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Numerical Simulation of Installation Ship's Dynamics by Crane's Cargo Loss

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Master Thesis

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ABSTRACT

Crane load lifting failures in the past have raised some serious concerns about the safety of offshore operations. Numerous efforts are being made to make the large offshore wind turbine installation operation safe and reliable. A vessel is being designed by Neptun Ship Design (NSD) to carry out offshore wind turbine installation operation in deep sea. The purpose of master thesis is to carry out a time domain simulation of installation vessel in roll, pitch, and heave motion due to sudden load loss. A comparative study of corresponding rules of classification societies (DNV, BV, and ABS) have been documented and followed. Based on the classification recommendations, corresponding mathematical modelling is designed and time domain simulation in case of sudden loss of crane load is carried out and analysed in single degree of freedom and coupled motion. An estimation of added mass, damping and restoring coefficient is made on the bases of literature review, analytical approximation and using contrikov's method. A sensitivity analysis of added mass is carried out to check the response of the vessel especially maximum heel angle. Moreover, the ship ability to minimize the roll angle by discharging of water ballast is analysed and its practical implementation is accessed.

1. INTRODUCTION

Due to recent global warming issues, global and European countries are putting an effort to decarbonize the economy by 2050. In 2017, another significant move taken place by shifting from fixed based to floating offshore wind turbine. Offshore wind is a highly favorable renewable energy source that is capable to make a huge contribution in this regard. High wind currents in deep sea can produce more energy than the coastal area. Installation of wind turbines in deep sea imposes some challenges in terms of cost, transportation, and installation. With gradual increase in the size and capacity of offshore wind turbine, the new ships are required for transportation, handling, and installation. Neptun Ship Design has proposed a solution of the crane mounted vessel to install offshore wind turbine directly on site and subsequently operational cost reduction. This project is under the framework of R&D project OWSplus, supported by Germany's Federal Ministry of Economic Affairs and Energy.

In May 2020 [1], crane mounted vessel's (Orion 1) hook collapsed during the load test of crane at the port of Rostock which resulted in a collapse of complete crane. To avoid such incidents during actual operation and to ensure installation ship integrity, safety and reliability, numerical simulation of proposed installation ship is to be carried out in case of sudden load loss. A numerical simulation in time domain is to be performed to investigate the effect in roll, pitch and heave motions in case of sudden loss of load.

The proposed installation vessel is demi hull catamaran type. The ship dynamics have been studied for the vessel carrying 1500 tons of load on the port side while the heel angle of the ship made 0^o by adding ballast to the starboard side of the ship. Simulation starts from the point when a sudden loss of 1500 load occurs in aforementioned condition.

The roll motion of ship is rotation of the ship around its longitudinal axis. Rotation around the transversal axis is known as pitch and translational motion of the ship along the vertical direction of the ship is called heave motion. The properties of motion of the ship due to sudden cargo loss of the load are determined by the ship hull shape, the mass distribution, and the motion of the surrounding fluid. The current study will be carried out considering the calm sea condition i.e., no induced motion due to sea waves.

The evaluation of dynamic stability of ships in the early design phase is needed. In modern naval engineering practice, the motion of a ship is evaluated with computational methods based on the potential theory and are sufficient for ship design purposes. However, they cannot predict viscous damping, which are determined by other means such as model tests, viscous field

methods or empirical prediction methods. Physical tests are costly and time consuming, but also can have error due to scale effects. Empirical methods like the Ikeda method has its own limitations as their development was based on a restricted number of hull shapes (e.g. slender bodies) and may not be able to work for today's modern ship designs [2].
However, the master thesis focuses on the method proposed by classification societies and corresponding coupled coefficients are estimated based on available analytical formulas and contrikov's method [3].

The main dimension and characteristics of the ship to be analyzed are mentioned in table (1.1).

Table 1.1 Main characteristics of ship

Characteristic	**Value**
Ship Type	Catamaran Twin Hull
Length Between Perpendiculars L_{pp}	158 m
Breadth of Hull	75 m
Design Draught	10 m
Working Draught	15.5 m
Crane Hanged Load	1500 Ton
Transverse Metacentric Height (After Load Loss)	25.83 m
Longitudinal Metacentric Height (After Load Loss)	104.29 m
Total Weight of Ship (Upright Position)	81539.9 Ton
Total Weight of Ship (After Load Loss)	80039.9 Ton
Availability of Ballast System	Yes
Vertical Center of Gravity (Upright Position)	44.2 m
Longitudinal Center of Gravity (Upright Position)	75.48 m
Transvers Center of Gravity (Upright Position)	0.00 m
Vertical Center of Gravity (After Load Loss)	41.37 m
Longitudinal Center of Gravity (After Load Loss)	75.55 m
Transvers Center of Gravity (After Load Loss)	2.83 m

Pictorial view of twin Hull catamaran under examination can be seen in the figure (1.1). The twin hull catamaran vessels generally posses better stability, provide great resistance to capsizing and stabilize the roll at rest position which is crucial in operating conditions of this particular ship. However, these ships are more sensitive to pounding while underway in heavier seas than conventional monohull ships.

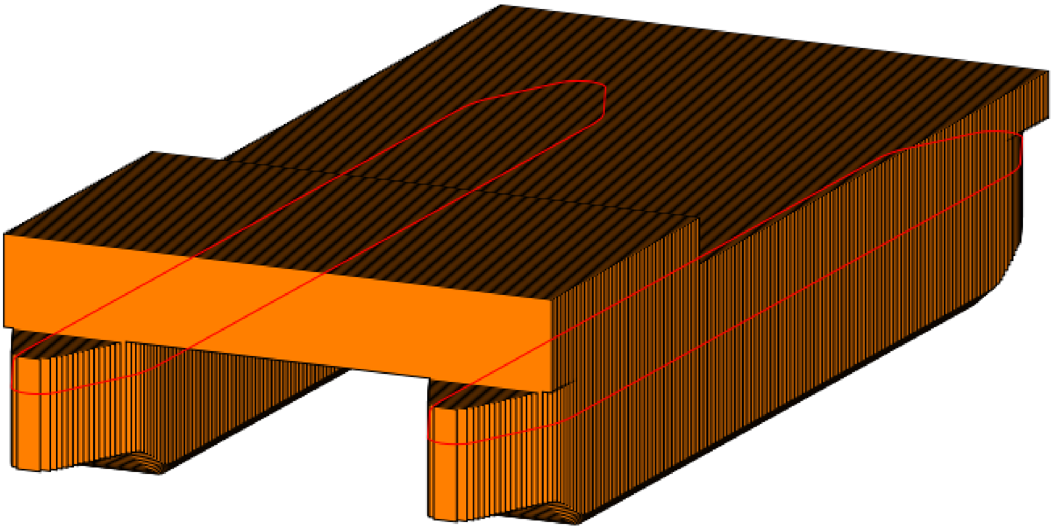

Figure 1.1 Hull shape of crane mounted vessel

2. CLASSIFICATION SOCIETY RULES

To start with the analysis, a literature review of classification society is conducted in case of sudden loss of cargo load. Three classification societies have been consulted, i.e., Det Norske Veritas (DNV), Bureau Veritas (BV) and American Bureau of Shipping (ABS).

2.1 Det Norske Veritas Rules

In case of counter ballasting measures used during the lifting operations, it is recommended to consider the effect of sudden loss of load due to hook failure. If cargo hatch covers are open, then these hatch covers shall be considered as down flooding points. It is recommended to find a maximum dynamic heeling angle by performing simplified analysis of a one degree of freedom differential equation of motion or a fully viscous RANSE-based CFD calculation.

2.1.1 Simplified Roll Motion Analysis

The dynamic heel angle after a sudden loss of load is estimated by considering single degree of freedom motion (Roll motion) starting with the ship at equilibrium and releasing the hook load, simulating a sudden failure of hook load. The roll equation of motion is described in equation (2.1).

$$(I_{roll} + I'_{roll})\varphi'' + B\varphi' + C\varphi = 0 \tag{2.1}$$

Where

I_{roll} = Mass roll moment of inertia of the ship in loading condition after loss of hook load, in kNm-s^2

I'_{roll} = Added mass roll moment of inertia of the ship in loading condition after loss of hook load, in kNm-s^2

φ'' = Roll acceleration, in deg/s

φ' = Roll velocity, in deg/s

B = Linearized roll damping coefficient

C = Roll restoring moment, in kNm, as a function of roll angle

It is recommended to take the linearized roll damping coefficient as 2% of critical damping and higher damping coefficients are acceptable if roll decay test results or valid numerical calculations are available. GZ curve after the loss of hook load as a function of roll angle must be used [4]. The maximum heel angle φ_3 during the roll motion must not exceed the smaller of angle of down flooding φ_F with a safety margin of 3^o or angle of vanishing stability φ_R with a safety margin of 7^o as described in equation (2.2).

$$\varphi_3 < \min(\varphi_F - 3^o, \varphi_R - 7^o) \tag{2.2}$$

2.1.2 Reynolds Averaged Naiver Stokes (RANS) simulation of roll motion

Dynamic heel angle of the ship because of the sudden loss of hook load can also be assessed using RANS equation solver which provides non-linear numerical simulation, but during the simulation, it must account for the ship's hull, stability pontoon and those parts of super structure that contribute to the righting moment [4]. The maximum heel angle φ_3 during the roll motion must not exceed the smaller of angle of down flooding φ_F with a safety margin of 2^o or angle of vanishing stability φ_R with a safety margin of 7^o as described in equation (2.3).

$$\varphi_3 < \min(\varphi_F - 2^o, \varphi_R - 7^o) \tag{2.3}$$

2.2 Bureau Veritas Rules

Bureau Veritas rules recommend considering the case failure of crane hook or lifting cable whenever counter ballast is used.

2.2.1 Intact Stability Criteria during Lifting Operation

The following intact stability criteria must be complied

a) The angle of static heel angle θ_c in equilibrium must be less than one of the following depending on whichever occurs first [5]
   - 10 degrees, or
   - Angle of deck immersion, or

- Allowable value of crane list and trim

b) The area under the GZ curve measured from angle of static equilibrium θ_c to the down flooding angle θ_f or 20 degrees, whichever is less must be at least 0.03 m rad.

