data for similar hull types. In practice, designers use tabulated constants for preliminary prediction, while refined estimates may be obtained through performance testing.

Table 2.2 summarizes representative C -values for common planing hull categories. These values are drawn from experimental practice and design guides, but it must be stressed that they are approximations and depend on surface finish, deadrise, trim angle, and propeller efficiency.

Hull type	C (Imperial)	C _{SI}
Heavy runabout / utility boat	150	1.9
Medium-V recreational planing hull	180	2.28
High-performance V-hull (deep-V)	200	2.53
Light racing boat / hydroplane	210–240	2.66–3.04
Record-class / extreme lightweight	≥250	≥3.17

Table 2.2.: Representative Crouch constants C for different hull types.

Remarks on Application The appeal of Crouch’s formula lies in its simplicity, which makes it suitable for preliminary design studies and comparative evaluation of different hulls. In this work, it is implemented in the BCD software alongside wave resistance-based methods (Froude) to generate speed–power–resistance graphs. This allows direct comparison of displacement hull behavior (where length-based Froude scaling dominates) and planing hull behavior (where mass-to-power ratio is critical). However, the limitations of the approach must be recognized: the method does not account for trim angle dynamics, porpoising, appendage drag, or propulsor efficiency variations (compensated in BCD). As such, Crouch’s relation is best regarded as a semi-empirical performance indicator, to be validated against model testing or CFD simulations.

2.1.8. Hull Geometry

In this section, the general ideas are listed concerning the three main sections of a ship: the bow, midship, and stern. Every section has its own function and should be matched with each other to come up with a design that fits the over-all purpose of the hull. This is the first step in hull design, apart from selecting a mono-hull versus multi-hull design. The three main sections will briefly summarize the options. Nevertheless, know that a multitude of types, variants and more specified geometry parameters are carefully studied to make well-considered decisions in this thesis.

Bow

There is much to talk about in the evaluation of different bow geometries. In essence, every bow should try to minimize wave impact, limit greenwater (water and sea spray on the decks) and add as least resistance as possible. Often, goals are quantified by limiting pitch accelerations for certain wave conditions [16]. Reducing pressure drag mainly implies smooth water deflection with minimal wavemaking as highlighted in Section 2.1.7. Though, this is true for low and medium sea states, yet the bow has most influence on safety and wave impact related seakeeping, where ultimate efficiency is less prioritized.

As in all elements of ship design, a balance between characteristics must be found. A sharp bow limits the deflection rate to minimize drag, but offers little response to incoming waves. So, generally these types of bows have bad seakeeping as illustrated by the sharp, low bow of the Torpedo MK.II in Figure 2.4. Designing for high flat-water efficiency often leads to significantly increased drag and diving behavior in mild sea conditions, while bows designed for mild sea conditions usually offer acceptable ef-

iciency ratings for both mild and flat sea conditions. For Figure 2.4 and the future design of bows, it is important to distinguish between sea spray and greenwater as mentioned by Datawave Marine Solutions [16]. In the UASB’s situation, spray may damage solar panels and wires or exposed sensors and cockpit equipment because of corrosion or direct contact (short circuiting) with conductive seawater. In contrast, greenwater is considered much worse as it can cause severe structural damage to decks, hull, antenna

poles, etc. Spray is formed by a wave hitting the bow, greenwater is formed by a wave splashing over or on top of the front deck. Such a wall of water may ever threaten stability in extreme conditions. In an open cockpit, the danger of flooding is also increased and on top of that, wave encounter and greenwater is often the largest factor

in drag increase. A wave may cause a vessel to instantaneously lose a large amount of forward momentum. In large seagoing ships a complete stop after a wave impact is not an exception. In summary, greenwater must be avoided at all times and spray should be minimized to achieve a suitable seagoing racing vessel.



Figure 2.4.: Torpedo MK.II's susceptibility to greenwater and deck-slamming

Subsequently, common bow types are evaluated in Appendix C.1.16, forming the basis for adaptations and new hull design considerations. Note the Shark also has a bit of an axe-bow design, as defined in the appendix. It has a sharp slender bow and the keel drops down (minimal) towards the bow. However, in the Shark, the axe-technique is poorly adopted. Featuring insufficient keel dropping geometry and minor aft placed CG causes unconventional maneuvering issues and has near no advantage in seakeeping compared to the Torpedo Mk.II EVO. Also, the hull suffers from the same issue commercial axe designs do, namely the lack of air under the hull, reducing wetted area and drag. This effect is of the utmost importance in the context of high-speed performance planing. The majority of commercial X-bow hulls do not attain speeds sufficient to necessitate this consideration; however, the Shark, given its racing-oriented design, does reach speeds that make this a relevant factor. The mitigation techniques will be addressed subsequently in this study.

Midship

The midship has the least influence on drag (insignificant hydrodynamic pressure), except for viscous resistance including friction. So the focus of a naval engineer designing the midship section mainly considers two things: (1) the required displacement and cargo / load and (2) the vessel's stability. Note that the (semi-planing) behavior of in case of planing craft is also defined in the midship. For example, steps for air introduction under the hull and reduced wetted area are placed here.

Note for wide and medium beam vessels, a rectangular cross-section can be considered drag increasing compared to a round hull. This is true when compared with a round hull (same displacement, less wetted area), having less shape stability as a drawback as indicated in Section 2.1.6. However, for slender designs ($LD > 8$), the influence of a hull's cross-section shape be-

comes negligible when evaluating drag as indicated in experiments by Molland et al. [17]. The length-displacement or LD-ratio is most significant when comparing displacement efficiency. Other hydrodynamic drag reduction strategies are described in Section 2.3. Hull-specific issues discovered when analyzing the existing hulls can be found in Section 2.2.

Stern

The stern of a ship plays a critical role in the overall hydrodynamic performance, particularly in terms of wake resistance. It constitutes the aftmost part of the hull and governs the flow conditions near the propeller and the pressure recovery region (see Section 2.1.7). The design of the stern must ensure a minimal pressure gradient to prevent flow separation, which would otherwise increase viscous resistance and reduce propulsive efficiency.

In conventional displacement ships, the stern is typically designed to promote laminar-to-turbulent transition at controlled locations, thereby stabilizing the wake field entering the propeller. A well-designed stern minimizes asymmetric flow patterns that could induce unbalanced thrust loads or vibrations. For twin-screw configurations, the geometry must also account for wake alignment and the interference effects between the propellers and rudders.

From a hydrodynamic perspective, the pressure recovery at the stern is one of the most decisive

parameters affecting total ship resistance. Optimized transom sterns, for instance, allow for partial pressure recovery and minimize the size of the low-pressure recirculation region (see Figure C.13). In contrast, fine-form or cruiser sterns are preferable for vessels prioritizing reduced wave-making resistance and smoother wake profiles at sub-hump speeds. The effect of energy regeneration in displacement ships is discussed in Section 2.1.7, where fundamental flow interactions between the hull and a laminar environment are examined in more detail.

Finally, stern shape optimization has become a more popular research area supported by CFD and model testing. Parametric design tools and adjoint solvers enable systematic evaluation of design trade-offs, such as resistance reduction versus flow uniformity. These techniques have led to the development of stern forms that balance hydrodynamic performance with structural and manufacturing constraints, making the stern a focal point in the evolution of high-efficiency ship design. The use of transom sterns, popular in planing vessels, is discussed in Appendix C.1.17. In essence, a vertical stern provides clean flow detachment promoting predictable, efficient flow in high-speed craft.

2.1.9. Drag Decomposition

The total resistance acting on a ship or small craft can be subdivided into multiple physical components, each associated with a distinct hydrodynamic mechanism. Several decomposition frameworks exist, depending on whether the analysis is derived from theoretical fluid dynamics, empirical testing, or standard conventions in naval architecture. The goal of this section is to clarify how these perspectives relate and to establish the terminology adopted in this thesis.

In Appendix C.1.18, a theoretical approach is presented to separate idealized flow phenomena — highlighting which forces persist or vanish under specific physical assumptions. This provides a conceptual foundation for understanding how drag arises in viscous and free-surface flows. In subsequent sections, the decomposition is redefined following the conventions used in naval architecture, particularly those standardized by the International Towing Tank Conference (ITTC). This practical framework introduces the coefficients of resistance commonly

employed in model testing, numerical simulation, and full-scale extrapolation, bridging the gap between theoretical hydrodynamics and experimental practice.

Preceding introduction and theoretical intermezzo of Appendix C.1.18 indicated that the resistance a hull shape provides at a fixed speed can be decomposed into many contributing parts adding to the total drag. Understanding the influence and contribution magnitude provides important insights in the potential of individual efficiency optimization techniques.

$$R_T = R_V + R_W + R_A \quad (2.10)$$

- R_T = total hull resistance [N]
 R_V = viscous & friction resistance [N]
 R_W = wave making resistance [N]
 R_A = combined aerodynamic / air resistance [N]

The near linear viscous resistance once again indicates that this is the most important factor for planing hulls as this is verified by tow tank responses. The exponential growth shows the same for wave making and displacement hulls accordingly. Note that multiple humps can show in resistance-speed measurements due to change of trim, spray, etc.

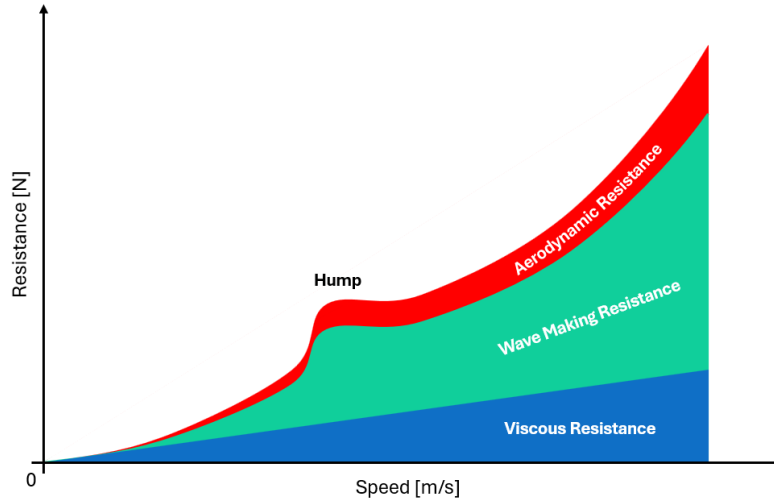


Figure 2.5.: Resistance decomposition for (semi-)displacement Vessels

Dimensionless coefficients are used by all engineering disciplines, including naval architecture, to characterize a system's performance or to compare systems. An automobile's aerodynamic efficiency is quantified by automotive engineers using a "drag coefficient" similar to aeronautical engineering. A ship's hull's performance and design are described by naval architects using a variety of dimensionless coefficients. The naval architect can evaluate the performance of various ship types or compare model test results to full-scale ship data thanks to these dimensionless coefficients. Standard dimensionless coefficients described by the ITTC are widely used in the field of ship propulsion and resistance. Dimensional analysis is discussed in other sections, based on the procedures and guidelines of the ITTC (Recommended Procedure 7.5-02-03-01.4 1978 ITTC Performance Prediction Method and Recommended Procedure 7.7-02-05-01 2017 High Speed Marine Vehicles Resistance Test). Usually a model coefficient is derived from a tow test and the full scale coefficient is calculated followed by the R_T . This results in a relation between speed and required shaft power. The Boat Curve Dashboard (BCD) software developed in this project helps illustrating and clarifying that relation. The R_T can be described in terms of dimensionless coefficients by:

$$C_T = C_V + C_W \quad (2.11)$$

- C_T = coefficient of total hull resistance
 C_V = coefficient of viscous resistance (or C_f)
 C_W = coefficient of wave making resistance

Remark that these coefficients are derived from scale models, therefore aerodynamic resistance is not represented. Most tow tank models doesn't have superstructures making detailed aerodynamic resistance analysis irrelevant.

The relation between the R_T and C_T is illustrated in equation 2.12. Note that often imperial units are used in literature, so particular care is required when comparing these coefficients to SI-based values of the ITTC.

$$C_T = \frac{R_T}{\frac{1}{2}\rho_w S_0 V^2} \quad (2.12)$$

C_T = coefficient of total hull resistance
 R_T = total hull resistance [N]
 ρ_w = mass density of water [kg/m³]
 S_0 = nominal wetted surface area [m²]
 V = speed [m/s]

The specific construction of hydro- and aerodynamic drag is discussed and partially quantified in Appendix C.1.19 and C.1.20 respectively. The analysis is based on the HSMVs procedures of the ITTC (2017) [3] and the Secrets of Tunnel Hull Design book [11].

2.1.10. Pro/Cons for High-Speed Autonomous Applications

High-speed autonomous boat racing imposes stringent requirements on perception, communications, and hull-aero-hydro coupling. Sensor performance is tightly coupled to sea state, structural vibration, and the hydrodynamic/aerodynamic flow field generated by the hull. The analysis in Appendix C.1.21 distinguishes flat and medium sea states and focuses on LiDAR, stereo-camera sensing, and GNSS/-communications antennas, together with implications for hull geometry and high-speed cornering being a reflection of the requirements for AI-class racing in the MEBC competition.

In essence, an air cushion catamaran with piercing/wave riding bows and a well-managed tunnel offers the best balance between dynamic stability, reduced spray, and low vibration transmission. This hull supports high-speed planing efficiency while enabling stable, predictable cornering when combined with trim control and modest aerodynamic stabilization. A flat-bottom, maximum efficiency planing hull is not feasible due

to high magnitude accelerations during wave encounter, low lateral grip, and high cornering inclination. Across sea states, platform motion and spray-induced occlusions are the dominant constraints on LiDAR, cameras, and GNSS reliability. Hull geometry that damps accelerations and minimizes spray directly increases autonomy robustness.

Based on the discussion, changes to the existing designs evaluating the requirements of autonomous craft are proposed in Appendix C.1.22. Through analysis, the list will be narrowed down until only the upgrades with the most potential remain.

2.2. Continued Discussion on Geometry

Based on previous and actual research, personal experience in hull design and offshore engineering, and extensive feedback for hull optimization and design professionals like marine consultants and researchers from various institutions, there are additional remarks on the potential and drawbacks of the current designs available within the UASBT highly specific to their unique design. The discussion will mainly focus on seakeeping, basic structural remarks and drag. In this section the prospective will be unveiled. The most promising upgrades are presented in extend to Appendix C.1.22. As previously

mentioned, this forms the basis for the selection of designs that will pass onto the next stage of CFD analysis and practical experiments.

2.2.1. Remarks on the Torpedo Mk.II EVO

The Torpedo Mk.II EVO has served as the baseline hull of the UAntwerp Solar Boat Team for several competition cycles and represents a semi-displacement concept with slender proportions and pronounced lateral overhangs featuring trimaran amas/sponsons. Its long service life and proven race record make it a valuable reference platform, yet it also embodies limitations that become increasingly relevant as the team aims for higher operating speeds and harsher seakeeping conditions. In particular, the geometry of the center hull and sponsons results in a draft/roll-dependent waterline beam, leading to sensitivity in wave-making resistance and appendage drag. Furthermore, structural aging due to material degradation in composite laminates introduces uncertainty in safety margins when exposed to elevated slamming loads that scale with V^2 . These characteristics necessitate a systematic form analysis to identify both the potential of the current hull and its constraints. The following sections provide an additional technical assessment of the Torpedo's geometry, hydrodynamic coefficients, and operational behavior in calm and moderate sea states, with emphasis on resistance, slamming, and maneuvering performance.



Figure 2.6.: Default hull of the torpedo (without decks)

From the section cuts in Figure C.15 can be learned that the draught of the Torpedo heavily influences the width at waterline (WWL). A high WWL encourages wave making and therefore displacement / wave making resistance. For maximizing efficiency on the torpedo it is mandatory to lift the hull near 0.1m draft, enabling the use of the lowest available width to be used while avoiding appendage drag of sponsons. A more vertical freeboard in the displacement center hull would significantly improve efficiency consistency over different mass conditions and sea states. Highlighted by the BCD software, a vertical freeboard would improve the resistance at 20 km/h of approximately 355 N to 300 N, improving fuel economy with 15.5%. Applying the same thrust of 355 N, the speed increases from 20 km/h to 22.5 km/h. Another insight provided by the BCD is the potential without geometric changes. When using a way to lift the body higher out of the water (e.g., assisted foils) with a force around 130 kg reducing draft to 0.1m, the 355 N at 20 km/h reduces to 270 N, improving fuel economy with 23.9%. Ap-

plying the same thrust of 355 N, the speed increases from 20 km/h to 25 km/h. Note that the appendage drag of for example assisted hydrofoils is not taken into account. For flat water, the potential of the Torpedo Mk.II EVO might still be relevant.

A second remark on the steep freeboard overhang underscores issues with wave resistance. When riding over a wave, temporarily, the draft locally increases which also significantly increases resistance due to the large waterplane width and submerged sponson drag. Having wave slamming against the hull from below (bow) and in the center (local draft 0.25m) and exponentially increased wetted area indicate the almost worst possible hull loading and drag increase. This makes the Torpedo the least suitable hull for medium sea states, especially at higher speed. In the front, the overhanging top-deck should be removed to prevent structural damage in waves. The Torpedo will still have a large flat bow deck as part of the hull itself, however, it will be reduced. As noted by the pilot(s) sail-

ing the Torpedo, the slamming is substantial and most probably near its maximum still ensuring a safe ride. Note that the Torpedo has served for around 7-8 years, and carbon fiber composites and resins degrade over time, especially in salt water and high temperature conditions. The original designed strength cannot be guaranteed, as minor fractures in carbon fiber composites are susceptible to progressive rupture. The wave slamming loads scale with V^2 as illustrated in Figure 2.7. This is a major concern, given that the future speed objectives are well above 35 km/h to ensure competitiveness. Lifting the boat by assisted foils may be the best way to mitigate the risk of wave impact failure when sailing in speed regimes over 30 km/h (16 kn). Attaching a full lifting foil system is not feasible due to stability/control complexity combined with cornering limitations and the large mass of the Torpedo Mk.II EVO (approx. 312 kg incl. pilot). It is evident that fully foiling systems are predominantly influenced by air drag and weight, consequently determining lift and drag in the foils.

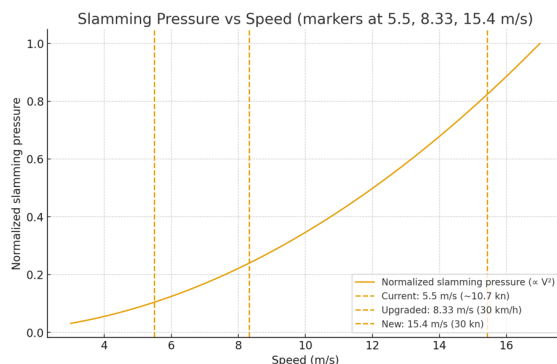


Figure 2.7.: Slamming pressure versus speed

The third remark concerns the maneuverability of the Torpedo Mk.II EVO. To ensure efficient cornering, it is imperative that the velocity of a craft is maintained, minimizing resistance, thereby preserving its momentum as it moves along the curve. The Torpedo has been observed to demonstrate a turning roll of approximately 8° to 15° at elevated speeds during competitive events. This implies that the side tunnel between the sponsons and center hull is fully closed (see Figure C.16) forcing a dramatic reduction in speed highlighted in the same performance indicators as discussed in wave encounter (hull slam-

ming, WWL, and wetted area). As this happens at one side only, it helps turning slightly faster initially, but after the speed reduction, the turning rate drops. To mitigate this, a larger sponson lift/displacement or more aggressive sponson draft-displacement could provide a smaller roll angle, not fully closing the side tunnels during turning. The sponsons may not be deeper in the water as the current setup as they also add drag in straight lines, limiting top speed. A second solution could be to add an aggressive draft-lift passive foiling system (near horizontal angled water piercing foils) that at (1) tries to lift the Torpedo for straight line efficiency and (2) tends to keep the roll as minimal in cornering as possible, preventing the dramatic speed reduction. It is imperative that the foils do not widen the craft to a considerable extent. This is because doing so would restrict the craft's ability to maneuver in close proximity to other vessels and buoys, thereby increasing the susceptibility of the foils to structural damage. Additionally, it is imperative to exercise caution with regard to foil ventilation when implementing shallow angled foils. Additionally, the stem length is a substantial factor in increasing the roll angle during cornering (Figure C.38). A long stem not only increases drag (trim influence, form drag and viscous drag), but also increases the roll moment of the vessel. Thus, a short stem is preferable in all cases. Specifically for cornering, if the force balance still tends to roll the Torpedo inwards, it might be beneficial to set the CG to a slightly higher position (e.g., lift the batteries) in order to prevent tunnel collapse.

Lastly, it is fundamental to examine the coefficients of form. That naval architects often use coefficients to address the complexity of hull performance was already delineated in Section 2.1.9. The most popular form coefficients in naval sciences provide dimensionless measures of the fullness or fineness of a vessel's geometry. They reduce the complex three-dimensional shape of a hull into ratios that compare its actual displacement or cross-sectional areas to those of simple reference volumes as cuboids. These coefficients are widely used in naval architecture because they allow systematic comparison between different hull forms, facilitate scaling, and may serve as input parameters in resistance,

propulsion, and stability predictions. The definitions of the block, midship, area of waterplane and prismatic coefficients are given in Appendix C.1.24.

For high-speed marine vehicles, these coefficients play different roles in design optimization. The block and midship coefficients are most critical in displacement ships, as high-speed craft often operate in semi-displacement

or planing regimes where dynamic lift dominates over buoyancy. In contrast, the prismatic coefficient becomes a decisive parameter, since it directly affects resistance at higher Froude numbers and determines whether a hull can transition smoothly into planing conditions. Similarly, the waterplane coefficient remains important, as it influences both seakeeping and stability at high speeds, especially during rapid maneuvers.

Thus, while all coefficients serve to characterize hull form in a concise and comparable manner, their relative importance shifts with vessel type and operational regime. In high-speed applications, emphasis is placed on prismatic and waterplane coefficients, whereas traditional displacement-related measures such as the block coefficient may provide less decisive guidance. The coefficients for the Torpedo's center hull are given by Figure C.18 and will be discussed in Section 2.2.3.

2.2.2. Remarks on the Shark

The Shark exhibits several geometry-related deficiencies, which led to its retirement after only one to two years in service. Nevertheless, it might be a better solution at higher speeds due to its planing characteristics, with some hydrodynamic adaptations in mind.

- **Stability** — Limited roll stiffness. Sponsons, passive hydrofoils... might be a straightforward fix.
- **Sliding stern** — During races, the pilot indicates the sliding of the stern (pivot around a vertical axis in front of the CG) when going around the track. Due to its flat bottom and soft chines, the stern has no “grip” or directional guidance. Ribs, hard chines, fins and/or keel appendages might address this issue. Remark this might increase the (turning) resistance as a drawback of improving controllability.
- **Dive** — The Shark has demonstrated diving behavior increasing drag and making its planing transition near impossible. The issue has been addressed by adding mass aft the CG, improving trim. This concern should have been anticipated for in the development phase, as it is a typical behavior for axe-type planing bows / hulls (see Section 2.1.8). Note that adding stern mass might sufficiently adapt for improved trim. However, is it a brute and sub-optimal way to improve performance locally and mitigate one specific issue — especially in planing-type craft — as it results in a list of other performance concerns.
- **High mass** — Despite being approximately 40 kg lighter than the Torpedo Mk.II EVO, the mass impact is severe as a planing hull suffers or gains more from a certain total mass number (see Section 2.1.7).
- **Underpowered** — Based on testing and insights from former team members with competitive experience, it appears that the Shark hull has not yet demonstrated its full performance potential. Owing to its current mass distribution, vertically displaced bow, and limited available power, the vessel has not reached the high-efficiency planing regime in which a controlled layer of air is distributed beneath the hull. Refinements to the hull geometry may help advance the onset of this “air introduction” region along the speed curve; however, a concurrent increase in available propulsion power may ultimately be required to fully exploit the design's capabilities.

On the left, the hull of the Shark is shown, without its overhanging decks.

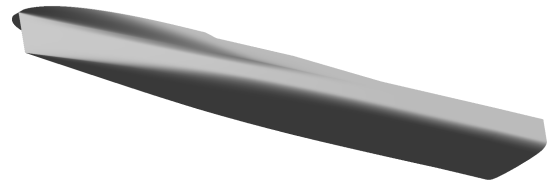


Figure 2.8.: Default hull of the shark

The discussion of graph C.18 will follow below. This plot facilitates geometry comparison based on numerical evaluation.

A small waterplane has been shown to be advantageous due to its reduced wave-making properties, as exemplified by the SWATH vessel (Appendix C.1.7). The waterplane area directly governs the hydrostatic stiffness in heave and pitch. A large waterplane means strong restoring forces when the vessel meets a wave. This leads to high accelerations, discomfort, and higher added resistance in waves. In other words: the larger the waterplane, the stronger the coupling with the sea surface.

When analyzing roll stiffness based on the waterplane, the hulls show significantly different properties. In general, derived from displacement theory, an over-all low waterplane encourages limited wave making and higher displacement efficiency. The Torpedo is a great example of that with an average waterplane of approximately 3.5 m^2 over a roll range of zero to eight degrees. The substantially steep exponential waterplane curve shows the aggressive roll stiffness and accelerations that the pilot also indicated during sea trials. This is also related to hull slamming as a steep curve indicates high-intensity wave encounters on the underside of a hull. The exponential submerged frontal area of the torpedo makes perfect sense, as the vessel tends to lose all its momentum when cornering. This is supported by the staggering plot of the width at waterline, which easily climbs to 1.1 m.

In comparison, the Shark has an high — and also more roll independent — waterplane curve compared to the Torpedo. This implies that the Shark experiences significant higher wave making resistance in displacement regimes with an average waterplane of approximately 4 m^2 over a roll range of zero to eight degrees. In contrast, the frontal submerged area of the Shark is exceptionally low and constant over the analyzed roll range. That means the Shark is capable of shifting to planing regime more quickly and steadily, even while cornering. This makes the Shark a more predictable hull in terms of maneuvering efficiency compared to the Torpedo Mk.II EVO. However, it must be noted that the current configuration of the Shark usually leads to larger roll angles. This can be directly related to the limited roll stiffness observed in the Shark's geometry graphs.

2.2.3. Coefficients of Form Reviewed

Figure C.18 in Appendix C.1.25 reports the principal form coefficients for the Torpedo Mk.II EVO (semi-displacement) and the Shark (flat-bottom planing). The Shark exhibits consistently higher values: C_B increases by ≈ 0.052 ($\sim 10.9\%$), C_M by ≈ 0.053 ($\sim 7.5\%$), C_{WP} by ≈ 0.042 ($\sim 5.8\%$), C_P by ≈ 0.021 ($\sim 3.2\%$), and C_{VP} by ≈ 0.031 ($\sim 4.8\%$). These trends are technically consistent with a transition from a semi-displacement geometry to a more planing suitable geometry.

A thorough report is added in Appendix C.1.25, discussing both the comparison between the Torpedo and Shark and quantifying speed-regime relevance while highlighting limitations. In essence, the practical reading for the present hulls is as follows. Given C_P values of 0.673 (Torpedo) and 0.694 (Shark), both hulls are well positioned for the $Fr \approx 0.5\text{--}1.0$ band where transom ventilation and trim stabilization matter most; the Shark's slightly higher C_P and C_{WP} favor a smoother, more robust planing condition in $Fr \gtrsim 0.6$, whereas the Torpedo's lower C_B and C_M are advantageous in $Fr \lesssim 0.55$ to mitigate the hump and wave-making as it transitions through semi-displacement.

2.3. Hydrodynamic Drag Reduction Strategies

The resistance of high-speed craft is usually determined largely by the hydrodynamic forces acting on the hull, and by extension, the proportion of the vessel's wetted surface area in contact with water. For small vessels in the 5–10 m class, operating in the semi-displacement to planing regime, hydrodynamic resistance accounts for the vast majority of the propulsive power demand [18, 14]. At speeds between 20 and 40 knots, hydrodynamic drag typically represents 85–95% of the total resistance, while aerodynamic drag seldom contributes more than 5–15%, depending on hull form and superstructure exposure [19].

This disparity underscores why hydrodynamic optimization is the most important focus for performance boats in this class. Once the hull is partially or fully lifted, total resistance is increasingly governed by aerodynamic drag, which remains significantly lower than hydrodynamic resistance. Consequently, reducing wetted surface area and wave-making resistance early in the speed range, ideally achieving lift-off at low speeds, yields a net reduction in total drag. In that case, the aerodynamics become more important. This is not only because of the smaller water-boat interaction, but also because the boat will move much faster from this point on due to the quadratically increasing influence of aerodynamics. So, lifting the boat not only increases attainable maximum speed but also delivers disproportionately better energy efficiency and fuel economy. For electrically powered craft, where available energy is tightly constrained by battery capacity, the importance of minimizing hydrodynamic drag at low-to-moderate speeds is magnified. A more detailed hydro-aero drag balance discussion, backed by simple equations, is provided in Appendix C.1.26, quantifying the importance of aerodynamics for the new hull.

As highlighted, a central principle of high-speed craft design is that performance improvements depend on progressively lifting the hull out of the water while simultaneously maintaining stability and safety. A boat that achieves dynamic lift or foil-borne operation at 15–20 knots will require far less propulsive power to reach 30 knots than one that transitions at 25 knots. Early lift reduces the drag curve, shifts the operating point into a more favorable regime, and may cut shaft power demand by more than half compared to a purely displacement-bound hull.

This logic motivates a stepwise evolution of hull configurations, each representing an established drag-reduction strategy. Beginning with a slender displacement hull, successive transitions to planing, tunnel-hull designs, and finally to hydrofoil-assisted or fully foil-borne vessels reflect a continuum from buoyant to dynamic or aerodynamic lift [18, 20, 21]. In the following comparative assessment (see Figure C.19), a representative 8 m, 350 kg displacement vessel operating at 30 knots was selected as the baseline case. The displacement hull was assumed to require 100% of the reference power.

Through successive modifications, the succeeding approximate improvements can be anticipated:

- **Planing hull** - Reduction of wetted area by dynamic lift yields a 30% reduction in required power at 30 kn. For constant installed power, speed increases by 25%.
- **Tunnel hull** - Entrapment of air between demi-hulls reduces wetted surface and spray drag, cutting power demand by an additional 15% and increasing speed by 8%.
- **Hydrofoil-assisted hull** - Partial lift provided by submerged foils decreases hull–water interaction significantly. Power at 30 kn can be reduced by 35% relative to the tunnel hull stage, with 18% speed improvement for constant power.
- **Passive hydrofoil** - At this stage, the hull is completely lifted clear of the water, with lift and stability provided by foil geometry. A further 30% reduction in power is achieved, with 10% speed increase.

- **Active hydrofoil** - The addition of real-time control enables minimization of induced drag and adjustment to sea conditions. This offers 10% additional power reduction and 3% higher speed relative to passive foils.

Expressed cumulatively, these steps illustrate the profound effect of minimizing hull–water interaction: a vessel requiring 100% of baseline shaft power at 30 kn in displacement mode can, after successive upgrades, operate at the same speed on only 24% of that power. Conversely, with unchanged installed power, top speed can increase by 80%, from 30 kn to above 50 kn. This is illustrated in Figure C.20.

These comparative figures highlight a crucial point: the earlier in the speed range that lift is achieved, the greater the lifetime benefit in performance, range, and economy. For the 5–10 m class of small high-speed craft, drag reduction is thus not a matter of marginal gains but the defining factor in whether a design can achieve efficient, safe, and sustainable operation.

Notes

- These are mid-case values for calm water that racing teams and boat manufacturers experience with well-optimized geometry and good setups; reality swings with sea state, CG, trim, ventilation/cavitation margins, and propulsor match.
- “Tunnel hull” assumes effective air-entrapment (pad/tunnel) done right; poorly designed tunnels can add drag.
- “Passive hydrofoil” means surface-piercing or fixed-incidence foils that are self-stabilizing by geometry; “active” adds control authority, letting one run smaller wetted area and better angles of attack more often and reducing the risk of ventilation.
- At 8 m and 30 kn the Froude number is very high, so a “fast displacement” hull at that speed is already near semi-planing behavior; the step to true planing still yields a big gain via reduced wetted area and wave-making.
- Due to the slenderness, the Torpedo Mk.II EVO is more of a semi-planing hull in this context. Derived from Figure C.19 the efficiency gain from well-designed hydrofoils is in the order of magnitude of 66% ($= (70 - 24)/70$). This is realistic as fully foilborne hydrofoil craft have demonstrated power savings exceeding 60% relative to equivalent planing or semi-displacement hulls at the same speed, as documented in naval and industrial hydrofoil programs since the 1950s [22]. Candela excels in this technology, claiming to save over 80% of power [23].
- The baseline displacement hull is not seen as slender-type configuration. This implies that a wide range of improvement is possible (30% is a gross estimation). Improvements indicated from planing all the way up to active hydrofoils are more accurate and supported by discussions with industry experts and broad ranging scientific literature.

2.3.1. Hull Length, Beam, and Cross Section Considerations

Optimizing the principal dimensions of a hull—its length, beam, and cross-sectional geometry—is one of the most effective strategies for reducing hydrodynamic drag. During the preliminary design stage, adjustments in these parameters significantly influence both total resistance and the magnitude of the characteristic “hump” in the resistance curve associated with the transition between displacement and semi-planing regimes.

Increasing the length-to-beam ratio (slenderness) generally reduces the wave-making component of resistance by mitigating wave interference, as the greater spatial separation between the bow and stern wave systems decreases the transverse wave amplitude [24]. Conversely, hulls with small slenderness ratios exhibit earlier and more pronounced humps in the resistance curve. The effect of hull length becomes particularly evident in displacement and semi-displacement vessels, where a longer hull postpones the critical Froude number corresponding to the wave-making hump, resulting in smoother performance and lower peak resistance. This phenomenon is insignificant for planing craft, where dynamic lift dominates over buoyancy and wave-making effects (see Section 2.1.7).

For relatively wide hulls, the geometry of the transverse cross-section plays an important role in determining viscous resistance. A rounded bilge shape typically produces a smaller wetted surface area for a given displacement compared to a rectangular or hard-chine section, thereby reducing skin friction. However, as the slenderness ratio (length-to-beam) increases beyond approximately $L/B > 8$, the influence of cross-sectional shape diminishes and viscous drag becomes primarily a function of surface area and flow quality rather than local curvature effects as shown in the empirical work of Molland et al. [19]. The same principle applies when comparing deep versus shallow draft configurations: reducing the underwater cross-sectional area lowers wetted surface and frictional drag but may impact static stability and payload capacity.

As discussed in Section 2.2.3 on the coefficients of form and in Section C.1.7 on SWATH-type configurations supported by the first theoretical decomposition of hydrodynamic drag in Section 2.1.9, reducing the waterplane area provides a further advantage in wave-making resistance by limiting the generation of surface disturbances. Extending hull length while maintaining displacement therefore increases slenderness and delays the wave hump, whereas reducing beam or waterplane area mitigates wave energy losses as the flow deflection is smoother energy regeneration becomes larger (see Section 2.1.7).

In the planing regime, total resistance becomes more dependent on wetted surface area and dynamic lift generation. A sufficiently flat stern region near the keel can enhance lift and trim, thereby reducing the effective wetted area at speed. Further reduction can be achieved by employing lift strakes — longitudinal chines or spray rails designed to redirect flow downward and outward. These features both deflect water away from the hull and promote clean flow separation, reducing drag while maintaining lift (see Section 2.3.2).

Finally, beam width influences both hydrodynamic and aerodynamic performance. A wider beam increases static stability and can provide additional lift through planform area, as seen in air-cushion or tunnel-hull configurations. However, once aerodynamic forces become dominant—typical in the high planing or semi-airborne regime — drag reduction increasingly depends on aerodynamic shaping rather than further hydrodynamic refinement.

In summary, the interplay between hull length, beam, and cross-section defines the hydrodynamic efficiency envelope of a vessel. Longer and more slender hulls minimize wave-making resistance; rounded sections and moderate drafts reduce frictional losses; and in planing craft, optimized beam and stern geometry ensure a favorable balance between lift generation and wetted area, ultimately determining overall drag performance.

2.3.2. Lift Strakes and Fins, Trim Tabs, and Shaft Trim

As widely recognized in literature and confirmed by numerous experiments, reducing the wetted area of planing craft is one of the most effective ways to enhance hydrodynamic performance. Beyond hull geometry alone, several appendages can be applied to improve lift generation, trim control, and flow separation, thereby directly influencing resistance and stability. Common devices include lift strakes, fins, hydrofoil-assist elements, trim tabs, and propeller shaft trim systems.

Lift strakes and fins serve as simple yet highly effective devices to manage wetted area and pressure distribution during acceleration and steady planing. They function by deflecting the flow downward and outward, producing local dynamic lift while promoting earlier flow separation along the hull underside. This reduces both wetted surface and skin friction, improving efficiency once the vessel reaches its steady running attitude. The correct placement and angle of such strakes are crucial: if too deep or too flat, they may increase wetted surface or induce turbulence, whereas well-designed strakes can stabilize the transition to planing, suppress or damp porpoising, and improve trim behavior at intermediate speeds. Inverted chine designs share similar characteristics with lift strakes (lift, deflection, stability) but usually doesn't reduce wetted area, crucial when transitioning to significant planing Froude numbers (> 3). Side-mounted or bow fins also follow similar principles as lifting strakes, locally supporting the hull and providing improved pitch damping and stability. In more substantial configurations, hydrofoil-assist elements can also be incorporated, most easy to integrate in tunnel-hull or catamaran designs. When mounted between sponsons, such foils contribute to early lift generation and smoother transitions to planing at moderate speeds. However, as discussed in Section 2.1.6, their design must remain fully passive, and they should never support the full vessel mass at racing speeds, as overall stability must remain ensured by the hull itself. Ventilation and cavitation prevention are critical design constraints, as once these occur, lift becomes unstable and the benefits are rapidly lost.

Even for displacement or semi-displacement vessels, partial lifting of the hull with strakes can significantly improve hydrodynamic efficiency by reducing both wavemaking and viscous resistance. A clear example of this concept is the

Hull Vane, a fixed hydrofoil positioned below the stern of the hull. Developed originally for large displacement vessels, this passive foil provides stern lift, reduces pitching motion, and generates a forward thrust component by interacting beneficially with the upward flow in the wake. Experimental data and company documentation [25] indicate that the Hull Vane can carry up to 60% of the vessel's mass, depending on hull form and operating condition, while simultaneously damping the stern wave system. Similar foil-assist principles are increasingly applied in smaller catamarans and high-speed craft, where the device functions as an auxiliary lifting surface that helps the hull reach the planing regime more efficiently and with improved stability.

Trim control appendages also play a critical role in drag management and overall safety. Trim tabs — flat, adjustable surfaces mounted on the transom — are among the most common control devices used to manipulate hull attitude during acceleration or steady running. By deflecting downward, the tabs generate a hydrodynamic moment that forces the bow downward, thereby reducing the trim angle. This can be particularly beneficial when the vessel has an aft-biased center of gravity, for example due to a heavy outboard motor or stern drive, which otherwise promotes excessive bow-up trim, porpoising or a delayed transition to planing. Properly adjusted trim tabs can thus improve efficiency, handling, and safety. However, their main disadvantage lies in the fact that these appendages greatly increase the effective wetted surface area — potentially by 3-12% in high-speed planing conditions, as supported by the relatively small running wetted area shown in Figure C.23 [26] compared to common trim tab sizes. Consequently, their deployment must balance the gains from improved trim against the penalty of drag. They are most profitable in configurations where trim imbalance is the dominant source of inefficiency.

Shaft or motor trim, briefly discussed in Section 2.8.3, provides another method of adjusting hull attitude through propulsion vector control. Tilting the thrust line downward (trimming in) applies an upward force on the transom and plants the bow down, while tilting upward (trimming out) has the opposite effect. Moderate ad-

justments in motor trim can fine-tune the planing angle and wetted area, but excessive angles risk propeller ventilation and safety. Therefore, propulsor trim is most effective and efficient as a supplementary control mechanism rather than a primary means of adjusting hull balance.

In summary, the reduction of wetted area through the use of lift-generating and trim-controlling appendages remains an essential principle in high-speed hull design. Lift strakes and fins offer a lightweight and passive method for managing hydrodynamic lift and flow separation, while hydrofoil-assist concepts such as the Hull Vane demonstrate the potential for integrated, passive lift systems to improve efficiency, even in displacement modes. Trim tabs and motor trim allow real-time control of running attitude and performance but must be used judiciously to avoid counterproductive drag increases. When appropriately combined and tailored to the specific hull form—such as a tunnel-hull catamaran or high-performance racing craft—these devices collectively enhance stability, efficiency, and seakeeping while ensuring safe and predictable dynamic behavior across the full operating range.

Review of Preliminary Trailing Wave Analysis within the Team

In previous studies [27], a first attempt was made to derive the resistance magnitude of the Torpedo and a potential new hull based on the amplitude of a trailing wave. As known in naval architecture, much can be learned from the analysis of trailing waves. However, as illustrated by Figure 2.5, wavemaking — closely related to the shape and magnitude of the trailing waves — is only a part of the total resistance. Depending on the hull's design focus (displacement versus planing) and aimed cruising speed range, the ratio of impact on the resistance components varies significantly. Displacement hulls (low speeds) are mainly affected by wavemaking and high-speed planing craft suffer mainly from viscous resistance as reflected in the Crouch's Equation 2.8.

In the analysis of the aforementioned examination [27], the experimental frame indicates the fixation of the hull during the free surface CFD calculation. This outcome is generally considered invalid in the field of naval studies, particularly with regard to semi-displacement and planing craft, as evidenced by the observed discrepancy between CFD test results and full-scale drag test findings. During tests on any scale, the minimal allowed degrees of freedom (DOF) should include pitch (trim) and heave of a hull, as these factors significantly impact wavemaking and wetted area resistance, as well as transition effects and spray at higher speeds. The sole means by which this study could have

yielded more precise results is at minimal velocities ($Fr \ll 0.4$). At such speeds, the effects of dynamic trim and heave are less substantial and are known for the static pose (though still not negligible) or for every higher fixed speed based on the established trim and heave. In this particular instance, it is imperative that the spatial constraints of the hull be established at a fixed position that has been determined through comprehensive full-scale analysis. It is important to note that this approach does not encompass the evaluation of stability and behavioral effects. Nevertheless, it is a worthwhile idea to evaluate the waveform generated behind the hull to assess partial resistance components.

2.3.3. Surface Microstructures: Sharkskin, Riblet Film & Hydrophobic Coatings

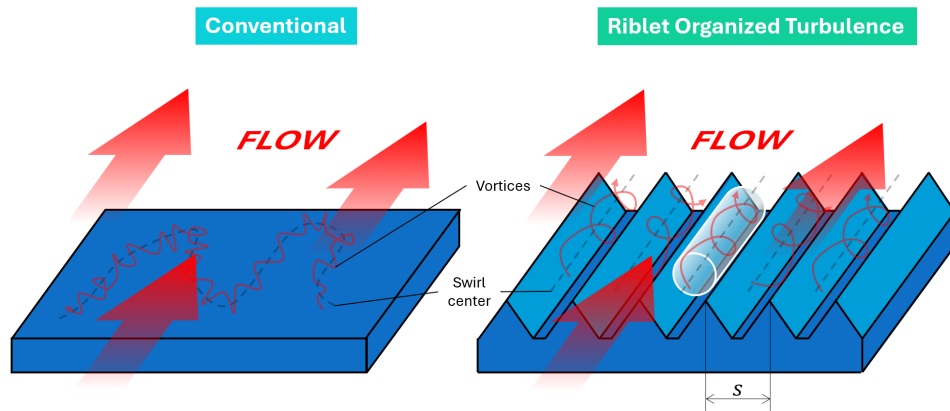


Figure 2.9.: Riblet film geometry

Biomimetic microstructures inspired by sharkskin (3D structure) — most commonly implemented as engineered riblet films (2D extrusion along flow direction, see Figure 2.9) — aim to reduce macro skin-friction by subtly reorganizing near-wall vortices. A large body of foundational work shows that longitudinal, streamwise-aligned riblets can yield up to about 3 to 8% skin-friction reduction in turbulent boundary layers when sized near an optimal non-dimensional spacing ($s^+ \approx 10\text{--}20$, see Figure C.21 and Equation C.19 for the conventional non-dimensionalization in wall-bounded turbulence theory) and carefully aligned with the local flow (e.g., Walsh reports net drag reductions up to $\sim 8\%$ in skin friction with strong sensitivity to size, yaw, and Reynolds effects) [28, 29]. Modern airline retrofits of sharkskin-inspired riblet films (e.g., AeroSHARK) report about $\sim 1\%$ fuel-burn reduction in daily service, consistent with partial-area application and operational constraints [30, 31]. In athletic swimsuits, the practical application showed significant drag reduction, but it was permanently banned in competitive swimming in 2010 as it is considered a major technological advantage [32]. More details and equations can be found in Appendix C.1.28 and are recommended to read when mi-

crostructure drag reduction use is considered.

In contrast, hydrophobic coatings do not create riblet-like turbulence manipulation, but they can indirectly preserve low surface friction. Superhydrophobic schemes that trap stable air layers (e.g., Salvinia-inspired) can achieve high laboratory drag reductions, but durability and air-retention at planing speeds remain open challenges [33, 34].

In conclusion, for a well-aligned, well-sized application on the running underside of a planing hull, a practical expectation is on the order of 3-8% reduction of local skin friction, translating to < 1 up to a few percent in total resistance reduction at race speeds, depending on how much of R_T is frictional. Benefits diminish if the stern flow is highly three-dimensional, intermittently separated, or influenced by spray, misaligned or contaminated (biofilm, salt crust). This can negate the effect or even add drag. As riblet films must be aligned with the mean flow to reduce drag, CFD may suggest optimal orientation. Finally, sharkskin replicas with 3D denticle crowns are not guaranteed to reduce drag. Several controlled studies show denticle arrays can increase drag while riblets of comparable scale reduce it [35, 36].

2.3.4. Biofouling Prevention Coatings

Biofouling — the accumulation of microorganisms, algae, or small marine organisms on wetted surfaces — is a major contributor to increased frictional resistance in marine vessels. Even a thin biofilm or light

algal growth can raise total drag by 10-30% within weeks of immersion, depending on temperature and salinity conditions [37, 38]. Advanced antifouling or fouling-release coatings, including silicone-based and biomimetic microstructured surfaces inspired by sharkskin, have been shown to delay attachment and maintain controlled boundary layers. However, such treatments are less critical for racing or experimental vessels that are not continuously immersed, as short operational durations prevent significant fouling buildup. Nevertheless, maintaining a clean and undamaged hull surface remains essential, since even minor surface roughness, salt-crystal deposits, or degraded microstructures can noticeably increase skin-friction resistance and reduce the repeatability of hydrodynamic measurements.

Note

Riblet geometry creates micro-scale flow separation zones that discourage the settlement of algae and bacteria [39], while also reducing unorganized turbulence and consequently viscous drag as discussed in the previous section [40].

2.3.5. Bulbous Bows

The bulbous bow remains one of the most recognizable hydrodynamic drag reduction strategies in ship design, intended primarily to reduce wave-making resistance through constructive interference between two distinct wave systems. As described by Datawave Marine Solutions (DMS) [41], the bulb generates its own bow wave whose trough coincides with the crest of the vessel's primary bow wave, thereby causing partial cancellation of the total wave system. When correctly tuned, this mechanism results in a measurable reduction in wave-making resistance. However, this effect is highly speed-dependent and only effective within a narrow range — typically no more than ± 2 knots around the design speed. Beyond this range, the interference pattern becomes unfavorable and can increase drag rather than reduce it. More info can be found in Appendix C.1.29.

2.3.6. Steps — Wetted Area Reduction or Air Lubrication?

Many insightful drag reduction studies arise from the field of naval engineering. Stepped hulls have become increasingly common in the design of high-speed monohulls and multihulls, particularly in offshore racing and in the design of experimental autonomous craft. They aim to reduce the wetted surface area and associated viscous resistance at planing speeds by promoting controlled flow separation at one or more transverse discontinuities, or steps, along the bottom of the hull. This concept originated in the design of early aviation floatplanes and was later refined for powerboat applications; it is now applied across a wide range of displacement-to-length ratios. Contemporary practice typically sees single-step geometries used for sports and patrol craft, while multi-step configurations are employed in long offshore racing hulls. Here, successive steps help sustain lift at high Froude numbers and distribute dynamic pressure more evenly over the hull's bottom [42, 43, 44].



It's inaccurate to describe that “Steps introduce air under the boat. The water that runs aft of a step contains small bubbles, and since aerated water produces less drag than solid water, stepped hulls are more efficient.” according to Jim Russell

A common oversimplification in popular literature describes stepped hulls as introducing air under the boat to reduce drag through water aeration. As Jim Russell clarifies, this explanation is inaccurate: the hydrodynamic benefit arises not from bubbles or aerated flow, but from flow separation and re-attachment, which decreases the wetted area while maintaining sufficient lift to support the craft [42]. When correctly designed, the step geometry forms a controlled ventilation cavity behind each discontinuity, allowing the aft surfaces to re-enter the water flow at a lower wetted fraction and reduced overall drag. In mono-hulls, the effect can substantially improve planing efficiency. In catamarans for example, steps are sometimes applied in each demi-hull (see Figure C.22) to stabilize the pressure distribution and reduce wetted beam at high speed.

The transition from single to multi-step configurations is primarily justified when operating at very high Froude numbers or when the hull must sustain lift across a broad longitudinal range. AeroMarine Research's analyses and experimental data indicate that multi-step arrangements are most effective above approximately 60 knots, where the flow behind the first step remains ventilated and attached along the succeeding surface. Below this range, additional steps often yield minimal benefits or even increased drag

due to premature ventilation or re-attachment losses [43].

Pascal AirHull and Beneteau Air Step are exemplar companies who both experiment and further extend this principle by integrating controlled air channels and cavities beneath the hull to regulate the pressure field artificially, actively reducing wetted area, but increasing system complexity. It is worth mentioning, but still, this type of controlled air introduction is rarely used as it is not yet a matured technology.

However, recent computational and experimental studies have shown that stepped hulls exhibit distinct performance limits. Numerical simulations by [26] on the Taunton Series C1 planing hull revealed that as speed increases, the air cavity initially expands — reducing the wetted area by up to 78% — but later contracts as the vessel enters the “triangle-planing” regime. While transcending through this stage, additional wetting of the aft bilge (wave interference) increases resistance and concentrates hydrodynamic pressure near the center of gravity, leading to reduced longitudinal stability and the onset of porpoising. The study concluded that for this particular geometry, optimal operation occurs below $Fr_V \approx 6$ (see Appendix C.1.31), beyond which the hydrodynamic benefits of the step diminish.

In summary, stepped hulls remain a dominant design strategy for modern high-speed craft, offering significant drag reduction when properly engineered and operated within their optimal speed envelope. While single-step designs are common in high-performance mono-hulls, multi-step configurations appear primarily in offshore racing hulls and specialized large scale catamarans. Their successful implementation requires precise control of cavity formation, step geometry, and center-of-gravity alignment to avoid instability and performance degradation at extreme speeds. This complexity is mitigated by the use of AeroMarine Research's design software, estimating the hull's step performance.

2.4. Special Considerations in High-Speed Craft

The design and operation of high-speed craft require specific attention to phenomena that are either absent or negligible at lower velocities but become dominant once the vessel transitions into high-speed (> 20 kn) semi-displacement or planing regimes. These effects arise from the interplay between hydrodynamic, aerodynamic, and structural mechanisms, and they critically influence performance, safety, and reliability.

A primary concern is drag, which manifests in multiple forms at high speed. Spray formation, turbulent flow separation, and induced lift at the hull-water interface are among the most common phenomena. Advanced design strategies, such as spray rails, optimized hull geometries, or the controlled introduction of air beneath the hull, are often employed to mitigate losses and enhance efficiency. Closely related is the issue of noise generation, which typically indicates turbulent vortices around the hull or superstructure. Beyond its impact on comfort and stealth, excessive noise is also symptomatic of energy losses that limit overall performance.

Structural and operational safety represent another central dimension. The high accelerations, vibrations, and dynamic loads encountered during operation necessitate careful consideration of material selection, reinforcement strategies, and redundancy in critical systems. Pilot safety is equally crucial, as cockpit ergonomics, impact mitigation, and exposure to extreme conditions significantly affect operational limits.

These aspects underline that high-speed craft cannot be approached merely as scaled versions of conventional displacement vessels. Instead, they represent a unique class of marine systems where performance optimization, hydrodynamic

control, and structural resilience must be addressed simultaneously. The following section will outline the most common instabilities encountered in high-speed craft, providing a foundation for the subsequent analysis of their mitigation and control strategies.

Speed related dangers highlighted in boat racing include [11]:

- **Blowover** - Longitudinal (negative) instability resulting in a bow-up moment leading to an airborne hull.
- **Barrel Roll or Chine Tripping** - Lateral instability caused by abrupt direction change or change in momentum with a high CG.
- **Nose-Dive or Stuff** - Longitudinal (positive) instability resulting in a nose dive and abrupt speed reduction.
- **Chine Walking** - Side-to-side oscillation (mainly vee hulls) [45].
- **Porpoising** - Cyclic oscillation in pitch and heave in planing regime.
- **Airborne Hull** - Due to large lift forces or blowover, the hull gets fully airborne (stability criteria).

In general, one can state that dynamic instabilities of a well-designed hull mostly arise from improper trim, hence its importance in the study. An advice for maximizing safe, stable behavior: use as little positive trim as possible (for example $1-1.5^\circ$). Increased trim (higher angle-of-attack/AoA) leads to a set of instabilities more quickly and with more drama [45]. As pounding and porpoising are most common and can occur at lower speeds, thorough analysis of the phenomenon is required. The extensive description and definition of both wave and aerodynamic induced variants along with mitigation strategies are delineated in Appendix C.1.32. When evaluating both stability and safety of a hull, it is highly recommended to read this section first. Note the susceptibility to porpoising behavior is a critical performance concern, often non-linear and changing with even the slightest design tweaks. In Figure C.27 an example of a theoretical porpoising characteristic is shown, similar to the predictive plots of the Tunnel Boat Design Program (TBDP) used in this project.

2.5. Simulation Methods and Limitations

2.5.1. Value of CFD

Computational Fluid Dynamics (CFD) represents one of the most powerful tools for modern naval hydrodynamic design and performance evaluation. Within this project, it is employed to analyse and compare numerous hull configurations, appendage layouts, and operating conditions more efficiently than would be possible through purely physical model testing (see Section 5.3.3). As outlined in Section 2.6.2, conventional towing tank experiments are constrained by scaling effects (Froude–Reynolds similarity), facility size, and model construction accuracy. These limitations are particularly critical for high-speed vessels, where small discrepancies in trim, surface roughness, or inaccurate free-surface deformation can affect the results. Large-scale test campaigns that could mitigate these issues typically require investments exceeding \$10k per use [46], making them unfeasible within the scope of this research project.

CFD overcomes these limitations by directly resolving the governing flow equations at full scale, eliminating the need for model rescaling and associated uncertainties. Additionally, it allows time-dependent simulations that capture dynamic trim, spray formation, and variable thrust vector effects, which are challenging or impossible to implement experimentally at small scale. Furthermore, CFD provides the flexibility to include unsteady boundary conditions such as variable thrust or wave encounters, enabling realistic analysis of transient hydrodynamic phenomena. This is particularly valuable for assessing wave-induced behavior, added resistance and

cornering, which would otherwise require complex wave-making systems and large-scale facilities.

The approach also offers a major advantage in terms of development time. Design iterations can be completed rapidly, providing feedback on geometry changes, force balances, and flow patterns. Combined with complementary empirical tools such as the Tunnel Boat Design Program (TBDP) by AeroMarine Research, CFD forms the core of a hybrid analysis methodology that maximizes, flexibility, accuracy and efficiency within the available resources.

2.5.2. CFD Approaches and Multi-physics Constraints

Several CFD methodologies are available for marine hydrodynamics, varying in accuracy and computational demand. For this study, simulations are performed using a multi-phase, time-dependent solver with a resolved free surface. This approach captures the dynamic interaction between water and air, including planing lift generation, spray formation, and wave–hull interference. Steady-state solvers were deemed inadequate, as they cannot reproduce the transient effects of hull motion response.

Transient simulations enable the inclusion of changing speed, time-dependent thrust vectors, and environmental perturbations, allowing evaluation of the hull’s dynamic stability and trim response under realistic conditions. While the present analysis does not include Fluid–Structure Interaction (FSI) coupling for structural deformation and stresses. Load patterns are extracted directly from the CFD results as reliable indicators of hydrodynamic loading and indirectly highlighting local stress concen-

tration zones. The description of the most common mathematical CFD frameworks for hull design are added in Appendix C.1.33.

Dynamic Fluid–Body Interaction (DFBI) extends CFD by coupling body motion and flow field dynamics. The solver iteratively calculates the hydrodynamic forces and moments acting on a floating or moving body, updates its motion in six degrees of freedom, and recomputes the flow based on the new position and attitude. This

two-way coupling enables the prediction of trim, heave, and roll in response to external forces such as thrust or waves. However, DFBI simulations are numerically sensitive and computationally more demanding and often 50% [47] more costly than comparable fixed-body RANS (Reynolds-Averaged Navier–Stokes) cases since both the body’s inertia and transient fluid effects must be resolved simultaneously. Despite this, DFBI is invaluable in studying planing stability, porpoising, and dynamic lift behavior in racing hulls and forms a natural extension of RANS-based CFD frameworks used in this project.

CFD cannot replace empirical testing and analytical methods, but should be seen as a complementary tool. A good simulation needs good reviewing. It is strongly advised to let a CFD setup and its results be reviewed by an indepen-

dent fluid mechanics expert. Nevertheless, empirical validation and mesh independence studies are always useful in proving CFD accuracy. Consequently, CFD can play a pivotal role in the project. [48]

For validation of qualitative predictions, empirical models, and hull optimization strategies, the software Flow3D HYDRO was selected. It is specifically tailored for naval and coastal applications, providing computationally efficient solvers for free-surface flows. Its proven capability in simulating planing and semi-planing dynamics makes it well suited for analyzing behavior, resistance, and seakeeping characteristics in the present context. The following section provides references to a more detailed technical discussion of the solver.

2.5.3. Flow3D HYDRO Computational Fluid Dynamics Framework

Flow3D HYDRO [49] is a commercial, volume-of-fluid (VOF)–based CFD solver developed by Flow Science Inc. and specializes in free-surface hydrodynamics. It is widely applied in civil, coastal, and naval engineering for predicting wave formation, resistance, and dynamic stability of marine structures and vessels. The solver provides a physically accurate representation of the air–water interface, making it particularly suitable for hull resistance, pressure distribution and seakeeping studies of high-speed craft configurations in comparison analysis. Arguments for the specific use and properties of the Flow3D Hydro solver can be found in Appendix C.1.34, discussion strengths and limitations, applications, etc.

In summary, Flow3D HYDRO provides a reliable CFD framework for analyzing resistance, wave-making, and dynamic stability of hulls in free-surface flows. Its VOF formulation and embedded-geometry approach offer an optimal compromise between numerical accuracy, modeling flexibility, and computational efficiency, making it an appropriate tool for iterative hull evaluation and comparative studies in high-speed craft.

2.5.4. Wave Influence and Spray Modeling

The total resistance of high-speed craft does not solely depend on the hull geometry and the flow conditions in calm water, but is strongly affected by wave–hull interaction and spray formation. Even in moderate sea states, short-period surface waves introduce fluctuating hydrodynamic loads that alter trim, heave, and effective wetted surface area, resulting in an increase in mean resistance. The ITTC recommendations for high-speed marine vehicles account for this through empirical correction factors that quantify the added resistance in waves relative to calm-water resistance. For small, short-crested waves of approximately 0.4 m in height, such as those commonly observed near Port Hercule (Monaco), the added resistance for semi-planing and planing hulls may range from 5% to 15% depending on wave encounter frequency and relative heading [3]. Experiments [50] demonstrated that added resistance in waves for planing hulls can be significant and non-linear with wave height [51]. In general, a correction

factor increases with the Froude number and wave steepness, emphasizing the importance of considering sea state when evaluating endurance or power demand during near-port races. For analysis added resistance of a specific wave height, the ITTC [3] recommends testing regular waves of different length or test in irregular waves, directly finding the resistance by averaging the measurement.

Beyond wave-induced resistance, the modeling of spray and its interaction with the hull surface plays a key role in predicting hydrodynamic efficiency and stability. In planing regimes, hulls generate distinct spray sheets at the chines and strakes, which contribute to both dynamic lift and additional aerodynamic drag. AeroMarine Research's Tunnel Boat Design Program (TBDP) integrates empirical and semi-empirical formulations (e.g., Whisker Spray) to estimate various spray effects including induced drag by accounting for local hull deadrise, speed, and strake geometry [52, 53]. The TBDP facilitates the analysis of spray trajectories and wetted area distribution under various trim conditions.

Strakes and spray rails — introduced in Section 2.3.2 and 2.3.6 — represent one of the most effective passive geometrical technique to manage spray behavior and local lift distribution. Properly positioned spray rails deflect the water flow away from the hull surface, thereby reducing wetted area and frictional drag. Conversely, poorly aligned or oversized strakes can trap spray under the hull, increasing resistance and reducing stability. As shown by AeroMarine Research [52], fine-tuning the angle, spacing, and height of these appendages are essential to balance hydrodynamic lift and spray-induced drag.

In summary, accurate prediction of total resistance at high speed requires accounting for both the direct hydrodynamic influence of incident waves and the aerodynamic and hydrodynamic effects of spray. Computational and semi-empirical models, such as those implemented in the TBDP framework, provide a practical approach to quantify these effects and inform iterative hull optimization for varying sea states.

Sea Condition	Significant Wave Height [m]	Added Resistance (% of R_T)
Calm water	0.0	0
Light chop (harbor exit)	0.3	3–7
Small coastal waves	0.5	5–15
Moderate swell	1.0	10–25

Table 2.3.: Indicative added resistance factors in small waves (Based on ITTC Recommended Procedure 7.5-02-03-01.5 and [3, 50])

2.6. Experimental Methods for Validation

2.6.1. Wind Tunnel Testing of Models

Wind tunnel testing remains one of the most established experimental techniques for the investigation of aerodynamic and hydrodynamic flow phenomena. It enables controlled, repeatable testing of scaled models or components under steady or transient flow conditions, allowing the direct observation and measurement of surface pressures, flow separation, wake development, and vortex structures. In the context of marine and high-speed surface vehicles, wind tunnels provide an efficient platform for assessing hull appendages and superstructures when water-based testing is impractical or preliminary insights must be gained quickly. Although originally developed for aeronautical applications, the fundamental principles of similitude and non-dimensional analysis make wind tunnels a valuable complementary tool to towing-tank experiments and computational fluid dynamics (CFD) simulations. Nevertheless,

the challenge of achieving dynamic similarity, particularly regarding the Reynolds number at small geometric scales, imposes limitations on the quantitative interpretation of wind tunnel data for marine configurations.

The Reynolds number is defined as

$$Re = \frac{VL}{\nu} \quad (2.13)$$

where V is the flow speed, L is a characteristic length, and ν is the kinematic viscosity of the fluid (air or water). Matching Re between a model and the full-scale prototype (i.e. $Re_{model} = Re_{full}$) ensures that inertial and viscous forces scale appropriately, so that boundary layer thicknesses, separation behavior, and transition loci are dynamically similar [54, 55].

Suppose a full scale prototype operates at 30 knots (≈ 15.4 m/s). Consider a scale model ($\lambda = 20$), with characteristic length $L_{model} = L_{prototype} \cdot \lambda^{-1}$. If one attempted to match Reynolds number in air (with $\nu_{air} \sim 1.5 \times 10^{-5} \text{ m}^2/\text{s}$), the required model velocity would be

$$V_{model, required} = V_{prototype} \times \lambda = 15.4 \times 20 \approx 308 \text{ m/s},$$

which is completely infeasible in conventional low-speed tunnels. More advanced installations often aim to allow for large scale models or offer high-speed capabilities instead.

Hence in practice, wind-tunnel models are often run at much lower speeds (in this study $V_{model} \leq 10 \text{ m/s}$). That implies

$$Re_{model} \ll Re_{prototype},$$

so that viscous effects, transition, and separation may occur differently in the model than in full-scale operation. Wind-tunnel data then require careful interpretation, corrections (when available), or supplement by CFD [56, 57].

One classical approach to mitigate the Reynolds-mismatch problem is to test in a fluid with much lower kinematic viscosity, such as water. The kinematic viscosity of water at room temperature is about $\nu_{water} \approx 1.0 \times 10^{-6} \text{ m}^2/\text{s}$, roughly 10–15 times smaller than that of air ($\nu_{air} \approx 1.5 \times 10^{-5} \text{ m}^2/\text{s}$) [58]. Because $Re \propto V/\nu$, a model in water will achieve the same Reynolds number at a velocity much lower than in air.

As an illustrative example: continuing with the 30 kn full-scale case and a 1/20 model, the same Re is targeted. In water:

$$V_{model, water} = \frac{Re_{prototype} \nu_{water}}{L_{model}} = \frac{V_{prototype} L_{prototype}}{\nu_{air}} \cdot \frac{\nu_{water}}{L_{prototype}/20} = 20 \frac{\nu_{water}}{\nu_{air}} V_{prototype}.$$

If $\nu_{water}/\nu_{air} \approx 0.0667$, this becomes $V_{model, water} \approx 2 V_{prototype}$. Concretely, at $V_{prototype} = 15.4$ m/s, one might need $V_{model, water} \approx 20.6$ m/s, which is far more manageable (though still high for many water tunnels). This is orders of magnitude lower than 308 m/s required in air for the same scale. In fact, NASA and other agencies have explored cryogenic or variable-density tunnels to further increase Reynolds numbers in air without having to deal with excessive flow velocity [57, 59].

Note that the analysis in Reynolds similarity is explained for the air flow assessment of the superstructure of the new boat. Quantitative evaluation of submerged appendages or hull geometry is unfeasible as the transition of a 30 knots full-scale water flow to a down-scaled air flow model is practically unachievable.

