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Stability Assessment of a Catamaran Using Sea Trials

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Abstract: Despite the continued rise in popularity of powered catamarans for recreational and commercial applications, there is limited published research on the factors that improve or reduce a powered catamaran's hydrodynamic stability. There is no definitive research on a catamaran's hydrodynamic stability during a turn, and current regulations to control the risks are proving ineffective for modern speeds and power. Research on the hydrodynamic stability of catamarans was conducted using a custom-built vessel and its multi-sensor data logger. Test results confirmed serious concerns for safety regulation and acceptance testing of powered catamarans for hydrodynamic stability. The experiments have produced new insights as to why powered catamarans are at risk of capsizing and created a baseline for future testing. The work reported in this article provides an original characterisation of the multi-factor relationships that impact the instability of a powered planing catamaran. The results provide a starting point for the creation of a predictive model and approach to improve catamaran design and safety.

Keywords: catamaran; hydrodynamic; stability; planing; safety; design

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1. Introduction

Small powered catamarans (6–12 m in length) have increased in popularity over the last 30 years [1–3]. At the same time, increases in outboard development have seen an explosion in high-horsepower outboard motors [1,4]. The evolution of high-speed planing catamarans has developed vessels that can have as many as four 400 kW outboards, with many vessels achieving a Froude number higher than 2 [1,5–8].

While powered catamarans have several advantages, efficient hull form, stability at rest and an increased deck size for any given length, they are prone to capsize [3,6,9–13]. The Australian Maritime Safety Authority (AMSA) controls the registration of all commercial vessels in Australia, including approximately 650 catamarans that measure less than 12 m in length overall (LOA) [12].

AMSA regulates vessel design and safety standards through the National Standard for Commercial Vessels (NSCV) [14]. The NSCV is generally detailed, complex and prescriptive. The structure of the NSCV is applied in sequence by topic, operational area, vessel length, vessel type or exemption. AMSA does not publish accident data, despite AMSA data being the most accurate of any Australian accident data due to the compulsory reporting requirements for commercially registered vessels [14].

The experience of the lead author from a capsizing of a commercially registered power catamaran in 2020 provides a unique insight into the complexity of a powered catamaran's hydrodynamic stability [15,16]. Two expert reports by Renilson [15] and Wise-Mann [16] into this capsize found:

1. The vessel passed the AMSA simplified stability test and then failed the comprehensive stability test.

2. The residual stability left at the top of the righting arm curve (GZ) curve was insufficient to act as a righting arm.

At the 2023 International Maritime Conference, [17] built on the conclusions of the expert reports by Renilson [15] and Wise-Mann [16] by proposing focused vessel experiments. The proposed research was to use a high-throughput test (HTT) design to determine catamaran design factors that may influence hydrodynamic stability [17].

The straight-line experiments presented in this paper are a precursor to the HTT test plan presented by Matthews et al. (2023), developed with the support of Renilson (2023). Using a custom-built, 8.5 m, powered catamaran, a hydrodynamic inclining experiment was conducted to create baseline data for future comparison. The test vessel was run at a range of speeds on a straight course, and masses were shifted systematically on the transverse axis while measuring the induced angle of heel.

The experiment has yielded a new and unexpected insight into the hydrodynamic stability of powered catamarans. The results contrast the hydrodynamic inclining tests of a monohull by Blount [18], where, as the vessel speed increases, the heel decreases. The data from the powered catamaran experiment demonstrate for the condition tested that, as a powered catamaran increases speed, there is an increase in the vessel's heel.

Extensive searches of published research on planing powered catamarans suggest the hydrodynamic behaviour as observed in the experiment is novel [17]. This raises new questions and possibilities that must be considered for future research. The insight provided by this experiment emphasises the need for a change in the way the Australian Standards and the Australian Maritime Safety Authority deal with powered catamaran design and safety regulation. Likely, this process should also support changes to the international regulatory framework.

2. Powered Catamarans

In the context of the paper, a powered catamaran means a twin-hull vessel with symmetrical demihulls, capable of a Froude number greater than 1, less than 12 m long, and powered by outboard motors.

The iterative design process of modern powered catamarans has not changed since their early development [6,8,19,20]. The typical design process starts with a known hull form, and small incremental changes are applied and then tested [20–24]. The limited published research on powered catamaran design and hydrodynamic performance is detailed in an extensive literature survey presented and discussed herein in Section 3. The published research on planing twin-hull vessels that contribute to our understanding is normatively theoretical, and/or uses scaled models, and/or uses asymmetrical hulls (Formula One racing boats) [6,7,10,18,19,25,26]. Powered catamarans with symmetrical hulls are predominantly used in commercial working boat applications [12]. The hydrodynamic stability of a planing powered catamaran cannot be predicted on a first-principles basis yet, as the mathematics to support this process has not been readily developed [6,27–29].

There are as many as ten powered catamaran designers and builders in Australia, with several global brands also imported. The current stock of vessels available to purchase or build are all equipped with large-horsepower outboards and can achieve planing speeds over 30 knots [1,5]. Further, the mathematics of computational fluid dynamics (CFD) and known naval architecture principles are not yet able to fully calculate and predict a powered catamaran's hydrodynamic performance [7,9,29–31].