2.2.2 Weather Criteria during Lifting Operation

The following criteria must be complied

a) The area under the righting moment curve (A+B) as shown in figure (2.1), to the down flooding angle θ_f or second intercept, whichever is less must be more than 40% of the area under heeling moment due to wind (B+C) as shown in equation (2.4).

$$\frac{A+B}{B+C} \geq 1.40 \tag{2.4}$$

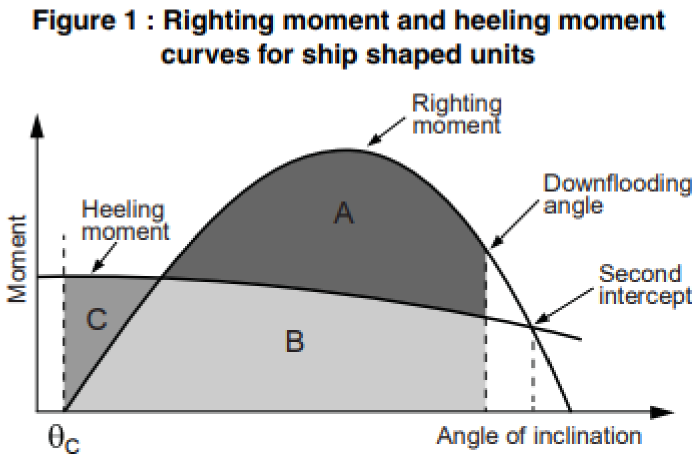

Figure 2.1 Righting moment and heeling moment curves [5]

2.2.3 Intact Stability Criteria in the Event of Sudden Load Loss

Apart from recommended intact stability criteria during lifting operation and weather criteria during lifting operation, the floating unit to withstand the effect of sudden lifted load loss shall be complied.

a) Prior to the load loss, the ship is at static equilibrium θ_{c0}. Then after sudden loss of load, ship will heel towards ballast side and make an equilibrium heel angle θ_{c1}. The new static heel angle must not exceed the 15 degrees from upright as mentioned in the equation (2.5).

$$\theta_{c1} \leq 15^o \qquad (2.5)$$

b) under this case area B as indicated in figure (2.2), under the righting arm curve GZ_1 to the second intercept or the angle of down flooding θ_f must not be less than 40% in excess of area A under the righting arm curve to the initial static equilibrium heel angle θ_{c0} [5]. The ratio of the areas to be complied is mentioned in the equation (2.6)

$$\frac{B}{A} \geq 1.40 \qquad (2.6)$$

Figure 3 : Righting moment curve after sudden loss of the lifted load

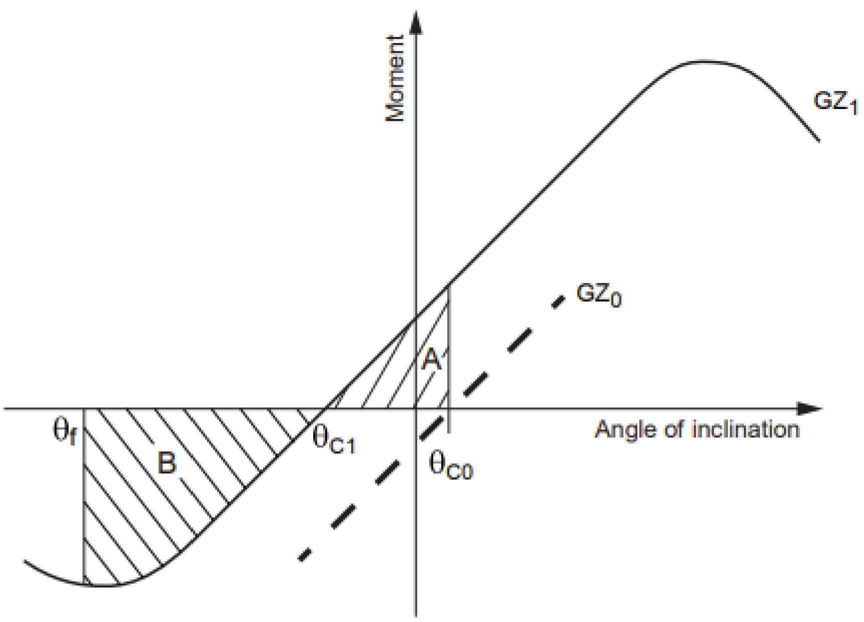

Figure 2.2 Righting moment curve after sudden loss of lifted load [5]

2.3 American Bureau of Shipping Rules

American Bureau of shipping recommends special stability requirement for counter ballast ships along with general intact stability requirements.

2.3.1 General Intact Stability Requirements

Each ship should comply with the following requirements

a) The area under the righting arm curve from the angle of heel at equilibrium up the one of the following whichever comes first must be at least 0.080 meter-radians [6]
   - Angle of down flooding
   - The second intercept
   - 40 degrees
b) Down flooding point or the lowest portion of weather deck must not be submerged at the equilibrium heel angle.
c) Heel angle due to heeling moment and effect of beam wind should not exceed the maximum heel angle of crane manufacturer.

2.3.2 Additional Intact Stability Requirements for Counter Ballast Vessels

It is recommended to correct for the increase in the vertical center of gravity due to the load. The following are to be complied with the vessel at the maximum allowable center of gravity in order to provide sufficient stability in case of sudden loss of hook load during lifting operation.

a) The area under the GZ curve between first intercept up to the angle of down flooding or second intercept whichever is less or comes first, must be more than 30% of Area A_2 as mentioned in the figure (2.3). Mathematical expression can be seen in equation (2.7)

$$A_1 \geq 1.3\, A_2 \tag{2.7}$$

b) Angle of first intercept between the GZ curve after the loss of crane load and the maximum allowable counter ballast lever curve must be less than 15^o [6].

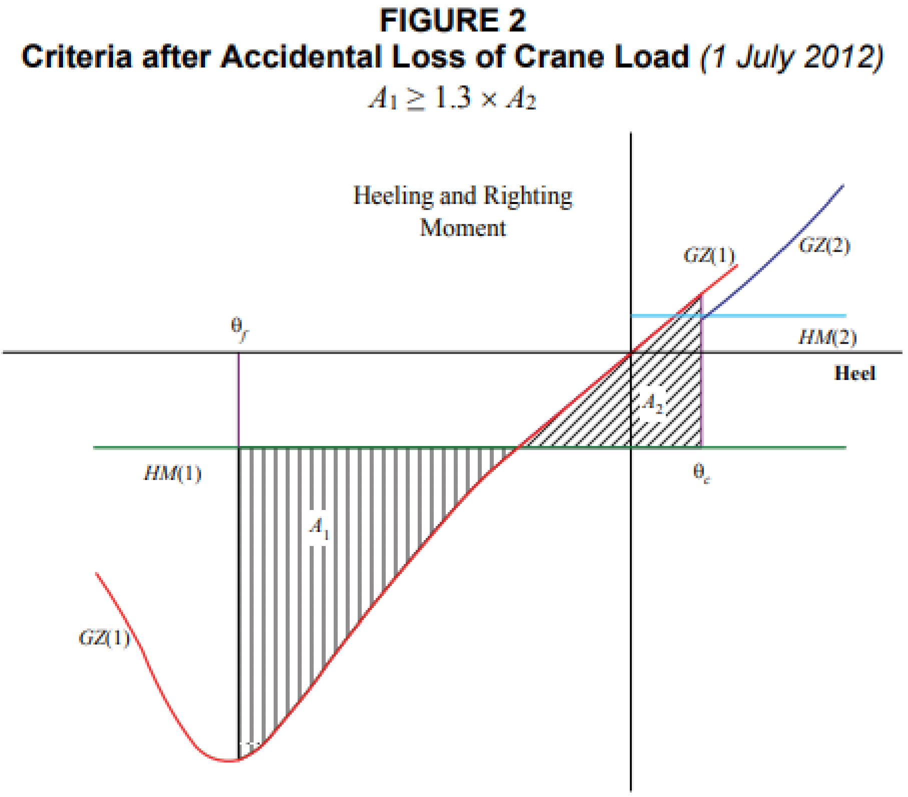

Figure 2.3 Criteria after accidental load loss of crane [6]

Where in figure (2.3)

$GZ(1)$ = Righting moment curve corresponding to the vessel without hook load.

$GZ(2)$ = Righting moment curve corresponding to the vessel with hook load.

$HM(1)$ = Curve of heeling moment due to the heeling moment of the counter-ballast without hook load.

$HM(2)$ = Curve of heeling moment due to the heeling moment of the counter-ballast with hook load.

$\theta_f =$ Angle of down flooding

$\theta_c =$ Angle of static equilibrium due to the combined hook load and counter-ballast heeling moment.

3. THEORETICAL BACKGROUND

To be able to develop a mathematical model, theory of motion analysis has been summarized. In the given case study, the most critical ship motion is roll motion of the ship due to sudden loss of the cargo load. The coupling of roll, pitch and heave motion can be done on the latter stages.

3.1 Ship Motions

The ship generally has six degrees of freedom which are known as

1. Surge (Translational motion in longitudinal axis)
2. Sway (Translational motion in transversal axis)
3. Heave (Translational motion in vertical axis)
4. Roll (Rotational motion around longitudinal axis)
5. Pitch (Rotational motion around transversal axis)
6. Yaw (Rotational motion around vertical axis)

The motion depiction can also be seen in figure (3.1).

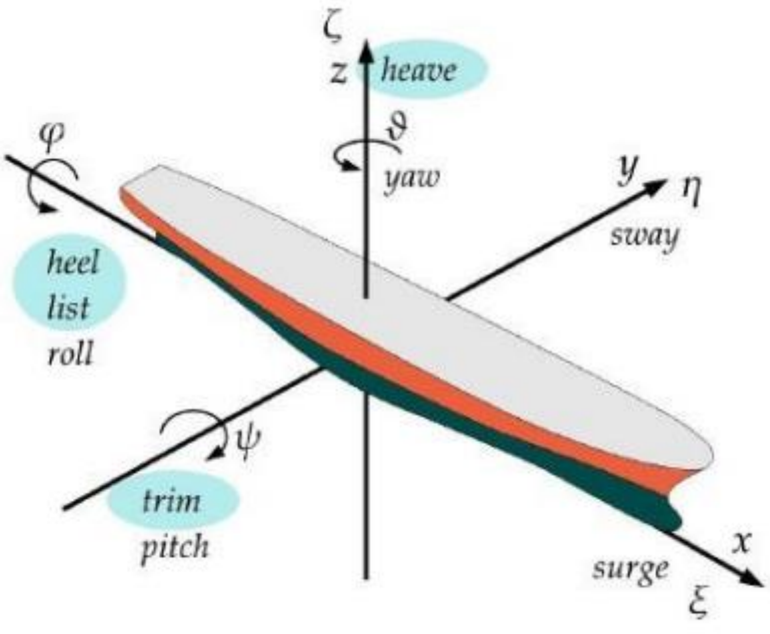

Figure 3.1 Depiction of ship motions [7]

The case will be considered with zero forward speed since the dynamic analysis is performed on the static ship i.e., zero forward motion.

3.2 Coupling of Ship Oscillations

Coupling of ship motions can be divided into the types of coupling [7].

1. Hydrostatic coupling
2. Hydrodynamic coupling
3. Gyroscopic Coupling

3.2.1 Hydrostatic Coupling

Due to the change in ship draught, vertical hydrostatic centre shift towards the stern which causes the negative pitch angle and therefore, heave oscillation causes the pitch and the same holds for the reverse case. This type of coupling is called hydrostatic coupling.

3.2.2 Hydrodynamic Coupling

If the ship is moving in the transversal direction, i.e., sway motion, a yaw moment will produce because the ship is asymmetric with respect to amidships. That means, sway and yaw motions are coupled, and this type of coupling is called hydrodynamic coupling.

3.2.3 Gyroscopic Coupling

Gyroscopic coupling happens if a transverse force acts on the ship during roll motion and it starts to perform the pitch motion. In ship gyroscopic coupling doesn't have a significant effect and can be neglected to avoid the complexity of motion equations.

3.3 Components of Ship Equation of Motion

3.3.1 Added Mass

During the ship oscillation, the energy required to accelerate the added mass must be considered to accurately estimate the response of the ship. Roughly the added mass is almost equal to the 1/3 or ¼ of the ship mass [7]. Since the density of water is not negligible as compared to air density, the added mass contribution cannot be ignored. Integration of the added mass of ship frame in longitudinal direction can be done for the slender bodies but roll added mass cannot be determined using this method because the viscosity is more dominant in this case and hence, added mass depends on the frequency ω.

3.3.2 Damping Coefficients

Hydrodynamic damping arises due to two factors. First is viscous damping that is proportional to square root of viscous velocity. This damping is considered as zero with respect to the given problem because of zero ship velocity at the time of lifting operations. The most important contribution to the damping is due to radiated waves. These waves are formed during the oscillations of the ship in calm water a part of kinetic energy of the ship radiate into the wave formation on the surface of water. Thus, damping occurs and ship oscillations decay [7].