Despite the Reynolds mismatch in the wind tunnel tests of this study, the results can be useful for:

- Strouhal number calculation (quantify vortex shedding, $St = \frac{fL}{V}$, high $Re \rightarrow St \approx C^e$)

- Qualitative visualization
- Design comparisons
- Flow behavior insight
- Educational/demonstration purposes
- Input for CFD validation

Laser-Particle Interaction & Optical Flow Visualization Techniques

In extend of the argumentation on wind tunnel testing, flow measurement techniques like Particle Image Velocimetry (PIV) should be mentioned as they are used to visualize and quantify the flow acquired in the wind tunnel. PIV measurements capture at least a pair of high-speed images to calculate discrete velocity vector fields of a fluid flow cross-section by particle illuminance with a laser sheet and tracking the (often microscopic) particles. In this study, PIVlab (Matlab-based) is used for computation and analysis. Results of the PIV evaluation can be found in Section 5.2.1 (if time constraints allowed discussion) and background is provided in Appendix C.1.35. Optimized image pair settings found through experimentation with the PIVlab software are discussed in Appendix C.1.36.

Construction of a Medium Size Wind Tunnel For this study, a simple wind tunnel setup was built in order to gain quick insights in the flow around a specific body, superstructure, or appendages on a hull. The tunnel features a test volume of $0.3 \times 0.3 \times 0.5$ m or 0.045 m³, which is sufficient for small-scale tests or analysis of individual components. The input flow is laminarized by adding a long intake duct and a tube grid for which the diameter is around 1/10 th of its length, also ensuring the flow is stabilized when entering the tunnel space. Still, it is not advised to test model flow right in front of the intake grid due to the micro turbulences induced in the flow by the surface friction of the tube grid. As a consequence, even for models placed far away from the intake, it is not possible to test super-small scale models.

To limit undesired turbulences, the glass top and side of the tunnel is properly sealed and all surfaces are smooth to limit the size of the boundary layer. The decreased internal pressure due to speeding air (Bernoulli) might suck air in through leakages generating local turbulences disturbing the measurement.

The air is extracted by an appended exhaust chamber driven by a large centralized custom EDF-turbine (Electric Ducted Fan). The cubic

chamber ensures a smooth air extraction limiting the streamline deflection near the extraction tube grid. The control system is connected to both the ESC (Electronic Speed Controller) of the EDF as well as the laser and mirror oscillator (if used). In the used setup, the used cinema box camera is not wired to the wind tunnel controller. However, the camera is connected to the PC via WiFi to speed up the testing procedure and process images instantly. The setup is illustrated in Figure C.29.

Seeding Techniques and Tracer Particle Types

Accurate optical flow diagnostics (e.g., PIV, particle tracking) require an ensemble of tracer particles seeded into the flow that both scatter sufficient light and follow the fluid motion well. A variety of seeding strategies exist, broadly categorized as solid particles, liquid aerosols / fog, and gas-filled or bubble tracers. Each liquid or gas tracer has trade-offs in terms of particle size, light scattering, density,

and flow fidelity. In Table C.12 common options are summarized. The first three categories have been experimentally explored in this study. In choosing a tracer, one aims for a compromise: smaller particles track rapid accelerations and turbulence better (lower inertial lag) [60, 61], while larger particles improve scattering and therefore produce stronger images.

Gravity / Settling Concerns and “Too-Fast Falling” Particles When tracer particles have a density significantly greater than the fluid, gravitational effects can bias measurements, especially in orientations where the flow is horizontal (i.e. particles falling down the wind tunnel floor). If tracers settle faster than they can be carried by the flow, they will drop out of the measurement region or distort the apparent velocity.

Gravitational tracer effects mitigation might imply the use of a vertical wind tunnel. The tunnel operator may reduce the flow velocity slightly below nominal so that the upward flow counters the settling velocity, keeping particles at the intended test rate. However, compensating for fast settling by lowering flow rate complicates the test: the required slower flow may depart further from the intended conditions, and the particles’ gravitational acceleration can still introduce bias. Ideally, tracers should reach terminal velocity before entering the measurement space. In essence, the particles’ mass/flow resistance must not be too high relative to the fluid. - If the flow direction is horizontal, one must ensure the particles’ terminal velocity is negligible rel-

ative to the flow speed so that their vertical drift does not contaminate the in-plane velocity field.

In summary, one must carefully choose particle size, density, and seeding orientation to balance visibility versus flow fidelity. Even when perfect conditions are unattainable, qualitative flow features and comparative studies remain very useful.

Construction of a Vapor Separator & Atomizing Particle Accelerator The build of an experimental wind tunnel in this study has led to the construction of a custom local seeder device called the Particle Accelerator. The seeder-atomizer uses distilled water and excites droplets using a piezo-electric oscillator membrane as shown in Figure C.31. The output of the device — a pressurized air-water mix — is fed through a tube which is inserted as a “smoke” nozzle in the test chamber. To ensure laminar vapor outflow of the nozzle, the Particle Accelerator intake flow can be continuously restricted to match the wind tunnel flow rate. Specifications and a corresponding simulation of the flow are provided in Appendix C.1.39.

In the next paragraphs, the two build setups for making laser section-cuts of the tracer flow are briefly discussed. A third method with a commercial digital projector was also used to introduce extra functionalities and flexibility. The goal was to be able to change colors and section-cut shapes, blocks and perform multiple, simultaneous analysis, but was depreciated due to limited performance (brightness and contrast issues, wide line projections, etc.).

Construction of a Laser-Mirror Oscillator for Particle Flow Section-Cuts The first laser setup — serving as an experiment — has proven to be a good, adaptive way of making section-cuts for observation, but not for PIV-measurements. A small mirror is mounted together with a vibration motor on a compliant, wide sheet spring. On the fixed base, a laser is mounted, aiming at the mirror at a set angle. By regulating the voltage from 0.7-1.13 V, the motor triggers the first mode of the mirror-motor assembly, resonating at 54 Hz. Increasing the

voltage over 1.13 V triggers the first lateral bending mode resonance, causing the laser pattern to shift from a straight line to a figure eight. Below 0.7 V, the motor cannot start properly. The voltage controls the angular range of the laser: low voltages make a small bright laser section-cut and a high voltages make larger oscillations providing wider, less bright, projection angles. To achieve a high-brightness, small-angle projection, the control algorithm on the wind tunnel controller anticipates on the required start voltage. A voltage of at least 0.7 V is applied to over-

come static friction and, in a few milliseconds, is lowered to minimum 0.5 V to achieve a smaller stable oscillation magnitude. A more accurate, high-speed technique to reach a more homogeneous projection is by using a polygon mirror as demonstrated in laser-printers. Motor-mounted mirrors used in festival equipment may allow for more shape and color flexibility in exchange for added complexity and cost.

For high-speed images, the disadvantage of the oscillation mirror technique lies in its relatively

low frequency and time-dependent position. At high the shutter speeds and framerates required for PIV analysis, it is unfeasible to exploit such a method. Therefore, the mirror-oscillator applications within this project are limited to visuals only, and were never used for measurements. The demonstrations illustrated in Figure C.30 are performed in combination of a full-domain smoke generator. A green laser is chosen as this is the brightest color to the human eye (maximum sensitivity) and most cameras.

Construction of a Laser-Optical Line Projection To disperse a focused laser beam (collimated) in a line projection, multiple types of lenses are available. In this study, two types of lenses were tested: (1) a cylindrical lens and (2) a Powell lens. The core difference can be found in its intensity distribution. A cylindrical lens has a near-Gaussian distribution and a Powell lens propagates a more rectangular intensity distribution [62]. That makes the Powell lens an attractive solution for image capture, as it proves to show more consistent readings in PIV-analysis. Note that glass lenses provide increased line clarity compared to the plastic variants tested. Because of these advantages, a glass Powell lens is used in the final setup. The setup is illustrated in Figure C.32.

2.6.2. Water Testing of Scale Models

Apart from full-scale experiments — which have optimal accuracy — reduced models are often used to test various adaptations quickly and often is the only option that is economically acceptable. Unfortunately, due to inconceivable physics scaling, the translation is quite complex. For accurate results, Froude similarity scaling is insufficient, especially above hump speed.

Data Scaling and Performance Translation

Over the last two centuries, the prediction of ship speed and power has evolved into the accurate scientific methods we know today. From the digital modeling in CFD (Section 2.5.1) to empirical testing in advanced high-precision towing tanks compensating for the curvature of the earth, each technique has its place in hull development. While CFD can be faster to set up and achieve similar accuracy, the value of tow tank testing is still indisputable in the field of naval architecture. The tow tank test is still considered the absolute standard for resistance and power linked studies and serves a judicial role if full-scale power requirements deviate from the binding contracts of a shipyard [46]. The importance of tow tank testing is likewise reflected in test details the ITTC's guidelines imposes.

In this study, the empirical testing on scale and full-scale models mainly serves validation purposes. Superior accuracy is not as strict in this study, but well-chosen translation procedures still apply. To succeed in ideal transformation of a ship model measurement to a full-scale prototype, both geometric and dynamic similarity

must be achieved, as generally stated throughout this literature review [19]. Geometric similarity exists between a model and a prototype if ratios of length, area and volume are equal and characterized by a single scale factor λ (be cautious about the inverted convention for λ in some equations used in literature).

$$\lambda = \frac{L_{prototype}}{L_{model}}, \quad \lambda^2 = \frac{S_{prototype}}{S_{model}}, \quad \lambda^3 = \frac{\nabla_{prototype}}{\nabla_{model}} \quad (2.14)$$

On the other hand, dynamic similarity exists only if the ratios of all forces are the same. As illustrated in Section 2.1.9, the resistance of a ship can be decomposed in two major elements: (1) the wave making resistance (Froude) and (2) the frictional resistance (Reynolds). Note that dimensionless parameter C_{wave} doesn't scale as $C_{viscous}$.

$$C_{viscous} = f(Re) \quad \Rightarrow \quad V_{model} = V_{prototype} \frac{V_{model}}{V_{prototype}} \cdot \lambda \quad (2.15)$$

$$C_{wave} = f(Fr) \quad \Rightarrow \quad V_{model} = \frac{V_{prototype}}{\sqrt{\lambda}} \quad (2.16)$$

Thus, geometric and dynamic similitude cannot be satisfied at the same time (changing the fluid to a less viscous fluid like mercury for model testing is theoretically possible solution). It is proclaimed: “one can only achieve partial dynamic similarity in ship resistance scaling.” As wavemaking is considered most important in numerous naval studies, the industry standardization states that model tests must be conducted with Froude similarity and that Reynolds inconsistencies must be accounted for by implementing compensation coefficients. Partial dynamic similarity will be achieved by translation of model data for a corresponding test speed defined by the Froude similarity construction. A more detailed, illustrative explanation of the procedure is given in Appendix C.1.41.

The key of using estimates in towing experiment procedures lies in its consistency by standardization: tests and rescalings are never perfect, but within certain margins, designs are comparable. This approach provides a reliable method to compare scaled vessels in test tanks around the globe.

The conversion methodology used in this project is based on the ITTC HSMV 7.5-02-05-01 (2017) guidelines. To improve conversion calculation in studies and future tests, a computation script “Model-Ship Converter” (see Appendix C.7.1) was developed, incorporating all relevant equations and parameters for HSMV (High-Speed Marine Vehicles) translation. This is part of the research transfer and accessibility goal, highlighted in the introduction. This principle has been extended further by writing all scripts in Python, a programming language that is publicly available to all stakeholders and future members/sponsors, etc. unlike Matlab.

The input file for the Model-Ship converter script is formatted as a CSV-file containing 2 or 3 colons: model velocity V_M [m/s], resistance R_{TM} [N] and running wetted area S_{run} [m²] (S_{run} is not mandatory, if not provided $S_{run} = S_0$ and scales as λ^2) and should be put in the same folder as the script. After filling in a number of constants related to water conditions, detailed resistance components (if known) and other data, the script generates a CSV file containing all data available, including the full-scale translation of model data. A distilled version of the data, containing only speed and resistance, can be imported in, for example, BCD to conduct a more advanced analysis or comparison of the results.

Scaling of Foils, Fins & Stabilizers When considering scale model testing of fins and foils, it is important to analyze the true scaling effects of such appendages. Similarly to the drag Equation C.18, the (wing) lift equation is defined as

$$L = \frac{1}{2} \rho V^2 S C_L \quad (2.17)$$

Due to geometric Equations C.2 for S and ∇ and the relation of Froude similarity (Equation C.2) the laminar flow lift scales as

$$\frac{L_{model}}{L_{prototype}} = \left(\frac{V_{model}}{V_{prototype}} \right)^2 \left(\frac{S_{model}}{S_{prototype}} \right) = \left(\frac{1}{\sqrt{\lambda}} \right)^2 \left(\frac{1}{\lambda^2} \right) = \frac{1}{\lambda^3} \quad \Rightarrow \quad L_{model} = \frac{L_{prototype}}{\lambda^3} \quad (2.18)$$

As the lift force L scales identically with the mass or static displacement of a model (λ^3), the foils and lift fins may simply be geometrically scaled with the model. Remark that cavitation and flow separation effects do not scale correctly ($Re_{model} = \frac{Re_{prototype}}{\lambda^{3/2}}$) and results related to these phenomena are therefore invalid. $C_{L_{model}} = C_{L_{prototype}}$ is true assuming the same flow regime and angle of attack (AoA).

Note

- Full-scale prototype is often shortened as “S” (ship), model as “M” in literature equations.
- ρ deviations are not accounted for in Equation 2.18, but could also be included resulting in

$$\frac{L_{model}}{L_{prototype}} = \frac{\rho_{model}}{\rho_{prototype}} \cdot \frac{1}{\lambda^3} \quad (2.19)$$

- In calculations, it was preferred to use the length-based Froude number rather than the volumetric Froude number. This makes the analysis more accessible and easier to understand for those unfamiliar with naval engineering and more complex non-dimensionalization (see Appendix C.1.47). For the sake of completeness, the volumetric Froude number has been added to the calculation software. In contexts studying true planing behavior, it may be more accurate to switch between the two and use the following speed relation accordingly:

$$V_S = \begin{cases} \sqrt{\lambda} V_M, & \text{if matching } Fr_L \text{ (displacement and semi-disp.)} \\ \lambda^{1/6} V_M, & \text{if matching } Fr_V \text{ (true planing)} \end{cases} \quad (2.20)$$

Sea States

High-speed racing events such as the Monaco Energy Boat Challenge (MEBC), held annually in early July in Port Hercule, place stringent constraints on sea state conditions due to the fragility and high speeds of the participating craft. While the event’s technical regulations (published by Yacht Club de Monaco) do not explicitly stipulate allowable sea states, it is universally understood that races proceed only under benign conditions — typically “smooth” to at most “slight” waves — given the vulnerability of lightweight, low-drag race hulls to slamming, capsize, and general loss of control.

Sea states are classified according to the significant wave height H_s (the mean height of the highest one-third of waves), using standardized scales such as the WMO or Douglas sea state scale. The WMO’s sea state definitions are summarized in Table 2.4 [63, 64].

Degree	Description	H_s [m]
0–1	Calm (glassy, rippled)	< 0.1
2	Smooth (wavelets)	0.1 – 0.5
3	Slight	0.5 – 1.25
4	Moderate	1.25 – 2.5

Table 2.4.: Douglas sea state scale — significant wave height H_s ranges

Regional marine climatology of the Ligurian Sea indicates that in early July, Port Hercule experiences predominantly minimal wave activity — generally confined to sea states 2 and 3 — thanks to Mediterranean sheltering and light summer wind conditions (NOAA/ORCA Mediterranean wave climatology) [65]. Such conditions help explain why F1H2O-style race hulls — characterized by lightweight composite construction and sharp planing — are considered appropriate: they excel in calm-to-smooth seas, where hydrodynamic predictability and pilot control are maximized. However, with a slightly raised deck, the tunnel hull designs prove to have excellent sea-keeping at higher speeds as well.

Although the MEBC does not publicly define a maximum sea state threshold for races, prudent design practice dictates that racing hulls be engineered to operate safely up to at least sea state 2 (“smooth”) at high speed, with H_s up to approximately 0.5 m and sea state 3 (“slight”) at low speed, with H_s up to approximately 1.25 m, assuming significant wavelengths ($\gg$ LWL). Beyond this threshold, risks such as hull slamming, foil ventilation, and loss of control escalate rapidly. The emphasis on flat water is not solely for efficiency (minimizing drag and maximizing lift-to-drag ratio), but also for maintaining pilot safety and preserving structural integrity under dynamic loads.

In the context of hull design, the imperative is therefore twofold: (1) minimizing hydrodynamic resistance via early and efficient lift (planing or foiling), and (2) ensuring sea-keeping capability and stability under the light-to-moderate wave environment most commonly encountered during summer racing in Port Hercule. This

dual focus underpins the rationale for high-speed racing hull design, particularly in the energy-constrained and technologically ambitious context of the MEBC.

When transferring real sea conditions to scale-model testing, it must be noted that wave height is scaled according to Froude similarity. Under this principle, length and wave height scale linearly with the geometric scale ratio, while time and velocity follow the square-root law of the same ratio. Thus, a prototype wave height of 1.0 m at sea state 3 corresponds to only 0.1 m in a 1/10 physical model. This linear scaling of wave height ensures that model tests reproduce the hydrodynamic loading environment realistically, provided that Froude scaling laws are respected. This is described in more detail in the Recommended Procedures 7.5-02-05-01 2017 High Speed Marine Vehicles Resistance Test section 4.7.3 Calculating the added resistance by the ITTC.

Notes

- For sea trials on the North Sea of both the existing and future hulls, the team aims to collaborate with the Belgian Navy base located at Bruges-on-Sea.
- It is possible to extract wave data live from the Blue Accelerator research platform in Ostend for checking conditions and matching with the test conditions in this study.
- For stability evaluation, only subjects relevant to the final design have been approached. For example, stability related to wave patterns and wave-speed relations like in following and quartering sea conditions will not be evaluated (encounter wave grouping, surf-riding, parametric rolling, etc.) [66]. These phenomena are mainly relevant at open sea with high sea states and larger vessels and the race will not be held at critical sea conditions.

Common terminology

- Swell — long-period, smooth, rolling waves generated by distant weather systems (limited during MEBC)
- Chop — short-period, confused, locally generated wavelets that make the water “choppy”, typically seen as a rough, bumpy water surface with small but closely spaced waves (dominating during MEBC)

2.7. Competition Regulations and Compliance Criteria

Since the AI competition is fairly new, few rules apply to the design of the hull. Essentially, the rules are limited to the maximum length of 12 meters for the vessel, the requirements for the chosen propulsion power source, and the specifications for custom communication systems. A design should focus on both performance and safety. If the distance between driver and battery cannot be preserved in a new design, a fireproof wall or battery encapsulation is required.

Currently, there is no power restriction, which allows for a significant increase over the currently used 6 kW. For the 2026 MEBC edition, the team's goal is to keep and optimize the use of 6 kW, but prepare subsystems and hardware for the switch to 15 kW. For more information, consult the MEBC race documentation.

2.8. Industry Best Practices

2.8.1. Materials and Manufacturing in Industry Practice

Industry best practices for material selection in high-performance planing hulls, such as those used in F1H₂O racing, prioritize an optimized balance between stiffness, strength, impact resistance, and mass reduction. The current standard employs carbon fiber-reinforced polymer (CFRP) laminates combined with lightweight structural cores (e.g., PMI, SAN, or Nomex honeycomb) to achieve high stiffness and low hydrodynamic drag. Strategic reinforcement through internal ribs and localized thickening enhances load distribution and crash safety without excessive weight gain.

The recent adoption of natural fiber composites, such as the flax-based cockpit in the Torpedo hull, reflects a growing trend toward sustainability while maintaining energy absorption capacity. For a next-generation F1H₂O hull, a hybrid carbon-flax composite sandwich structure with the latest high-performance core, delivered by one of the team's sponsors, is recommended — combining top-tier stiffness-to-weight ratios, excellent fatigue and impact resistance, and improved environmental performance when paired with bio-based resins. In the new hull, total mass and stiffness should be a key design focus.

2.8.2. Conventional Hull Geometry & USV Selection Approaches

In extend of Section 2.1.5, it is important and enlightening to observe what hull types are common for USVs and autonomous test platforms. In discussion with Vlaams Instituut voor de Zee (VLIZ, Marine Robotics Center), VIRO and Offshore Positioning Systems (OPS), stability is pointed out as the major concern for perception under (sonar) and above water (cameras, LiDAR, radar) both for scientific data acquisition as well as (GNSS-independent) navigation as facilitated in short term path planning and obstacle avoidance. Therefore, catamarans are often the first choice for small platforms that require moderate seakeeping abilities (XOCEAN).

For larger scale vessels, all types of hull shapes appear from X-bows (USV Gobelijn, 5000 kg - 9 m, operated by VLIZ) to medium speed deep-V (Exail, Sea Machines Robotics) and slender displacement configurations (Saildrone). Some are build for medium speeds (EXAIL), some are build for ultimate seakeeping at lower speeds featuring large displacement capabilities (USV Gobelijn) and build for endurance or continuous operation (Saildrone). In conclusion, all vessels can be used as an USV platform. The selection process is near identical as for a manned vessel. However, its key difference lies with

increased stability and reliability at the intended speeds being the most critical design requirements. Therefore, the adopted hull selection method in this project is valid and supported by the main stakeholders and naval architects.

2.8.3. Trim Control in Tunnel Hull Racing and Autonomous Systems

Trim control plays a decisive role in the hydrodynamic performance of high-speed planing craft. For tunnel hull designs such as those used in F1H₂O competition, minute adjustments in the outboard motor trim angle can significantly alter drag, lift distribution, propeller ventilation, and cornering capability. In racing applications, skilled drivers continuously adjust trim to balance speed, stability, and turning efficiency. In autonomous platforms, replicating this practice demands novel control strategies that can sense and adapt to dynamic conditions without human intuition.

Trim in Conventional and Tunnel Hull Design

Conventional Hulls In traditional planing hulls, trim directly influences wetted surface area and thus drag. Positive trim lifts the bow, reducing wetted area and increasing speed on straights, whereas negative trim plants the bow, increasing wetted surface and enhancing directional stability at the expense of drag. A schematic is provided in Appendix C.1.42.

Tunnel Hulls Tunnel hulls introduce an additional complexity due to the aerodynamic lift generated by the tunnel effect between the twin sponsons. Proper trim is essential to balance hydrodynamic lift from the sponsons and aerodynamic lift from the tunnel. Excessive positive trim risks blowover, while excessive negative trim increases gearcase drag and destabilizes the yaw axis. The narrow operational window of tunnel hull trim makes active control particularly critical.

F1H₂O Specifics In F1H₂O racing, outboard motors are equipped with power trim systems that allow dynamic adjustment. Drivers typically employ positive trim of approximately +1° to +3° in straight-line racing to minimize drag and maximize top speed. During cornering, trim is returned toward neutral or slightly negative values (0° to -1.5°) to increase sponson engagement and initiate sharper turning. The full operational range rarely exceeds 5°, yet within this range lies the fine balance between competitive lap time and catastrophic instability. In general, trimming down is always the safer choice. This helps stabilize the craft.

Cornering Sequence in F1H₂O

1. **Entry** — At the onset of a corner, experienced drivers preemptively reduce trim to stabilize the bow and prevent understeer. This maneuver occurs before steering input is maximized.
2. **Mid-Corner** — In mid-corner, trim remains slightly negative to keep sponsons engaged and maintain lateral grip while acceleration forces can exceed 6 G. Excessive negative trim, however, can lift the stern and increase the risk of propeller blowout due to high slip angles.
3. **Exit** — As steering input decreases and acceleration resumes, trim is returned progressively toward positive values. This sequence restores efficiency and avoids bogging the craft in transition to the next straight.

Autonomous Implementation

Sensor Requirements An autonomous control system must replicate the human driver's ability to sense dynamic states. This requires integration of inertial measurement units (IMU) for pitch, yaw, and roll; GPS for speed and trajectory; propeller torque sensors for ventilation detection; and hydrodynamic pressure sensors to monitor flow attachment.

Control Strategies Autonomous trim control may be implemented through three primary strategies: (1) rule-based deterministic control, where trim is adjusted according to predefined speed and turn-radius maps; (2) adaptive feedback control, where trim is dynamically altered to maintain optimal pitch and yaw stability; and (3) data-driven learning methods, where telemetry is used to train models that predict the optimal trim angle from measurable states. During normal, straight-line operation of a well-designed craft, the motor trim should remain parallel to the water's surface.

Simplified Steering-Proportional Trim

Concept A practical approach for autonomous systems is to link trim adjustments proportionally to steering input, optionally corrected by yaw-rate feedback. This method ensures that increased steering angles are accompanied by modest trim-down commands, while straight-line operation defaults to positive trim. A similar approach is to link trim mechanically to the steering input by tilting the steering axis as shown in Figure C.35. Graphs and concept diagrams are provided in Appendix C.1.43. Developed in this study, the concept was inspired by the world of RC boating where trim control is conventional too. Although simplifications are common, the fact that trimming systems are standard-equipped in RC setups underscores their importance.

Such a proportional system offers both robustness and predictable behavior. It can approximate the human practice of trimming down in corners and trimming up on straights. Unfortunately, the simplification sacrifices performance in nuanced conditions. Unlike human drivers, a proportional controller cannot anticipate corner entry or exit. Furthermore, environmental disturbances such as wind and chop are not inherently compensated. An active calibration with a proportional trim system may be the best hybrid approach. A comparison with manual control is given in Figure C.37.

Risks of Improper Trim and Mitigation

Porpoising Excessive positive trim leads to unstable pitch oscillations, or porpoising, which reduces efficiency and risk loss of control. Automated systems must detect periodic pitch oscillations (2–6 Hz) and respond with corrective down-trim.

Propeller Blowout Excessive negative trim, especially at high yaw angles, can destabilize flow around the gearcase, resulting in ventilation and propeller blowout. Preventive measures include monitoring propeller torque and yaw rate spikes to trigger corrective up-trim.

Blowover Uncontrolled bow lift at high speeds can result in aerodynamic blowover. Autonomous systems must therefore limit maximum positive trim values as a function of speed to prevent this catastrophic failure. However, paradoxically, a blowover can also be initiated by an excessively nose-down trim setting. When the outboard is trimmed too far negative, the lower unit and gearcase are forced deeply into the water, generating significant drag at the stern. This drag tends to pivot the hull around the center of lift, producing an abrupt nose-up reaction. As speed increases, aerodynamic pressure under the tunnel amplifies this pitch-up motion. In such

conditions, even though the initial trim setting forces the bow downward, hydrodynamic imbalance can cause a sudden rotation that lifts the bow uncontrollably. This counterintuitive mechanism has been observed in high-performance tunnel hulls: a driver attempting to stabilize the craft with down-trim instead triggers a violent pitch excursion. To prevent this, control systems must not only cap positive trim but also enforce speed-dependent limits on negative trim authority. Monitoring pitch rate and tunnel pressure provides an additional safeguard, ensuring that corrective nose-down trim cannot unintentionally load the stern to the point of inducing blowover.

Propeller Depth and its Influence on Trim

In general, a low depth propeller position is more optimal than a deep one (ignoring hydrofoil setups). What comes first to mind is the extra drag caused by the strut, stem or driveshaft housing (see Section 5.1.5). However, there is another noteworthy phenomenon related to trim which is causing substantial drag and potential dangerous behavior at high speeds when not accounted for. A stem that is excessively large (i.e., a deep propeller), as used in the 2025 competition, will generate more torque on the hull body. This will push the bow out of the water depending on the

amount of thrust generated. An illustration is provided in Appendix C.1.45.

To compensate for the additional bow lift, trimming “in” is required as illustrated in Figure C.38, depending on the throttle conditions. This additional trim will provide the most optimal performance for the setup, but it will cause extra losses as the thrust vector is not aligned with the direction of travel (may provide lift and therefore decreased wavemaking resistance, slightly compensating for the negative effect). Note that a short stem also has these losses, but they are much less significant. High surface piercing props are another way to limit these types of losses even more. However they suffer from many other effects beyond the scope of this thesis.

As discussed in the previous section, shortening the span between the keel and the propeller’s “torpedo” provides hydrodynamic advantages. However, in bottom mount systems, these gains are constrained by minimum clearances mandated between the propeller blades and hull or rudder structures, in order to avoid vibration, cavitation, and critical flow interference. The table in Appendix C.1.46 summarizes recommended clearance proportions (relative to propeller diameter D) from various maritime authorities.

In summary, trim control in F1H₂O-style tunnel hulls is a highly sensitive balance that determines both speed and safety. For autonomous vessels, a steering-proportional trim approach can provide a safe and functional baseline, though at the cost of peak performance. Advanced sensing and predictive algorithms remain essential for replicating human-level lap time while avoiding the inherent dangers of improper trim.

Note

The optimal / maximum high (without ventilation) of a stern mount propulsion system can best be fine-tuned on the full-scale vessel, for example with an adjustable jack plate (compatible with outboards), highlighted during industry feedback sessions.

3. New Hull Concept & Design

This chapter presents the development of a new high-performance hull concept, translating the insights obtained from experimental, numerical, and theoretical investigations into a coherent and optimized design. The objective is to define a hull configuration that significantly outperforms existing designs in terms of speed, efficiency, stability, and cornering behavior, while remaining feasible within practical constraints such as power availability, manufacturability, safety, and long-term operational reliability.

The chapter is structured to progressively refine the design, starting from clearly defined performance targets and operational requirements. These targets reflect not only hydrodynamic and aerodynamic considerations, but also broader system-level constraints, including stakeholder expectations, sponsorship value, and competition relevance. Based on these requirements, a conceptual design framework is established that integrates both functional performance and aesthetic identity, acknowledging the increasing importance of visual presence and recognizability in racing.

Subsequently, the chapter specifically focuses on hull geometry optimization and performance prediction. A tunnel hull configuration is selected as the most promising baseline due to its superior high-speed efficiency, dynamic stability, and cornering potential. Dedicated velocity prediction tools and expert-driven iterative optimization are employed to explore the influence of key geometric, mass, and balance parameters. This approach enables rapid and reliable performance estimation while maintaining sufficient physical fidelity for design decision-making.

The outcome of this chapter is a finalized hull concept (outer geometry) with well-defined characteristics, operational limits, and performance envelopes. The resulting design serves as a foundation for future structural, control, and autonomous system integration, while providing a technically sound and competitive platform for high-speed solar and AI-focused autonomous racing.

3.1. Performance Targets

Minimal top speed and cornering should be significantly better than current designs to be technically and economically feasible. Determining the appropriate limits involves a complex balance that encompasses much more than performance alone. For example, the significance of advertising and value for sponsors may not be underestimated as various important stakeholders pointed this out themselves. In general, it is stated that the hull must be useful along a wide range of speeds above 50 km/h. Note that the Shark is recommended below this speed and tested up to 36 km/h in this study. The new design will consider 15 kW as input power, while the Shark currently uses 6 kW. Additionally, greenwater and hull slamming must be minimized identical to all other hull's requirements. Cornering must be as good or better than the current fleet (38°/s minimum according to the improved Shark's simulations) and include level turning (small roll deviation).

3.2. Conceptual Design Development

The conceptual design development phase establishes the foundation for the vessel's geometry, configuration, and overall design philosophy focusing on aesthetics. It bridges the transition from performance-driven requirements and flow dynamics to a coherent physical concept that embodies both functional efficiency and visual identity.

3.2.1. Aesthetic Design Considerations

Although flow dynamics and structural performance remain the foremost priorities in new hull's design, aesthetics form an increasingly relevant aspect of the team's identity and public perception. Previous generations of the UAntwerp Solar Boat were perceived as functional but visually uninspiring, according to feedback from stakeholders, sponsors, and the general public. The next-generation design must therefore integrate a clear aesthetic vision that complements technical performance while reflecting the team's ambition, professionalism, and innovative character.

The objective is to achieve a visual language that is both futuristic and purposeful, much like contemporary hypercar and yacht concepts that blend aerodynamic function with sculptural form. The hull and superstructure should visually express motion, precision, and performance even at rest. Although aesthetics will never outweigh hydrodynamic optimization, the combination of smooth curvature continuity, well-placed break lines, proportion, and surface effects can enhance perceived quality, evoke emotion, and attract media attention, promoting sponsorship interest. Bright and vivid color schemes — potentially determined by the main sponsor — are encouraged to differentiate the vessel from competitors and to increase visibility on the water during racing events. In case no principal sponsor defines a palette, the team may establish a consistent visual identity based on its own color scheme to ensure continuity in public communication and long-term recognizability. This is of particular importance given that the hull is intended to serve many years of racing with the team. This consideration is particularly important, as the hull is expected to serve the team reliably over several years of competition.

The concept has been designated “Falcon,” an homage to the fastest native bird of prey and a symbol of the Belgium. The name is congruent with the existing fleet. The Torpedo model, distinguished by its displacement. The Shark model, on the other hand, exhibits higher speed with the sharp bow, aided by its fin-like appendages (next chapter). The Falcon model utilizes aerodynamic design advancements, ensuring minimal water contact during operation, racing across the surface with ease and precision.

To ensure a professional design approach, the aesthetic development process will adopt established practices from automotive concept design

studios. The initial phase involves the preparation of mood boards and style guides that compile reference imagery, material textures, geometry, and visual inspirations aligned with the project's design philosophy. These serve as qualitative tools to define tone, color language, and surface dynamics before committing to geometric development. Subsequently, concept hand sketches and digital renderings are produced to explore proportions, silhouette, and design flow, translating conceptual intent into tangible form proposals. Shadow, reflections and shapes are reviewed and inspired by Mazda designer Ikuro Maeda and concepts by Volkswagen and Bugatti.

Once preliminary forms are established, physical modeling becomes essential to evaluate real-world visual behavior under natural lighting. The utilization of small-scale clay models or rapid 3D printing prototypes facilitates the analysis of shadow lines, curvature transitions, and surface reflections. These elements are critical parameters that influence the perception of quality and speed. This phase offers insights that are frequently overlooked by digital environments, such as the subtle distortions caused by compound curvature or joint placement. Iterative clay re-surfacing, combined with basic color and reflection studies, enables qualitative assessment of how the vessel will appear under sunlight or in dynamic motion. Throughout this process, manufacturability and hydrodynamic performance remain the governing constraints and primary focus. Aesthetic adjustments must always comply with structural and fabrication limitations, especially with respect to composite techniques, aerodynamic smoothness, and weight distribution. Nevertheless, within these boundaries, careful shaping of the superstructure, canopy, and sponson transitions can yield a distinct and appealing identity without measurable performance penalties. This balance

mirrors practices in advanced yacht, super car and Formula 1 (livery) design, where visual appeal supports technical excellence rather than compromising it.

It is imperative to emphasize that the focal point of subsequent studies must be the elements of modularity, pilot safety, canopy, air ducts, cable routing, and other pertinent elements. It is recommended that the following subjects be investigated: the use of airplane latches, detachable canopy and the installation of a gas cartridge-powered safety cushion/balloon at the top of the craft in the event of an upside-down landing.

Ultimately, the new hull's design philosophy aspires to merge high performance with compelling aesthetics, establishing a recognizable and media-friendly signature for the UAntwerp Solar Boat Team. Beyond mere appearance, such a design becomes a communication tool — embodying the team's innovative mindset, attracting sponsors and students, and promoting the university's image in sustainable engineering and competitive solar/AI racing. All design, livery and concept drawing iterations can be found in the portfolio.

3.3. Hull Geometry Optimization & Performance Prediction

As previously stated, hull performance is a complex matter to model and accurately predict. In practice, only empirical approximations serve a useful contribution in both early and middle drafting stages, usually on hull type and features selections. However, when focusing on a niche hull type and proven geometry additions, and executing a high number of various full scale testing complemented by extensive experience, it is possible to accurately ($< 1\%$) predict performance similar to CFD outputs, with much less computational effort. By assessing the influence of hundreds of parameters on the behavior of a specific hull type, the prediction of porpoising, dynamic stability, power to speed mapping, etc. it is possible with remarkably low-computational power on a level that is useful in racing where every percent counts. This allows for high speed development with excellent reliability within the experimental frame of the niche hull type. In case of the UASBT, which currently lack performance and stability predictability, the approach has the potential to be the most useful ever exploited. As the preliminary literature study boiled down the most suitable hull types to tunnel hull designs, the Tunnel Boat Design Program (TBDP) by AeroMarine Research is undoubtedly the best way to tackle the high speed development concern considering the limited study time frame. It is a dedicated Velocity Prediction Program or VPP (e.g., NavCad by HydroComp Inc.) and much more. To achieve the best results possible, there is a col-

laboration with the developer, researcher, engineer, and Canadian & North American champion tunnel hull racer Jim Russell who was kind enough to sponsor the software for this analysis. Note that the parameter-input procedure and approximation technique is very similar to the approach used in BCD, except for the much more advanced level of calculation in AeroMarine Research's software. However, unlike the TBDP, BCD's lightweight processing allows for live updates and reviews.

This section will discuss the results of geometry, power and balance parameter optimization based on iterative (industry expert) review and improvement processes.

3.3.1. Geometry Optimization

The F1 tunnel hull is the most compact and high-performance hull shape for higher speeds and fast turns. It also has the lowest center of gravity and the cockpit is optimized for a single pilot. This means that the F1 tunnel hull (Seabold) will serve as the base design from which derivatives will follow. To estimate the influence of certain parameters on a standard F1 tunnel hull design, calculations are performed on various configurations with iteratively adjusted values. These configurations are then compared. This maximizes the TBDP's calculation advantage over slower engineering cycles using CFD.

General, Preliminary Insights

To gain insight in the performance of a tunnel hull design compared to the current designs it is vital to test some base configurations in the dedicated software. Remember that the goal is to reduce the mass below 200 kg in operational conditions (including pilot). Competing teams prove this is achievable with a adequate margin. The used power efficiency factor for the drivetrain is set to 0.85.

Run	Mass [kg]	Top Speed [km/h]	Motor Power [kW]
F1 Config 1	290	31.0	6
F1 Config 2	165	49.2	6
F1 Config 3	165	86.0	15

Table 3.1.: Preliminary tunnel hull insights

The first configuration mimics the weight range of the Shark and Torpedo. Considering the effective power output of 5.1 kW at 31 km/h derived from Table 3.1, the tunnel hull's default configuration outperforms the full-scale measurements of the Torpedo with 6.9 kW output at 31 km/h based on the corrected regressing in validation; uncorrected regression still indicates at least 5.9 kW output and simulation holds even higher values due to tunnel collapse. The Shark's simulations highlight 5.4 kW at this speed and measurements show 3.5 kW. Due to various reasons discussed in Section 5.4, the accuracy of the simulation is estimated to be better for output power prediction. As expected, the tunnel hull performs similarly to the Shark at this speed because it is a low-deadrise planing hull too. As is well known, the performance of these hulls depends heavily on their overall mass. Specifically, tunnel hulls lean toward airplanes in requirements at higher speed because they must be supported by an air cushion or ground effect to reduce drag. This is why there is an increased focus on reducing the weight of these specialized planing designs. Featuring a mass of maximum 165 kg (defined as the practical ultimate reduced mass) really shows its advantage as the F1 hull configuration gains speed up to 49.2 km/h using 5.1 kW output power. A new or combined battery system using the team's available packs could have a maximum peak power of approximately 15 kW lasting a few seconds (10 kW continuous). This leads to a top speed of 86 km/h, eligible for the high-speed classes of the MEBC.

The used base configuration exploited in subsequent analysis (within this section) has customized parameters to match the current ranges

of the UASBT fleet. The boat's overall length was set to 6.1 m, the overall hull mass is set to 165 kg to meet new requirements. The balancing is default.

Tunnel Height

The first parameter trend analysis implies the geometric tunnel height at transom (deck underside to keel at transom) with an analysis accuracy of 1%. Note the height must be as low as possible to benefit from ground effect, but should be height to address seakeeping issues and slamming loads. Based on CFD simulations in this study, a hull has sufficient seakeeping with a tunnel height of around 30-35 cm (fore) considering a low deadrise. A bow with a larger block coefficient (less piercing-type bow) and large protruding part may raise the bow earlier making lower tunnel heights acceptable. Ultimately, the aft tunnel height is set to 25 cm in this set of tests. The results can be found in Figure C.41 of Appendix C.2.

Selecting a lower deck does not significantly improve performance. The default F1 hull has a 15-centimeter tunnel height and performs best on flat water. Tunnel heights of 35 cm result in a 2 km/h decrease in speed compared to the default 15 cm tunnel height. This maintains a good balance of performance and seakeeping.

Asymmetrical Sponson Deadrise

Asymmetrical deadrise geometry is most common in racing hulls as they provide good sea-keeping combined with predictable cornering due to limited sliding effects and effective outward spray deflection (keeps the tunnel open and free of water). Variable deadrise (transom versus fore) may improve wave piercing at the cost of increasing efficient planing speed transition. As the hull is traveling rather slow compared to the F1 competition, early transition is desired.

Results of the analysis are given in Table C.14. Note the length to Deadrise Fore is set to 2.44 m measured at the transom. This ensures limited effect of the variation of deadrise on high-speed performance as the wetted area usually is behind this position. A variation may also reduce wetted area depending on spray formation, requiring careful analysis in the design stage, especially when combining step design. WAngle is fixed at 6°, identical to the previous configurations. Flattening the deadrise increases dynamic lift and efficiency, advancing transition while risking the effect of pounding and more vibrations. As inaccurate hull balancing may be common in future changes of the evolving hull hardware it is safe to consider a slightly larger deadrise than one found to be more optimal. For this purpose, configuration 14 is attractive, balancing both seakeeping, mass balance variation, early transition and adequate bow wave response (relatively aggressive at 30°). By enlarging the “Length to Deadrise Fore” up to 3 m, the efficient running area increases while keeping its wave deflection capabilities. The calculated top speed maxes out at 46.2 km/h while also using configuration 14.

The deadrise analysis shows that a moderately conservative asymmetrical deadrise provides the best balance between early planing transition, seakeeping, and handling robustness. Although flatter deadrise angles yield marginal efficiency gains, they increase sensitivity to mass balance variations and dynamic loading. Configuration 14 therefore represents the most suitable compromise, offering predictable behavior and sufficient performance while maintaining tolerance to future design and operational variations.

Tunnel, Sponson & Deck Width

Another important set of parameters to optimize include the tunnel, sponson (pad) and deck width. As the lift of the deck is higher for higher speeds (power increase), the influence of the deck compared to the sponsons also increases. At 6 kW the aerodynamic lift is around 13% of total lift and at 15 kW it is around 40%, depending on tunnel geometry. Note that adding steps proved to add no top speed advantage for the used configuration at 15 kW. Section 2.3.6 claimed that steps are most effective above 60 knots, aligning with the results found in this paragraph. Note that wider sponsons allow for more efficient planing at low speeds and higher tunnel widths for high speeds. There is always an optimum found for a certain speed. However, lifting strakes under the hull allow for clean flow separation making the sponson wetted width adaptive; fitting both low- and high-speed efficiency requirements. A sufficiently wide hull is required to allow for stable cornering.

In order to facilitate the transportation of the hull, it would be more expedient to employ the current trailer, with no substantial adaptations required. Consequently, the maximum width of the hull is restricted to 170 cm, corresponding to the current width of the Torpedo. In the event that widening the hull would prove to be a significantly beneficial measure — a scenario that is not currently under consideration based on the results in Table C.15 — the hull could be transported by tilting it onto an inclined support cart.

The tunnel, sponson, and deck width study demonstrates that overall hull width and lift distribution must be selected as a compromise between low-speed planing efficiency, high-speed aerodynamic lift and cornering stability. The selected configuration achieves an acceptable lift distribution across the targeted speed range without reliance on stepped sponson geometry, while remaining compatible with existing transport constraints. Deviations from the optimized values may be justified in the final design to accommodate structural, manufacturing, or integration requirements.

Extended Geometry Optimization

This thesis does not extensively discuss further geometric experimentation and optimization; however, the full analysis does take into account the extended fine-tuning of tunnel design and center pod geometry. The center pod effectively lowers the center of gravity (CG), thereby making the hull lighter while maintaining safe coring without compromising performance. The meticulous design of the center pod had two notable effects. First, it was possible to halt the efficiency loss. Second, it was possible to enhance performance and seakeeping. Note that the locally lowered tunnel roof is not defined as the center pod in the TBDP because it is not designed to provide hydrodynamic lift.

In summary, a well-designed center pod has been shown to offer the following advantages:

- It prevents nose diving.
- It has been demonstrated that, at cruising speed, the center pod does not come into contact with the water, thereby ensuring that the wetted area remains low and constant.
- A reduction in center of gravity (CG) is imperative for achieving stability and optimal cornering performance.
- The device has been demonstrated to lift the hull, thereby dampening impact and deflecting water in the event of a collision with a wave. This has been shown to protect the tunnel against slamming.
- The overall dimensions are reduced by the lowered cockpit, which sags between the sponsons.

The optimization process entailed a comprehensive consideration and design of spray rails (situated above and below the chine) and lift strakes (designed to ensure efficiency across diverse speed ranges) of both sponsons and the center pod. The final design takes into account:

- Safety
- Stability
- Seakeeping
- Efficiency
- Fool-proofing
- Hull slamming

- Greenwater & spray
- Performance predictability
- Reduced geometric complexity
- Variation in speed and power ranges
- Variation in overall mass

While a deviation of 165 kg from the standard mass is permissible based on the findings in the TBDP, performance is still strongly dependable. Given the significance of mass-influence in this regard, it is imperative that particular care be taken during the manufacturing process for all planing-type hulls. It is important that the CG is accurately set and that the motor is (actively) trimable to maximize the efficient and safe use of the design. The wetted area and, consequently, drag are contingent on this adjustment. Furthermore, correct trimming can expedite cornering and significantly improve safety (refer to Section 2.8.3 for further details). In order to optimize the straight-line efficiency of autonomous systems, it is recommended to adjust the inclination of the motor to regulate the actual trim of the hull (gyroscopic IMU measurement) to match the corresponding setpoint. The analysis of the TBDP provides pitch setpoints that the hull should match for every speed (e.g., optimized for max acceleration, see power reports). In flat water conditions, a correctly balanced hull will naturally change its angle of attack and the drive/torpedo must normalize itself to neutral: parallel to the water surface. The trim controller configuration constitutes an ancillary optimization objective, but running angles for different optimization objectives are provided in the hull's documentation. Note that the drive-prop may also generate lift depending on the configuration (propeller type and submersion level).

While a default F1-style hull is comparable to the Shark's monohull design at 30 km/h for the same weight, experts emphasize that the hull class demonstrates superior performance compared to monohulls at higher speeds in both efficiency and stability. Monohulls are more prone to exhibiting an exponential drag curve and porpoising requiring dampening techniques that reduce efficiency (high-speed range), while a tunnel hull demonstrates a different behavior due to its lift being generated aerodynamically, enforced by the ground effect. This advantage, combined with the low hull roll and pitch, which is crucial for antennas and perception, make them ideal for autonomous racing.

Theoretical maximum velocity of an optimized setup (165 kg) based on configurations in Table C.15 ultimately reached 92.9 km/h (effective speed of 90.0 km/h at $WAngle = 8^\circ$ is set and confirmed, calculated at 0.06% accuracy) using an output power of 12.8 kW (15% loss). The aerodynamic drag reaches up to 54% (28.6 kg) of total drag while being responsible for 30% (49 kg) of the total lift. It has been determined that in order to exceed a theoretical velocity of 100 km/h, a minimum of 17 kW of output power is required. It is important to note that this is a highly improbable scenario, as the mass will inevitably increase due to the augmented structural requirements. Hypothetically, it has been calculated that 123.7 km/h can be attained using only 25.5 kW of output power (30 kW of input, with an 85% power efficiency factor), considering the original weight of 165 kg. The aerodynamic drag contributes to a total drag of 74%, amounting to 55.6 kg, while aerodynamics account for 58% of the total lift, which is equivalent to 96 kg. Implementing a 6 kW motor ($\eta = 85\%$) results in a theoretical top speed of 48.2 km/h in which aerodynamic lift reaches 11% or 19 kg. However, it is wise to conduct a systematic analysis to ensure that certain transitions does not induce complications using a strongly underpowered setup (6 kW), such as a shift of lift effects (high-speed hump zone, next section). Further analysis will focus on stability optimization in discussion with Jim Russell. Note that input parameters change as the geometry is improved further. The final design is presented in the next sections and the reports can be found in Appendix C.9.

Recommendations in case of Infeasible Mass or Superstructure

The augmentation of cockpit space may be a viable option in the event that the existing optimized design is deemed impractical due to the constrained space behind the driver. It is important to note that the performance of the system is less influenced by the tunnel hull top deck than by the inner tunnel surfaces. The portfolio presents both CAD designs with and without a cockpit, thus allowing for the exploration of alternative designs. It is imperative to maintain the precise positioning of the CG at all times.

A rise in the vehicle's overall mass (e.g., 230 kilograms) was associated with a decline in performance, leading to instabilities such as porpoising. Porpoising was found to be more pronounced at higher speeds ($porpoise_{min}^{1.0} = 0.74$), and it was observed to extend further to medium and high speed regimes (33-60 km/h). Mitigation strategies are required. In order to ensure dynamic stability in rougher seas and at higher speeds, it is necessary to position the CG at a slightly aft position (0.6 m in front of the transom) in comparison to the standard 165 kg configuration, which has a CG location of 0.72 m. The increased weight of the hull also leads to a smaller operational domain. The lower limit of 55 km/h is increased to 60 km/h, while the upper limit is reduced from 90 km/h to 80 km/h. It is imperative to acknowledge that this final limit represents a power constraint rather than a stability constraint. To ensure an accurate analysis of a renewed setup, the manufacturable configuration must be entered into the TBDP and subsequently reviewed. The default design configuration is provided in the portfolio.

Stability Optimization and Final Design

A substantial body of information exists that describes the final geometry. However, due to time constraints, a comprehensive discussion on this topic is not feasible at this juncture. The final geometry was reviewed by James D. Russell upon completion of an extensive internal analysis during the design phase of this part of the project. Subsequently, brief overview is given.

The inner walls of the tunnel have undergone modifications (negative inner low freeboard at sponsons with reversed chine) to provide enhanced traction. The deadrise angle varies over the length with 20 degrees at 2 meters and 8 degrees at 0 meters (measured at the transom). This configuration is designed to optimize the equilibrium between efficiency and stability, achieving peak velocity while maintaining effective wave-cutting capabilities. The 38-centimeter pads are equipped with a negative chine for spray deflection and utilize different hard edges for effective flow separation. The configuration of the tunnel has been designed to measure 22 cm in height at the transom. It was lowered from the original design to enhance the ground effect and limit porpoising. The chord line of the airfoil features a static angle of attack of 1.5 degrees, the purpose of which is to promote the lift gained from the tunnel at lower speeds (<100 km/h). The center of gravity was reduced through the strategic local lowering of the tunnel ceiling, set at 0.7 m, thereby ensuring dynamic stability within the speed range of 55 to 100 km/h.

It is imperative to note that the design is not stable during airborne flight (blowover). To achieve this objective, it is required to shift the CG towards the front (center of lift by the wing). This technique is only advantageous when higher speeds are attained. This issue is discussed and found acceptable as it is very common and safe at the aimed operating range. In this configuration, the wing is responsible for carrying one-quarter of the total weight, while the remaining weight is supported by the hydrodynamic pressure generated by the sponsons. The running angle ($2.93\text{--}4.72^\circ$ WAngle) at the maximum velocity contributes to the overall aerodynamic performance of the craft. The combined running and static wing angle of approximately 6° will most likely stall the craft when going over the safe operating limits. This can be used as an inherent safety measure to keep the craft down. Active aerodynamic surfaces like canards or active hydrodynamic surfaces like trim tabs (strongly discouraged for efficiency, encouraged to reduce porpoising) may be an alternative, more complex safety feature. Trimming the motor may also be a suitable alternative, while it is expected that the mechanism will be too slow to react. While

the aerodynamic wing's lift is 1/3th of the total supporting forces, the wing's drag is responsible for approximately 41% (21.4 kg) of total drag at the top speed of 90.2 km/h, while the cockpit accounts for approximately 8% (3.99 kg), the sponson drag is 41% (21.2 kg) and the lower drive unit causes 10% (5.41 kg) of total drag by extending a torpedo of 5 cm diameter 14 cm (center) down.

The total mass is configured to be 165 kg (see decomposition in TBDP Design Input document). The plausibility of this claim is substantiated by the findings of other race teams that have built larger hulls, thereby attaining the specified total mass, inclusive of the driver. However, during the manufacturing and the structural design stage, it might be that loads demand a heavier construction. Therefore it is important to note that while increased total mass affects overall performance, as long as the CG is positioned correctly, this hull design can be effective. It is recommended not to exceed roughly 220 kg total mass (including driver). The final design measures 170 centimeters in width and 4.2 meters in length. 15 kW motor power (12.8 kW propulsive power) is a requirement for normal operation.

It has been determined that true planning is reached at 32 km/h (planing velocity criteria). Furthermore, given the stability regime that has been previously mentioned, the optimal operating range is between 55 and 90 km/h (more power may safely extend up to 100 km/h). The construction of this hull is rendered futile if one intends to operate at speeds below 55 kilometers per hour.

To ensure the integrity of the data, random checks are performed to compare the results of CFD analyses with the TBDP reports. For example, the findings indicate that aerodynamic lift is accurately estimated (a 2% reduction in CFD), while aerodynamic drag is approximately 34% lower according to CFD (at a maximum speed of 90.2 km/h). This discrepancy may be attributable to the less precise configuration of the CFD simulation (boundary conditions, waterplane effects, surface conditions, etc.) and may also be caused by the superior performing superstructure not accounted for by the TBDP.

The high-speed “hump zone,” as identified in various plots of the TBDP, occurs at approximately 43.5 km/h and signifies the range of velocities associated with the observation of intrinsic “instability” resulting from a substantial transfer of the lift mechanism. It can be observed in various performance plots across its velocity range. Every performance boat has this “hump” or “transition zone,” which differs depending on the hull and configuration. The enhanced functionality of the lift mechanism is accountable for the observed alteration in lift experienced by the operator while maneuvering the vessel. In the context of tunnel hulls, the “hump zone” is defined as the transition velocity at which aerodynamic lift forces from the deck or tunnel begin to exert a substantial influence on the hull’s performance. Concurrently, a reduction in wetted surface area enhances the overall efficiency.

The size of a vertical stabilizer can be calculated according to Appendix C.1.49. Note that an aft placed cockpit may function as stabilizer geometry without the need of adding more aerodynamic friction area by using a separate tail sec-

tion. Fully submerged propellers may be less effective for high speed applications, yet the use requires a small rudder section for stable operation. A surface piercing prop like a 3-blade (recommended) or 4-blade cleaver is conventional for this type of racing craft, but requires a skeg for stability. It is important to acknowledge that both the rudder, as incorporated in the final analysis, and the skeg, may have a substantial positive impact on stability. It should be noted that submerged appendages can contribute significantly to drag.

It is evident that the design is suboptimal yet usable without a canopy, similar to most Sport C class craft, which operate in comparable speed ranges. F1 uses a canopy to protect the driver from high water impact in the event of a control failure. Other design features like a sensor wing and cable routing, internal structure and stress analysis should be part of a follow-up study. Extensive performance reports can be found in Appendix C.9. A scaled drawing of the new hull can be found in Appendix C.9.21.

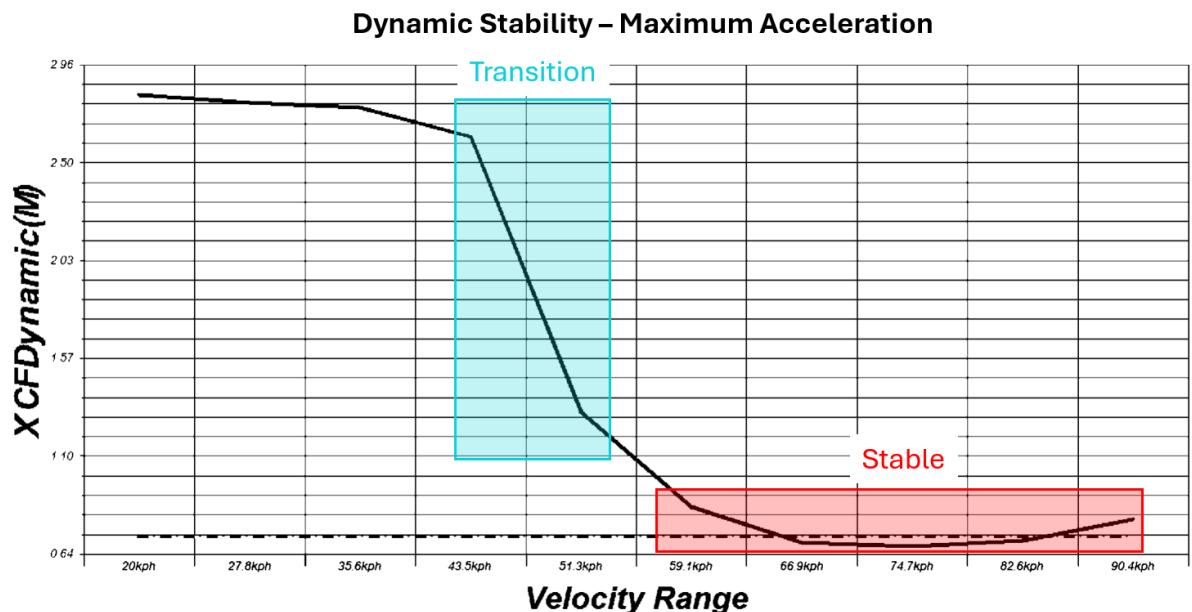


Figure 3.1.: Dynamic stability of for a maximum acceleration calculation indicating the transition of lift and the deviation between the static CG and the dynamic center of forces (XCFDdynamic)

Cornering

Thanks to the F1-style configuration, an exceptionally high angular velocity or yaw rate is made possible. The craft is expected to achieve tight inshore/hairpin cornering performance of 60–90 km/h, with an effective turning radius of 12–25 m, corresponding to 4–6 g lateral acceleration and 70–120 °/s yaw rate. These values are mutually consistent through $a = \frac{v^2}{r}$ and $\omega = \frac{v}{r}$, and represent a realistic upper-working regime for aggressive inshore maneuvering. Structural integrity is expected to be the limiting factor. The primary critical load case is anticipated at the sponson-to-centerbody interface, where sponsons will tend to peel/tear away from the wing body under combined loads. Local reinforcements and robust load-path design at the sponson roots are therefore required, and should be verified by targeted stress analysis (including increased safety factors due to wave impacts and the potential of unsteady ventilation). The ratio of exiting versus entering velocity is estimated to be around 65–75 %.

“Safe” cornering characteristics in case the inner sponsons loose grip (worst case) are calculated (see Figure 3.2) and appended to the portfolio. It is imperative to acknowledge that these velocities are considerably lower or the turns are significantly wider in comparison to their potential under normal conditions (flat water, inner sponson planted down). In discussion with Jim Russell, it was stated that the hull configuration will act very similar to real F1-type configurations in both handling, performance (for the aimed velocity range) and cornering. While not everyone in the racing world agrees that a negatively inclined inner freeboard (-4°) will provide significant extra grip, it is very common to have this feature in F1. Since it has no pronounced negative effect on performance and it is believed to

improve cornering in rougher sea conditions, it is implemented in this design. Note that the 8° shallow deadrise is configured to dampen porpoising while increasing aft placed lift and overall efficiency. The sponson lift forces must be accurately balanced with the aerodynamically generated lift coming from the ground effect tunnel and wing, which is represented by a forward placed vector. The overhanging wing section behind the transom increases the aerodynamic versus hydrodynamic lift ratio and places the wing lift vector aft to provide better balance in Del-Press plots of the TBDP.

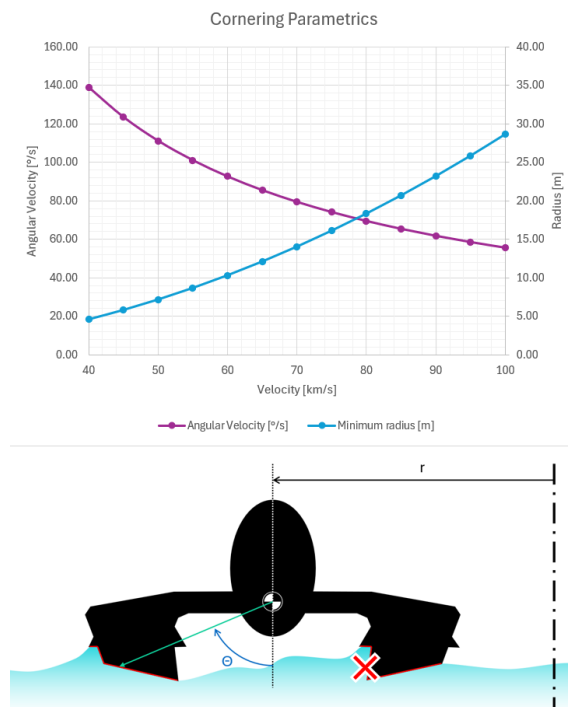


Figure 3.2.: Approximated maximum cornering performance in case of inner sponson grip failure (rough sea conditions)

4. Simulation and Computational Analysis

With nearly 1 TB of simulation results generated during this study, the hulls are subjected to a wide spectrum of conditions and evaluated within the different aspects of hull design. However, due to time limits, scope of the study, license and computational constraints, not all cases of interest could be evaluated as thoroughly. Though, it has been a highly valuable study which has proved valid to claim all kinds of performance statements. Industry standards, review and simulations highlight concerns, limitations, improvements and the great potential for both current hulls and the new hull. Note that it was impractical and noncritical to discuss both hulls as thoroughly, hence the Shark was put forward as the reference model. This hull proves to have the most potential during the evaluation of the existing fleet.

4.1. Hydrodynamic Simulation

The hydrodynamic analysis consists of continuous curve speed-resistance, fixed speed (looking for steady-state drag), stability, and maneuvering analysis. All are discussed below. Note that more simulations were conducted to thoroughly understand the behavior, simulation environment, etc. Only the most important results made it into the thesis.

4.1.1. Speed-Resistance Curves

In this section, the full-range speed-resistance curves are briefly discussed. This type of simulations aims to produce continuous drag curves as an output by defining time dependent inflow boundary conditions. In most simulations of this type, the duration is set to 120 s and the speed is raised from 0 to 10 m/s. Over 150 parameters (e.g., drag decomposed forces, wetted area, vector component resistance, motion metrics, etc.) are monitored during the simulation run for every time step. Spatial data is recorded every 100 ms. Configuration parameters for critical simulation runs are noted in a separate Excel file added to the portfolio documentation.

Drag Decomposition

Simulations illustrated the drag decomposition components defined in Section 2.1.9. Figure 4.1 shows the Shark's total drag decomposed in a shear force (viscosity, friction) and pressure force (wavemaking) plot for both zero surface roughness (red) and the true 0.0001 m roughness (blue). Note the influence of the roughness on the shear forces.

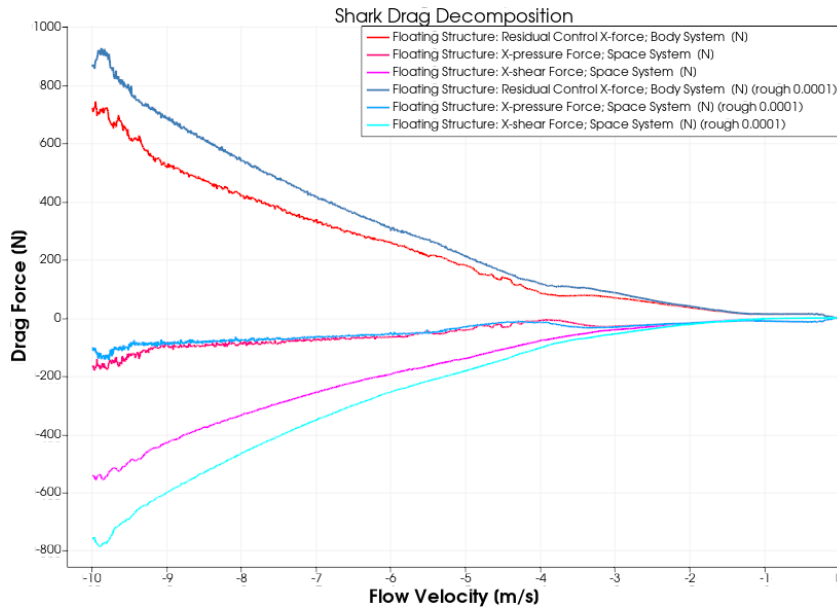


Figure 4.1.: Shark drag decomposition versus flow velocity

The decomposition comparison for both hulls is shown in Figure 4.2 based on the calibrated surface roughness configured simulations. Note the Torpedo's jump at 7.5 m/s caused by an unstable trim-sinkage behavior at higher speeds (see Section 4.1.2). At 5 m/s, a drop can be observed. This is a startup effect highlighted in the same Section 4.1.2. The Torpedo's plot is subordinate 5 m/s (separate calculation) was run in reverse, lowering the flow velocity from 5 to 0 m/s. Later, this partial plot was replaced to an improved simulation setup before validation in Section 5.4 as it proved to be invalid due to inaccurate settings. This is clearly visible in Figure 4.2 by observing the unconventional slope.