Catamarans as a vessel type are known for their statical stability at rest [6,19,32]. However, the standards that regulate the design and performance of powered catamarans in Australia fail to adequately capture and predict a powered catamaran's hydrodynamic stability at speed (where velocity is not equal to zero). There are several Australian and international standards a powered catamaran may be designed to, including Australian Standard 1799.2, American Boat and Yacht Council (ABYC) and the International Standards Organization (ISO). None of these standards provide a required performance metric for this "hydrodynamic" stability at speed. In this context, the connotation of hydrodynamic

stability has a parallel with the well-known statical stability and should not be confused with the equally common reference to "dynamic stability" which relates to the area under the righting arm curve (GZ curve).

As a regulatory framework, the NSCV [14] is explicit in its methodology for the calculation of a vessel's stability [33]. However, the NSCV provides no metric for the calculation or prediction of hydrodynamic stability for a planing vessel [33].

The NSCV Section C, Part A, Intact Stability, is both complex and prescriptive. It provides static stability requirements for catamarans in Section 5, Table 11 [33], which is applied commonly to all vessels regardless of the vessel's operating range of Froude number. There is no metric to measure hydrodynamic stability; rather, the NSCV vicariously leverages the scope of [33] Section 6A–C to make clear the obligation to consider other factors. However, this generalised scope is at odds with the prescriptive nature of NSCV compliance. Equally at odds is AMSA's decision not to publish accident data to support awareness for vessel designers and builders. In response to the NSCV's regulatory framework, the commercial industry typically takes the path of least resistance to reduce cost.

3. Literature Summary

A literature review undertaken in 2023 demonstrated the limited research on small dynamically supported power catamarans [17]. Using the University of New South Wales (UNSW) library search tool, the following keywords were used: "Planing Catamaran", "Catamaran Hydrodynamics", "Catamaran Design", "Catamaran Stability", and "Dynamic Stability". The same keywords were used in a supplementary Google Scholar search, with additional research identified and collected via the UNSW Library. The literature has been synthesised into areas of knowledge.

3.1. Dynamically Supported Vessels

The absence of an agreed technical definition of when a vessel is dynamically supported (planing) underlines the challenge of this research [18,34,35]. For example, a semi-displacement hull that rises only fractionally from its static position to Formula One race boats that barely have any hull in the water are both described as dynamically supported [30,36]. The vessels that are the focus of this research spend operational time moving between states of displacement, semi-displacement, and planing. They require stability in all modes, including the transitioning phases [37].

Two demihulls provide more lift than a monohull of an equivalent displacement, as much as 40 per cent [7]. Haase et al. [10] consider the performance of asymmetrical hulls, whereas most small catamarans are designed with symmetrical hulls. Symmetric demihulls provide more dynamic lift than asymmetric ones [38]. The waterline beam/demihull (B/b1) ratio that controls the tunnel size and ratio also impacts dynamic support. Demihulls that are closer together provide greater dynamic lift than those further apart [7,8,19,30].

The beam-to-demihull relationship dictates how the waves generated from one hull impact the other and the vessel's dynamic lift. As the demihulls move further apart, proportionally to the size of the demihull, the impact of the wave on dynamic lift reaches a point whereby greater distance has no further effect [7,25].

The recent popularity of stepped hulls to improve planing in monohulls has not reached the catamaran literature. This change will add additional complexities to predicting catamaran performance [39].

The existing literature can articulate how a powered catamaran will plane, and 3D software like MaxSurf (V23.06.00) and Orca3D (V3.0.15) will provide high-accuracy hydrodynamic predictability on a 3D hull form. However, the research offers no consideration of what happens when the vessel turns while on the plane. The literature demonstrates that multihull configurations are as different from each other as they are from planing and displacement monohulls.

3.2. Length-to-Beam and Beam-to-Demihull Ratios

The waterline length-to-beam (L/B) and beam-to-demihull (B/b1) ratios dictate the size of each demihull and the relationship between the beam and the tunnel. The approach taken to design, the recommended B/b1 ratios, and the predictability of existing vessels' performance are of specific interest to this research. A length/beam ratio from 4:1 to 5:1 is suggested as appropriate for displacement catamarans [6,19,26]. Length-to-beam ratios above 3:1 for vessels below 12 m LOA create an abnormally slender vessel, which is, therefore, less stable. This scaling impact has not been discussed in the research reviewed for this paper.

Wahidin et al. [34], Yun et al. [6] and Dubrovsky [27] all provide guidance for beam-to-demihull ratio and tunnel size. These works need to be considered, not in isolation, but simultaneously with the work on hydrodynamics and stability. The B/b1 ratios directly contribute to the vessel's metacentric height, driving a catamaran's core stability. The metacentric height is positive when the M is higher than the G. The larger the demihull spacing, the larger the righting arm, as can be seen when plotting a GZ curve with all factors being the same except for the vessel's beam [19].

3.3. Stability during Planing

Predicting the hydrodynamic stability of a powered catamaran during a turn presents an opportunity. Blount [18,28] considers the force required at different speeds to create a heeling moment. Blount [18,28] is also emphatic about the centre of gravity and its role in stability, as well as dynamic instability being speed-dependent, with changes as small as 3 knots (Fn = 0.2) changing a vessel's behaviour. The importance of the VCG on hydrodynamic stability has generally not been addressed by the literature [35], though it is called out in the Renilson [15] report. The adverse impact of relative speed (Fn = 1 to Fn = 2) on stability is discussed by Wang et al. [9] and Faltinsen [25].

