

Influence of Key Design Parameters on the Stability of Suspension Ships

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Abstract

This study presents a stability analysis of suspension ships, with a specific focus on assessing how the key design parameters influence the ship's stability. These parameters include the total mass, mass ratio, beam of ship, spring stiffness, allowable travel, and ride height of the suspension system. The stability analysis is performed based on a hydrostatic force model and a static analysis model including the suspension deflection induced by ship inclination and load variations on the cabin. The present numerical method is verified by a sensitivity study of hull surface discretization and comparisons of the initial and intact stability obtained using Orca3D. Numerical results show that the coupling effect of the heeling and trimming occurs when the total mass of the suspension ship exceeds a critical value. An increasing mass ratio significantly reduces the righting arm (GZ) and increases trim, while spring stiffness, allowable travel, and ride height have relatively less influence on the initial and intact stability of the suspension ship. The proper total mass of the investigated ship is suggested to be 95 to 125 tonnes based on the International Maritime Organization (IMO) intact stability resolution and total mass threshold defined in this study. A dynamic stability analysis based on IMO weather criterion indicates that the mass ratio should not exceed 0.75. In damaged conditions, flooding can intensify heel-trim coupling due to the added weight of floodwater, further deteriorating the ship's damage stability. This study provides new insights into the stability characteristics of suspension ships, highlighting the effects of suspension parameters that differentiate suspension ships from conventional ships.

Keywords Marine suspension; Stability analysis; Ship design; Crew transfer vessel; Catamaran

1 Introduction

Small ships, owing to their compact size, are inherently more prone to the influence of waves, resulting in heightened motions such as rolling, pitching, and heaving in rough sea conditions. These heightened motions can cause discomfort and motion sickness among passengers and crew members

on board. Current strategies for mitigating motion primarily concentrate on addressing rolling motion. These include various techniques such as bilge keels, fin stabilizers, rudder roll stabilization systems, anti-rolling tanks, gyrostabilizers, and moving weights. Jiang et al. (2020) conducted a comprehensive study on the configuration of the bilge keel, revealing that the height, length, and installation angle of the bilge keel play significant roles in the roll damping and roll response amplitude. Liang et al. (2018, 2019, 2022) have conducted a series of numerical studies on the design of fin stabilizers and the controller during ship turns and zig-zag motion. Liu and Liu (2024) developed a finite-time disturbance observer for roll stabilization. Lin et al. (2022) proposed an anti-rolling controller using a deep deterministic policy gradient algorithm and the Magnus effect. Sun et al. (2023) proposed a full-speed range lift calculation method for the Magnus rotating roll stabilizer. Chen and Yang (2018) experimentally investigated and verified the effectiveness of a hybrid tuned mass damper and tuned liquid damper on motion reduction. For motion reduction in heaving and pitching, the effectiveness of T-foil has been experimentally studied by Jiang et al. (2022) using a hybrid control strategy on a trimaran in head waves. Furthermore, ride control systems were studied to minimize slamming of high-speed craft to improve ride comfort

Article Highlights

- A stability analysis framework incorporating suspension deflection and six design parameters is developed for suspension ships.
- Excessive total mass or high mass ratio causes heel-trim coupling and reduced GZ, compromising intact and damage stability.
- Spring stiffness and beam notably affect intact stability, and neglecting suspension deflection leads to overestimation of GZ values.

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(AlaviMehr et al., 2019; Alsalah et al., 2024).

Besides these technologies, suspension ships have emerged as a novel ship type, attracting interest from researchers and engineers. This concept is inspired by the proven effectiveness of suspension technology in mitigating motion reduction and enhancing ride comfort in land vehicles. A suspension system typically consists of springs, dampers, and linkages, which perform as shock absorbers to isolate the disturbance from road surfaces, consequently, improving ride comfort and road-holding performance. When applied to small vessels, suspension technology enables relative motion between the suspended and unsuspended bodies, facilitated by the extension and retraction of the suspension linkages. This separation offers new opportunities to control the ship motions using a variety of strategies.

Since 2007, several companies and research groups have developed different catamarans equipped with distinct suspension designs as shown in Figure 1. Their specifications are summarized in Table 1. A common aspect in these designs is that the cabins of those ships were fully suspended upon the two hulls. The suspended cabin is called the sprung body, whereas the two hulls are called unsprung body. Due to the movement of the suspension mounted between the cabin and the hulls, the cabin's motion can be decoupled from that of the hulls. Sea trials with full scale ships and laboratory towing tank experiments with proto-

types have demonstrated significant motion reduction on the suspended cabins.

These suspension ships utilize different suspension designs and control methods to achieve motion mitigation. Wave Adaptive Modular Vessels (WAM-V) in Figures 1(a) and 1(b) incorporate springs, transverse hinges, and ball joints to isolate the impact caused by waves. Meanwhile, inflatable hulls are used to reduce the high frequency vibration by adjusting the air pressure inside (Conti and Gundersen, 2011; Ocean Power Technologies, 2023). The suspension of the Wave Harmonizer in Figures 1(c) and 1(d) is comprised of Watt's linkage, pantograph, and helical springs. Motion mitigation of the cabin (heave, pitch, and roll) utilizing the classical Proportional-Integral-Derivative control was proposed and experimentally studied (Han et al., 2019a, 2023a). By using generators to convert the kinetic energy of the oscillating cabin into electrical energy, Wave Harmonizers can also perform as wave energy converters (Han et al., 2019b, 2022). The suspension of Martini 1.5 in Figure 1(e) consists of crossed-bar linkages located at the front and aft of the ship. The real-time space position of the cabin was tracked to allow the active air suspension systems to compensate the wave motions, keeping the cabin stable (Servo-yachts, 2023). Nauti-Craft 2play in Figure 1(f) has front and aft hinge linkages mounted between the cabin and the hulls. Motion reduction of cabin is achieved by a



Figure 1 Examples of different suspension ships

Table 1 Specifications of the existing suspension ships

Name (Country)	Dimension Length $\times$ Beam (m)	Separable motion	Actuator	Control system	Application
Wave-Adaptive Modular Vessel (USA)	Protceus: 30 $\times$ 15	Pitch	–	Passive	Survey vessel
Wave Harmonizer (Japan)	Type 7: 3.3 $\times$ 1.6	Heave, roll, pitch	Electric	Active	Leisure boat,
Martini series by Servo Yachts (USA)	Martini 1.5: 10.7 $\times$ 5.8	Heave, pitch	Pneumatic	Active	work boat,
Nauti-Craft Marine suspension technologies (Australia)	2Play: 8.5 $\times$ Not available	Heave, pitch	Hydraulic	Passive, active	search/rescue boat, and ferries

passive reactive interlinked hydraulic system. Results of the sea trials showed that the vertical acceleration of the cabin can be reduced by 80% compared to a similar size of a normal catamaran (Nauti-Craft, 2023).

The concept of suspension ships presents opportunities for ship motion reduction and ride comfort improvement. Advances in automotive engineering can provide a strong theoretical foundation for suspension structure optimization (Mollajafari and Nejad, 2025) and control system design (Mohtavipour et al., 2023). In the marine industry, suspension ships have the potential applications as security and rescue ships, leisure boats, passenger ferries, and working boats leveraging their motion reduction capabilities. Additionally, by integrating energy harvesting with motion stabilization, suspension ships could serve as survey vessels, benefiting from extended power supply durations and reduced oscillations.

Meanwhile, the development of suspension ships may also raise challenges to regulations for the existing classifications and registry. For conventional ships without suspension system, the stability requirements follow the rules published by the International Maritime Organization, with intact stability addressed in Resolution MSC. 267(85) and damage stability in Resolution MSC. 429(98). The evolution of international intact stability criteria has been historically demonstrated by Francescutto (2016). Some stability research primarily focuses on various ship types under different operational conditions. Mauro et al. (2022) applied Randomised Quasi-Monte Carlo methods to investigate the static and dynamic studies of two passenger ships in flooding conditions. Qi et al. (2025) examined the effect of ice accumulation on the static and dynamic stability of a polar ship. Some studies have proposed practical methods for stability assessment. Sun et al. (2021) utilized three-dimensional design data of ships with an integration method to evaluate the impact of liquid load on damage stability in flooding conditions, while Bulian et al. (2019) developed a simplified probabilistic method for collision damage below the waterline. For suspension ships, Han et al. (2023b) conducted a study on how the suspension design parameters influence the initial transverse stability of a monohull suspension ship. The findings indicated that the inclusion of a suspension system can result in a large GM, which makes it difficult for monohull suspension ships to meet the initial stability requirements. While this issue does not appear to affect catamaran configurations, but further and comprehensive investigation is necessary.

In the present study, the focus is given to investigate the influence of the suspension design parameters on the stability of suspension catamarans and gaining knowledge on decision-making policy for the design specifications of suspension ships. The initial, intact and damage stability with free trim are analysed using a hydrodynamic force model and a static analysis model based on Wen et al. (2023,

2024) and Zhang et al (2025), with additional consideration of suspension deflection caused by ship inclination and load variations on the cabin. The influence of six design parameters, including the total mass, mass ratio, beam of the ship, spring stiffness, allowable travel, and ride height of the suspension system, on the stability of a suspension ship are investigated. The present study is structured as follows. The design parameters of interest and the numerical method adopted are introduced in Section 2. In Section 3, a parametric study is conducted on a suspension ship with a length of 24 m. The metacentric height (GM), righting arm (GZ) and free trim of the suspension ship are calculated and analysed. A sensitivity study of the hull surface discretization is performed, and the results are compared with those of using Orca3D to prove the method's accuracy. Then the GM, GZ and trim are numerically calculated at various design parameter values to investigate their influences on the ship's stability. Following this, dynamic stability and damage stability analysis are conducted. The main findings of this study are summarized in Section 4. This research presents an engineering approach for assessing the stability of suspension ships, offering fundamental insights that can support the design and development of marine suspension structures.

2 Ship design and methodology

2.1 Design of a suspension ship

2.1.1 Initial specifications

A conceptual prototype of the investigated suspension ship is demonstrated in Figure 2. The ship is a crew transfer vessel designed for offshore wind turbine access purposes. A certain working area is guaranteed in the front part of the cabin's deck. The cabin and twin-hull are connected by four suspension units which are symbolically shown as springs and dampers. Suspension linkages employed in suspension ships are designed to mitigate relative surge, sway, and yaw motions between the sprung and unsprung bodies, meanwhile allowing the heave, pitch, and roll motions of the cabin and the twin-hull. As illustrated in Figure 1, although suspension linkages exhibit significant diversity among different designs, they serve similar functions in motion separation.

Typically, a floating rigid body possesses six degrees of freedom. When two rigid bodies are joined, the total degrees of freedom can increase up to twelve. Due to the constraints imposed by the suspension linkages, the total degrees of freedom of a suspension ship can be reduced to nine if the two hulls are rigidly connected (Figure 1(c)), and to ten if they remain independent (Figure 1(f)). These degrees of freedom are important for further studies on the motion responses of suspension ships in waves.

The dimensions of the twin-hull and the description of

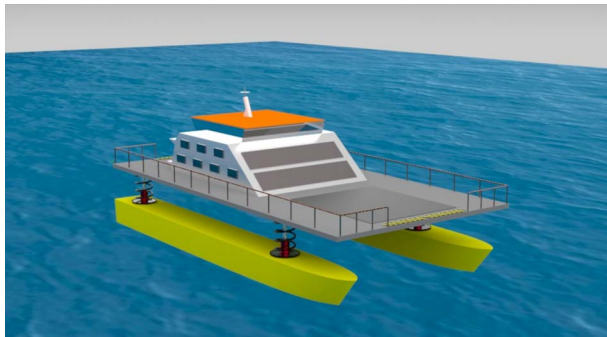


Figure 2 Conceptual prototype of a suspension ship in length 24 m designed for offshore wind turbine access purposes

the global coordinate system OXYZ are shown in Figure 3. The origin of the global coordinate OXYZ is located at the centre along the longitudinal line on the baseline of the twin-hull. The X-axis points from the stern to the bow, while the Z-axis points vertically upward. The Y-axis is oriented according to the right-hand rule. The main specifications of the investigated suspension ship are given in Table 2. The centre of gravity is defined within the global coordi-

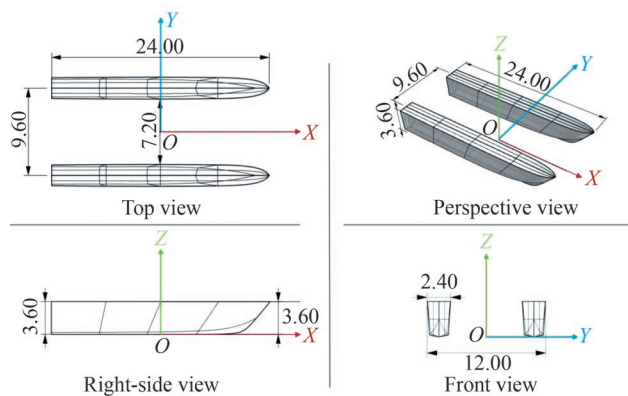


Figure 3 Dimensions of the planing twin-hull and description of the global coordinate system OXYZ. Dimensions are in meters

nate system OXYZ, while other parameters are illustrated with a simplified structure sketch in Figure 4.