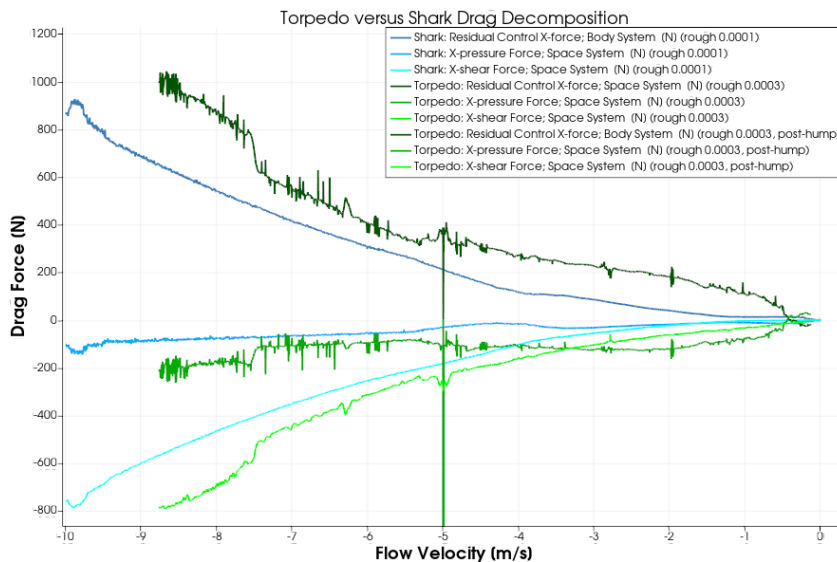


Figure 4.2.: Shark versus torpedo drag Decomposition

Surface Roughness

The difference in configured surface roughness drastically affects results. Initially, tests were conducted with a surface roughness set to 0.003 m. The convergence study in Section 4.1.7 used those results as input values. However, subsequent investigation indicated no significant influence on convergence or change in calculated wetted area based on the roughness settings as illustrated for a selection of tests in Figure 4.3. Later, the roughness was calibrated to 0.0001 m for the Shark and 0.0003 m for the Torpedo to match the experiments at full-scale discussed in Section 5.4 on validation. Note that the higher roughness of the Torpedo is caused by fine sanding of the hull. Repainting is strongly advised in order to reduce drag. It doesn't have an effect on behavior or wetted area, supported by simulation.

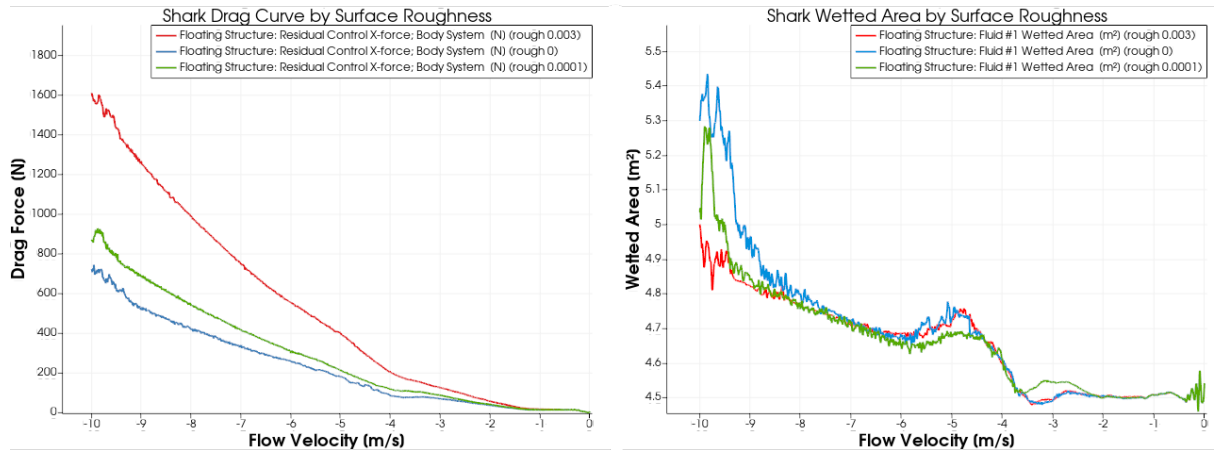


Figure 4.3.: Shark roughness influence

Simulations indicate that repainting the Torpedo hull with a smooth glossy coating may be of interest to reduce the viscous drag. The Shark already features a glossy surface, promoting resistance decrease. Reduction techniques discussed in Section 2.3.3 should be considered with their fundamental theory in mind. Correctly dimensioned and oriented riblet type structures are key for advancing the hull's performance.

4.1.2. In-Depth Examination

The ensuing findings are predominantly derived from steady-state simulations conducted at specific constant velocities, namely 0 m/s, 5 m/s, and 10 m/s. These simulations are maintained for a duration of five seconds to ensure the establishment of a stable drag and wetted area measurement. The conclusions are also supported by the other, continuous simulation results.

Transom Flow

Clean flow separation for both Shark and Torpedo, as explained in Section C.1.17, are shown in the results. The flow and turbulence analysis in Flow3D at the stern proves effective use of the transom design. However, the Torpedo shows issues at higher regimes due to high trim and stern sinkage.

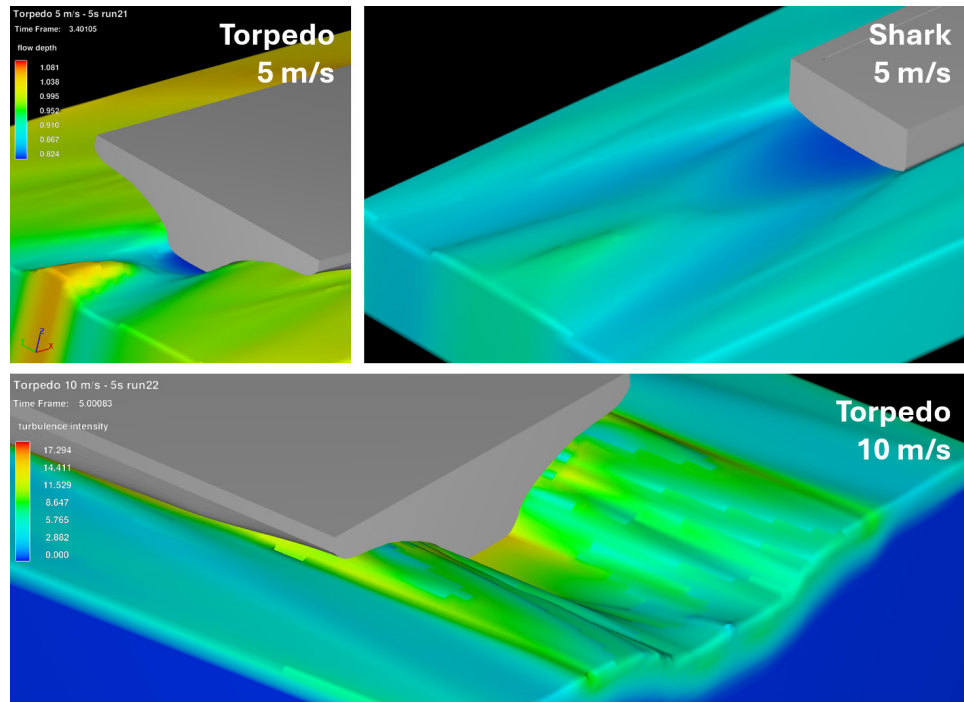


Figure 4.4.: Transom flow Torpedo (left) and Shark (right) at 5 m/s and Torpedo turbulence at 10 m/s (bottom)

Wetted Area

The Shark shows a lower wetted area than Torpedo in all speed regimes (grossly 20%). Contradictory, planing hulls usually have higher wetted area due to dynamic lift promotion compared to (slender) round bilge hulls whom only suffer more drag from wetted area. Comparative numbers and detailed analysis are provided in subsequent sections.

Trim

Since the trim of the existing hulls is fixed, it is essential to examine how the existing fleet naturally trims. It is learned from experience that the Shark tends to nose dive when approaching planing speeds (see Section 5.1.5). This is also reflected in the full-range towing test (see Figure 4.5, Section 4.1.1), and steady-state (fixed flow velocity) simulations. Once the Shark is reaching higher planing speeds approaching 10 m/s, the negative trim starts to decrease as the midship-bow zone experiences minor planing behavior. The wetted area increases up to the flared bow below the gunwale. This enforces wavemaking: raising the resistance, reaching approximately 910 N and wetted area peaking at around 5 m^2 in the 10 m/s steady-state simulation.

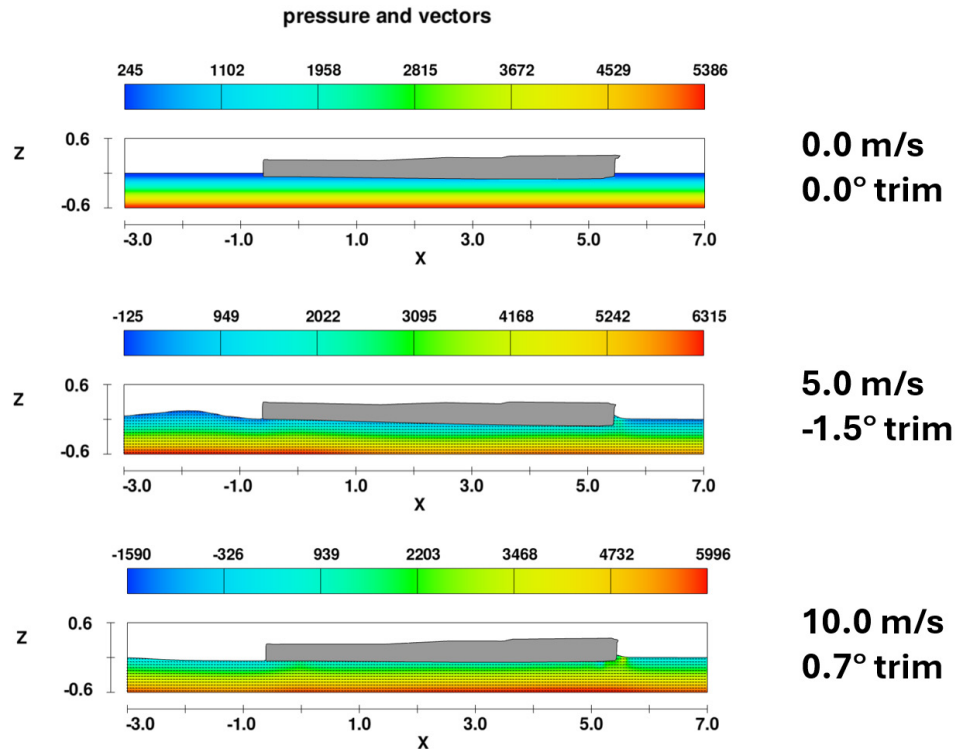


Figure 4.5.: Shark trim at different speeds 5 m/s

The diving effect should have been mitigated by a more aft placed CG as explained in Section 2.1.8 according to the enlarged hull concept adopted by most axe-bow type vessels. What causes the bow to sink? There are two driving forces enforcing this phenomenon:

- Once the stern lifts due to planing, the force balance requires lift from the bow, which is compensated by increased displacement due to the absence of dynamic lift in the vertical piercing bow.
- The lowered keel at the bow generates a lowered pressure zone under the hull's bow and midship effectively keeping the hull down in the water rather than planing over the surface.

Based on the simulation results, it can be concluded that the Shark's hull was designed to operate in the 18-32 km/h or Froude 0.64-1.14 range. With beyond growing speed, the wavemaking starts fluctuating and severely increasing drag and wetted area due to the water deflection of the bow overhang plane.

On the other hand, the Torpedo dynamically rises its bow when increasing speed due to the more conventional round bilge and flared hull shape. The bow generates more dynamic lift compared to the stern as the hull is slightly pitched down at 0 m/s for the default CG-conditions derived from previous studies (UASBT documentation). Note that the Torpedo features a neutral pitch in the range of 4.5-7.5 m/s. Above, the tunnel may close and the Torpedo heavily pitches up and sinks its stern as described in subsequent Section 4.1.2.

Observed Wave & Behavioral Effects

In this part of the simulation results discussion, the hull specific effects are described at different speeds (0, 5, and 10 m/s). The analysis is based on continuous full-range towing and steady-state simulations.

Shark Stern sliding of the Shark, showed in full-scale experiments (see Section 5.1), Shark review (see Section 2.2.2, and model experiments in Figure 5.11) is now also observed in simulation. The steady state 5 m/s simulation with a perfectly aligned hull, default balance and a single constraint for X-translation produced the results given in Figure 4.6. The figure illustrates the pressure distribution plane 75 mm (cross-hull) and 125 mm (underneath the hull) below the initial environment waterplane at approximately 3.2s simulation time. The pressure distribution is as predicted in the *Literature Review* (2.2.2), indicating the driving force behind the effect.

The graph of both wetted area and x-force (drag inverted) highlights four different zones of behavior.

1. In the first zone, the hull settles from the initially set position to the actual steady-state position mainly by correcting trim and sinkage.
2. After 0.7 s, the hull is in the right condition for drag or X-force derivation suggesting a magnitude of approximately 407 N and wetted area reading of 4.65 m^2 . Note that the surface roughness of 0.003 m led to the steady-state drag of 407 N illustrated in the Figure 4.6. However, in calibrated simulation, an actual drag force of 214 N was found at 5 m/s, validated with experiments.
3. In the third phase of the simulation, starting from 2.5 s, force data noise and wetted area increase appears due to the initialization of the course deviation.
4. The last zone is part of a failed result. As the sudden wetted area reduction and 3D-data show, the hull model is sailing of the computational domain. Obviously, the results are invalid.

Subsequent Shark towing simulations will apply more constraints (e.g., fixed yaw and Y-translation), making no compromises on accuracy, but ensuring correct flow alignment.

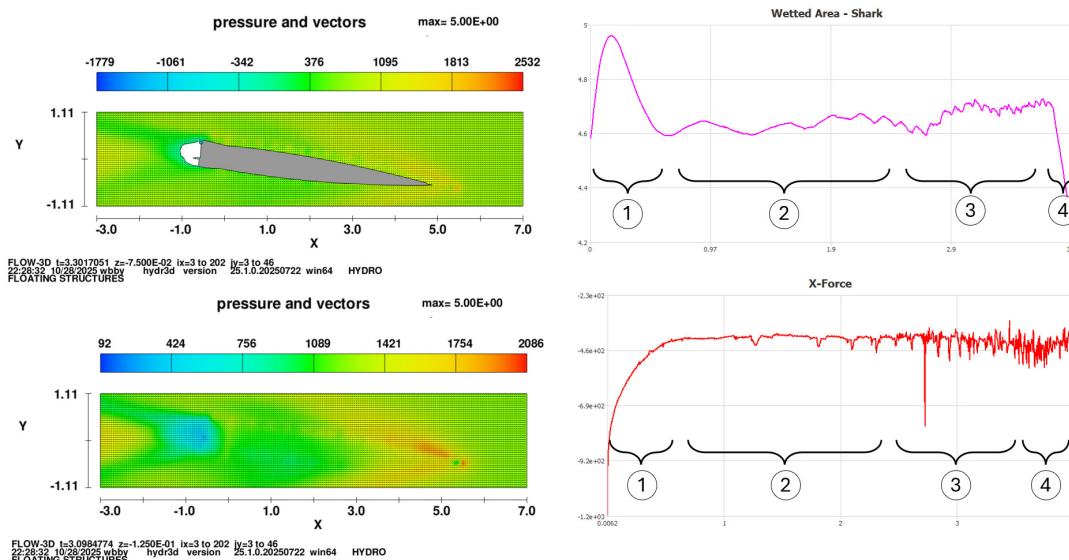


Figure 4.6.: Shark yaw instability at 5 m/s (high roughness (0.003 m) simulation)

Torpedo The Torpedo hull at 5 m/s (current mean operation speed) suffers from sponson wave interference, a red flag in trimaran design as stated by Datawave Marine Solutions [73]. This was discovered during the simulation analysis depicted in Figure 4.7. In the design of trimaran sponsons, a naval engineer should aim for optimal placement of amas or sponsons so that the main hull does not generate a bow wave with negative drag influence. In the simulations, it becomes clear that the second (outer) bow wave has its crest aligned with the bow of the sponsons, locally increasing the submersion, wavemaking and wetted area. Note that the angle of the bow wave changes for $Fn > 0.5$ (see the end of Section 2.1.2). Despite the speed dependence of the bow wave angle, the interference is present in the operating range of 0 all the way up to 8 m/s beyond its sponson influence diminishes.

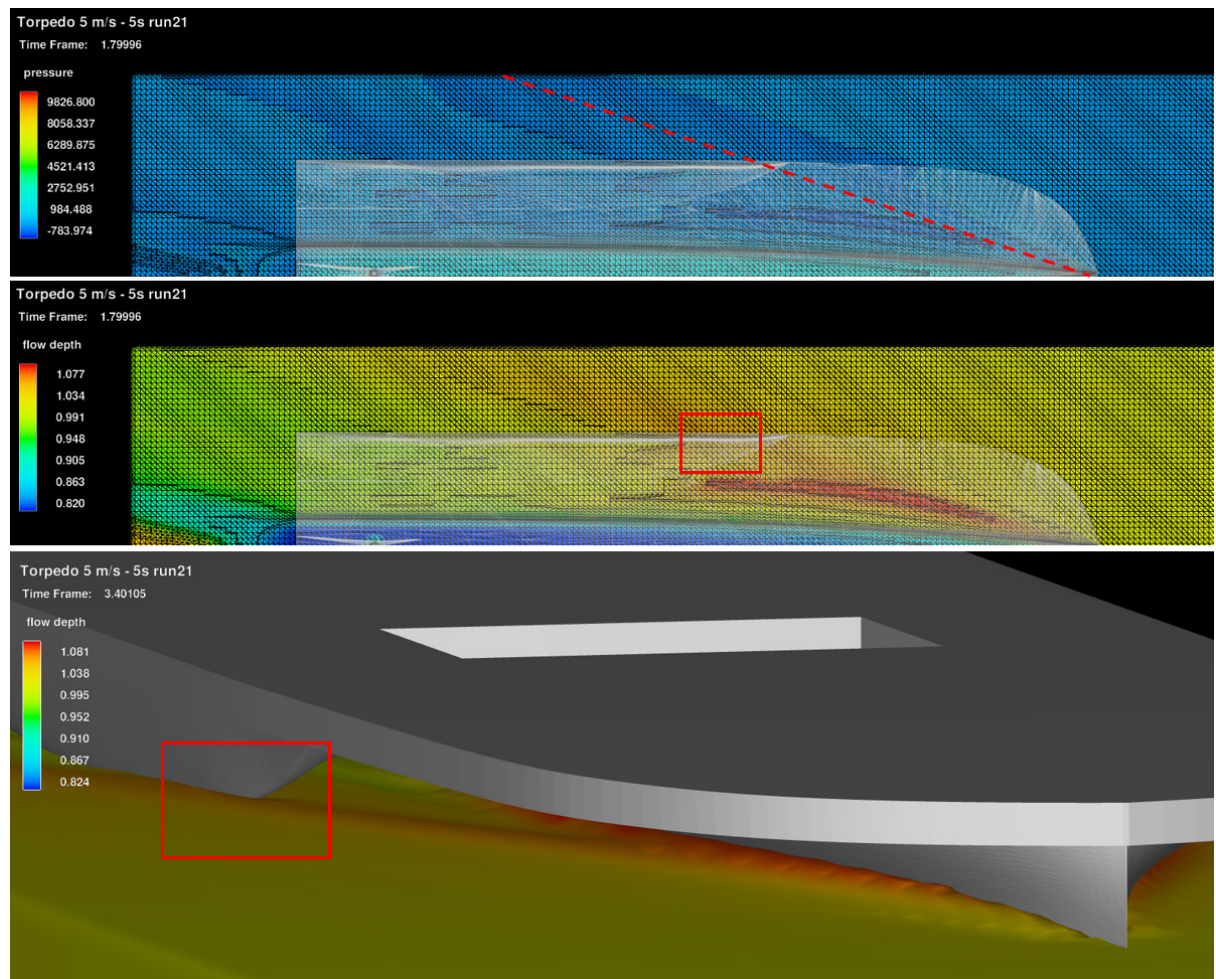


Figure 4.7.: Torpedo wave interference 5 m/s

At 10 m/s, the drag (~ 1700 N, 0.003 m surface roughness) and wetted area ($\sim 6.38 \text{ m}^2$) of the Torpedo drastically increases due to sinkage of the stern during planing, closing the steeply flared tunnel of the trimaran configuration entirely and starting to plow incoming water (see Section 2.2.1 and Figure 4.10). This is not only a hydrodynamic concern, it is also a loss of aerodynamic advantage. While increasing speed, the tunnel closes, slowing the hull in parachute type aerodynamics. Once reaching approximately 8 m/s, the tunnel may close and the pressure rises to a maximum, enabling unstable flow effects similar to the effect illustrated in Figure C.26 of Appendix C.1.32. It is noteworthy that the hull does not demonstrate uncontrollable porpoising, a phenomenon that would typically occur at higher speeds and with a less substantial mass. However, the hull experiences elevated levels of drag. In calculations, one may use the frontal area including tunnel for aerodynamic drag calculation for this

case. However, sinkage and trim is not the only and certainly not the main cause for the tunnel to close. Full-range simulation highlights that the decreasing bow wave angle is responsible for the increasing bow wave magnitude, eventually heading for the center of the tunnel (see Figure 4.8). That implies that the main bow wave is initially hitting the inner freeboard of sponsons and effectively filling the tunnel with turbulent water flow. Locally, in the tunnel, the water level rises, eventually closing the tunnel at ~ 2.8 m in front of the transom.

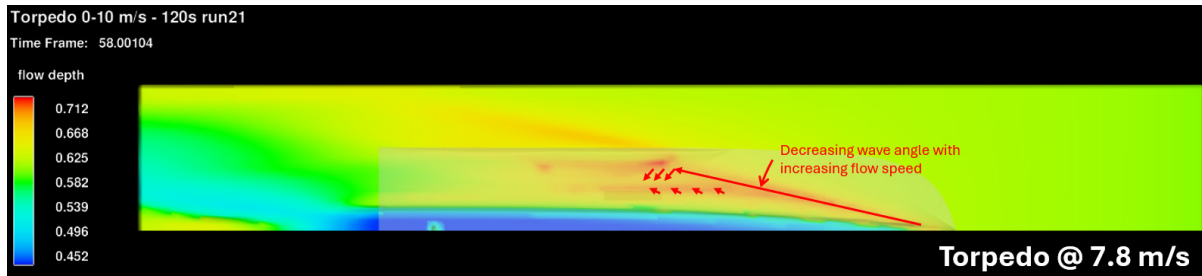


Figure 4.8.: Torpedo's before tunnel closure at 7.8 m/s

Once the tunnel fully closes, the hull's sinkage and pitch increases even more as illustrated in Figure 4.10 and the drag increases by over 23%. This unintended behavior underlines the hull is unsuitability for high-speed (> 9 m/s) planing operation without hydrofoils or fundamental adaptations (e.g., vertical main hull freeboard) with flat water conditions. Rough water lowers the speed threshold for tunnel closure. Note that the tunnel closure is an instability. The phenomenon under discussion is not defined by a particular velocity. However, the probability of closure increases strongly with speeds exceeding 9 m/s. For example, in flat water conditions, rising the speed very gently (0-10 m/s in 120 s) up to 10 m/s, the effect does not present itself. Nevertheless, the slightest unsteadiness in waves or hull movement triggers the negative response as it showed in the steady-state 10 m/s simulation and continuous flow slope simulation at 7.5 m/s. Figure 4.2 indicates that both pressure (+60%) and shear (+40%) forces suffer from the effect adding about 25-30% increased drag and difficulties in maneuvering. It was not simulated, although it is expected that the recovery after the tunnel closure by decreasing speed will exhibit speed-resistance hysteresis when reducing speed.

Analysis of the drag curves indicate that the trim and exact sinkage has to set itself at the start of every simulation, usually within one to five seconds of physical runtime, depending on the initial speed and initial pose matching. This causes temporal spikes or drops in the plot as observed at 5 m/s in Figure 4.2 and zone one of Figure 4.6 for the Shark. In Figure 4.9, the gently risen flow speed of the continuous towing test at 10 m/s cross-section turbulence is illustrated, indicating the small air gaps left in the tunnels right before collapse.

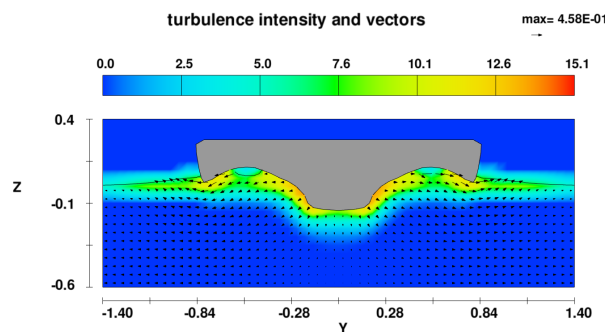


Figure 4.9.: Torpedo turbulence and tunnel air gap cross-section at 10 m/s, gently increased over its testing range of 3 up to 10 m/s

Solutions for mitigation of sponson interference — and thus low-tunnel closure — include

1. widening the sponson placement and cross-deck span;
2. reducing the sponson length;
3. more aft placement of sponsons;
4. forward sponson placement as adopted in 3-point hydroplane design, assuming that the wave-making of the sponsons does not cause significant interference with the main hull.

The proposed solutions can be categorized:

- Stay behind the main bow wave in the intended speed range (2, 3)
- Stay in front of the main bow wave ($\lesssim 19.47^\circ$, see Section 2.1.2), this is most common in trimaran design as it offers the more stability and clean inflow (Earthrace, military vehicles, etc.) (1, 2, 4)

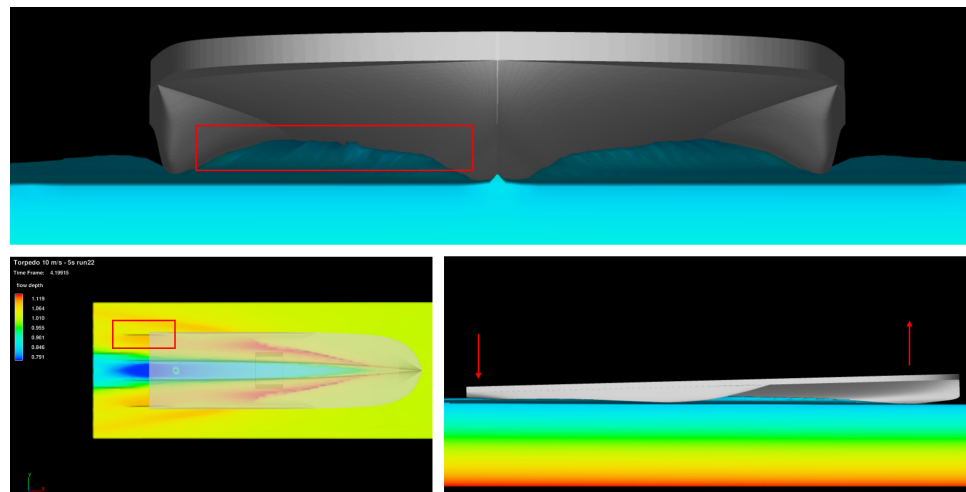


Figure 4.10.: Torpedo's closed tunnel response at 10 m/s

Direct solutions for delay of the tunnel closure effect include

- lifting the hull significantly by fins or foils;
- making the hull significantly lighter;
- shifting the balance by adaptive thrust vector or pad trim.

4.1.3. Waves & Stability Analysis

Research delineated the worst case scenario applicable to the MEBC competition. At high hull-to-wave speed ratios (> 4 , estimated for $wavelength = LWL$), the head-on wave impact is most threatening for seakeeping in hull structures. Resonant oscillations of the various degrees of freedom of a free body have not been taken into account because this would require a much more extensive analysis. Furthermore, it is impossible to study all waveforms and dynamic patterns (see Section 2.1.6), some of which only become apparent after several cycles. Moving at different speeds and making turns further complicates the analysis. In practice, model ships are exposed to varied conditions and patterns for very long periods of time in a 2D wave tank.

Consequently, this study simulates the impact waves on the bow at medium (5 m/s) and high (10 m/s) speeds. A wave pattern outside the domain with a crest-to-trough distance of 0.6 m and a wavelength of 6 m is fed into the domain, which contains a partially constrained body with a fixed sailing speed. The wave outflow of the domain features a wave absorbing layer of 1 m and sides are set to continuative. In general, a hull-to-wave length ratio of one is considered most dangerous for impact loading, hence the configuration of 6 m wave length. Note that slow wavelength experiments in simulation demonstrate that a ratio of 2/3 induces nosediving, notwithstanding that slamming stays focus in this specific study. The wave is approximated by the Stokes and Cnoidal wave algorithm. The average water depth is approximated at 20 m, based on the close-to-port conditions in Port Hercules. The pitch, acceleration, local pressure (average bow peak pressure), rigid body forces and wetted area are monitored over a period of 6 seconds, containing at least 5 wave cycles.

Shark in Waves

The following explanation is based on the results of simulation run 40 (5 m/s) and 41 (10m/s). Run 40 shows an average peak pressure X-force increase of 1.8 kN and a local bow surface pressure of 16.1 kPa and an average of 7-8 kPa over the 6 cycles captured. The slamming pressure was found over a length of 2 m, traveling from the bow to the midship, where the peak diminishes. The greenwater (50% cell isosurface, 100% fluid fraction) reaches over 5% of deck wetting by the scooped wave crest, traveling down the deck. Analysis shows that these maximum wave conditions may endanger seakeeping. With the current cockpit, the Shark drives at its maximum for deck wetting. The temporary pressure and force peaks are still acceptable, yet significant. For the analysis of spray (5% cell isosurface, 10% fluid fraction), it was found that over 70% to 90% of the deck was covered in droplets and fine aerosol. A screenshot of the results is added to the *Appendix* in Figure C.45.

The instantaneous pressure forces in the x-direction reach up to 2.938 kN when passing over a wave in run 41. The increased speed leads to a local impact pressure of 45.02 kPa, which is considerably higher compared to run 40, as predicted in Figure 2.7. The results of wave test 2 (run 41) are presented in Figure C.46. The greenwater is increased to 8% and spray to 90%, showing slightly denser fluid-air mixture.

Note

During the stabilization of the pitch angle in flat water planing conditions at 10 m/s, the Shark shows effective deflection of the upwards accelerated bow wave, generating minor dynamic lift on the overhanging freeboard. However, the observed behavior also indicates that design of the vertical bow below the overhanging freeboard generates no dynamic lift at all. Therefore, at high flow velocities, the planing character of the bow is completely nonexistent. This typical property of a sharp vertical bow is aiming for a smooth ride, but lack safe seakeeping as the bow pierces through high waves. Wave simulations at 10 m/s acknowledge this effect by demonstrating a pitch variation of only 5°. A common technique suggests to flare the bow's flanks and add rake. This provides both dynamic lift and locally increases buoyancy (see Appendix C.1.16). This keeps the deck (and sensors) dry at the cost of increased pitch deviation.

Torpedo in Waves

The bow of the Torpedo Mk. II EVO is remarkably lower and has a semicircular frontal gunwale edge (top view) perpendicular to the direction of travel. Additionally, it holds a large overhanging deck, worsening seakeeping capabilities altogether. As indicated by various stakeholders, this bow rightly raises concern of both pilot danger and structural threats in the open sea (see Figure 2.4). If the Torpedo is exposed to the same maximum conditions as the Shark for competition at sea, several indicators that point to the risks of these waves. In run 44, the Torpedo is traveling at 5 m/s, clearly demonstrating destructive bow pressures of > 15 kPa over a centrally focused area of 4 dm^2 . Around 9.5 kPa was found over the entire width of the frontal gunwale. Once the wave propagates over the body, large pressure zones of > 14 kPa form both on deck and at the wave front underneath the hull. There is no point in discussing greenwater if the entire deck is flooded and the local pressure is likely to cause immediate damage. The pressure distribution and front propagation is displayed in Figure 4.11. Note the decreased number of test cycles in Figure C.49. The simulation became unstable as the pressure peaks skyrocketed on impact at the second wave encounter. Also, pay attention to the sharp spike at the highest value of angular velocity (1.13 s). This indicates severe angular acceleration and hard slamming of the hull.

A final simulation of the Torpedo was performed at 10 m/s. This presents less dramatic motion because the Torpedo can ride the waves. Nevertheless, it is unrealistic to achieve this speed given the effects at lower speeds. In addition, the pressure at the bottom of the hull is not evenly distributed and still very high (10-12 kPa) along with strong localized peaks up to 30 kPa.

To summarize, it is not advisable to operate the Torpedo Mk.II EVO in medium sea state conditions in which the trough-to-crest height difference exceeds 0.6 m. Lower thresholds may apply, with estimated values ranging up to 0.4 m.

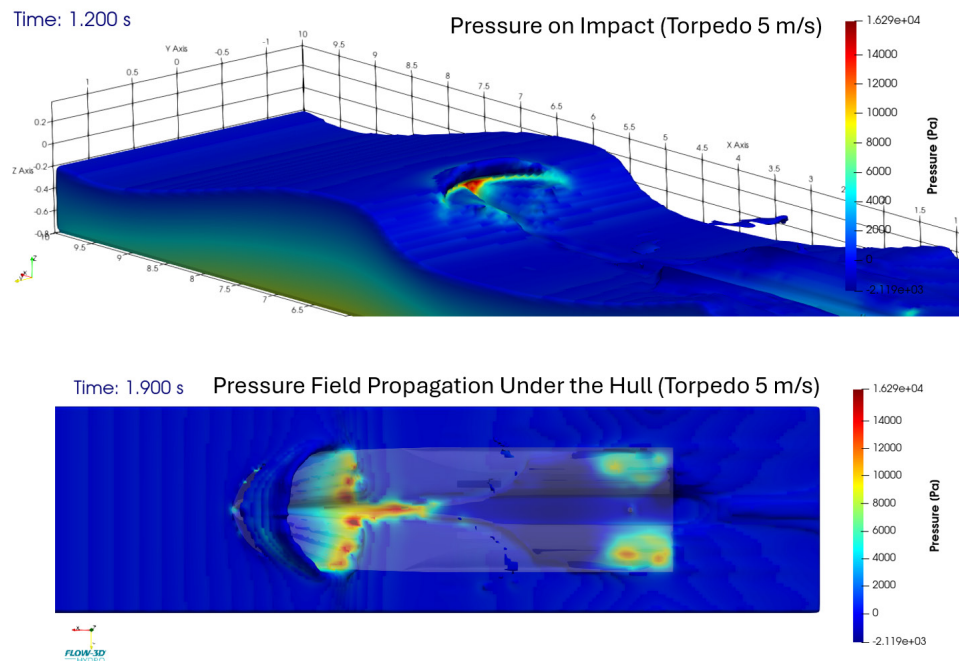


Figure 4.11.: Torpedo's wave impact and pressure distribution at 5 m/s

4.1.4. CG Positioning and Balance

For all simulations, the force balance, CG position, and mass are configured based on the Teams shared Excel file “Torpedo VS Shark.” The calculation of inertia is based on a homogeneous volume density distribution, which closely aligns with the actual values measured in previous studies conducted by the team. This is mainly important for the wave response evaluation and wave resonance estimation. However, CG positioning and total mass is most important for normal operation during investigation of high-speed performance.

For the Shark, experiments have shown that slight aft placement of the default center of gravity (CG) is not beneficial. The default CG is 2.75 m measured from the transom. This distance was reduced by 0.13 m in the simulation. Only a slight reduction in drag is observed in Figure 4.12 in the hump region between 4.1 and 5.9 m/s showing an average of approximately -4%. Above 5.9 m/s, the performance gain diminishes. This is caused by the dropping bow geometry. The theory behind axe-bows (see Section 2.1.8) states that a CG place close to the stern would should prove to be beneficial. However, in discussion with the team, it was found this is an impossible case eventually rejecting any further analysis because implementation (new cockpit, repositioned battery and electronics) would be impracticable. Adding lifting bow fins is more convenient and has more potential to reduce drag, similar to the assisted hydrofoil principle — stability is controlled by the nature of the hull, not foils. Placing extra mass (0-60 kg) near the transom is only worth experimenting with if the stern extender is removed and bow foils are practically infeasible. The bow should be lifted free from the water at cruising speed.

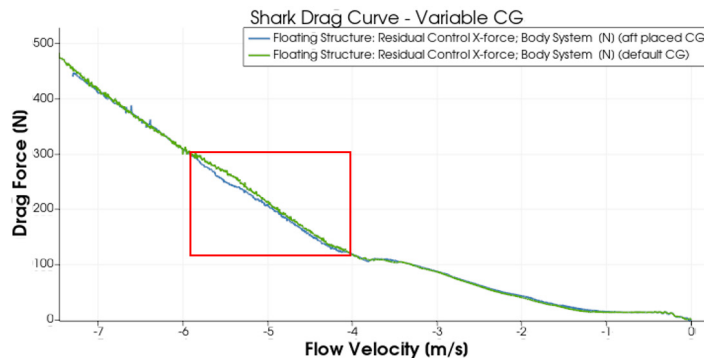


Figure 4.12.: Shark changed CG placement

In the next analysis, the default Shark is compared with the same model with an added stern ballast of 400 N, a stern ballast of 400 N combined with the removal of the stern extender and the removal of the stern extender combined with bow foils (v3 configuration, high non-optimized pitch, see discussion Section 4.1.6). Note that adding weight to the equation is a brute force method to influence the trim and thus transition to planing.

- **Stern ballast** — Adding significant weight of 400 N at the transom proves to have negative effect in the low-speed region up to 4 m/s (near theoretical hump). This is expected as BCD suggests increased wavemaking in this regime. It shows like a (proportional) offset in drag curves. Increasing flow velocity up to 9 m/s indicates no discrepancy in drag compared to the default configuration. Once reaching high-planing regimes above 9 m/s, the hull starts to trim and the design benefits from the ballast.
- **Stern ballast & removed stern extender** — Removal of the stern allows the hull to trim more easily. This effect is reflected in the advancement of the ballast benefit at 7 m/s. The gain in drag reduction (at 10 m/s) is doubled compared to the previous configuration. Note the decrease in wetted area of 0.6 m^2 .

- Bow foils & removed stern extender** — Being the holy grail in drag reduction, foils offer the benefit of reducing resistance without the need for adding ballast. Remember that planing vessels are highly dependent on the increase in mass (see Section 2.1.7). The inclined, surface piercing passive foils lift the bow out of the water at 5.4 m/s. However, the hull already experiences decreased wavemaking as the hull is partially lifted below this threshold. Once reaching 7.5 m/s, the drag starts to decrease with increased speed. The profit really shows in this operating range. Note that the actual drag will be higher at 10 m/s as stated in Section 4.1.6 due to the optimization, controllability and ventilation prevention of the bow foils. The foil's pitch should be fine-tuned during physical full-scale testing. The true drag will be near 400 N, still highlighting the benefit of this technique. In general, lifting the bow by 400 N (foils at 7 m/s) is far more effective than adding 400 N at the transom. The addition of the localized foils keeps the stability benefits of the foil-assisted configurations (see Appendix C.1.12).

Note that the unstable drag recording of the last configuration is a simulation issue and no physical concern is raised by reviewing the 3D plots.

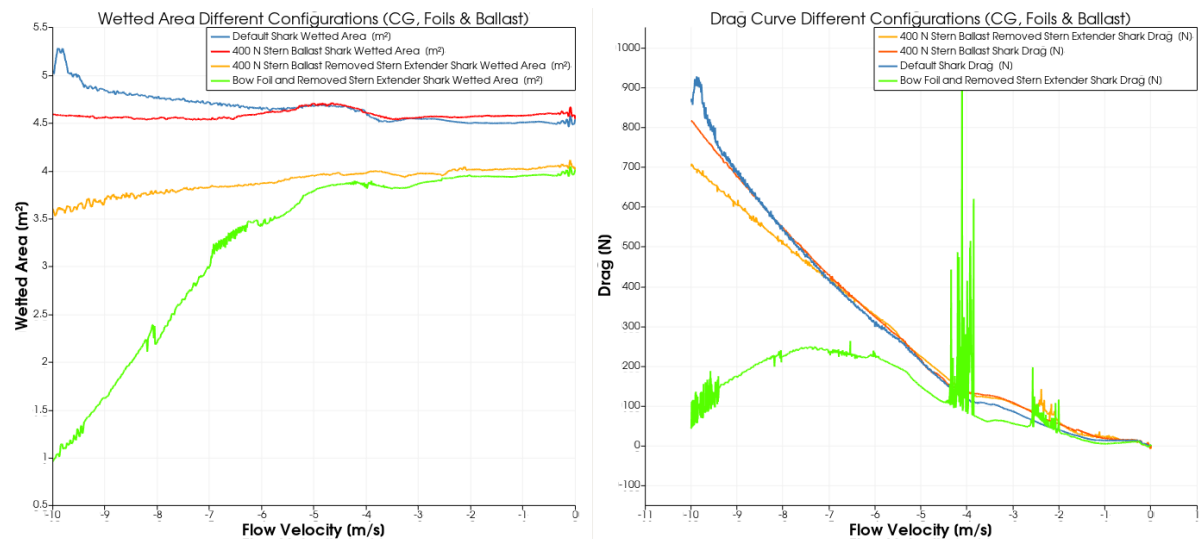


Figure 4.13.: Shark foils & ballast

Following on from the above simulations, a request was made to also conduct a test with the cockpit moved rearward (80 kg 1 m aft). As mentioned previously, the team rejected this proposal, but for the sake of completeness, the simulation plot has been added to Appendix C.50. It is noteworthy that an increased resistance is observed at a velocity of 3.3 m/s. The enhanced initial trim, which exceeded the standard setting, resulted in diminished clean flow separation at the transom. Once the hull gains speed, the effect diminishes (at 3.5 m/s). The bow draught at 6 m/s is zero, which has the effect of reducing wavemaking and enhancing the vessel's handling. However, this does not result in a reduction in wetted area thus viscous resistance.

A similar set of results was obtained in the planing regime when compared with the 400 N stern ballast without stern extender simulation, decreasing the resistance by approximately 5% due to the overall lower mass. Additionally, the wetted area exhibited only a marginal decrease in size. Consequently, it can be argued that the repositioning of the cockpit does not warrant the necessary effort. A rudimentary bow foil capable of generating a minimum of 20 kilograms of lift has demonstrated superior performance in comparison to all CG displacement configurations that have been examined as they do decrease wetted area. It is important to note that the bow foils that soon will be presented have not been engineered for peak performance; rather, their design is intended to address issues related

to unstable handling and minor erroneous installation. These vessels possess the capacity to accept errors in set speed and are resistant to unstable sea conditions, a trait that is attributed to

their over-dimensioned, under-pitched (AoA of 2°) design. This results in higher resistance, but it ensures proper functionality.

It is concluded that the Torpedo would not benefit significantly from a repositioned CG. This is counter-intuitive as the Torpedo is strongly pitched down at standstill. Nevertheless, it was discussed in Section 4.1.2 that the trim becomes neutral or even slightly pitched up at designed cruising speeds of 7-8 m/s, optimal for more planing-like behavior, already promoting post-hump linearity in the drag curve.

4.1.5. Cornering and Maneuverability

In the various racing disciplines of 2025, it has become clear that fast, sharp cornering is essential for achieving a good lap time. Various simulations and experiments were conducted to monitor the angular velocity and behavior, and the results were comparable. The reference simulations are discussed here.

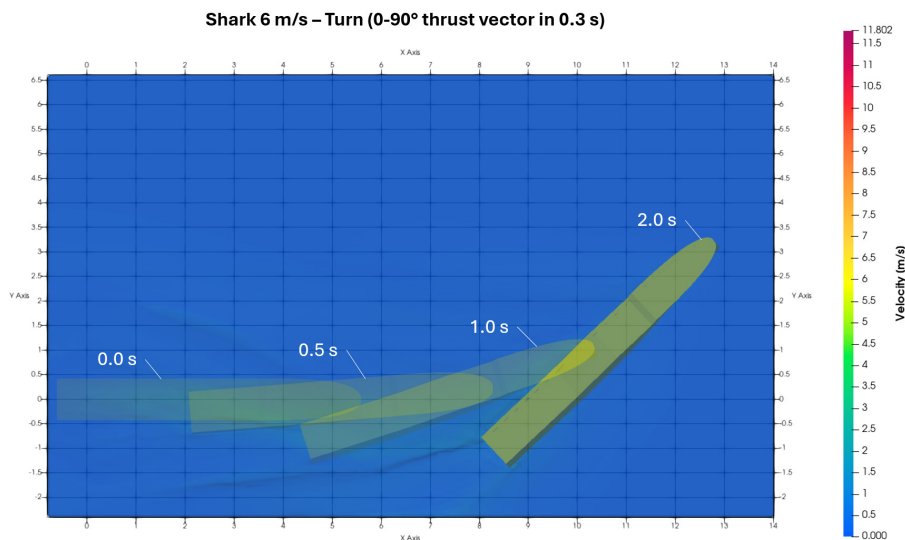


Figure 4.14.: Shark's default cornering at 6 m/s

The thrust vectoring is mimicked by time-dependent body-referenced forces. The simulation estimates the 90° thrust vector deflection time by 0.3 s (0.15 s for 45° simulations respectively) and does not take the influence of the rudder surface into account (widening the turn). Still, this simulation is highly valuable for the acquisition of comparative data, isolating the reaction of the hull's geometry. The application point is identical in every simulation of the same hull (Shark equals upgraded Shark) and aims to be as close to reality as possible. The thrust magnitude is matched by the resistance acquired from the continuous drag simulation of the specific hull, simulating no thrust changes along the turn.

For 90° thrust vector rotation at 6 m/s (Figure C.43), the hull's turning rate reaches up to $45^\circ/s$. In 45° thrust vector rotation at 10 m/s, the converged turning rate is approximately $42^\circ/s$. Both simulations showed similar, safe, and maximized operational roll at 38° . Therefore, the roll inclination proves to be an acceptable measure for the turning rate. Due to the fact that it takes longer for the 6 m/s simulation to reach the full deflection and a steady turning rate (and predictably also the breaking affect of the transverse rudder surface). Consequently, the 6 m/s will take longer to complete a full turn, despite the similarity in converged turning rate. This is demonstrated in extensive simulation analysis shown

in Figure C.44. Hence, in race path planning, a wide high-speed corner and early start of the curve is beneficial for improved lap time. Obviously, this theory states a global maximum not making the corner even wider. While not very extensively analyzed in simulation, the $45^\circ/s$ proves to be a good compromise between keeping its momentum and taking the curve. Note that both 6 and 10 m/s simulations show that an added central, vertical stern fin would be fully submerged at all time. This will be kept in mind when designing adapted geometry.

A more aggressive simulation including 90° thrust vector deflection at 10 m/s demonstrated failed seakeeping in flat water conditions for the default geometry (run 35 in simulation logging, see fourth quadrant of Figure C.44). Loss of control is found only 0.6 s after simulation start. The addition of roll limiting forward placed sponsons may allow for more aggressive cornering and may help the piercing bow to lift and turn. The loss in speed along a random turn is found acceptable for all cases in contrast to the Torpedo.

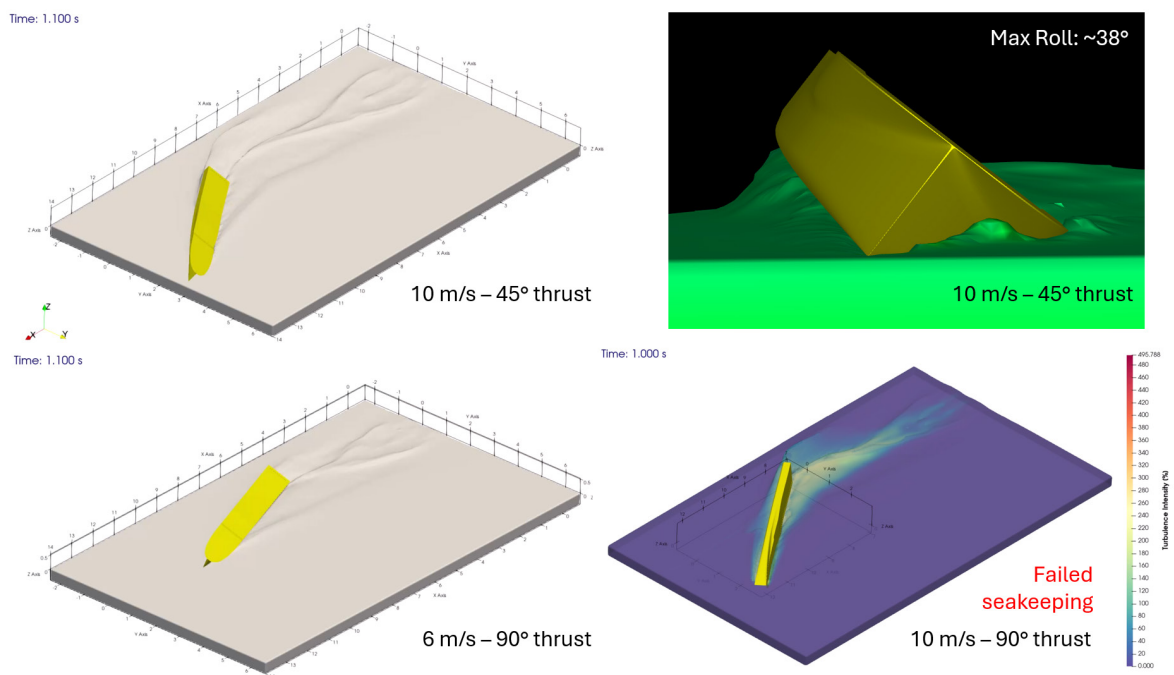


Figure 4.15.: Shark's default configuration cornering at 6 and 10 m/s

The Torpedo's cornering ability has only been tested at 6 m/s, with the same timing as the Shark's thrust vectoring. The Torpedo's time-dependent thrust magnitude was defined using the constant thrust principle, as in previous simulations.

Because the Torpedo showed single (Figure C.44) and potential double tunnel collapse consecutively — strongly slowing the hull down —

and the path's radius is significant compared to the Shark (slow turning rate, grossly 60% of the Shark's turning rate), the testing was discontinued. There is no confidence in the existence of better cornering capabilities of the Torpedo, either with or without modifications, compared to the Shark. This has to do with both increased resistance and limitation of the bank or roll angle (7° at 0.7 s), previously indicated as a measure for angular velocity.

4.1.6. Adapted Hull Simulations

In this Section, the potential of various geometry upgrades are discussed. The Shark is tested for straight line performance and cornering.

Shark

Geometry updates:

- Replaced decks (25.8 kg reduced to 6 kg, 7.3% total mass reduction). Water deflection with sea wall inspired geometry.
- Reduced solar panel area (16.56 kg reduced to 5 kg, 4.3% total mass reduction). This increases roll allowance providing better cornering and safer operation.
- Bow foil raises pitch, increases planing efficiency, and reduces wetted area and bow directional guidance authority.
- Stern fin(s) provide directional guidance. To limit drag, the mainly the projected frontal area and also the wetted area should be limited to a minimum. However, a sufficiently large lateral area should be present to stabilize the craft, similar to the aerodynamic tail area described in Section C.1.49.
- Sponsons (neutral inner camber or vertical inner freeboard) provide roll stability and both grip and guidance in sharp cornering. They may add about 5-10 kg of mass, depending on the chosen construction materials. Sponsons limit the cornering roll (maxed out by test) for safety and therefore turn radius for a given speed. However, the same turn path may be followed at higher velocities; increasing turning rate and governing high entry and exit speeds: the key racing metrics for effective cornering.
- Removed stern extender. This reduces wetted area and allows for more dynamic trim. Also, the design was slightly deflecting transom flow down, planting the bow at planing speeds.

The addition of lifting strakes (reverse chine) and spray rails to the Sharks geometry proved to have negative influence on various performance indicators (wetted area, drag, etc.) due to the absence of a steep deadrise in the midship and stern region. As noted, they are more common in higher speed use cases (> 45 mph, according to Hydrofin). Thus, based on relevant flat water simulations, these adaptations are not recommended. Also, as found in the wave simulations, the addition of an inclined wave deflection front deck is not mandatory if sponsons (and fins) are installed. The sponsons and fins add bow lift reducing the chance of greenwater getting on the deck and eventually ending up in the cockpit. A flat deck, without any adaptations to the cockpit's walls, is totally acceptable in normal sea conditions.

Performance Increase In order to analyze the performance of various geometry upgrades, numerous CFD analysis has been executed. It is important to both isolate and combine improvements as they might interact in certain situations. The fully assembled hulls are subjected to various speed conditions, aiming to guarantee no reduction in performance for a certain operation condition and achieving significant gains at the intended high-speed ranges during racing with increased power. As previously mentioned, drag reduction, cornering, and safety are critical for every advancement tested. Using both innovative and conventional design principles, a dozen simulations revealed an increase or decrease in

performance. For every environmental settings change, a benchmark (default configuration) calculation has been executed in order to measure the precise increase in performance using exactly the same conditions.

In preliminary tests on fins and foils, a single ultra fine mesh full body simulation was conducted. This proved to be impractical due to the heavy workload. In addition, thin geometry is not accurately captured by the meshing techniques used in Flow3D. To reduce the computational demand and increase accuracy any further, a separate locally refined mesh CFD analysis was completed in Solidworks Flow simulation. The results of this study are configured as body-referenced input forces in the Flow3D geometry configuration. Within the Solidworks simulations, flow velocity and surface roughness was matched with the connected Flow3D setup. This allows for the use of conventional mesh sizing in Flow3D Hydro while accurately incorporating forces produced by thin structures. The calculation time of the entire workflow has been reduced by over 80% using this approach.

The results of the analysis is given in Table C.17 (extract from the simulation reference Excel), listing the performance benefit for every situation. The drag is derived from the converged steady-state observations. Note that the stern extender limits the hull to trim in planing regime, enforcing wavemaking and diving behavior. The pressure distribution at the bottom of the extender supports this statement. At medium speeds (5 m/s), the Shark does not benefit significantly for removing the stern. However, at high speeds, removal allows for effective dynamic trim and will reduce drag significantly as found in test 13.

Upgraded Shark in Waves The following explanation is based on the results of simulation run 42 (5 m/s) and 43 (10 m/s), holding exactly the same water conditions as in 4.1.3. When riding a wave at 5 m/s, the sponsons accelerate the bow up by both dynamic lift and displacement. The high deadrise of the sponsons ensure soft entry while limiting deck slamming of the main hull. As a consequence, the hull experiences a higher pitch deviation up to 8° , 3° higher than in the 10 m/s default configuration in run 41. The angular velocity was increased by 40%. The removal of the stern extender also contributes to

the increased pitch. Still, no issues for most perception sensors. The greenwater is reduced to only 2% of the decks and spray to only 15%. The rear deck is completely free of water. Note that is is not entirely true for any kind of cornering both in waves and flat water conditions. The drag force increases to approximately 2 kN on impact and show a short, instantaneous spike due to the first contact with sponsons reaching up to 3 kN for about 10 ms. Nevertheless, the average bow impact slightly reduces to a magnitude of grossly 6.5 kPa ($\approx 13\%$) as the sponsons dampen some of the impact (sponsons may show higher local pressure across its distribution). The energy on wave encounter accelerates a wall of water vertically between the sponsons and the freeboard of the main hull. The height is acceptable and doesn't tend to enter the cockpit. However, in cornering, this requires special care. Results of the 5 m/s simulation can be found in Figure C.47.

Figure C.48 shows the results of run 43, indicating a drag peak of keeping below approximately 1 kN. The bow pressure is significantly reduced to 7 kPa compared to run 41 of the default configuration at 10 m/s, which demonstrated a staggering 45 kPa. What is the driving force behind this result? 3D rendering shows that the bow foils keep the bow high and prevent the overhanging bow flanks from approaching to wave crests. In the same way, even the sponsons are fully preserved from contact with water. Ultimately, splashing waves are minimized and the Shark demonstrates excellent seakeeping at increased speeds, in this type of waves. Note that the foils themselves have to withstand limited impact and bending loads. When studying the deck water and spray, it is noticeable that no water reaches higher than the gunwale. The angular velocity was halved compared to run 42, underlining the smoothness of ride. Note that the removal of the stern extender makes the pitch variations slightly higher, but (long) waves can be followed more accurately, increasing seakeeping at low speeds too. The risk of cockpit flooding at low speed and standstill reduces.

This is the priority order for implementation of

the suggested upgrades:

1. Stern fin
2. Bow foils + stern extender removal
3. Sponsons (allows for faster cornering) or limitation of the rudder angle (to guarantee safe cornering without sponsons at higher speeds)

The third advancement can be implemented without the second. However, the bow foils and stern removal combination has most impact on (straight line) racing efficiency. Added bow mass (sponsons) should be compensated by bow foils or stern ballast. In the event that the implementation of bow foils is not feasible, consideration should be given to the utilization of transom ballast, as discussed in Section 4.1.4.

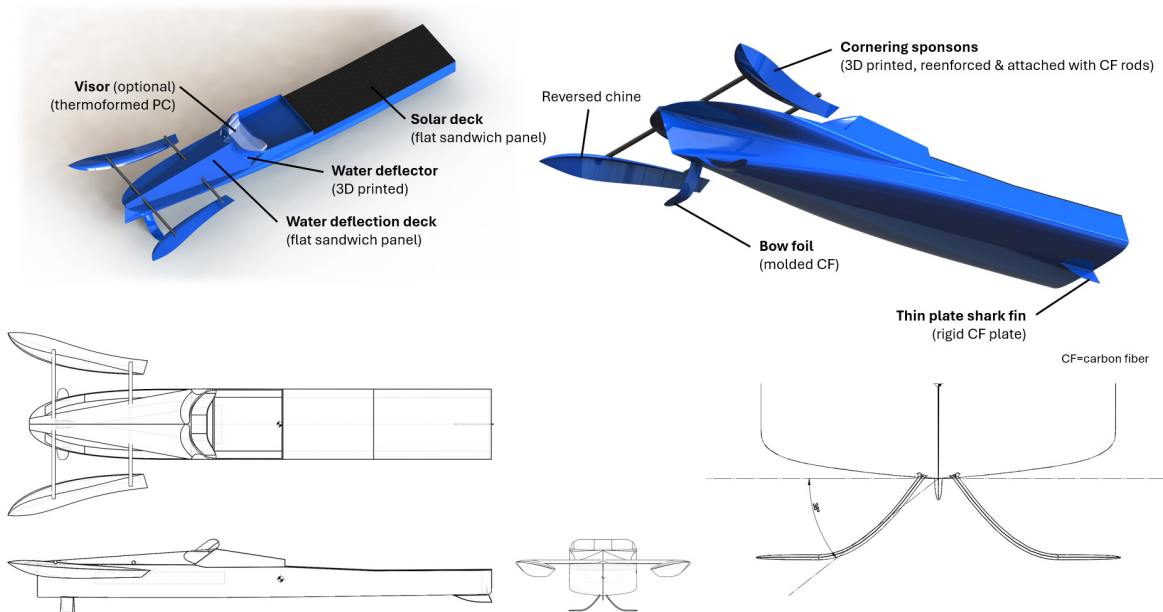


Figure 4.16.: Shark's upgrades (v3)

Note that version 4 of the geometry upgrades in documentation is the final design. The difference between v2 and v3 are in the positioning and geometry of the sponsons. Version 4 is performing exactly the same as v3, but has a simplified front deck as wave simulation indicated a more pronounced, complex deflection deck is not required and also has optimized (similar performance) foils to reduce local material stress. Solidworks Simulation demonstrated 10-13 MPa in the Von Mises plot, depending on the load conditions. A $SF \geq 8$ is suggested if the fin will be 3D printed with carbon fiber composite filament (choose an impact resistant variant for wave and cornering loading). The fins can easily be 3D-printed with carbon fiber filling of 15%. Version 4 simplifies implementation, manufacturing and cut costs. In this version, the connec-

tion between the sponson and the carbon fiber tube is modeled as the sponson geometry is prepared for production.

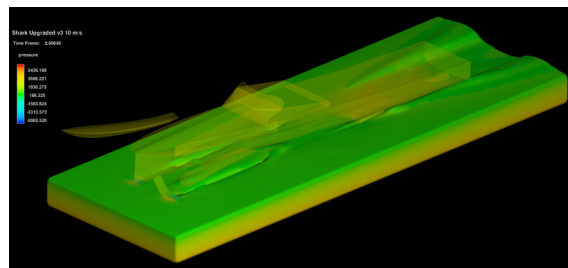


Figure 4.17.: Shark's geometry v3, test 15

In conclusion of Table C.19, adding fins and foils is not beneficial for low-speed operation (except seakeeping). However, once higher speeds ($> 5 \text{ m/s}$) are relevant, they provide significant drag improvement reaching up to 21.3% (test 8). Combining this with the removal of the stern extender the team can gain up to 51.6% in drag reduction requiring only 417 N of thrust at 10 m/s

in flat water conditions (test 15, Shark geometry v3). The improvements are designed for an operation speed of 10 m/s. Based on the findings in the validation Section 5.4, it is expected that even lower thrust is required and higher speeds are possible in practice. Seakeeping at 10 m/s is also improved significantly and safety is guaranteed with increased confidence.

Note

Fine-tuning trim at practical maximum operating speed for racing is strongly advised. Procedure: sail at maximum speed and add mass up to approximately 10 kg in the bow (or stern). The key is to use leverage by placing the payload as far away from the CG as possible in order to change balance without significant influence on total mass. Search for maximized speed on straight line racing. A less efficient technique is to slide a 10kg weight from the CG forward (or backwards) until optimal (stable) trim is achieved and top speed is measured. A third, completely different but better fine-tuning technique that definitely takes precedence over the first and second (adding mass) is to adjust the AoA (angle of attack) of the bow fin. This can be done with a wedge, but a motorized AoA that takes speed, turns, and wing stall into account is obviously even more interesting increasing both efficiency and consistency in optimal trim. However, this study recommends a fixed fin to limit complexity and increase hull response predictability.

Very short motor strut is recommended. Long motors stems/struts/legs (see Section 2.8.3) may require less lift from the bow fin(s) or need increased bow ballast. Also, note that the drag might decrease with increased speed (high region) due to reduced wetted area and trim, requiring the power to be designed for the peak thrust (including strut drag).

Cornering Improvement The modified Shark with iteratively designed bow sponsons exhibits exceptional turning capacity. The sponsons are free of the water with ample clearance when sailing straight ahead and making slight turns. However, when fast, short turns are necessary, the sponsons engage aggressively and dynamic lift on the outside ensures that the Shark always maintains a safe roll. This makes the design foolproof for experiments with the potentially unstable autonomous system. During flat water testing, the sponsons remain “planing” on the water surface, allowing the Shark to maintain as much momentum as possible. Note that the sponsons have a remarkable shape. They are

curved upward and outward to make the turn as effectively as possible and to steer. In a sharp turn, the inner bottom surface of the sponson hull is positioned almost vertically (at right angles) to the water surface. This causes a violent upward spray sheet that is deflected by a spray rail or lifting strake on the inner gunwale. The reoriented spray provides extra lift and prevents the sponson from diving. In addition, this also protects the cockpit from incoming water. This, together with the wave-deflecting front deck, ensures that both on straight and curved paths, the rear deck with solar panels is protected from corrosive salt water (greenwater).

The roll angle is limited to approximately 23° without compromising cornering performance. This is beneficial for limited antenna field of view and therefore more reliable communication with the vessel. The angular velocity reaches 38° as shown in Figure 4.18. Reaching 38° , the turning rate is comparable with the default configuration of Section 4.1.5. However, far less roll is required making the turn safer to execute. This happens because of the added bow lift and increased directional guidance of geometry

advancements. Also, note that the cornering thrust of 417 N is significantly lower compared to the default 90° thrust simulation's of more than 800 N. Using increased power with the new geometry allows for even faster turns. The sponsons show only partial submersion, indicating a safe margin for the application of grossly 20-40% more thrust. It is noticeable that the sponsons significantly limit the required turning radius (3-4 m) as they guide the boat into the curve. If the racing environment allows larger turns, it is expected that the limitation of the rudder angle ($[45^\circ; 90^\circ]$) enables increased cornering efficiency and higher exit speeds. Experiments are required. Turning in a spiral with slowly decreasing rudder deflection and vertical top down video recording by drone enables quick and effective analysis of the steady-state angular velocity for different input controls. The acquired optimal steady-state angle will be close to the actual optimized angle for use during races including both entry and exit behavior.

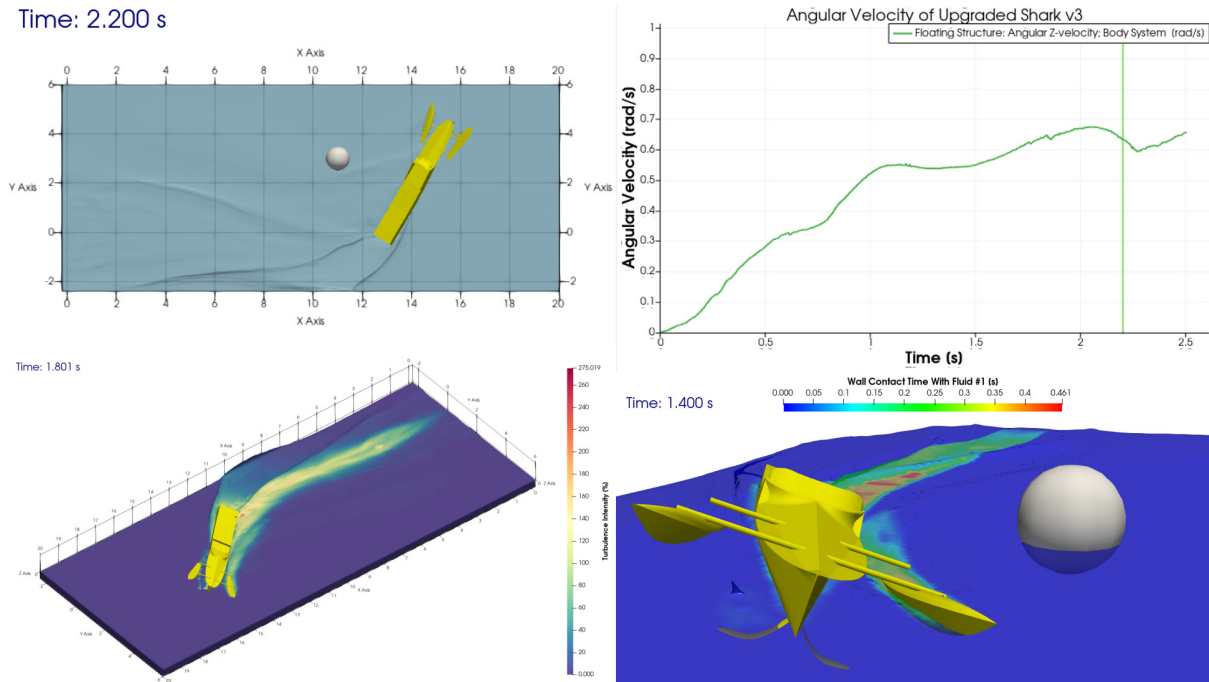


Figure 4.18.: Shark's geometry v3 cornering behavior at 90° Rudder Deflection in 0.3 s

Torpedo

As simulations, experiments, and theory negatively advise the use of the Torpedo for future competition, only suggestions for upgrades are delineated. In order to devote more time to concepts with greater potential, it was decided not to carry out extensive simulations, analysis, and CAD modeling for the modifications to the Torpedo. Geometry updates to consider:

- Added passive lifting foils under sponsons increasing running stability and decreasing drag by lifting the main hull (partially) out of the water.
- Bow fins to make the trim more optimal and consistent along the operating range. Active control allows for higher speeds and better cornering.
- Repainted bottom of the hull, roughness reduced to 0.0001 m or less.