The beam and demihull design and ratios are critical factors in a catamaran's hydrodynamic stability. The closer the hulls, the more dynamic lift they have; the trade-off is that the closer they are, the greater the tendency for instability [25,26].

Hydrodynamic stability is also affected by wave formation generated by the two demihulls. The wave interaction is not accounted for in any of the dynamic or turning calculations considered by other researchers. The demihull B/b1 relationship influences wave formation, where a larger spacing reduces interference and makes the vessel more stable [19].

Blount and Codega [28] offer a pragmatic test to measure the hydrodynamic stability of a vessel. This in-water test would suit first-of-class trials and other specialist applications. While it does not provide a prediction of performance, the testing of existing vessels and the use of the DOE methodology may provide a predictive data set over time. Honaryar et al. [35] and Pandey and Hasegawa [29] undertake to validate numeric models of the hydrodynamic performance of symmetric powered catamarans and provide a base for further mathematical work that could be built on with the addition of a turn.

The literature on vessel stability uses static measurement and analysis to determine heeling and righting moments [14,40]. Using the vessel's VCG and metacentric height (GM), the force required to move the vessel to a predicted heeling moment can be calculated. This approach allows for plotting a GZ curve and identifying residual stability.

Using static stability to determine hydrodynamic stability, including through a turn and/or in a given sea state, fails to consider the hydrodynamic resistance on each demihull [29,35]. The Blount [18,28] hydrodynamic stability test suggests that the powered catamaran will behave differently when dynamically supported. It is hypothesised that, unlike the static heeling of a powered catamaran, where the counterforce demihull submerses when a heeling moment is applied, when planning, where the force applied is less than the force of the dynamic support, but more than half the mass of the vessel, the applied-force demihull will lift, pivoting on the counterforce demihull while it remains dynamically supported.

4. Test Vessel

An aluminium-powered catamaran with a proven hull form was built and customised to conduct the experiments from the HTT test plan; a summary of the HTT plan is provided in Appendix A. The vessel has several sister ships in commercial operation.

At 8.5 m length overall (LOA), the vessel was built with an adjustable beam to control the length-to-beam ratio (See Table 1). The demihull size (b1) is fixed. The vessel's transom and tunnel were modified to allow for the engines to be mounted both on the demihulls, as is normal (see Figure 1), and closer to the centreline of the vessel, as per Figure 2. The centreline placement has both engines in the catamaran's tunnel, in a similar position to that of an asymmetrical Formula One race boat. The engine manufacturer's recommendation for twin rigging was followed, with the engines' centres 800 mm apart. Therefore, in this configuration, the propellers are each 400 mm from the vessel's centreline. Presented in Figures 1 and 2 are images of the test vessel displaying the two engine positions for HTT testing (sample engines used to display engine location). Photo of the test vessel in Figure 3.

Table 1. Vessel Configuration.

Options	L1	L2	L3
Beam (mm)	3500	3250	3000
Engine placement	Demihull	Centre	-

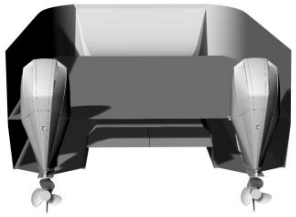


Figure 1. Test vessel with engines set wide.

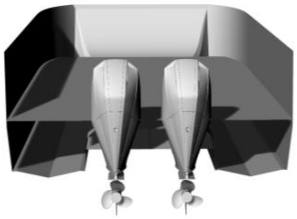


Figure 2. Test vessel with engines set close.



Figure 3. The test vessel.

Two Mercury® 200 HP (147 kW) SeaPro® outboards (Mercury Power International, Sydney, Australia), the port engine counter-rotating, with four-blade stainless steel 19-inch pitch propellers, were fitted such that the propellers turn outward in opposite directions. The engines were paired with Mercury® Digital Throttle and Shift (DTS) controls, and the active trim function was disabled during testing. The vessel was fitted with Multisteer® power-assisted hydraulic steering and outfitted with chandlery from RWB Marine® (RWB Marine, Sydney, Australia).

A Raymarine® AR200 positioning sensor was used to track the vessel's heading, water depth, rate of turn, yaw, pitch and roll (heel), speed over ground and GPS coordinates. This sensor was connected to a Raymarine Axiom Pro® Multifunction Display (MFD) (Raymarine, Sydney, Australia) to record the vessel's movement on five axes via the Raymarine® MDF National Electrical Manufacturers Association (NEMA) protocol data logger, with data capture at two-second intervals (0.5 Hz data rate).

Solid ballast racks were positioned below deck, on deck, and on the gunwale, to contain the lead inclining mass and to vary the VCG, mitigating any free-surface effect on results from using liquid ballast. The gunwale rack is clearly shown in Figure 4. Forty-two 25 kg lead ingots were used as the adjustable ballast. The vessel has a displacement of 3.77 tonnes when all the ballast is loaded. The only variable load during the experiment was due to fuel burn with a maximum variation in the order of only 10–20 kg.