As a crew transfer vessel, the investigated suspension

Table 2 Main specifications of the investigated suspension ship

Parameter Description	Symbol	Values	
		Lightship	Full load
Sprung mass	M_s	50 tonnes	100 tonnes
Unsprung mass	M_u	25 tonnes	50 tonnes
Draught	d	1.13 m	2.00 m
Length overall	LoA	24 m	
Beam overall	BoA	12 m	
Beam of hull	b	2.4 m	
Distance between hulls' centrelines	B_{CH}	9.6 m	
Transverse clearance of the two hulls	B_{HH}	7.2 m	
Depth overall of the two hulls	DoA	3.6 m	
Centre of gravity of the ship	CoG	(-2.70 m, 0 m, 6.14 m)	
Centre of gravity of the twin-hull	CoG _u	(-1.65 m, 0 m, 2.00 m)	
Vertical height from STL to CoG _s	H_{GS}	1.5 m	
Vertical height from baseline to CoG _u	H_{GU}	2.0 m	
Ride height of suspension	H_{RD}	$\geq \text{DoA}$	
Allowable travel of suspension	H_{AT}	≥ 1.5 m	

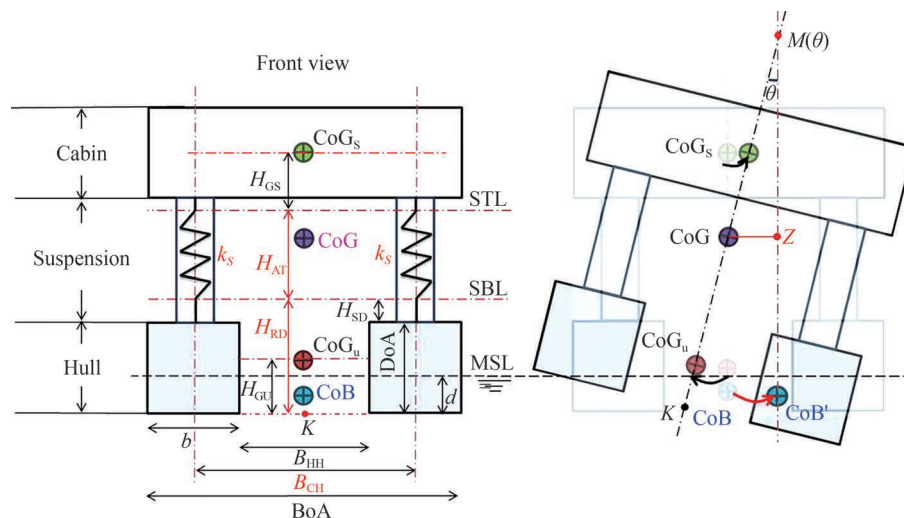


Figure 4 Structure sketch of the investigated suspension ship and the main design parameters

ship is designed with a load capacity of 30 tonnes, including turbine components, equipment, and fuel. It is capable of reaching speeds of 30 knots. The maximum wave height for transient and transfer phases is set at 2.5 m considering the feature of motion compensation. The vessel can carry up to 24 passengers, with each passenger accounting for 0.5 tonnes of payload (including body weight, gear, fresh water, and life/safety provisions), totalling 12 tonnes. For this study, a broad mass variation range of 75 tonnes has been considered to assess the allowable mass margin for the given hull form and to evaluate the impact of design parameters on the ship's stability under heavy load conditions. It is important to note that the full load used in this study represents an initial design assumption rather than a recommended operational full load. The allowable load margin and the full load capacity vary depending on the hull forms, necessitating case-specific investigations. Optimizations of hull forms and operational loads are not covered in this study.

The mass of the whole ship is distributed into two parts: the sprung and unsprung mass. The sprung mass denotes the mass of the suspended cabin while the unsprung mass denotes the mass set below the suspension system. The suspension linkages symbolized as springs and dampers are simplified as massless. The lightship condition represents the ship's lightest weight, while the full load condition represents the maximum loading weight that includes the weight of the ship, crew, fuel, and devices required for off-shore wind turbine maintenance.

2.1.2 Aspects and assumptions

Suspension ships differ from conventional ships in several key aspects. Their structure consists of three main components: the suspended cabin, the suspension system, and the twin-hull. These structural differences introduce the following unique aspects:

1) Variations in the loaded weight on the cabin's deck affect both the twin-hull's draught and the deformation of the suspension system.

2) Shifting loads on the cabin's deck induces inclination in both the suspended cabin and the twin-hull, but to different extents.

3) Encountering waves cause the deformation of the suspension system, inducing motions in both the suspended cabin and the twin-hull, but to different extents.

To simplify the analysis, the following assumptions regarding mass distribution are made:

1) The two hulls are rigidly connected as a twin-hull, whose centre of gravity is denoted as CoG_U .

2) The H_{GS} and H_{GU} remain constant regardless of changes in the ship's total mass. H_{GS} represents the vertical distance between the centre of gravity of the sprung mass (CoG_S) and the suspension top line (STL), while H_{GU} represents the vertical distance between the centre of gravity of the unsprung mass (CoG_U) and the twin-hull baseline.

3) The CoG_U remains constant regardless of changes in the unsprung mass and is always positioned at the centroid of the twin-hull.

4) The CoG is vertically aligned with the centre of buoyancy (CoB) at any given total mass, ensuring zero trim under initial load conditions.

2.1.3 Design parameters

The six design parameters, shown in Table 3, are of interest in the present study to explore the stability characteristics of the investigated suspension ship. Five of them relate to the suspension design, and the other one B_{CH} refers to the ship configuration. The initial values of the design parameters and their tuning ranges are also listed in Table 3. These initial design values are estimated based on the design parameters in the previous study of a suspension ship described in Han et al. (2022, 2023a). A scale ratio of 15 is adopted.

Table 3 Initial value and tuning ranges of design parameters

	$M_S + M_U$ (tonne)	M_S/M_U	B_{CH} (m)	k_s (tonne/m)	H_{AT} (m)	H_{RD} (m)
Initial value	150	2	9.6	56	1.5	4.32
Range	[50, 150]	[1, 4]	[7.2, 10.8]	[30, 100]	[1.5, 2.0]	[3.6, 4.32]

The mass terms are further discussed as the total mass of the ship and the mass ratio of the sprung and unsprung mass. The mass and spring stiffness units employed hereafter are tonne and tonne/m, chosen for their simplicity and convenience regarding spring deflection.

- Total mass $M_S + M_U$, and mass ratio M_S/M_U

The total mass and mass ratio vary instantaneously due to the loading/offloading operations. The mass ratio is 2 at both the lightship and full load conditions. The mass shown in Table 2 gives a mass ratio range of 1 to 4.

- Distance between the hulls' centrelines, B_{CH}

The distance between the hulls' centrelines is the transverse distance between the longitudinal centrelines of the two hulls. The initial B_{CH} is 9.6 m, and the initial length-beam ratio of the twin-hull is 2, while the length-beam ratio of one hull is 10. The choice of a length-beam ratio of 2 is due to two main considerations. First, the suspension design increases the vertical height of the vessel's CoG , leading to a lower transverse GM. A wider deck is adopted to mitigate the negative effects of the reduced GM. Second, a wider deck reduces the transverse inclination caused by load shifting on deck. It can improve operational stability and reduce the workload of the motion control system.

- Spring stiffness, k_s

The spring stiffness of the suspension system can affect the positions of the cabin during loading/offloading operations, which is one of the key parameters that dominate the ship's motion responses in waves. The total spring stiffness of four springs is given based on the value of a scale

model WHzer-6 according to Han et al. (2022, 2023a). The initial value is calculated as 56 tonne/m based on the Froude scaling.

- Allowable travel, H_{AT}

The allowable travel of the suspension system is defined as the vertical distance between the static initial load condition (static equilibrium) to the most compressed condition of the suspension. In Figure 4, H_{AT} is the distance between the suspension top line (STL) and the suspension bottom-out line (SBL). It signifies the physical limitation of the motion compensation capacity of a suspension ship in waves. This value varies with changes in sprung mass and spring stiffness, and it plays a crucial role in designing control strategies to prevent the suspension from bottoming out. In the initial design, the minimum value of H_{AT} is assumed to be 1.5 m at the full load condition.

- Ride height of suspension, H_{RD}

The ride height of the suspension is the vertical distance between the hull's baseline and the suspension bottom-out line. It has a limited tuning range in the design phase once the suspension system and the ship configuration are determined. For suspension ships in the configuration of a catamaran as those shown in Figure 1, the H_{RD} is the sum of DoA and the solid height of suspension. Whereas, for suspension ships in the configuration of monohull, this value could be smaller than DoA when the suspension is mounted to a location lower than the deck of the hull as presented by Han et al. (2023b).

2.2 Strip theory in the hydrostatic calculation

The hydrostatic loads acting on the hulls are calculated by integrating the hydrostatic forces acting on the 2D hull sections along the longitudinal direction of the twin hull. The hull sections of the investigated suspension ship are shown in Figure 5. The geometries of the 2D sections are expressed with a group of points lying on the boundaries, as shown in Figure 6. There are two different coordinate systems to describe the locations of the twin-hull: global coordinate system OXYZ and body-fixed coordinate system oxyz. The origin of the body-fixed coordinate system is located at the centre of the baseline of the twin-hull. The x-axis points from aft to fore along the direction of the keel. The y-axis points from starboard to port. The global coordinate system initially coincides with the body-fixed coordinate system. When there are heel and trim, the body-fixed coordinate system will rotate with respect to the CoG. The initial hull sections (dashed frames) and heeled hull sections (solid frames) are shown in Figure 6. The transformation from the body-fixed coordinate system to the global coordinate system is given using Equation (1), where $(x_{CoG}, y_{CoG}, z_{CoG})$ and $(X_{CoG}, Y_{CoG}, Z_{CoG})$ are the locations of CoG in the body-fixed coordinate system and the global coordinate system, ψ and θ are the trim and heel of the ship.