To make use of the Torpedo hull at high speeds without balancing concerns, hydrofoils are the only proper way to do so. Control fins with trim authority complicates the system just as hydrofoils do.

Performance Increase Coating the hull is the least invasive approach and has the greatest potential for reducing friction, enabling efficient use at moderate speeds. Up to 7.5 m/s is considered safe at all time, except in moderate wave conditions where slamming becomes a structural concern. Coating is one of the most influential properties, demonstrated in simulations. Further analysis and quantification of improvements was not effectuated.

4.1.7. Conditions, Mesh & Convergence

For a comprehensive overview of the primary constraints, boundary conditions, objectives, and simulation parameters, refer to the Excel file provided in the portfolio. The report of the convergence study is appended below.

Convergence Study

To verify of the mesh refining dependency of calculations (apart from full-scale validation), the overarching Shark high roughness (0.003 m) continuous drag test simulation setup was selected. First, a mesh convergence study was conducted in which the three 0-10 m/s towing datasets of the Shark were exploited, experimenting with a uniform mesh of 0.07, 0.05 and 0.03 m. Note that the latter does not contain data in the speed range above 5.2 m/s due to calculation limitations and that the 0.07 m coarse mesh results are invalid above 7 m/s due to lateral sliding of the hull in the computational domain. It should also be noted that the X-axis in continuous towing plots are shifted by approximately -0.08 m/s as the speed axis is based on the inflow speed (X-max velocity) rather than the mean speed at the position of the hull. In most cases, this is negligible.

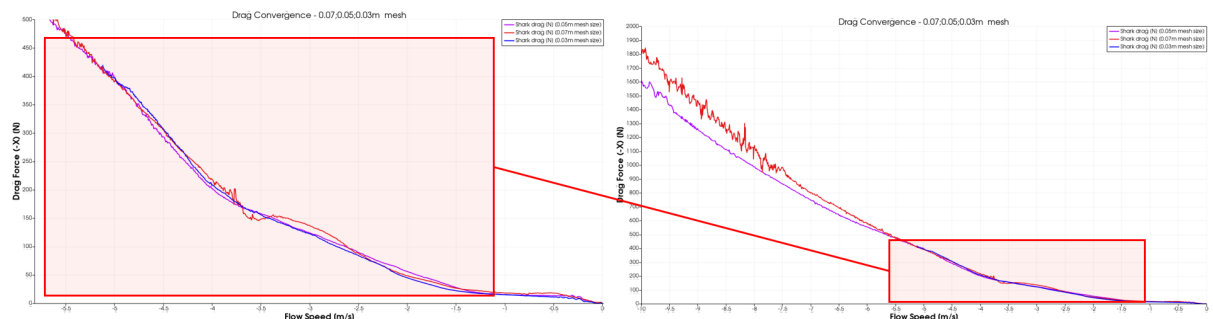


Figure 4.19.: Shark's drag mesh convergence study for continuous towing results

In Figure 4.19, this mesh comparison was plotted, showing that the results hardly change for speeds ≤ 5.5 m/s. The mesh refinement mainly results in a smoother curve, rather than better average values around a small speed range. The higher speed curves require an additional convergence test. For this purpose, it was chosen to perform three steady-state simulations at 10 m/s and determine an averaged response for all mesh sizes. This analysis reveals that the speed-resistance response exhibits a dependence on the mesh in high regimes. In conclusion, to reach

the study's required accuracy, a sufficiently fine mesh (≤ 0.05 m) must be selected.

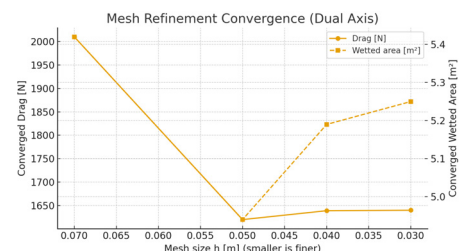


Figure 4.20.: Shark's mesh refinement of the drag & wetted area at 10 m/s

Spatial Grid-Convergence A four-grid refinement was run at $h = \{0.07, 0.05, 0.04, 0.03\}$ m for the 10 m/s steady-state analysis (see Figure 4.20). Following the NASA WIND tutorial (Roache/Celik three-grid procedure, Appendix C.7.6) [74], the formal analysis used the triplet $(h_1, h_2, h_3) = (0.03, 0.04, 0.05)$ m with constant physics and numerics. Drag is treated as the primary quantity; wetted area is included as a consistency check since it is more sensitive to free-surface discretization. The observed order of convergence p , Richardson-extrapolated value ϕ^{ext} , and Grid Convergence Index (GCI) are reported with $F_s = 1.25$. Between $h = 0.05 \rightarrow 0.04$

and $0.04 \rightarrow 0.03$, drag varies by only $\sim 1.2\%$ and 0.06% , respectively, indicating a practical plateau for $h \leq 0.05$ m. Wetted area shows somewhat larger but still moderate changes ($\sim 5.1\%$ and 1.2%), consistent with its higher sensitivity. On this basis, $h = 0.05$ m was selected as the main mesh for the simulation studies, as it balances accuracy with computational cost while lying on the plateau indicated by the finer solutions. Convergence of refined time steps (transient analysis) is not discussed because the solver employs adaptive time stepping that automatically refines when exceeding the stability limit.

Quantity	p	ϕ^{ext}	$\text{GCI}_{21} [\%]$	$\text{GCI}_{32} [\%]$	Asym. ratio	ϕ_1 (fine)
Drag [N]	13.332	1640.02	0.002	0.078	0.000	1640.0
Wetted area [m^2]	6.820	5.260	0.234	1.681	0.020	5.25

Table 4.1.: Three-grid Roache/Celik metrics using (fine, medium, coarse) = (0.03, 0.04, 0.05) m. Safety factor $F_s = 1.25$.

Drag shows a high observed order of convergence ($p \approx 13.33$) and a very small fine-medium GCI ($\text{GCI}_{21} \approx 0.002\%$), because the medium and fine values are nearly identical — typical of a plateau rather than clean asymptotic behavior. Compared to drag, wetted area shows a lower observed order of convergence and higher GCIs, which is expected given its stronger dependence on free-surface resolution. The asymptotic-range ratios are $\ll 1$ (drag ≈ 0 , wetted ≈ 0.02), indicating the sequence is not strictly in the asymptotic range, but the plateau conclusion for drag below $h = 0.05$ m remains supported by the small successive changes.

Extended Discussion Although the wetted area (WA) is not the only important parameter governing the hydrodynamic performance of high-speed hulls, it remains an essential planing performance indicator. The wetted area directly influences viscous resistance and the distribution of hydrodynamic pressure, which in turn affects trim, lift, and overall flow behavior at varying speeds. Accurate estimation of this area is therefore essential to ensure that the solver correctly captures the transition between displacement and planing regimes. A brief convergence study on the wetted surface provides confidence that the mesh resolution and numerical setup adequately represent the free-surface interaction and pressure distribution. While total resistance and dynamic stability are ultimately more critical design criteria, the wetted area serves as a practical and sensitive intermediate quantity for assessing numerical consistency and model fidelity.

In Figure 4.21, the coarse mesh plot highlights severe deviations compared to the other curves. This is as expected based on the observations of the 3D mesh; a coarse mesh removes sharp edges (bow) from the input model, resulting in reduced wetted area in the speed range below 7 m/s. Above this speed, the results are considered invalid due to lateral displacement of the hull. However, based on the results of the 0.05 and 0.03 m mesh, the wetted area quickly converges.

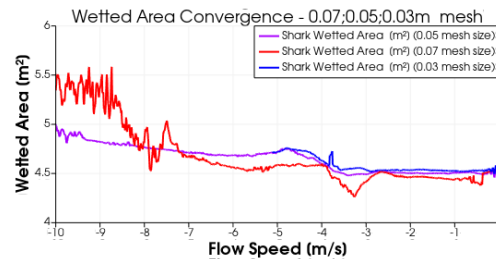


Figure 4.21.: Shark's WA mesh convergence study for continuous towing results

4.2. Boat Curve Dashboard — BCD

4.2.1. Goal & Distinction

There are many mathematical and experimental graph correlation studies conducted in the field. Often with great results and good accuracy. However, global theoretical derivation with acceptable curve accuracy has still a long way to go. Currently, only CFD analysis offers results capable of approaching the quality and preciseness of full-scale and official ITTC tow tank tests. Recent studies try to find experiment approaching curves by regression and deep-learning tactics [75] rather than direct empirical derivation [79].

The development of the Boat Curve Dashboard or BCD software included in this study has a distinct purpose: theoretical coupling based on experimental calibration for both displacement, semi-displacement, and planing hull types. It makes it easy to match a very specific hull type by integrating experimental data points and rearrange the data with parameter curve sliders shown in the main window. In this version, matching is done manually. A subsequent update to the software might encompass the incorporation of mean square error (MSE) optimization for curve fitting. As highlighted, strength of this software lies in the theoretical coupling from base formulas which influence the shape of the graph(s). That implies a user can change for example the mass of a base design, and monitor the changes on a resistance and power graph in the BCD environment. Please note that this is only a first release and that the software is still in its infancy. Further development may improve quality and expand functionality. The interface is illustrated in Figure C.57.

While not extremely accurate, it is possible to get a sense of scale/magnitude. To achieve increased accuracy, geometry and balance should be known and the exact and changing decomposition of drag components should be known as in Figure 43 of “Benchmark Testing of Spray Deflection Technologies for High-Speed Planing Craft” of B. Molchanov [76]. Each component can be coupled to physical parameters and has its percentage of contribution to the total drag (known from experiments or CFD). This would advance the software to a more accurate morphing tool. In the current version, the curve is sectioned in three parts: (1) pre-hump, (2) hump, and (3) post-hump. Each is characterized by a mathematical (theoretical) model that tends to estimate the drag in its section. This model

is mostly defined — except for correction factors — by the most significant effects that add to the drag decomposition. For example, idealized planing craft in a post-hump behavior react based on theoretical approximations of friction drag, displacement / semi-displacement are mostly depending on pressure and wavemaking and only partially on friction drag. The three-section decomposition principle indicates poor estimation in very high planing regimes above Froude 3.5, as the spray and air drag kick in and thus require additional “in-planing” corrections. Additionally, during the transition from $Fn_{\nabla} = 2.5$ to 5. Still, BCD is a very useful tool for experimentation.

A naval design engineer can derive suggestions based on the changes that provide the most advantage in power reduction: making it slender, reducing mass, reducing air resistance, making the hull longer, etc. Note, prediction accuracy drops when adapting the hull’s parameters by a significant number. For insight on the accuracy of every parameter, a very extensive cross-correlation CFD or experimental analysis should be conducted. In extend, empirical papers [17] may serve as auxiliary validation material.

Another benefit of the software is that it simplifies the transfer and transparency of information incorporated in this study. That particular principle is key, as the study itself will serve as a team reference work on hull design (specialized in the UASBT hulls). The idea is reinforced by integrating features like a detailed Parameter Help window, explaining the meaning of every variable and providing insight into realistic ranges to set. Providing an easy-to-run EXE-file and the original source code, increases usability and makes future upgrades and in-depth analysis more accessible.

4.2.2. Functions

In short, these are the main functionalities embedded in BCD:

- Live adaptation of parameters providing a live side-by-side comparison between designs, validation and inter-/extrapolation
- Visual correlation and calibration by CSV point import
- Export of graphs to CSV, parameter-graph screenshot
- Simple scaling, zooming, range adaptation,...
- Parameter Help documentation

Note

The pre-hump stage of the planing-type craft is not accurately modeled as this is a region of minor interest in planing craft. Still, as theory and full scale experiments suggest, for comparable light-weight vessels in near hump planing regime, a true planing model is showing higher resistance compared to the (semi-) displacement variant.

5. Experimental Validation

The experimental validation phase serves to verify the numerical predictions and design assumptions established during the analytical and simulation-based stages of this research. A combination of full-scale and model-scale testing was conducted to ensure both accuracy and scalability of the results across different Reynolds and Froude regimes.

The first section presents the full-scale towing tests, which provide a direct measurement of hydrodynamic resistance and form the reference datasets for subsequent scaling and comparison. These results are complemented by controlled towing tank experiments performed on scaled models to investigate hydrodynamic performance under repeatable laboratory conditions. To evaluate aerodynamic influences and qualitative flow behavior around above-water ge-

ometries, wind tunnel testing was performed. These tests primarily aim at visualizing flow for quick evaluations. Furthermore, scale model trials using (radio-controlled) prototypes were carried out to assess handling and maneuvering characteristics, dynamic hull behavior in realistic conditions and towing tests. These experiments bridge the gap between idealized simulations, predictions based on literature and experience, and real-world operational performance.

Finally, the collected datasets are compared and analyzed to validate the numerical models. The results are discussed in terms of consistency across test scales and accuracy of resistance predictions. The chapter concludes with a critical evaluation of the model's predictive capability and the reliability of the experimental and simulation methodology.

5.1. Full-Scale Towing Test

5.1.1. Goal

The goal of the full-scale tests is to validate both scaled experiments and simulation of the Shark and the Torpedo. Additionally, predictions derived from theory can be either confirmed or rejected and various behavioral effects can be observed. In previous years, towing tests of (e.g., Torpedo) became available for further analysis. Despite these realistic graphs, there are several reasons to retake the experiments. For example, the general test conditions are unknown (density, water depth, wind, setup, appendages, etc.). Also, unidentified artifacts appear at the extremes of the graph obtained. For example, instability is visible at the maximum of the re-

sistance curve. Is this a physical phenomenon of the hull's dynamics? As tested in the lab with the same measuring equipment, it is probably due to an “end-of-scale” phenomenon. The maximum scale deflection is below the expected maximum tensile force that can be extrapolated from the rest of the graph. In addition, there are rumors that the towing test was performed with a very short line (≈ 3 m), straight in the pronounced wake of the heavy tugboat. In this study, simulations show that small waves and underflow have a major impact on the measurement, making renewed, independent tests relevant.

5.1.2. Procedure

Effective testing requires careful planning and preparation. Therefore, the team is briefed about the goals, setup and constraints related to these tests and both the planning for testing procedures and hull transport are defined. In essence, the experimental procedure is planned according to the following rules:

- If wind conditions are significant (>3 m/s), the test will be conducted twice; both headwind and tailwind. These measurements will be averaged for increased accuracy and wind independency. In that case, the influence of aerodynamic drag can be derived by measuring the stationary craft wind speed. For extra resulting (qualitative) data, the decks were covered with a grid of air-flow tels.
- The speed recordings are done on the Xsens (PC at towed hull) and backed up by an accurate phone GNSS speed estimation in the recording overlay on the tugboat.
- The length of the towing line is determined based on the wake of the tug boat at full speed (36 km/h, ≥ 25 m).
- The environmental conditions are recorded for the tests as defined in Section 5.1.4.
- The speed of the test is gradually increased over 2×650 m to ensure a steady-state measurement at the recorded ground speed.
- In advance of the towing test, the 30 HP tugboat is tested for maximum speed and the outboard is sufficiently trimmed in to prevent porpoising during towing.
- To calibrate the results, the line is gradually towed to maximum speed and the force is measured. The significance and correction of the results will be evaluated.
- The influence of the strut used should be determined for correction over the entire speed range, especially if one of the hulls cannot be isolated in a test without appendages.
- The loading conditions are set to default for normal operation. Validation will prove whether derived loading condition simulations may be considered valid.

5.1.3. Setup

The issue of the range and static friction of the analog scale will be mitigated by adding an additional digital load cell measurement device in series and connecting the towing rope to a single pulley system dividing the force measured by two, as shown in Figure 5.1.

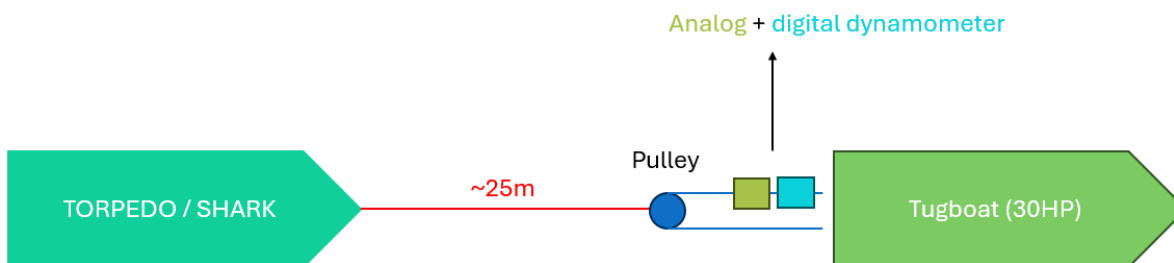


Figure 5.1.: Full-scale towing setup diagram

5.1. FULL-SCALE TOWING TEST

The range is doubled up to 1000 N while ~ 600 N is the expected global maximum force during tests. To increase the data rate of the measurement (increased scatter graph results, ~ 600 points for every full range dataset), the force measurement is recorded with an overlay including position, distance, speed (0.1 km/h resolution, 1 Hz update rate, serves as a speed measurement backup for the Xsens IMU/GNSS set at an update rate of 4 Hz) heading, timestamp (sync data), etc. This is illustrated in Figure 5.2.



Figure 5.2.: Full-scale towing recording

5.1.4. Conditions

To expand the usability of the test results in future studies and accurately model tests in simulation, it is of great importance to record environmental test conditions (ITTC).

Parameter	Value	Remark
Mean wave amplitude [m]	0.05	
Mean wave period [s]	1.3	0.45 m
Salinity [ppt] / density [kg/m ³]	35 / 1026	
Water temperature [°C]	10	
Air temperature [°C]	11	
Wind direction (relative to towing test axis) [°]	0	ZZW
Wind speed [m/s]	1.5	
Length towing rope [m]	25	
Resolution force measurement [N]	1	
Water depth [m]	6	
Date	29-10-2025	Afternoon

Table 5.1.: Test conditions of the full-scale towing test

5.1.5. Results

Processing Data

During the experiments, the Xsens system intended for velocity recording failed due to technical issues caused by cable corrosion. Fortunately, a backup measurement device was anticipated and provided sufficiently accurate speed data to complete the analysis.

The raw data from the four towing experiments — towing rope, Torpedo with short strut, Torpedo without strut, and Shark with short strut — were first transferred to a structured Excel work-

sheet. Except for the towing rope test, which contained only analog measurements, all other datasets included both analog and digital load cell recordings for each corresponding speed. Each dataset covered a recording period of approximately five minutes, with samples taken at one-second intervals, leading to approximately 600 force recordings for every completed measurement.

Occasional measurement artifacts were observed, such as load cell “hold” states caused by temporary freezes in the digital acquisition device and transient peak values originat-

ing from stuttering throttle control, resulting in brief line-tension spikes. These irregularities were removed through filtering to ensure a consistent and representative dataset. The cleaned data were subsequently exported as semicolon-separated value files (CSV) to enable automated Matlab processing.

In the second stage, a dedicated script (see Appendix C.7.4) was used to import each CSV file and sort the data by ascending towing speed. The dataset was then divided into an equal number of point groups. For each group, the mean towing speed and corresponding mean resistance were calculated, effectively reducing data scatter and improving the clarity of the overall trend. This averaging process also ensured that regions with denser measurements contributed proportionally to the final curve resolution.

Subsequently, a smooth interpolation was applied to the mean values, producing a continuous resistance curve with a fixed resolution of

approximately 1000 points per 30 km/h. This approach was compared to various regression methods, proving to be the most suitable tested. Both the scatter data and interpolated curve were plotted together, and the resulting curve was exported as a new output CSV file. This procedure was repeated for all datasets.

In the final stage, a second Matlab routine combined the processed CSV files and derived the strut resistance contribution by subtracting the Torpedo without strut from the Torpedo with strut dataset. This step enabled a direct estimation of the appendage drag, from which the theoretical correlation for the strut frictional resistance was obtained in terms of the product $SC_f = 0.00435461 m^2$, derived from Equation C.7. When evaluating the total form drag, S should be replaced by the frontal projection A , similar as in Equation C.10. The quadratic regression curve is used to derive the Shark's resistance without the strut.

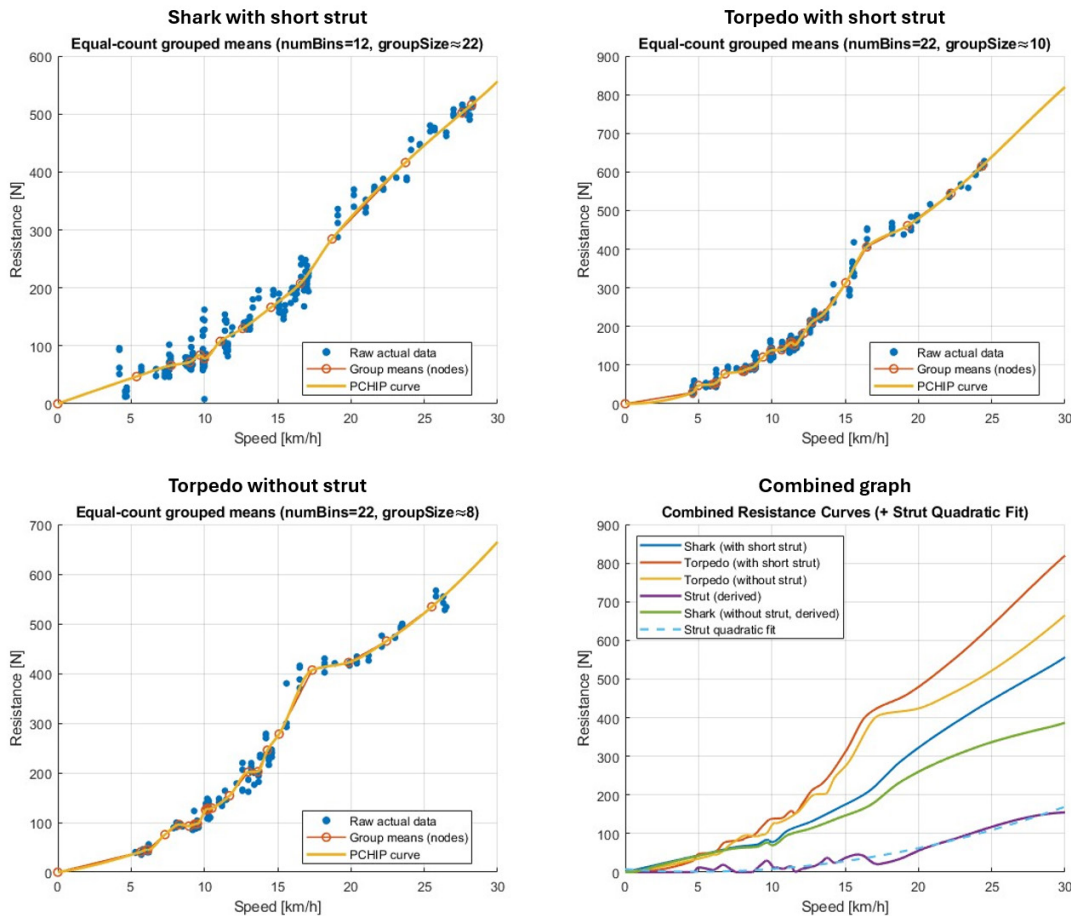


Figure 5.3.: Torpedo and shark full-scale results

Air Flow

The air flow tels indicate the turbulence zones as they where predicted. Behind the cockpit — at 25 km/h — the tels indicate disturbed air flow and recirculation up to 1.5 m aft the pilot. The cockpit's dashboard clearly deflects the flow up-and outwards. Also, for which the tels where originally placed, these ribbons — at the deck's edges on the port and starboard sides — do not turn inwards or outwards. This shows that no lift nor downforce is generated by the decks because they do not experience any significant angle of attack (trim) during operation. The tels on the middle deck of the Shark do show limited lift due to the locally inclined deck geometry. As there was minor wind ($< 3 \text{ m/s}$) during the tests and the expected aerodynamic drag is low ($< 5 \%$), the aerodynamics where not analyzed any further.

Hull Behavior & Directional Stability

During the first towing trial of the Shark, the issue with the directional stability of the Shark made measuring impossible. Once the hull gains only minor speed, the stern slides and the nose grips tight all the way to the tip of the bow, leaving no lever for the tugboat to straighten the course by increasing power. This effect is known from the *Literature Review* and demonstrated in the simulations. Due to this issue, the Shark could not be used in an isolated hull test, highlighting the need for derivation in Figure 5.3. Adding the strut is the most convenient way to keep the hull aligned. The Torpedo has a lifted bow keel curve and cutting / gripping hull (draft) which is more evenly distributed over its length, promoting a natural directional stability defined by its geometry. During towing, the Shark also shows nose diving behavior, similar to the trim reduction found in full-range CFD analysis.

5.1.6. Discussion

In this section, the observation of effects observed in the results in Figure 5.3 are discussed.

Planing Regime Trend

For both the Torpedo and Shark, the displacement regime shows exponential curves up to the hump observed at 18 km/h for the Shark and 16 km/h for the Torpedo. In the planing regime, the Torpedo resets the exponential curve and propagates along a renewed exponential trend, including a slightly flattened effect in high regimes. This is a typical observation in semi-planing hulls. The Shark propagates from the less pronounced hump into a quasi-linear line, indicating its planing capabilities, despite pushing its bow down while the stern is dynamically lifted.

Hump

The displacement-to-dynamic lift transition of the Torpedo is clearly visible. For the Shark, the hump is less pronounced as the midship and stern smoothly transitions to planing. It is to be expected that the Shark — being slightly longer — features a hump around 1.44% (speed percentile) later than the Torpedo. However, the gap is wider with its 2 km/h or 12% delay.

Usually, a naval architect aims for a hump delay, effectively expanding its displacement domain and often smoothing the hump. Remarkably, the hump of both hulls is delayed (see Figure 5.3) compared to the theoretical hump speed of 13.8 km/h for the Torpedo and 14.0 km/h for the Shark. This delay indicates that the increase in total resistance occurred at higher Froude numbers than theoretically expected. This phenomenon is well known in ship hydrodynamics and primarily relates to the complex interaction between viscous and wave-making resistance components. The following mechanisms may contribute to such a delay.

- **Hull geometry** — A slender hull form or fine entry can postpone the onset of significant wave-making. The smoother flow alignment reduces the amplitude of the bow and stern waves at low to moderate speeds, thereby delaying the increase in wave-making resistance. This behavior is typical for hulls with a low beam-to-length ratio as featured by both hulls.

- **Dynamic trim and sinkage effects** — At higher speeds, dynamic pressure can alter the trim and sinkage of the hull. A small bow-up or bow-down motion may change the effective waterline length, which in turn shifts the Froude number at which resonance between the bow and stern wave systems occurs. Such dynamic effects can therefore postpone or advance the hump, depending on the direction of trim change. This is not applicable to the vertical piercing bows of both hulls (only slightly for the Torpedo). Note, if these effects are significant, one should consider to use displacement based Froude numbers.
- **Partial dynamic lift** — For semi-displacement or semi-planing hulls, partial hydrodynamic lift begins to support a fraction of the vessel's mass as speed increases. This reduces wave amplitude and the associated wave-making resistance, resulting in a flatter or delayed hump in the total-resistance curve. This is probably the second largest effect in the hump-delay of the Shark, indicating the reason for the enlarged gap between the Shark and Torpedo's hump.
- **Experimental and setup influences** — The towing arrangement can also affect the apparent position of the hump. Variations in towline height or angle introduce vertical forces that modify running trim. Additionally, unsteady carriage motion, throttle oscillations, or wave reflections in the tank may distort resistance readings, particularly around transitional speeds. This is mitigated as much as possible by being aware of the limitations and trying to collect as much data as practically manageable.
- **Hull-appendage interaction** — Appendages such as struts, keels, and shafts modify the local pressure field and stern-wave pattern. These interactions can suppress or delay the hull's natural wave system, shifting the hump or smoothing its amplitude. This is not conclusive in the results. Also, in the Shark's strutless derivation, this cannot be accounted for.

- **Scaling effects** — When comparing model- and full-scale data, differences in Reynolds number may alter the balance between frictional and residual resistance. At low Reynolds numbers, the higher relative contribution of viscous drag can mask the onset of wave-making effects, making the hump appear at higher speeds in the scaled data. Obviously this is not applicable to these tests, but worth mentioning for the sake of completeness.

In summary, a delayed resistance hump generally arises from a combination of hydrodynamic and experimental factors. Hulls with fine entries or early hydrodynamic lift tend to show a delayed onset of wave-making resistance, while dynamic trim and appendage interference can further influence the speed at which the hump develops. In the present study, the delayed hump observed for the Shark configuration is likely associated with a combination of early lift generation and the fine entry, which is cause for the delay in the Torpedo too.

Strut Influence

Given the Shark's inability to operate without a strut, it was imperative to ascertain the resistance of the strut itself and compensate for it on the Shark's hull. At 30 km/h, the strut accounts for approximately 30% of the total drag of the Shark. For the Torpedo at 25 km/h, the strut accounts for approximately 17% of the total drag. Estimating the influence at 30 km/h is less accurate for the Torpedo as no scatter data is available and partial (above hump) data extrapolation indicates higher total resistance values compared to the 4th graph (Torpedo with short strut plot) in Figure 5.3. The theoretical quadratic relation of the strut drag is shown in Figure 5.3 and was already discussed in Section 5.1.5. The Matlab script can be found in Appendix C.7.5 along with the rest of the data analysis code.

Simplifications

- Shallow water effects are ignored. For 6 m and a max speed of 28 km/h (= 7.78 m/s), using Equation 5.4 and 5.2, the shallow water critical depth and critical Froude number is

$$h_c = \frac{V^2}{g} = \frac{7.78^2}{9.81} \approx 6.17 \text{ m/s} \quad (5.1)$$

being in a transition zone of $0.8 < Fr_h < 1.0$ while in higher regimes and only supercritical ($1 < Fr_h \approx 1.014$) at the maximum speeds reached only by the Shark. For $Fr_h > 1$, the resistance increases (mostly for wavemaking effects).

- Due to wave jumping of the knot (middle of the towing line), fluctuations show in Figure 5.4. The resistance drops at 25 and 35 km/h are showing due to the tensioning of the rope at higher speeds and alignment of the knot with the trough of the wake waves of the tugboat. These line resistance results were later classified as insignificant as the rope under tension (during towing) lifts free of the water ($> 200 \text{ N}$). This

is acceptable as we are not interested in the ultimate accuracy of lower speed regimes.

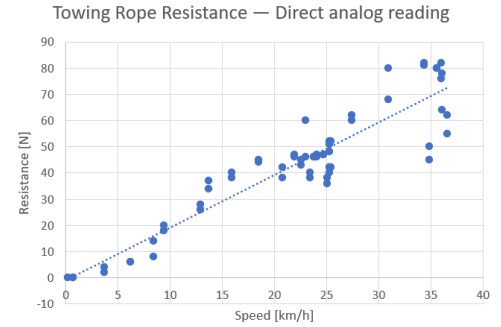


Figure 5.4.: Towing rope resistance

- The trim, heave and yaw effects may have been influenced by the bow tug configuration. However, due to the significant length of the rope, forces are damped and deviating towing force angles are limited. Also, the trim expected for both hull types is rather small for a flat water test, mainly due to their wave piercing bow.
- Waves and wind are ignored as they were reported insignificant at the time of measurement.

5.1.7. Conclusion

The full-scale towing experiments provided a coherent and reproducible resistance dataset for the Torpedo and Shark configurations under documented environmental conditions (see Table 5.1). Despite the Xsens failure, the pre-planned backup ensured sufficient speed accuracy, and the combined analog–digital force acquisition — together with the pulley arrangement — extended the usable range and sampling density of the measurements. The data-processing pipeline (see Section 5.1.5) effectively removed artifacts (device “hold” and acceleration spikes), reduced scatter through equal-count grouping, and yielded smooth, non-negative resistance curves suitable for subsequent analysis and comparison (see Figure 5.3).

Physically, both hulls exhibited the characteristic displacement-to-dynamic-lift transition, with a delayed hump relative to first-order expectations. For the Shark, the hump appears later and less pronounced, consistent with early partial lift and a fine-entry geometry that moderates wave-making at intermediate speeds. Directional stability constraints necessitated the use of a strut on the Shark; accordingly, a strut-only resistance

curve was derived and modeled with a quadratic fit, enabling the reconstruction of the Shark’s strutless resistance. From this decomposition, a theoretical drag constant for the strut was obtained in the form SC_f via $D_f = \frac{1}{2}\rho_w V^2 SC_f$ (with $\rho_w = 1026 \text{ kg/m}^3$), providing an interpretable metric for appendage drag across the tested range.

The present results establish a transparent baseline for the hydrodynamic behavior of both hulls at full scale and supply processed inputs for cross-comparison. Limitations remain — most notably the inability to test the Shark without appendages in isolation, potential residual sensitivity to towing setup, and the inherent coupling of viscous and wave-making components in open-water measurements. These aspects, together with flow-field diagnostics, will be addressed by the validation steps in the subsequent sections: towing-tank model tests, qualitative wind-tunnel observations, RC-scale manoeuvring trials, and correlation against simulation and theoretical estimates. Collectively, these will indicate uncertainties, confirm (or refute) observed trends, and consolidate the final resistance models for both hulls.

5.2. Wind Tunnel Testing Results

A qualitative wind tunnel evaluation was conducted to assess the flow behavior around the Shark’s bow foils and appendages, with particular emphasis on wingtip vortex formation and potential flow disturbances. The primary objective of this experiment was to verify whether the simulated flow structures correspond to physically observable patterns and to identify any indications of unstable or unpredictable aerodynamic behavior.

To this end, flow visualization techniques based on laser–particle interaction were employed to examine sectional flow fields in the vicinity of the bow foils. The length, coherence, and expansion of the wingtip vortex cone were evaluated while systematically varying the angle of attack. A particular emphasis was placed on the initial formation and subsequent evolution of vortex structures, along with their interaction with proximate appendages.

Across the tested operating range, the observed vortex topology and growth trends exhibited qualitative agreement with the steady-state simulation results. The wingtip vortices demonstrated a consistent and predictable expansion downstream, with no abrupt breakdown, oscillatory behavior, or asymmetric development detected. Additionally, no substantial flow separation or unanticipated appendage-induced disturbances were observed within the investigated angles of attack due to the smooth CAD design following the original hull shape.

While the wind tunnel evaluation was not designed to provide quantitative validation, the qualitative correspondence between experimental flow patterns and simulated laser–particle interaction section cuts supports the physical plausibility and predictability of the numerical

model. These findings provide substantial evidence that the simulations effectively capture the predominant aerodynamic mechanisms governing the bow foil behavior under the tested conditions and that the foils indeed show a very forgiving geometry as originally intended.



Figure 5.5.: Bow foil wing tip vortices

5.2.1. Particle Image Velocimetry

Results (via PIVlab) are not discussed due to time constraints. As already highlighted, mainly qualitative analysis was performed (recirculation zones, vortices, etc.).

5.3. Scale Model Experiments

In the maritime racing industry, remote control (RC) modeling has proven to support quick and low-cost development. Scale model testing will allow for real-world validation of adaptations and current performance of base designs. The test includes a simple spontaneous deceleration analysis, tow tank resistance test according to the ITTC guidelines and a seakeeping / stability experiment mapping seaworthiness under various wave conditions.

5.3.1. Evaluation of Relevant Previous Research Experience

In light of the findings from prior projects, the succeeding experiments are intended to serve as a foundation for the discourse on hull shape, with the objective of bridging the gap between theoretical principles and practical applications. The utilization of a diverse set of hull types is expected to provide support for the study's theoretical framework, thereby enhancing the comprehension of naval architecture.

UASBT Test Platform



Figure 5.6.: UASBT test platform 2024-25 during autonomy tests

The first of these analysis row was conducted on the geometry of the UASBT autonomous system test platform of last academic year. Here, the basic flow theory discussion on length of a vessel and trailing wake effects (transom wrongly installed on a displacement hull) are extended and illustrated by simplified single-fluid CFD simulation (non-free-surface, computational domain limited to the water body). This approach can only produce usable results at low speeds, far below hump speed. It cannot be used as an accurate analysis as the wavemaking is not properly modeled due to the absence of the free surface. This

has also effect on the wetted surface and pressure distribution near the waterplane.

However, the simulation demonstrates the advantageous outcomes of augmenting the length of the catamaran test platform. The findings indicate that the currently used wide demi-hulls demonstrate an elevated degree of resistance to bow pressure, a consequence of the aggressive flow deflection (see Section C.1.14). The short length (approximately 1 m) causes the hull speed to be very low, namely 1.4 m/s or 2.74 kn. Due to the large bow wave and limited power, the vessel cannot surpass hump speed. Easing flow de-

flection and making the hulls generally longer increases hull speed, decreases wavemaking resistance, and reduces wake turbulence and therefore increase energy regeneration at the stern. Despite the incorporation of a transom in the design, this study advises against its implementation due to the vessel's low cruising speed.

Note that the setup is simplified by expecting 50% sponson submersion (static balanced buoyancy) for every run and simplified thruster-rudder geometry is included. As a result, the longer configurations also estimate more load carrying capacity, promoting the drag decrease even more. The configurations are listed in Figure C.42 of Appendix C.2.

Run	Description	Speed @ 19,5N drag [m/s]	Drag @ 1 m/s [N]	Speed advantage vs default setup	Cause
0	Default (bow 20 cm)	1.00 (ref.)	19,69	-	-
1	Add tail stern (30 cm)	1.27	11.96	27%	Reduce drag by reducing rear turbulence.
2	Long bow (20 to 60 cm)	1.30	11.5	30%	Lowered nose pressure.
3	Tail stern (30 cm) & long tip (20 to 60 cm)	1.70	6.81	70%	Reducing rear turbulence & lowered nose pressure.
4	Long tip and tail (both 60 cm, cutting edge 5 mm)	1.82	5.80	82%	Reducing rear turbulence & lowered nose pressure.

Table 5.3.: Results of the basic limited-domain CFD simulation

For every configuration, the Froude number is calculated at the characteristic speed, reaching a drag force of 19.5 N. The results of this calculation in Table 5.5 indicate that no value is near the threshold of $Fr = 0.45$. So, ignoring the limitations described earlier in this section, the speed advantage can be considered a valid outcome. In the table, “OK” means that the result is valid, but indicates magnified wavemaking causing inaccuracy due to the transcendence over $Fr = 0.4$, approaching the hump and relying more on the resistance effects modeled by free-surface CFD.

Test	Bow [m]	Midship [m]	Stern [m]	L_{WL}	Speed [m/s]	Fr	Valid
0	0,2	0,8	0	1	1	0,319	Yes
1	0,2	0,8	0,3	1,3	1,27	0,356	Yes
2	0,6	0,8	0	1,4	1,30	0,351	Yes
3	0,6	0,8	0,3	1,7	1,70	0,416	OK
4	0,6	0,8	0,6	2	1,82	0,411	OK

Table 5.5.: Displacement regime validity for the UASBT test platform 2024-25 CFD simulations

Foil Research Platform

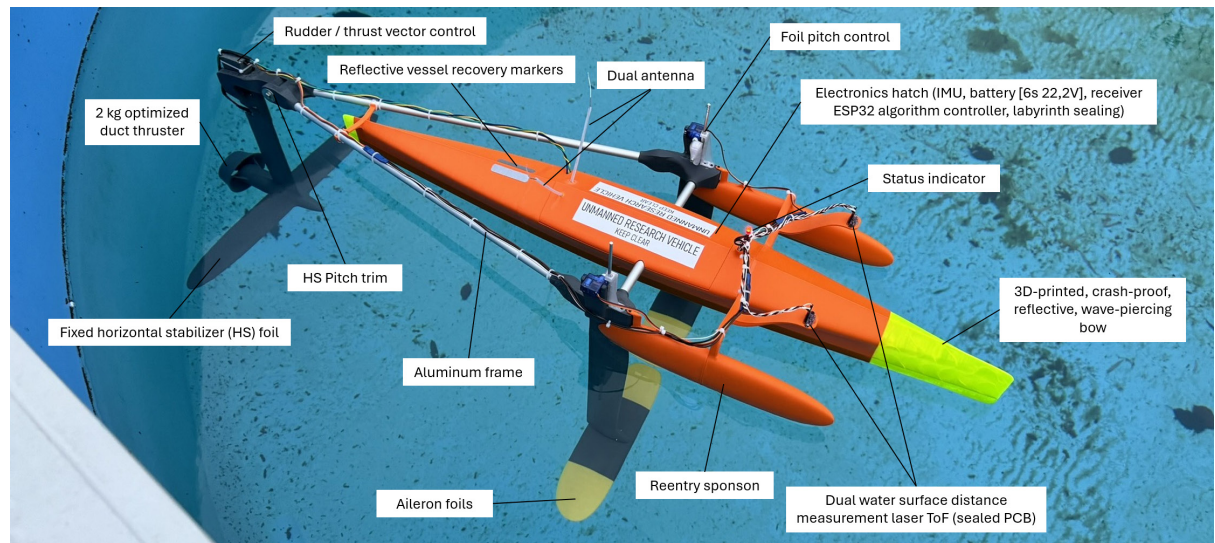


Figure 5.7.: Foil research platform

The foiling test platform was not constructed to have an optimal foiling configuration, but build to mimic the active foil configuration of the Torpedo MK.II. The study proved a suspicion stated: the foils of the Torpedo are too far forward (CG way back of the front foils) to achieve stable foiling. What also can be derived from subsonic aerospace engineering principles is the unstable pitch control: because the setup relies on a fixed, very large, pitched up horizontal stabilizer, it is not capable of controlling the foil angle of attack properly. As a consequence, the front foils have to compensate for that extra heave/pitch balance, often leading to the failure of one of the two aileron foils. Moving the CG forward helps to achieve a stable foiling craft as demonstrated by the test platform. However, as has been noted during trials, the variable speed requires different pitch compensation input as the horizontal stabilizer generates different lift forces at different speeds as long as the CG is not on top of the center of lift (CL) of the aileron foils. So, a variable horizontal stabilizer is more optimal and makes it easier to control the craft. However, experiments have suggested that a symmetrical, smaller horizontal stabilizer also improves control drastically, provided that the CG is corrected.

Another parameter on foil configurations that makes the project goals more manageable is the

horizontal distance in between the aileron foils. Placing them further apart would increase roll controllability due to the decrease in roll sensitivity and lowering of the relative metacenter. On the other hand, tight cornering is limited due to the need of foils to stay submerged (keep predictable lift). Widening the setup limits the allowable roll before reaching the foil ventilation criterion. Stable cornering on-foil is even a bigger challenge as demonstrated by multiple teams across the MEBC. This requires meticulous analysis of aerospace principles for airplane control surfaces. Within the domain, a rudder turning horizontal stabilizer as currently fitted is a questionable design due to the reduced and re-positioned lift vectors when angled (HS-foil width might be the main disadvantage, see Figure 5.7), inducing a disturbance in the aileron control algorithm due to geometry changes and generating increased lift on one side and decreased lift on the other side while changing the rudder's angle setpoint.

In essence, this configuration might not be the most optimal. However, various experiments across the field have proved this configuration has the potential working in corners with widely spaced aileron foils. Due to the wing-shaped front struts, cornering slip is minimized and cornering without roll is made possible, with increased vertical foil forces — to prevent falling

over — as a consequence. The increase is limited by widening the strut spacing as the generated reversed roll moment is the same for a specific cornering radius. Over-dimensioned foils are mandatory in order to prevent stall and cavitation, as implemented in this test platform.

The slightly over-sized aileron's surface of the test platform ($\approx 30\%$) reduces the susceptibility to cavitation and foil-stall as mentioned above, with slightly more drag as a consequence. High aspect-ratio foils or wings, leads to increased efficiency due to decreased induced drag, limiting the tolerance for operational variations as a consequence. Another, more experimentally investigated (e.g., by NASA) technique to reduce induced drag is the implementation of a Prandtl-wing design, which by definition slightly inverts the lift towards the tips of a wing, effectively reducing the vortices generated by the difference in pressure under and above the wing tip. However, to keep the platform compact and because efficiency was not a major design objective in this test craft, the aspect ratio is rather limited and no advanced wing optimizations were applied. The only two techniques used to reduce induced drag are:

1. More or less elliptical lift distribution over the wings [77]. This limits the size of trailing vortices at the wingtips.
2. The aileron foils nearly touch in the middle of the craft. If vortices are formed, they will cancel out each-other as observed in CFD. This was inspired by the flexible wing concept by the Swedish foiling boat company Candela.

That being said, the main discussion should be about the design of the hull. In general, the 2024 research platform (foil demo) — build for control algorithm tests, detailed analysis and simulation — is not actively part of this study but serves as great example of narrow or high L/B hull design: slender, near vertical freeboard for flat water efficiency, featuring a wider flared top freeboard to reduce deep submersion and therefore limit greenwater on the electronics hatch. The cambered deck deflects greenwater when encountering a wave or during a nosedive maneuver. The sponsons add trimaran-type stability, but quickly lift free of the water once the speed increases (at a lower Froude number than the Torpedo). They mainly serve an important role during tilted water reentry or pitch-down landings.

A sharp, vertical bow does not prevent greenwater nor encourages the nose to pitch up by hydrodynamic pressure (only by the limited increase in buoyancy), and is similar to the Torpedo's and Shark's bow. The choice for a slender displacement hull showing minimal wave-making resistance and wake is obvious: only in low-speed regime, sub-hump, the hull touches the water. Once accelerated to a characteristic speed (approaching hump), the boat is lifted and leaves the water, quickly accelerating to cruising foil speed. The sharp nose is selected based on the deceleration water reentry sequence expected. As noted during experiments, a slightly forward tilted bow with more overhang and some more camber, might have been more optimal. The hull tends to dive if the reentry is too steep. This can happen if the power is cut abruptly and the craft was high-on-foil.

Functional, Autonomous, Displacement Model of the MCM M940 Oostende

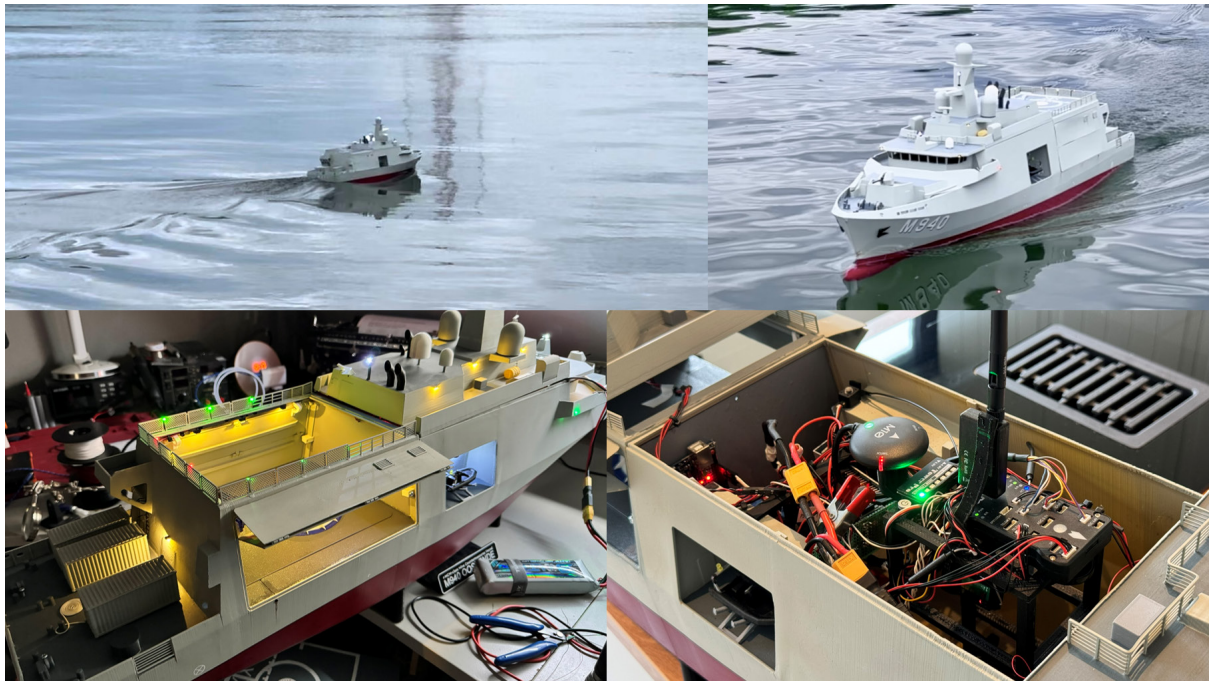


Figure 5.8.: Mine countermeasures vessel M940 autonomous scale model

The M940 Oostende model has been build and tested as an autonomous test platform in previous research and is a great example of a large, modern, displacement-type hull. The geometry is designed to operate efficiently at wavelength $\lambda = 114 - 171$ cm at scale (see Figure 5.8), reaching optimal hull speed with excellent yaw (directional) stability (in contrast with the Shark), confirmed by sea trials and short tank tests. At sea trial experiments, the 1.14x0.22 m model ($L/B = 5.18$) sails 1.36 m/s only requiring 1.93 A at 8.36 V (16.13 W) — according to the autopilot — with a displacement of around 6 kg. The referenced speed and length indicate an S/L-ratio of 1.37 ($\text{kn}/\sqrt{ft}$) equivalent to $\text{Fn} \approx 0.406$, which is clearly right in front of the resistance hump: the last efficient displacement speed.

Due to a high CG and wind-catching superstructure, the hull suffered from excessive roll oscillations at higher sea state. Overall, the rectangular midship cross-section with near vertical freeboard and straight bow overhang ensures increased seakeeping and proved to never suffer from greenwater in all tested conditions. The integration of a well-dimensioned gyroscopic sta-

bilizer accelerated to over 8000 rpm or approximately 133 Hz proves to be very effective reducing the roll amplitude with more than 75-80% in high sea states.

Observe the high bow wave at maximum speed in Figure 5.8, which suggests a revision of the bulbous bow geometry according to the description in Section 2.3.5.

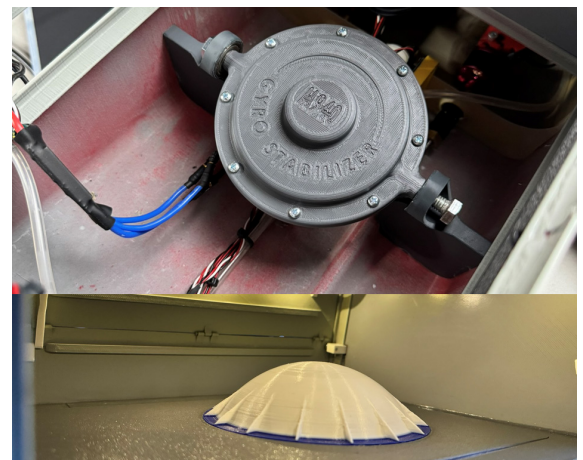


Figure 5.9.: M940 scale model gyroscopic stabilizer

As a full-scale theory validation for the efficient displacement speed, a simple Froude calculation was conducted during the visit of the F931 Louise-Marie (see Figure C.56), an operational Belgian multi-role frigate stationed in Zeebrugge Naval Base. Build by Damen Schelde Naval Shipbuilding, its length stretches up to 122 m and equivalent to approximately 120 m L_{WL} , with a beam of 14.4 m, and a draft of 6.2 m. The $L/B \approx 10.5$ indicates its operational capabilities and underlines its intended design: a displacement hull featuring limited wavemaking for high-speed military operations, providing security for aircraft carriers or fleets by sub-surface, surface and aerial defense mechanisms.

Given: Froude equation 2.3, $g = 9.81 \text{ m/s}^2$, $L_{WL} = 120 \text{ m}$

$$\sqrt{gL} = \sqrt{9.81 \times 120} = \sqrt{1177.2} = 34.32$$

$$V = Fn \times \sqrt{gL}$$

$$\text{For } Fn = 0.40: \quad V = 0.40 \times 34.32 = 13.73 \text{ m/s}$$

$$\text{For } Fn = 0.45: \quad V = 0.45 \times 34.32 = 15.45 \text{ m/s}$$

$$1 \text{ m/s} = 1.94384 \text{ kn}$$

$$\text{For } Fn = 0.40: \quad V = 13.73 \times 1.94384 = 26.7 \text{ kn}$$

$$\text{For } Fn = 0.45: \quad V = 15.45 \times 1.94384 = 30.0 \text{ kn}$$

$Fn = 0.40 \Rightarrow V = 13.7 \text{ m/s} = 26.7 \text{ knots}$ $Fn = 0.45 \Rightarrow V = 15.5 \text{ m/s} = 30.0 \text{ knots}$
--

Froude number calculations indicate that the last efficient speed or so called hull speed is 26.1 kn, with the resistance hump located at directly above 30 kn. The 30 kn confirmed maximum operation speed of the frigate indicates a perfectly matched propeller-motor configuration, keeping the vessel in efficient displacement regime at all time and maxing-out that speed range. This allows for fast maneuvering and quick acceleration, while being economically feasible for its size. The slightly flared, sharp bow and rectangular midship cross-section suggests good seakeeping in high sea states.

5.3.2. First Test — Indicative Comparison

The first model-on-water experiment implies an accelerated vessel that starts to decelerate naturally by water friction the moment the experiment begins. The deceleration is a measure for the resistance acted on the model for each moment in time. It is important to note that this is a very simple indicative test (spontaneous body deceleration) measured based on high-speed camera footage recorded from the bottom (1.3 m) of the freshwater test tank (2.6×7 m). This was the very first experiment on the hulls in of this study and will only indicate gross deviations in hull performance.

Expectations

It is to be expected that the Shark will suffer from increased surface friction due to the large wetted area. Planing behavior of this type of hull does not scale accurately using length-based Froude similarity. Based on the slight difference in mass (see Table 5.6), the Torpedo Mk.II EVO has a limited advantage in slower deceleration due to its larger inertia. However, they are still comparable. It can be predicted that the Torpedo will most probably suffer from wavemaking and that the Shark's deceleration is dominated by surface friction.

Limitations for Quick Comparisons

Presuming Froude scaling is inaccurate transferring data to full-scale (an overly simplistic statement), it is still a valuable tool comparing hulls at the same scale. One should note that physics as displacement scales cubically with volume and surface friction scales quadratically (see Section 2.6.2). Therefore, a 1/10 scale indicates a displacement of 1/1000th and the surface friction 1/100th. This implies that the surface friction has a significantly larger effect on the model than it does on the full-scale prototype. That should be kept in mind during evaluation or using correction formulas.

Results

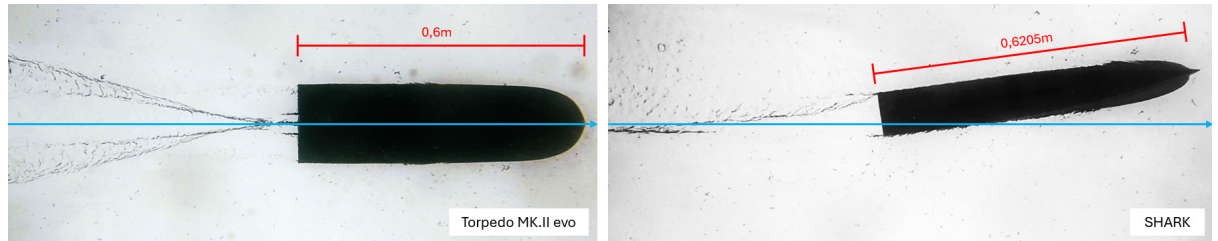


Figure 5.10.: Torpedo Mk.II EVO and Shark during tests

Parameter	Torpedo Mk.II EVO	Shark
Model CG (measured from the stern) [m]	0.2000	0.2300
Model Mass [kg]	0.312	0.273
Model Width [m]	0.1690	0.0965
Model Length [m]	0.6000	0.6205
Model Start Speed [m/s]	3.537	3.078
Scaled CG (measured from the stern) [m]	2.000	2.300
Scaled Mass [kg]	312	273
Scaled Width [m]	1.690	0.965
Scaled Length [m]	6.000	6.205
Scaled Start Speed (Froude similarity) [m/s]	11.1845	9.7325
Scale of the Model	1/10	1/10
Tank/Environmental Temp. [°C]	18/23	18/23

Table 5.6.: Parameters of the first test

During testing, it was observed that the Shark’s flat-bottom hull tended to turn to one side after release (note that no drive shaft housing is mounted, potentially limiting the lateral sliding behavior as demonstrated in full-scale experiments). This is in line with the experience stated by pilots navigating the vessel in previous competitions. This obviously influences the deceleration curve. The theory behind this phenomenon is simple to explain. The bow — a sharp displacement configuration — grips laterally in the surrounding mass of water gently pushing it aside (it has to pick a side). Be-

cause the stern is “free” to slide and the center of mass is the driving force pushing against all drag forces an undesirable maneuver occurs. As the boat keeps its momentum, the CG will push and — due to an unstable singularity — the bow aside trying to overtake the most dragging parts of the hull (momentarily: the bow). This increases the pressure on one side of the bow, pushing it further aside until all forces are balanced and the boat rolls and follows a steady turning rate (assuming the speed would be constant and the rolling hull grips a certain curved flow path).

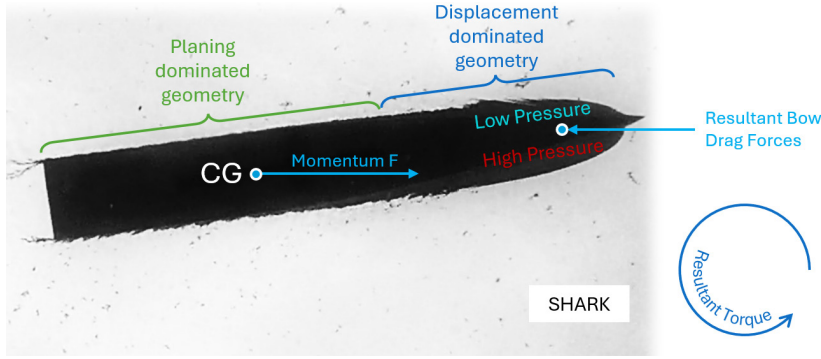


Figure 5.11.: Shark turning torque during first test

Note that the chosen operating range of 5-10 m/s ($10 \text{ m/s} = 36 \text{ km/h} \approx 19.4384 \text{ kt}$) for a full-scale prototype is relevant as this is the transition zone between the currently used speeds of the UASBT and the high-speed range that the team is aiming for in the near future.

Shallow-water critical depth and test validity Before reviewing the results, it is mandatory to check whether they are valid for the test tank conditions used. Therefore, the shallow-water critical depth should be checked. Too shallow water increases resistance disturbing test results.

Definition Shallow-water effects are governed by the depth Froude number:

$$Fr_h = \frac{V}{\sqrt{gh}}, \quad (5.2)$$

with vessel speed V , gravitational acceleration $g = 9.81 \text{ m/s}^2$, and water depth h . The critical shallow-water condition occurs when

$$Fr_h = 1 \iff V = \sqrt{gh}. \quad (5.3)$$

Equivalently, for a given speed V , the critical depth is

$$h_c = \frac{V^2}{g}. \quad (5.4)$$

For $Fr_h < 1$ (subcritical), long gravity waves propagate faster than the vessel and shallow-water “critical” amplification is avoided; for $Fr_h > 1$ (supercritical), the flow is dominated by a divergent wave pattern and strong depth effects.

Torpedo Mk.II EVO — $L_{WL} = 0.60 \text{ m}$, $V_{\max} = 3.537 \text{ m/s}$ Critical depth at this speed:

$$h_c = \frac{(3.537)^2}{9.81} \approx 1.275 \text{ m}. \quad (5.5)$$

With the tank depth $h = 1.30 \text{ m}$,

$$Fr_h = \frac{3.537}{\sqrt{9.81 \times 1.30}} \approx 0.990 < 1, \quad (5.6)$$

leaving a margin of $h - h_c \approx 0.025 \text{ m}$. The test is therefore subcritical and formally valid with respect to shallow-water criticality, but it operates very close to the critical point where depth effects (e.g., increased wave resistance) can be significantly amplified. This proximity should be noted when interpreting resistance peaks near top speed.

Shark — $L_{WL} = 0.6205\text{ m}$ ($\approx 0.55\text{ m}$ at speed), $V_{\max} = 3.078\text{ m/s}$ Critical depth at this speed:

$$h_c = \frac{(3.078)^2}{9.81} \approx 0.966\text{ m} \quad (5.7)$$

With the same tank depth $h = 1.30\text{ m}$,

$$Fr_h = \frac{3.078}{\sqrt{9.81 \times 1.30}} \approx 0.862 < 1 \quad (5.8)$$

leaving a comfortable margin of $h - h_c \approx 0.334\text{ m}$. This run is clearly subcritical with respect to depth and therefore valid from a shallow-water perspective.

Remarks on length-based Froude numbers For context, the length-based Froude numbers at top speed are well into the planing regime:

$$\begin{array}{ll} Fr_L = \frac{3.537}{\sqrt{9.81 \times 0.60}} \approx 1.458 & (V=3.537, L=0.60) \\ \hline Fr_L \approx 1.248 & (V = 3.078\text{ m/s}, L = 0.6205\text{ m}) \\ Fr_L \approx 1.325 & (V = 3.078\text{ m/s}, L = 0.55\text{ m}) \end{array}$$

These values confirm that any resistance peaks observed near maximum speed are not the part of the classical displacement “hump” (typically $Fr_L \approx 0.4 - 0.5$), and interpretations should focus on planing/trim transitions, ventilation, and interference effects rather than wave-making hump relocation.

Conclusion on validity Both tests are conducted below the shallow-water critical condition ($Fr_h < 1$), hence they are valid with respect to shallow-water criticality. The Torpedo Mk.II EVO test is near-critical and may exhibit noticeable depth-induced amplifications near the maximum test speed; The Shark model test is comfortably subcritical. Any analysis of resistance or stability near the top speeds should account for this difference in proximity to $Fr_h = 1$. However, this is not the goal of this test.

Position & Magnitude of the Hump In the Torpedo Mk.II EVO model, a hump speed could not be observed, although not very conclusive in this test. This is as expected because slender hulls do not show a high hump magnitude. The exponential growth shown indicates that the hulls is still characteristically behaving as a displacement vessel as has been shown by the BCD-software and accompanied formulas (including Froude). The linearity shown at high speeds in the Shark model test reflects the same effect observed in BCD using planing formulas (including Crouch). Additionally, the presence of a hump speed is not clearly discernible. The step in the graph might potentially be part of a displaced hump, better visible in following experiments. The position of the whole speed is appropriately scaled by Froude similarity. However, the magnitude is not, since vicious effects are governed by the Reynolds number. Both Froude and Reynolds scaling cannot match in the model tests. Above hump speeds (importance of dynamic lift and spray formation) Froude scaling must be corrected by scaling corrections or CFD as stated by the ITTC, especially for planing hulls.

Using Froude equation 2.3, the hump speed can be calculated:

The classical wave-making “hump” occurs around a length-based Froude number

$$Fr \approx 0.4 - 0.5, \quad Fr = \frac{V}{\sqrt{gL}} \Rightarrow V_{hump} = Fr \sqrt{gL}. \quad (5.9)$$

For Froude similarity with scale factor $\lambda = L_{\text{proto}}/L_{\text{model}}$,

$$V_{\text{proto}} = V_{\text{model}} \sqrt{\lambda}. \quad (5.10)$$

Here $g = 9.81\text{ m/s}^2$ and $1\text{ m/s} = 1.9438\text{ kn}$.

Hump speed estimation: model vs. prototype The classical wave-making “hump” occurs around a length-based Froude number.

$$Fr \approx 0.4-0.5, \quad Fr = \frac{V}{\sqrt{gL}} \Rightarrow V_{\text{hump}} = Fr \sqrt{gL}. \quad (5.11)$$

For Froude similarity with scale factor $\lambda = L_{\text{proto}}/L_{\text{model}} = 10$,

$$V_{\text{proto}} = V_{\text{model}} \sqrt{\lambda}. \quad (5.12)$$

Here $g = 9.81 \text{ m/s}^2$ and $1 \text{ m/s} = 1.944 \text{ kn}$. Note that the input lengths L are those of the *model* (1/10 scale), while the *prototype* results are extrapolated from the model values using Froude similarity.

Torpedo Mk.II EVO (model $L = 0.60 \text{ m}$)

$$\sqrt{gL} = \sqrt{9.81 \times 0.60} = 2.427 \text{ m/s}.$$

Model hump speeds (theoretical):

$$Fr = 0.40 : 0.970 \text{ m/s (1.89 kn)}, \quad Fr = 0.45 : 1.092 \text{ m/s (2.12 kn)}, \quad Fr = 0.50 : 1.214 \text{ m/s (2.36 kn)}$$

Prototype hump speeds ($\lambda = 10$, $\sqrt{\lambda} = 3.162$) (theoretical):

$$Fr = 0.40 : 3.069 \text{ m/s (5.97 kn)}, \quad Fr = 0.45 : 3.452 \text{ m/s (6.71 kn)}, \quad Fr = 0.50 : 3.836 \text{ m/s (7.46 kn)}$$

Shark (model $L = 0.6205 \text{ m}$)

$$\sqrt{gL} = \sqrt{9.81 \times 0.6205} = 2.467 \text{ m/s}.$$

Model hump speeds (theoretical):

$$Fr = 0.40 : 0.987 \text{ m/s (1.92 kn)}, \quad Fr = 0.45 : 1.110 \text{ m/s (2.16 kn)}, \quad Fr = 0.50 : 1.234 \text{ m/s (2.40 kn)}$$

Prototype hump speeds ($\lambda = 10$) (theoretical):

$$Fr = 0.40 : 3.121 \text{ m/s (6.07 kn)}, \quad Fr = 0.45 : 3.511 \text{ m/s (6.83 kn)}, \quad Fr = 0.50 : 3.901 \text{ m/s (7.58 kn)}$$

This means that it is not possible to prove the effects of the hump in one of the tests as they are conducted above 1.617 m/s (Torpedo Mk.II EVO) and 1.588 m/s (Shark) respectively.