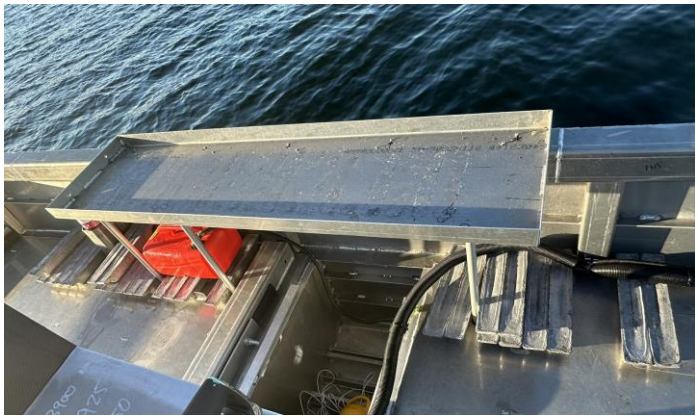


Figure 4. Lead ingots positioned on the deck, moved port and starboard to change the vessel's heeling moment, and tray at gunwale height to change the vessel's vertical centre of gravity.

5. Baseline for High-Throughput Testing

To create a baseline for the test vessels' hydrodynamic values, straight-line experiments were designed using the design of experiment principles [41,42]. These straight-line experiments would provide a baseline and context for the higher-order factor testing with the HTT design planned by Matthews et al. (2023) in which data were to be collected while the vessel was in a turn [43]. The results from the straight-line tests could be compared to the HTT testing conducted during a turn, allowing for the isolation and contrast of factors.

The planed experiments are listed in Table 2 for each of the vessel's six beam and engine configurations as outlined in Table 1. The vessel's maximum speed was achieved by placing the throttle in the wide-open throttle (WOT) position and trimming the engine position to achieve maximum speed.

Table 2. Straight-line baseline testing.

Factor	Levels	L1	L2	L3	L4
(A) Speed (rpm)	1	WOT	-	-	-
(B) Thrust (%)	3	100	75	50	-
(C) Hydrodynamic stability (degree)	4	0	2500	3500	4500

Test (A)—Speed

Placing 525 kg of weight on each side of the vessel's deck, accelerate to wide-open throttle, being the vessel's maximum trimmed speed, to measure the vessel's maximum speed and the vessel's roll (heel) deviation.

Test (B)—Thrust

Placing 525 kg of weight on each side of the vessel's deck, accelerate to 4500 rpm on both the port and starboard engines. Once on the plane, reduce the port engine to 75 per cent or 3375 rpm. This process was repeated with the port engine reduced to 50 per cent or 2250 rpm.

Test (C)—Dynamic Stability

Re-creation of Blount's hydrodynamic stability test [18]. By incrementally placing more weight on the starboard side of the vessel than the port side to create a heeling moment (see Table 3), the vessel would accelerate in a straight line, to a manageable speed, ideally WOT. This process was repeated with the range of weight differences listed in Table 3, measuring any change in heel angle at each of the different speeds.

Table 3. Test C—hydrodynamic stability.

PORT—STBD Kg	Levels	L1	L2	L3	L4
525–525	4	0	2500	3500	4500
475–575	4	0	2500	3500	4500
400–650	4	0	2500	3500	4500
350–700	4	0	2500	3500	4500
200–850	4	0	2500	3500	4500
100–950	4	0	2500	3500	4500

The test track used for baseline testing was a straight line where the tested protocol was held for a minimum of 100 m. With three repetitions of each configuration to be tested, a total of 18 runs were manageable and completed.

6. Test Methodology

The test site was Toronto Bay on Lake Macquarie, NSW Australia. It is a saltwater lake, with an average depth of 6 to 8 meters. Testing was conducted over several days. The conditions for each day were considered perfectly calm with the water surface displaying less than 0.1 m of movement and the wind below 10 knots.

The vessel was configured while at rest. The Mercury auto-trim feature was turned off, and the engine trim was set at a constant 'trimmed-in' position for all testing. The three repetitions of each experiment were recorded in one data log file, with each pass completed consecutively with only the time to turn the vessel between each pass. Data from each test run (three passes) were collected on a USB data stick.

Raw data from the NEMA data log were imported into an Excel macro where the data logs were converted into time-stamped instances. The run data were then homogenised, with any instance that had capture errors removed. None of the instances removed contained a unique minimum or maximum data event.

7. Vessel Configuration for Testing

The vessel was configured with the beam at 3500 mm (maximum setting) and the engines were placed on the demihulls. The moveable mass of 1050 kg comprised 42 lead ingots which were placed on the deck. Baseline hydrodynamic inclining testing was only conducted in this configuration. Tests were conducted with only the master of the vessel onboard to ensure a consistent and controlled mass distribution.

8. Results

Test A, Speed. The speed test at wide-open throttle provided the vessel's top speed of 42 knots. Shown in Table 4 are the speeds relative to the average rpm of the two

engines, averaged over the three repetitions. The raw data demonstrate the challenges with interpreting data and small samples, where small amounts of hydrodynamic resistance can impact results, particularly at slow speeds.