3.3.3 Hydrostatic Force

During the equilibrium state of the ship, ship weight is balanced by the opposite hydrostatic lift force. During the perturbation, the equilibrium condition violates and a change in hydrostatic force occurs. If the perturbation is small, then hydrostatic force in heave can be calculated from equation (3.1)

$$\Delta F_{\zeta} = \left\{-\rho g \int \cos(nz)\, zdS\right\}_{T+\zeta} - \left\{-\rho g \int \cos(nz)\, zdS\right\}_{T} = -\rho g A_{wp}\zeta \qquad (3.1)$$

Where

ζ = Increment of ship draught
T = Draught at equilibrium state
A_{wp} = Water plane area

Similarly, in case of roll or pitch motion, corresponding hydrostatic moment can be determined by the equation (3.2) and (3.3) respectively.

$$M_{\varphi} = -\rho g \nabla_{o} GM_{\gamma}\varphi \tag{3.2}$$

$$M_{\psi} = -\rho g \nabla_{o} GM_{L}\psi \tag{3.3}$$

Where

φ = Roll angle
ψ = Pitch angle
GM_{γ} = Transverse metacentric height
GM_{L} = Longitudinal metacentric height
∇_{o}= Ship displacement

3.4 Ship Free Oscillation

The studied case has zero ship speed during the lifting of offshore turbine load. Therefore, if any perturbation occurs due to loss of load will have a motion in Heave, roll and pitch. Other motions i.e., yaw, surge and sway didn't arise in calm water conditions. The free ship oscillation in calm water can be represented as mentioned in the equation (3.4)

$$(m_{33} + A_{33})\zeta'' + B_{33}\zeta' + \rho g A_{wp}\zeta = 0$$

$$(I_{roll} + A_{44})\varphi'' + B_{44}\varphi' + \rho g \nabla_{o} GM_{\gamma}\varphi = 0 \tag{3.4}$$

$$(I_{pitch} + A_{55})\psi'' + B_{55}\psi' + \rho g \nabla_{o} GM_{L}\psi = 0$$

To solve these equations, first normalization of the above-mentioned equation is done and can be represented as equation (3.5)

$$\zeta'' + 2\vartheta_{\zeta}\zeta' + \omega_{\zeta}^{2}\zeta = 0$$

$$\varphi'' + 2\vartheta_{\varphi}\varphi' + \omega_{\varphi}^{2}\varphi = 0 \tag{3.5}$$

$$\psi'' + 2\vartheta_{\psi}\psi' + \omega_{\psi}^{2}\psi = 0$$

Where $\vartheta_\zeta, \vartheta_\varphi$ and ϑ_ψ are damping co-efficient as described in equation (3.6)

$$\vartheta_\zeta = \frac{B_{33}}{2(m_{33} + A_{33})}$$

$$\vartheta_\varphi = \frac{B_{44}}{2(I_{roll} + A_{44})} \tag{3.6}$$

$$\vartheta_\psi = \frac{B_{55}}{2(I_{pitch} + A_{55})}$$

ω_ζ, ω_φ and ω_ψ represents eigen frequencies of non-damped oscillations and can be described as mentioned in equation (3.7).

$$\omega_\zeta = \sqrt{\frac{\rho g A_{wp}}{m_{33} + A_{33}}}$$

$$\omega_\varphi = \sqrt{\frac{\rho g \nabla_o GM_\gamma}{I_{roll} + A_{44}}} \tag{3.7}$$

$$\omega_\psi = \sqrt{\frac{\rho g \nabla_o GM_L}{I_{pitch} + A_{55}}}$$

Free ship oscillation equation of mentioned in (3.4) are fully independent from each other and can be written in generalized as mentioned in equation (3.8).

$$\varphi'' + 2\vartheta\varphi' + \omega^2\varphi = 0 \tag{3.8}$$

In case of homogeneous second order differential equation, Corresponding φ can be represented as mentioned in equation (3.9).

$$\varphi = Ce^{pt} \tag{3.9}$$

By substituting equation (3.9) into equation (3.8) yields equation (3.10)

$$p^2 + 2\vartheta p + \omega^2 = 0 \tag{3.10}$$

Solution to the equation (3.10) is represented in equation (3.11)

$$p_{1,2} = \vartheta \pm \sqrt{\vartheta^2 - \omega^2} \tag{3.11}$$

Considering the system has no damping, then solution yields equation (3.12)

$$p_{1,2} = Ce^{i\omega t} = C\cos\omega t + i\sin\omega t \tag{3.12}$$

That means the system will oscillate with constant amplitude and frequency ω but in case of the real vessel damping phenomenon would occur because of radiation wave generation during the oscillations. In case of damping, equation (3.13) describes the solution of the system in free ship oscillation mode.

$$\begin{aligned} p_1 &= -\vartheta + i\sqrt{\omega^2 - \vartheta^2} = -\vartheta + i\bar{\omega} \\ p_2 &= -\vartheta - i\sqrt{\omega^2 - \vartheta^2} = -\vartheta - i\bar{\omega} \end{aligned} \tag{3.12}$$

In turn, the solution of a differential equation is described in equation (3.13)

$$\varphi = Ce^{-\vartheta t}(\cos\bar{\omega}t \pm i\sin\bar{\omega}t) \tag{3.13}$$

Equation (3.13) indicates that the response decay amplitude governs by $e^{-\vartheta t}$ and rate of decay is dependent on damping coefficient ϑ. $\bar{\omega}$ is damped oscillation frequency that is less than the undamped natural frequency by a factor of ϑ^2 and can be represented as equation (3.14).

$$\bar{\omega} = \sqrt{\omega^2 - \vartheta^2} < \omega \tag{3.14}$$

It can be concluded from the equation (3.15) that damped oscillation time period is more as compared to undamped oscillation.

$$T = \frac{2\pi}{\bar{\omega}} \tag{3.15}$$

From the above equations, we can calculate Eigen frequency, damped oscillation frequency and oscillation time period. A summary of these characteristics for heave, roll and pitch motion is represented in table (3.1).

Table 3.1 Roll, pitch and heave frequency, damped frequency, and time period relations [7]

Oscillation	Eigen frequency	Damped oscillation frequency	Oscillation time period
Heave	$\omega_\zeta = \sqrt{\frac{\rho g A_{wp}}{m_{33} + A_{33}}}$	$\bar{\omega}_\zeta = \sqrt{\vartheta_\zeta^2 - \omega_\zeta^2}$	$T_\zeta = 2\pi\sqrt{\frac{m_{33} + A_{33}}{\rho g A_{wp}}}$
Roll	$\omega_\varphi = \sqrt{\frac{\rho g \nabla_o GM_\gamma}{I_{roll} + A_{44}}}$	$\bar{\omega}_\varphi = \sqrt{\vartheta_\varphi^2 - \omega_\varphi^2}$	$T_\varphi = 2\pi\sqrt{\frac{I_{roll} + A_{44}}{\rho g \nabla_o GM_\gamma}}$
Pitch	$\omega_\psi = \sqrt{\frac{\rho g \nabla_o GM_L}{I_{pitch} + A_{55}}}$	$\bar{\omega}_\psi = \sqrt{\vartheta_\psi^2 - \omega_\psi^2}$	$T_\psi = 2\pi\sqrt{\frac{I_{pitch} + A_{55}}{\rho g \nabla_o GM_L}}$

The damping ratio can be characterized by the logarithmic decrement that can be calculated by the ratio of oscillation time period at the instance t to the instance time t + T. It can be represented in logarithmic form as mentioned in the equation (3.17).

$$\frac{\varphi(t)}{\varphi(t+T)} = \frac{e^{-\vartheta t}}{e^{-\vartheta(t+T)}} = e^{\vartheta T} \tag{3.16}$$

$$ln\frac{\varphi(t)}{\varphi(t+T)} = ln\frac{e^{-\vartheta t}}{e^{-\vartheta(t+T)}} = ln e^{\vartheta T} = \vartheta T = \frac{2\pi\vartheta}{\bar{\omega}} = 2\pi\bar{\vartheta} \tag{3.17}$$

Where $\bar{\vartheta} = \frac{\vartheta}{\bar{\omega}}$ and called as refered damping factor. Therefore, oscillation amplitude decay is 2π times referred damping factor. This statement is valid only if $\omega \gg \vartheta$.

Another conclusion can be drawn from the table (3.1) that oscillation time period is dependent on transversal metacentric height and longitudinal metacentric height for the case of roll and pitch motion respectively. The greater the value of metacentric height, the less is the time period of oscillation and vice versa. Also, referring to the table (3.2), smaller the metacentric height, larger the oscillation time period and less oscillations are necessary to decay the oscillation amplitude. Hence, time of decay doesn't depend on metacentric height, but only on damping.

Table 3.2 Referred damping factors in heave, roll and pitch motion [7]

Oscillation	**Referred damping factor**
Heave	$\bar{\vartheta}_{\zeta} = \dfrac{B_{33}}{2\sqrt{(m + A_{33})\rho g A_{wp}}}$
Roll	$\bar{\vartheta}_{\varphi} = \dfrac{B_{44}}{2\sqrt{(I_{44} + A_{44})\rho g \nabla_{o} GM_{\gamma}}}$
Pitch	$\bar{\vartheta}_{\psi} = \dfrac{B_{55}}{2\sqrt{(I_{55} + A_{55})\rho g \nabla_{o} GM_{L}}}$

4. MATHEMATICAL MODELING

To determine the response of the ship due to sudden load loss of cargo in time domain, a coupled motion analysis in pitch, heave and roll motion must be determined. To determine the response of the system, first analysis with the single degree of freedom is accessed and coupled to get the overall response. The major challenge is to determine or estimate the added mass, 2nd moment of inertia, damping and restoring coefficients. These values will be estimated based on analytical formulas and the literature available. Therefore, a sensitivity analysis will be carried out by varying the added mass estimated value and corresponding maximum inclination or displacement will be checked accordingly.

The time domain simulation is carried out using Python coding platform, whereas for better results fourth order Range Kutta method [8] is selected to get precise and accurate results. However, Range Kutta method can fail if the differential equation is stiff except if the time step taken is too small [9]. Therefore, these results are also compared with odeint solver and resulted in equivalent solution which also confirms the integrity of the code and method implemented.

4.1 Single Degree of Freedom Roll Motion

The roll motion can be described by a linear homogeneous differential equation consisting of inertia, damping and restoring moment of the ship as shown in equation (4.1)

$$(I_{roll} + I'_{roll})\varphi'' + B\varphi' + C\varphi = 0 \tag{4.1}$$

Where

I_{roll} = Mass roll moment of inertia of the ship in loading condition after loss of hook load, in kNm-s^2

I'_{roll} = Added mass roll moment of inertia of the ship in loading condition after loss of hook load, in kNm-s^2

φ'' = Roll acceleration, in deg/s

φ' = Roll velocity, in deg/s

B = Linearized roll damping coefficient

C = Roll restoring moment, in kNm, as a function of roll angle

The above equation can be modified according to particulars of the ship. The ship is supposed to carry a 1500 ton of load and is upright at the initial stage by the addition of ballast. Then after load sudden loss of load happens that induce inclining moment and change in the center of gravity of the ship. Stepwise pictorial representation can be seen in figure (4.1-4.2)