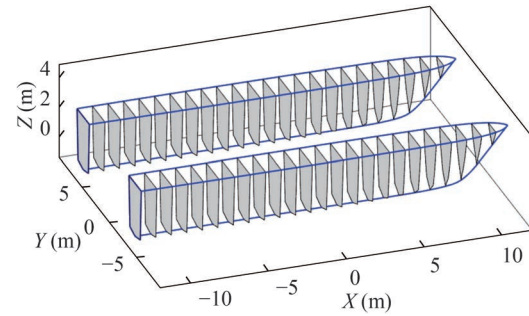


Figure 5 Hull sections of the investigated suspension ship, showing 24 sections for clarity

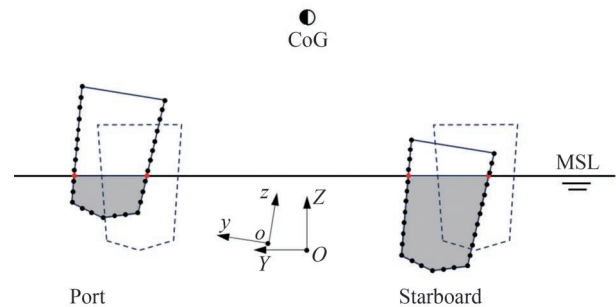


Figure 6 Description of the truncated sections of the suspension ship

$$\begin{bmatrix} X - X_{CoG} \\ Y - Y_{CoG} \\ Z - Z_{CoG} \end{bmatrix} = \begin{bmatrix} \cos \psi & 0 & \sin \psi \\ 0 & 1 & 0 \\ -\sin \psi & 0 & \cos \psi \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \theta & -\sin \theta \\ 0 & \sin \theta & \cos \theta \end{bmatrix} \begin{bmatrix} x - x_{CoG} \\ y - y_{CoG} \\ z - z_{CoG} \end{bmatrix} \quad (1)$$

The hull sections are truncated by the mean sea level (MSL) and the intersection points between the section boundaries and the MSL are calculated. The area and the first moments of area for the truncated sections are calculated using the equations in Table 4. The areas of submerged sections are used to calculate the volume of the displaced water. S_y and S_z are used to calculate the centroid of the truncated section as $y_0 = S_z/A$ and $z_0 = S_y/A$. The surface integrals of these variables are transferred to line integrals based on Green's first identity. The boundaries of the line integrals for the immersed region are discretized as N number of points (y_k, z_k) . In Table 4, $\bar{y}_{1,k} = 0.5(y_k + y_{k+1})$, $\bar{z}_{1,k} = 0.5(z_k + z_{k+1})$, $\bar{y}_{2,k} = 0.5(y_k^2 + y_{k+1}^2)$, $\bar{z}_{2,k} = 0.5(z_k^2 + z_{k+1}^2)$, $\Delta y_k = y_{k+1} - y_k$ and $\Delta z_k = z_{k+1} - z_k$. The centroid of the submerged area (X_0, Y_0, Z_0) in the global coordinate system is transferred from (x_0, y_0, z_0) using Equation (1).

The 2D results are integrated along the longitudinal direction to provide the buoyancy force and moments using Equations (2)–(4). When the gravity force and moments of the water inside the damaged tanks are considered, the directions of the force and moments in Equations (2)–(4) are opposite.

Table 4 Area and first moments of area of the hull sections based on Wen et al. (2023, 2024)

Variables	Surface integral	Line integral	Discretization
A	$\iint dydz$	$\oint -zdy$	$\sum_{k=1}^N -\bar{z}_{1,k} \Delta y_k$
S_y	$\iint zdydz$	$-\frac{1}{2} \oint z^2 dy$	$-\frac{1}{6} \sum_{k=1}^N (2\bar{z}_{1,k}^2 + \bar{z}_{2,k}) \Delta y_k$
S_z	$\iint ydydz$	$\frac{1}{2} \oint y^2 dz$	$\frac{1}{6} \sum_{k=1}^N (2\bar{y}_{1,k}^2 + \bar{y}_{2,k}) \Delta z_k$

$$F_z = \rho g \int A dx \quad (2)$$

$$M_x = \rho g \int A(Y_0 - Y_{CoG}) dx \quad (3)$$

$$M_y = -\rho g \int A(X_0 - X_{CoG}) dx \quad (4)$$

2.3 Static analysis method of the free-floating ship

During the intact stability calculation, the ship is considered to be in free trim condition. This means that when the ship is heeling, trim can be simultaneously induced due to the unsymmetrical shape at the bow and stern. The static responses of the suspension ships with free trim are calculated using a static analysis based on the hydrostatic calculation in Section 2.2 and the static equilibrium equations in draught, heel, and trim in Equations (5)–(7). In these equations, the subscript “P” denotes the port hull, “S” denotes the starboard hull and “W” denotes the water inside the damaged tanks. These equilibrium equations are numerically solved using the Newton-Raphson method as shown in Equations (8)–(10), where $C_{33} = \rho g A_w$, $C_{44} = mgGM_t$ and $C_{55} = mgGM_l$ are the restoring hydrostatic coefficients of the ship in heave, roll and pitch motions. $m = M_s + M_U$ is the total mass of the ship.

$$F_z = F_{P,z} + F_{S,z} - mg - F_{W,z} = 0 \quad (5)$$

$$M_x = M_{P,x} + M_{S,x} - M_{W,x} = 0 \quad (6)$$

$$M_y = M_{P,y} + M_{S,y} - M_{W,y} = 0 \quad (7)$$

The initial values of the draught d_0 , heel θ_0 and trim ψ_0 are chosen as zeros. The waterplane A_w , GM_t and GM_l are adopted as those of the normal case with zero trim and heel.

$$d_{j+1} = d_j - \frac{F_z}{C_{33}} \bigg|_{d_j} \quad (8)$$

$$\theta_{j+1} = \theta_j - \frac{M_x}{C_{44}} \bigg|_{\theta_j} \quad (9)$$

$$\psi_{j+1} = \psi_j - \frac{M_y}{C_{55}} \bigg|_{\psi_j} \quad (10)$$

For the intact stability calculation, Equation (9) is not included, and the heel angle is given as a known value. For the damage stability calculation, Equations (8)–(10) will be all updated.

When considering the elastic effects of the springs in the suspension system, the CoG_s in the body-fixed coordinate system is modified based on the balance in the direction of the springs. For a given heel angle of θ_j , the extension in the direction of the springs is calculated as

$$\Delta z = \frac{M_s g (1 - \cos \theta_j)}{k_s} \quad (11)$$

The workflow of the present numerical method is depicted in Figure 7. Initially, the sections of the hulls are initialized as illustrated in Figure 5. For a given heel angle, the static analysis model, enclosed within the dashed frame, is applied to determine the free trim and GZ of the suspension ship across three typical steps. The hull sections are rotated into the global coordinate system using Equation (1). The hydrostatic forces are then calculated using strip theory as outlined in Equations (2)–(4). The draught, heel, and trim are updated iteratively using Equations (8)–(10), accordingly. Importantly, the CoG of the entire suspension ship is updated to account for suspension deflection due to heeling and load variations on the cabin, thereby providing an accurate representation of the suspension ship's dynamic characteristics.

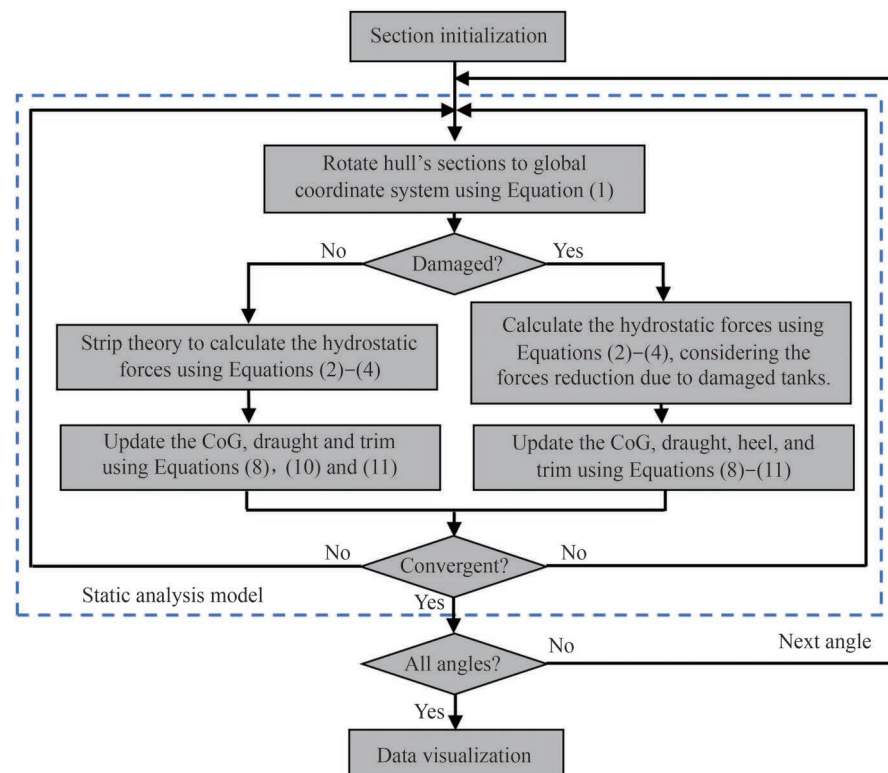
3 Results and Discussion

3.1 Verification study

3.1.1 Sensitivity study of hull surface discretization

In the present method, the hull surface is discretized into several sections, as shown in Figure 5. Each section is discretized into a number of points (see Figure 6). To ensure a proper hull surface discretization solution, sensitivity studies of the section interval and the number of points on each section are conducted.

Tables 5–8 show the sensitivity study results on the initial stability characteristics including the draught, longitudinal centre of buoyancy (LCB), vertical centre of buoyancy (VCB), transverse metacentric height GM_t , and longitudinal metacentric height GM_l at the lightship and full load conditions, respectively. The initial stability characteristics of the four tested section intervals and the numbers of points in each section show excellent consistency between each other. Notably, for the three finer discretization schemes, the discrepancies across all parameters are negligible. This indicated that the calculation has reached convergence. Therefore, the sensitivity of the initial stability is verified.

**Figure 7** Workflow of the present numerical model**Table 5** Initial stability obtained with different section intervals at the lightship condition

Section interval (m)	Number of points on each section	Draught (m)	LCB (m)	VCB (m)	GM_t (m)	GM_l (m)
1.2	19	1.131	−2.700	0.646	20.027	31.499
0.1		1.127	−2.694	0.643	20.008	31.453
0.2	301	1.127	−2.696	0.643	20.008	31.453
0.4		1.127	−2.695	0.643	20.008	31.453

Table 6 Initial stability obtained with different section intervals at the full load condition

Section interval (m)	Number of points on each section	Draught (m)	LCB (m)	VCB (m)	GM_t (m)	GM_l (m)
1.2	19	1.996	−2.261	1.109	9.838	18.325
0.1		1.992	−2.259	1.105	9.828	18.302
0.2	301	1.992	−2.259	1.105	9.828	18.302
0.4		1.992	−2.260	1.105	9.828	18.302

Figures 8–11 show the GZ and trim with different section intervals and different numbers of points on each section at the lightship and full load conditions, respectively. Good agreement of the GZ and trim is observed across all the section intervals and the numbers of points in each section. Therefore, the sensitivity of the intact stability is also

Table 7 Initial stability obtained with different number of points at the lightship condition

Section interval (m)	Number of points on each section	Draught (m)	LCB (m)	VCB (m)	GM_t (m)	GM_l (m)
	19	1.131	−2.690	0.646	20.026	31.496
0.2	151	1.127	−2.695	0.643	20.009	31.454
	301	1.127	−2.695	0.643	20.008	31.453
	601	1.127	−2.695	0.643	20.008	31.453

Table 8 Initial stability obtained with different number of points at the full load condition

Section interval (m)	Number of points on each section	Draught (m)	LCB (m)	VCB (m)	GM_t (m)	GM_l (m)
	19	1.996	−2.257	1.109	9.838	18.326
0.2	151	1.992	−2.259	1.105	9.828	18.302
	301	1.992	−2.259	1.105	9.828	18.302
	601	1.992	−2.259	1.105	9.828	18.302

verified. Notably, increasing the number of sections and points does not significantly extend calculation time using the present method. It can be concluded from the sensitivity study that the section interval and the number of points in each section can be chosen as 0.2 m and 301, respectively.