Notes

- The range $Fr = 0.4 - 0.5$ brackets the classical hump; $Fr = 0.45$ is a practical nominal value.
- If effective waterline length shortens at speed (trim/planing), the corresponding hump-speed estimates increase or decrease with $\sqrt{L}$ accordingly.

Direct Comparison in Drag Performance As previously emphasized, the deceleration can be utilized as a metric for resistance, akin to the slope of the curve. Using linear regression for a similar speed range ($1.6-3.05 \text{ m/s}$ model speed), the slope can be calculated. The steeper curve — a lower negative value — indicates higher resistance in this range. This is remarkable as there are numerous extra factors increasing the resistance for the Shark compared to the Torpedo model:

- Sliding sideways increases wavemaking and form drag.
- The wetted area of the Shark is around 20-40% larger than the Torpedo (for the used mass and CG). As demonstrated in Section 5.3.2, this issue is of considerable importance. The corrected value should indicate even less resistance than the Torpedo at full-scale.
- The slightly lighter Shark model has less momentum (driving force to continue motion). Note this is more complex to label as an advantage in evaluating deceleration as the form drag also will reduce due to the weight difference.

The deceleration for a similar speed range (1.6-2.7 m/s model speed) can be averaged to gain the same insight. Note the accuracy of these acceleration values is less accurate due to double integration of the measured position values. Changes in speed are exited. All results can be found in Appendix C.3. Note that the resolution is provided in order to present a measure of accuracy in these tests.

5.3.3. Tow Tank Resistance Test

Towing tank experiments were conducted on geometrically similar scale models at a scale ratio of 1/10, corresponding to a static displacement scaling factor of 1/1000 (1/10 length scale). The tests were performed in the same towing basin as used for the preliminary free deceleration experiments (Section 5.3.2), and therefore the depth influence analysis remains identical. The limited basin length of 7.6 m necessitated the execution of sequential steady-state towing tests, rather than extended transient measurements.

Model tests were performed under specific environmental conditions. Water temperature and general basin conditions were monitored throughout the test campaign to ensure repeatability in future research. The detailed input configuration of the test models, including principal dimensions, displacement, and environmental parameters, is documented in the project portfolio and is not repeated here for brevity.

Three geometrically similar models were tested, each representing a distinct hull configuration within the investigated design space. The models were manufactured to comparable tolerances and prepared using the same surface finishing procedure to minimize variability in frictional resistance. It is noteworthy that the Shark v4 model exhibited a marginally rougher surface, a consequence of time constraints, yet this deviation remained within acceptable limits.

The three models tested are:

1. Default Torpedo
2. Default Shark
3. Upgraded Shark (v4)

Setup and Instrumentation

Each model was mounted on a the towing rig equipped with a calibrated force balance. Prior to testing, the balance was zeroed and verified under static load conditions. The models were carefully aligned to ensure straight-ahead motion and free trim and sinkage during towing, as prescribed by the ITTC.

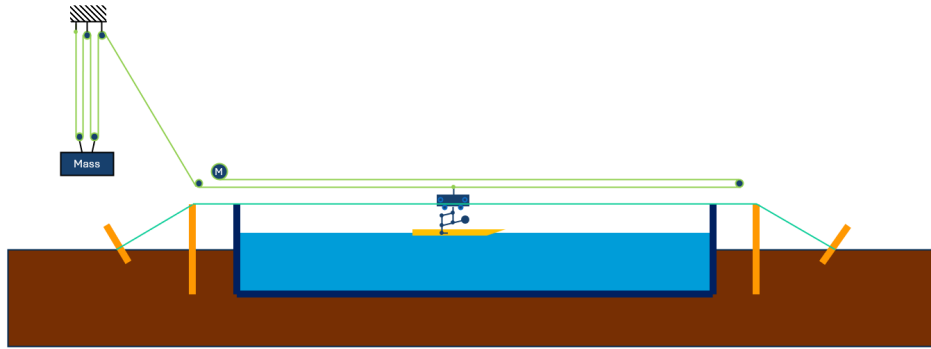


Figure 5.12.: Tow tank setup diagram

The experiments were conducted as a series of steady-state tests over a range of towing speeds. For each speed, the carriage velocity was held constant until the measured resistance signal reached a steady mean value. Resistance data were then recorded over a sufficiently long time window and subsequently averaged to reduce the influence of short-term fluctuations. This approach resulted in a set of discrete data points with an acceptable degree of repeatability, allowing the construction of reliable resistance–speed curves despite the limited tank length. In Figure 5.12, a diagram of the main towing setup can be found.

Measured model-scale resistance data were processed using the dedicated Model-to-Ship Converter (MSC). The conversion methodology follows the scaling approach described in Section 5.3, employing volumetric Froude number similarity appropriate for high-speed and

semi-planing craft. Frictional resistance was corrected using the ITTC-1957 correlation line, while residuary resistance components were extrapolated to full scale under equal Froude number conditions.

Full-scale resistance and speed curves were obtained for each configuration and subsequently compared with numerical simulation results (Section 5.4), including the findings in Figure 4.13 concerning the bow foils. Available full-scale tow test data were used as an external reference to assess the validity of the scaling methodology.

The test rig holds the IMU, load cell, signal conversion electronics, battery pack, monitoring display, recording memory (micro SD), ESP32 micro controller, cable guides and ball bearings, high-accuracy micro single point LiDAR (TFS20-L for accurate velocity calculation), etc. A picture is provided below in Figure 5.13.

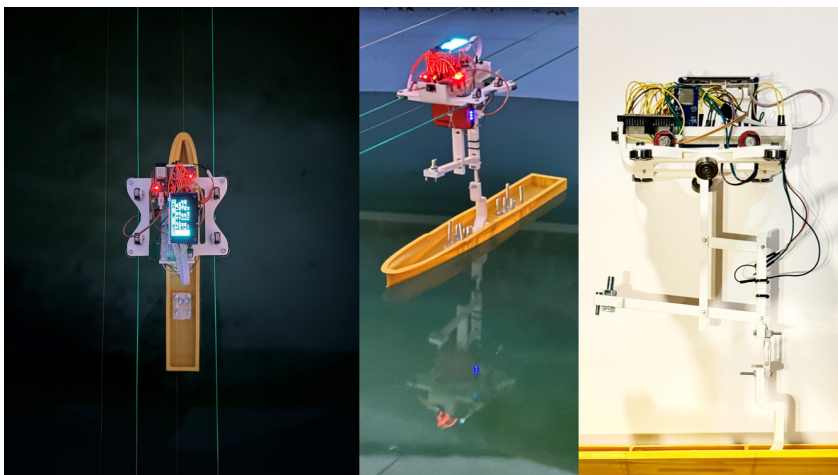


Figure 5.13.: Towing rig

Results

The scaled full-scale resistance curves show good agreement with predictions over the investigated speed range. Trends in resistance growth and characteristic changes in slope are consistently captured by both the experimental and numerical approaches. Comparison with full-scale tow test data further confirms the validity of the applied scaling procedure, with deviations remaining within acceptable engineering margins. Therefore, it can be assumed that the new Shark (v4) will deliver the predicted efficiency results based on theoretical assumptions and simulations.

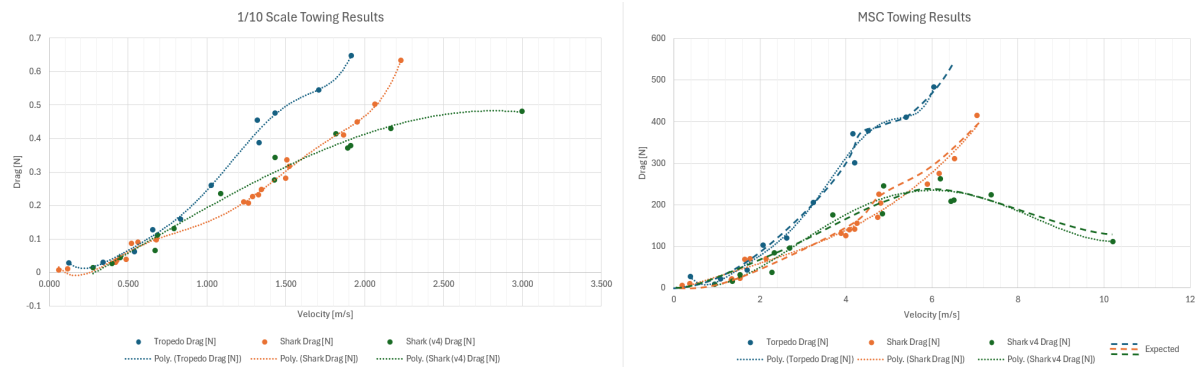


Figure 5.14.: 1/10 Scale towing results

Upgraded Shark In a separate free body test, the upgraded Shark exhibited exceptional directional stability when equipped with a 0.028 m^2 stern fin measuring 0.1 m in vertical length (converted full-scale values). This observation was made at both slow speeds and high-speeds on foil. The wetted area and natural trim behavior is identical to the simulations. Tests prove that, when the Shark's bow fins ventilate, the foils tend to ski over the water, effectively limiting the trim to a safe and efficient angle. For every trim angle (WAngle) in the v4 configuration, the Shark showed to have improved stability and handling. As the simulations suggested, the drag reduction is significant as measured in the 1/10 towing tests and trimming the bow fins allows for accurate behavioral optimization. The tests of the upgraded Shark show that the partial passive foiling configuration does not compromise the stability governed by the hull's geometry as also

stated in the discussion in Section 2.1.6. Note the increased drag at lower speeds; the hydrodynamic resistance of the foils and stern fin is not yet compensated for, as the foils do not generate enough lift to reduce wetted area and lift the bow. Comparing the 1/10th scale results with the MSC-translation, it is noteworthy that the benefit of reduced wetted area is accentuated in the high-speeds regimes by the conversion. The evaluation of the drag curve shape and magnitude highlights very effective use of the bow foils similar to the findings in simulation Figure 4.13. Validation using the simulation shows approximately 15% higher drag in the towing test. This might be caused by the fact that the Shark (v4) model was not sanded and coated due to time limitation. Also, a small error in the model's bow foil (warping in 3D print) could have influenced the result as the angle of attack is a very sensitive parameter of sailing efficiency.

Overall, the results demonstrate that the combination of steady-state towing experiments, volumetric Froude-based scaling, and post-processing through the MSC provides a robust, low-cost framework for predicting full-scale performance of the investigated hull forms, especially useful in comparative analysis.

5.3.4. Remote Control Model Fabrication and Instrumentation

History shows that various innovative concepts arise more quickly by improving remote control (RC) models than expensive and time consuming trial and error on full scale. It gives instinctive insight into how a design feels when piloting and how a model hull can be adjusted independently of purely scientific numerical research. This valuable insight issue is still difficult to address in real-world situations by simulation. Unfortunately, in this study, it is not possible to build all three fully functional RC models of the final proposed designs due to financial and time-related constraints. However, the new design is picked based on sponsor and general public interest. The fully functional design serves as a proof-of-concept as-well-as blue-chip advertising material. The innovative concept has the potential to attract new sponsors and team members as it has also proven to show increased media interest.

RC Radio Transmission Regulations

Remote control applications require special care, regarding regulations on radio wave transmissions, depending on the location. To exploit transmission equipment, often a power limit (mW), signal distribution/strength (dB) and frequency range is defined for specific uses and licenses. For every frequency band, it is defined whether it may be used by the public, military, satellite communications, etc. An application specific overview in discussion with the Belgian Institute of Postal services and Telecommunication (BIPT) is provided in Appendix C.1.48 (regarding telemetry transmissions).

ESC Data Recording & Upgrade Options

Bluetooth recording of performance parametrics. Default RC protocols allow for easy integration of autonomous flight controllers (e.g., ardupilot-based systems). The hardware configuration includes an F1-type model sterndrive, which employs the proportional trim principle discussed in Section 2.8.3. This sterndrive is illustrated in conjunction with the receiver, electronic speed controller (ESC), Bluetooth link, 4s battery pack, and digital steering servo.

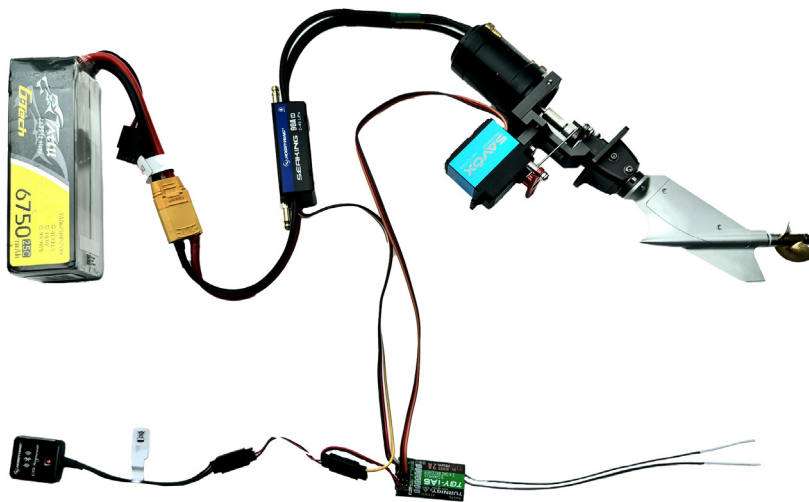


Figure 5.15.: Falcon RC hardware

Model Seagoing Handling Discussion

A short seagoing handling evaluation was conducted using a 1/5 scale RC model on the Galgeweel. Despite limited test duration, the model demonstrated stable and predictable behavior across the tested speed range. The trial primarily served as a qualitative assessment of handling characteristics, trim sensitivity, and cornering performance under realistic environmental conditions.

Due to rough water, an aggressively trimmed out drive shaft, and a rapid throttle increase, the model experienced a blowover event. This resulted in water ingress and damage to the receiver, the only electronic component without water protection, leading to the premature termination of the test. Although the setup was not originally intended for operation in brackish water, practical constraints necessitated testing at this location. The damaged receiver was subsequently replaced, fully restoring functionality. The modular drive configuration allows for rapid and precise trim-angle adjustment, and the setup has since been updated to reduce the likelihood of blowover under similar conditions.

The following qualitative observations were recorded during the test:

- A trimmed-out motor configuration reduces the wetted area and increases effective aerodynamic lift, which is a beneficial configuration at lower speeds.
- The combined test conditions resulted in a blowover at approximately 57% throttle, corresponding to a current draw of 34 A and a measured rotational speed of approximately 56k eRPM.
- The two-blade cleaver propeller exhibited no observable cavitation.
- Cornering behavior was consistently tight and controlled at all tested speeds, up to approximately 7 m/s.
- Due to the overloaded mass configuration at scale (+59%), the water surface contacts the trailing edge of the tunnel when stationary, a behavior typical for this hull type.
- Roll angles remained very low during maneuvering, indicating strong lateral grip.
- The proportional steering–trim coupling effectively increases bow-down attitude during cornering, with a small and expected response delay.
- The propulsion system provides excess power relative to the model scale, enabling the hull to reach the intended scaled speed range with ease. While the scaled equivalent of the full-scale design of 90 km/h corresponds to approximately 43 km/h, significantly higher model speeds are feasible on flat water with optimized drive tuning.

Although the sea trial was brief, the observed handling characteristics and stability trends indicate strong potential for the proposed hull concept. The qualitative behavior of the scale model aligns well with the predicted performance and handling characteristics derived from numerical and empirical analyses. The model's advertisement poster, livery process, and phosphorescent paint, which adds aesthetic value, are illustrated in Figure C.54. The measurement data of a stationary water test of the propulsion system is provided in Appendix C.3, Figure C.55. It is noteworthy that the plot employs eRPM (electrical RPM). The propeller rpm is equal to half of the eRPM, given that the SSS 2960/2881KV motor has four poles.

5.4. Comparison to Simulation Results

As hydrodynamics in hull design is a complex matter requiring a lot of background and deep understanding, multi-facet validation has been conducted on various metrics. The cogent evidence of known conditions allows for comparison resulting in reliable range extrapolation and geometry type expansion. In this section, the used predictive tools are compared with experiments, proving the validity of statements, behavior, conclusions, etc. for every preliminary section of analysis.

5.4.1. Validation Metrics

The validity of the data can be ascertained by examining the total hull drag, which serves as the basis for all performance metrics. Therefore, simulation, full-scale and scaled experiments of the known configuration are measured against each-other. Given the simplicity of reproducing the test conditions of flat water towing experiments on other scales and in simulation, the respective results are selected for evaluation.

5.4.2. Full-Scale versus Simulation

The most significant validation analysis is the comparison of full-scale models with simulations. This is due to the fact that simulations serve as the primary evaluation tool for assessing hull geometry upgrades.

What first comes to mind in observation of the simulation results of the Torpedo versus the Shark are the instabilities in the spiking data points. While not yet substantiated, there is a hypothesis that the input geometry file may be the source of the observed effects. Analysis and improvement in Solidworks revealed sharp edges, complex cavities and invalid (singularity points) surfaces. Not all issues could be removed. Hence, some modifications are critical to make the simulation run without errors. Consequently, it was not possible to accurately simulate the 3-5 m/s region. Still, the delayed hump (see Section 5.1.5) was correctly captured at approximately 17 km/h for the Torpedo and 18.5 km/h for the Shark. However, the experiment did not capture the jump in the Torpedo's curve (drag and wetted area) at 27.5 km/h. This holds two causes. Firstly, the tunnel closure effect, which is responsible for the observed increase in drag, is an instability characteristic that becomes more pronounced at higher speeds. However, this effect is not a constant occurrence at velocities of exactly 27.5 km/h and also depends on the wave conditions, as explained in Section 4.1.2. Secondly, the curve of the Torpedo is extrapolated from 25 km/h onward, not revealing the sudden

increase in drag. Note that this drag extrapolation, based on the data of the full-scale measurement in Figure 5.3, is underestimated compared to the scatter data. This issue is related to the regression methodology exploited. The steeper slope of the simulation is therefore more accurate than the smoothed experimental curve (see Figure 5.16).

The Torpedo simulation should probably use a surface roughness of 0.0004 m instead of the used global surface roughness of 0.0003 m used in Figure 5.16 to better match the full-scale experimental results. Note the drop after the hump. This level of detail could not be captured in experiments. To measure the local minimum, a towing force decrease by the tugboat would be required. However, the sharpness of the hump indicates the displacement behavior and supports the theory of the steep exponential slope post-hump. A comparison of the Torpedo's simulation run 32 with the experimental data from 2023 (portfolio, range [19;26] km/h) reveals the post-hump data and slope to be highly accurate. This finding potentially underscores a regression and a potential tugboat wake issue in this region of the 2025 measurement. Note that it is not known

whether the strut was taken into account. The 2025 result suggest inclusion of the strut in 2023. It cannot be derived whether the long or short strut was used as the long strut was not tested in 2025.

In the low-speed region, a decelerating flow simulation is employed to approach the experiment, as the accelerating flow CFD revealed unfeasible flow artifacts for the Torpedo simulation. This results in an overestimation of the drag, as the momentum of the water cells is unable to be influenced instantaneously by the inflow boundary condition alone. Conversely, the Shark exhibited no such issues. Steady-state analysis confirms the accuracy of the experimental plot in this low-speed region for the Torpedo.

In conclusion of the Torpedo, the utilization of simulations is deemed legitimate for the evaluation of the actual performance of the hull. Nevertheless, it has been demonstrated that the 2025 experiment may exhibit a slight overestimation in the context of high-speed planing regimes.

In contrast, the Shark's simulation is smooth and accurate as the cleaned geometry in Solidworks resulted in a better and less complex model.

All the way up to 20 km/h, the curve is precisely following the measurement. The simulation, in ideal conditions, shows more detail revealing theoretical phenomena that could not be derived from the experiment. At 13.8 km/h — the theoretical, non-delayed hump speed of the Shark — a bend in the simulated graph could be noted, followed by a short plateau. This effect is most probably caused by the deeply planted sharp bow, locally acting as a displacement vessel. After the plateau, from 14 km/h onward, the bow drops (see Figure 4.5) further increasing drag. The actual, most pronounced hump (18–19 km/h) is correctly captured as stated in the previous paragraph. From the hump onward, the curve propagates quasi-linear, withal slightly exponential. Similar to the post-hump overestimation for the Torpedo, the Shark's curve slightly overestimates the drag above hump speed. The underlying mechanisms that precipitate this phenomenon remain to be fully clarified and substantiated. Simulation highlights no significance in minor ballast repositioning causing the slope to change (see Section 4.1.4) and the dense scatter plot of measured data in the region above 20 km/h indicates the good accuracy of the regression. Consequently, these potential causes have been ruled out.

The potential causes that could be found in full-scale experimentation include:

- As the speed increases, the tugboat rises, promoting the positive pitch of the Shark when pulling the towing line. However, this is unlikely due to the significant length of the rope.
- The digital measurement was not used in the high region due to a “hold” condition. However, the measurement coming from the analog device at the end of the scale range may be inaccurate showing lower values as the spring is tensioned. Also static friction in the device might have influenced the scatter of data.
- Lack of the configured simulation physics model to accurately evaluate high-speed surface interaction. The simulation model currently only holds gravity, rigid moving bodies (motion coupled), turbulence and viscosity models. High-speed inflow may affect viscous interaction and is potentially out of the accurate range of the used model. Models for surface tension, dynamic void pressure, air entrainment, cavitation dispersed multi-phase flow, free particles, wind, etc. are available but set inactive to reduce the computational demand.
- Delay and inaccurate speed data as it was not possible to use high-end GNSS and/or inertial recorders (see Section 5.1).
- The full-scale towing experiment is inconsistent with the setup and procedure the ITTC defines (force balance, thrust moment compensation, etc.).

- The resistance influence of the strut may be overestimated by the towing test as the drag induces a pitch down moment leading to an increase in wavemaking (see Figure 5.3). This suggestion enforces the validity of simulation accuracy. In reality, the pitch down is positively overcompensated by the position of the thrust vector.
- Last but not least, the wake behind the tugboat may have influenced results as the Shark had a fixed rudder setup and could not be kept at a fixed position, aligned with the tugboat. In general, the over-all position was reviewed acceptable on scene. Hence, in data analysis, the behavior is showing like it has surfed down the high-speed enlarged central wake wave of the deep-V tugboat in the second half of the last run. Furthermore, surfing on top of that same wake may also reduce drag, as the wetted area may be reduced. As the Torpedo showed the same issue with the curve's slope when evaluating the 2023 data, the deviation is found consistent for both hulls above hump speed. This highlights a potential issue with the 2025 tests at high speed.

Note that these are all difficulties that are eliminated using idealized towing tank setups and that these simulations were conducted with a water density of 998.2 kg/m^3 . Salty water slightly promotes buoyancy due to increased density, essentially leading to reduced drag. Future work recommendations suggests re-measurement in the $> 20 \text{ km/h}$ regions with even longer (smooth) towing lines combined with linearity analysis of measurement devices. Capturing the trim and multi-angle scientific video recordings may be a valuable tool for assessment.

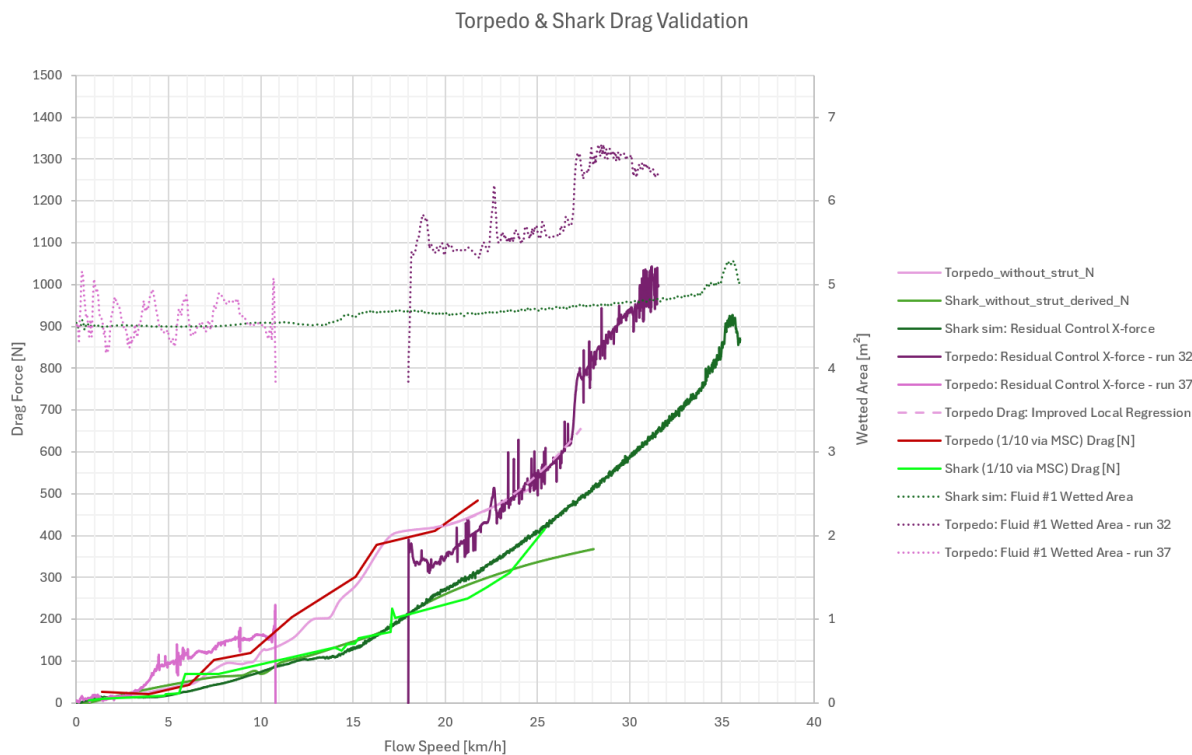


Figure 5.16.: Torpedo and Shark in simulation and full-scale experiments compared

When drag tunnel experiments from scale models are included in the validation analysis, it can be concluded that the values obtained by all methods are similar. The post-hump trend of these scale drag tests also indicates limited error in full-scale measurements, which the simulation correctly approximates. The combined graph highlights acceptable accuracy of all hydrodynamic simulations, experiments and related conclusions found across the entire project.

5.4.3. Scale-Model versus Scaled Simulation

This short section highlights the additional revision layer based on simulation-to-simulation validation. This is a valuable way to double check the ITTC scale-model translation script (Model-Ship Converter or MSC) written for this study.

The procedure is as follows. First, a simulation is prepared on a 1/10 scale model. The flow velocity is increased from 0 to 3.162 m/s in approximately 30 seconds, which translates to 10 m/s on full-scale according to Froude similarity calculations. The results are processed and exported via a Flow3D Post spreadsheet. The resulting CSV file is then structured according to the input required for the custom-built software (V_M, R_T_M, S_run_M). This data is automatically processed by the MSC script, resulting in a comprehensive file containing all kinds of ITTC-relevant coefficient data and the resulting speed versus resistance. This data is compared with a full-scale simulation for validation. For information purposes, the full-scale drag of the towing test is also plotted with a calibrated roughness of 0.0001 m. Note that both validation simulations were performed with ideal surface roughness of zero, resulting in a slightly lowered graph. The discrepancy in the results is strongly suspected to be related to the fact that the ITTC's surface roughness requirements are not being considered

here. The model is generally much smoother — close to ideal — compared to the real (painted steel) ship. This inconsistency may result in a lower MSC result. The error increases along with the trend in surface roughness deviation or influence on drag. This shows that the results of the translation software are accurate and the code is correct.

The analysis presented here uses the length-based Froude number as a preliminary validation. However, the scale model experiments of this research project eventually use the volumetric Froude number for high-speed marine vehicles as prescribed by the ITTC [3]. This produces comparable but slightly more accurate results, which, according to the previous validation section, closely resemble the full-scale experiments. The results of this study are excellent and highly accurate as competing teams and companies have found that resistance approximation is an extremely complex task throughout the industry, often requiring very expensive instrumentation.

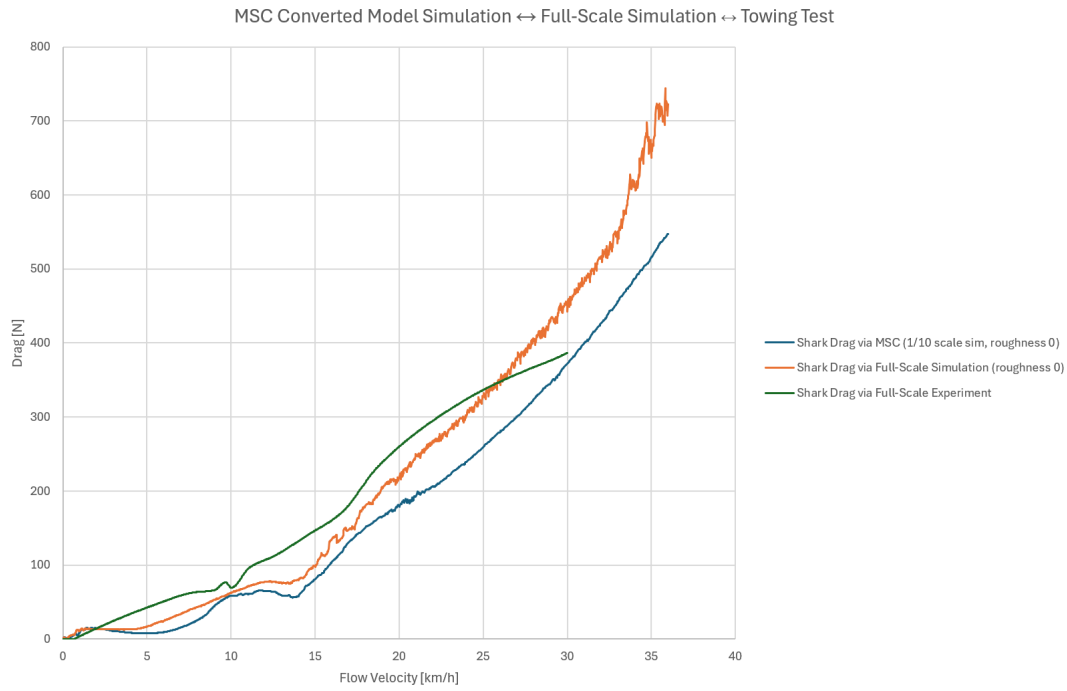


Figure 5.17.: MSC software validation

6. Future Work

6.1. Improvements of BCD

For detailed insights on the accuracy of every BCD-included parameter, a very extensive cross-correlation CFD and/or experimental analysis should be conducted. Also, a future update could include MSE or mean square error optimization for curve fitting to accelerate analysis. Note that the user should judge whether parameters are realistic. The BCD Parameter Help can be used as a guide.

6.2. Theory Completeness

The overview of hull design can never be all-inclusive as it is a complex and mostly empirical science domain. It is stated as incomplete, even though this very thesis will serve as a hull design reference work for the team. Therefore, it is strongly advised to complement this thesis by sources like the ITTC's database, Ship Resistance and Propulsion book by Anthony F. Molland, Stephen R. Turnock, and Dominic A. Hudson [79] and the Secrets of Tunnel Boat Design book by James D. Russell [11] who served as a major industry expert reviewer in this thesis.

6.3. Further Use of Custom Towing Tank Setups

If similar towing tank setups are built and used in the future, it is recommended that a reliable, smooth winch setup be used to improve measurement accuracy. This will make it possible to further automate the process, as the speeds will be more consistent and predictable. Using a carriage over rigid rails is preferable as well as a longer tank. The implementation of a permanent setup within the UASB laboratory has the potential to enhance the quality of designs and accelerate the testing of hulls and propulsion systems. This, in turn, would ensure optimal performance and increase the comprehensiveness of students designs. The approach necessitates a one-time investment, thereby providing a cost-effective and lasting solution.

Since testing directionally unstable designs, such as the Shark, proved difficult, it is recommended to stabilize them before testing. Directional deflection puts high torsion loads on the rig and can influence the measurement. Adding a shallow stern fin may slightly increase frictional resistance but improves operational satisfaction.

6.4. Structural Analysis

As the (detailed) structural analysis and material selection (e.g., via Ashby plots) is out of the scope of this study, it should be investigated in a following-up study report. A material stress focused external simulation (not direct fluid structure interaction or FSI) might seem a sufficient technique for linking

this study to the structure analysis and reduce hull mass as much as possible. However, one should keep in mind two things: (1) the pressure distributions are dynamic and thus different in every simulated timeframe (use the max pressure in each hull surface cell), and (2) it is impossible to simulate all possible hull impacts / pressure distributions (increased safety factors apply). An inconvenience of the first issue implies the fact that "taking the maximum of each cell" is good for a local stress estimation, but a specific distribution of a simulated timeframe can result in bending moments greater and or at least different for an (accumulated) maximum matrix. This requires special care when configuring an extended stress model.

In the context of the implementation of modifications to the Shark, such as the bow fins and sponsons, it is recommended that strength calculations and finite element method (FEM) analysis be performed during the design of mounting materials and the interaction with and deformation of the existing composite hull. In light of the findings from this dynamics study, it is recommended that further tests be conducted as follows. The foil test load was determined to be approximately 0.85 kN at the center. The second test should include a load identified as a single fin (hull torsion) load of approximately 0.5 kN, positioned off-center at the center of pressure of the fin. This configuration was based on

the assumption that the other fin was fully ventilated entering a sharp corner. The third test holds a single sponson load of approximately 0.3 kN (also leading to torsion) and a double sponson load of 2 kN due to heavy wave impact (most is absorbed by the hull). These loads were determined, based on the pressure distributions of the tests in the relevant simulation chapter. Note that the latter wave load (2 kN) significantly exceeds the standard cornering load (0.3 kN) at 10 m/s. It is reasonable to conclude that the worst-case scenario would involve the primary carbon crossbeam tube fully absorbing the load of the sponsons. One should define the safety factor based on the risks involved.

6.4.1. Shark's Use Beyond 10 m/s

The maximum test speed of the Shark in this study was limited to 10 m/s, as this represents a practical compromise between achieving a substantial increase in current velocity (22.5 km/h with the Torpedo) and avoiding the need for additional installed power during the current development cycle of 2025-26. The steady-state simulations furthermore indicate that the modified Shark operates close to the onset of porpoising. It is important to emphasize that the maximum attainable speed of a hull is not a strict physical limit; rather, it is highly sensitive to parameters such as trim, mass distribution, deadrise angle, and the longitudinal position of the center of mass. Consequently, higher velocities cannot simply be achieved without careful optimization in practical tests.

For planing-type hulls, preventing porpoising at elevated speeds generally requires reducing trim — by lowering the effective angle of attack of the bow fins — or applying a rapid increase in thrust to counteract the instability, which can also suppress porpoising. Note excessive power may in fact re-increase the trim (WAngle), re-inducing porpoising at a higher speed. A reduction in trim will also increase the wetted surface area of the Shark, a highly sensitive effect in its design. Due to the very small deadrise — almost absent at the transom — an increase in speed with corrected bow fin angle of attack will not necessarily result in a reduction of the wetted area

due to a heave motion, contrary to what is typically observed in conventional planing craft with a stronger deadrise. As a result, the hydrodynamic resistance of the Shark is expected to rise more steeply in the high-speed regime (above 10 m/s) compared with conventional V-shaped or deep-V hulls. Although the Shark benefits from a low deadrise and thus more efficient dynamic lift, yielding a generally lower resistance curve at moderate speeds, this advantage diminishes progressively at higher velocities. It is recommended to fine tune the angle of attack of the bow foils at every new practical maximum velocity. It is also advisable to experiment with the

limitation of the rudder angle at higher speeds to ensure the autopilot cannot compromise pilot safety and that cornering is most effective in practical racing conditions.

In addition to porpoising sensitivity and dynamic stability, the structural integrity of the sponsons and fins will become increasingly critical as the speed intensifies. Therefore, defining a strict upper velocity limit is not feasible, since small adjustments in center of gravity, fin angle of attack, and structural reinforcement can significantly alter the high-speed capabilities. Nonetheless, based on estimates of effectiveness for this hull type from the literature and industry insights, operation above approximately 45–50 km/h is not recommended. Both the stability characteristics and the limited wave-deflection capability at such speeds suggest potentially hazardous operating conditions, irrespective of structural integrity.

This study highlights that high-performance autonomous craft may benefit more from alternative geometries, such as low-center-of-gravity catamarans with features like negative inner freeboard for fast and stable cornering. For lightweight high-efficiency vessels, ground-effect and air-cushion configurations become particularly attractive. The present work shows that a feasible hull of comparable size to current fleet assets can theoretically operate across a broad speed envelope of 55–100 km/h using only 15 kW of propulsion power. With appropriate engine trim, this speed range can be extended further at higher power levels. Air-cushion vehicles such as F1H₂O craft exemplify geometries whose dynamic characteristics are less sensitive to speed, maintaining comparable stability across their operating envelope. Similarly, the proposed design in this study demonstrates nearly constant stability metrics throughout the full 55–100 km/h range, underscoring the potential of such configurations for future high-performance autonomous applications with great flexibility for upgrades.



Figure 6.1.: Overview Shark v4 & Falcon

6.5. Future Research

In accordance with the principle of curved sponsons, it would be a worthwhile subject to investigate whether morphing or snake-sectioned hulls for low and medium speed (displacement) vessels can lead to significant drag reduction in maneuvering. This may also offer interesting opportunities for stability and reduction of green water over strong waves (hull follows the shape of the wave) and furthermore for applications with snake-inspired USVs and diving robots that have the potential to be highly efficient and extremely maneuverable.

6.6. Remarks for the New Hull

6.6.1. Advanced Materials and Manufacturing Techniques

A smooth surface finish is important (use suitable molding techniques accordingly). Aim for a lightweight core assembly and minimize total weight by adding optimized rib structures and local reinforcements including sharp inner corners prone to rupture (e.g., inner corner of the reversed chines of the sponson attachments). Make the design stiff to ensure safe and reliable operation. For the pilot ergonomics a M5-recumbent bike style cushioned seat is recommended. Initial testing can be done without canopy (slightly compromising efficiency). Carefully design all safety measures; consider a 5-point harness, small air tank, inflatable canopy cushion to keep the cockpit above the water in case of emergency, cooling and breathing ram air intake, helmet (limit lateral motion for cornering with high Gs), etc. before testing at full power/speed. A canopy should be lightweight, impact resistant and may be tinted to reduce the greenhouse effect (similar to sunglasses). This may also hold aesthetic value.

6.6.2. Design Changes of the Proposed New Hull

Within certain margins, variations of geometry and mass is allowed featuring acceptable performance (e.g., enlarged superstructure for more spacious internals). Note that this must be done with much caution and reassessing performance and stability with the TBDP is advisable. An expert review is recommended for correct interpretation of the results.

7. Conclusion

This thesis presented the design and analysis of an AI-class solar boat race hull, with a primary focus on performance, stability, and competition compliance. A simulation-driven methodology, complemented by experimental validation and expert review, was employed to systematically evaluate the limitations of the existing fleet and to develop a future-proof hull concept tailored to the requirements of high-speed autonomous racing.

A comprehensive assessment of the current UAntwerp Solar Boat Team fleet demonstrated that neither the Torpedo Mk.II EVO nor the default Shark hull configuration provides a viable long-term pathway toward sustained operation beyond approximately 10 m/s without significant compromises in stability, efficiency, or safety. The Torpedo Mk.II EVO exhibits fundamental limitations related to tunnel collapse, sponson interference, and an exponential post-hump drag increase, which are characteristic of displacement-dominated behavior and inherently restrict its usable speed range. As such, further performance increases for this platform are neither technically nor strategically justified.

The Shark's hull, in contrast, is identified as the only platform within the current fleet that retains genuine potential for further speed increases. Its planing behavior, near-linear post-hump drag trend, and comparatively lower mass provide a stronger foundation for higher-speed operation. However, simulations and experimental observations clearly indicate that the default Shark configuration is directionally unstable, exhibiting stern sliding, nose-diving tendencies beyond the hump region, and failing cornering near and above 10 m/s. Consequently, safe and predictable operation at higher speeds can only be achieved through targeted geometric and hydrodynamic adaptations. These include the integration of bow foils, stern fin(s), and specialized sponson geometry, all of which were demonstrated to significantly improve cornering behavior, directional stability, and handling predictability in the updated Shark configurations. Operation beyond the simulated 10 m/s regime

remains subject to uncertainty and is therefore explicitly identified as future work requiring further validation.

To overcome the intrinsic limitations of monohull-based concepts at higher speeds, a novel hull design was developed based on a tunnel hull catamaran configuration optimized for a wide operational envelope. Dedicated velocity prediction tools, combined with industry expertise and iterative geometry optimization, enabled accurate and computationally efficient performance forecasting across a broad range of mass, power, and trim conditions. The resulting design demonstrates stable and predictable behavior between approximately 55 and 90 km/h, with feasibility extending toward 100 km/h under increased power availability. The analysis definitively demonstrates that the development of a tunnel hull is not warranted for target speeds below 55 km/h. At these speeds, a tunnel does not attain the requisite stability and does not transition into the aerodynamic ground effect benefiting regime. This finding establishes a clear speed-based decision boundary for future hull selection within the team.

The proposed tunnel hull exhibits superior cornering performance, minimal roll and pitch sensitivity, and a favorable lift distribution dominated by controlled aerodynamic ground effect. These characteristics are particularly beneficial for autonomous operation, as predictable attitude behavior directly supports sensor reliability, antenna field-of-view, and robust perception under dynamic race conditions. The deliberate use of passive stability mechanisms — such as opti-

mized tunnel geometry, deadrise variation, and lift balance — eliminates dependence on complex active control surfaces, thereby enhancing system robustness, reliability, and offshore behavior.

A key strength of this work lies in the extensive cross-method validation. Computational fluid dynamics, velocity prediction modeling, full-scale drag experiments, towing tests, qualitative wind tunnel observations, and 1:5 scale RC seagoing trials consistently revealed aligned trends across performance, stability, and handling metrics. While some absolute numerical discrepancies remain, the strong qualitative and relative agreement confirms the reliability of the adopted methodology and substantiates the physical plausibility of the derived conclusions.

Beyond the hull designs themselves, this thesis delivered a structured analytical framework and a suite of custom software tools, including

the Boat Curve Dashboard and a Model–Ship Converter based on ITTC procedures. These tools significantly enhance the team’s capability to rapidly evaluate design changes, assess feasibility limits, and support future development cycles, thereby establishing a durable reference framework for performance-driven hull analysis within the UAntwerp Solar Boat Team.

From a strategic perspective, two clear development pathways emerge. In the short term, performance increases should be pursued through the use and upgrading of the Shark’s hull, incorporating stabilizing and performance elements and such as bow foils, stern fins, and sponson adaptations to safely extend its operational envelope toward the upper limits examined in this study. In the longer term, once sustained speeds beyond 55 km/h are required, a transition toward the newly developed AI-class tunnel hull becomes both technically necessary and competitively advantageous.

In conclusion, the tunnel hull concept presented in this thesis represents a technically sound, scalable, and future-oriented platform for high-speed autonomous solar racing. Together with the upgraded Shark as an intermediate solution, it provides a coherent and validated roadmap for performance progression, balancing innovation with safety, predictability, and competition relevance. While further work remains necessary in structural optimization and control system integration, this thesis establishes a robust and defensible foundation for the next generation of AI-class boats.

A. Symbols

UASBT	University of Antwerp Solar Boat Team
MEBC	Monaco Energy Boat Challenge
USV	Unmanned Surface Vehicle
UAV	Unmanned Aerial Vehicle
ROS	Robot Operating System
SLAM	Simultaneous Localization and Mapping
L/B	Length-to-Beam ratio
LWL	Length at the WaterLine
WWL	Width at the WaterLine
S/L	Speed-to-Length ratio
SWATH	Small Waterplane Area Twin Hull
SES	Surface Effect Ship
ACV	Air Cushion Vehicles
CFD	Computational Fluid Dynamics
ITTC	International Towing Tank Conference
DOF	Degrees Of Freedom
CG	Center of Gravity
CL	Center of Lift
AC	Aerodynamic Center
CoB	Center of Buoyancy
HSMV	High-Speed Marine Vehicle
LiDAR	Light Detection And Ranging
GNSS	Global Navigation Satellite System
AoA	Angle of Attack
EMI	ElectroMagnetic Interference
VLf	Very Low Frequency (radio waves)
GCS	Ground Control Station
CFRP	carbon fiber-reinforced polymer
TBDP	Tunnel Boat Design Program
INS	Inertial Navigation System
AR	Aspect Ratio
RC	Remote Control
FHSS	Frequency-Hopping Spread Spectrum
MAVLink	Micro Air Vehicle Link
GCS	Ground Control Station
VOF	Volume-Of-Fluid
RANS	Reynolds-Averaged Navier–Stokes
FAVOR	Fractional Area/Volume Obstacle Representation
PIV	Particle Image Velocimetry
FFT	Fast Fourier Transform
EDF	Electric Ducted Fan
ESC	Electronic Speed Controller
BIPT	Belgian Institute of Postal services and Telecommunication
FSI	Fluid-Structure Interaction
LR	Lloyd's Register
DNV	Det Norske Veritas (Norwegian classification society; formerly DNV GL)
NPL	National Physical Laboratory (United Kingdom)
NSMB	Netherlands Ship Model Basin (now part of MARIN)
GL	Germanischer Lloyd (merged with DNV to form DNV GL in 2013)

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C. Appendix

Additional supplementary material such as theoretical background, raw data, detailed reports, or technical drawings are appended here.

C.1. Literature Study Background

C.1.1. Hump Speed and Resistance Increase

The phenomenon of hump speed is closely related to hull speed and can be characterized as follows:

- The hull speed is considered as the fastest efficient displacement speed before the transition zone.
- The hump speed zone refers to the local maximum in resistance encountered when the vessel transitions from pure displacement motion towards semi-displacement or planing behavior. At hump speed, the vessel's bow climbs over the crest of its own bow wave, leading to a sharp rise in trim angle and resistance. Overcoming this “resistance hump” requires either a significant increase in propulsion power or a hydrodynamically favorable hull form designed for high-speed operation.

For conventional displacement ships, surpassing hump speed is energetically prohibitive, as the required effective power scales steeply with increasing resistance. However, for slender or light vessels with flatter bottoms or hydrodynamic lifting surfaces, it is possible to overcome the hump and enter a regime of reduced resistance per unit speed: the planing regime. Whether this is true planing or high-speed semi-displacement fully depends on the geometry. True planing is claimed to be beyond $Fr \approx 1.1$ by Figure 2.1.

In essence, the distinction between displacement and planing hulls is not absolute but represents a continuous transition. Whether a hull is called and optimized as a planing hull or displacement hull usually is defined based on which cruising speed it will operate: in displacement (below hump speed) or planing regime (above hump speed). However, slender hull designs going above hump speed might still be considered a displacement hull as the hump itself has not a significant magnitude and the displacement characteristics are extended above hump speed. This raises a question: “When are hulls called semi-displacement?” This is a highly relevant question. One could state that a semi-displacement hull shares characteristic geometry and hull features of both displacement and planing designs. In this way, they have the potential to have excellent properties in both regimes, not reaching ultimate efficiency as a trade-off. This is mostly used in small to medium-sized, high-speed ($> hullspeed$) boats that require good seakeeping and cornering (piercing waves: displacement) but also want to be efficient above hull speed (planing regime). Note that in commercial (non-racing) boats the length-to-beam ratio (L/B-ratio) is often relatively low. A higher L/B-ratio lowers the resistance “barrier” at hump speed. The speed at which the hump occurs is almost entirely controlled by the length at the waterline (LWL). Intuitive statement:

- A lower L/B (beamy hull) pushes more water aside $\rightarrow$ larger bow wave $\rightarrow$ higher hump resistance.

- A higher L/B (slender hull) disturbs the water less → lower hump resistance, making the hump easier to overcome.

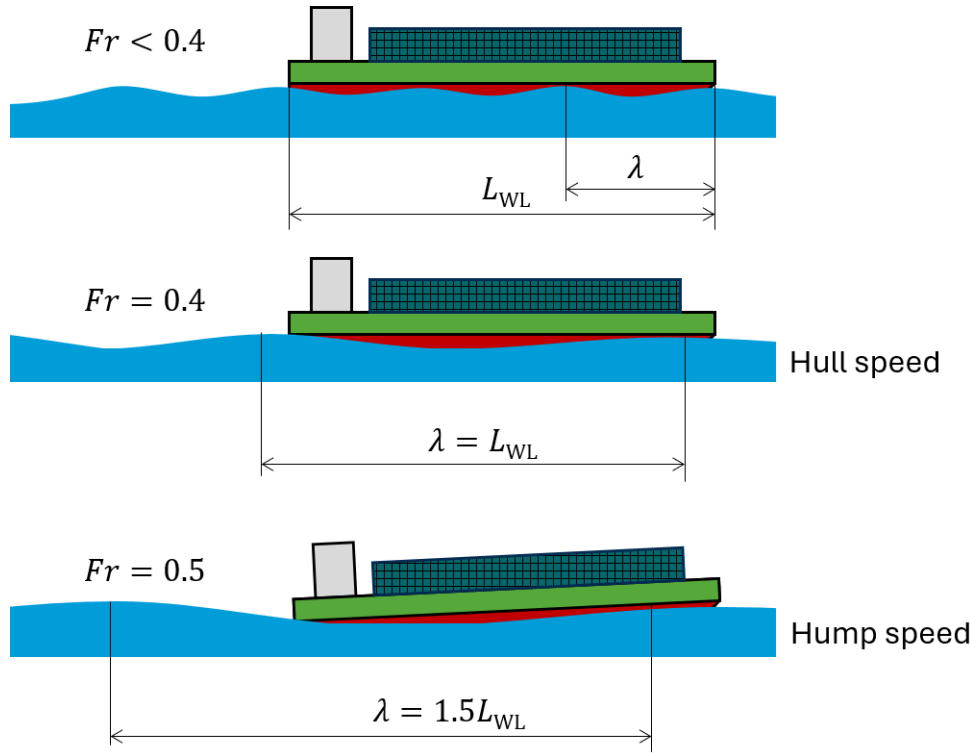


Figure C.1.: Hump speed & hull speed

C.1.2. Mach Number versus Froude

The theory behind the Froude number is somehow comparable to the mach number equation, a dimensionless value for the division of the object velocity and the speed of sound in the medium.

$$Ma = \frac{V}{c} \quad (C.1)$$

Ma = mach number [-]

V = flow velocity relative to the medium [m/s]

c = speed of sound in the medium [m/s], given by $a = \sqrt{\frac{\gamma RT}{M}}$

γ = ratio of specific heats [c_p/c_v]

R = specific gas constant [J/kg·K], $R \approx 8,314$ J/(mol·K)

T = absolute temperature of the medium [K]

M = molar mass [g/mol]

An easy way to make the comparison more understandable is to evaluate physical phenomena. Imagine the waves in a 2D field undergoing the Doppler Effect due to the linear uniform displacement of its source. As the source speeds up, the speed of the waves traveling outwards will not be able to catch up with the source.

Now, critical source velocity is reached (Mach 1). When further increasing speed, a shock wave cone will form as seen around supersonic aircraft and during high speed bullet imaging. The similarity for boats can be visualized by the Froude number. The shockwave is the angled wave formed at the bow and stern of the vessel.

The waves own outward traveling velocity cannot catch up with the velocity of the boat [80]. A more specialized (CFD) analysis and discus-

sion of the Torpedo's waveform was conducted in prior research [27] by J. Peeters in 2020.

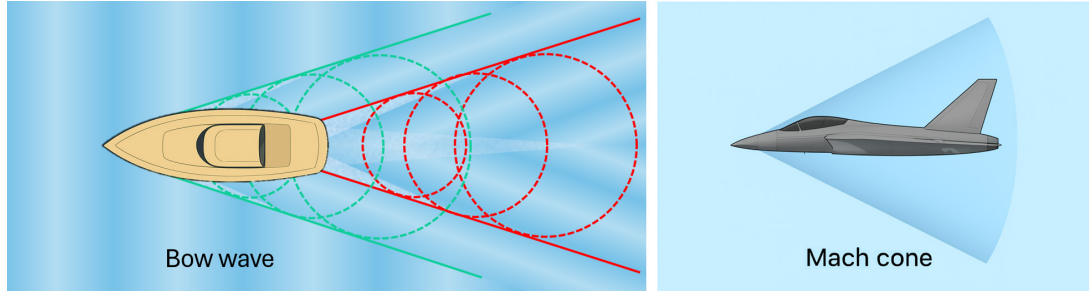


Figure C.2.: Bow wave versus mach cone

However, it is important to note that the analogy between the Mach cone of supersonic flight and the bow wave of a vessel is limited and only applicable in planing regimes for $Fn > 0.5$. While the Froude number resembles the Mach number in form, the underlying wave dynamics differ significantly. In water, for $Fn < 0.5$, the wave pattern generated by a moving hull follows the classic Kelvin wake, which is characterized by a fixed half-angle of about 19.47° , independent of speed in deep water [81]. This results from the dispersive nature of surface gravity waves, where different wavelengths travel at different phase velocities, in contrast to the non-dispersive propagation of sound waves in air. Only under certain conditions, such as finite depth or interference effects, does the wake angle narrow, as observed in experimental and computational studies [82, 83, 84]. Thus, although the Froude number provides a useful scaling parameter, the wake

of a ship cannot be equated directly to a Mach cone, since dispersion and surface-boundary effects govern the wave geometry rather than a simple supersonic shock relation.

In problems of fluid flow there are always at least four parameters involved: the pressure difference ΔP , velocity V , characteristic length d , and fluid density ρ . In open-channel hydraulics, gravity acceleration g is an additional governing parameter. Considering a closed pipe circuit, and provided that the liquid remains in the liquid phase (i.e. no cavitation), the piezometric pressure ($P + \rho gz$) is merely affected by gravity, since the static pressure P may take arbitrary values. In an open channel, however, the existence of a free surface imposes the atmospheric pressure P_{atm} at the boundary, and gravity therefore directly influences the piezometric pressure and flow regime [5].

C.1.3. Example of Froude Similarity Calculation

Full scale prototype: length $L_p = 8$ m, beam $B_p = 1.8$ m, speed $V_p = 25$ m/s. **Model:** $L_m = 1$ m (geometric scale $\lambda = L_p/L_m = 8$) using equation 2.3.

$$\text{Froude similarity:} \quad Fr_m = Fr_p \quad (C.2)$$

Prototype Froude number

$$Fr_p = \frac{V_p}{\sqrt{gL_p}} = \frac{25}{\sqrt{9.81 \cdot 8}} = \frac{25}{8.862} \approx 2.82$$

Model speed (from $Fr_m = Fr_p$)

$$V_m = V_p \sqrt{\frac{L_m}{L_p}} = 25 \sqrt{\frac{1}{8}} = \frac{25}{\sqrt{8}} \approx 8.84 \text{ m/s}$$

Model beam (geometric scaling)

$$B_m = \frac{B_p}{\lambda} = \frac{1.8}{8} = 0.225 \text{ m}$$

Time scale (useful for run durations)

$$t_m = \frac{t_p}{\sqrt{\lambda}} = \frac{t_p}{\sqrt{8}}$$

Figure C.3 illustrates the relationship between prototype speed, Froude number, and the equivalent model speed under Froude similarity. On the left axis, the Froude number increases linearly with prototype speed, reaching a maximum of approximately 2.8. The right axis shows the corresponding model speed for a geometrically scaled 1 m hull, which follows directly from the condition of equal Froude number. At the maximum prototype speed, the equivalent model velocity is approximately 8.8 m/s. This correspondence enables the use of small-scale models to investigate hydrodynamic instabilities such as wave-making, porpoising, and pounding while preserving the correct balance between inertial and gravitational forces.

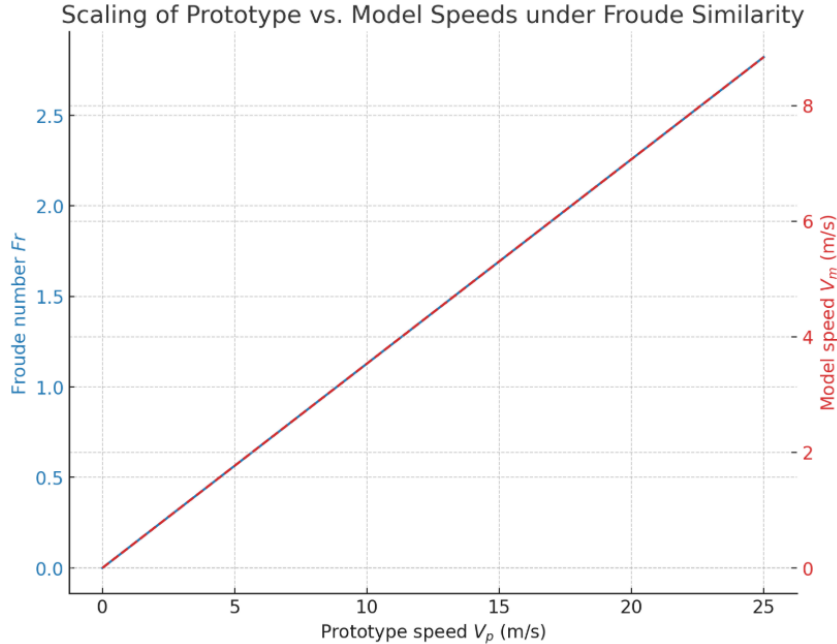


Figure C.3.: Prototype/model speed & Froude number relation

- Blue curve (left axis): The Froude number as a function of prototype speed up to 25 m/s.
- Red dashed curve (right axis): The corresponding model speed for a 1 m model, obtained by enforcing Froude similarity with the 8m prototype.

C.1.4. Implications of Testing at Matched Froude Number

Experiments with a model scaled according to Froude similarity ($Fr_m = Fr_p$) provide reliable insight into gravity- and wave-dominated phenomena. The following effects can be studied with good accuracy:

- Wave pattern, wake formation, and wave-making resistance as functions of Fr .
- Dynamic sinkage and trim with increasing speed; onset of porpoising and pounding in terms of Fr .
- Heave and pitch oscillations near resonance; likelihood of slamming events and their dominant frequencies.
- Spray formation envelopes and free-surface interactions governed by the $\sqrt{gL}$ scaling law.

Certain effects, however, are not fully captured by Froude scaling alone due to the mismatch in Reynolds number ($Re = VL/\nu$):

- Viscous resistance components, boundary-layer development, and transition, requiring frictional corrections.
- Scale-sensitive phenomena such as air entrainment, ventilation, and spray breakup.
- Aerodynamic lift contributions at very high Fr , where blowover or pitch divergence may occur.

C.1.5. Tabled Summary of Popular Hull Families

Category	Hull Type	Typical Uses
Displacement	Round-Bilge Hull	Sailboats, traditional yachts
Displacement	Full-Displacement Monohull	Cargo ships, trawlers
Displacement	X-Bow	Offshore supply ships, expedition vessels
Displacement	Displacement Catamaran	Passenger ferries, sailing cats
Displacement	Displacement Trimaran	Ocean-crossing yachts, ferries
Displacement	SWATH (Small Waterplane Area Twin Hull)	Offshore platforms, research vessels
Semi-Displacement	Semi-Displacement Monohull	Fast yachts, patrol boats
Semi-Displacement	Wave-Piercing Catamaran	High-speed ferries
Semi-Displacement	Slender Flat Bottom Hull	Naval ships, efficient long-range craft
Planing	Flat-Bottom Hull	Skiffs, small lake boats
Planing	Shallow-V Hull	Recreational boats, calm-water cruisers
Planing	Deep-V Hull	Offshore racing boats, performance yachts
Planing	Pad-V Hull	Bass boats, small sport boats
Planing	Step Hull	High-performance speedboats
Planing	Tunnel Hull (High Performance Catamaran)	F1H ₂ O racing, offshore racing
Planing	Hydroplane (3-Point)	Drag racing, circuit racing
Planing	Air-Entrapment / Canard Hull	Extreme racing prototypes
Specialized	Hydrofoil Hull	Fast ferries, experimental craft
Specialized	Surface Effect Ship (SES)	Military vessels, fast transports
Specialized	Hovercraft	Amphibious transport
Specialized	Piercing / Inverted Bow	Naval stealth ships, offshore vessels
Specialized	Proa / Asymmetrical Hull	Traditional sailing, rare in powered craft

Table C.2.: Overview of popular hull families per cruising mode

C.1.6. Deadrise

Within the planing regime, the bottom geometry of a mono-hull strongly influences its performance and seakeeping capabilities. The most defining geometric parameter is the deadrise angle — the inclination of the bottom panels relative to the horizontal plane, typically measured at the transom. Industry practice generally classifies a transom deadrise of approximately 20°–25° as a deep-V configuration. Deep-V hulls exhibit superior wave-piercing capability and ride comfort in rough seas due to their sharper entry and smoother vertical accelerations, but this comes at the expense of reduced static stability, greater draft, and increased wetted surface area, which together lead to higher hydrodynamic resistance.

In contrast, hulls with a smaller transom deadrise — typically less than 20° — are referred to as semi-V designs. These combine several advantages of deep-V and flatter-bottom hulls, offering a compromise between stability, efficiency, and seakeeping. At the opposite end of the spectrum, nearly flat-bottomed designs — characterized by negligible transom deadrise — provide excellent load capacity and minimal draft, making them efficient and stable in calm conditions. However, such hulls tend to experience strong slamming and high vertical accelerations in waves, resulting in limited comfort and seakeeping capability at speed.

In general, increasing the deadrise angle enhances seakeeping and wave-piercing perfor-

mance but decreases static stability and overall hydrodynamic efficiency. Consequently, the selection of the deadrise angle represents a fundamental design trade-off between flat-water efficiency and rough-water performance. Between these extremes lies an almost infinite range of intermediate geometries, including round-bilge and multi-chine variants, each tailored for specific operational and stability requirements.

Deadrise depends on:

- **Speed** — Different conditions at different speeds imply different optimal deadrise angles. How much efficiency does one want to sacrifice in exchange for cornering “grip”?
- **Wave conditions** — Wave-piercer required?
- **Step and lifting strake usage** — High-speed planing might advance from step and lifting strake integration, influenced by the deadrise angle (see Section 2.3.6).
- **Warping** — How does the hull deflect water? How does the deadrise angle propagate towards the stern?

C.1.7. What, SWATH?

Ship design has historically revolved around the balance between stability, buoyancy, and hydrodynamic efficiency. Conventional mono-hull vessels rely on a single main hull, offering high cargo capacity but exhibiting significant motions in rough seas due to large variations in buoyancy with wave encounter. Catamaran designs reduce this effect by splitting buoyancy across two slender hulls, which increases transverse stability and reduces wave-making resistance. However, their hydrodynamic performance remains strongly linked to the hull-water interaction at the free surface.

A more advanced concept is the Small Waterplane Area Twin Hull (SWATH) vessel, invented by Canadian Frederick George Creed [85]. The key principle of the SWATH design is to minimize the waterplane area by submerging most of the buoyant volume below the wave-affected region and connecting it to the deck with slender struts. In this configuration, buoyancy becomes less dependent on wave elevation, as the increase in displaced volume with increasing draft is relatively small. This results in improved seakeeping and a reduction in vertical accelerations, making

the platform significantly more stable than both mono-hull and catamaran designs. The ability to “pierce” incoming waves rather than ride over them (similar to hydrofoils) further enhances its stability in high sea states. These effects are further discussed on page 168, section Pounding. [86]

The next illustration provides valuable insight of the gain in roll stability for a particular hull type. Note the values shown are just for indicative purposes as this fully depends on the exact sea dynamics and hull geometry.

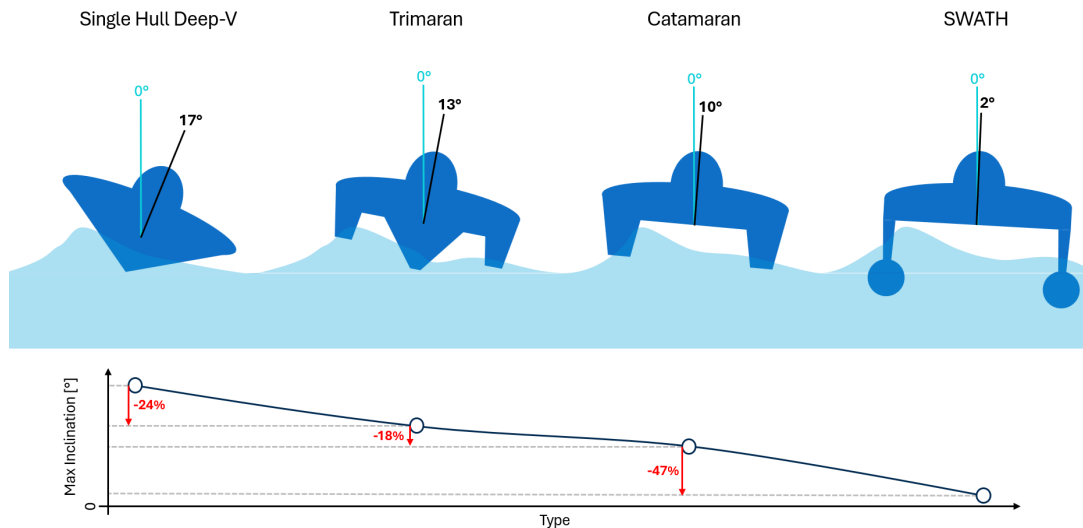


Figure C.4.: SWATH platform stability compared

Despite these advantages, the SWATH concept presents notable limitations. Because buoyancy is concentrated in submerged demi-hulls, changes in payload or vessel weight strongly influence draft. This makes SWATH vessels less suitable for heavy cargo transport, where large payload variability is common. The design is therefore better suited for applications where stability is critical and payload variation is limited, such as naval operations (Sea Shadow (IX-529), Sea Fighter (FSF-1)), research platforms (Type 751 Planet), and offshore services. Its highly stable deck offers a reliable platform for the launch and recovery of helicopters, or unmanned aerial vehicles, where vertical accelerations and deck inclination must be minimized. [85]

From an efficiency point of view, SWATH vessels achieve reduced resistance through minimized waterline contact, but further efficiency

gains can be obtained by reducing the cross-sectional area under the water line, stepping away from the SWATH design. So, despite the high stability gain and increased efficiency, the design proves not to be suitable for a racing prototype this project is aiming for.