Table 4. Speed in knots.

rpm	Knots	Froude
4900	39	2.37
4500	35	2.13
4000	30	1.82
3800	25	1.52
3700	20	1.21
3300	19	1.15
3200	17	1.03
2700	15	0.91
2100	12	0.73
1900	10	0.57
1380	8	0.61
980	6	0.48

Test B, Thrust. This test demonstrated the vessel could be held in a straight line while the engines produced different levels of thrust. The power-assisted steering made this task easy to handle. The ability to precisely control the rpm of each engine proved very difficult, though. When the throttle was pulled back on one engine, the propeller of that engine was dragged through the water and rotated at a coefficient relative to the vessel's adjusted speed. The engine's safety systems allowed the engine to free-spin at this self-determined rpm. The adjusted speeds achieved are displayed in Table 5.

Table 5. Speed relative to depowering one engine.

PORT—STBD rpm	Knots	PORT %	STBD %
3900–3900	30	100	100
3100–4600	28	75	100
2000–4000	17	50	100

The port engine was selected to be depowered, as the helm was right-handed and the port engine control was closest to the master of the vessel.

Test C, Hydrodynamic Stability. The data presented in Table 6 are the average of the repetitions. During testing, the vessel was damaged. Data logs from runs two and three were lost. The data from all 15 repetitions set out in Table 6 follow the same pattern as shown in Figure 3 and are available in the extended data. The lost data were assumed to be consistent with all other data.

Table 6. Hydrodynamic stability.

PORT—STBD Kg	Knots	Rest	Max	Mean	%>	STDEV	Rep
525–525	38.5	0	0.2	0.4	00	0.36	3
475–575	-	-	-	-	-	-	3
400–650	-	-	-	-	-	-	3
350–700	35.8	0.82	2.96	1.99	260	0.47	2
200–850	34.8	1.38	3.5	1.8	153	0.47	3
100–950	27.2	2.65	4.1	3.16	54	0.36	4

The data demonstrate a high level of consistency, with the standard deviation and range of the data sets of each repetition being consistent, given the human operation of the vessel. The vessel's roll angle increased as the vessel transitioned from a Froude of below 1 to a Froude of above 1, as shown in Figure 5.

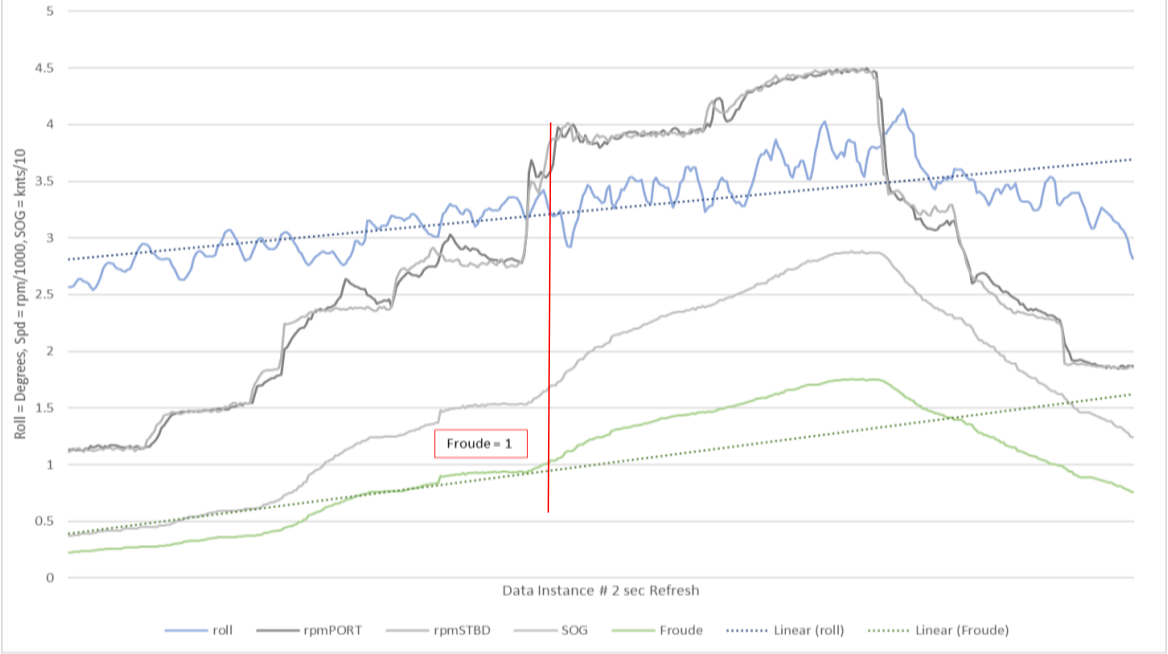


Figure 5. The increase in roll correlating to the increase in speed over ground (SOG and the corresponding Froude number) for the 100 kg port and 950 kg starboard configuration in Repetition 2. The red line shows Froude = 1.

Shown in Figure 6 is the righting moment increasing as the engine's rpm increases. Observe the consistent increase in heeling moment across each moment (kg.m) across each engine rpm range.