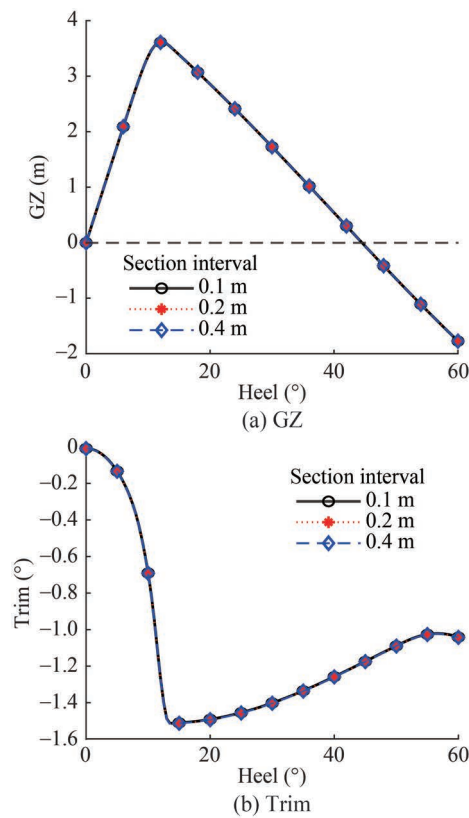


Figure 8 GZ and trim with different section intervals at the lightship condition

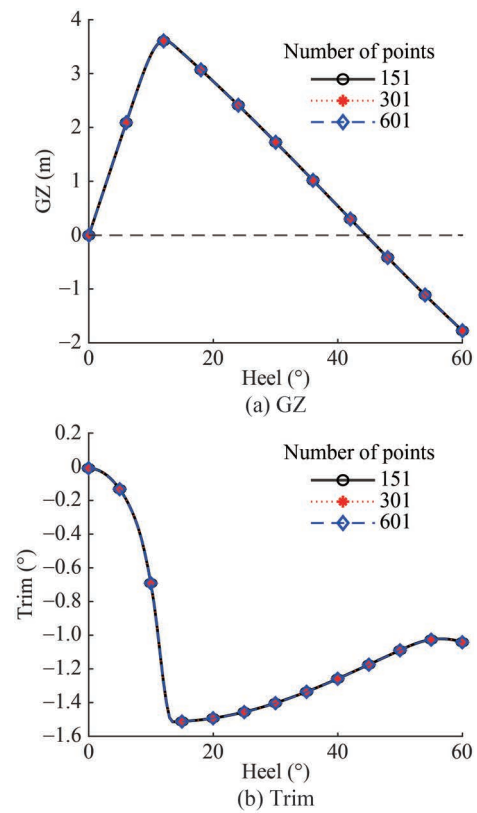


Figure 10 GZ and trim with different numbers of points on each section at the lightship condition

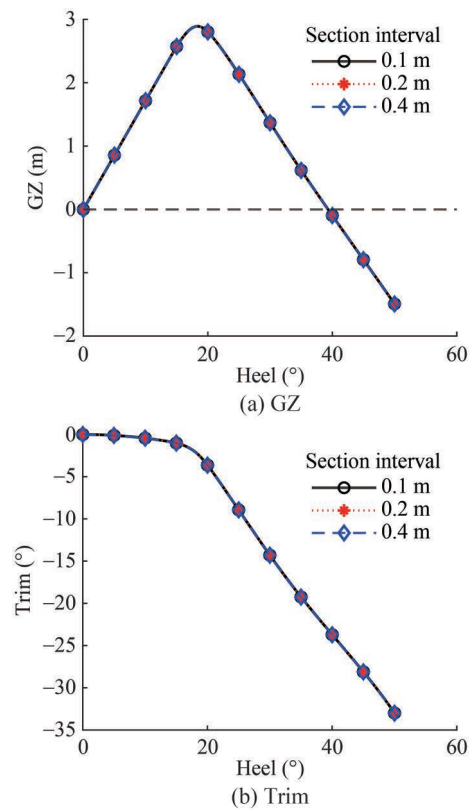


Figure 9 GZ and trim with different section intervals at the full load condition

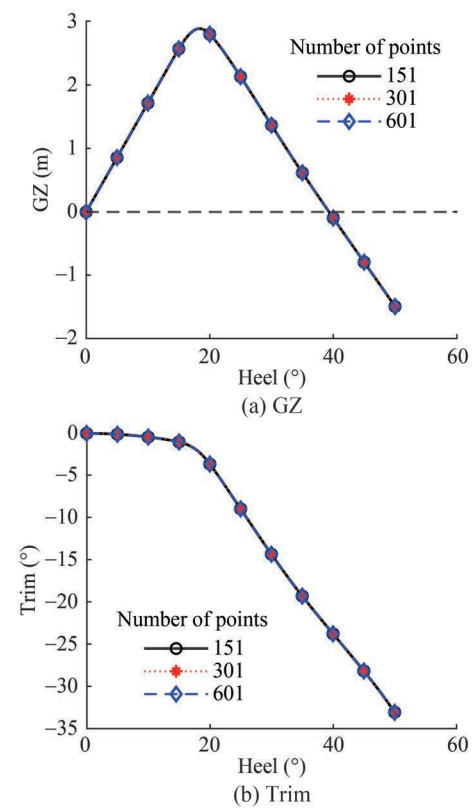


Figure 11 GZ and trim with different numbers of points on each section at the full load condition

3.1.2 Comparison with Orca3D

Orca3D is a plug-in marine design suite for Rhinoceros®. The hull design in Orca3D is done by using Non-Uniform Rational B-Splines surfaces. Hydrostatic properties and stability information are obtained by the surface model which adopts three-dimensional volume integration.

The accuracy of the present method in stability calculation is verified by comparing the intact stability results with those obtained from Orca3D under lightship and full load conditions. To ensure calculation consistency with Orca3D, the suspension deflection caused by ship inclination is temporarily disabled in this section when applying the present method.

The comparisons of the initial stability characteristics between the present method and Orca3D are shown in Table 9. A good agreement is achieved with a maximum error of less than 0.5%. Figure 12 shows the comparisons of GZ, trim, and righting moment between the present method and Orca3D at the lightship and full load conditions. An excellent agreement is observed, which proves the reliability of the present method in assessing the initial and intact stability of the investigated suspension ship.

Table 9 Initial stability results obtained by the present method and Orca3D

Condition	Section interval (m)	Draught (m)	LCB (m)	VCB (m)	GM_t (m)	GM_l (m)
Lightship	Present method	1.1266	-2.6945	0.6426	20.0082	31.4529
	Orca3D	1.1269	-2.7001	0.6428	20.0052	31.4461
	Error	0.03%	0.21%	0.03%	0.02%	0.02%
Full load	Present method	1.9920	-2.2597	1.1049	9.8277	18.3017
	Orca3D	1.9933	-2.2709	1.1056	9.8269	18.2997
	Error	0.07%	0.49%	0.06%	0.01%	0.01%

3.2 Influence of design parameters on intact stability

A parametric study involving the abovementioned six design parameters is performed to explore their influences on the intact stability of the investigated suspension ship. The transverse metacentric height, righting arm, righting moment, and trim are analysed.

The intact stability criteria of ships in length 24 m and over is defined in IMO Resolution MSC.267(85)–Adoption of the International Code on Intact Stability, 2008. This regulation is adopted to verify the safety of the investigated suspension ships. Regarding the GZ curve properties, it is required that the initial transverse metacentric height should not be less than 0.15 m; the area under the GZ curve up to 30 degrees (°) should not be less than 0.055 m·rad; GZ shall be at least 0.2 m at the heel equal to or larger than 30°; and

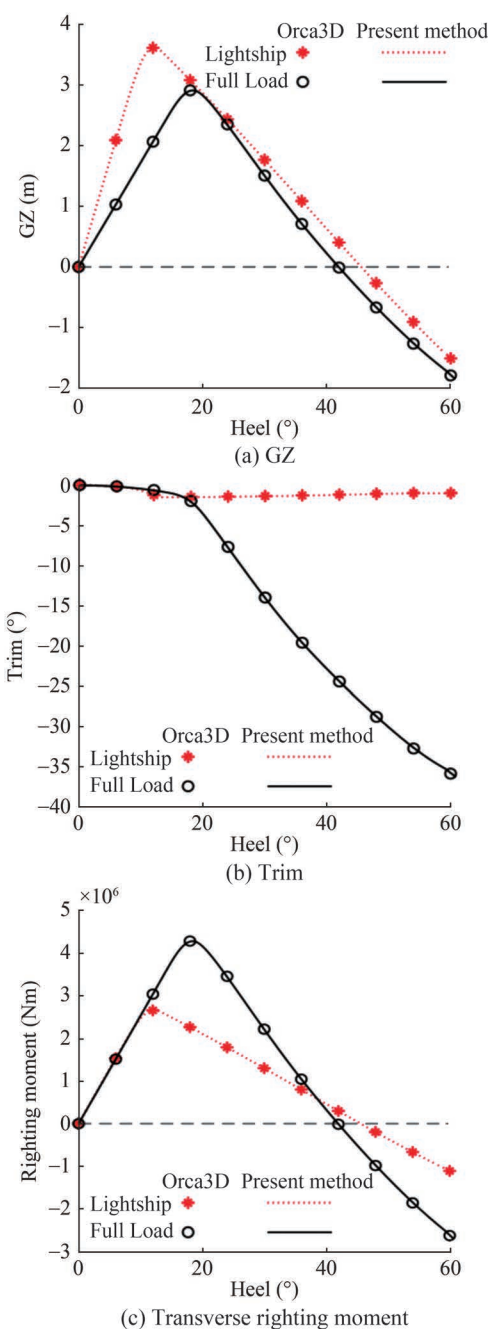


Figure 12 Comparisons of GZ, trim, and transverse righting moment between the present method and Orca3D

the maximum GZ shall occur at an angle not less than 25° (IMO, 2008).

3.2.1 Total mass

The GZ, trim, and transverse righting moment of the suspension ship are investigated at three load conditions (lightship, full load, and a representative operational condition). In these conditions, the mass ratio is kept as 2, and the mass distribution of the sprung and unsprung mass adheres to the assumption outlined in section 2.1.2, thereby the centre of gravity of the unsprung mass remains unchanged and

the ship's initial trim remains at zero.

In Figure 13, it can be seen that variation in total mass leads to a significant change in GZ, trim, and righting moment. An increasing total mass results in a decreasing GM_t and maximum GZ, as shown in Table 10. However, the heel of the maximum GZ increases with the increasing total mass. The heel of vanishing stability does not show an increasing or decreasing trend with the increasing total mass. The GM_t and the area under the GZ curves up to 30° satisfied the criteria, whereas the heel of maximum GZ did not.

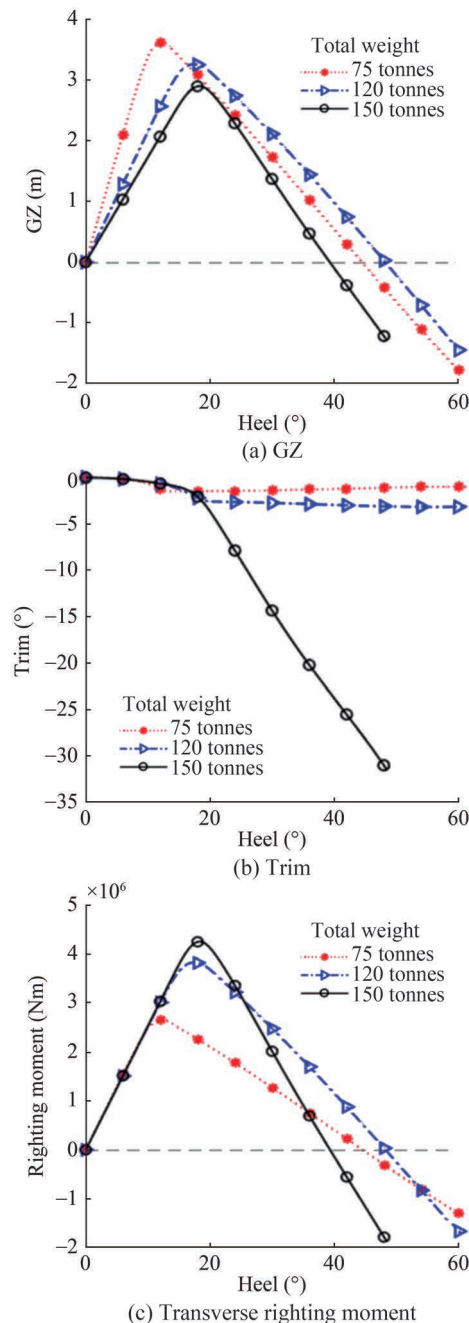


Figure 13 GZ, trim, and transverse righting moment of the suspension ship with different load conditions

Table 10 Initial and intact stability of the suspension ship with different load conditions

Total mass (tonne)	75	120	150
GM_t (m)	20.008	12.234	9.828
Maximum GZ (m)	3.610	3.251	2.892
Heel of maximum GZ (°)	12.0	17.5	18.5
Heel of vanishing stability (°)	44.5	48.0	39.0
Area under GZ curve up to 30° (m·rad)	1.284	1.156	0.952

It is interesting to notice that the trim of the suspension ship with a total mass of 150 tonnes shows a significantly different trend compared with those of 75 tonnes and 120 tonnes. To demonstrate the causes, the TCB, LCB, trim and GZ of the suspension ship with total mass varying from 125 tonnes to 131 tonnes at various heel angles are shown in Figure 14. In Figures 14(a) and 14(b), as the heel increases, the centre of buoyancy gradually shifts from the longitudinal centreline of the ship towards the longitudinal centreline of a side-hull, meanwhile progressively moving towards the stern. At a certain heel angle, sharp drops in TCB and LCB are observed at the total mass of 126 tonnes and 129 tonnes. Those drops indicate a sudden shift of the centre of buoyancy towards the centre-aft side of the ship. Consequently, there is a rapid increase in the trim of the ship and a sudden decrease in GZ, resulting in the suspension ship experiencing a significant coupling effect of heeling and trimming.