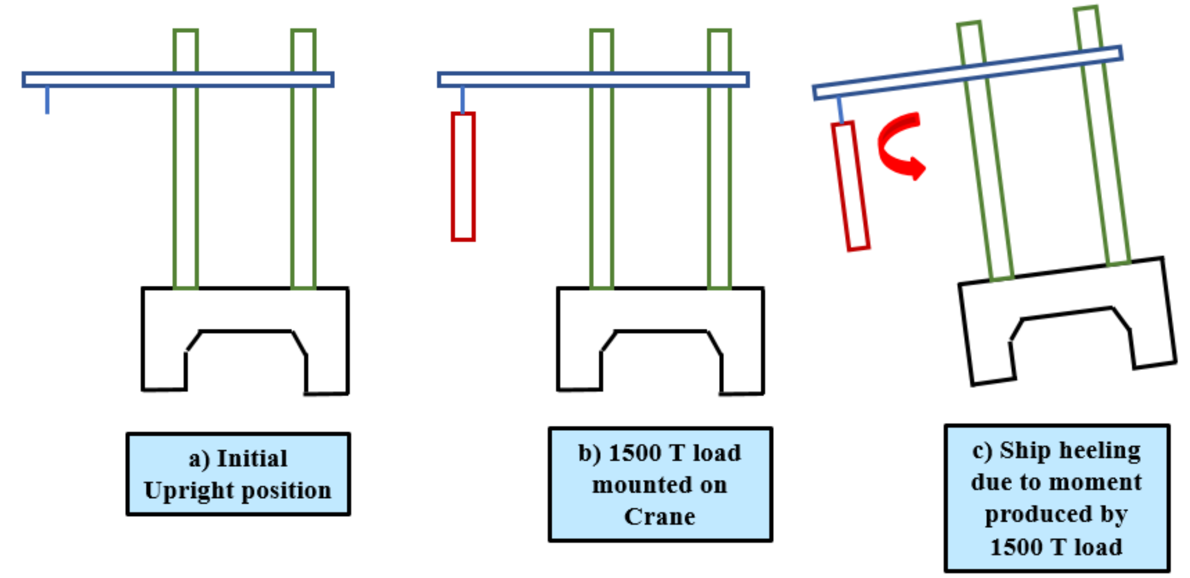

Figure 4.1 Depiction of ship motion in roll due to load loss (part a)

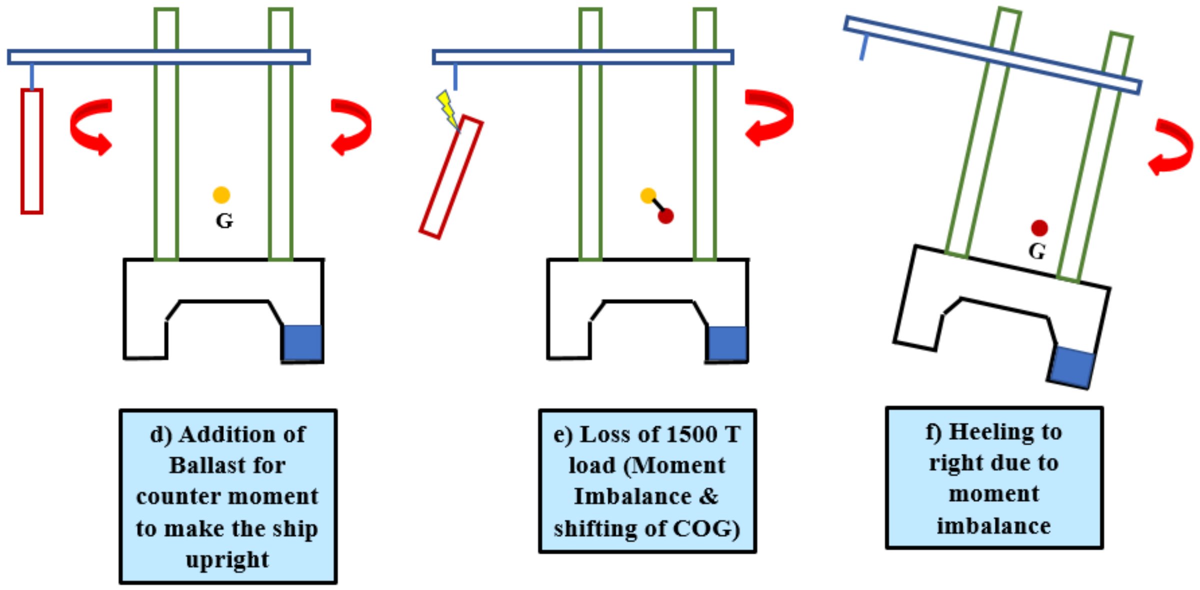

Figure 4.2 Depiction of ship motion in roll due to load loss (part b)

The modified nonhomogeneous equation of motion including the external inclining moment is given in equation (4.2)

$$(I_{roll} + I'_{roll})\varphi'' + B\varphi' + C\varphi = M_T \quad (4.2)$$

Where

$M_T =$ Inclining moment in transverse direction due to load loss

4.1.1 Determination of Added Mass Moment of Inertia in Roll Motion

To determine the added mass in roll motion a relation between vertical added mass can be established [3] as shown in equation (4.3)

$$\lambda_{33}{}^{d} = 0.85\frac{\pi}{4}\frac{\gamma}{g}Lb^2\frac{\alpha^2}{1+\alpha} \quad (4.3)$$

Where

$L =$ Length of demihull (m)

$b =$ beam of demi hull (m)

$\alpha =$ waterplane area coefficient

$\gamma =$ density of water (kg/m^3)

$\lambda_{33}{}^{d}$ = The water added mass of a demihull in the vertical direction (kg/ms^2)

Considering the wave interference between the demihulls, the heave added mass can be estimated by equation (4.4)

$$\lambda_{\zeta} = 1.6\lambda_{33}{}^{d} \quad (4.4)$$

Now the added roll moment of inertia can be estimated from equation (4.5)

$$\lambda_{44} = 2.5\lambda_{\zeta}k_d{}^2 \quad (4.5)$$

Where

$k_d =$ demihull spacing from its centerline to the catamaran's longitudinal centerline (m)

4.1.2 Determination of Mass Moment of Inertia in Roll

The mass moment of inertia of the ship in roll can be estimated by the analytical formula provided by Lian Yung, 2019 [3]. The mass moment of inertia of each demihull around x-axis can be determined by the equation (4.6)

$$I_x{}^d = \frac{D}{2g}\left(\frac{b^2\alpha^2}{11.4\delta} + \frac{H^2}{12}\right) \tag{4.6}$$

Where

H = Depth of demihull (m)

D = Displacement of demihull (kg)

δ = Block Coefficent

b = Breadth of demihull

α = waterplane area coefficient

The mass moment of inertia of catamaran ship in roll motion, including demi hulls and spacing can be estimated by equation (4.7).

$$I_x = \frac{D}{12g}\left(B^2 + 4{z_g}^2\right) \tag{4.7}$$

Where

B = Catamaran breadth (m)

z_g = Center of gravity height above baseline (m)

4.1.3 Determination of Damping Coefficient in Roll Motion

The damping coefficient is responsible for the decay of roll motion and can be estimated by using equation (4.8).

$$N_\theta = \nu_\theta + \nu_\zeta {k_d}^2 \tag{4.8}$$

Where

ν_θ = Damping coefficient for angular oscillation of demihull

The term ν_ζ is considered zero because the ship analysis is being taken at the zero ship speed. Whereas the ν_θ can be determined equation (4.9).

$$\nu_\theta = 0.1\sqrt{\frac{D}{2}h\left({I_x}^d + {\lambda_{44}}^d\right)} \tag{4.9}$$

Where ${I_x}^d$ and ${\lambda_{44}}^d$ can be calculated from equation (4.6) and (4.10) respectively.

$${\lambda_{44}}^d = 2{\lambda_{33}}^d k'^2 \tag{4.10}$$

Where

${\lambda_{33}}^d$ can be determined by the equation (4.3) and k' is the radius of moment of inertia of added mass about x-axis of demihull; in general, it can be taken as b/4, where b is the beam of the demihull

4.1.4 Determination of Restoring Coefficient and Inclining Moment in Roll Motion

The restoring coefficient in roll motion can be calculated in from the equation (4.11)

$$C = GM_T D \sin\theta \tag{4.11}$$

Where

GM_T = Tranverse Metacentric height (m)

D = Displacement of ship (kg)

θ = Roll angle

The inclining moment generated by the 1500 T load is equal to the righting moment made by addition of ballast in order to make the ship upright. When load will loss, the righting moment produced by the ballast addition will work as an inclining moment of ship and can be calculated from equation (4.12)

$$M_T = 1500 * 1000\, r \tag{4.12}$$

Where

M_T = Inclining moment in roll motion (kg m)

r = Righting arm of 1500 T load (m)

4.1.5 Roll Motion Time Domain Analysis

The roll motion analysis is carried out by starting simulation just before 1500 ton load loss. The nonhomogeneous second order differential equation is solved by taking roll angle and rate of change of roll angle equal to zero as an initial condition. The maximum roll angle of the ship is 5.904^o.

Maximum Roll angle = 5.904^o

Oscillation Time Period = 16.286 sec

Natural Frequency = 0.386 rad/sec

Equilibrium Position = 2.952^o

The maximum roll angle achieved is quite less as compared to the angle of vanishing stability i.e., maximum roll angle achieved is 5.904^o whereas, angle of vanishing stability is 40.8 ^{o}. However, the angle of down flooding of ship is unknown but still it can be accessed that determined maximum roll angle would have a quite big from the angle of down flooding of ship. Hence, it complies with the DNV rules and results are satisfactory.

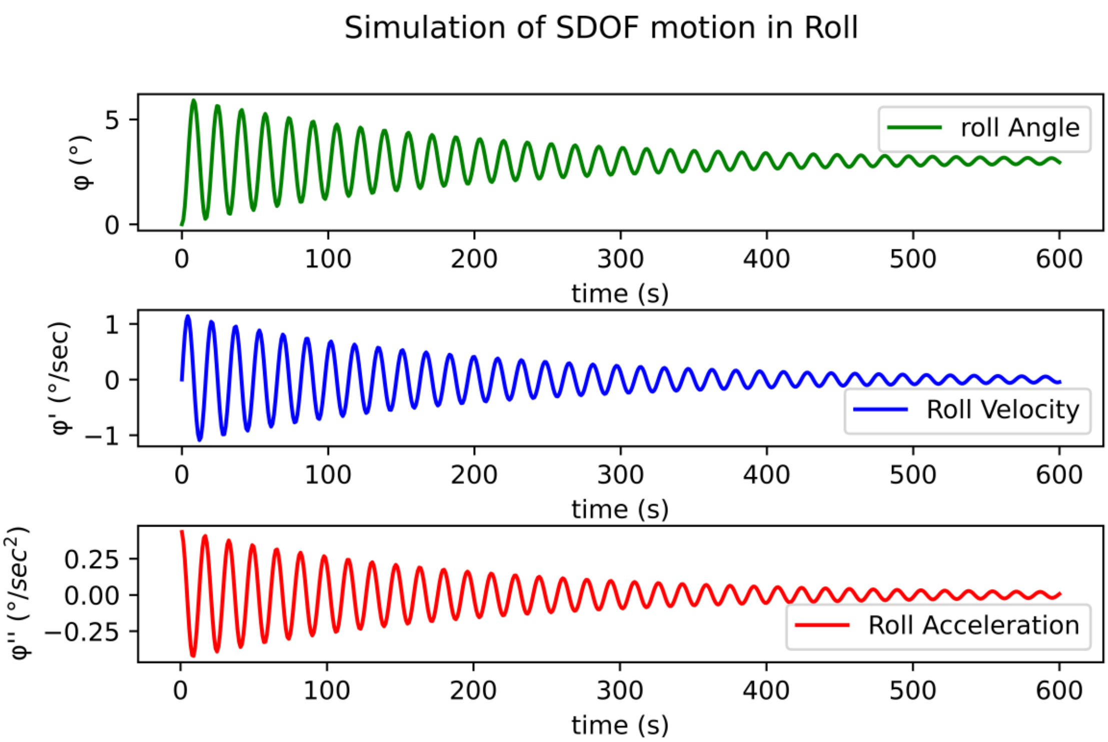

Figure 4.3 Simulation of single degree of freedom motion in roll using analytical damping

The damping estimation can also be done with the guided rules of DNV [4] that recommends using damping coefficient in roll motion as 2% of the critical damping of the system in single degree of freedom. The maximum roll angle remains the same as previous analysis. However, the damping rate is faster than the previous analysis because calculated damping coefficient value is higher according to DNV rule as compared to calculated from analytical formula. Response of system by taking 2% of critical damping can be seen in figure (4.4).