One emerging direction to further increase efficiency at high speeds with a small vessel is to partially or fully lift a lightweight hull out of the water by combining hydrodynamic and aerodynamic lift. In this manner, vessels transition from displacing water to traveling partly through air, drastically reducing drag. Such hybrid concepts represent a step beyond the SWATH in efficiency, suggesting that while the SWATH offers a stable and robust solution for certain applications, the greatest potential for high-speed and efficient operation can be found in designs that minimize hull-water interaction entirely.

C.1.8. Catamaran

In the design of high-speed and autonomous racing vessels, the hull form must achieve an optimal balance between hydrodynamic efficiency, stability, and structural integrity. Stability in surface craft is generally governed by a combination of buoyancy distribution and the waterplane area. A wide beam increases the righting arm and thus enhances transverse stability; however, it also increases hydrodynamic resistance, particularly due to the enlarged wetted surface and greater wave-making potential. Consequently, simply increasing beam width is an inefficient way to achieve stability for high-performance racing hulls.

Multi-hull configurations, such as catamarans and trimarans, offer an elegant solution to this trade-off. By separating the buoyant volume into multiple slender hulls, a large effective waterplane width can be achieved while maintaining narrow, low-resistance hull forms. In a catamaran, the two demi-hulls (sponsons) are positioned laterally apart, providing a wide waterplane and therefore excellent initial stability, while each hull remains hydrodynamically efficient due to its slenderness ratio. From a hydrodynamic standpoint, this configuration combines high stability with low total resistance, which is the ideal combination for autonomous craft operating efficiently at high speed, where both seakeeping and control precision are critical.

However, these advantages introduce additional structural considerations. The wide separation of the sponsons results in significant torsional and bending loads across the cross-deck structure. A deck must resist twisting moments generated by differential wave impacts and asymmetric loading, making structural stiffness and material design crucial. For lightweight, high-performance craft — especially those built for autonomous operation — minimizing structural deflection is essential to maintain sensor alignment and preserve the fidelity of control and perception systems.

Naval engineers often follow a rule of thumb to ensure favorable roll behavior and seakeeping performance in multi-hull designs: both demi-hulls should remain immersed under typical operating conditions. This implies sufficient free-

board and draft in the sponsons to avoid loss of stability or abrupt dynamic behavior when encountering waves. In practice, catamarans tend to outperform trimarans in terms of stability and seakeeping under comparable displacement and beam conditions, as both demi-hulls contribute continuously to buoyancy and restoring moments. Nevertheless, a poorly optimized catamaran design — particularly one with excessive beam or insufficient draft — may suffer from intermittent hull emergence, resulting in reduced righting energy and undesirable roll accelerations [87].

In racing or autonomous applications, where rapid course corrections and minimal roll motion are required for accurate sensor perception, such instabilities are especially problematic. The initial stability, characterized by the metacentric height and righting arm, becomes a key design criterion for assessing lateral accelerations typical for catamaran configurations. To mitigate excessive roll motion, designers frequently employ passive or active control devices such as bilge keels, roll-damping fins, or adjustable trim tabs. These features contribute to smoother dynamic response, improving sensor reliability, thereby making the catamaran hull a favorable platform for modern autonomous racing vessels. In conclusion: to advance from the concept of catamarans and their stability while keeping good cornering and benefiting from enlarged ship effects [88] as well as having a increased slenderness ratio, the catamaran may not be too wide. The commonness of slender racing cats and racing tunnel hulls affirm that idea.

In summary, the catamaran features the following advantages:

- **Large waterplane stability** — the wide separation of demi-hulls is minimizing roll motions. Reduced heave and pitch amplitude improve data consistency (LiDAR, cameras and GPS).
- **Spacious cross deck** — provides ample area for solar panels, lift surfaces, and aerodynamic lift (tunnel).
- **Reduced wave-making resistance** — slender demi-hulls generate less wave interference at higher Froude numbers, improving hydrodynamic efficiency.
- **Favorable wave interference** — proper spacing of hulls can cause partial cancellation of divergent waves.
- **Improved sensor performance** — stable deck and large beam allow more optimal sensor place-

ment and minimal interference between instruments.

- **Damage resilience** — if one hull is compromised, the vessel typically retains buoyancy and limited controllability.
- **Flexible payload distribution** — wide beam enables balanced weight distribution along the centerline for stable operation.
- **Predictable high-speed handling** — minimal heeling in turns improve course-keeping and autonomous path tracking.
- **Superior seakeeping in moderate seas** — slender hulls reduce slamming and impact loads, allowing higher average speeds in rough water.
- **Ease of modular construction** — hulls and deck modules can be separated for transport, maintenance, or rapid prototyping, depending on the design.

and disadvantages:

- **Structural implications** — the sponson–cross-deck connection requires significant reinforcement to handle torsional, bending, and slamming loads, increasing structural complexity. Uneven wave impact or asymmetric loading can induce twisting and bending moments in the bridge deck.
- **Limited tight cornering** — due to the wide hull spacing, making quick heading corrections and tight turns is more difficult. Cause: higher yaw inertia and high lateral grip.
- **Potential interference drag** — hydrodynamic interaction between the two hulls can increase viscous and wave resistance if spacing is suboptimal.
- **Heavier overall structure** — additional material is required for stiffness and to prevent cross-deck deflection under racing loads compared to mono-hulls.
- **Wider beam** — complicates transport, launch, and docking operations.
- **Slamming under the tunnel** — in rough seas, wave impacts on the bridge deck can cause dynamic loading, noise, reduced comfort and may affect sensitive electronics.
- **Complex hydrodynamic optimization** — achieving balanced lift, trim, and interference control between demi-hulls requires extensive CFD tuning and empirical validation.
- **Limited roll compliance** — while stable, the hull might transmit more vertical accelerations to the sensors and electronics during high-frequency wave encounters. Cautious modeling is required for proper mitigation (e.g., inclined planing surfaces for wave cutting/dampening).

Note

In tunnel hull and catamaran design, the outer hulls are often termed sponsons in racing practice, a carryover from naval usage describing projections from a vessel's sides. In hydrodynamic and academic literature, the term demi-hulls is more common, emphasizing their role as full lifting hull forms analogous to catamaran hulls. Sponsons is a term often used in racing and less precise alternatives such as pontoons, amas, floats, or outriggers are sometimes used in casual or explanatory contexts, but are avoided in technical descriptions.

C.1.9. Trimaran

Trimaran configurations represent a natural evolution of the multihull concept, combining aspects of both monohull and catamaran hydrodynamics. A trimaran typically consists of a central main hull, or *vaka*, accompanied by two smaller outer demi-hulls (in this context *amas* is more common). This arrangement provides a hybrid balance between stability, efficiency, and structural feasibility. In contrast to catamarans, where the buoyant volume is symmetrically distributed across two equally sized hulls, the trimaran relies primarily on the main hull for buoyancy and load carrying, while the *amas* contribute mainly to transverse stability and dynamic roll damping.

From a hydrodynamic perspective, the trimaran benefits from the slenderness of its central hull, which maintains low wave-making resistance and efficient flow characteristics similar to a mono-hull (usually longer than a comparable catamaran). The smaller *amas*, located at a moderate lateral distance, should remain partially submerged during steady conditions. This configuration yields less wetted surface area compared to a catamaran of equal beam, and therefore consequently slightly lower viscous resistance at cruising speed. At the same time, when the vessel heels or experiences asymmetric wave loading, the *amas* progressively contribute buoyancy, providing a soft and predictable restoring moment that improves comfort and roll control. The Torpedo Mk.II EVO has roll issues while cornering because of the poorly dimensioned *amas*.

For high-speed or autonomous racing applications, stable roll behavior can be advantageous. The trimaran's narrow central hull minimizes drag and promotes efficient acceleration, while the outrigger hulls act as stabilizing elements that only generate lift when necessary. This dynamic form of stability reduces the risk of asymmetric drag penalties at low heel angles and usually enables sharp, efficient maneuvering.

Nevertheless, these benefits are accompanied by several design and operational challenges. The partial immersion of the *amas* implies that the vessel's stability margin is lower than that of a catamaran of comparable displacement. When

roll angles exceed the design threshold, one *ama* may emerge entirely, sharply reducing righting energy and potentially inducing large roll accelerations. For this reason, the lateral spacing, volume, and immersion depth of the *amas* must be optimized to balance stability, drag, and structural load transmission. If the *amas* are positioned too far from the centerline, they experience high vertical loads and slamming forces in rough seas; if too close, their hydrodynamic interference with the central hull increases total resistance.

Structurally, the trimaran introduces complex load paths between the main hull and the *amas*. The connecting beams must resist both bending and torsion induced by differential wave impacts and dynamic coupling between hulls. These structural demands are particularly relevant for lightweight autonomous platforms, where maintaining stiffness and alignment is essential to protect sensors, control electronics, and propulsion systems from vibration and misalignment.

In practice, trimarans can achieve a high ratio of stability to drag if the cross-deck geometry, *ama* volume distribution, and longitudinal positioning are carefully designed. Their hydrodynamic versatility makes them suitable for hybrid operation modes—planing, semi-displacement, or foil-assisted—depending on the performance envelope required just as catamarans do. However, achieving optimal trim and roll characteristics requires precise control of mass distribution and often implies active damping strategies.

In summary, trimarans occupy an intermediate position between mono-hulls and catamarans [73], offering a refined compromise between slenderness, stability, and efficiency. For autonomous racing vessels, they present an attractive alternative when compact dimensions, high maneuverability, and directional stability are prioritized. Yet, because of the higher design sensitivity to hull geometry and load distribution, their development demands careful simulation, experimental validation, and structural optimization

before competitive deployment and is not very flexible in changing the load and its distribution.

In essence the advantages include:

- **Slender main hull** — offers low wave-making resistance, similar to a mono-hull.
- **Dynamic stability** — amas contribute buoyancy only when heeled or under asymmetric loading, reducing drag in straight line sprints in calm sea conditions.
- **Reduced wetted surface area** — smaller total hull area compared to an equivalent catamaran, lowering viscous resistance.
- **Predictable roll behavior** — progressive righting moment from the amas provides smooth, controllable roll damping.

Disadvantages include:

- **Reduced initial stability** — only the central hull is fully buoyant, making the vessel more sensitive to lateral accelerations at rest.
- **Complex structural loads** — crossbeams must withstand torsion and bending between the main hull and amas.
- **Risk of ama emergence** — excessive heel or wave impact may lift one ama completely, influencing righting energy.
- **Hydrodynamic interference** — poor ama placement can cause wake overlap, instability, increased submersion while cornering, and increased resistance.
- **Higher design sensitivity** — stability and roll response depend strongly on precise volume, spacing, geometry, and immersion depth of the amas and become even more complex in planing regimes as submersion must be limited.

C.1.10. Common Racing Hull Types

Racing Hull Type	Subtypes/Examples	Shape & Lift	Speed Range	Use Case	Strengths	Weaknesses
Hydroplane (3-point)	Drag Hydroplane, Unlimited Hydroplane	Two front sponsons + 1 rear skid; rides mostly on air	200–500 km/h	Straight-line drag racing	Minimal drag, extreme top speeds, air-lifted hull	Very unstable in rough water- Limited or no turning ability
Short Closed Catamaran / Tunnel Hull	F1H ₂ O (tunnel)	Catamaran with aerodynamic air tunnel	220–300 km/h	Closed-course circuit racing	High lift, fast cornering, tight lateral grip	Prone to blowover (tunnel)- Less agile than tilting monohulls in tight turns
Long Closed Catamaran / Offshore Racing Catamaran	Offshore racing cats	Long twin hulls with air cushion	180–320 km/h	Offshore endurance races	Stable at high speed in moderate chop, very fast	Wide beam, less agile in tight turns
Deep-V Monohull	Cigarette boats, offshore racers	Single sharp V hull, planing surface	150–220 km/h	Offshore, rough-water racing	Cuts through waves- Good stability in chop	More wetted surface- Less top speed than cats
Pad-V / Mod-Vee Hull	Bass boats, small sport boats	Flatter aft section, small planing “pad”	100–160 km/h	Lake racing, fast recreation	Simple & cost-effective, good acceleration	Poor performance in waves- Less stable at very high speed
Specialized Hulls	Air-entrapment, Hydrofoils, Canards	Specialised, (partially) submerged wing	60–300+ km/h	Records, experiments, ferries	Very low drag, novel lifting principles	Rare in competitive racing, costly maintenance and increased design complexity

Table C.4.: Overview of racing hull types

C.1.11. Performance Hull Anatomy

In this section only project-relevant parts, shapes, types, and terms are introduced in order to clarify the vocabulary used in the following chapters. The terms are grouped by category, their typical position on the vessel is indicated, and each is briefly described. This overview serves as a reference to the detailed hydrodynamic discussions that will follow.

General Nautical Terms

- **Bow** - The forward (front) end of the vessel.
- **Stern** - The aft (rear) end of the vessel.
- **Port** - The left side of the vessel when facing forward.
- **Starboard** - The right side of the vessel when facing forward.

Structural and Hull Features

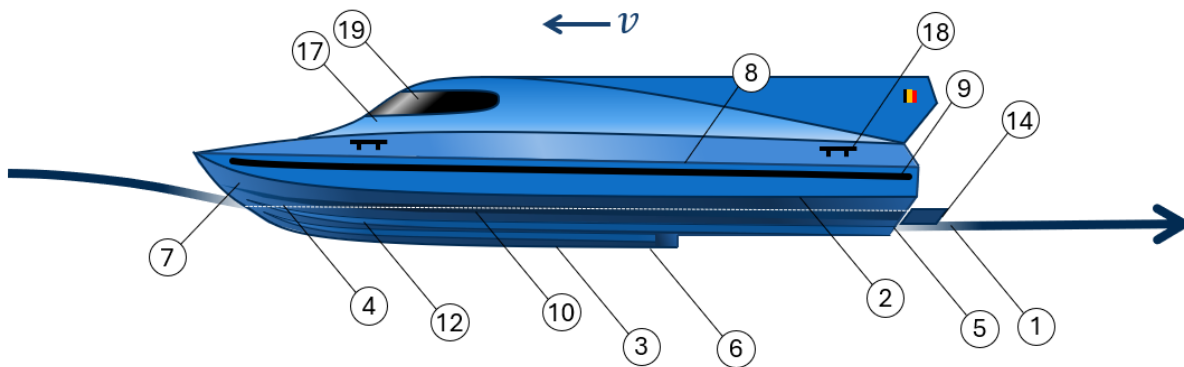


Figure C.5.: Performance tunnel hull anatomy

1. **Tunnel** - The air channel formed between demi-hulls in a catamaran, providing aerodynamic lift and stability.
2. **Chine** - The sharp angle along the hull side where the bottom meets the topsides, influencing spray deflection and roll stability.
3. **Keel** - The central lowest longitudinal line of the hull bottom, providing alignment and directional stability.
4. **Waterline** - The line separating submerged and unsubmerged parts of the hull at rest or in motion.
5. **Transom** - The flat vertical or near-vertical surface forming the stern of the hull, where the propulsion system is mounted.
6. **Bilge** - The lowest interior part of the hull where water may collect.
7. **Hull** - The main watertight body of the vessel, excluding fittings and superstructure.
8. **Gunwale** - The upper edge of the hull sides, often reinforced for structural strength.

9. **Rub rail (bumper)** - A protective strip running along the gunwale to absorb impacts against docks or other vessels.
10. **Sprayrail** - A small longitudinal ridge on the hull bottom or sides designed to deflect spray outward and increase lift.
11. **Deadrise** - The angle between the hull bottom and a horizontal plane, measured at the transom; determines how “V-shaped” the hull is.
12. **Lifting strake** - Longitudinal protrusions on the hull bottom that enhance hydrodynamic lift and stability.
13. **Wetted surface** - The portion of the hull in direct contact with water, critical to resistance and efficiency.
14. **Trim tabs** - Adjustable plates on the transom that control hull trim and attitude by changing lift at the stern.
15. **Propeller** - The rotating device mounted on the drive shaft that converts engine torque into thrust.
16. **Rudder** - A steering surface, typically aft of the propeller, that deflects flow to control yaw.
17. **Helm** - The steering control interface (wheel, tiller, or joystick) used by the operator to command rudder or drive angle.
18. **Cleat** - A deck fitting used for securing ropes and mooring lines.
19. **Visor** - A cockpit fitting to protect the pilot and advance aerodynamic efficiency.

Step Geometry

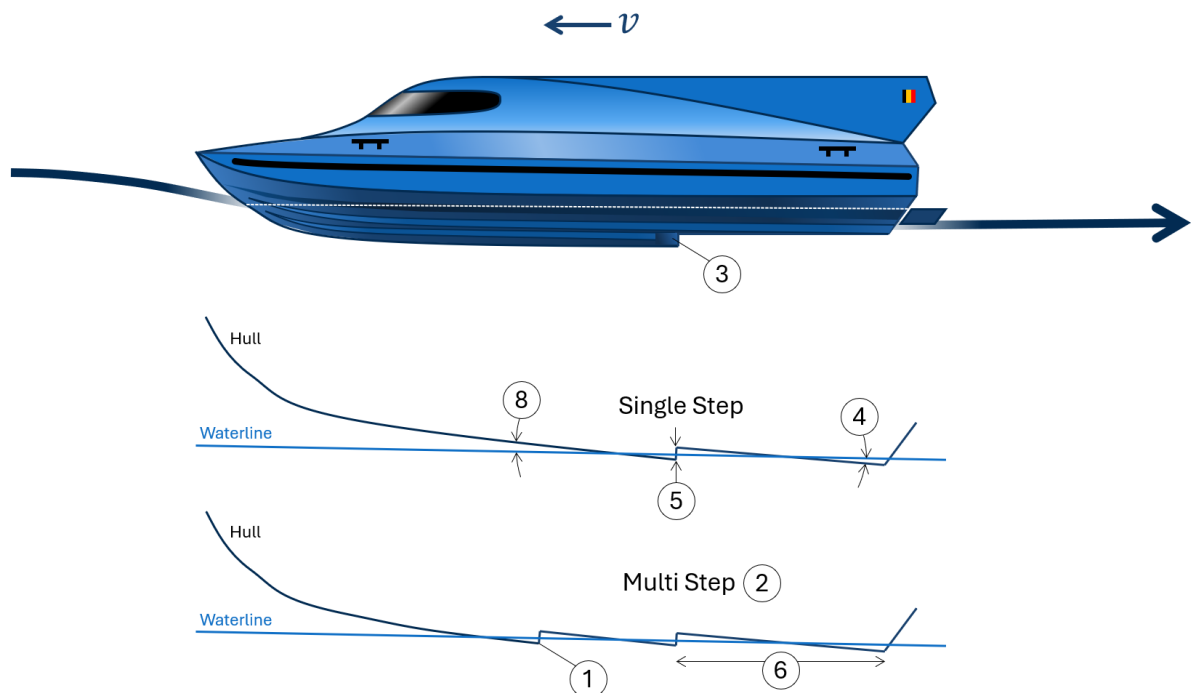


Figure C.6.: Steps geometry

1. **Steps** - Transverse notches in the bottom of the hull that introduce air under the running surface to reduce wetted area and drag.
2. **Multi-step** - A hull incorporating more than one step to further break up wetted surface and improve efficiency at high speeds.
3. **Step ventilation channel** - A channel feeding air to the step, improving aeration and ensuring clean flow separation behind the step.
4. **Step angle** - The angle of the step relative to the hull bottom, influencing flow separation and reattachment.
5. **Step height** - The vertical offset between the forward and aft running surfaces at the step.
6. **Step length** - The longitudinal distance over which the step extends across the hull bottom.
7. **Motor trim** - The angular adjustment of the outboard or sterndrive relative to the transom, controlling thrust line and hull trim. See Section 2.8.3.
8. **Trim angle (angle of attack)** - The angle of the hull bottom relative to the water surface when underway.

Lower Outboard Terminology

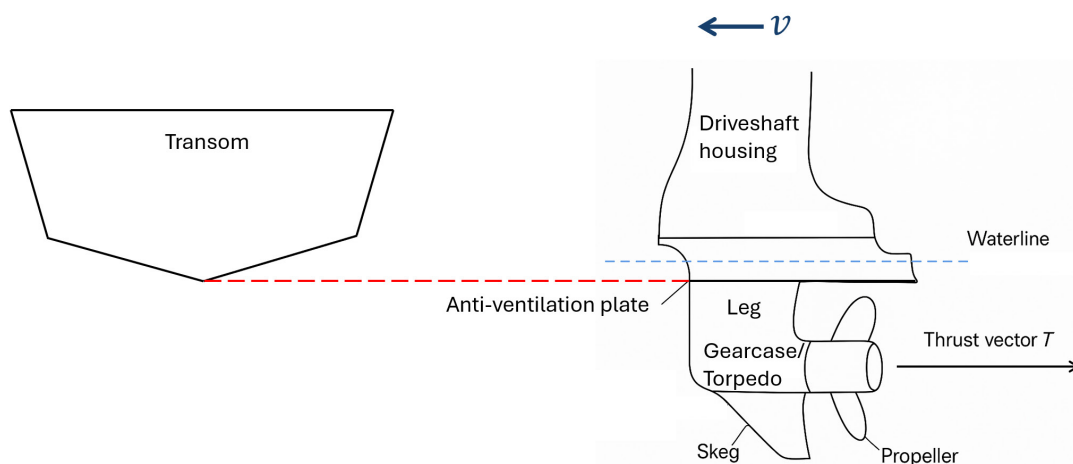


Figure C.7.: Lower outboard terminology

C.1.12. Comparison of Stability Characteristics

Configuration	DOF	Stability Characteristics	Control Method
Hull-only	Heave / Pitch	Stability from hull pressure distribution and trim; prone to porpoising at high F_n	Geometry, weight distribution
Hull-only	Roll	Stability from beam and chines; chine walking possible in vee-hulls	Beam, spray rails
Foil-assisted	Heave / Pitch	Foils unload hull, sensitive to immersion and angle of attack	Foil geometry, fixed incidence
Foil-assisted	Roll	Symmetry of foil lift critical; asymmetric ventilation induces instability, but the hull controls roll (stern) in normal operation	Foil placement
Passive foil	Heave / Pitch	Fully foil-borne, but no adjustment; sensitive to waves and disturbances	Fixed foil design
Passive foil	Roll	Limited inherent stability; recovery relies on geometry: water piercing foil	Difference in wing submersion
Active foil	Heave / Pitch	Active feedback stabilizes flight; suppresses disturbances	Similar to aircraft, elevator control
Active foil	Roll	Roll actively damped; stability margin highest of all configurations	Similar to aircraft, aileron control
Active foil	Yaw	Managed via differential foil lift or rudders	Similar to aircraft, rudder-aileron control

Table C.5.: Comparison of stability characteristics across hull-only, foil-assisted, passive, and active foil configurations in relation to degrees of freedom.

C.1.13. Stabilization Techniques

For most ships, roll stabilization is most critical. In the next paragraphs, the most common techniques are listed alongside with their potential and drawbacks.

Shape of the hull The initial and overall stability of a floating body is primarily governed by the relative positions of the center of buoyancy (CoB) and the center of gravity (CG). The CoB is the centroid

of the displaced water volume, and its movement with heel angle depends directly on the underwater hull geometry. The metacentric height, GM , defined as the distance between the metacenter and the CG (see Figure C.9), provides a first-order measure of transverse stability (small-angle). A larger GM corresponds to greater righting moments at small angles of heel. A graphical representation of an inherent stability is given by Figure C.8 (inspired by Shyh-Kuang Ueng's diagram [89]).

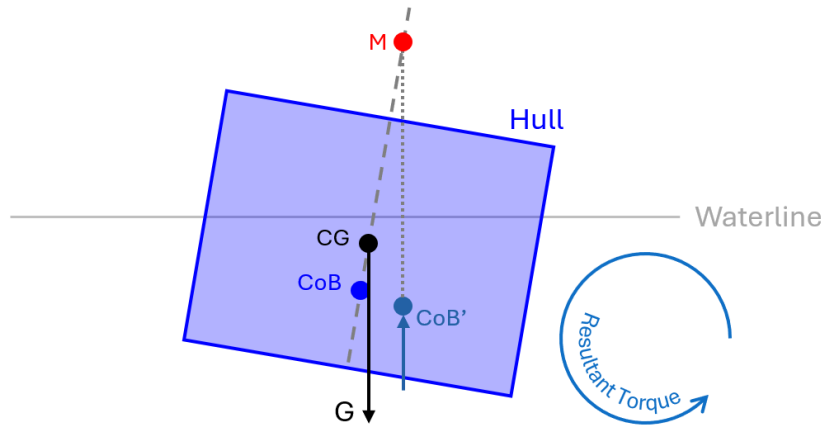


Figure C.8.: Self righting behavior of an inherently stable hull

- **Rectangular bodies** — Vessels with prismatic, rectangular cross-sections exhibit large waterplane areas and thus large shifts of the CoB with heel. This results in a high initial metacentric height, which provides strong righting moments and excellent transverse stability. Such forms are common in barges and pontoon-type platforms, where stability outweighs hydrodynamic efficiency. However, the disadvantage is increased drag due to blunt waterlines.
- **Round hulls** — Circular or round-bilged cross-sections, typical of traditional displacement ships and submarines, generate only limited movement of the CoB with heel. This leads to lower metacentric heights and reduced initial stability. The advantage of such hulls lies in their hydrodynamic efficiency, since the smooth curvature minimizes viscous resistance and wave-making. Over a roll range, these hulls are generally less roll-efficiency coupled, making it ideal for sailing. Round hulls therefore achieve low drag at moderate speeds, but require careful placement of the CG (ballast, keel) to ensure adequate stability.
- **Wide, expanded round vessels with tumbelhome or negative flare** — Large sailing battleships of the 17th–19th centuries adopted broad, flared round hulls in order to carry heavy artillery high above the waterline while maintaining sufficient stability. The expanded beam increased the waterplane area, producing higher GM values and ensuring safe operation under sail despite a high vertical CG due to multiple decks, masts, and cannons. The compromise was a reduction in speed potential and efficiency compared to finer merchant hulls, but the geometry delivered the required stability for combat platforms.
- **Wide flat bottom** — Wide flat bottoms offer a high roll stiffness and potential for efficient planing behavior, accompanied by the drawback of large resistance at low speeds compared to other hull forms.
- **Multihull** — An alternative solution to the stability–efficiency trade-off is the multi-hull concept. Catamarans and trimarans achieve high stability not through large individual waterplane areas, but through widely spaced hulls. Waves heave is averaged over the span of a vessel, thus decreasing roll. The separation of buoyant volumes produces very large righting arms even at small heel angles, resulting in exceptional transverse stability without resorting to excessively broad or inefficient individual hulls. This geometry allows multi-hulls to combine slender, low-resistance hull

forms with high transverse stability, making them highly competitive in modern sailing, passenger ferries, and high-speed autonomous vessels. Note that stability is not only about limiting the angle and positional variance, but also in dampening. More specifically, a tunnel hull, for example, is an Air Cushion Vehicle (ACV) configuration that dampens heave and pitch oscillations. Similarly, sharp deep-V hulls tend to dampen wave landings and limit wave heave/pitch accelerations on impact.

Metacentric height GM is the small-heel measure of transverse stability.

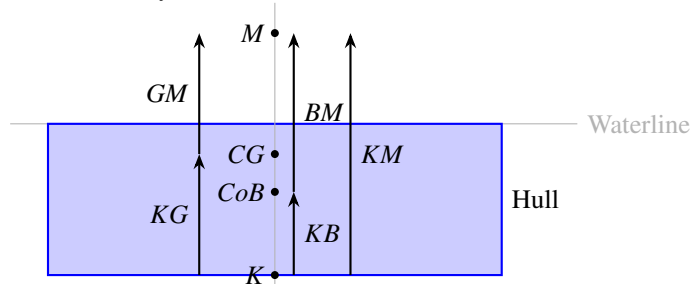


Figure C.9.: Schematic definition of GM : K (keel), CoB (center of buoyancy), CG (center of gravity), M (metacenter).

Active stabilization fins

Stabilizer fins are appendages fitted to a vessel's hull to reduce roll motions by generating hydrodynamic lift forces that oppose angular displacements. Their action is inherently flow-speed dependent: at higher forward speeds, the relative flow over the fin surface produces lift proportional to the square of the velocity, providing effective damping of roll oscillations. At low flow velocities, however, the fins lose effectiveness, requiring alternative stabilization methods.

The integration of stabilizer fins introduces added complexity in design and operation. They require control systems to adjust fin angle actively, balancing effectiveness against drag penalties. Poorly configured fins may increase resistance or induce undesirable yaw moments, highlighting the importance of careful hydrodynamic design.

Hydrofoils, traditionally employed for dynamic lift and drag reduction, can also serve a stabilizing function. When configured laterally, small hydrofoil surfaces generate moments similar to stabilizer fins, but usually with greater authority at speed. This depends on the intended function of the hydrofoil (assist versus passive/active). This dual role allows hydrofoils to enhance both efficiency (by reducing wetted sur-

face) and seakeeping performance (by damping roll), though it also magnifies the demands on structural strength, control algorithms, and actuation systems depending on the type of configuration implemented.

Active tank stabilizer

Active tank systems reduce roll motions by transferring water between port and starboard tanks in response to the vessel's oscillation. When excessive roll is detected, water is pumped (active) or allowed to flow (passive) across transversely, generating a counteracting moment that damps the motion. Such systems are effective in compensating for wave-induced oscillations, particularly at low speeds or when the vessel is stationary, where flow-dependent devices such as fins are ineffective.

The drawbacks of active tanks lie in the added structural weight, volume requirements, and ineffectiveness for the persistent roll moment during cornering maneuvers. Because the water mass responds with inertia, the system may amplify rather than dampen turning-induced roll, making it unsuitable for vessels requiring rapid maneuverability. Consequently, active tanks are best applied in conditions dominated by regular

wave excitation, where their ability to mitigate oscillatory roll enhances passenger comfort and operational safety without imposing significant dynamic penalties.

Gyroscopic stabilizer

Gyroscopic stabilizers rely on the angular momentum of a high-speed rotating flywheel to generate roll-counteracting moments through gyroscopic precession. When the vessel rolls, the gyroscope's gimbal mechanism is actuated, producing a torque that resists the roll motion. So, the flywheel is limited to roll (axis fixed in the hull) but is free to pitch. This approach does not depend on forward speed and can therefore be highly effective at zero or low speed too, where flow-based devices are inoperative.

The main disadvantages are cost, weight, and technical complexity. Gyroscopic stabilizers are precision devices requiring robust bearings, control systems, and continuous monitoring to prevent imbalances. Improper alignment or maintenance can lead to high-frequency vibrations transmitted to the hull structure. This induces a negative effects in various sensors for autonomous operation. While their added weight is generally less than that of active tanks, the devices are quite expensive to install and maintain. Moreover, gyroscopic stabilizers are less effective during prolonged steady turns, where the persistent roll moment cannot be fully countered by precession alone. Remark that a higher rotational speed requires less inertia, meaning less added mass, to generate the same counteracting torque. The goal of a compact, lightweight solution is thus having the rotor run at the highest possible rpm. The demand for specialized features like vacuum chambers or magnetic bearings further increases costs.

Despite these limitations, gyroscopic stabilizers

are increasingly employed in luxury yachts, research vessels, and other specialized craft, where their ability to reduce roll without any reliance on forward speed justifies the investment.

Reaction wheel

Reaction wheels function on the same fundamental principle of counteracting angular momentum as gyroscopic stabilizers, but instead of relying on precession through gimbals, they directly change the speed of rotating flywheels to generate counteracting roll moments. A reaction wheel is therefore not a dual axis device, but uses only one longitudinal axis fixed in the hull. By accelerating or decelerating the wheel, torque is transferred to the vessel structure in the opposite direction of the roll. A reaction wheel mechanical system coupling (rotor-hull) is found in the motor.

Although the concept is mechanically straightforward, reaction wheels are rarely practical in marine applications. To provide sufficient roll control, the flywheels must carry substantial inertia, resulting in prohibitive size and mass penalties. The system is also very energy demanding, as continuous acceleration and deceleration cycles are required to track roll oscillations. Furthermore, the added mass competes with payload capacity and lowers overall efficiency.

For these reasons, reaction wheels are typically considered impractical for ship stabilization, despite their use in aerospace and satellite applications where vacuum environments and precise attitude control justify the mass and energy cost. In naval contexts, they are studied primarily as theoretical analogues or as experimental concepts rather than as operational systems.

An overview of the (roll) stabilizer systems included above are summarized in Table C.6.

System	Effectiveness range	Added mass	Relative cost	Limitations / Remarks
Rectangular / prismatic wide-beam hull	Effective at all speeds; very high initial GM from large waterplane	Low (geometry-driven)	Low	High calm-water resistance; poor seakeeping at speed; prone to slamming; suited to barges/pontoons where stability outweighs efficiency
Round-bilged mono-hull (+ ballast keel)	Effective ultimate stability in seaway; low initial GM (small-heel) but efficient hydrodynamics	Moderate–High (ballast mass)	Moderate	Deeper draft; slower roll period (comfort) but less transverse stiffness at small heel; not ideal for high-speed planing
Expanded round / beamy flared hull (e.g., sailing battle-ships)	Effective at all speeds; high initial GM to support high topside weights	Moderate–High	High	Increased wave-making and form drag; harsher motions; structural mass/complexity for broad decks
Multi-hull (catamaran / trimaran)	Effective at all speeds; exceptional transverse stiffness from hull spacing (large righting arm at small heel)	Moderate (bridgedeck + connections)	High	Very wide beam (berthing/turning constraints); tunnel/bridgedeck slamming if clearance inadequate; significant torsional loads in bridging structure
Stabilizer fins	Effective at medium to high forward speeds	Low to moderate	Moderate	Ineffective at low or zero speed; adds drag; requires active control
Hydrofoils (as stabilizers)	Effective at high speeds; dual role (lift + roll damping)	Moderate to high (structural)	High	Structural complexity; susceptible to impact damage; may affect maneuverability
Active tanks	Effective at low or zero speed for oscillatory roll	High (water mass + structure)	Moderate	Persistent roll moment in turns; unsuitable for maneuvering craft; space-consuming
Gyroscopic stabilizers	Effective at all speeds including zero	Moderate	Very high	Ineffective in prolonged steady turns; requires precision balancing; potential vibration issues
Reaction wheels	Theoretical roll control at all speeds	Very high (large inertia required)	Impractical	Excessive size, mass, and energy demand; primarily studied in aerospace, rarely used in ships

Table C.6.: Comparison of stabilization systems for marine vessels.

C.1.14. Bernoulli Based Flow Theory

As highlighted by DMS Marine Consultant [12] and other naval hydrodynamic studies, the effectiveness of a hull form is strongly governed by the minimization of changes in fluid velocity along the hull. In practical terms, this translates into smooth surface curvature and slender proportions. A long and narrow hull with the same displacement as a shorter and wider hull will induce smaller accelerations in the flow, thereby reducing resistance. The advantage of slenderness is clearly illustrated in competition rowing shells, where the athlete is positioned on top of

the hull to allow an extremely narrow design with minimal beam. Another outcome of slenderness is a reduction of the resistance “hump” that occurs near a speed-to-length (S/L) ratio of 1.34 (used in yacht design), $\approx V/\sqrt{L_{WL}}$ (imperial units, kn & feet), corresponding to a Froude number of $Fr \approx 0.40$. This softening of the hump due to a slender design, also observed in the Boat Curve Dashboard (BCD) software analysis, reflects smoother pressure transitions and reduced wave-making in practical tests.

A second principle indicated by DMS Marine Consultant [12] relates to the orientation of the hull surface relative to the flow. Besides minimizing velocity changes, it is advantageous to arrange surfaces so that they do not generate excessive longitudinal force components that oppose motion. In essence, hull curvature should deflect flow smoothly rather than obstruct it. An important remark is that most of the theory explained in this section is only applicable to displacement analysis of hulls, not to planing behavior (except for hydrodynamic pressure support). A schematic representation of streamline curvature and local pressure interaction around the hull surface can be seen in Figure C.10, which illustrates how laminar flow theory informs practical hull design.

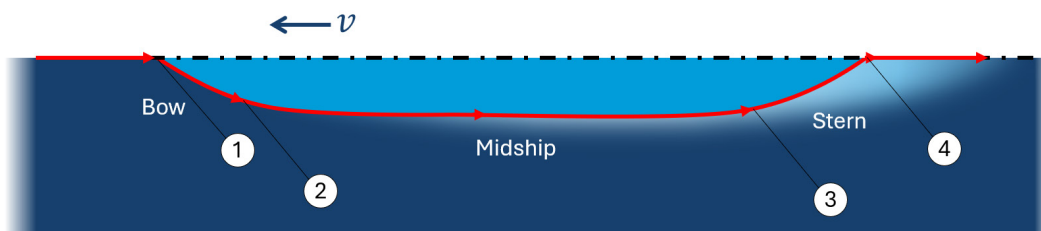


Figure C.10.: Basic flow around a hull

In this paragraph the pressure and velocity components of a point in the flow making contact with a hull are analyzed. The path is highlighted in red. Transition layers will be ignored as this is a simplified macro evaluation in a 2D-plane, with constant hydrostatic pressure. The velocity vector magnitude is the inverse of the rescaled and shifted pressure distribution [14] described below.

1. In front of point one, the pressure rises exponentially when approaching the bow with an initial relative velocity, reaching its hull pressure peak in point one, where the velocity becomes zero.
2. During the transition of the particle from point one to point two, the pressure drops below zero reaching its local minimum in point two. In contrast, the velocity rises above the initial velocity. Bernoulli indicates the increased pressure on the bow just after point one, as the flow is deflected. The deflection is responsible for increasingly reduced dynamic pressure longitudinally and increased pressure transversely.
3. Traveling from point two to point three, the midship effects are observed. In this stage, the pressure is more constant. Hull pressure is slightly negative compared to the undisturbed flow pressure in the environment (reference pressure 0 Pa) and the flow velocity of a particle along this section is slightly higher than the initial

velocity. Equation 2.7 proves this statement. As hydrostatic pressure is constant and the flow velocity is higher than the reference — therefore the dynamic pressure — then the static pressure on the hull should be lower than the reference static pressure as the sum needs to be constant. If skin friction is ignored, a long parallel midbody can be assumed as a good low resistance geometry, not suffering from excessive pressure drag.

4. In this diagram the stern and bow are nearly identical. So, the pressure and ve-

locity distribution is almost a mirror image with a positive pressure peak at point 4 (reduced dynamic pressure longitudinally and increased pressure transversely). This implies that ideally, the resultant forces on the hull turn zero, having no combined flow resistance at all. What is happening at the stern is somewhat counterintuitive, as the flow pushes the body forward. That is what is known as energy regeneration in naval engineering. Note that the peak of this pressure is often topped off due to turbulence or so called wake, limiting the energy regeneration.

An in-depth theoretical flow analysis is out of the scope of this study and will be addressed by using computational fluid dynamics or CFD. The more detailed mathematical synthesis on the flow around a hull can be found in the book “Principles of Naval Architecture Series: Ship Resistance and Flow” by Lars Larsson and Hoyte C. Raven [14]. The pressure and velocity distribution for a similar diagram is given on page 20 of the book.

C.1.15. Savitsky, Clement, and Navier-Stokes

- **Savitsky (1964)** — Daniel Savitsky introduced a widely adopted analytical method for predicting the hydrodynamic performance of planing hulls, accounting for lift, resistance, and trim in steady-state planing conditions on flat water [18]. His work provided the first systematic approach that linked hull geometry to running attitude and resistance, and it remains a cornerstone of planing craft design.

Savitsky’s analysis treats a prismatic planing surface (two symmetrically angled surfaces) with constant deadrise β moving at constant speed V and trim angle θ . By equating forces and moments, he established a method to predict the running trim, lift, and resistance. The governing relations can be summarized as follows [18]:

$$C_L = \frac{W}{\frac{1}{2} \rho V^2 b^2} \quad (C.3)$$

C_L = lift coefficient
 W = vessel weight [N]
 ρ = fluid density [kg/m³]
 V = forward velocity [m/s]
 b = wetted beam at the transom [m]

$$\frac{L_k}{b} = f(C_L, \beta, \theta) \quad (C.4)$$

L_k = wetted keel length [m]
 b = wetted beam at transom [m]
 $f()$ = empirical relation including lift C_L , deadrise β , and trim θ

$$C_D = C_f + \frac{C_L}{\tan(\theta)} + \Delta C_{\text{spray}} \quad (\text{C.5})$$

C_D	= total drag coefficient (dimensionless)
C_f	= skin-friction coefficient (from boundary-layer theory)
$\frac{C_L}{\tan(\theta)}$	= induced drag term due to trim angle
ΔC_{spray}	= empirical spray drag correction

Here, the lift coefficient C_L is determined from the requirement that hydrodynamic lift balances vessel weight. The wetted keel length ratio L_k/b is evaluated empirically as a function of deadrise, trim angle, and C_L , reflecting the reduction of wetted area with increasing speed. The total drag coefficient C_D combines viscous skin-friction, pressure-induced drag, and a spray correction term calibrated against experiments.

From these relations, Savitsky demonstrated that both the running trim θ and wetted geometry can be solved iteratively for a given speed and displacement, yielding estimates of resistance and performance. One of the critical insights was the existence of an “optimum trim angle,” typically in the range of 3° – 5° , where resistance is minimized due to the balance between lift generation and spray losses. This trim angle of minimum resistance is still used as a benchmark in the design of modern planing craft and in the calibration of active trim-control systems.

Unfortunately, in practice, a varying deadrise along the keel makes the direct implementation of this theory more difficult. The Shark is a true planing craft (with a displacement bow), but adoption of Savitsky’s analysis is complex as the hull is curved and has no sharp keel. As a simplification, one could work with the curves of a small to non-existing deadrise, but only computational fluid dynamics (CFD) and high-speed model testing can truly make an accurate estimation of the (optimal) trim, lift, and wetted area relation.

- **Clement (1960s)** — Eugene Clement proposed modifications to conventional planing hull forms — principally the introduction of cambered planing surfaces — that led to measurable reductions in resistance at high speeds. His innovations emphasized improved pressure recovery and smoother flow detachment at the stern, resonating with later CFD-based design studies by C. D. Gray [90, 91].
- **Navier–Stokes (1822)** — Claude-Louis Navier and George Gabriel Stokes formulated the fundamental partial differential equations governing viscous fluid motion. The famous Navier–Stokes equations generalize Newton’s second law to continuous fluids, accounting for inertia, pressure, viscous stresses, and external forces [92, 93]. While an exact analytical solution for ship flows remains unattainable due to turbulence and free-surface complexity, these equations form the basis of modern CFD for hull resistance and seakeeping studies.

C.1.16. Common Bow Types

In the first place, the conventional bow type, as seen on most (larger) vessels. They are often designed from an above and below the surface perspective. Below the surface flat water hydrodynamics matter most. If the ship operates at a fixed cruising speed most of the time, a bulbous bow (see Section 2.3.5) is often considered in order to limit the bow wave magnitude, further improving wavemaking efficiency. Above the waterline, most seagoing vessels use a flared bow (frontal stem rake and lateral flare). This supports the theory of better seakeeping as increased bow submersion leads to an exponentially increasing bow displacement. Therefore, greenwater is limited in rough seas and cruising efficiency is optimized for calm water situations that are most common during the operational lifespan of these vessels. The exponentially increasing displacement can be controlled by changing the flare and rake of the bow. Note

that longer L/B-ratio vessels may need an even steeper rake and flare as they tend to follow the wave's shape less due to their slender, piecing design. For impact mitigation and raising the vessel on top of the incoming wave, in order to keep the boat's momentum, a steep rake is most important. This is because the moment are of the raked bow is larger than the flare, propagated over the bow and midship. Large flare angles mostly serves a wave deflection function and keep the hull floating. Remark that flare causes peak accelerations and rake tries to mitigate that effect.

Ulstein shipbuilding suggested a second alternative bow shape called the X-bow that is characterized by smoother operation for large scale ships compared to the conventional bow geometry and may increase efficiency for certain situations as claimed by CFD research [94]. The main buoyancy is shifted aft, which is most influential cause of smoothing an incoming wave. While the wave travels along the hull, the cantilevered planar sides of the hull prevent changes in water deflection speed (compared to flared hulls) limiting wave encounter pitch accelerations. For high sea states, the hull is capable of nosediving (increased spray) insuring a smooth ride and limiting greenwater by placing the front deck high up. Datawave Marine Solutions [16] suggest using this type of bow only for large vessels, which is a valid statement considering that the wave stability is governed by large inertia as illustrated in the CFD-analysis [94]. A short, lightweight racing hull cannot benefit from the main advantages of the X-bow design. Therefore, there is no point in discussing the X-bow any further.

In the third place, the axe-bow, developed by Damen, may be of particular interest due to its unique and distinctive design. The fine — often vertical — nose drops near the bow and combined with the flat sides make it look like an axe, hence its name. It is generally focused on the limitation of pitch accelerations and linear flow deflection as the X-bow does, but for another type and scale of craft. Axe types are often used for smaller boats in contrast to the X-bows of Ulstein's utility vessels. Due to its fine long-stretched bow, the CG should be placed more aft. This results in a lightweight bow (large moment arm) trying the pitch the vessel up slightly before the main mass and thus displacement reaches the wave. Adjusting the approach angle slightly before reaching the wave, results in less over all pitching oscillations and peak accelerations as demonstrated in the Enlarged Ship Concept proposed by Delft University [88]. The shape allows use for planing craft too, but the technique is not

fully matured in the science of shipbuilding and still raises some questions regarding broaching, wind influence, lack of sub hull air introduction, maneuverability, etc. But why has the hull increased draft near the bow? That question can be answered in extend of the discussion on porpoising accelerations coming from hull reentry (see Section C.1.32). The issue of wave induced porpoising mainly imposes issues of structural impact coming from reentry. If an axe bow is raised — which is already limited because of the low-displacement piercing bow, the keel never leaves the water. Because reentry is absent if the bow never leaves the water, no reentry phenomena occur.

The forth and last type of bow that is discussed is the inverted bow, also known as submersible or wave piercing bow and is often seen in catamaran, trimaran and yacht designs. Note that within naval engineering, there is no consensus on naming conventions. In contrast of the conventional and X-type bow, the inverted bow is not focusing on keeping the deck above the water. Instead, it tries to pierce the waves and deflect water above the waterline. This technique focuses on keeping the vessel as level as possible and is often used in a SWATH-setup. Additionally, using such a bow/hull in slender configurations leads to strongly reduced wave pressure drag slowing the vessel down. That causes slender, sharp keel hulls to have maximum lateral grip in the water on straight lines making cornering more difficult. That's why sailing trimarans that use sponsons with wave piercing bows, often lift one side out of the water, apart from reducing drag and redistributing tilting sailing forces. In essence, the strongest argument for using sharp, wave piercing bows is to keep the craft's momentum.

For fast planing hulls, the bow is very important in keeping the craft stable, safe, and reliable. At high speeds, not only the water flow but also the airflow must be predictable, reducing the dan-

gers discussed in Section 2.4. Increasing oscillations in all DOF must be limited and the bow must keep a steady angle of attack, even when encountering waves. In general, a steep, almost

horizontal bow pushes an air cushion / lubrication under the hull, maximizing efficiency. This technique is also demonstrated in simple craft like wakeboards and water skis.

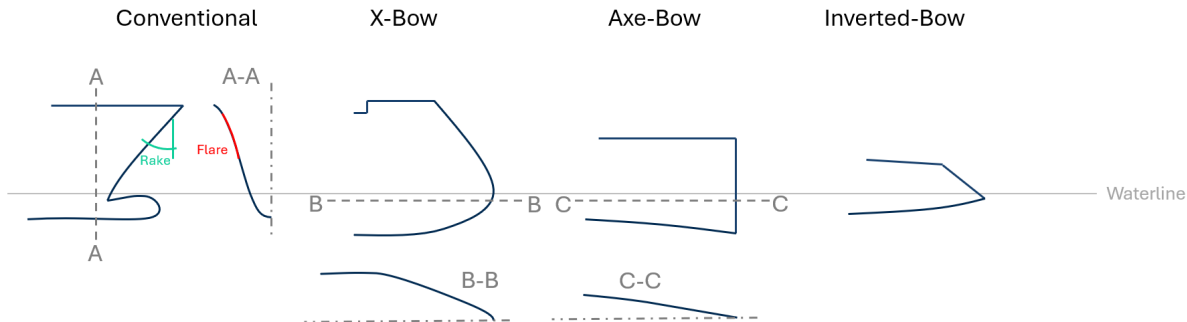


Figure C.11.: Common bow types and characteristic cross-sections

C.1.17. Transom Sterns in High-Speed Hulls

In ship and boat design, a transom is the (typically flat and near-vertical) aft termination of the hull at or near the design waterline. The adoption of transom sterns in high-speed craft is primarily driven by hydrodynamic considerations. For vessels operating above hump speed, the flow around the aft body can no longer follow a fine, curved contour due to flow separation at high Froude numbers. A transom stern provides a sharp edge for clean flow detachment, reducing the formation of vortices and low-pressure regions that otherwise increase residuary resistance [14, 95]. In racing hulls, this is critical, since wave-making and viscous pressure drag dominate the resistance components once the vessel transitions beyond the displacement regime. The vertical cut-off also improves the pressure recovery at the stern, lowering the stern

wave amplitude and thus reducing energy losses in the wake, similar to induced drag in airplanes. From a hydrodynamic lift perspective, the flat transom facilitates the transition into planing by enabling a stable stagnation line at the stern, which increases dynamic lift and allows the hull to rise onto its running surface [18]. Additionally, a transom stern provides a clean inflow to propulsors and control surfaces, minimizing unsteady forces and ventilation phenomena at high speed. Although more material and structural reinforcement may be required compared to fine-ended hulls, the performance gains in reduced resistance, enhanced stability, and reliable propulsion integration make the transom stern the technically superior choice for racing hulls operating above hump speed.

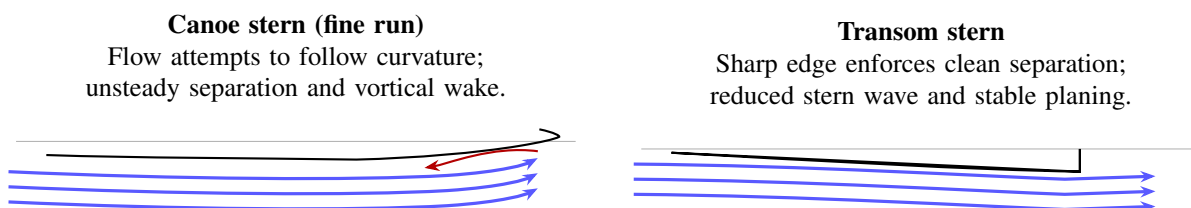


Figure C.12.: Flow topology at the stern above hump speed.

C.1.18. Theoretical Drag

Based on the lectures of the Technical University of Lisbon within the course “Resistance and Propulsion,” the following overview was developed. Table C.7 distinguishes between physical effects and theoretical assumptions to illustrate how different resistance components contribute to total drag.

Resistance type	Fully submerged? (No free surface, infinite depth)	Ideal fluid? (Zero viscosity)	Non-zero?	Reason
Frictional	Yes	Yes	No	No shear stresses
Pressure	Yes	Yes	No	d’Alembert paradox
Viscous	Yes	Yes	No	No viscosity
Wave	Yes	Yes	No	No gravity wave system
Frictional	Yes	No	Yes	Viscous friction
Pressure	Yes	No	Yes	Viscous pressure
Viscous	Yes	No	Yes	Viscous fric. + press.
Wave	Yes	No	No	No gravity wave system
Frictional	No	Yes	No	No shear stresses
Pressure	No	Yes	Yes	Wave pressure
Viscous	No	Yes	No	No viscosity
Wave	No	Yes	Yes	Wave generation
Frictional	No	No	Yes	Viscous friction
Pressure	No	No	Yes	Viscous & wave pressure
Viscous	No	No	Yes	Viscous fric. + press.
Wave	No	No	Yes	Wave pressure

Table C.7.: Alternative theoretical drag decomposition

Note that in the sections following the drag decomposition, the viscous pressure resistance and frictional resistance are referred to as one. The decomposition approach will be different due to the significance in hull resistance and relation to Froude and Reynolds-depending components. More details and a schematic representation of drag decomposition and related conventions in naval architecture can be found in the reference book by Molland et al. [19].

C.1.19. Hydrodynamic Drag

Hydrodynamic resistance is traditionally decomposed into several components:

$$D_{\text{hydro}} = D_f + D_{\text{form}} + D_w + D_i + D_{\text{spray}} + D_{\text{app}}, \quad (\text{C.6})$$

where D_f is viscous surface friction, D_{form} is pressure or form drag due to geometry, D_w is wave-making resistance, D_i is induced drag associated with lifting surfaces, D_{spray} is momentum lost to spray formation, and D_{app} is drag due to appendages (rudders, struts, shafts).

Spray is formed at the stagnation line (see Section 2.3.6) on high-speed marine craft in planing regime; a visible line that separates water flow under the hull from spray formation flow that

forms a thin sheet of water on the underside of the hull’s surface (deflected flow) on the other side of the stagnation line. This is a significant effect accounting for around 15-18% of total re-

sistance at true planing speeds [76]. The importance of the spray resistance further increases with raised speed and can be mitigated by spray rails (see Section C.1.11 and multi-step in Figure C.22). These rails have the potential to partially convert the reduced spray losses into a minor lift advantage. Therefore, spray rails are often called lifting strakes, depending on their geometry and the convention in the field of application.

The viscous component is estimated by the ITTC'57 correlation line:

$$C_f = \frac{0.075}{(\log_{10} Re - 2)^2}, \quad D_f = \frac{1}{2} \rho_w V^2 S C_f \quad (C.7)$$

with wetted surface area S , Reynolds number Re , and water density ρ_w . This can be decomposed into two separate resistance effects: (1) the friction resistance due to shear stresses tangential

to the body and (2) viscous pressure resistance acting normal to the hull. In extend, the stream around the hull is characterized by the type of flow: laminar versus turbulent flow. Physically, the white zone around the ship indicates the turbulent boundary layer. Similarly, the magnitude of the wake waves and white water stream provide an indication of the resistance the hull is exposed to. For example, a large wake wave indicates large displacement and/or abrupt changes in flow direction (see Hull Vane in Section 2.3.2). The flow separation indicated in Figure C.13 is responsible for the separation or so called Eddy resistance; the main contribution to total form drag. As highlighted in Section 2.1.7, it is important to limit abrupt changes in flow deflection, especially for slow-speed vessels, in order to minimize Eddy turbulence.

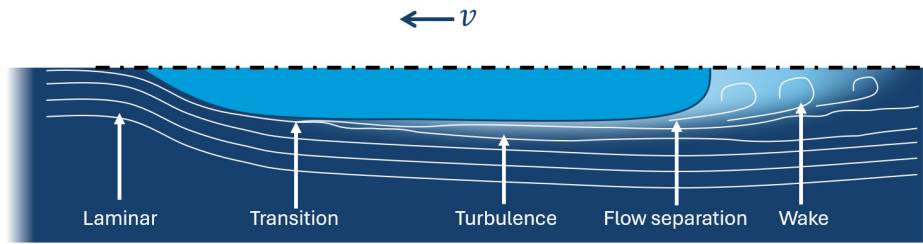


Figure C.13.: Flow around a ship

Wave-making drag dominates near Froude numbers $0.4 < Fn < 0.5$, often scaling as $D_w \propto V^n$ with n between 3.5 and 5. For foil-borne or lifting hulls, induced drag is critical and given by

$$D_i = \frac{L^2}{\pi e AR \frac{1}{2} \rho_w V^2} \quad (C.8)$$

where L is lift, AR the aspect ratio of the lifting surface, and e the span efficiency factor. Spray and appendage drag are smaller but non-negligible, especially in complex high-speed hull configurations.

C.1.20. Aerodynamic Drag

Above the free surface, the exposed superstructure and hull surfaces experience aerodynamic resistance. This can be decomposed analogously:

$$D_{aero} = D_{f,a} + D_{form,a} + D_{int,a} + D_{i,a} \quad (C.9)$$

where $D_{f,a}$ is skin friction drag from the air boundary layer, $D_{form,a}$ is pressure or bluff-body drag (dominant form drag) from topside geometry, $D_{int,a}$ is the interference drag propagating from the edges connecting bodies like cockpit and tunnel, and $D_{i,a}$ is induced drag associated with any lifting surfaces interacting with air (e.g., tunnels) in which wake turbulence is formed due to downward flow deflection. The first three types of drag defined are also known as parasitic drag in aerodynamics. Note the goal of a performance hull is also to eliminate bluff bodies on the top side geometry. However, sensors (cameras, LiDAR, radar, etc.) and antennas (GNSS, GCS) are often neglected to preserve optical / radiation performance. The total aerodynamic drag is expressed as

$$D_{aero} = \frac{1}{2} \rho_{air} V^2 A C_D \quad (C.10)$$

where ρ_{air} is air density, V is the vessel speed, A is the frontal projected area and C_D the aerody-

dynamic drag coefficient. Aerodynamic drag is a minor fraction of total resistance at 20–40 knots (see section 2.3), but increases in importance at speeds above 45–50 knots, especially once hydrodynamic drag is governed by significant dynamic lift or foiling. The decomposition of aerodynamic drag over a speed range is clarified by Figure C.14.

The ITTC air resistance coefficient used for scal-

ing is given by:

$$C_T = \frac{R_T}{\frac{1}{2}\rho_w S_0 V^2} \quad (\text{C.11})$$

- C_T = coefficient of total hull resistance
- R_T = (or D_{aero}) total hull resistance [N]
- ρ_w = mass density of water [kg/m^3]
- S_0 = nominal wetted surface area [m^2]
- V = speed [m/s]

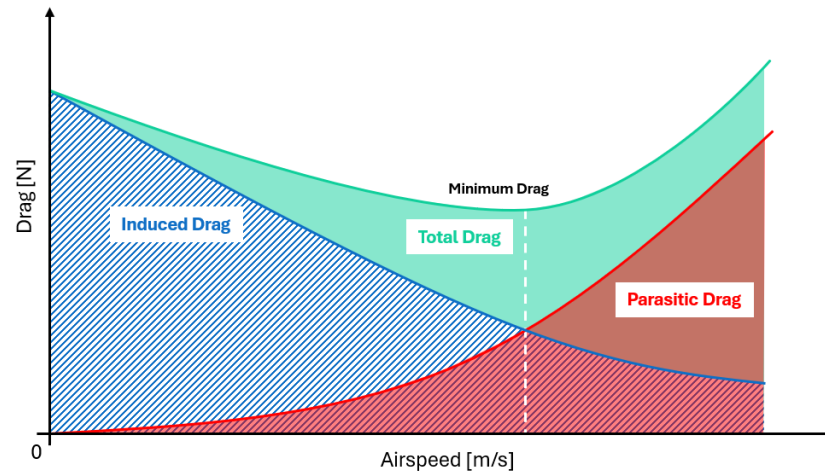


Figure C.14.: Aerodynamic drag decomposition

C.1.21. Overview of Design Implications for High Speed Autonomous Craft

Perception Sensors

LiDAR

- **Flat sea** - Specular reflections from calm water produce sparse or aliased returns, particularly at shallow grazing angles [96, 97]. Range bias and dropout increase when the beam footprint skims the surface [97]. Benefits include low spray and minimal occlusions, enabling long-range obstacle detection if the LiDAR has sufficient vertical range — or is compensated by a gimbal setup — and mounted high enough to maintain favorable incidence angles [96].
- **Medium sea** - Breaking crests, spray, and aerosols increase backscatter and false positives [97]. The rapidly changing free surface yields dynamic horizon clutter; wave faces can be misclassified as obstacles. Mechanical scanners are sensitive to high-frequency vibration; solid-state units reduce moving-parts sensitivity but still require isolation. Protective windows must be hydrophobic/oleophobic and heated to shed droplets.

Design implications — Vibration-isolated, high-mount installations with controlled pitch angle are required. Near-field masking of the water plane in software, fusion with roll/pitch-stabilized motion, and filters help suppress spray-induced noise. Hull forms that minimize spray generation — such as tunnel hulls or fine, wave-piercing bows — enhance LiDAR data integrity, whereas poorly designed tunnels can eject spray in front of the craft that increase deck wetting and potential optical interference [96].

3D Cameras and Imaging

- **Flat sea** - Stereo disparity fails on mirror-like surfaces; ToF suffers from low return energy and multi-path from glints. Polarizing filters help, but reduce light reflections. Dynamic exposure and HDR improve horizon detail. [98]

- **Medium sea** - Droplet occlusion, spray curtains, and rapid platform motion degrade feature tracking and optical flow. Rolling-shutter sensors amplify motion distortion. Thermal cameras maintain horizon/obstacle contrast in glare but have reduced small-object resolution. [98]

Design implications — Global-shutter sensors, short exposure with high-sensitivity optics, and active stabilization (gimbals or electronic) are recommended. Window heaters or air-knife purge systems mitigate droplet interference. Rotating lenses or wipers may also increase reliability but add cost and complexity. Algorithmically, horizon segmentation and temporal consistency checks improve robustness. [98]

Antennas: GNSS and Communications

- **Flat sea** - Sky visibility is high; GNSS multipath is moderate but present from deck reflections [99]. High-gain antennas on masts provide robust RTK/PPP corrections. Low frequency RF path is rarely blocked by spray [100].
- **Medium sea** - Pitch/roll excursions cause intermittent satellite masking and Doppler dynamics; spray and water films can detune antennas [99]. Tunnel and canopy flows can introduce vibration at mast tips, degrading signal stability [101]. For comms, sea clutter and aerosols attenuate higher microwave bands [100]. Significant craft attitude may induce temporal signal loss for certain antenna configurations [101].

Design implications — Stiff, low-deflection masts with damping are recommended, together with dual-antenna GNSS baselines and multi-band receivers. Radomes with hydrophobic coatings and drainage improve reliability. For comms, frequency diversity (recommend frequency hopping to mitigate interference) and EMI shielding are beneficial. For all radio communication, a position as far away from the wa-

ter body as possible is mandatory. The higher the frequency, the more water absorbs the radio waves. Therefore, maritime radios, geophysical measurements and navigation systems usually rely on long wavelength VLF (100-10 km; 3–30 kHz). Communication system redundancy is mandatory as demonstrated by the modern USV Gobelijn (VLIZ) featuring 4G, Starlink MBR 144 and 179 (LOS), Iridium VesselLINK satellite and a Iridium SBD backup system.

Hull-Form Consequences for Sensing

- **Catamaran and deeply-submerged-sponson trimaran** - Provides high transverse stability and predictable roll, improving sensor pointing and GNSS lock. The tunnel reduces wave impact and shelters payloads.
- **Piercing bows** - Reduce heave/pitch accelerations by penetrating waves rather than riding over them, improving LiDAR and camera temporal coherence.
- **Tunnel-hull air cushion** - Smooths pressure fluctuations, attenuates spray, and reduces slam frequency while providing lift with lower spray than planing surfaces.
- **Flat-bottom planing hull** - Generates high-frequency vibrations and severe spray, unsuitable for high-speed autonomous sensing in moderate seas.

High-Speed Cornering Requirements

- **Load paths and stability** - Cornering induces large lateral accelerations and dynamic heel.

- **Hydrodynamic control** - Predictable yaw authority requires appendages resistant to ventilation. Strakes and chine geometry should minimize spray into sensor fields.
- **Aero-hydro balance** - Aerodynamic lift reduces drag on straights, but cornering benefits from modest aero-downforce elements to stabilize roll and heave.
- **Actuation and damping** - Trim control should prioritize stability over drag. Sensor payloads require tuned isolators and gimbals to reduce jitter.

Systems Engineering Measures

- **Vibration environment** - Characterize and isolate engine and wave impact frequencies. Co-locate IMU/INS with GNSS timing sources to reduce errors.
- **Sensor placement** - Mount LiDAR and cameras high and forward, avoid spray trajectories, and place antennas at the highest points for unobstructed sky view. Reduced aerodynamic efficiency in exchange for sensor and antenna reliability is acceptable.
- **Environmental hardening** - Cooling, hydrophobic windows, ingress protection, and EMI shielding are required.
- **Perception robustness** - Fuse LiDAR, camera, and INS/GNSS; apply horizon and sea-surface modeling with redundancy.
- **Operational envelopes** - Define validated speed—sea-state envelopes, limiting speed in medium seas to maintain perception reliability.