To further illustrate the relationship between the heel angle and the trim angle, the positions of the twin-hull are examined at a mass ratio of 2 with a total mass of 120 tonnes and 129 tonnes for comparison. The cabin and suspension system are omitted for clarity. The results are depicted in Figures 15 and 16. Figures 15(a) and 16(a) show the initial positions of the suspension ship where the heel angle is zero. Figures 15(b) and 16(b) show the positions of the suspension ship where the deck edge of the port side hull is about to immerse into the water surface. The difference in the heel angle between the two load conditions is one degree. Figures 15(c) and 16(c) show the positions of the suspension ship where the entire body of the starboard side hull is about to rise out of the water. One-degree difference in the heel angle is observed between the two load conditions. In Figures 15(d) and 16(d), at the same heel angle, it is found that a larger area of the aft part of the port side hull is immersed at a total mass of 129 tonnes compared with 120 tonnes. This causes the longitudinal centre of buoyancy to shift towards the stern. As shown in Figures 16(d) and 16(e), when the heel angle increased by 0.5° from 34.5° to 35.0° , the trim angle increased dramatically from -4.3° to -18.1° with a span of about 14° . This pronounced coupling effect of the heeling and trimming is consistent with the sharp drops observed in Figure 14.

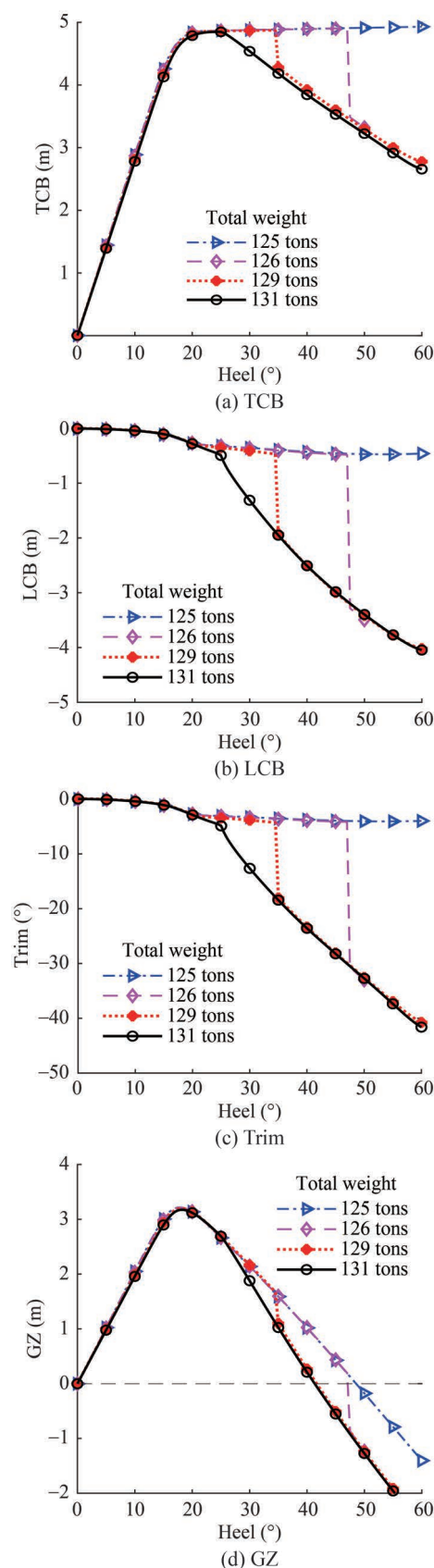


Figure 14 TCB, LCB, trim, and GZ of the suspension ship at different load conditions with a mass ratio of 2

With continuous heeling, the ship shows noticeable differences in position at the heel angle of vanishing stability in Figures 15(f) and 16(f).

The results indicate that total mass has a significant impact on the intact stability of the suspension ship. When the total mass exceeds a critical threshold, the intact stability undergoes a noticeable change due to the coupling effect of heeling and trimming, driven by the shifting in CoB. For the investigated suspension ship, this critical mass is 125 tonnes. Additionally, compared to conventional catamarans, suspension ships may have a smaller freeboard, meaning the deck edge can become immersed at a lower heel angle. This could result in different deck-water interactions than in traditional catamarans. Therefore, special attention should be given to the implications of reduced freeboard when assessing the operational limits and design considerations of suspension ships.

Based on these characteristics, it is recommended that the operational limitation of suspension ships could be defined by the maximum heel angle at which neither the hull's edge submerges, nor the hull's bottom lifts above the waterline. This limitation is crucial because the longitudinal stability of suspension ships may dramatically diminish once this heel angle is exceeded. Furthermore, hull edge submersion poses an additional risk: the potential ingress of seawater into the suspension linkages. This can increase fatigue risks and lead to higher structural stresses in the transverse direction of the suspension linkages.

3.2.2 Mass ratio

Figure 17 shows the GZ (a) and trim (b) of the suspension ship with a total mass of 120 tonnes at different mass ratios. The GZ decreases and the trim increases as the mass ratio becomes larger. Similar trends of GZ and trim are observed at a total mass of 150 tonnes, as shown in Figure 18. Tables 11 and 12 show the initial and intact stability of the suspension ship with a total mass of 120 tonnes and 150 tonnes, respectively. The GM_i , maximum GZ, heel angle of maximum GZ, and angle of vanishing stability all exhibit a decreasing trend as the mass ratio increases. This reduction is attributed to the elevation of the ship's CoG. While GM_i and the area under the GZ curves up to 30° satisfied the criteria, the heel of the maximum GZ did not meet the required value.

3.2.3 Beam of ship

The initial design value of the ship's beam is 12 m with a length of 24 m, which yields a length-to-beam ratio of 2. Two other length-to-beam ratios of 1.82 and 2.22 are chosen to investigate the influence of the beam on stability. Their corresponding beams are 13.2 m and 10.8 m, respectively. Figure 19 shows the GZ and trim of the suspension ship with different length-to-beam ratios at the lightship condition. The GZ shows a significant decrease with the increasing length-to-beam ratios. The trims with different length-to-

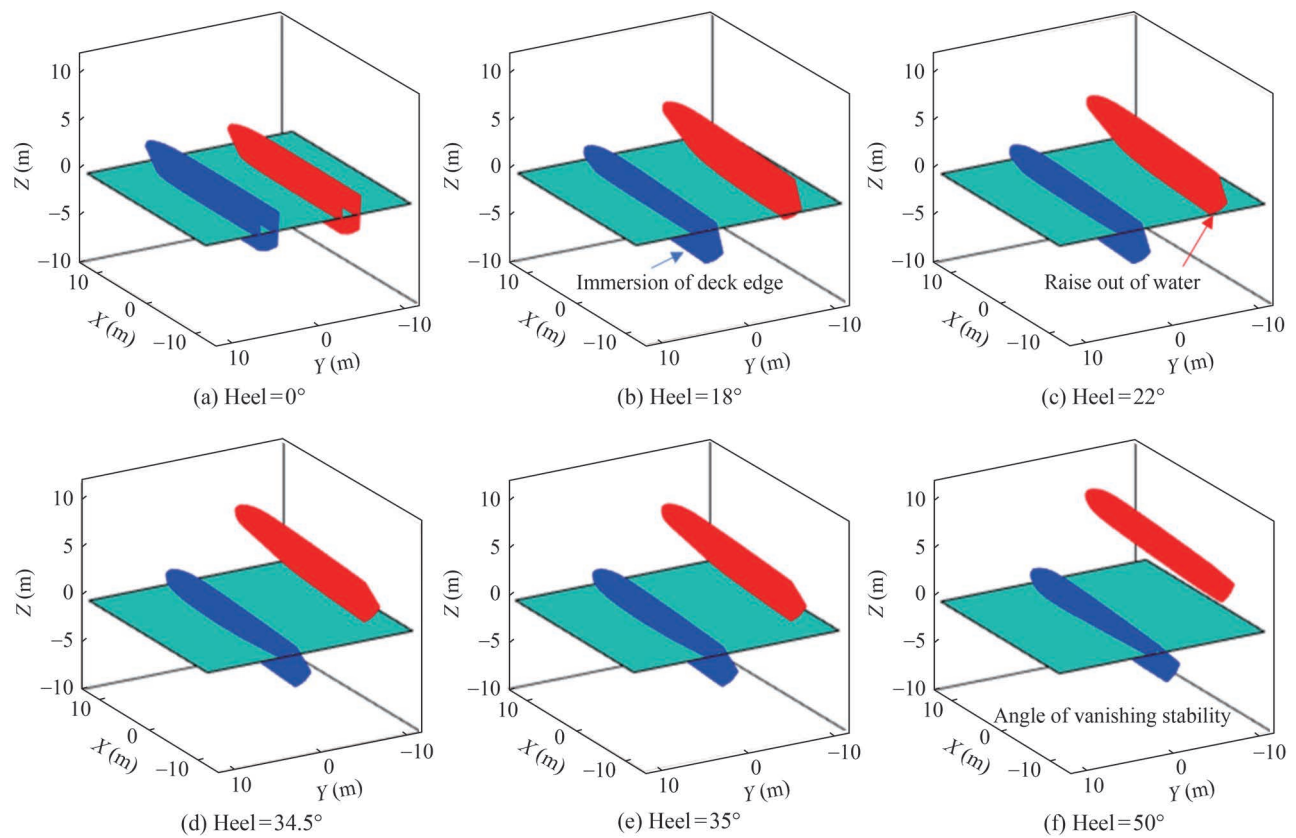


Figure 15 Positions of the suspension ship at different heel angles with a total mass of 120 tonnes