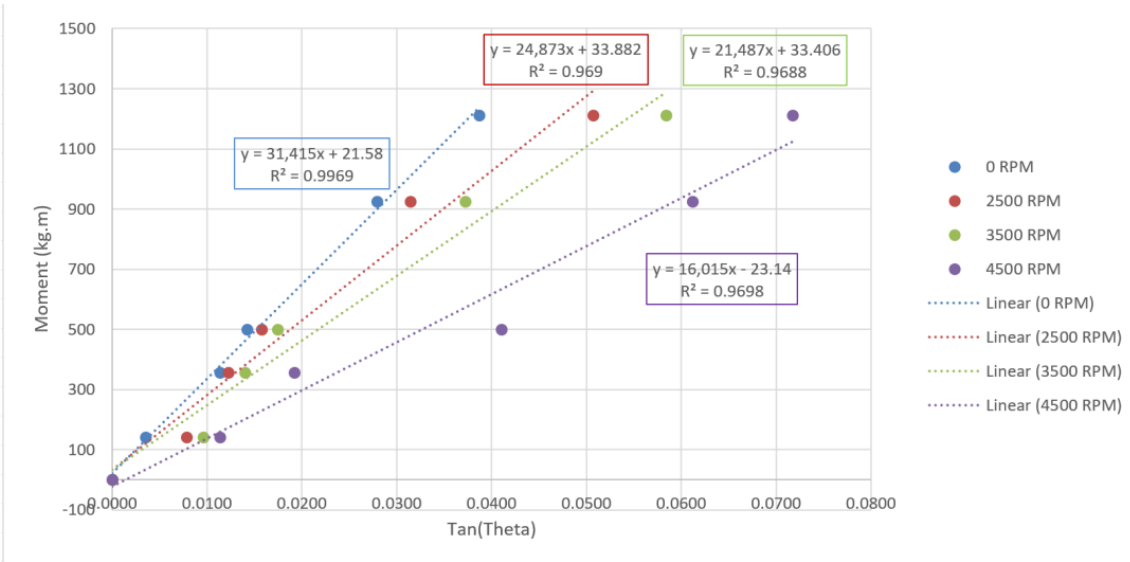


Figure 6. The different heeling movements produced a constant increase in righting moments correlating to the increase in rpm and Froude.

Shown in Figure 7 is the loss of the vessel's metacentric height (GM) as the Froude increases for the configuration tested (maximum beam). This result is consistent with the other observations and shows the decline in the vessel's stability as a correlation to the vessel's speed.

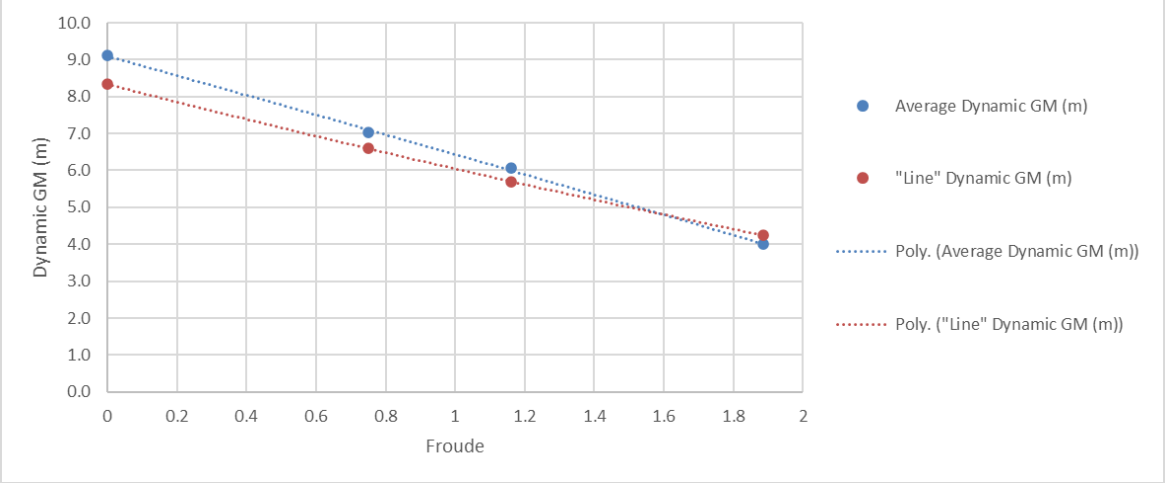


Figure 7. Shows the loss of metacentric height as the rpm and Froude increase.

9. Discussion

The stability of displacement vessels is understood through the calculation of a heeling and righting moment plotted on a righting arm (GZ) curve [40,44,45]. This method has proven to be a reliable predictor of hydrodynamic stability for displacement vessels, as the waterline relative to the vessel's displacement is largely the same when the vessel is static or dynamic [18,23,40,44]. When the same approach is applied to a planing mono-hull, the result again provides a reliable predictor, not because the righting moment accurately predicts the vessel's hydrodynamic stability, but rather because the hydrodynamic support, as described by Blount [18,28], creates its own righting moment, separately, but in parallel to the role of gravity and buoyancy as seen in a GZ curve. The quantum of the hydrodynamic lift, correlated to the vessel's speed, also works to right the vessel.

Calculating a planing vessel's hydrodynamic stability via a GZ curve at a single point in time, while the vessel is travelling with a Froude greater than 1, would yield extremely large values due to the very small denominator of buoyancy (B) and produce results that have no practical application. Therefore, we cannot provide a reliable prediction of hydrodynamic stability. There is not currently a sensor or practical method that would allow for the accurate measurement of B or the metacentric height (M) as it correlates to a specific Froude and corresponding heeling moment (when planing). Specifically, it is currently impossible to determine the accurate displacement of each demihull at any given time. This presents an opportunity for future investigation.