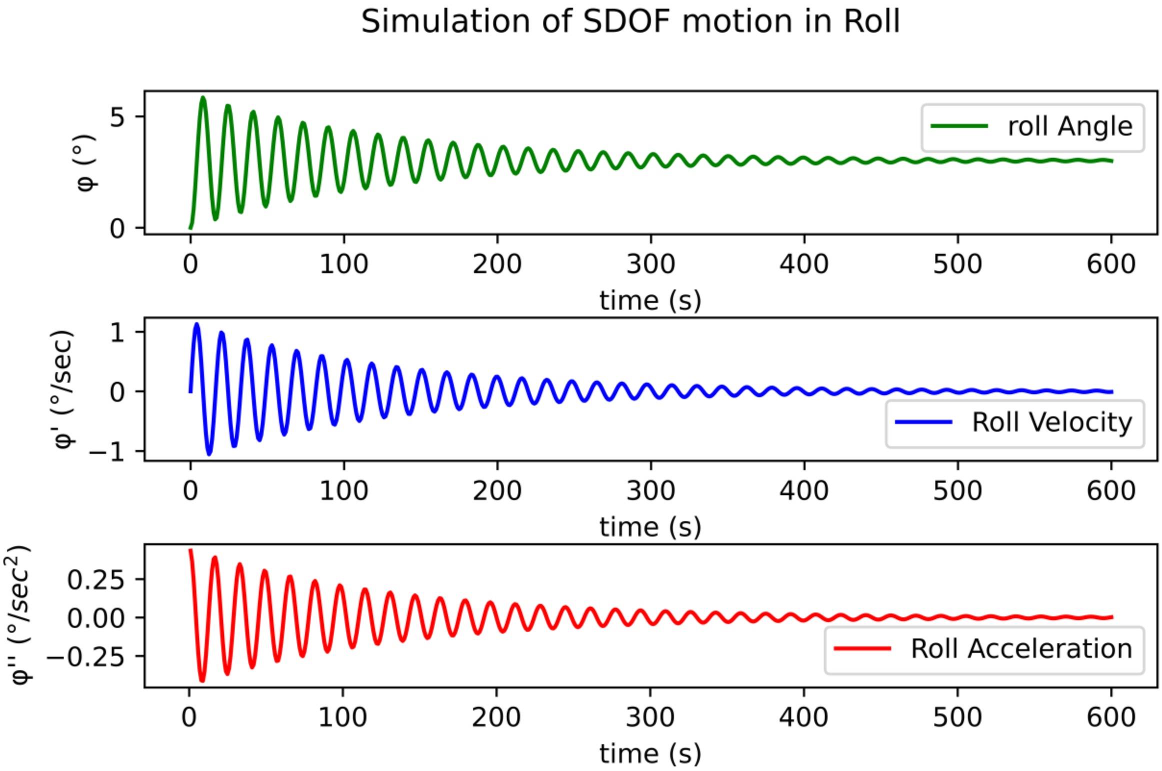

Figure 4.4 Simulation of single degree of freedom motion in roll using 2% of critical damping

4.1.6 Roll Motion Sensitivity Analysis

To evaluate the effect of change in added mass, a sensitivity analysis is carried out by varying the percentage of added. The added mass is varied from 80 percent of the estimated value as a lower bound while 120 percent of estimated value taken as an upper bound. The graph shows, with the increase in added mass the roll angle increases till 94% of the estimated value. Thereafter, a reverse effect can be seen. This behaviour can be evaluated based on the fact that with lower added mass, the acceleration due to load loss is more dominant than the inertia of the ship till 94% of estimated value. Thereafter, the inertia of ship is more dominant as compared to acceleration generated by 1500 Ton load loss. Maximum roll angle at 94% of the estimated added mass value is 5.9052^{o}.

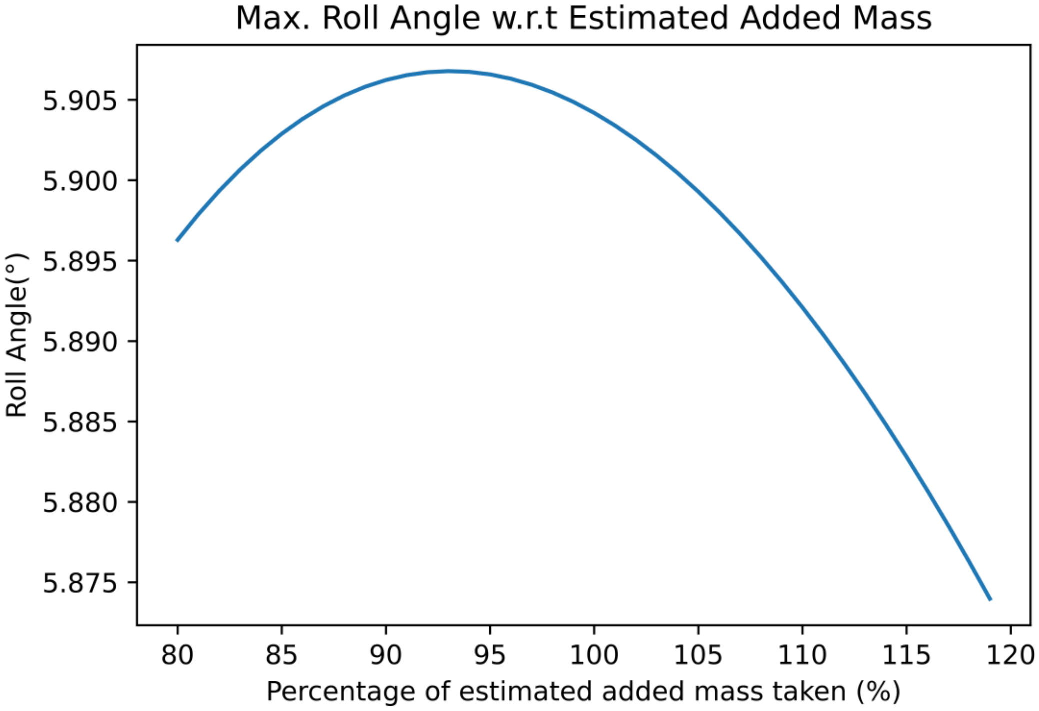

Figure 4.5 Sensitivity analysis of maximum roll angle with respect to added mass

Response of system by considering 94% of estimated added mass can be seen in figure (4.6).

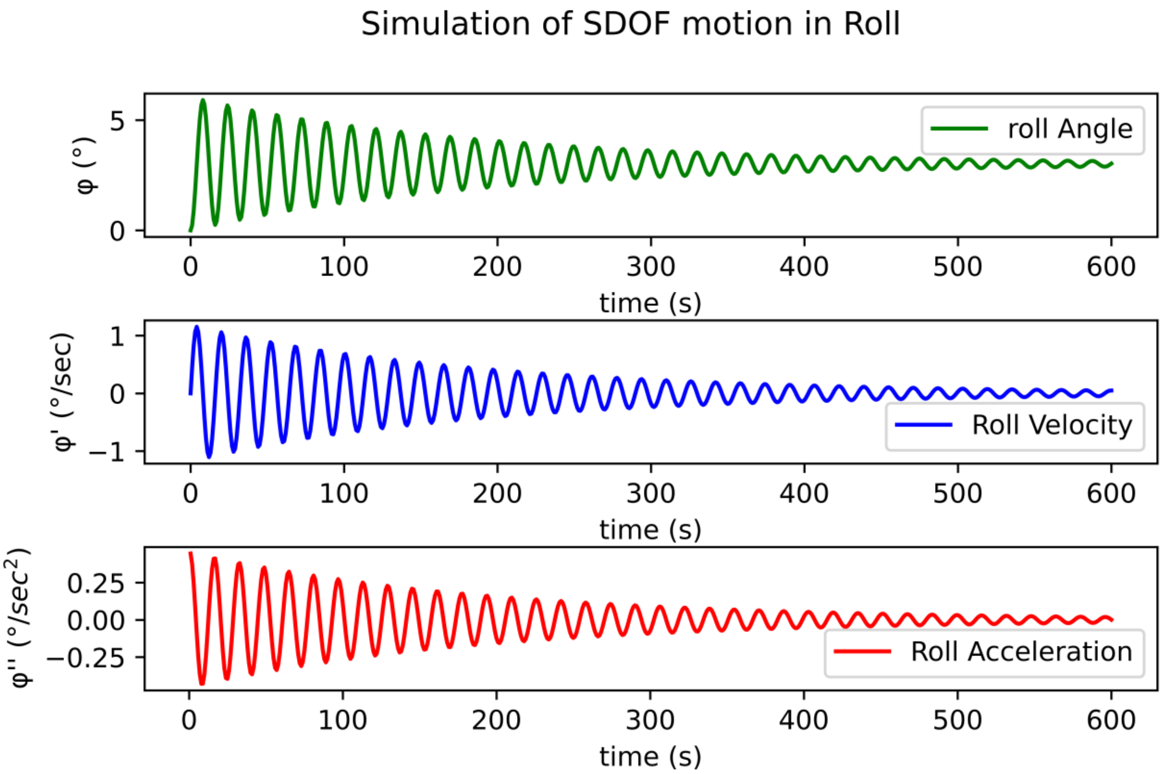

Figure 4.6 Response of ship in roll at 94% of estimated added mass

4.2 Single Degree of Freedom Pitch Motion

The pitch motion can be described by a linear homogeneous differential equation consisting of inertia, damping, and restoring moment of the ship as shown in equation (4.13)

$$(I_{pitch} + I'_{pitch})\psi'' + B\psi' + C\psi = 0 \tag{4.13}$$

Where

I_{pitch} = Mass roll moment of inertia of the ship in loading condition after loss of hook load, in kNm-s^2

I'_{pitch} = Added mass roll moment of inertia of the ship in loading condition after loss of hook load, in kNm-s^2

ψ'' = Pitch acceleration, in deg/s

ψ' = Pitch velocity, in deg/s

B = Linearized roll damping coefficient

C = Roll restoring moment, in kNm, as a function of roll angle

Similar to roll motion, the above equation can be modified as per given problem statement. The ship is supposed to carry a 1500 ton of load and is upright at the initial stage by the addition of ballast. Then after sudden loss of load happens that induce inclining moment and change in the center of gravity of the ship. The modified equation, including inclining moment is shown in equation (4.14).

$$(I_{pitch} + I'_{pitch})\psi'' + B\psi' + C\psi = M_L \tag{4.14}$$

Where

M_L = Inclining moment in longitudinal direction due to load loss

4.2.1 Determination of Added Mass Moment of Inertia in Pitch Motion

Added mass moment of inertia in pitch motion is estimated on the basis of empirical formula provided by Liang Yun 2019 in the book "High speed catamarans and multihull'' [3]. The empirical formula is a function of water plane area co-efficient, density of water, length, and beam of demi hull as shown in equation (4.15).

$$I'_{pitch} = 0.055\frac{\gamma}{g}\, b^2L^3\,\frac{\alpha^2}{(3-2\alpha)(3-\alpha)} \tag{4.15}$$

Where

$\gamma =$ Density of water in kg/m^3

$g =$ Gravitational acceleration in m/s^2

$b =$ Beam of demi hull in m

$L =$ Length of demi hull in m

$\alpha =$ Water plane area coefficient of demi hull

4.2.2 Determination of Mass Moment of Inertia in Pitch Motion

Added mass moment of inertia in pitch motion is estimated based on empirical formula shown in equation (4.16). The mass moment of inertia in pitch motion is a function of water plane area coefficient, displacement, and length of the demi hull.

$$I_{pitch} = 0.07\frac{\alpha}{g}\, DL^2 \tag{4.16}$$

Where

$D =$ Displacement of ship in kg

$g =$ Gravitational acceleration in m/s^2

$L =$ Length of demi hull in m

$\alpha =$ Water plane area coefficient of demi hull

4.2.3 Determination of Damping Coefficient in Pitch Motion

The most used method for determination of damping coefficient is to analyze a decay test data and obtain the logarithmic decrement of the consecutive oscillations in order to examine the behavior of the decay process. This method requires that all the peaks including positive and negative peaks of the decay test should be recorded and thereafter, analyzed from the experimental time-domain record to find the damping coefficients [10].