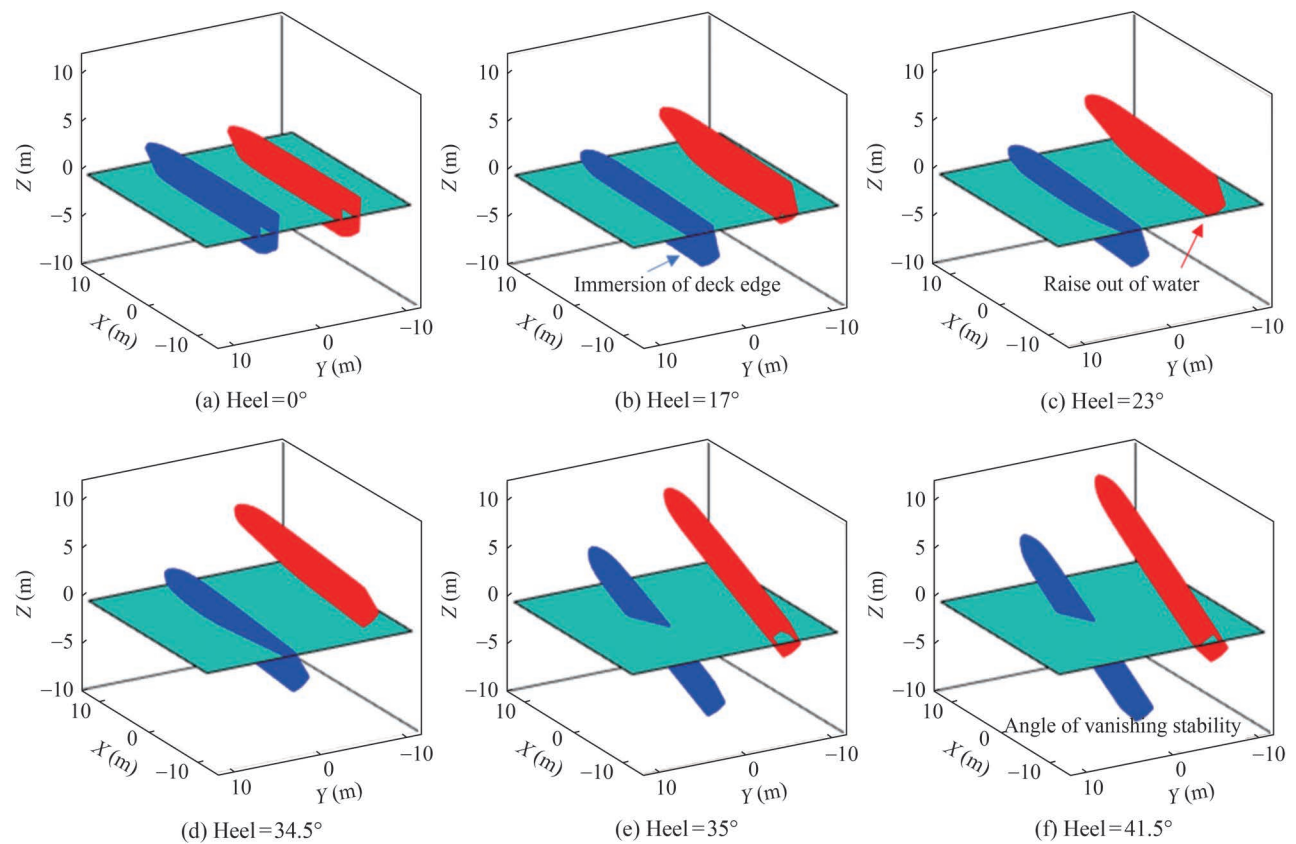


Figure 16 Positions of the suspension ship at different heel angles with a total mass of 129 tonnes

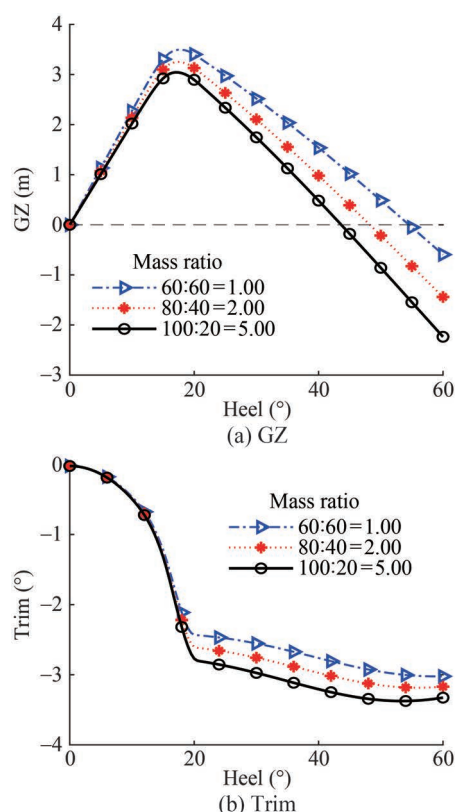


Figure 17 GZ and trim of the suspension ship with a total mass of 120 tonnes

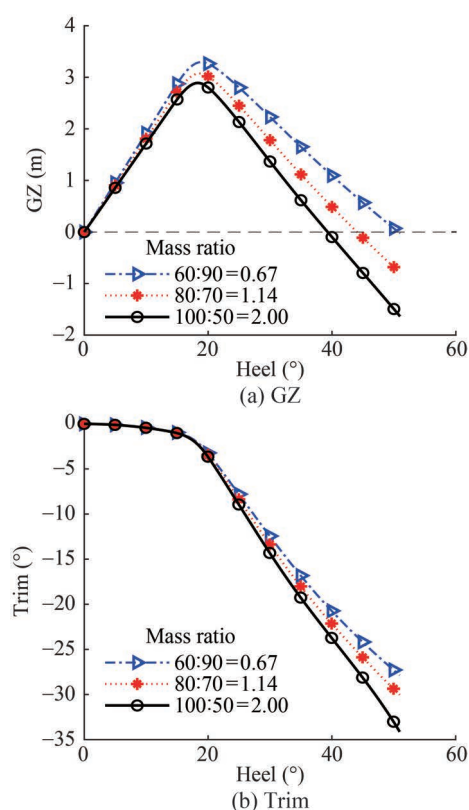


Figure 18 GZ and trim of the suspension ship with a total mass of 150 tonnes

Table 11 Initial and intact stability of the suspension ship with a total mass of 120 tonnes

Mass ratio	1	2	5
GM_t (m)	13.002	12.233	11.585
Maximum GZ (m)	3.496	3.251	3.044
Heel of maximum GZ (°)	17.5	17.5	17.0
Heel of vanishing stability (°)	54.5	48.0	43.5
Area under GZ curve up to 30° (m·rad)	1.263	1.156	1.063

Table 12 Initial and intact stability of the suspension ship with a total mass of 150 tonnes

Mass ratio	0.67	1.14	2
GM_t (m)	10.962	10.347	9.828
Maximum GZ (m)	3.298	3.078	2.892
Heel of maximum GZ (°)	19.0	18.5	18.5
Heel of vanishing stability (°)	50.5	44.0	39.0
Area under GZ curve up to 30° (m·rad)	1.140	1.041	0.952

beam ratios are close to each other. Figure 20 shows the GZ and trim at the full load condition. Similar to the lightship condition, the GZ shows a decreasing trend as the length-to-beam ratio increases. However, the trim in this case exhibits a significant decreasing trend with the increasing length-to-beam ratio when the heel exceeds 20°.

Table 13 shows the initial and intact stability of suspension ships with various length-to-beam ratios. It is noticed that GM_t and maximum GZ dramatically decrease when the length-to-beam ratio of the ship increases. However, the heel of the maximum GZ has a slight increase. The heel of vanishing stability shows a decreasing trend with the increasing length-to-beam ratio at the lightship condition. The heels of vanishing stability at the full load condition are close among the three length-to-beam ratios. The GM_t and the area under the GZ curves up to 30° satisfied the criteria, but not for the heel of maximum GZ.

3.2.4 Spring stiffness

The GZ and trim of the suspension ship with various spring stiffness at the lightship and full load conditions are depicted in Figures 21 and 22. The variations in GZ and trim due to different spring stiffnesses are negligible when the heel angle remains below the heel of the maximum GZ. However, once the heel angle exceeds this threshold, GZ decreases and trim increases more for softer springs. The reason for this trend is similar to that discussed in Section 3.2.2 regarding the influence of mass ratio. Softer springs allow for larger extension when the ship heels, leading to a higher GoG_s in the body-fixed coordinate system, which in turn reduces GZ and increases trim. The spring extension remains small when the heel is below the heel of maximum GZ. However, at larger heel angles the spring extension becomes substantial. This indicates that

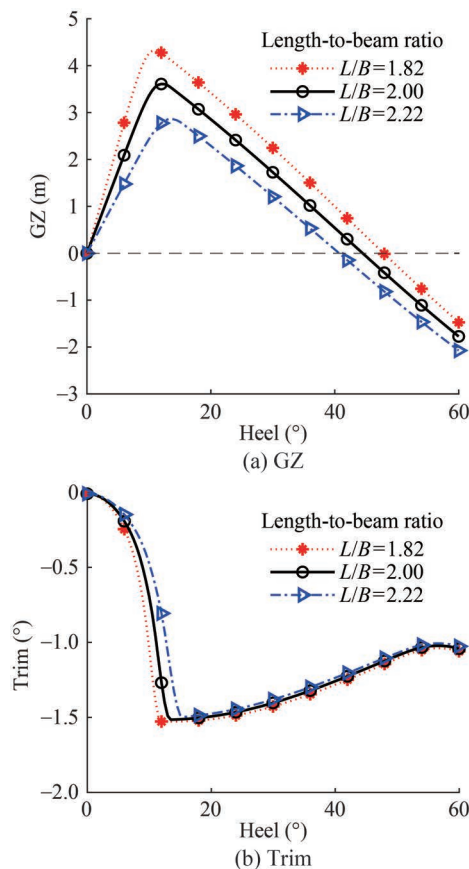


Figure 19 GZ and trim of the suspension ship with different length-beam ratios at the lightship condition

spring stiffness has a noticeable influence on intact stability for the suspension ship.

Additionally, the results shown in Figures 21 and 22 indicate that a stiffer spring leads to a larger value in heel of vanishing stability. This suggests that neglecting spring deflection, which is equivalent to assuming an extremely stiff spring, could overestimate intact stability. Therefore, it is essential to include the suspension system in stability analysis for suspension ships for accurate evaluation.

3.2.5 Allowable travel

Figures 23 and 24 show the GZ and trim of the suspension ship with different allowable travel at the lightship and full load conditions, respectively. The allowable travel varies from 1.5 m to 2.0 m. The GZ and trim of the suspension ship at the lightship and full load conditions show minimal differences between the two allowable travel conditions, which indicates that the allowable travel has limited influence on the initial and intact stability within the investigated range.

3.2.6 Ride height

The ride height of the suspension H_{RD} is set in three options, which are 3.6 m, 3.96 m, and 4.32 m, corresponding to 1.0, 1.1 and 1.2 times of the depth overall (DoA), respectively. When H_{RD} is equal to DoA, it means the bot-

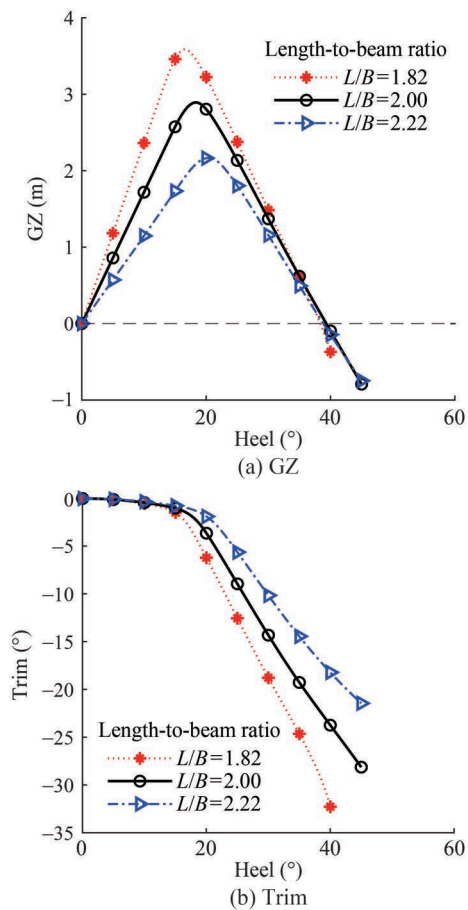


Figure 20 GZ and trim of the suspension ship at different length-beam ratios at the full load condition

Table 13 Initial and intact stability of the suspension ship with various length-to-beam ratios

	Mass ratio	1.82	2.00	2.22
Lightship	GM_i (m)	26.689	20.008	14.109
	Maximum GZ (m)	4.331	3.610	2.865
	Heel of maximum GZ (°)	11.0	12.0	13.5
	Heel of vanishing stability (°)	47.5	44.5	40.5
	Area under GZ curve up to 30° (m·rad)	1.584	1.284	0.984
Full load	GM_i (m)	13.560	9.828	6.534
	Maximum GZ (m)	3.588	2.892	2.167
	Heel of maximum GZ (°)	Mass ratio	18.5	20.5
	Heel of vanishing stability (°)	38.0	39.0	38.5
	Area under GZ curve up to 30° (m·rad)	1.180	0.952	0.707

tom-out line of suspension is located on the surface of the hull's upper deck. To achieve this, part of the suspension components needs to be mounted inside the hull, which