As a hydrodynamically supported vessel transitions from displacement to planing, the location of buoyancy (B) used to calculate the righting arm to create a GZ curve also moves. The vessel's displacement does not change; however, both the volume of water displaced and where the vessel sits in the water do change. If the vessel's displacement is now set as the volume of water displaced when on the plane and not the vessel's static displacement, with buoyancy (B) at or near zero (Figures 8 and 9), the use of a GZ curve as a predictor of hydrodynamic stability becomes invalid. The hydrodynamic lift force must also be accommodated.

Figure 10 shows applying this principle to a powered catamaran travelling at a Froude greater than 1 with a heel of 5 degrees. In this dynamic position, one demihull has a displacement like its static displacement, the other almost none. The traditional method of predicting a planing powered catamaran's hydrodynamic stability through the use of a static righting arm and righting moment on a GZ curve will not provide an accurate predictor of this vessel's hydrodynamic stability.

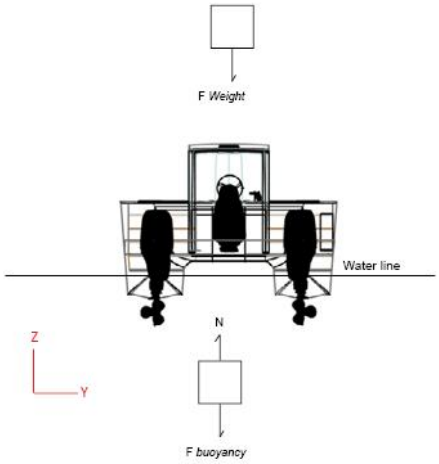


Figure 8. At rest, the vessel's displacement is equal to the water displaced. Buoyancy (B) can be calculated and used in a GZ calculation.

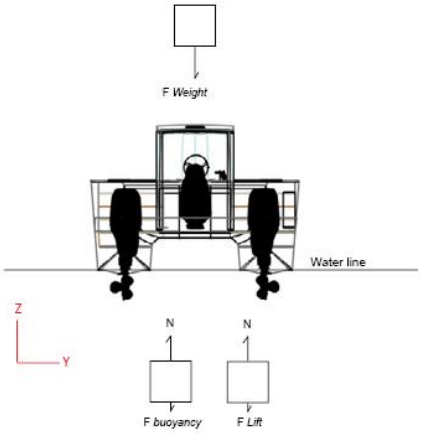


Figure 9. Dynamically supported (planing), the water displaced is not equal to the vessel's displacement. Buoyancy (B) is plotted at or near zero.

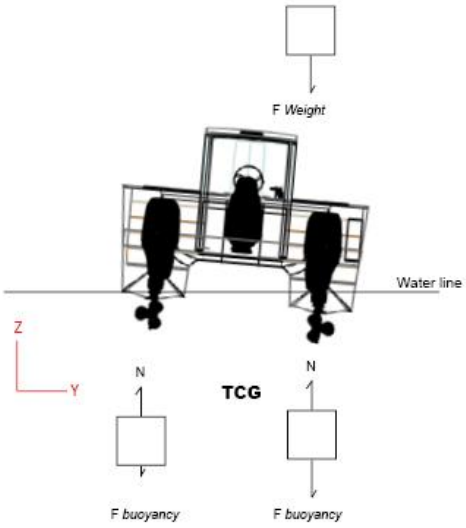


Figure 10. A snapshot of the powered catamaran travelling at 30 knots with a Froude of 1.82 with a heel of 5 degrees.

As observed, increasing speed increases the heel of a symmetrical planing powered catamaran when asymmetrically loaded. The lead author recreated and confirmed Blount's [18] hydrodynamic stability observations in a monohull in 2023 (Matthews 2023).

It was hypothesised that the powered catamaran would behave like the monohull, in that the applied-force (more heavily loaded) demihull would be hydrodynamically supported, with a positive correlation between increased speed and increased support. The results of the experiments conducted disprove that hypothesis and provide a novel understanding of the hydrodynamic behaviour of powered catamarans.

The advantages of a catamaran's sleek demihulls make it efficient in a straight line and stable at rest [6,19,27,32], but equally disadvantage the vessel's hydrodynamic stability. The catamaran's tunnel reduces the breadth of the vessel's planing surface area. The results of these experiments raise new questions about demihull design in catamarans, reinforcing the need for design clarity as to the effect a catamaran's length, beam and demihull ratio has on hydrodynamic stability [6,8–10,21,27].

The test results show that, as the powered catamaran speed increases, the heel also increases. A new question arises, as follows: is the phenomenon caused by the applied-force demihull sinking, the counterforce demihull rising, or both? Placing an altimeter on the vessel's NEMA data logger on each gunwale of the vessel would provide an insight into the behaviour of each demihull in relation to the vessel's overall position in the water. This sensor would experience less noise compared to nominal hull sensors that are impacted by spray and wash.