C.1.22. Existing Hull Adaptation Suggestions

Ref.	Hull	Section	Adaptation	Pro	Con
1	Torpedo Mk.II EVO	Deck/shoulder	Add deck camber or positive sheer	Reduces slamming pressures, sheds water laterally	Increases structural complexity, higher CG, more windage
2	Torpedo Mk.II EVO	Deck/shoulder	Spray knuckle/chine above waterline	Deflects greenwater, lowers slam frequency	Possible added wave-making resistance
3	Torpedo Mk.II EVO	Deck/shoulder	Shorten or perforate overhang	Breaks suction, improves drainage	Less dry deck space, structural penalties
4	Torpedo Mk.II EVO	Bow	Finer entry or wave-piercing bow	Reduces vertical accelerations, lowers deck wetness	Increases wetness at low speeds, sensitive to CG
5	Torpedo Mk.II EVO	Bow	Lifting strakes on forebody	Deflects spray, reduces wet deck loads	Additional drag at some trims
6	Torpedo Mk.II EVO	Control	Bow interceptor/small T-foil	Controls trim, reduces slam in head seas	Added wetted area, complexity, debris/ weed sensitivity
7	Shark	Bottom	Warped-plane or moderate-V aft	Reduces stern slide, smoother re-entry	Higher drag at low-mid speed, more complex build
8	Shark	Bottom	Add lifting strakes	Improves dynamic lift and tracking, channels spray	Risk of porpoising, harsh impacts if oversized
9	Shark	Bottom	Hard chines/spray rails	Stronger edge for cornering, keeps spray low/outward	Harsher impacts, increased spray if poorly aligned
10	Shark	Bottom	Delta pad or keel pad	Reduces wetted area on straights, improves tracking	Bow steer or skid risk if pad too wide/flat
11	Shark	Bottom	Sword	Stabilizes vessel, improved cornering	High increase in wetted surface area
12	Shark	Appendages	Keel skeg/turning strakes	Increases lateral grip, improves yaw authority	Extra drag, snag/trip risk in waves
13	Shark	Appendages	Trim tabs or stern interceptors	Controls bow dive, stabilizes turns	Extra drag when deployed, moving parts, complexity
14	Shark	Appendages	Sponsons/pods	Wider track, more roll stiffness, softer re-entry	Added weight and drag, fabrication complexity
15	Shark	Mass/ CG	Shift CG aft	Reduces bow dive and steer, improves trim	Risk of porpoising, stern squat, ventilation
16	Shark	Aero aids	Add modest aft downforce	Increases grip in cornering	Drag penalty, spray interaction, antenna shadowing
17	Shark	Bow	Nose fin/ski	Lifts the bow, improving handling and encourages planing	Higher drag, debris/ weed sensitivity, most effective on flat water, may induce porpoising

Table C.9.: Overview of hull adaptations with pros and cons

C.1.23. Geometry Section

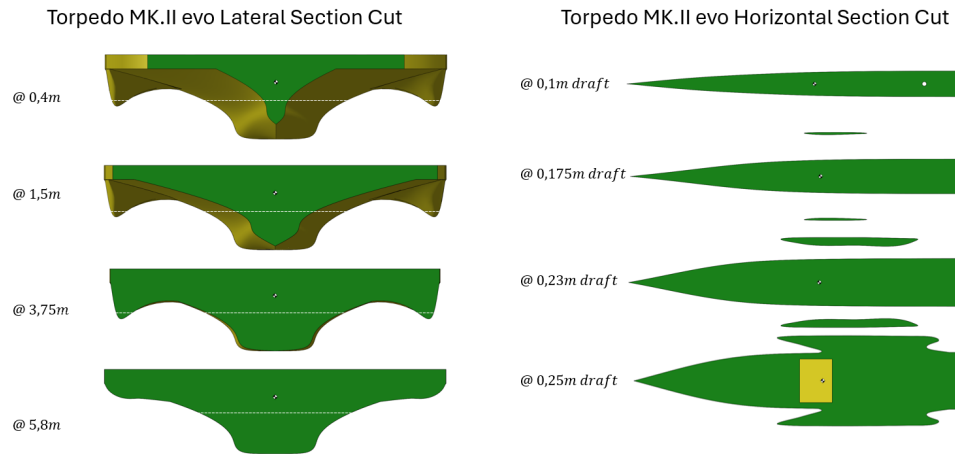


Figure C.15.: Torpedo lateral and horizontal section cut

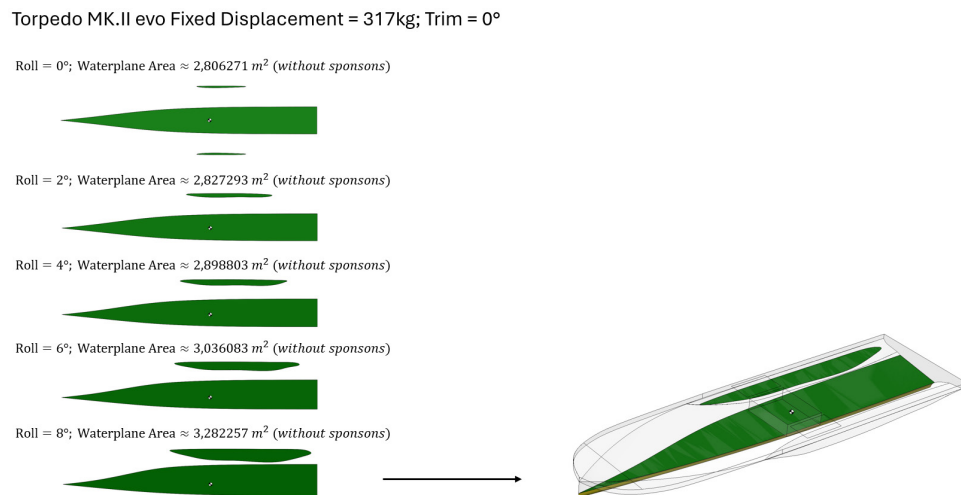


Figure C.16.: Torpedo Roll

C.1.24. Defining the Coefficients of Form

- **Block coefficient** — The block coefficient expresses the ratio between the vessel's displaced volume and the circumscribing rectangular block defined by length, breadth, and draft. It indicates the overall “fullness” of a hull and is particularly relevant for displacement vessels where low

resistance at moderate speeds is desired.

$$C_B = \frac{\nabla}{L_{WL} B_{WL} T} \quad (C.12)$$

∇ = displaced volume
 L_{WL} = length on the waterline
 B_{WL} = beam on the waterline
 T = mean draft

- **Midship coefficient** — The midship coef-

ficient compares the actual midship cross-sectional area to the rectangular section of breadth and draft; it highlights the fullness of the hull at its widest section.

$$C_M = \frac{A_M}{B_{WL} T} \quad (C.13)$$

A_M = molded midship immersed area (area of the max-immersion transverse section)

- **Area of waterplane coefficient** — The waterplane area coefficient relates the actual waterplane area to the product of length and breadth at the waterline, linking directly to initial stability and resistance in waves.

$$C_{WP} = \frac{A_{WP}}{L_{WL} B_{WL}} \quad (C.14)$$

A_{WP} = waterplane area at the design waterline

- **Prismatic coefficient** — The prismatic coefficients are divided into horizontal and vertical forms. The horizontal prismatic coefficient (commonly just called the prismatic coefficient) relates the displacement volume to the product of length and midship area, and it strongly influences wave-making resistance and the position of the “hump” in resistance curves. The vertical prismatic coefficient, by contrast, considers the distribution of underwater volume in the vertical plane and can be useful in assessing draft distribution and flow characteristics.

$$C_P = \frac{\nabla}{A_M L_{WL}}, \quad C_{VP} = \frac{\nabla}{A_{WP} T} \quad (C.15)$$

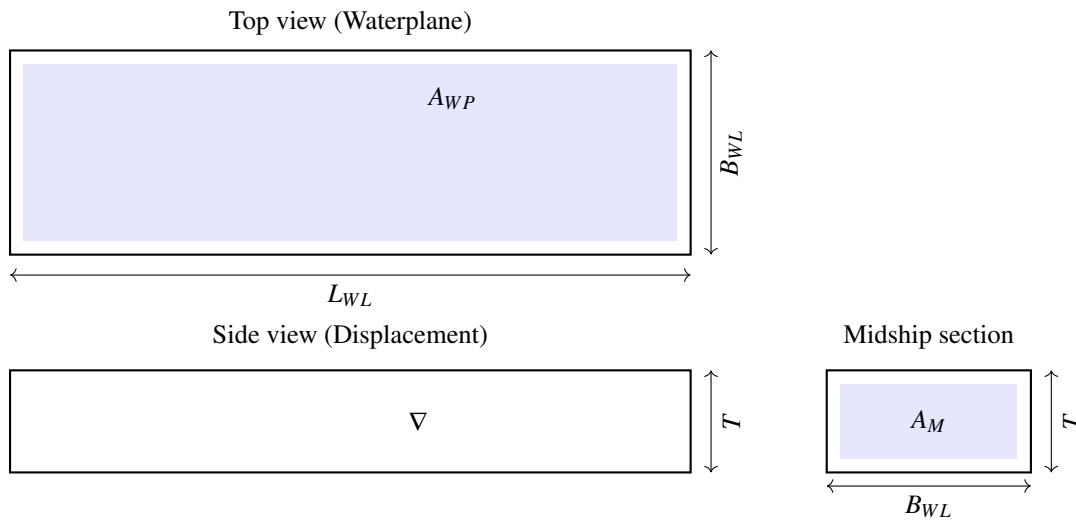


Figure C.17.: Definition of principal hull parameters for form coefficients.

C.1.25. Coefficients of Form Report

Block and Midship Coefficients

The higher block coefficient of the Shark ($C_B = 0.527$ vs. 0.476) and the fuller midship ($C_M = 0.760$ vs. 0.707) indicate a broader, more voluminous immersed body at the design waterline. At low Froude numbers this would increase residuary resistance; however, above hump speed the block and midship coefficients

are less decisive because dynamic lift dominates the support of mass, especially for a flat-bottom planing surface. For the semi-displacement Torpedo, the lower C_B and C_M are appropriate to limit wave-making near $Fr \approx 0.4$ – 0.6 and to delay the resistance “hump.”

Prismatic Coefficient (Horizontal)

Both hulls have C_P in the range 0.673–0.694, which is typical for efficient high-speed hulls and consistent with planing-surface analyses where moderately high prismatic fullness promotes favorable pressure recovery and a stable running trim. The Shark's slightly higher C_P aligns with its intended planing mode, aiding the development of a sustained stagnation line at the transom and smoother transition beyond hump speed. The Torpedo's slightly lower C_P reflects its semi-displacement role, balancing wave-making and viscous components while still enabling partial dynamic lift.

Waterplane and Vertical Prismatic Coefficients

The Shark's larger waterplane coefficient ($C_{WP} = 0.777$ vs. 0.735) implies greater initial transverse and longitudinal hydrostatic stiffness (not including the Torpedo's sponsons), which contributes to heading and roll stability at speed and improves control authority during rapid maneuvers. The trade-off is a stronger wave-structure coupling, potentially leading to higher slamming impulses for a flat bottom at high speed. The higher vertical prismatic ($C_{VP} = 0.679$ vs. 0.647) indicates a more uniform vertical distribution of volume, consistent with shallower mean draft and a broader near-surface footprint; this supports rapid lift buildup and reduced sinkage at the stern as the planing regime is established. The Torpedo's lower C_{WP} and C_{VP} fit a semi-displacement strategy: smaller hydrostatic restoring coefficients moderate wave-induced accelerations and help limit added resistance in a seaway while the hull still benefits from some dynamic support.

Interpretation and Limitations

In calm water comparisons at the design waterline, the Shark's higher C_P and C_{WP} are advantageous for planing performance and control above hump speed, whereas the Torpedo's lower C_B and C_M suit semi-displacement operation where wave-making remains relevant. Note that all

coefficients are defined at the static design waterline; in the high-speed regime the effective immersed geometry changes with dynamic trim and sinkage, so these ratios are best viewed as baseline descriptors rather than absolute performance predictors.

Speed-Regime Relevance by Froude Number

While all form coefficients are defined at the static waterline, their predictive power shifts with Fr as the immersed geometry, trim, and pressure field evolve:

- **Block coefficient C_B and midship coefficient C_M** — Most influential in the displacement regime and through the hump onset, i.e. $Fr \lesssim 0.40$ (displacement) and $0.40 \lesssim Fr \lesssim 0.55$ (classical hump band). Fuller forms (higher C_B , C_M) raise residuary resistance and can accentuate the hump; finer forms delay and soften it. Above $Fr \gtrsim 0.6$, their direct influence rapidly diminishes as dynamic lift grows and wetted geometry changes dominate.
- **Prismatic coefficient C_P (horizontal)** — Becomes decisive from late displacement through semi-planing and planing, roughly $0.45 \lesssim Fr \lesssim 1.2$. In 0.5–0.8, C_P strongly affects pressure recovery, transom ventilation, and the stability of the running trim; in 0.8–1.2 it correlates with sustainable planing and lower pressure drag. Very low C_P tends to prolong hump behavior; moderately high C_P supports a clean transition and stable trim at speed.
- **Waterplane area coefficient C_{WP}** — Governs hydrostatic stiffness and wave-structure coupling across the full range, but its performance impact is most evident in $0.3 \lesssim Fr \lesssim 0.9$. In 0.3–0.6, larger C_{WP} improves maneuvering and roll stability yet can increase added resistance in a seaway; in 0.6–0.9 it aids control authority as lift builds, at the cost of stronger slamming risk for flat bottoms.
- **Vertical prismatic coefficient C_{VP}** —

Most informative for vertical volume distribution and sinkage/trim response in $0.5 \lesssim Fr \lesssim 1.0$. Higher C_{VP} generally corresponds to more uniform vertical fill-

ing, aiding stern support and earlier/steadier planing (reduced stern squat) once the transom runs dry. Below $Fr \lesssim 0.4$, its effect trails that of C_B and C_M .

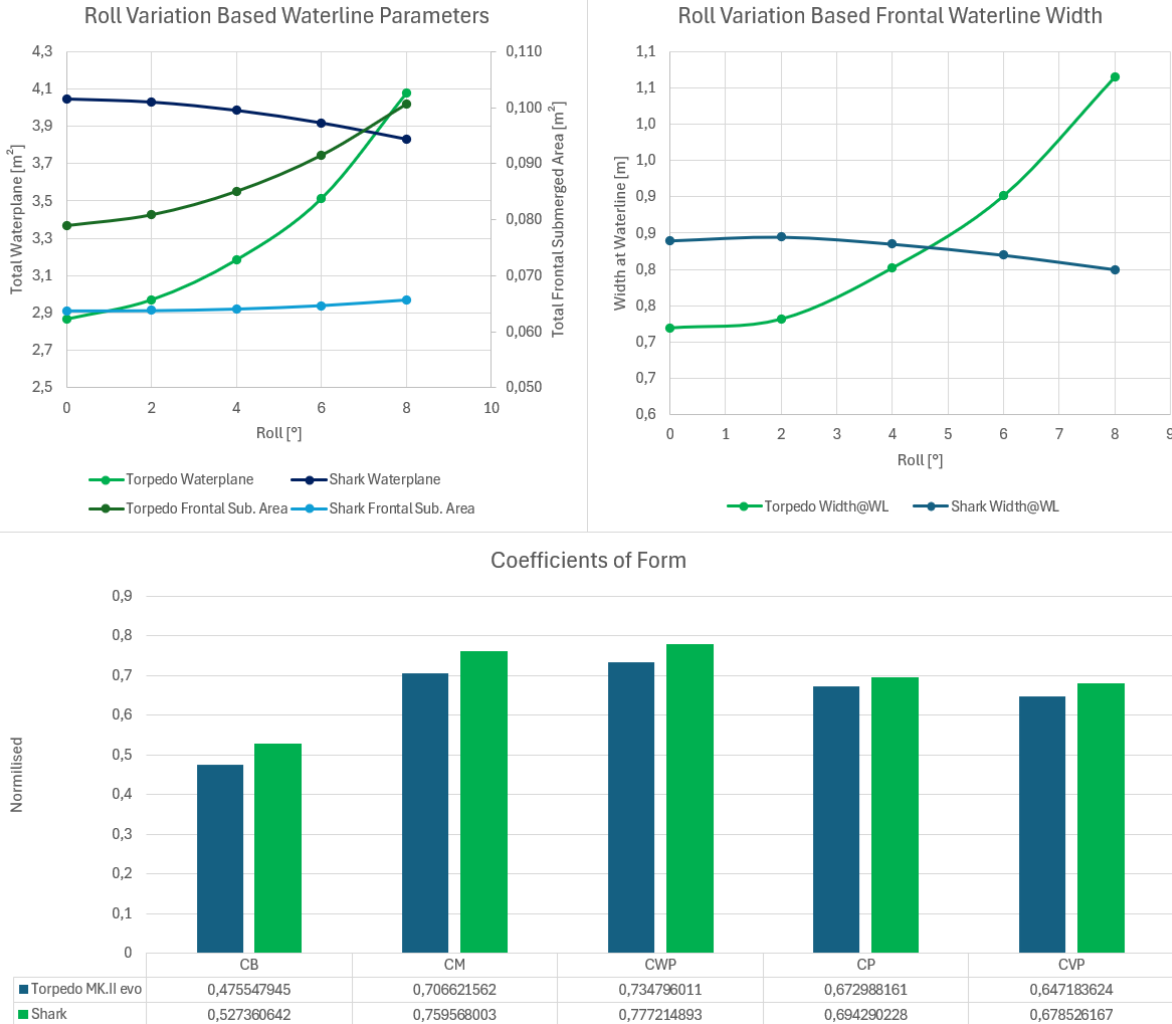


Figure C.18.: Coefficients of form and geometry comparison

C.1.26. Hydrodynamic and Aerodynamic Drag Balance

The resistance of small high-speed vessels in the 5–10 m class is dominated by hydrodynamic forces, with aerodynamic contributions becoming relevant only when the hull is largely lifted from the water. At typical operating speeds of 20–40 knots, hydrodynamic drag represents approximately 85–95% of the total resistance, while aerodynamic drag seldom exceeds 5–15% [19]. This distribution highlights the central design objective: to minimize hull–water interaction and achieve dynamic or foil-borne lift at the lowest possible speed, thereby improving attainable maximum velocity, reducing shaft power requirements, and enhancing overall fuel economy.

The aerodynamic drag of the above-water structure follows the classical quadratic dependence on speed and can be expressed as Equation C.10. Consequently, the corresponding aerodynamic power demand is

$$P_{aero} = D_{aero} V \propto V^3, \quad (C.16)$$

which illustrates that while initially small, the aerodynamic component increases rapidly with speed, especially once hydrodynamic drag has been reduced by lift devices.

Hydrodynamic drag, on the other hand, comprises viscous friction, wave-making, spray, appendage components, etc. The viscous resistance R_f or viscous drag D_f is typically defined or approximated as illustrated in Equation C.7. In addition, typical displacement hull components dominate including wave-making, form and induced drag. They grow more steeply with speed and may be approximated by a power-law dependence of the form

$$D_w \propto V^n, \quad 3.5 \leq n \leq 5, \quad (C.17)$$

depending on the hull form and operating Froude number. The total D_{hydro} is defined in Equation C.6 and leads to the total hydrodynamic power

$$P_{hydro} = D_{hydro} V, \quad (C.18)$$

in which hydrodynamic drag overwhelmingly dominates the energy budget of conventional displacement and semi-planing hulls up to at least 40–45 knots.

The comparison of these mechanisms makes clear that aerodynamic optimization is of secondary importance in the design of 5–10 m low profile craft at the intended cruise speeds. At moderate speeds, aerodynamic drag is one order of magnitude smaller than hydrodynamic resistance, and only when the hull is almost completely clear of the water—for example, when supported by hydrofoils— and operating at very high speeds does it approach comparable magnitude. As such, the design priority for medium to high-speed small craft must be to reduce hydrodynamic drag through hull shaping, wetted area reduction, and dynamic lift strategies, while ensuring sufficient stability and safety during the lift transition. Aerodynamic refinement becomes valuable mainly at very high speeds, typically above 45–50 knots, when it begins to represent a substantial fraction of the total resistance.

In conclusion, the geometry optimizations in terms of flow efficiency of the existing hulls will solely be related to hydrodynamic drag reduction. For the new hull, as future proofing (higher speed regimes) and maximum customization allows for the implementation of a more optimal planing configuration, aerodynamics will be taken into account, limited to a certain level.

C.1.27. Power Reduction & Speed Increase for Common Hull Upgrades

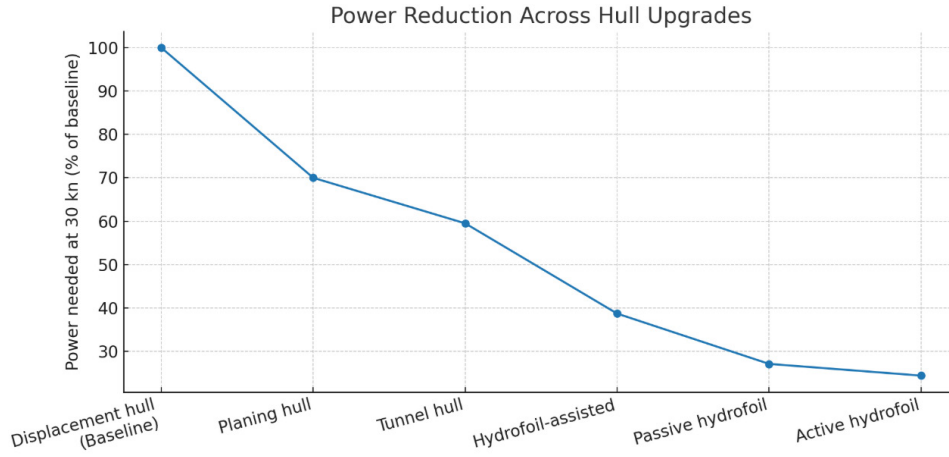


Figure C.19.: Power reduction for common hydrodynamic hull upgrades

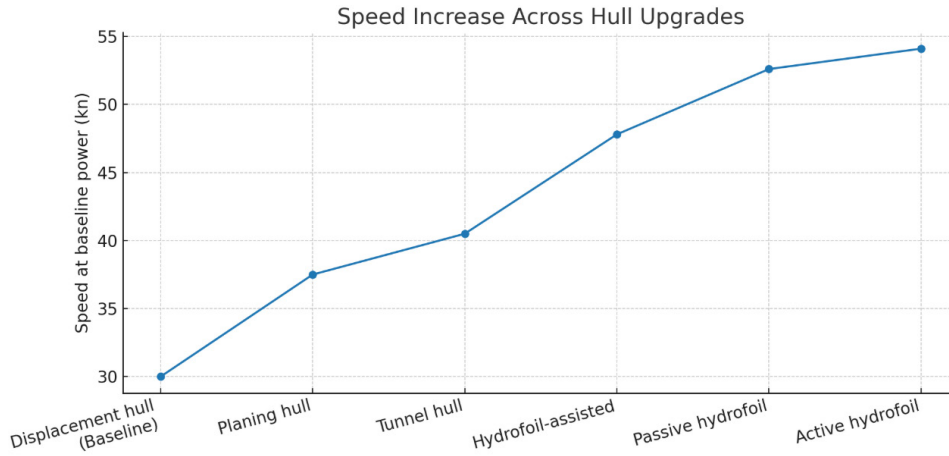


Figure C.20.: Speed increase for common hydrodynamic hull upgrades

C.1.28. Microstructure Drag Reduction — In-depth

$$\ell^* = \frac{\nu}{u_\tau} \quad \Rightarrow \quad s^+ = \frac{s}{\ell^*} = \frac{s u_\tau}{\nu} \quad (\text{C.19})$$

- ℓ^* = viscous length scale (wall unit) [m]
- ν = kinematic viscosity of the fluid [m²/s]
- u_τ = friction velocity (shear velocity) [m/s]
- s^+ = non-dimensional riblet spacing [-]
- s = physical riblet spacing [m]

The predominant mechanism for drag reduction is that shallow, streamwise micro-grooves guide the flow of water (or air) so that small turbulent vortices near the surface cannot move sideways as easily, illustrated by Figure 2.9. This keeps the flow more orderly and reduces the friction between the fluid and the surface. Importantly, physical replicas of shark denticles (with complex three-dimensional crowns) do not automatically yield drag reduction. Several high-fidelity numerical and experimental

studies show that some denticle arrangements can increase drag drastically, whereas engineered riblets of appropriate size and alignment provide modest, robust reductions. [35, 36, 102]

Optimal riblet spacing s and height h scale with the near-wall viscous length ν/u_τ (with u_τ the friction velocity). In practice, this means the physical optimal spacing tends to be on the order of a few $\times 10 \mu\text{m}$ at the Reynolds numbers relevant to aircraft skins and fast marine boundary layers. As speed increases (higher u_τ), the optimal physical spacing decreases slightly. Mis-sizing (too large or too small s, h) or misalignment to local flow can negate benefits and even add drag as previously presented. Therefore, design and practical implementation must be conducted with much care for geometry and orientation. [28, 103, 104]

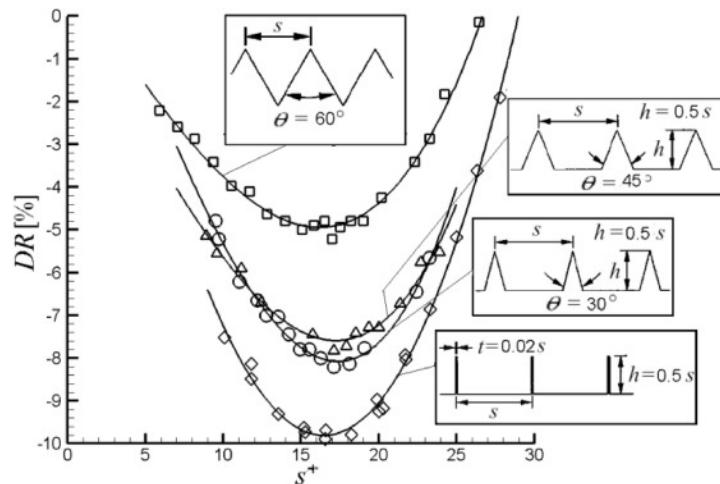


Figure C.21.: Drag reduction of various riblet geometries (source: [105, 106])

Planing Craft Implementation

At high Froude numbers, a planing mono-hull's wetted footprint contracts and is concentrated near the stern. Applying riblets only on that small, highly loaded patch targets the right physics (turbulent skin friction), but the global benefit is bounded by (1) the fractional area treated and (2) local flow quality (spray impingement, intermittent separation, flow yaw). In clean, attached flow on the running bottom, a well-sized riblet film might deliver a few percent reduction in the local friction component; at the total-boat level this often translates to small — sometimes sub-percent — reductions, since total resistance at high speed also contains pressure/spray and residual components. Thus, riblets are a plausible incremental efficiency enhancer for steady-state planing, provided alignment is maintained and the surface structure remains clean.

Commercial riblet films (e.g., aerospace-grade laminates) are available off-the-shelf and have been applied on race craft and transport vehicles. NASA/3M riblets were famously used on “Stars & Stripes” in the America’s Cup, similar textures have appeared in motorsport and on production airliners [107, 108, 109].

Surface	Characteristic size	Notes	Source
Shark denticle crown	$\sim 0.2\text{--}0.5$ mm	Ridge spacing	[110, 111]
White shark denticles	planform Ridge spacing $30\text{--}150$ μm ; height $3\text{--}26$ μm	$\sim 30\text{--}100$ μm 3D micro-CT quantification	[112]
Engineered riblets (lab)	Spacing/height typically $25\text{--}75$ μm	Optimal $s^+ \approx 15$; mis-sizing increases drag	[28, 32]
3M / wind-turbine airfoil tests	Optimal spacing $\sim 44\text{--}62$ μm	4–5% airfoil drag reduction (local)	[103]
AeroSHARK (aircraft)	Riblet height ~ 50 μm	$\sim 1\%$ fuel-burn reduction in service	[30, 31]

Table C.10.: Typical dimensional scales for sharkskin denticles and engineered riblet films.

C.1.29. Bulbous Bow Design

The effectiveness of a bulbous bow therefore depends strongly on both hydrodynamic precision and operational conditions. As emphasized by Nicholas Barczak of DMS [41], bulbs are often overrated in general marine design discourse. When not meticulously optimized for a vessel's exact displacement, speed, and wave environment, they can lead to higher resistance, flow separation, and added structural complexity without benefit. Bulbs are most useful for ships operating in steady-state conditions with consistent wave systems, such as large displacement vessels traveling in calm or near-flat water. Under variable sea states or off-design speeds, their performance advantage quickly diminishes.

The classical design procedure for bulbous bows was established by Kracht [113], who developed analytical relations between bulb geometry, wave interference patterns, and hull parameters. These methods provide an essential starting point for determining bulb volume, length, and immersion depth. In modern design practice, Computational Fluid Dynamics (CFD) is subsequently applied to refine and validate these preliminary calculations. Importantly, a base resistance test without a bulb is mandatory before introducing one, to quantify the net benefit and verify whether the added frontal area and flow deflection is justified.

The applicability of bulbous bows depends strongly on hull form characteristics, particu-

larly the block coefficient or C_B (see prior Section 2.2.3). Ships with high C_B values—such as tankers, cargo ships, and ferries—tend to benefit most from well-optimized bulbs because their wave-making resistance constitutes a major portion of total drag. In contrast, slender high-speed craft and semi-planing vessels with low block coefficients experience minimal wave interference and derive little or no advantage. In many cases, the added frontal area of the bulb can even increase total resistance at planing or semi-planing speeds. For this reason, bulbous bows are never found on modern speedboats, racing craft, or high-speed catamarans, where flow separation, dynamic lift, and spray formation dominate resistance characteristics.

In summary, while bulbous bows remain effective tools for reducing wave-making resistance in displacement vessels operating at steady speeds and calm-water conditions, they offer limited or even negative performance benefits for high-speed craft. Their usefulness is constrained by precise hydrodynamic geometry, narrow speed tolerance, and the vessel's block coefficient. In advanced design studies, the

inclusion of a bulb should always be justified by comparative CFD simulations supported by analytical estimates following Kracht's methodology for hand calculations.

C.1.30. Step & Multistep Illustrated

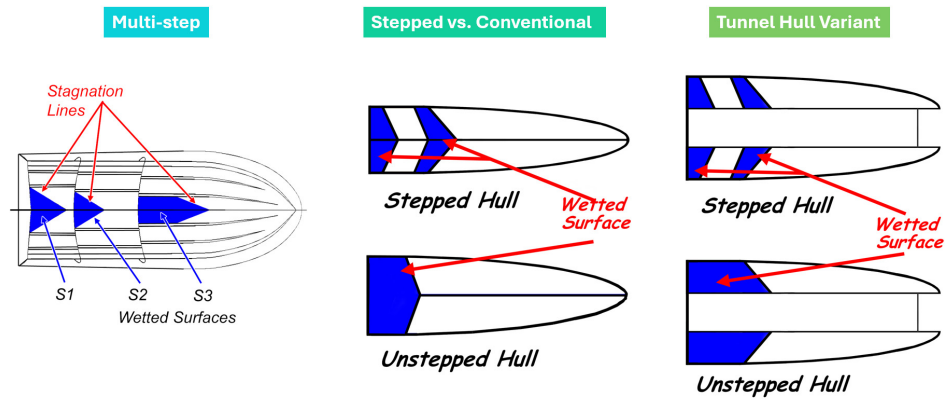


Figure C.22.: Stepped hull and stepped tunnel hull (source: AeroMarine Research)

C.1.31. Volumetric Froude Number in Stepped Designs

In contrast to the conventional length-based Froude number ($Fr_L = V/\sqrt{gL}$) used in the preceding sections, Yang et al. [26] introduced the volumetric Froude number (Fr_V) to characterize planing performance (see Figure C.23). The parameter is defined as

$$Fr_V = \frac{V}{\sqrt{g \nabla^{1/3}}}, \quad (C.20)$$

where ∇ denotes the vessel's displacement volume. This formulation normalizes the velocity by the cube root of the displaced volume rather than by a characteristic length, providing a more consistent representation of the hydrodynamic regime for planing and semi-planing hulls, in which the wetted length varies significantly with speed. More variants are included in Appendix C.1.47.

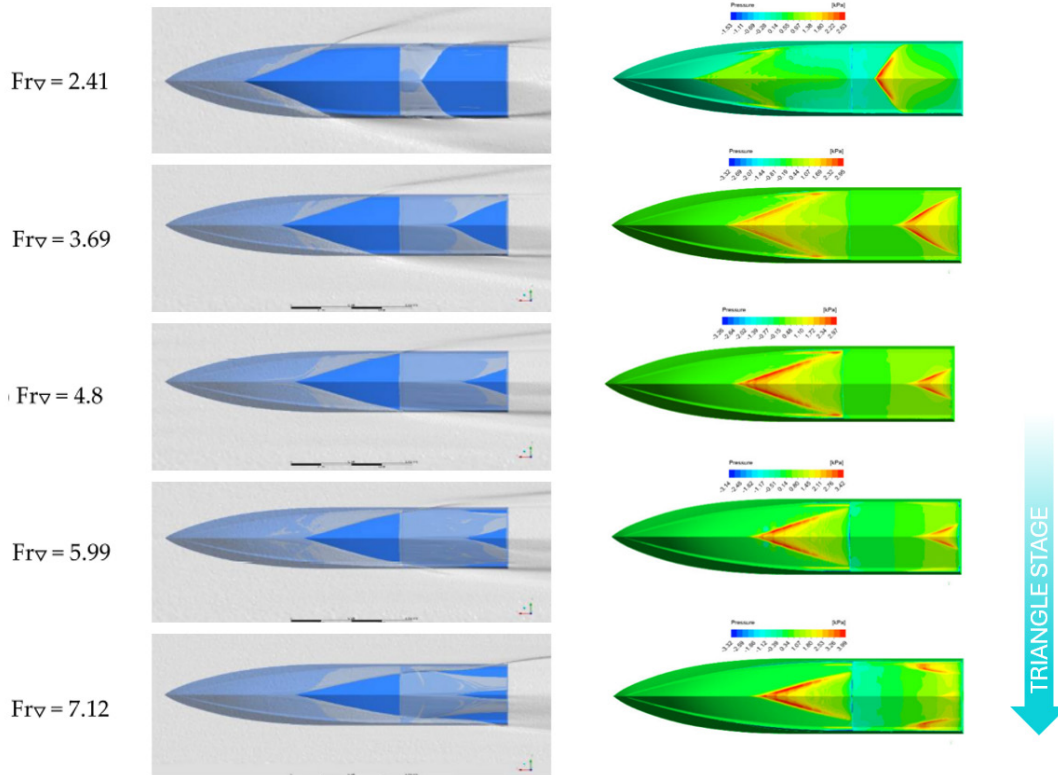


Figure C.23.: Single step hull and triangle-planing regime (source: [26])

C.1.32. Pounding and Porpoising

Boat pounding, also referred to as slamming or porpoising, denotes the phenomenon whereby the hull of a vessel experiences repeated impulsive loads as it re-enters the water surface in a seaway. This effect is particularly pronounced when operating at planing or semi-planing speeds in short or steep wave conditions. The physical process results in significant transient pressure peaks on the hull bottom, which can induce both structural stresses and severe onboard vibration, adversely affecting structural integrity, steering control, and crew comfort.

Cause of Wave Pounding

The primary mechanism of pounding arises from the relative motion between the vessel and the incident waves. When the bow lifts clear of the water, either due to high forward velocity or steep wave slopes, the subsequent re-entry generates a localized high-intensity impact as the hull bottom strikes the free surface. The magnitude of this impact is influenced by hull geometry, speed, displacement, and wave steepness. Flat-bottom or shallow-V hull forms exhibit elevated slamming loads due to a larger impact area and reduced deadrise angles, while deep-V hulls or multi-hull configurations typically distribute

the impact more progressively, reducing pressure peaks. Therefore, deep-V hulls are more suitable for offshore and medium to high speed vessels as they tend to smooth the wave impact.

Structural Implications

Slamming events contribute a cyclic load to the stresses on a body, which can accelerate fatigue in the hull plating, stiffeners like structural ribs, and connections. Additionally, localized material failures may arise in areas subjected to repeated impact, particularly at the bow bottom and forward sections of planing craft. Be-

yond structural considerations, severe pounding reduces operability by increasing pilot fatigue and decreasing navigational precision, as transient lifting of the hull may reduce controllability. Also, during pounding, the hull efficiency

will significantly deviate from optimal designed performance as this is highly affected by the trim angle, the angle of attack as well as energy that is lost during the slamming due to water displacement.

Mitigation Strategies

Several design strategies have been developed to mitigate pounding phenomena:

- **Hull form optimization** - Increasing the deadrise angle in the forward sections enables wave-cutting rather than wave-slamming behavior. Deep-V geometries, while reducing hydrodynamic efficiency at low speeds, significantly improve seakeeping and slamming resistance [86].
- **Adjust thrust trim angle and CG** - An often cause of porpoising behavior is a sub-optimal trim angle. The thrust vector is pushing slightly down, pushing the bow up into the air. When directly related to hull geometry (lift characteristics), often the CG is placed too far backwards.
- **Air cushioning** - Studies show that entrapped air cushions modulate slamming pressure, reducing peak impulses and yielding smoother load profiles [86, 114].
- **Spray rails and chines** - The integration of spray rails and hard chines alters flow separation, redistributes impact loads, and reduces wetted surface transitions during re-entry.
- **Stepped hulls**: The introduction of transverse steps reduces the wetted surface area during planing and promotes controlled ventilation, decreasing the magnitude of slamming loads.
- **Structural reinforcement** - Reinforcing critical impact-prone regions of the hull, particularly the forward bottom plating, increases resistance to localized stresses. Advanced materials, such as fiber-reinforced composites with tailored layups, provide additional fatigue resistance under repeated impulsive loads. CFD-FEA coupling may provide useful insights in this behaviour [115].
- **Shock-mitigating features** - Beyond hull geometry, shock-mitigating seats and flexible structural connections may be employed to protect occupants from vertical accelerations without significant alteration to hydrodynamic performance.

Although hull design is the primary determinant of slamming susceptibility, operational strategies remain critical. Reducing speed in adverse sea states, adjusting the trim angle, and altering the vessel's heading relative to incident waves can substantially reduce the severity of pounding events.

Aerodynamic Induced Pounding or Porpoising

In planing hull designs that rely on aerodynamic lift to reduce hydrodynamic resistance and wetted area at cruise speed, lift may transition from predominantly hydrodynamic to substantially aerodynamic as speed increases. Even in calm water, this can introduce a self-excited oscillation—porpoising—and impulsive re-entries (“aerodynamic pounding”) when the center of gravity (CG) is located forward of the center of aerodynamic lift (CL) [11]. The mechanism is an aero-hydrodynamic coupling as can be seen in Table C.11.

Time	Speed	Event	Angle of Attack
1	Low	Hull gains lift mainly from water pushing against the bow	Low
2	Medium	Hull gains lift mainly from aerodynamics	Medium
3	High	Hull lift collapse due to stall	High
4	Medium	Hull slamming down on the water surface.	Low
5	High	Hull gains lift mainly from aerodynamics	Medium
6	...	...	...

Table C.11.: Aerodynamic slamming sequence

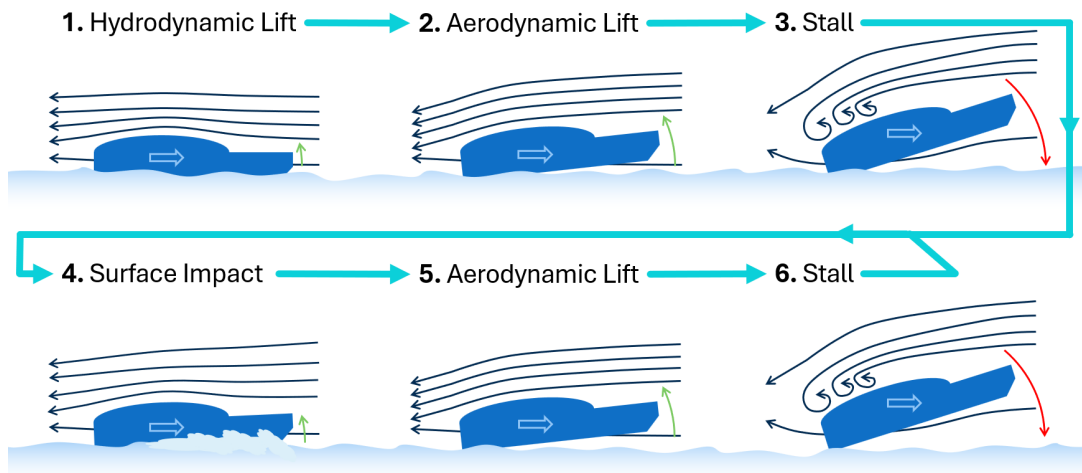


Figure C.24.: Aerodynamic pounding sequence

The mechanism in Figure C.24 can be described as follows: as speed increases, aerodynamic lift develops, reducing the wetted surface area and thereby decreasing hydrodynamic drag. However, when the lift forces act ahead of the CG, the effective angle of attack of the hull increases until a stall occurs. The subsequent stall causes a sudden loss of lift, resulting in an impulsive re-entry of the hull onto the water surface. This cycle may repeat at a characteristic frequency close to the vessel's natural heave–pitch resonance (first mode), thereby amplifying the oscillatory response and impact severity. The phenomenon represents an aero-hydroelastic coupling, where aerodynamic lift variations and hull motions reinforce each other, producing severe transient loading.

At even higher speeds, instability can manifest without stall. In this case, lift continues to increase with the angle of attack until the restoring

moment is insufficient to counteract the nose-up pitching moment. A “blowover” or back-flip can result, initiated either by a small wave encounter or by rapid acceleration. This represents an aero-hydrodynamic instability unique to lightweight planing craft and tunnel hulls at high Froude numbers ($Fr > 3$), where inertial and aerodynamic effects dominate over buoyancy. In this regime, even minor variations in trim angle can induce large changes in lift, driving oscillations that couple strongly with the natural pitch–heave modes and, at extreme values ($Fr > 3$), potentially causing blowover. Preventive design measures include positioning the CG sufficiently close or very slightly forward of the CL, modifying aerodynamic lift distribution, ensuring predictable and stable behavior when fully airborne, and ensuring adequate pitch stiffness to prevent resonance with the primary hull motion modes.

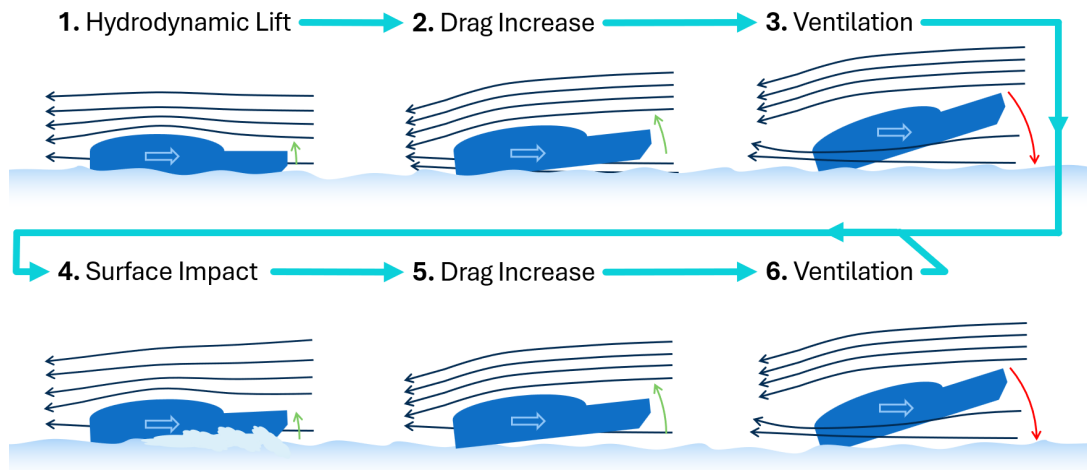


Figure C.25.: Aerodynamic porpoising sequence

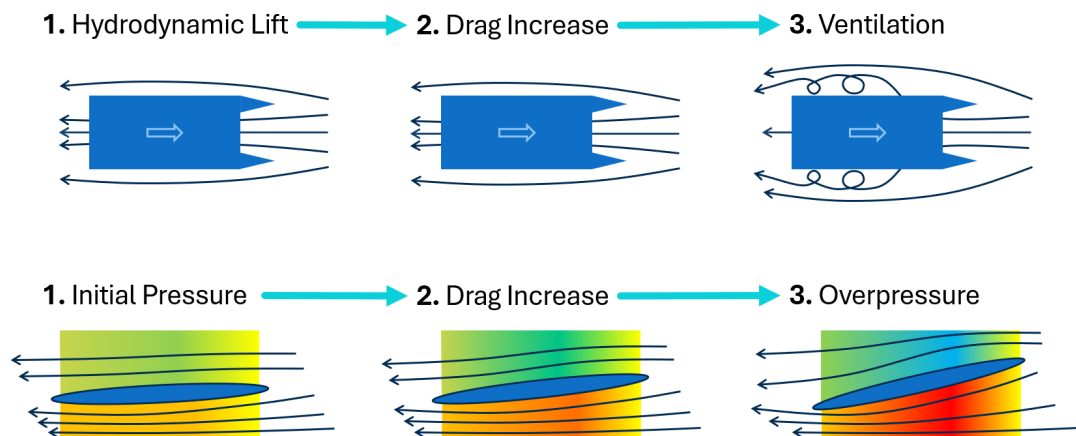


Figure C.26.: Aerodynamic porpoising pressure build-up

A similar event - often described as porpoising - is illustrated in Figure C.25. Tunnel hulls require thoughtful design as the tunnel can also cause an increase in drag rather than lifting the body out of the water, reducing drag. Porpoising is the phenomenon in which the pressure difference becomes too large causing major air leakage from the tunnel under the demi-hulls towards the side. This causes a sudden pressure drop causing the hull to repeatedly slam into the water surface, thus heave instability. The pressure build-up is illustrated in Figure C.26.

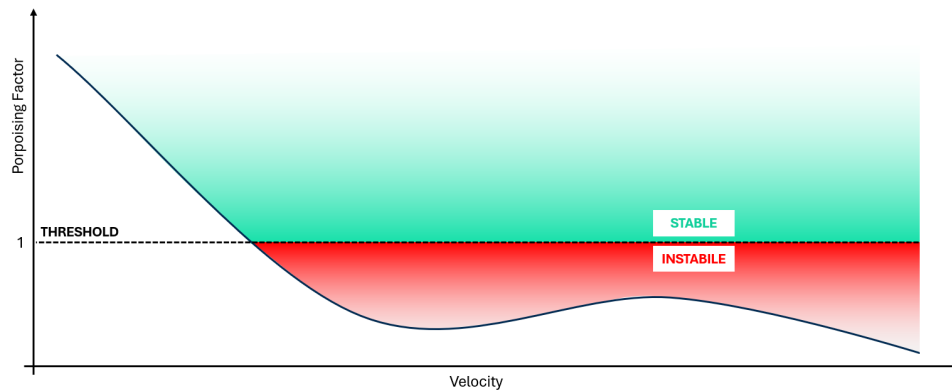


Figure C.27.: Theoretical porpoising susceptibility versus velocity - example

C.1.33. Mathematical CFD Frameworks

The foundation of computational hydrodynamics can be separated in two major historical levels: the development of the Boundary Element Method or BEM (fast, preliminary study) and the Finite Volume Method or FVM (slower, general purpose), which is most often coupled with Reynolds-Averaged Navier–Stokes (RANS) formulations. These approaches differ in the level of physical detail, computational cost, and historical development. An overview is given by Datawave Marine Solutions [47] and briefly discussed below.

Boundary Element Method (BEM) — The Panel Code Era

The Boundary Element Method, also known as the Panel Method/Code as it defines model boundaries as panel-fields, represents the earliest form of numerical hydrodynamics, primarily used from the 1960s to the 1980s. It solves the flow potential only on the body's surface boundaries and generally assumes an inviscid, irrotational, and incompressible fluid. As such, viscosity and turbulence effects are excluded from the governing equations. BEM is computationally light. Typical solutions can be obtained within seconds, making it particularly useful for preliminary design, optimization, etc. However, its inability to resolve viscous effects or local flow separation limits its predictive capability for drag and real-world resistance estimation. It remains valuable for conceptual hydrodynamic evaluations, wave pattern prediction, and propeller or foil analysis, particularly in two-dimensional applications where potential-flow assumptions are adequate.

Finite Volume Method (FVM) — Modern CFD Standard

The Finite Volume Method (FVM) directly discretizes the full Navier–Stokes equations (see Section 2.1.7), enabling the simulation of viscous, rotational, and unsteady flow fields. When combined with turbulence modeling through Reynolds averaging — yielding the RANS equations — this method captures both pressure- and shear-driven flow behavior with relatively low computational cost compared to more advanced turbulence-resolving models. While BEM computes solutions nearly instantaneously, RANS-based CFD typically requires between 30 minutes and 8 hours [47] for a converged solution, depending on mesh density, solver settings, and transient coupling. This approach remains the global standard for ship and propeller design, providing a practical balance between accuracy, stability, and computational cost. Today, CFD is implemented for both two-dimensional analyses, such as hydrofoil optimization, and full three-dimensional free-surface simulations. In many solvers, 2D results can be injected as initial boundary conditions for more complex 3D studies, enhancing convergence.

Turbulence Modeling

Turbulence remains the primary challenge in CFD. Its chaotic, multi-scale nature makes it impossible to fully resolve at engineering Reynolds numbers without substantial computational resources. As a result, several modeling strategies exist, each representing different compromises between accuracy and computational effort.

- **Laminar Flow** — Laminar simulations solve the Navier–Stokes equations without any turbulence model. They are occasionally used at high Reynolds numbers for preliminary evaluation or to isolate viscosity effects, but are unsuitable for realistic hull resistance predictions. They serve mainly as reference cases for model validation.
- **Reynolds-Averaged Navier–Stokes (RANS)** — The RANS approach averages the effects of turbulence and introduces a turbulence model (e.g., k - ϵ , k - ω , or SST) to approximate the influence of small-scale eddies. This reduces computational demand drastically while maintaining acceptable accuracy for streamlined bodies. It is currently the most widely used method in naval and aerospace CFD. Compared to laminar simulations, RANS typically increases computational cost by about 10–20 %, but improves the reliability of resistance and pressure predictions.
- **Large Eddy Simulation (LES) and Detached Eddy Simulation (DES)** — LES resolves large turbulent eddies directly while modeling only the smallest scales, leading to improved fidelity at the cost of high computational demand—roughly double that of RANS. DES combines LES in the flow field with RANS near the walls to reduce mesh requirements and is especially used for offshore structures, appendages, and unsteady wake analysis. For the current study, these methods are noted but not applied due to their computational expense.
- **Direct Numerical Simulation (DNS)** — DNS fully resolves all turbulent scales by directly integrating the complete Navier–Stokes equations without any modeling approximations. Although physically exact, it remains impractical for engineering applications, as the computational cost scales with the Reynolds number exponentially (roughly between Re^3 and Re^4 as the grid size $N \sim Re^{9/4}$ [116], see Appendix C.5). DNS is used almost exclusively for fundamental turbulence research.

C.1.34. Flow3D HYDRO Solver Arguments & Properties

Solver Characteristics

Flow3D HYDRO numerically solves the three-dimensional Navier–Stokes equations using a finite-volume discretization on a structured Cartesian mesh [117]. The key features of the numerical framework include:

- **Free-surface tracking** — the proprietary so called TruVOF algorithm captures the transient air–water interface, conserving mass accurately while resolving wave formation, spray, and air entrainment.
- **Geometry representation** — complex hull shapes are embedded in the Cartesian grid using the Fractional Area/Volume Obstacle Representation or FAVOR technique, eliminating body-fitted meshing and significantly reducing preprocessing effort. Both the Cartesian grid method and the reduced preprocessing time was clearly observed during the first experiments using the software.
- **Turbulence modeling** — several Reynolds-Averaged Navier–Stokes (RANS) closures are available, including the standard and RNG k – ϵ and the k – ω SST models, allowing turbulence modeling across laminar, transitional, and turbulent regimes. Discussing this in detail is far beyond the scope of this study.
- **Dynamic motion** — a fully coupled six-degree-of-freedom (6-DOF) rigid-body solver permits simulation of heave, pitch, roll, surge, sway, and yaw motions, enabling realistic seakeeping analysis. This, combined with the relatively simple setup workflow, was the most persuasive factor in adopting the software for this study.

Applications in Hull Resistance and Wave Analysis.

For naval architecture, Flow3D HYDRO is typically applied in the early and intermediate stages

of hull development to evaluate calm-water and seaway performance:

- **Resistance prediction** — by simulating calm-water towing or self-propelled tests, the solver determines total, viscous, and wave-making resistance components, enabling the comparison of alternative hull geometries such as stepped, slender, or multi-hull forms. This requires extended post-processing evaluation.
- **Wave formation and interaction** — the VOF formulation accurately reproduces wave patterns, wakes, and spray sheets, allowing quantitative assessment of wave interference effects (for example, in catamaran tunnel flows).
- **Dynamic stability and seakeeping** — the 6-DOF solver evaluates hull response in regular or irregular seas, quantifying heave, pitch, and roll motions, and assessing effects such as deck wetness or tunnel ventilation.

Strengths and Limitations

In essence, the main advantages of Flow3D HYDRO for naval hydrodynamics are:

- Efficient modeling of complex free-surface phenomena, including wave breaking and spray formation along coupled body interaction.
- Simplified geometry handling through the FAVOR method, which accelerates iterative hull development.
- Moderate computational cost compared with fully unstructured solvers, making it suitable for parametric design studies.

However, several limitations must be acknowledged:

- The structured Cartesian grid requires locally fine resolution around sharp features such as chines or transom edges.

- RANS-level turbulence models cannot fully resolve small-scale vortical structures or cavitation.
- Propulsor–hull interactions are typically approximated or treated via empirical thrust models rather than direct simulation. A relative to hull fixed time dependent force vectors were used in this study.

Role within the Naval CFD Workflow

Within a complete naval design framework, Flow3D HYDRO occupies an intermediate po-

sition between empirical prediction methods (e.g., ITTC-1957 friction correlations) and high-fidelity CFD (unsteady RANS or large-eddy simulations). It is best suited for:

- Parametric hull comparisons to determine resistance trends and wave interaction effects.
- Verification of design concepts such as step geometry, tunnel shape, or sponson spacing in multi-hulls.
- Early-stage optimization where computational turnaround time is critical and fine propulsor modeling is not (yet) required.

C.1.35. PIV Principle

In general, these steps are included in PIV measurement (also see Figure C.28):

1. **Constant flow** — In the first step, a steady-state flow is introduced in a wind tunnel's or water flow tank's intake.
2. **Seeding** — The seeder, responsible of introducing homogeneously distributed particles in the flow, will make sure the region of interest is sufficiently seeded with solid or droplet particles. The size should be matched with the capturing equipment regarding resolution.
3. **Laser marking of particles** — A light sheet is created by a mechanical mirror setup or beam scattering lens. For the best results, two subsequent laser beam pulses must be fired in synchronization with the camera shutter.
4. **CCD capture** — Two images are captured with a separation of for example 10 ms matching the flow range, particle size, and laser sheet thickness. Often, high-speed cameras are used as Δt is small.
5. **Preparation** — The images are calibrated and cleaned: optimized optical detection parameters like increased contrast, images are being scaled, and specific zones are excluded from the analysis to increase result quality.
6. **Calculation** — As the image/laser pulse timing is known, the particles of two images can be correlated as being the same on the condition that the resolution is sufficiently high and the images are close together on the timeline (depends on the maximum flow speed to be measured). To increase correlation accuracy, the maximum expected velocities are set and sub-pixel interpolation is used in most PIV-software available. The algorithm is searching for a peak illumination to distinguish a particle from the background.
7. **Post-processing & visualization** — PIV post-processing usually includes separating realistic values from faulty detections. PIVlab uses a 2D scatter graph for determining the valid domain. After post-processing the data acquired, various visualizations can contribute to analysis: heatmap (accuracy,...), streamlines, velocity vectors, vorticity, etc. or animations if sequential frames are also included.

To calculate the true 3D-displacement and velocity of particle, a third velocity component should be determined. This can be achieved by placing 2 cameras at an angle pointing towards the center of a laser sheet target area. By stereoscopic PIV, a 3D-displacement map can be analyzed. This can be useful for studying laser sheets perpendicular to the direction of flow.

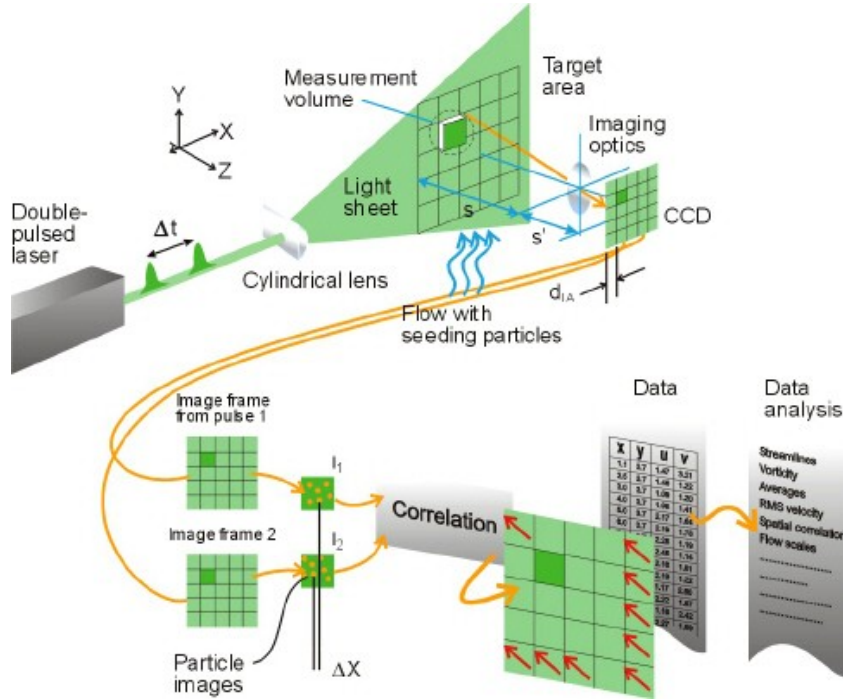


Figure C.28.: Particle Image Velocimetry (source: Dantec Dynamics A/S)

C.1.36. PIVlab Image Pairs Settings

Number of image pairs

Take N as the number of image pairs (frame-couples) used in a PIV run. The minimum is $N = 1$ as one pair produces one instantaneous displacement field. For better statistical quality (averaging, reducing noise), typical practice is

$$N \sim 10 \text{ to } 100 \text{ (or more).}$$

Pixel Displacement

Let $\Delta x, \Delta y$ be the pixel displacements of a particle tracer between two successive images along the x, y axes. Define the total displacement magnitude as

$$\Delta p = \sqrt{(\Delta x)^2 + (\Delta y)^2} \quad [\text{in pixels}].$$

Some practical guidelines found and validated by experimenting with PIVlab:

- For reliable cross-correlation, $\Delta p \approx 3$ to 7 pixels is often considered a “sweet spot.”
- In more permissive ranges, Δp up to 10–12 px may still work, depending on interrogation window size and overlap as found during experimentation with the software.

- Rule-of-thumb: the interrogation window size W (in pixels) should be larger than a multiple of Δp . For example:

$$W \gtrsim 4\Delta p,$$

so that the correlation peak does not “wrap around” or become ambiguous.

- In PIVlab’s documentation, it is stated that a displacement more than 32 pixels may flip the correlation peak to the opposite side of the correlation matrix, making correct measurement impossible as large displacements are misinterpreted as motion in the reverse direction.

Interrogation Window Passes in PIVlab

PIVlab typically uses a multi-pass FFT (Fast Fourier Transform) window deformation algorithm. It starts with a large interrogation window (e.g., 128×128 px), then reduces window size in subsequent passes (e.g., 64×64 , then 32×32). Using overlapping windows (e.g., 50 % overlap is common) helps to increase spatial resolution and get better correlation. However, making the last pass’ interrogation window too small relative to the displacement or particle density may lead to noise or erroneous vectors. A smaller window improves spatial resolution but reduces correlation strength (less particles leads to noisier results).

C.1.37. Seeding Types

Type	Size	Scattering	Pro	Con / remarks
Liquid aerosol (oil droplets, DEHS (Di-Ethyl-Hexyl-Sebacate))	0.5–1.0 μm [118]	good	excellent flow following in gas, good contrast, easy generation	may require precise seeder, potential deposition / optics contamination
Theatrical fog / glycol fog	1–5 μm [60, 119]	moderate to good	readily available, easy to generate	broad size distribution
Solid microparticles (e.g., powdered sugar, polystyrene, polyamide)	5–100 μm [120, 121]	excellent	controllable size, good contrast	inertia / lag in fast acceleration or shear zones
Hollow glass spheres (HGS)	9–13 μm [121]	good	low density (hollow), good in liquid	costly, use when size uniformity is important
Fluorescent particles (liquid or solid)	10–50 μm (in liquid) [121]	strong fluorescence signal	high signal-to-background contrast	requires matching optics / filters
Gas-filled bubbles / soap bubbles (e.g., helium-filled)	10–100 μm or more [122, 123]	strong scattering & reflection	very low effective density, excellent flow following in gas	generation complexity, bubble stability issues

Table C.12.: Comparison of tracer particle types for PIV / flow diagnostics

C.1.38. Wind Tunnel Setup

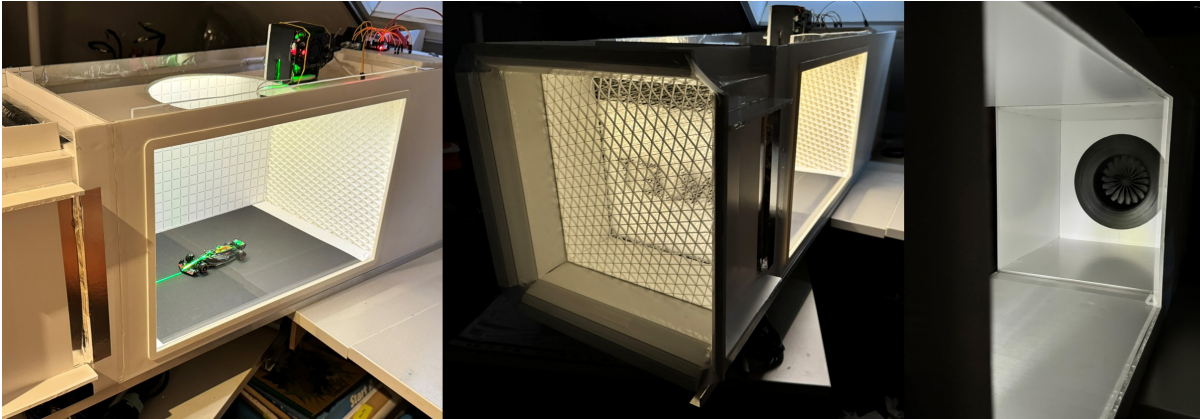


Figure C.29.: Wind tunnel during construction

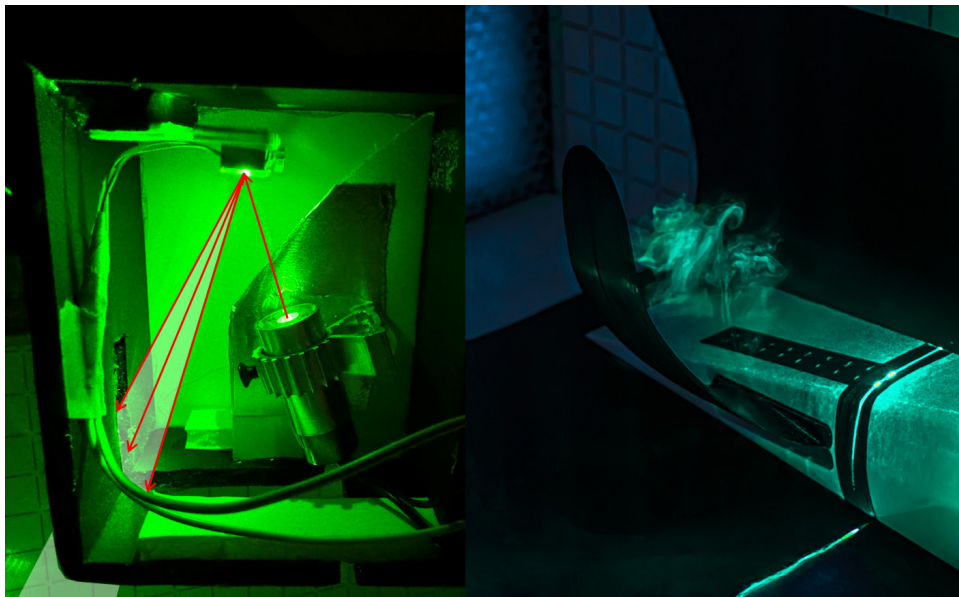


Figure C.30.: Laser-mirror oscillator

C.1.39. Particle Accelerator

Frequency	110 kHz
# Perforation diffusion holes	740
Average particle size	5 μm
Power	3 W (dual atomizer, 5 V, $2 \times 300 \text{ mA}$)

C.1. LITERATURE STUDY BACKGROUND

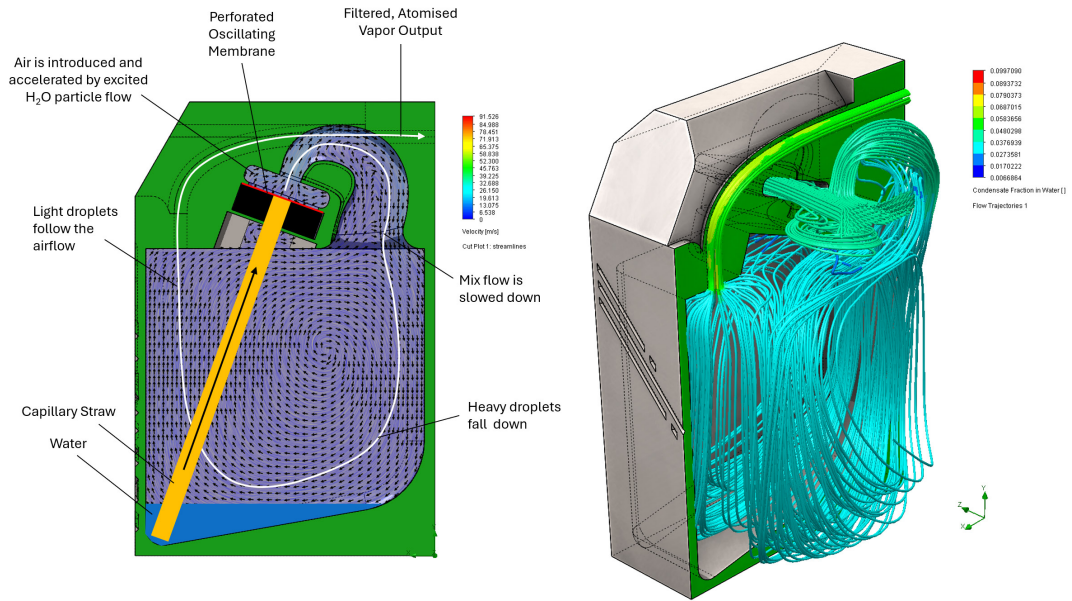


Figure C.31.: Particle accelerator & separator

C.1.40. Powell Lens Implementation

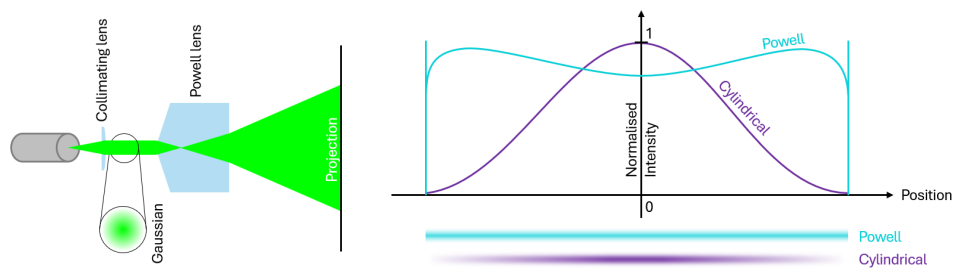


Figure C.32.: Powell versus cylindrical normalized laser intensity distribution

C.1.41. Froude & Hughes

In “Ship Resistance and Propulsion” by Mollard et al. and guidelines by the ITTC, the coefficient of wave making is identical for both the model and the full-scale prototype. In contrast, $C_{viscous,model} \neq C_{viscous,prototype}$ and should be corrected using the ITTC-1957 correlation line (equation C.7, estimating skin friction), a major breakthrough in naval sciences.