However, Decay test is not carried out for the ship. Therefore, an alternative method is used to determine the damping coefficient of the ship. Contrikov's method is used to determine the estimated damping coefficient of catamaran type ship. This method will be discussed in detail in the section of the coupled motion analysis.

4.2.4 Determination of Restoring Coefficient and Inclining Moment in Pitch Motion

The restoring coefficient in pitch motion can be calculated in from the equation (4.17)

$$C = GM_L D \sin\phi \tag{4.17}$$

Where
$GM_T =$ Tranverse Metacentric height (m)
D = Displacement of ship (kg)
$\phi =$ pitch angle

The inclining moment generated by the 1500 T load is equal to the righting moment made by addition of ballast in order to make the ship upright. When load will loss, the righting moment produced by the ballast addition will work as an inclining moment of the ship and can be calculated from equation (4.18)

$$M_L = 1500 * 1000\, r \tag{4.18}$$

Where
$M_T =$ Incling moment in pitch motion (kg m)
$r =$ Righting arm of 1500 T load in longitudinal direction (m)

4.2.5 Pitch Motion Time Domain Analysis

The pitch motion analysis is carried out by starting just before the 1500-ton load loss. The nonhomogeneous second order differential equation is solved by taking pitch angle and rate of change of pitch angle equal to zero as an initial condition. The maximum pitch angle of the ship is -0.0034^o. The pitch angle or perturbation in pitch motion is quite small. Therefore, contribution of pitch to the other motion in coupled analysis would be small or negligible.
Maximum pitch angle = -0.0034^o

Oscillation Time Period = 9.08 sec

Natural Frequency = 0.691 rad/sec

Equilibrium Position = -0.0017 ^{o}

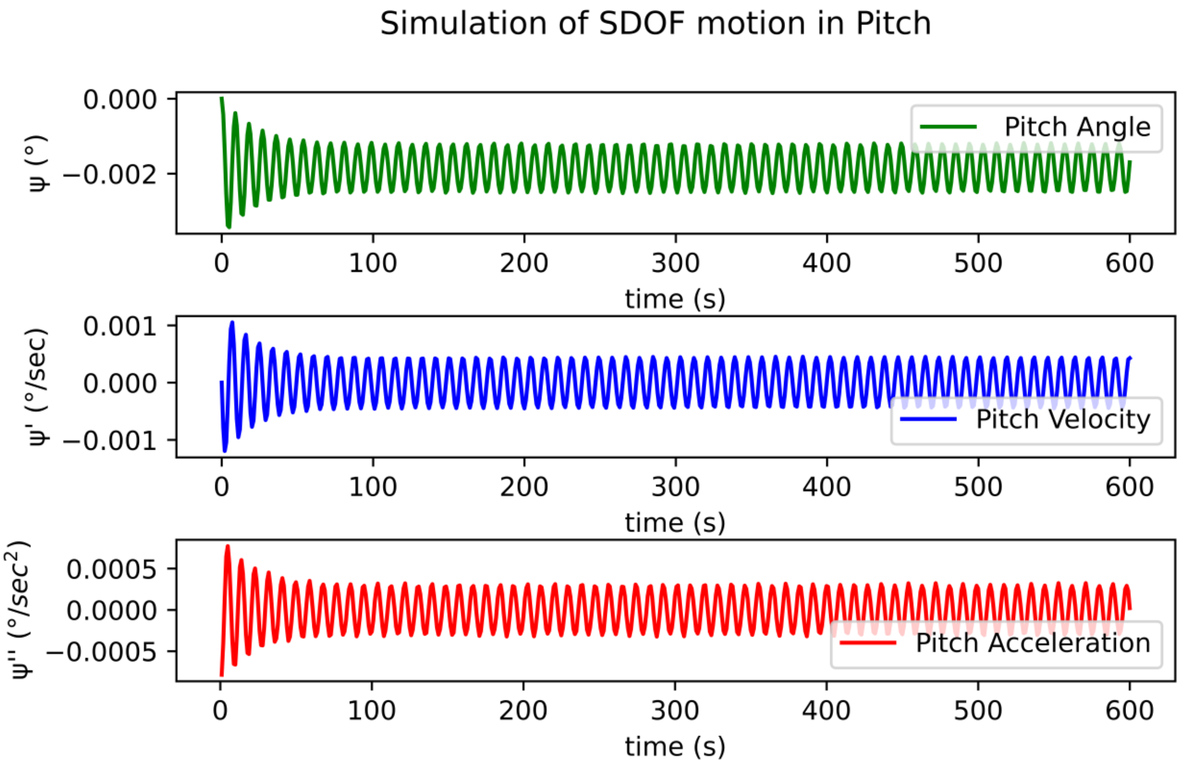

Figure 4.7 Simulation of single degree of freedom motion in pitch

4.2.6 Pitch Motion Sensitivity Analysis

In order to evaluate the effect of change in added mass, a sensitivity analysis is carried out by varying the percentage of added. The added mass is varied from 70 percent of the estimated value as a lower bound while 130 percent of estimated value taken as an upper bound. The graph shows, with the increase in added mass the pitch angle increases till 90% of the estimated value with lower rate. Thereafter, a sensitivity to added mass is enhanced effect. However, the effect on pitch angle can be neglected because overall pitch angle is quite less and also with the increase of 30% of the added mass to the ship, only 2% of change is observed.

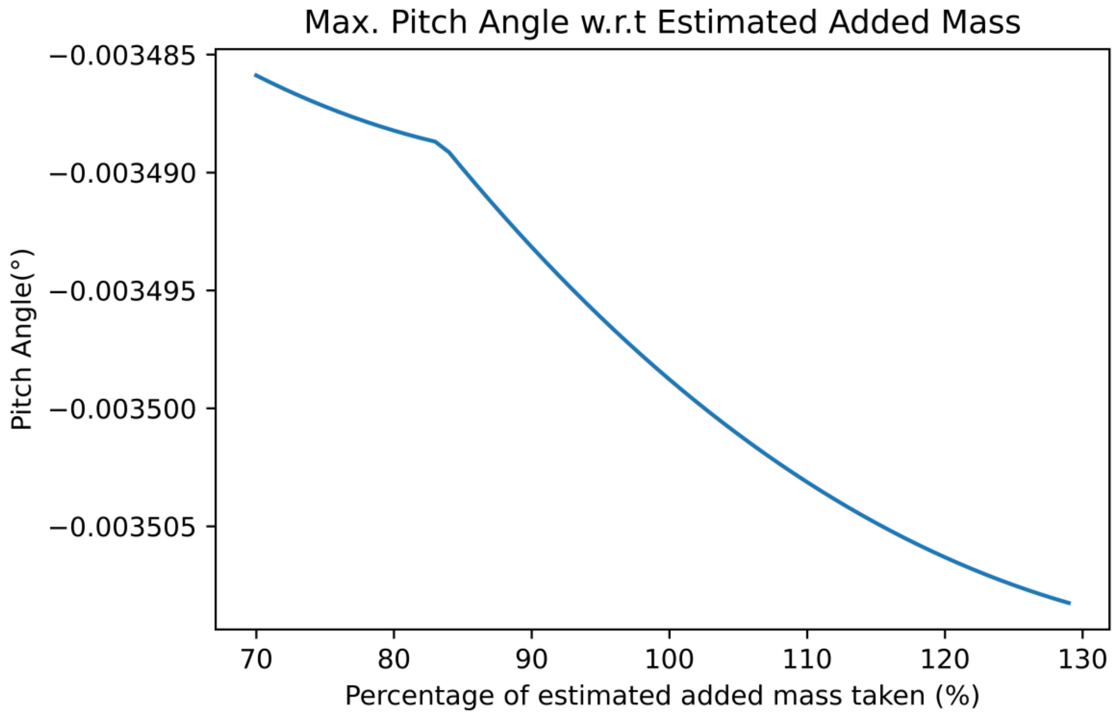

Figure 4.8 Maximum pitch angle sensitivity analysis with respect to estimated added mass

4.3 Single Degree of Freedom Heave Motion

Analysis of heave motion of ship due to sudden load loss has its own importance. Sudden load loss would cause inequality to load and buoyancy force. Therefore, heave oscillation can contribute inconvenience for ship operating staff. Heave motion analysis is also carried out just before load loss. Hence, initial conditions have been evaluated accordingly. Just after the load loss, ship is more submerged as compared to the downward load force requirement for equilibrium. Hence differential submerged depth is taken as the initial condition and calculated according to the equation (4.19)

$$\zeta = (\frac{1500 * 1000}{\rho * A_{wp}}) \tag{4.19}$$

Where

ρ = Density of water in kg/m^3

A_{wp} = Water plane area of ship in m^2

The equation of motion in heave can be represented as equation (4.20)

$$(m_{33} + m'_{33})\ddot{\zeta} + B_{33}\zeta' + C_{33}\zeta = 0 \quad (4.20)$$

Where

m_{33} = Mass of ship in kg

m'_{33} = Added mass of ship in heave motion in kg

B_{33} = Damping coefficient in heave direction in kg/s

C_{33} = Heave motion restoring coefficient in kg/s^2

4.3.1 Determination of Added Mass in Heave Motion

In order to estimate the added mass of ship in single degree of freedom motion in heave, equation (4.3) can be used to determine the λ_{33}^{d} and thenafter, to predict the interference effect of twin demihulls on wave making and, subsequently, on the catamaran water added mass, the additional added mass between the hulls is considered to be contained by an ellipse horizontally with maximum breadth at amidships and a parabola in the vertical plane, that can be represented as equation (4.21).

$$\Delta\lambda_{33} = 0.21\frac{\rho\pi TLb^3}{k_d^2}\frac{\alpha^3}{(1+\alpha)(1+2\alpha)} \quad (4.21)$$

Where

L = Lenght of demihull in m

α = Waterplane coefficient of demihull

k_d =Demihull spacing from its centerline to the catamaran's longitudinal centerline

The total added mass of ship in heave motion can be estimated by using equation (4.22)

$$\lambda_{33} = 2(\lambda_{33}^{d} + \Delta\lambda_{33}) \quad (4.22)$$

4.3.2 Determination of Damping Coefficient in Heave Motion

Like pitch motion, decay test is not carried out for the ship. Therefore, an alternative method is used to determine the damping coefficient of the ship. Contrikov's method is utilized to determine the estimated damping coefficient of catamaran type ship. This method will be discussed in detail in the section of coupled motion analysis.

4.3.3 Determination of Restoring Coefficient in Heave Motion

The restoring coefficient of catamaran ship in motion can be determined by using equation (4.23).

$$C_{33} = \rho g A_{wp} \tag{4.23}$$

4.3.4 Heave Motion Time Domain Analysis

The heave motion analysis is carried out by starting just before the 1500 ton load loss. The homogeneous second order differential equation is solved by taking initial displacement equal to 0.257 m due to load buoyancy imbalance and heave velocity equal to zero as an initial condition. The maximum heave displacement of the ship is 0.480 m. Obtained results clearly shows that in coupled motion analysis heave motion would definitely amplify the pitch motion.