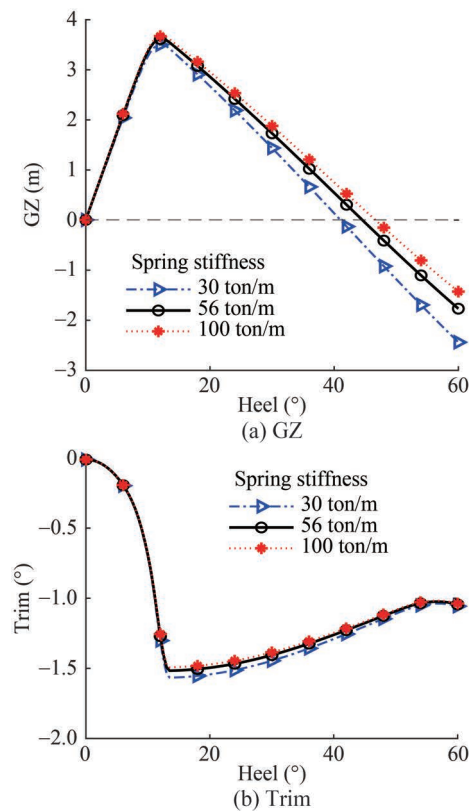


Figure 21 GZ and trim of the suspension ship with different spring stiffness at the lightship condition

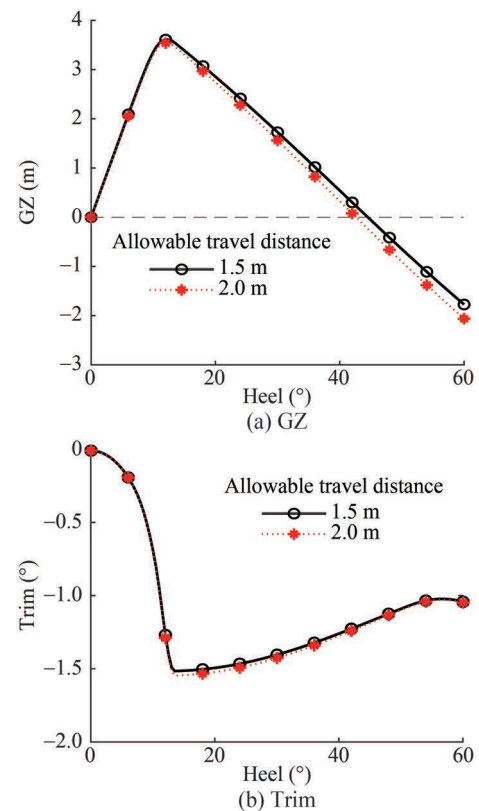


Figure 23 GZ and trim of the suspension ship with different allowable travel at the lightship condition

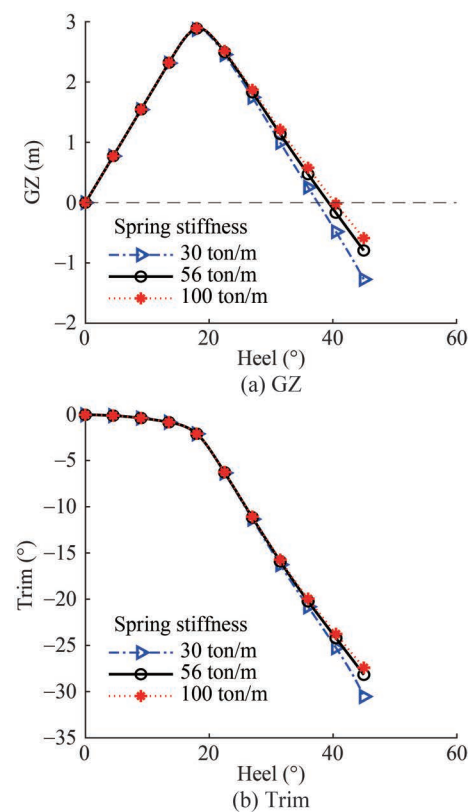


Figure 22 GZ and trim of the suspension ship with different spring stiffness at the full load condition

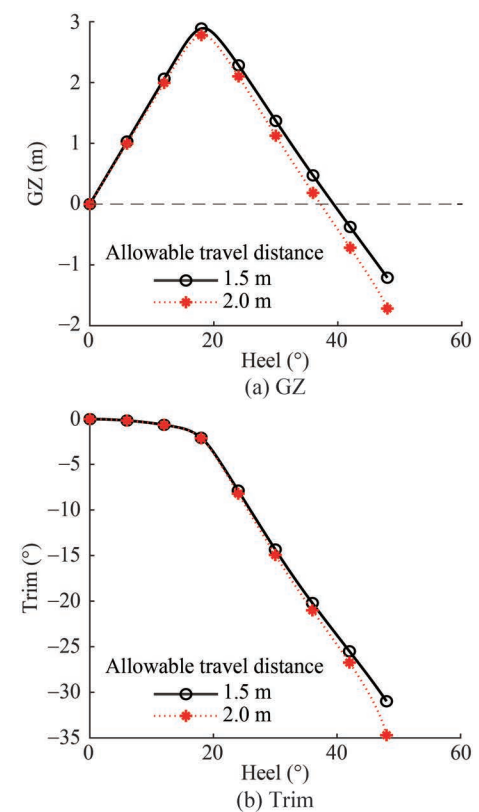


Figure 24 GZ and trim of the suspension ship with different allowable travel at the full load condition

could require specific efforts. In the initial design, the suspension is simply mounted on the hull's upper deck with $H_{RD} = 1.2 \times \text{DoA}$ as an assumption. Here, $0.2 \times \text{DoA}$ means the solid height of the suspension linkage.

Figures 25 and 26 show the GZ and trim of the suspension ship with different ride heights at the lightship and full load conditions, respectively. It is found that increasing ride height exerts limited impacts on the GZ and trim within the investigated range. These results are similar to those of the allowable travel. This is because the tuned range of these parameters has minimal influence on the variations of CoG_S .

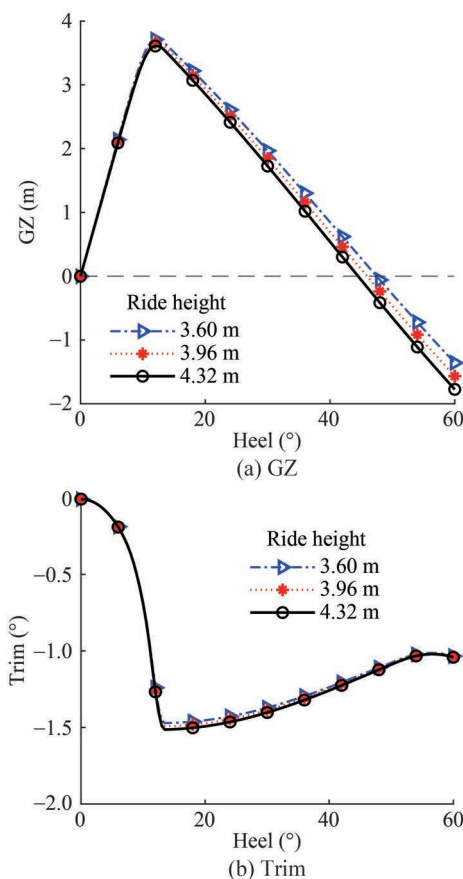


Figure 25 GZ and trim of the suspension ship with different ride heights at the lightship condition

3.2.7 Remarks

Based on the above analysis, the following key findings are summarized:

- The intact stability of suspension ships with free trim is investigated and the coupling effect of the heeling and trimming is revealed. The heel-trim coupling, caused by the shifting of CoB in longitudinal and transverse directions, becomes significant at large heel angles when the total mass exceeds a critical threshold. For the investigated ship, this threshold is 125 tonnes.
- Increasing the total mass results in a significant reduc-

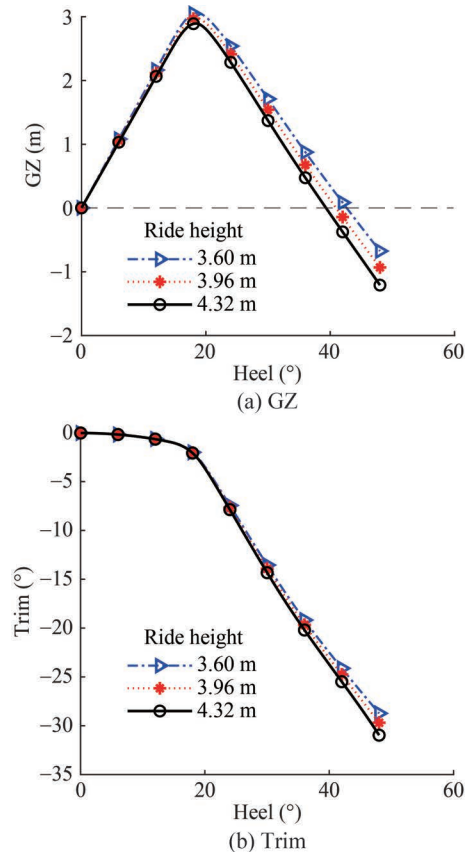


Figure 26 GZ and trim of the suspension ship with different ride heights at the full load condition

tion in both the transverse metacentric height and the maximum GZ.

- A higher mass ratio leads to a notable decrease in GZ and an increase in trim, particularly when the heel angle exceeds the heel of maximum GZ.
- The beam of the ship significantly influences GZ and trim. A wider beam improves intact stability but also results in larger trim under full load conditions.
- Spring stiffness has a noticeable effect on GZ and trim. Neglecting spring deflection can lead to an overestimation of the intact stability of suspension ships.
- Allowable travel and ride height of the suspension system have a limited impact on GZ and trim.

All initial transverse metacentric heights and the area under the GZ curves up to 30° in the investigated cases satisfy the criteria defined in Part A–Section 2.2 of IMO Resolution MSC.267(85) (IMO, 2008). However, the criterion related to the heel of maximum GZ is not met, as this value remains below 25° across all cases. For offshore supply vessels, alternative intact stability criteria are suggested in Part B–Section 2.4 of IMO Resolution MSC. 267(85), which requires the heel of maximum GZ to be not less than 15° (IMO, 2008). It is found that this requirement can be satisfied when the total mass of the investigated suspension ship exceeds 95 tonnes. Therefore, based on the intact

stability criteria alone, the minimum total mass of the investigated suspension ship should be set at 95 tonnes.

However, this study also reveals that when the total mass exceeds a critical threshold, excessive heel-trim coupling may occur due to significant shifts in the centre of buoyancy. For the current design, this threshold is identified at 125 tonnes. Accordingly, to ensure both compliance with stability criteria and avoidance of undesirable heel-trim coupling, a feasible total mass range for the investigated suspension ship lies between 95 and 125 tonnes.

Furthermore, while the present intact stability analysis accounts for suspension deflection due to inclination, it does not consider the effects of suspension submersion. To accurately assess the intact stability of suspension ships, particularly under conditions involving heavy load and submersion, caution should be applied, and modifications to existing stability criteria may be necessary to reflect the distinct structural characteristics of suspension ships.

It should also be noted that the investigated suspension ship has a large initial GM. It can lead to short roll periods and high lateral accelerations for the traditional catamarans without suspension system. However, prototype tank experiments have demonstrated that suspension systems are highly effective in attenuating oscillations in the vicinity of the resonance frequency (Han et al. 2023a). This suggests that short roll periods may not pose a primary design limitation during the early design stage of suspension ships. Nonetheless, a comprehensive evaluation, incorporating wave-induced motion response, control system performance, and operational conditions is essential to ensure safe and effective ship performance.

3.3 Dynamic stability

The restoring characteristics of the investigated suspension ship in wind and wave conditions are evaluated using the IMO weather criterion of Resolution A.562(14), which is also adopted in Resolution MSC.267(85). According to these criteria, a vessel is regarded as dynamically stable if the potential energy required to restore it to its upright position is not less than the energy imparted by wind forces causing it to heel. In practical terms, this means that the heeling angle θ_2 must not exceed either θ_c , θ_p or 50° . Here, θ_2 is defined as the heel angle at which the work done by wind equals the restoring potential energy, θ_c is the angle of second intercept between gust wind heeling lever and GZ curve, and θ_p is the angle of downflooding. In this analysis, θ_p is assumed to be greater than θ_c , making θ_c and 50° the limiting angles.