Acknowledging that the heel increases with speed requires researchers to consider in more depth what is happening to a powered catamaran during a turn. As force (inertia or other) is applied to the vessel's outer demihull during a turn, the heel increases, correlated to the speed of the vessel. Without further research, it appears a lack of hydrodynamic stability is an inherent design limitation of powered planing catamarans.

The HTT plan considered by the authors in Matthews et al. (2023) looked to identify which factors contributed to a powered catamaran's instability in turning. The test design will now need to be reviewed in light of these results. Factors including the centre of gravity and beam may no longer be relevant, as their impact on the vessel's heeling moment is known and understood. Some factors in the HTT design may not be a contributing factor; however, they may be mitigating factors.

These findings have wide-ranging implications for the safety and design of powered planing catamarans already in widespread use, as they illustrate that each class of vessel can have unique and uncommon characteristics. General principles that apply to a displacement monohull are not sufficient for catamaran hulls without first validating the conventional test methods to produce predictable results. The Civil Aviation Safety Regulations, regulated in Australia by the Civil Aviation Safety Authority, contrast with the NSCV by providing individual regulations for each class of aircraft [46]. These results provide merit for vessel regulators to move to class-specific regulations for vessels.

The predictability of a vessel's hydrodynamic stability must be a primary concern for national maritime regulators [47]. With a significant number of powered catamarans in commercial registration, many are involved in commercial fishing and or people transport where the likelihood that these vessels have cargo or people that can achieve unknown unsafe conditions should now be considered likely.

Similarly, powered catamarans built in Australia or imported into Australia for the recreational boating market should meet one of the Australian Standards or international equivalents [48]. Compliance with a standard is currently not regulated as a condition of sale, unlike for boat trailers [49]. The Australian and international standards that apply to powered catamarans do not provide a metric for measuring or controlling hydrodynamic stability [50,51].

AMSA's use of Exemption 02 (EX02) and Exemption 40 (EX40) registration allows the commercial registration and use of a powered catamaran without understanding the vessel's design limitations [14]. While EX02 registration restricts use to calm waters, this research and the lead author's own experience unquestionably demonstrate that the use of the EX02 registration places people and vessels at an unacceptable risk of capsizing.

The NSCV identifies a heeling moment of 14 degrees as a fail point for stability [14]. The results of our experiments demonstrate that these standards must be reviewed in light of these findings, noting that there are still insufficient data to arrive at a predictive equation to express the vessel's likely behaviour in any specific condition.

The data collected from these experiments demonstrate a novel observation of powered catamarans. There is a correlation between the vessel's speed and a declining ability to resist a heeling moment.

10. Conclusions

Contrary to the behaviour of a monohull, the impact of a heeling moment of a planing powered catamaran will increase (negatively), with correlation to the value of the force applied to create the heeling moment and the speed at which the vessel is travelling. This novel observation is an important development in the understanding of a powered catamaran's hydrodynamic stability.

The results of this experiment demonstrate that the static stability criteria of the NSCV should not be used as a predictor for hydrodynamic stability in a powered catamaran. Furthermore, there is now a compelling argument for all maritime regulators and all standards agencies to reconsider the current generic approach to the regulation of catamaran design. Catamarans should be provided with their own specific requirements and regulations to improve the predictability of their safety, with the added advantage of removing the existing complexity when reading and interpreting generic standards.

Maritime regulators should sponsor research into catamaran hydrodynamic testing to further understand the factors that influence the hydrodynamic stability of a powered catamaran, particularly in a turn. Placing an altimeter on each demihull configured to be read by a NEMA data logger may provide insight into the behaviour of each demihull relative to the vessel. The use of an altimeter on other planing vessels may provide new insights relative to a planing vessel's hydrodynamic displacement, yaw, pitch and roll as vessels cross the hydrodynamic to aerodynamic threshold.

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Appendix A. High-Throughput Test Plan—Proposed by Matthews et al., 2023 [17]

A high-throughput test plan was developed with Quantum XL® (Version 16) software ensuring that the experiment remains statistically significant [19,52]. Controlled variables and measured output are articulated through a DOE diagram [31,42]; see Diagram 3. This diagram is developed from the cause-and-effect diagram (see Diagram 1).

Using Quantum XL by SigmaZone®, an N-level factorial design was selected with five factors. Four factors were chosen to be tested at three levels and one categorical factor

with two levels [53,54]. Two outputs have been selected: the heeling moment measured in degrees from zero and the VCG measured in mm from the resting position. Three replications of each run were determined to be adequate [53]. Therefore, this test design produced an orthogonal test plan with a variance inflation factor (VIF) of one (i.e., perfect orthogonality), requiring 162 runs with three replications—the total number of test points in this test plan is 486. The DOE factors, levels and values are defined in Table A1.

Table A1. Experiment factors and levels.

Factor	Levels	L1	L2	L3
Beam (m)	3	3	3.25	3.5
Deadweight (kg)	3	0	500	1000
VCG (%)	3	0	+10	+20
Thrust (%)	3	100	75	50
Engine Position	2	Demihull	Centre	1.8

The test plan was then optimised using the Quantum XL® D-Optimal tool to reduce the required test runs while maintaining orthogonality (i.e., independence of factor and interaction) by trying several test designs. A test design with two-way interactions and quadratics produced a test design with 26 nearly orthogonal runs, with the VIF for interactions below 1.02 and quadratics below 1.21.

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