Maximum heave displacement = 0.480 m

Oscillation Time Period = 8.95 sec

Natural Frequency = 0.70 Hz

Equilibrium Position = 0 m

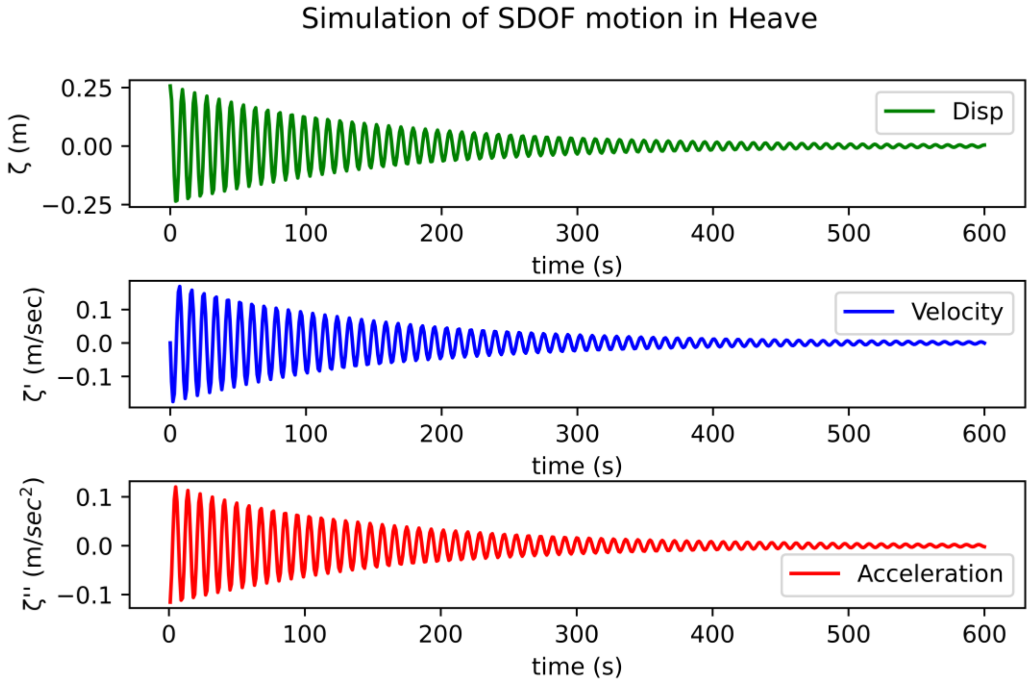

Figure 4.9 Simulation of single degree of freedom motion in heave

4.3.5 Heave Motion Sensitivity Analysis

To observe the effect of max heave displacement by varying the added mass value 30% from estimated value i.e., taking 70% value of estimated value as lower bound while 130% value of estimated value as upper bound. It can be seen from figure (4.10) that increased added mass from 70% of estimated value has an inverse effect on maximum displacement in heave motion till 104% of estimated added mass. Then after, Maximum displacement is directly proportional to the increased added mass estimated value. However, sensitivity of maximum heave displacement on added mass variation is not significant.

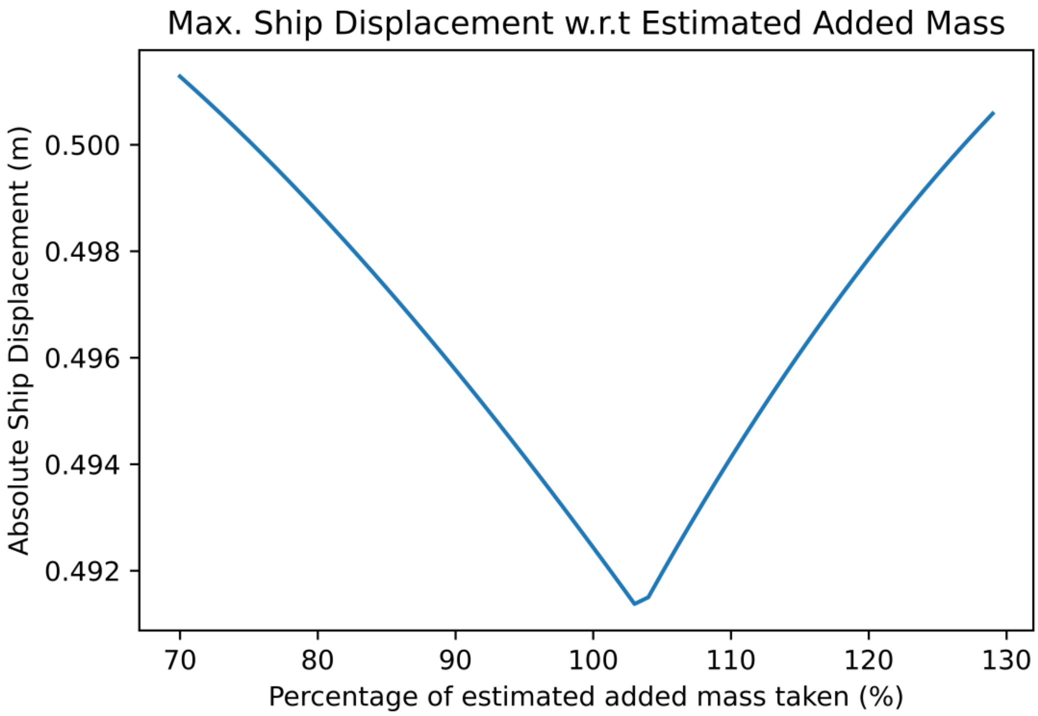

Figure 4.10 Maximum heave displacement sensitivity analysis with respect to estimated added mass

4.4 Coupled Motion Analysis

In this section analysis of coupled motion of ship due to sudden loss of load is examined. The coupled equation of motion in heave, roll and pitch can be formalized as mentioned in equation (4.24 - 4.26). To solve coupled equations, corresponding coupled added mass, damping and restoring coefficient must be determined. The approximate values can be determined by literature review of current available analytical formulas. This approximation can be used to estimate the coupled motion for preliminary analysis. Single degree of freedom analysis has revealed that pitch motion would have less contribution to the coupled motion. Moreover, roll motion is more critical in this case and it doesn't couple strongly with heave and pitch motions although heave and pitch do have strongly coupled with each other.

$$(I_{44}+A_{44})\ddot{\varphi}+b_{44}\dot{\varphi}+C_{44}\varphi+A_{43}\ddot{\zeta}+b_{43}\dot{\zeta}+C_{43}\zeta+\ A_{45}\ddot{\Psi}+b_{45}\dot{\Psi} + C_{45}\Psi=M_{44} \tag{4.24}$$

$$(m_{33}+A_{33})\ddot{\zeta}+b_{33}\dot{\zeta}+C_{33}\zeta+A_{34}\ddot{\varphi}+b_{34}\dot{\varphi}+C_{34}\varphi+\ A_{35}\ddot{\Psi}+b_{35}\dot{\Psi} + C_{35}\Psi=F_{33} \tag{4.25}$$

$$(I_{55}+A_{55})\ddot{\Psi}+b_{55}\dot{\Psi}+C_{55}\Psi+A_{54}\ddot{\varphi}+b_{54}\dot{\varphi}+C_{54}\varphi+A_{53}\ddot{\zeta}+b_{53}\dot{\zeta} + C_{53}\zeta\ =M_{55} \tag{4.26}$$

The coefficient b_{34}, b_{43}, b_{45}, b_{54}, A_{34}, A_{43}, A_{45}, A_{54}, C_{34}, C_{43}, C_{45} and C_{54} are equal to zero due to xz plane symmetry and location of coordination axis, no contribution from these coefficients will occurs in coupled motion. However, rest of remaining coefficients will have a contribution to the coupled motion and therefore, are being estimated.

To determine the values to the coupled mass matrix, damping matrix and restoring coefficient matrix, the Contrikov's method [3] is utilized to check the feasibility. Contrikov's estimated the regression coefficients for monohull ship that can be used for unknown values determination for initial design stage estimation. However, these coefficients are also applicable to catamaran type ship if the heel angle is small or negligible. Regression coefficient for the estimation of added mass damping and restoring coefficients are shown in table (4.1).

Table 4.1 Regression coefficients for added mass, damping, and restoring coefficient estimation [3]

Regression coefficients of a_{ijl}, b_{ijl}					
i, k, l	a_{ikl}	b_{ikl}	i, k, l	a_{ikl}	b_{ikl}
0,0,0	2.2102	6.5418	3,1,1	-78.8555	-74.6699
1,0,0	-11.0964	-28.2111	4,1,1	16.86	14.8633
2,0,0	27.3812	42.3544	0,2,1	-3.4149	-6.3947
3,0,0	-19.3812	-23.9681	1,2,1	24.0855	40.4441
4,0,0	4.4314	4.8685	2,2,1	-46.5159	-51.9258
0,1,0	-6.0134	-6.1183	3,2,1	30.394	27.2324
1,1,0	36.2004	38.7077	4,2,1	-6.4753	-5.2692
2,1,0	-80.3705	-51	0,0,2	0.4612	9.7853

3,1,0	56.9283	26.7735	1,0,2	-4.0683	-52.7271
4,1,0	-13	-5.1945	2,0,2	1.4359	81.6971
0,2,0	2.3129	2.4644	3,0,2	1.6235	-49.2273
1,2,0	-14.4029	-14.6185	4,0,2	-0.8189	10.2915
2,2,0	30.995	18.5078	0,1,2	-2.1326	-8.0976
3,2,0	-21.797	-9.2339	1,1,2	16.0122	66.585
4,2,0	4.9468	1.6751	2,1,2	-23.307	-87.4029
0,0,1	-2.8107	-15.1006	3,1,2	10.5205	48.0897
1,0,1	20.6434	77.402	4,1,2	-1.231	-9.8324
2,0,1	-37.3756	-116.744	0,2,2	0.8927	3.8935
3,0,1	23.1179	69.1435	1,2,2	6.3716	-26.1503
4,0,1	-4.8009	-14.2832	2,2,2	9.2028	33.3911
0,1,1	8.5736	15.272	3,2,2	-4.063	-17.6554
1,1,1	-61.3614	-106.6744	4,2,2	0.4603	3.4641
2,1,1	120.2025	138.9335			

Based on regression coefficient values in table (4.1), the coefficient $\bar{A}_3$ and C can be calculated from equation (4.27) and (4.28).

$$\bar{A}_3 = \sum_{i=0}^{4}\sum_{k=0}^{2}\sum_{l=0}^{2} a_{ikl}\xi_d^i \left(\frac{d}{B}\right)^k \sigma^l \tag{4.27}$$

$$C_o = \left({}^{d}/_{2}\right)^2 C = \sum_{i=0}^{4}\sum_{k=0}^{2}\sum_{l=0}^{2} b_{ikl}\xi_d^i \left(\frac{d}{B}\right)^k \sigma^l \tag{4.28}$$

Where ξ_d and σ can be calulated from equation (4.29) and (4.30) respectively.

$$\xi_d = \frac{\omega^2}{g} d \tag{4.29}$$

$$\sigma = \frac{S}{Bd} \tag{4.30}$$

Where

$S =$ Area of calulated section

$B =$ Width of calculated section

$d =$ Draught of calculated section

ω = Natural frequency

Damping and added mass value of 2D planes as a function of ship length can be calculated from equation (4.31) and (4.32) respectively.

$$N_z(x) = \frac{\rho g^2 \bar{A}_3^2}{\omega^3} \tag{4.31}$$

$$m_z(x) = \frac{1}{8}\rho\pi B^2 C \tag{4.32}$$

Integration of these functions along the length of ship gives corresponding values of damping coefficient, restoring coefficient, and added mass values of diagonal and non-diagonal values. Unfortunately, upon analysis, Contrikov's method doesn't well suit to the coupled mass, damping and restoring coefficients. Therefore, diagonal values for damping coefficients are estimated only and a general estimation is made based on the predetermined diagonal values and subsequently, a reasonable percentage of diagonal value is assigned to nondiagonal matrix values. The damping coefficient values are calculated using equation (4.33) and (4.34). Ship under observation is in the static condition and hence, velocity U is taken as zero.