Concluding, in the history of ship design, only two techniques for model-prototype translation

are commonly used:

- Froude’s methodology (1870s) — based on Froude similarity explained in Section 2.1.2, direct C_R transfer.
- Hughes’ methodology (adopted by the ITTC). (1950s) — more accurate, accounting for Reynolds inconsistency of Froude scaling (ITTC’57 & Prohaska form factor), correction via C_F and $(1+k)$.

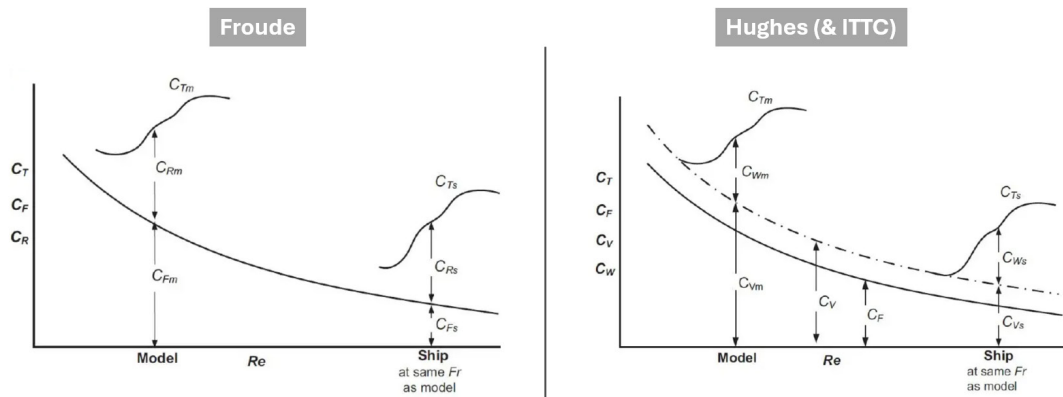


Figure C.33.: Froude vs Hughes (source: [19])

A detailed description of both methodologies can be found in chapter 4 “Model-Ship Extrapolation” of the “Ship Resistance and Propulsion” book written by Mollard et al. [19].

Simplified, the model testing procedure is [124]:

1. Data recording (model)
2. Determine resistance coefficient
3. Subtract out viscous coefficient (ITTC'57 correlation line, model)
4. Estimate new viscous coefficient (ITTC'57 correlation line, prototype)
5. Add in new viscous coefficient
6. Apply the full size and actual speed of the ship

C.1.42. Trim Angle Conventional Response

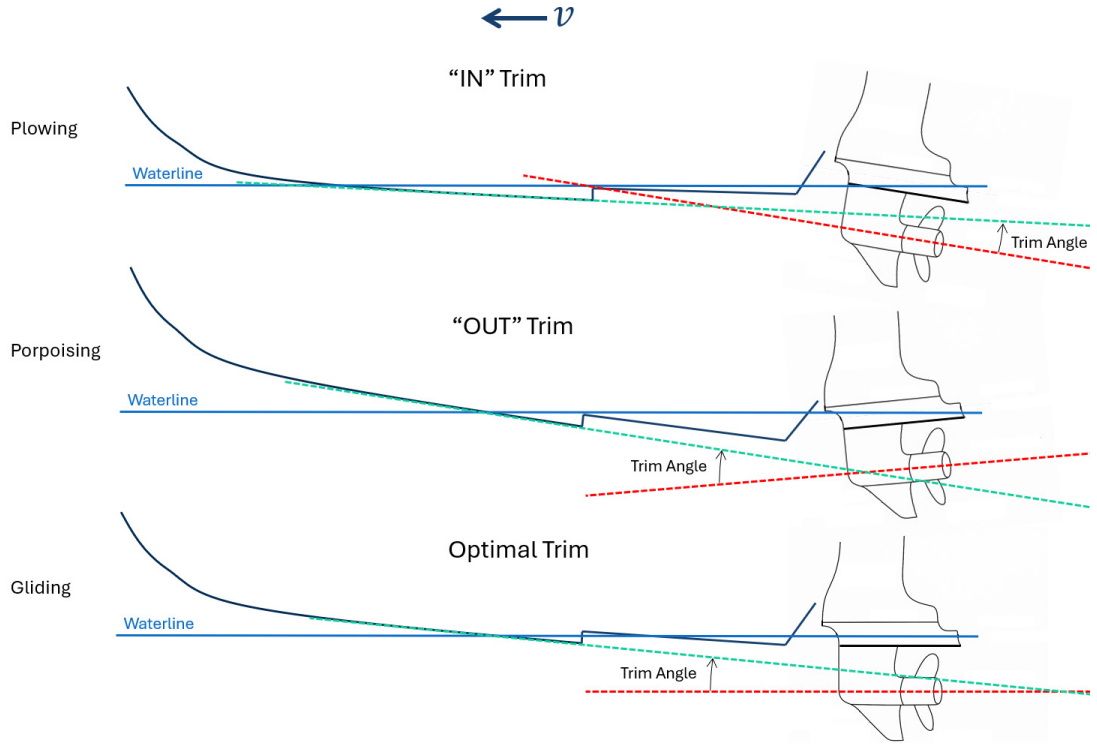


Figure C.34.: Trim angle

C.1.43. Steering-Proportional Trim Concept Illustrated

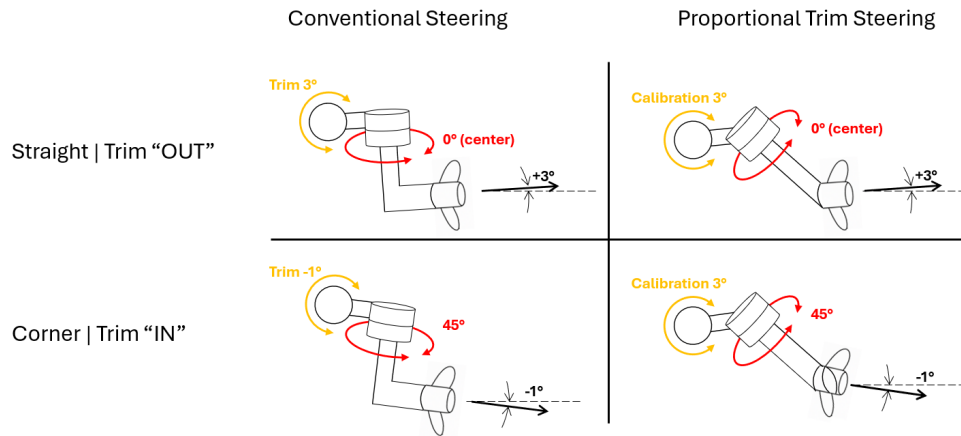


Figure C.35.: Tilted steering proportional trim

Trigonometry shows that the trim angle is given by:

$$\alpha_{trim} = (1 - \cos(\beta_{shaft})) \cdot \gamma_{ilt} \quad (C.21)$$

For the calculation of the trim angle, the calibration is set to zero. The influence of β_{shaft} and γ_{ilt} (45° is common in RC systems) on α_{trim} is plotted in Graph C.36.

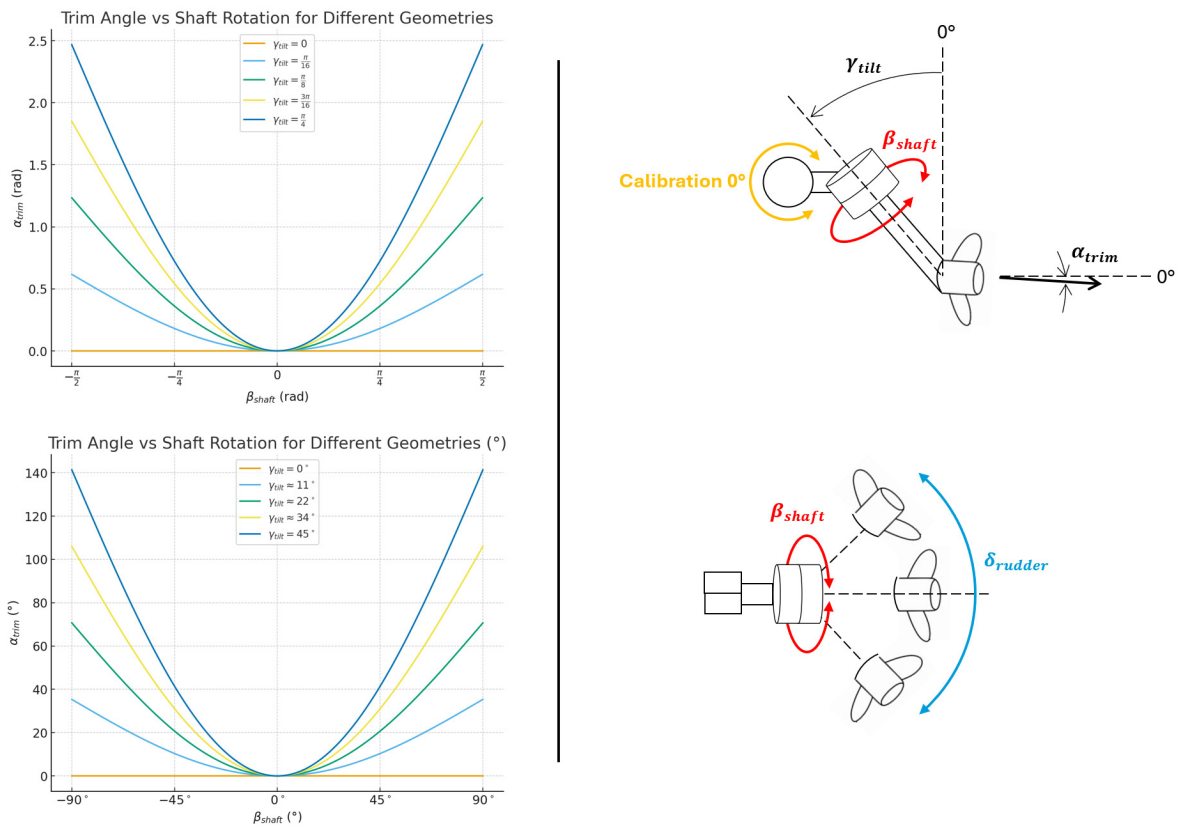
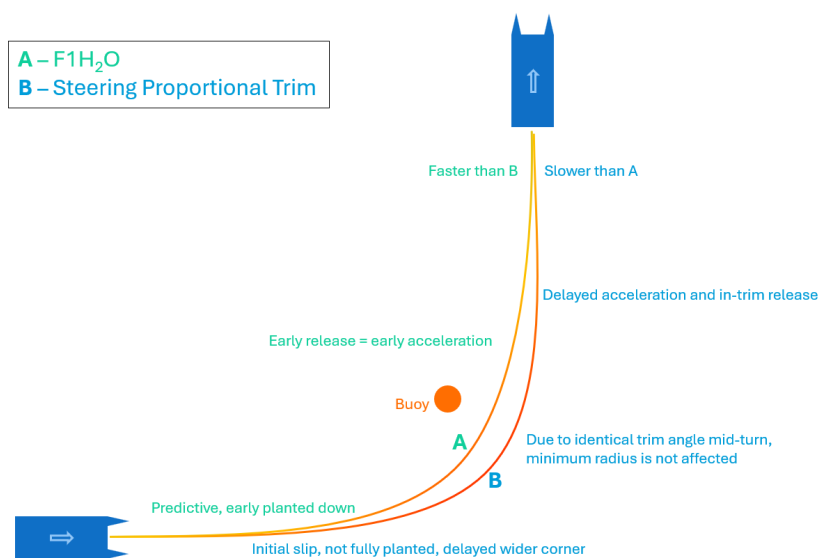


Figure C.36.: Trim Angle for Different Shaft Angles and Geometries

C.1.44. F1H₂O versus Proportional Trim Concept

Figure C.37.: F1H₂O pilot versus tilted steering proportional trim cornering

C.1.45. Trim Losses & Long Stem Compensation

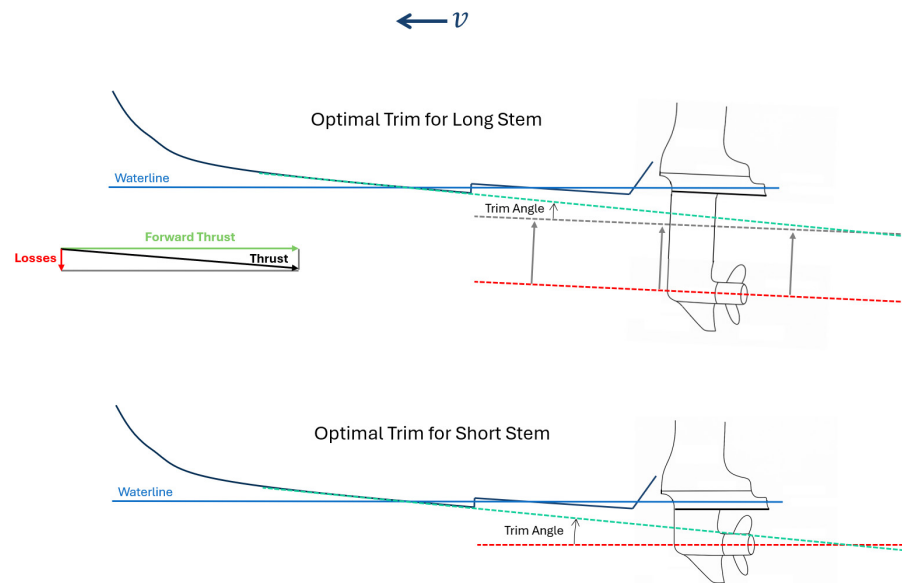


Figure C.38.: Additional trim losses using a long stem

C.1.46. Propeller Clearance

Table C.13 summarizes recommended clearance proportions (relative to propeller diameter D) from various maritime authorities (LR, DNV, NPL, NSMB, GL). These minima are typically expressed as fractions—such as $0.08D$ to $0.15D$, $0.10D$, or $0.18D$, depending on the classification of the maritime authority (see symbols; Section A).

The categories (a, b, c, d) refer to different clearance types — aft, front, radial (prop-hull), radial (prop-below) respectively — and $t/1$ is the thickness ratio of the rudder, while ϕ relates to the angle of run of waterline $0.5 R$ above prop CL (consult DNV). These baseline values help ensure that propeller-induced vibratory pulses do not resonate with the hull or excite unwanted structural noise. That said, many classification guides caution that these values are minimums; local flow conditions, cavitation allowance, and hull geometry should always inform final design decisions.

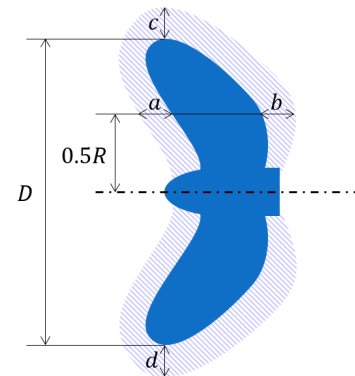


Figure C.39.: Propeller clearance

Clearance	LR	DNV	NPL	NSMB	GL
a	0.08–0.15 D	$(0.72 t/1) D$	0.08–0.15 D	0.08–0.12 D	0.10 D
b	0.15 D	$0.11 (1 - \phi)$	0.20 D	0.15–0.20 D	0.18 D
c	0.08 D	0.08 D	0.08–0.10 D	0.10–0.12 D	0.17 D
d	-	0.03 D	0.02–0.03 D	0.03 D	0.04 D

Table C.13.: Minimum recommended clearances between propeller and hull (relative to propeller diameter D) by classification rules (1970s–1990s)

C.1.47. Displacement-Based Non-Dimensionalization in Naval Architecture

In ship hydrodynamics it is conventional to use the displacement volume, denoted by ∇ (SI units: m^3), as a reference scale for geometry and resistance analysis. Many authors reserve Δ for displacement mass or weight; the link is $\Delta = \rho \nabla$ (mass) or $W = \rho g \nabla$ (weight). Powers of ∇ provide characteristic measures with clear dimensional meaning and enable comparisons across size while holding form constant as in the experiments of Molland et al. [17]. This custom is also adopted by standardization institutions as the ITTC [3]. The most frequently used forms are:

- ∇ (m^3) — displacement volume.
- $\nabla^{1/3}$ (m) — a characteristic length, often called the displacement length. It is used to non-dimensionalize principal dimensions, e.g. $L/\nabla^{1/3}$, $B/\nabla^{1/3}$, $T/\nabla^{1/3}$, which serve as displacement-based slenderness and fullness ratios. Note, T is the conventional symbol for draft.
- $\nabla^{2/3}$ (m^2) — a characteristic area, used to scale surface measures, e.g. $S/\nabla^{2/3}$ for wetted-surface normalization.
- $\nabla^{1/6}$ ($\text{m}^{1/2}$) — satisfies $\nabla^{1/6} = (\nabla^{1/3})^{1/2}$. It appears occasionally in empirical correlations that involve a square-root of a length scale; it is not a primary descriptor but a convenient shorthand when the underlying physics suggests $\sqrt{\text{length}}$ dependence.

These scalings are habitually combined with dynamic similarity parameters (e.g., Froude) to separate size from shape effects, to collapse experimental data across model-ship scales, and to construct dimensionless groups for regression and design-space exploration without introducing bias from absolute size.

C.1.48. Radio Transmission Regulations

In general (EU), 2.4 GHz ranges are typically used for public WiFi networking and are therefore less restrictive. This also makes them first choice in RC transmitter-receiver applications and digital/analog video streaming often using frequency-hopping spread spectrum (FHSS). 5.8 GHz is also common for video and telemetry streaming, unfortunately in Europe it is limited to only 25 mW of antenna transmission power (RTT&E regulations), which is in most cases unusable (note public WiFi and marine comms have specified access to some of these “5G” ranges, consult BIPT for more info).

For reliable telemetry transmission such as a MAVLink (Micro Air Vehicle Link) downlink to the ground control station or GCS — especially in boating environments — a lower frequency range is more popular due to less transmission absorption near water bodies. 433 MHz and 917 MHz are common ranges for this application. In Belgium, the Belgian Institute of Postal services and Telecommunication (BIPT) is responsible for the coordination of civil and military radio transmissions. They manage the domain sectioning and allowed power ranges of

certified equipment. In discussion with the authority, it was stated that 917 MHz transmissions in the range of 917.3-918.9 MHz (using radio interface B01-40) at a maximum antenna power of 500 mW using CE-certified transceivers are the most suitable and only legal method for this telemetry connection except for high-frequency WiFi ranges. For 433 MHz, the power is limited to only 10 mW, and therefore only used in low-power, short-range applications like a garage door remotes or children’s RC toys.

C.1.49. Vertical Stabilizer

Research indicated the usefulness of a vertical stabilizer at speeds around and exceeding 30 kn. As future team goals may imply even larger power increases, and thus high speeds, the proposal of a lightweight tail stabilizer is anything but overrated. Determining the size of a vertical stabilizer is mainly finding the balance between 2 major factors: (1) a large surface provides better stabilization and (2) a smaller surface adds less appendage surface drag. What is the minimum required surface area to provide sufficient stabilization? Note the aerodynamic stabilizer has a more pronounced effect at higher speeds ($> 30 \text{ kts}$) and is the most important for safety when the craft gets fully airborne due to wave jumping.

Method 1: Tail–Volume Coefficient

A robust first estimate for vertical-tail planform area S_V uses the vertical tail–volume coefficient [67, 68, 69]:

$$V_V = \frac{S_V \ell_V}{S_W b_W} \implies S_V = V_V \frac{S_W b_W}{\ell_V}. \quad (\text{C.22})$$

Here S_W and b_W (wingspan) are the reference area and span of the primary lifting surface, and ℓ_V is the moment arm from that surface’s aerodynamic center to the vertical tail’s aerodynamic center. For aircraft, typical V_V is 0.03–0.08 depending on category [67].

Adapting to F1H2O When a tunnel hull is partly airborne, its deck/tunnel including sponsons act as a low–aspect-ratio wing. Take S_W as the projected aerodynamic planform of the air-loaded surfaces when airborne (deck/tunnel roof between sponsons plus any aerodynamic cowl, excluding wetted areas), b_W as the lateral span (outer chine to outer chine), and ℓ_V measured to the proposed fin location on the engine cowl or transom support. Starting with $V_V \in [0.04, 0.10]$ reflects (1) short ℓ_V on boats and (2) stronger fuselage/engine-cowling destabilization than on slender aircraft. Raymer and Roskam claim that a tail-volume coefficient of 0.05–0.06 is optimal for “flying boat” and seaplane configurations, which a tunnel boat does share many relevant characteristics with [70, 71]. However, a refinement approach is required for a more optimal design choice.

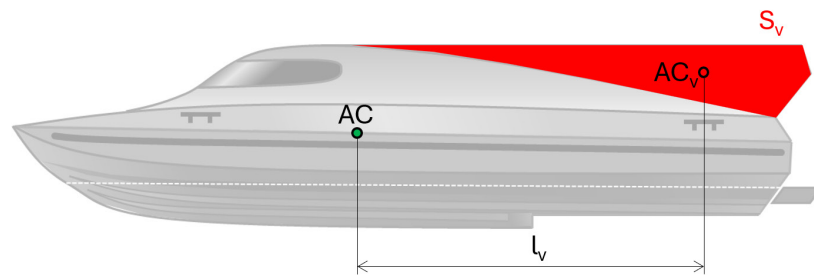


Figure C.40.: Tail-volume approach illustrated on the tunnel hull design.

A symmetrical airfoil is recommended for vertical stabilizers, as the net aerodynamic force reverses with rudder deflection. Thin symmetrical profiles, such as the NACA 0010 or NACA 0012, are preferred to minimize drag. Integrating the stabilizer with the cockpit can synergetically reduce drag by lowering tail frontal area and cockpit wake formation compared to a separate tail fin. From the sizing relations defined in equation C.22, placing the tail further aft of the aerodynamic center reduces the required surface area, which is advantageous for limiting skin-friction drag. This can be observed in the typical configuration of high performance sailplanes.

Method 2: Meet a target directional stability derivative (physics-driven)

Weathercock (yaw) stability — also known as yaw stiffness — requires a positive total derivative $C_{n\beta}$ (the yawing moment coefficient derivative with respect to sideslip).

$$C_{n\beta} \equiv \partial C_n / \partial \beta \quad (C.23)$$

with β sideslip angle.

The fuselage/cowl and exposed hull typically provide a destabilizing contribution $C_{n\beta}^F < 0$. The tail must offset this:

$$C_{n\beta}^{tot} = C_{n\beta}^F + C_{n\beta}^V > 0. \quad (C.24)$$

where

$$\begin{aligned} C_{n\beta}^F < 0 &= \text{contribution of fuselage/hull (estimate: } -0,02 \text{ to } -0,05 \text{ /rad for a wide body)} \\ C_{n\beta}^V > 0 &= \text{contribution of vertical stabilizer} \end{aligned}$$

Semi-empirical stability texts give the tail contribution approximately as [72, 67]:

$$C_{n\beta}^V \approx \eta_V a_V \frac{S_V}{S_W} \frac{\ell_V}{b_W}, \quad (C.25)$$

where a_V is the lift-curve slope of the tail airfoil (sideforce slope), and $\eta_V = q_V/q$ is the dynamic-pressure ratio at the tail (0,8–1,0; 1,0 if fully exposed to the flow, depending on the wake regions). Rearranging for area with a desired margin $\Delta C_{n\beta} > 0$:

$$S_V \geq \frac{(|C_{n\beta}^F| + \Delta C_{n\beta}) S_W b_W}{\eta_V a_V \ell_V}. \quad (C.26)$$

with $\Delta C_{n\beta}$ safety margin ($\approx 0,02$ – $0,05$ /rad is a typical target for good stability in small craft)

For preliminary work on a clean, free-streamed finite thin fin, a_V can be taken in the range 5 – 6 rad^{-1} for low-AR, swept plates at the Reynolds numbers of interest (less than the 2π thin-airfoil limit, see paragraph C.1.49). To accurately determine the a_V required, subsonic engineering approximations for limited sweep geometry, exist. These take the aspect ratio (AR), e the Oswald/ efficiency factor, the quarter-chord sweep and thin airfoil slope into account. However, it is beyond the scope of this thesis to investigate the exact numbers as the common approximations showed in this section are usually sufficient to achieve stable craft. $C_{n\beta}^F$ can both be computed from a simplified body method (e.g., DATCOM's fuselage model) or be estimated from CFD/windtunnel of the cowl and tunnel cross-section [72, 69]. Equation (C.26) then yields S_V and can be cross-checked with (C.22).

Thin-airfoil Lift Slope Theoretical Limit In classical thin-airfoil theory for incompressible, inviscid flow, the lift-curve slope of an ideal two-dimensional airfoil is given by

$$a_0 = \frac{dC_L}{d\alpha} = 2\pi \text{ /rad}. \quad (C.27)$$

Numerically, this corresponds to

$$a_0 \approx 6,28 \text{ /rad} \quad (\approx 0,11 \text{ /}^\circ).$$

This value represents the theoretical upper limit of the lift-curve slope; real wings and tail surfaces at finite aspect ratio and finite Reynolds number will always produce lower values.

C.2. Simulation Data

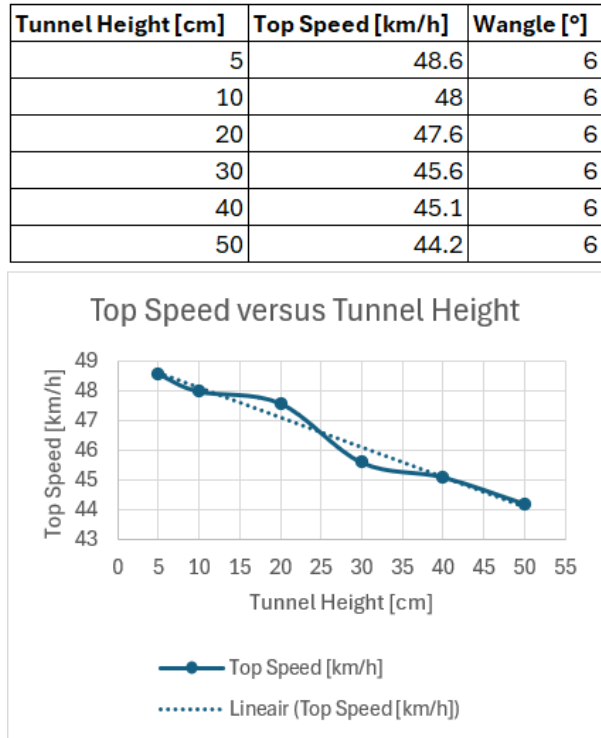


Figure C.41.: Tunnel height analysis

Run	Pad Deadrise [°]	Pad Deadrise Fore [°]	Top Speed [km/h]
F1 Config 4	10	60	44.7
F1 Config 5	10	30	44.9
F1 Config 6	10	15	45.6
F1 Config 7	10	10	45.8
F1 Config 8	6	6	46.6
F1 Config 9	0	0	47.7
F1 Config 10	6	15	46.2
F1 Config 11	1	30	46.4
F1 Config 12	15	30	44.0
F1 Config 13	8	15	46.1
F1 Config 14	8	30	45.2
F1 Config 15	8	40	45.0
F1 Config 16	8	60	45.0

Table C.14.: Deadrise variations

Run	Tunnel [cm]	Pad [cm]	Deck [cm]	Top Speed [km/h]	Motor Power [kW]
F1 Config 17	110	40	225	47.8	6
F1 Config 18	110	50	225	48.1	6
F1 Config 19	70	70	225	47.2	6
F1 Config 20	70	70	225	79.2	15
F1 Config 21	150	30	225	83.5	15
F1 Config 22	160	25	225	84	15
F1 Config 23	170	20	225	83.6	15
F1 Config 24	133	26	200	83.4	15
F1 Config 25	141	22	200	83.6	15
F1 Config 26	149	18	200	83.9	15
F1 Config 27	155	15	200	83.9	15
F1 Config 28	115	20	170	84.2	15
F1 Config 29	155	15	200	83.9	15

Table C.15.: Tunnel, sponson and deck width variations

C.2. SIMULATION DATA

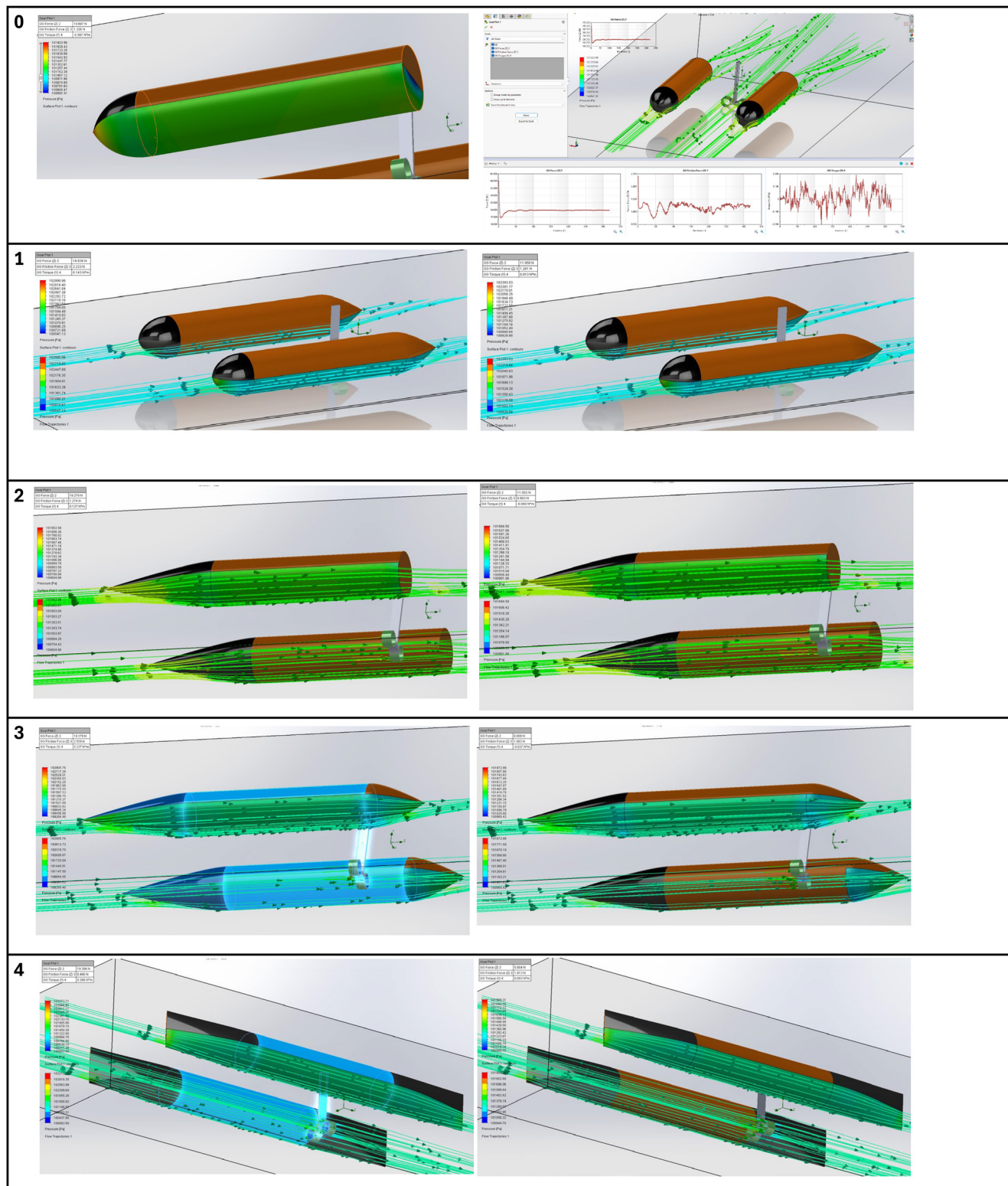


Figure C.42.: UASBT Test Platform Simulation Setups

Test Ref.	Flow Velocity [m/s]	Bow Fin L/D	Bow Fin Lift[N]	Bow Fin Drag [N]	Stern Fin Drag [N]	Hull Stern Extender (default)	Simulated Drag [N] (Flow3D)	Benchmark	Efficiency Compared w. Relevant Benchmark	Remark
1	5	0	0	0	0	Yes	217	Yes	0.0%	Benchmark simulation, no changes
2	5	4	168	43	7	Yes	264	No	-21.7%	
3	5	4	168	43	0	Yes	257	No	-18.4%	The shark does not benefit from lifting the bow by fins @ 5m/s
4	5	16	681	43	7	Yes	273	No	-25.8%	High aspect ratio bow fins (geometrically impractical & little tolerance for design speed deviation)
5	5	0	-200	0	7	Yes	260	No	-19.8%	Added local ballast of 200 N
6	5	0	-100	0	7	Yes	234	No	-7.8%	Indicating small bow ballast (100N) is possible with no significant compromise for drag
7	10	0	0	0	0	Yes	862	Yes	0.0%	Benchmark simulation, no changes
8	10	4	681	178	27	Yes	678	No	21.3%	21.3% drag reduction
9	5	0	0	0	0	No	214	Yes	1.4%	Minor positive influence on drag reduction (reduced wetted area)
10	5	4	168	43	7	No	255	No	-19.2%	Reduced by 3.4% compared to test 2, fins show increased positive influence @ 5m/s by removing stern extender
11	5	4	168	43	0	No	248	No	-15.9%	Removed stern fin
12	10	0	0	0	0	No	692	Yes	19.7%	19.7% drag reduction @ 10 m/s due to stern extender removal (free to trim dynamically and reduce wetted area)
13	10	4	681	178	27	No	540	No	22.0%	37.4% drag reduction compared to test 7, simulation showed minor porpoising ;0.5° oscillation (geometry will be adapted, foil max draft and surface area-depth will be designed to dampen the effect), wetted area reduced to 1.7 m² due to high trim

Table C.17.: Overview of simulations on the Shark's upgrades (Shark v2, Excel in portfolio)

Test Ref.	Flow Velocity [m/s]	Bow Fin L/D	Bow Fin Lift[N]	Bow Fin Drag [N]	Stern Fin Drag [N]	Hull Stern Extender (default)	Simulated Drag [N] (Flow3D)	Benchmark	Efficiency Compared w. Relevant Benchmark	Remark
14	10	3	878	318	-	-	-	-	-	Full submersion of fin foils, 0° hull pitch (inaccurate, the bow lifts dynamically only featuring partial submersion), wing profile features 2° AoA @ 0° hull pitch
15	10	8	846	102	5	No	417	No	39.7%	Lift and drag is based on the 2° hull pitch (due to dynamic trim), shows porpoising but stabilizes after 2 cycles, WA approx. 1.44 m², 51.6% drag reduction compared with the 10 m/s default Shark geometry response
16	7,5	0	0	0	0	No	505	Yes	0.0%	Benchmark simulation, WA approx. 4.3 m²
17	7,5	4	554	132	5	No	467	No	7.5%	Partial submersion of fin foils, 1° hull pitch, bow lifts free from the water, WA approx. 3 m²
18	5	6	215	35	5	No	258	No	-20.6%	Full submersion of fin foils, 0° hull pitch, WA approx. 3.9 m², noticeably reduced wave making at the bow compared to default response

Table C.19.: Overview of simulations on the Shark's upgrades (Shark v3, Excel in portfolio)

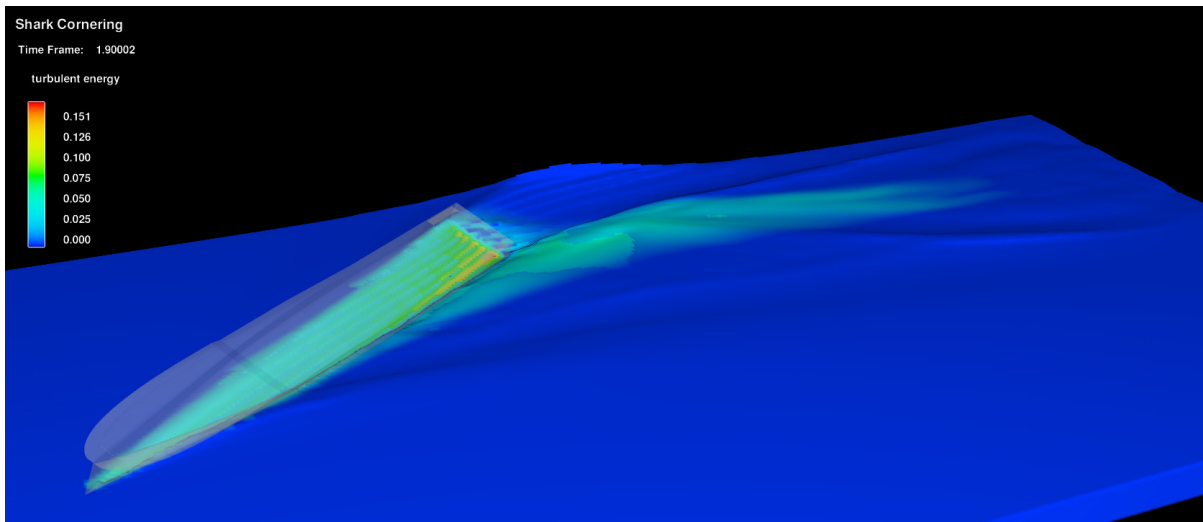


Figure C.43.: Shark's default configuration cornering at 6 m/s

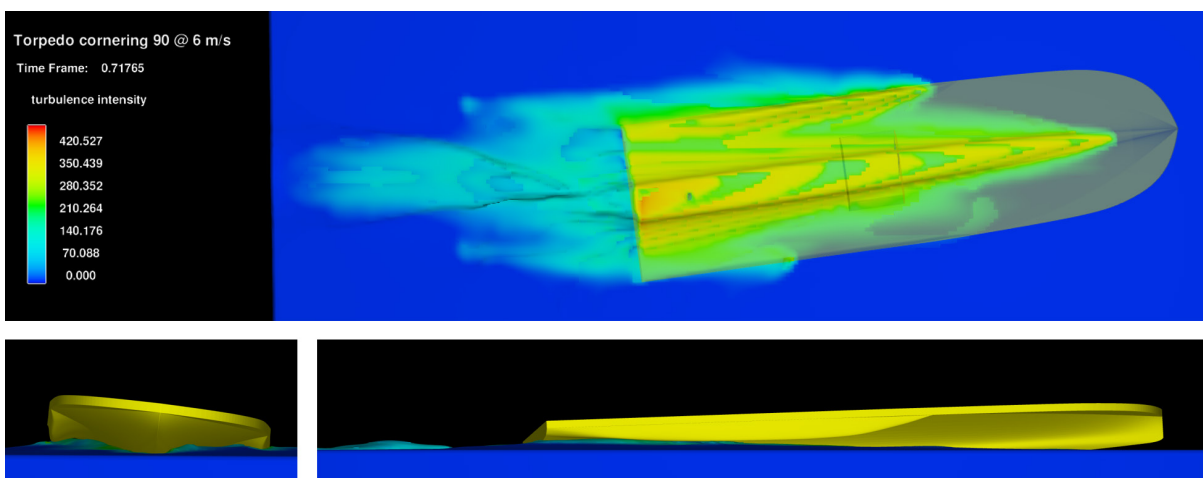


Figure C.44.: Shark's default configuration cornering at 6 and 10 m/s

C.2. SIMULATION DATA

Time: 2.000 s

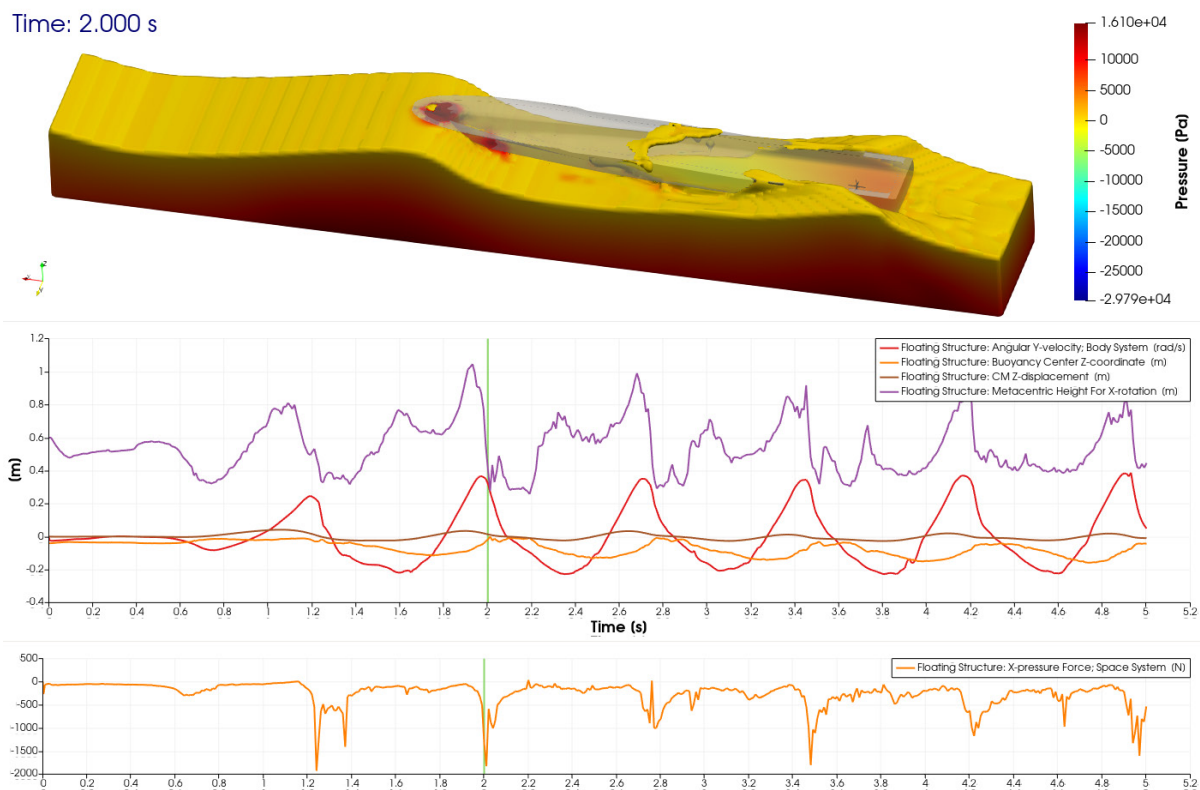


Figure C.45.: Results wave test 1 - Shark 5 m/s - 0.6x6m waves

Time: 1.900 s

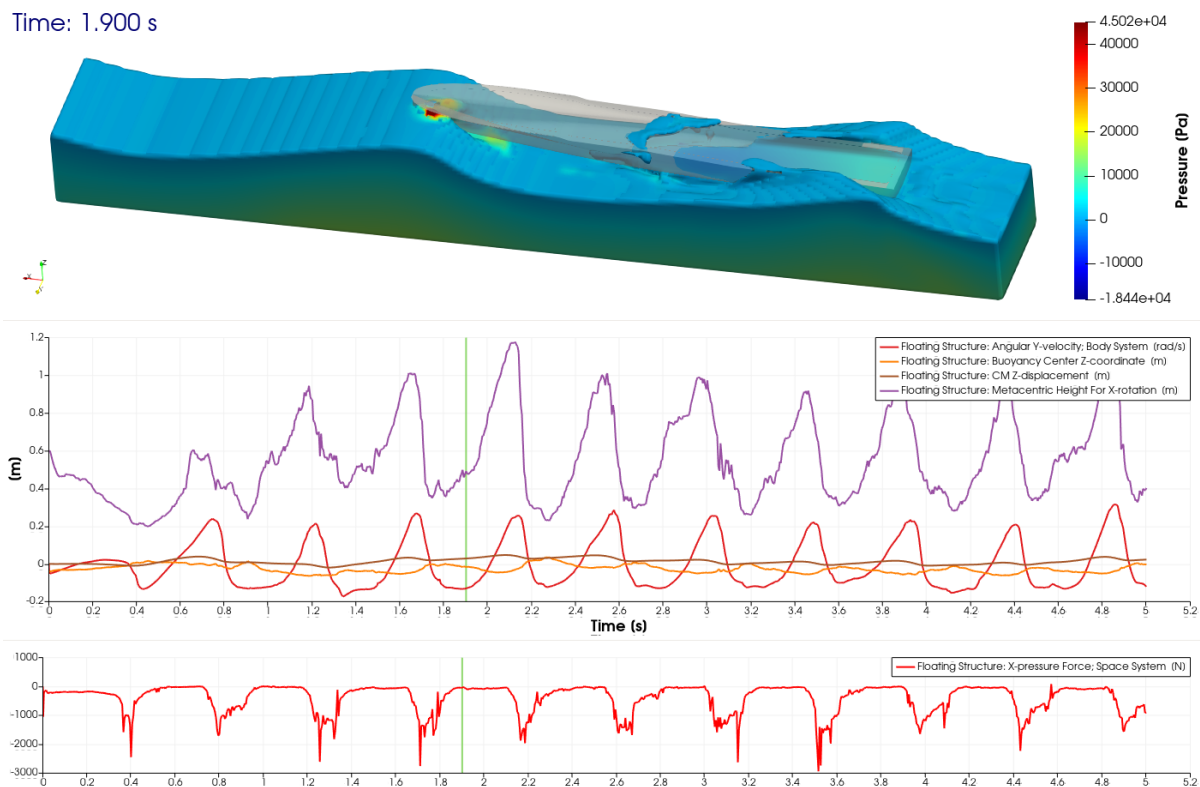


Figure C.46.: Results wave test 2 - Shark 10 m/s - 0.6x6m waves

Time: 2.200 s

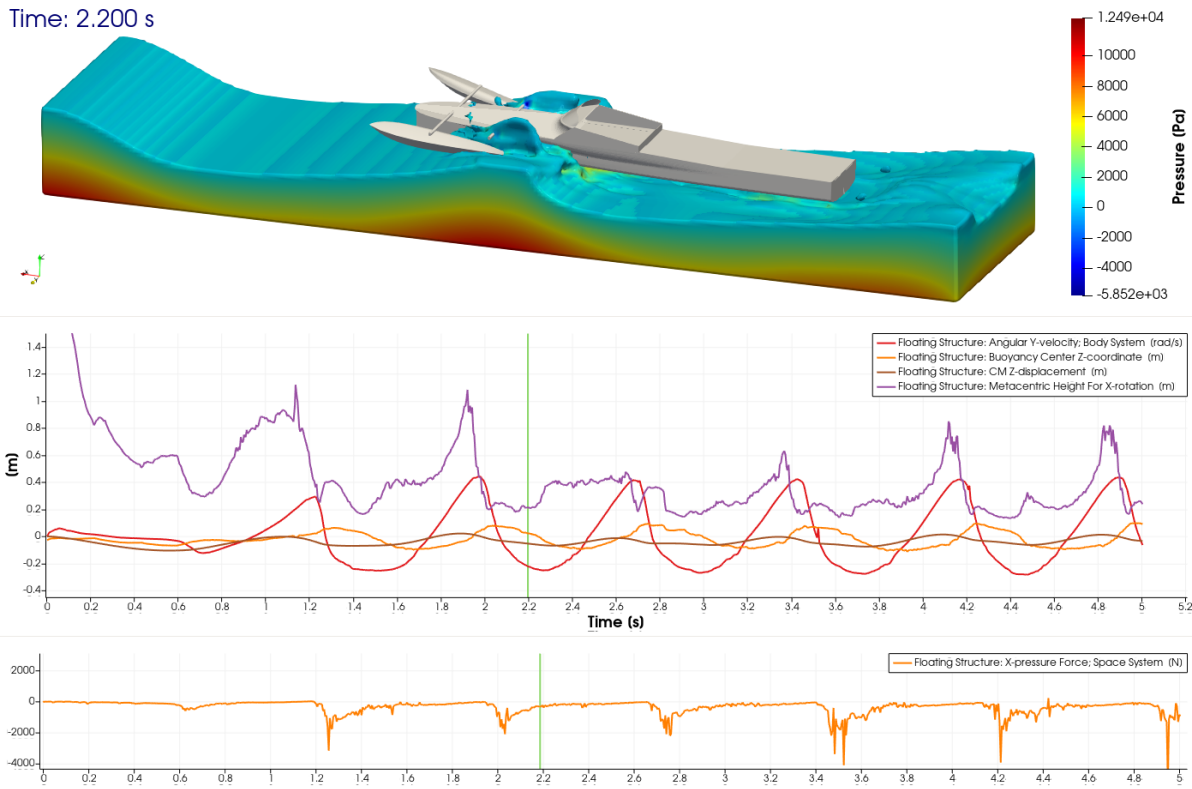


Figure C.47.: Results wave test 3 - upgraded Shark 5 m/s - 0.6x6m waves

Time: 1.300 s

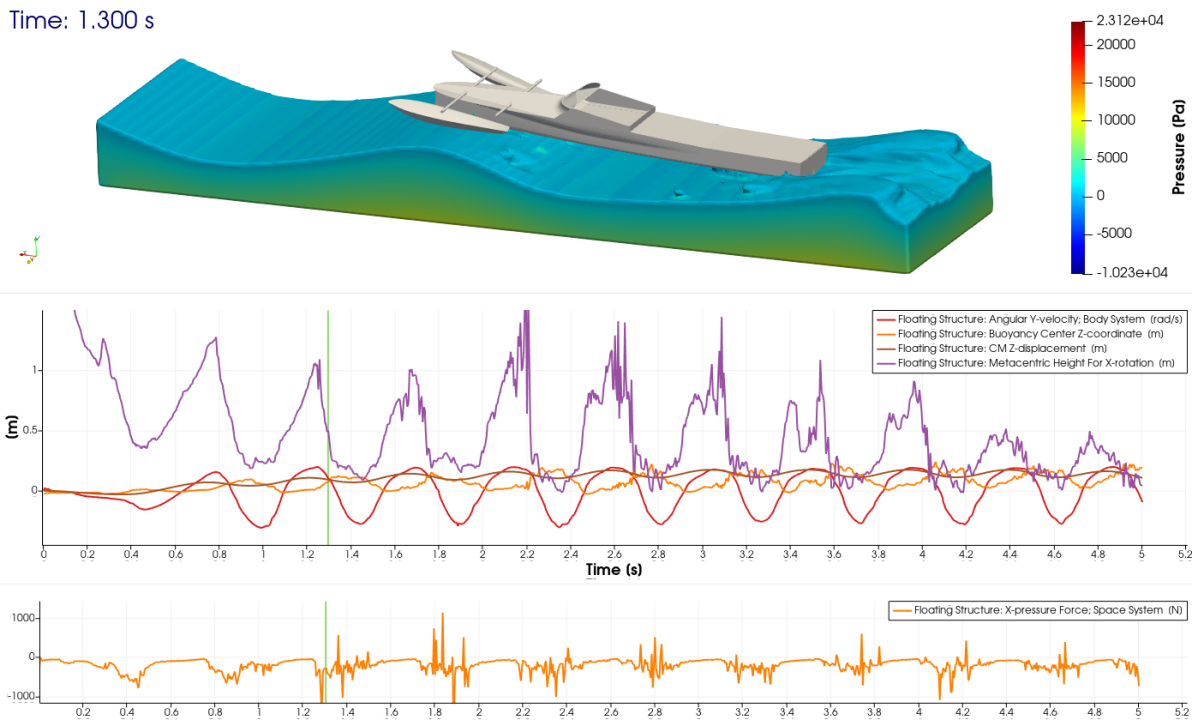


Figure C.48.: Results wave test 4 - upgraded Shark 10 m/s - 0.6x6m waves

C.2. SIMULATION DATA

Time: 1.328 s

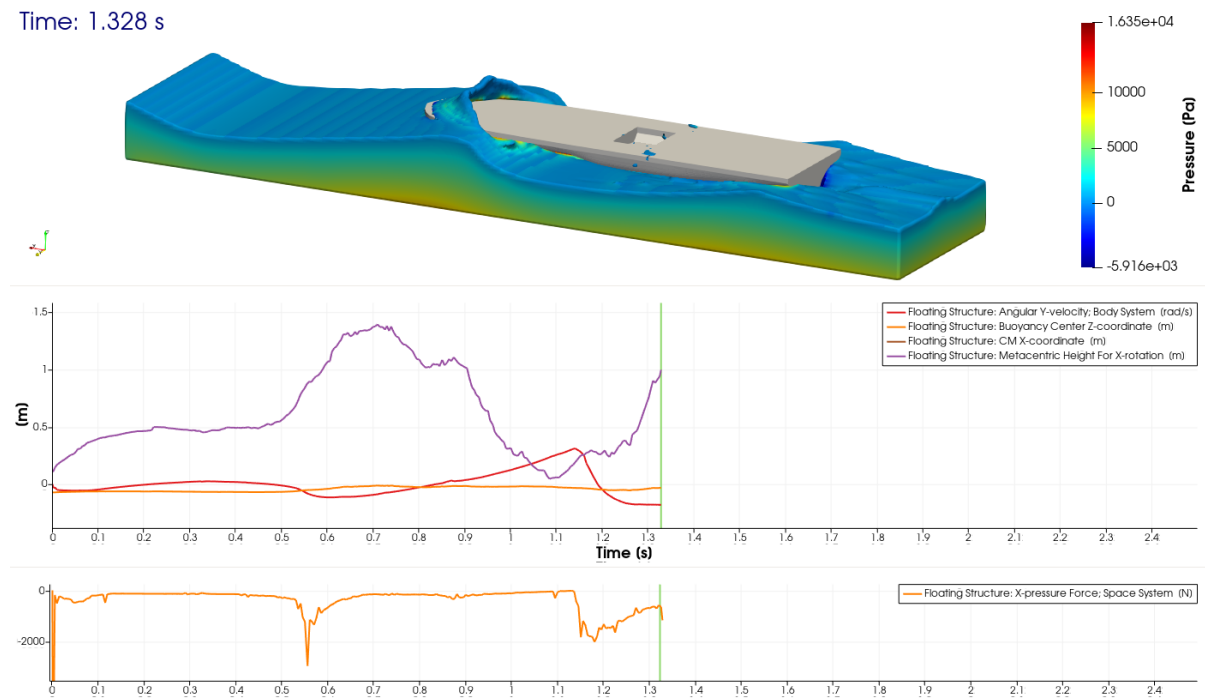


Figure C.49.: Results wave test 5 - Torpedo 5 m/s - 0.6x6m waves

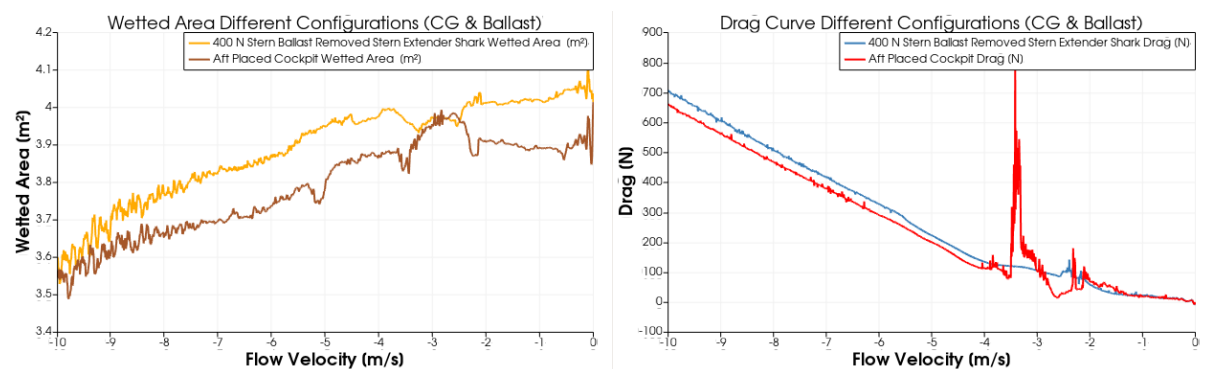


Figure C.50.: Resistance test of the Shark's cockpit (80 kg) aft placement compared with 400 N stern ballast (similar trim and CG, different total mass)

C.3. Experimental Data

Test 1 - Spontaneous Deceleration - 18-09-2025

Timescale	0,0625
PixelScale Torpedo (relative)	0,006315789
PixelScale Shark (relative)	0,006205
Recording FPS (stretched time)	15
Model Scale	0,1
Actual HS-CAM FPS	240

Torpedo	Video Timestamp [s]	0/0,000	Actual Time (model) [s]	0	1,0000	0,000	0,0063																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																							
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Figure C.51.: First test numerical data

Parameter	Torpedo Mk.II EVO	Shark	New Design
Slope [m/s ²]	-2.2244	-2.0995	x
Averaged Acceleration [m/s ²]	-0.4971	-0.4081	x

Table C.20.: Results of the first test

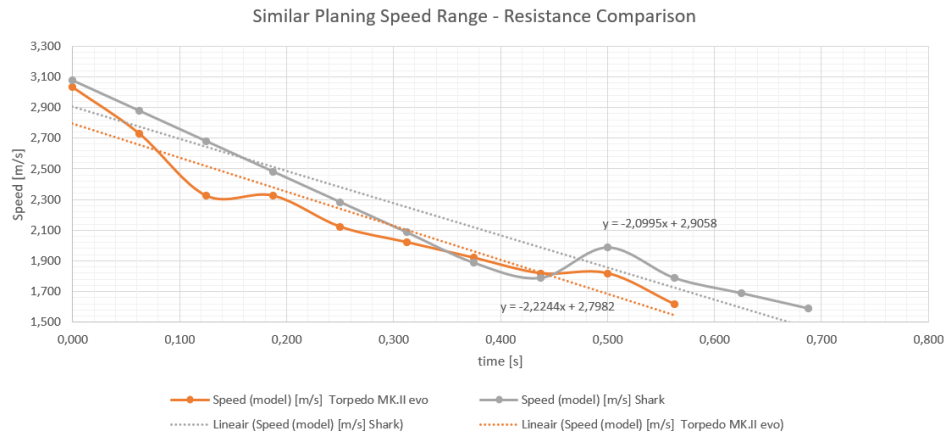


Figure C.52.: Torpedo Mk.II EVO and Shark slope resistance

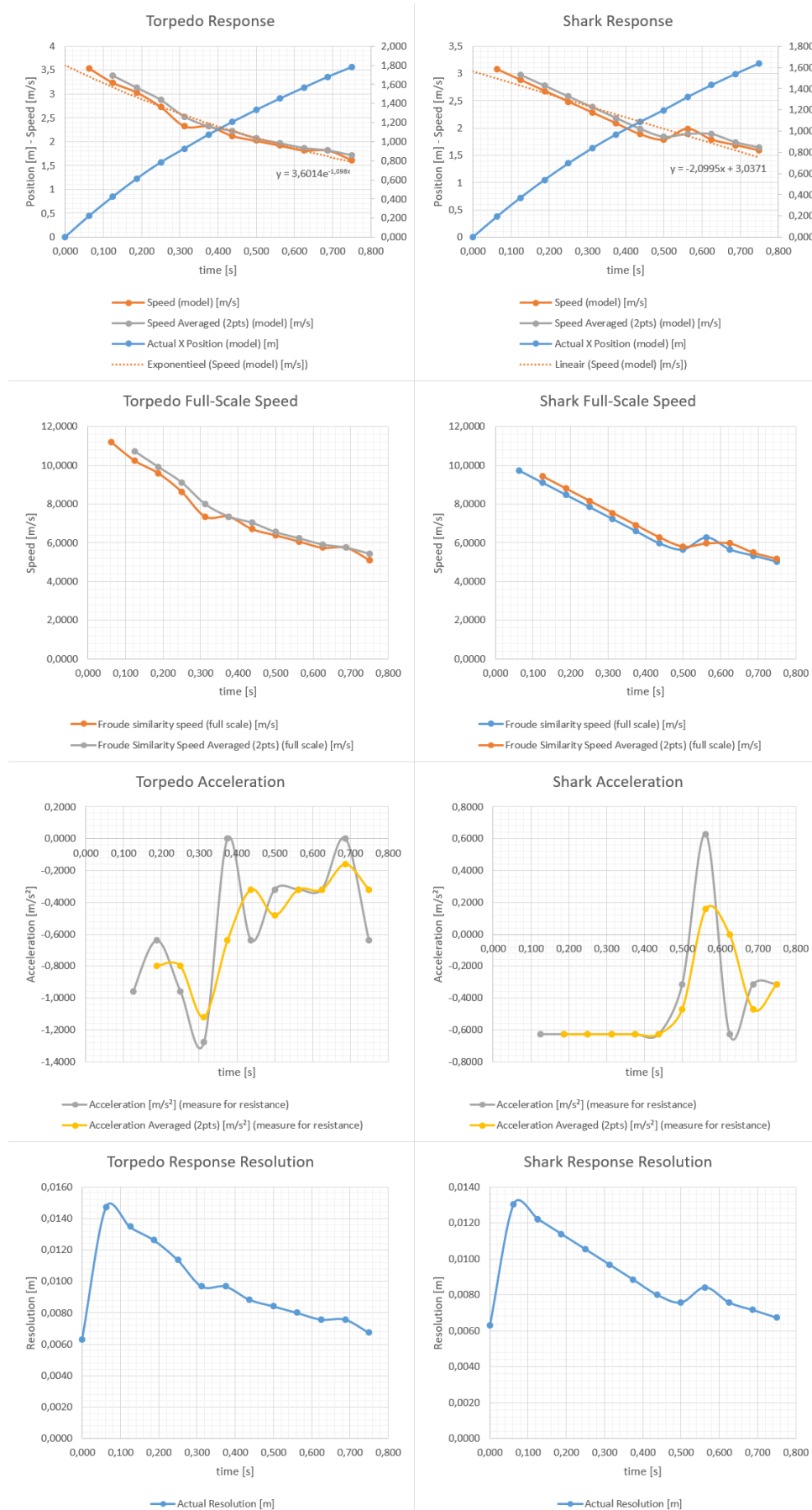


Figure C.53.: Torpedo Mk.II EVO and Shark during tests



Figure C.54.: Falcon livery and coating

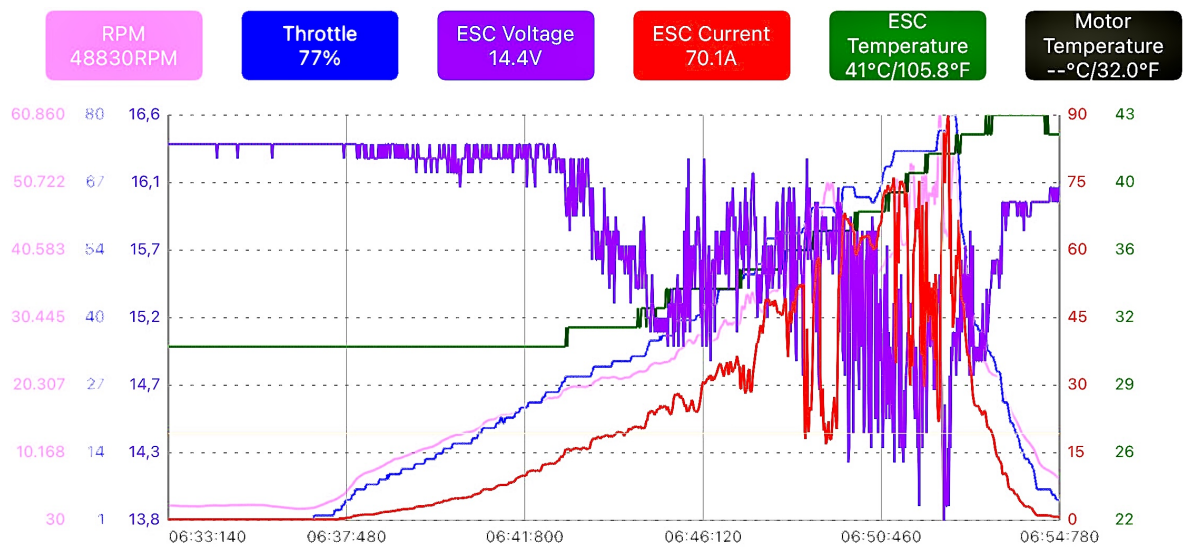


Figure C.55.: Falcon model stationary submerged propulsion ramp measurement

C.4. Analytical Data

See portfolio.

C.5. DNS Scaling Relations

The computational requirements of Direct Numerical Simulation (DNS) increase rapidly with Reynolds number, as all turbulent scales must be fully resolved. Following Pope [116], the key scaling relations are:

$$\frac{L}{\eta} \sim Re^{3/4} \quad (C.28)$$

L = integral (energy-containing) length scale [m]
 η = Kolmogorov (dissipation) scale [m]
 Re = Reynolds number [-]

$$N \sim Re^{9/4} \quad (C.29)$$

N = total number of grid points required for DNS [-]

$$\text{Computational Cost} \sim Re^3 \quad (C.30)$$

These relations indicate that the spatial resolution increases approximately with $Re^{3/4}$, the total grid requirement scales as $Re^{9/4}$, and the total computational cost grows roughly with the cube of the Reynolds number. This makes DNS impractical for engineering flows, where Re typically exceeds 10^6 .

C.6. F913 Louise-Marie



Figure C.56.: F931 Louise-Marie (source: marinebelge.be/f931.html)

C.7. Software

C.7.1. Model-Ship Converter

Not for publication.