Known that the dynamic stability is sensitive to the VCG of the whole ship, the influence of the VCG on the dynamic stability is investigated under a load scenario of 120 tonnes. Figure 27(a) shows the variation of θ_2 with different mass ratios. To satisfy the weather criterion, the mass ratio must remain below 0.75, so that the angle of θ_2 does not exceed

either 50° or θ_c . Figure 27(b) presents the corresponding VCG values for the whole ship. The critical case, where $\theta_2 = 50^\circ$, occurs at a mass ratio of 0.75, resulting a VCG of 4.65 m. Therefore, for the investigated suspension ship at 120 tonnes with a mass ratio of 0.75, the maximum allowable VCG is identified as 4.65 m.

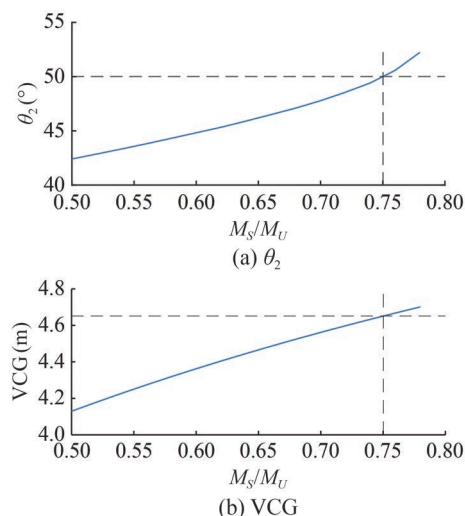


Figure 27 θ_2 and VCG of the suspension ship under various mass ratios at 120 tonnes in the dynamic stability analysis

However, these results should be interpreted with caution and validated through experimental methods, as the equations and loop-up tables defined in Resolutions A.562(14) and MSC.267(85) are derived from ships with BoA/draught ratios less than 3.5. The BoA/draught ratio of the investigated suspension ship exceeds 6, which lies substantially outside the criteria's original range of applicability. In such cases, Resolution MSC.267(85) recommends conducting model experiments following the procedure outlined in MSC.1/Circ.1200 (IMO, 2006).

3.4 Damage stability

Based on the initial and intact stability study, the suspension ship with a total mass not higher than 125 tonnes shows better intact stability than a heavier one. In the damage stability study the ship with a total mass of 120 tonnes is chosen whose mass ratio is 2.0. Other design parameters are the initial values shown in Table 2.

The IMO Resolution MSC.429(98) outlines a detailed methodology that uses the probability of survival as a measure of ships' safety after damage (IMO, 2020). In this section, the focus is given to the calculation of trim, heel, GM, and the righting lever curve (including GZ max and its range) for each possible single-compartment damage scenario. This serves as an initial step in evaluating the probability of survival for suspension ships following hull damage.

The free surface effect within the damaged watertight compartments is considered in the damage stability analy-

sis. When the hull is ruptured, sea water flows in and out as the ship heels or trims, resulting in continuous weight variation and the consequent shifting of the ship's CoG. To enhance damage control readiness, the hull is divided into eight watertight tanks, as shown in Figure 28. A horizontal bulkhead is incorporated to limit significant sinkage at the stern caused by flooding in one of the aft tanks. Each watertight tank has a height of 1.8 m. The length of the tanks No. 2–8 is 6.0 m, while that of tank No. 1 is 4.49 m. It should be noted that at 120 tonnes, the ship's draught is 1.657 m in intact condition, which is less than the compartment height. As a result, the upper tanks No. 5–8 remain above the mean sea level initially, which will only flood once the ship heels beyond a certain angle.

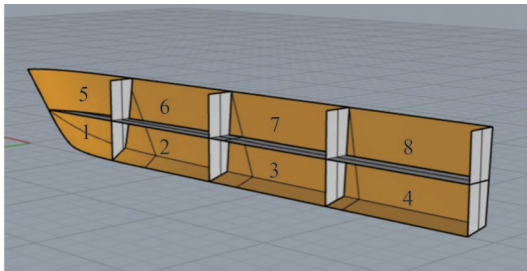


Figure 28 Layout of the eight watertight compartments

The results of the damage stability are summarized in Table 14, while the GZ and trim versus heel angle are illustrated in Figure 29. For damage occurring in the lower tanks from bow to stern (tanks No. 1 to 4), the list of the ship at the equilibrium position increases and the GZ range decreases. For damage in the upper tanks from bow to stern (tanks No. 5 to 8), the maximum GZ and GM decrease. Among all scenarios, the most critical condition occurs when tank No. 8 is ruptured, resulting in the greatest stability deterioration.

Table 14 Results of damage stability of the suspension ship with eight watertight compartments, the results of intact stability are given for comparison

Damaged Tank	Equilibrium		GZ Range (°)	Maximum GZ (m)	GM (m)
	List (°)	Trim (°)			
None	0	0	~ 48.0	3.251	12.234
No. 1	1.060	1.113	~ 48.5	3.336	12.434
No. 2	3.229	1.642	~ 49.5	3.323	12.506
No. 3	3.728	-0.657	~ 40.5	2.897	12.037
No. 4	3.903	-3.464	~ 34.0	2.133	11.522
No. 5	0	0	~ 42.5	3.212	11.114
No. 6	0	0	~ 48.0	3.210	9.987
No. 7	0	0	~ 37.5	2.768	9.738
No. 8	0	0	~ 30.0	1.731	7.905

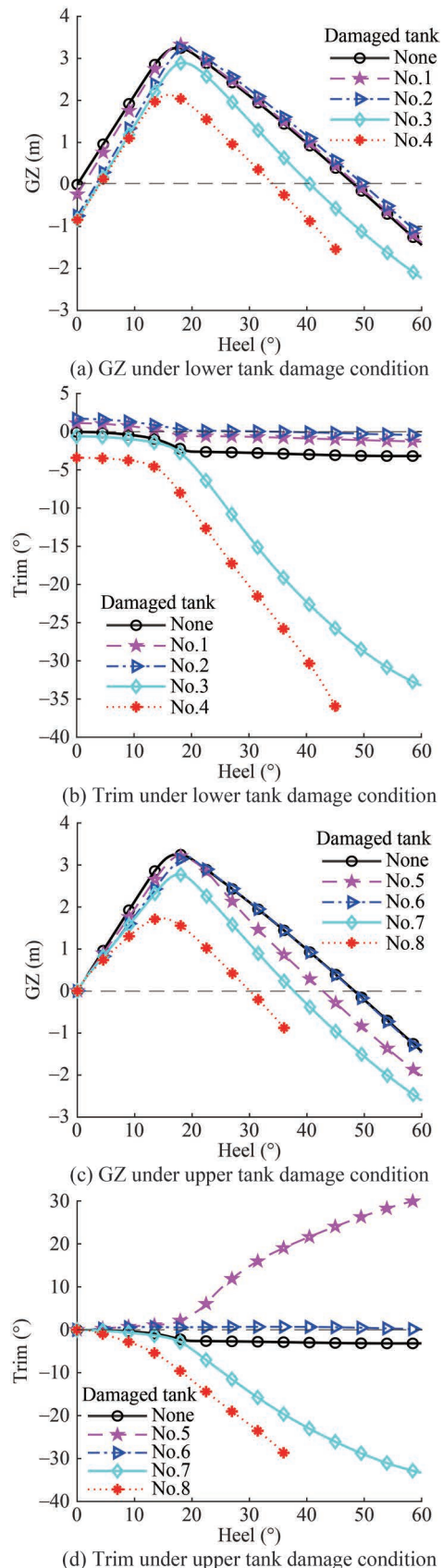


Figure 29 GZ and trim of the suspension ship with an eight-tank configuration at a total mass of 120 tonnes

Damage occurring at the aft of the ship has a greater impact on stability compared to damage at the front. As shown in Figure 29, a comparison between damage in the aft tanks (No. 3, 4, 7, 8) and the front tanks (No. 1, 2, 5, 6) reveals a substantial reduction in the area under the GZ curves and a more pronounced change in trim. Consequently, the ship experiences a larger inclination in both heel and trim when damage is located in the aft half of the investigated ship.

Furthermore, the added weight of floodwater could induce trim-heel coupling of the suspension ship, further deteriorating the ship's damage stability. Figures 29(b) and 29(d) show that when damage occurs in tanks No. 1, 2 or 6, the trim remains low even as the heel angle increases. However, damage in further tanks (No. 3, 4, 5, 7, 8) causes a significant reduction in the ship's longitudinal stability, leading to a stability condition comparable to that of an intact ship with a total mass of 131 tonnes (as seen in Figure 14).

4 Conclusions

A set of parameters unique to suspension ships, which are not typically considered in conventional ship designs, is introduced in the present study. The initial, intact, and damage stability analyses of a suspension ship are conducted using a hydrostatic force model and a static analysis model. The numerical method presented is verified by a sensitivity study of hull surface discretization and comparisons of the initial and intact stability obtained using Orca3D. The influences of six design parameters, including the total mass, mass ratio, beam of the ship, spring stiffness, allowable travel, and ride height of the suspension system, on the stability of the investigated suspension ship are summarized as follows:

1) The coupling effect of the heeling and trimming could occur when the total mass exceeds a critical threshold. It leads to excessive trimming, especially at large heel angles, owing to substantial shifts of CoB in longitudinal and transverse directions. As a crew transfer vessel, the total mass of the investigated suspension ship should be larger than 95 tonnes to meet the intact stability, and less than 125 tonnes to prevent severe heel-trim coupling.

2) A smaller mass ratio is preferable, as increasing the mass ratio significantly decreases the GZ and increases trim, particularly when the heel exceeds the heel of maximum GZ.

3) Neglecting suspension deflection can result in an overestimation of the intact stability of suspension ships. The spring stiffness and the beam of the ship have a noticeable impact on intact stability, while the allowable travel and ride height have limited effects.

4) A dynamic stability analysis based on the IMO weather criterion indicates that the mass ratio should not exceed 0.75, however, experimental validation is necessary.

5) The added weight of floodwater at the bow and stern can cause significant heel-trim coupling, leading to large trim angles when the ship heels further, thereby deteriorating damage stability.

This study provides insights into the stability analysis of suspension ships, incorporating novel design parameters that distinguish them from traditional ships. Future work will focus on analysing wave-induced motion responses of the investigated suspension ship and experimental validation of the dynamic stability analysis. The structural strength of suspension components will be evaluated to support their practical applications as crew transfer vessels of offshore wind farms. By integrating motion response studies, the suspension ship design optimization will be conducted to enhance performance and safety. These efforts aim to advance knowledge, contribute to guideline development, and support regulation modifications for marine suspension technologies.

Nomenclature

A	Area of a section
A_w	Area of waterplane
b	Beam of hull
B_{HH}	Transverse clearance between two hulls
B_{CH}	Transverse distance between longitudinal centreline of two hulls
C_{33}, C_{44}, C_{55}	Hydrostatic restoring coefficients
d	Draught of the ship
g	Gravitational acceleration
GM_l	Longitudinal metacentric height
GM_t	Transverse metacentric height
GZ	Righting arm
KB	Vertical distance between K and CoB
KG	Vertical distance between K and CoG
H_{AT}	Vertical distance between STL and SBL
H_{GS}	Vertical distance between CoG _s and STL
H_{GU}	Vertical distance between CoG _u and baseline
H_{RD}	Vertical distance between SBL and the baseline
H_{SD}	Vertical distance between the suspension bottom to SBL
k_s	Spring stiffness
M	Mass
S_y	First moments of area in the y direction
S_z	First moments of area in the z direction
x, y, z	Coordinates of the body-fixed coordinate system
X, Y, Z	Coordinates of the global coordinate system
x_0, y_0, z_0	Centroid of a section in the body-fixed coordinate system
X_0, Y_0, Z_0	Centroid of a section in the global coordinate system

θ	Heel angle
ψ	Trim angle
ρ	Density of sea water

Subscript

j	Iteration index in the static analysis
k	Section index
P	Port side
S	Sprung body or Starboard side
U	Unsprung body
W	Water inside the damaged tanks

Abbreviation

BoA	Beam overall
CoB	Centre of buoyancy
CoG	Centre of gravity of the whole ship
DoA	Vertical distance between the baseline and the top deck of the hull at midship
IMO	International Maritime Organization
LCB	Longitudinal CoB
LoA	Length overall
MSL	Mean sea level
SBL	Suspension bottom-out line
STL	Suspension top line
TCB	Transverse CoB
VCB	Vertical CoB

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