$$b_{33} = \int_0^L \left[N_z(x) - U\frac{dm_z(x)}{dx} \right] dx \tag{4.33}$$

$$b_{55} = \int_0^L \left[N_z(x)x^2 - 2Um_z(x)x - Ux^2\frac{dm_z(x)}{dx} \right] dx \tag{4.34}$$

The coupled analysis has shown almost similar results as compared to single degree of freedom system. The coupled analysis on roll motion has negligible effect because pitch and heave motions don't couple strongly with the roll motion. However, Pitch motion has shown an increase in pitch angle at initial stage and then dampens out with time. This behaviour can be due to resonance phenomenon in pitch motion and effect of heave coupling to the pitch motion. On the other hand, Heave motion has a little or non-significant impact in the coupled response. During the coupled motion analysis, coupled added mass, damping coefficients, and restoring coefficients have been taken 30%, 60% and 100% of the corresponding diagonal values and ship response is evaluated. At 30% of values, maximum roll angle, pitch angle and heave displacement noted respectively 5.898°, -0.022 ° and 0.488 m. Upon increasing coupled coefficient values to 60%, corresponding maximum roll angle, pitch angle and heave displacement turns out 5.8986°, -0.0464 ° and 0.485 m respectively. Pitch motion has minor influence on the heave motion as amplitude dampening has increased by increasing coupled coefficient values. Therefore, maximum heave displacement value has shown negligible variation but effect of heave coupling on pitch has significant effect on maximum pitch angle. It can be seen more prominently in 100% of diagonal values taken as coupled coefficients where maximum roll angle, pitch angle and heave displacement turn out 5.8986°, -0.0808 ° and 0.480 m respectively. It is important to realize that no effect on roll motion is taken place during the analysis because roll motion is not strongly coupled with heave and pitch motion. Therefore, corresponding coupled values in roll motion are taken as zero. Numerical simulation of coupled motion at 30%, 60% and 100% coefficient values are shown in figure (4.11), (4.12) and (4.13) respectively.

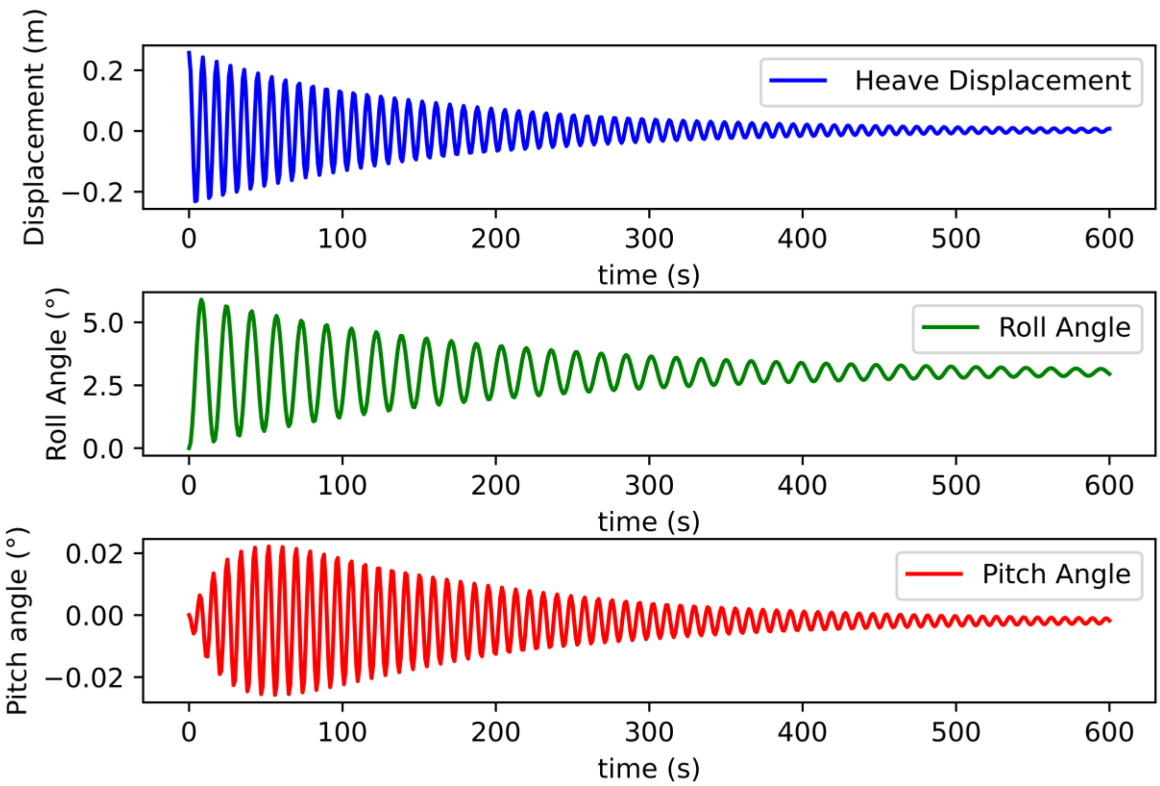

Figure 4.11 Coupled motion time domain analysis with 30% coupled coefficient values with respect to diagonal values

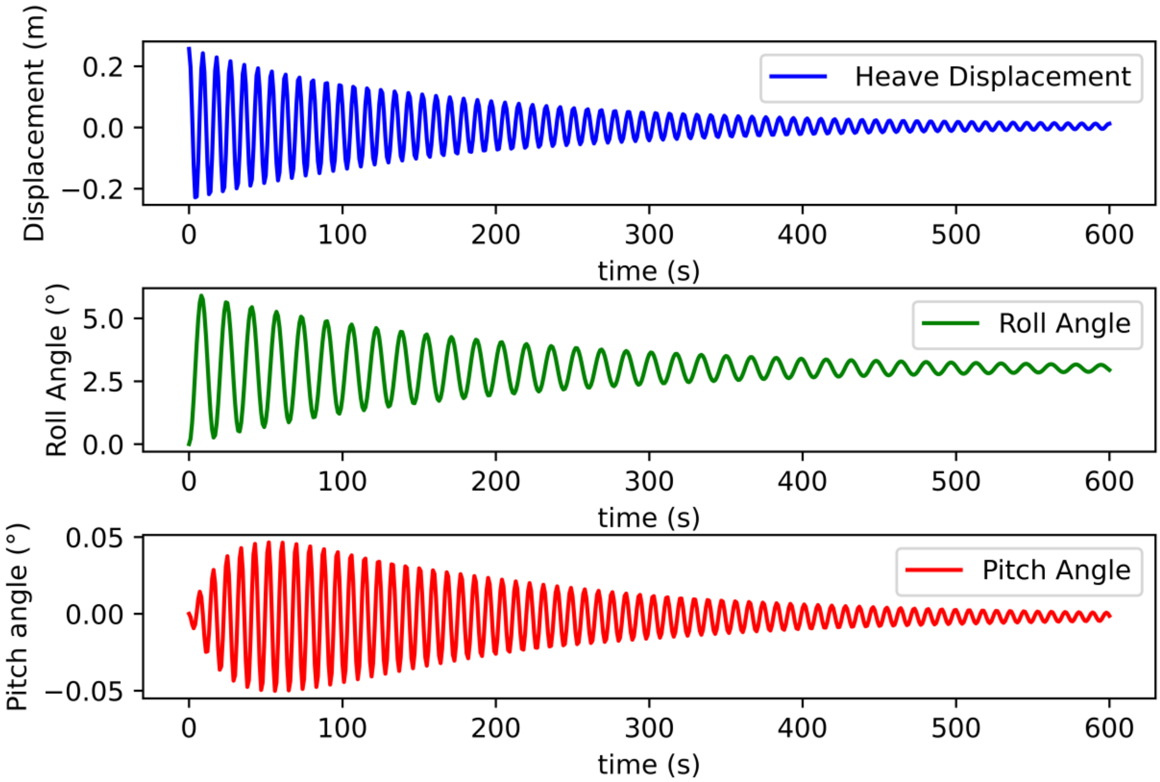

Figure 4.12 Coupled motion time domain analysis with 60% coupled coefficient values with respect to diagonal values

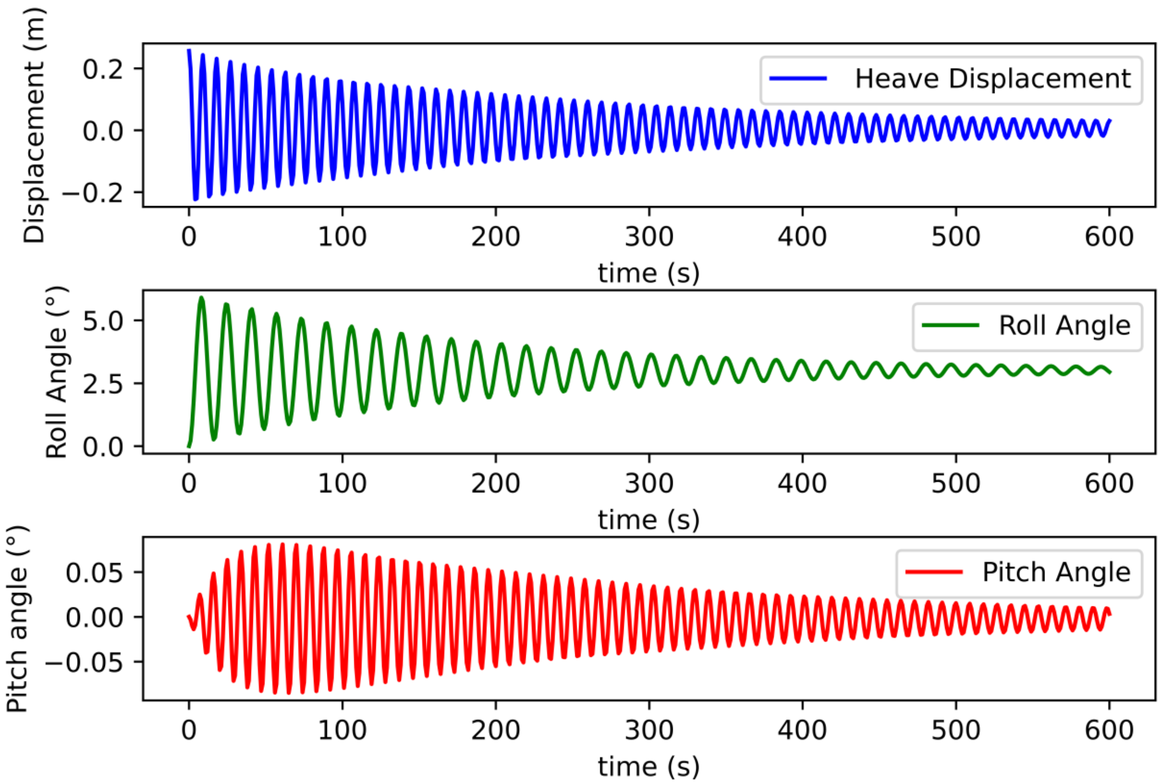

Figure 4.13 Coupled motion time domain analysis with 100% coupled coefficient values with respect to diagonal values

5. BALLAST WATER REMOVAL ANALYSIS

This section is devoted to analysing the practical possibility of ballast removal to lessen the maximum roll angle in case of sudden load loss. The idea is to remove the ballast water as soon as the 1500-ton load loss happens. Ballast arrangement can be seen in figure (5.1).

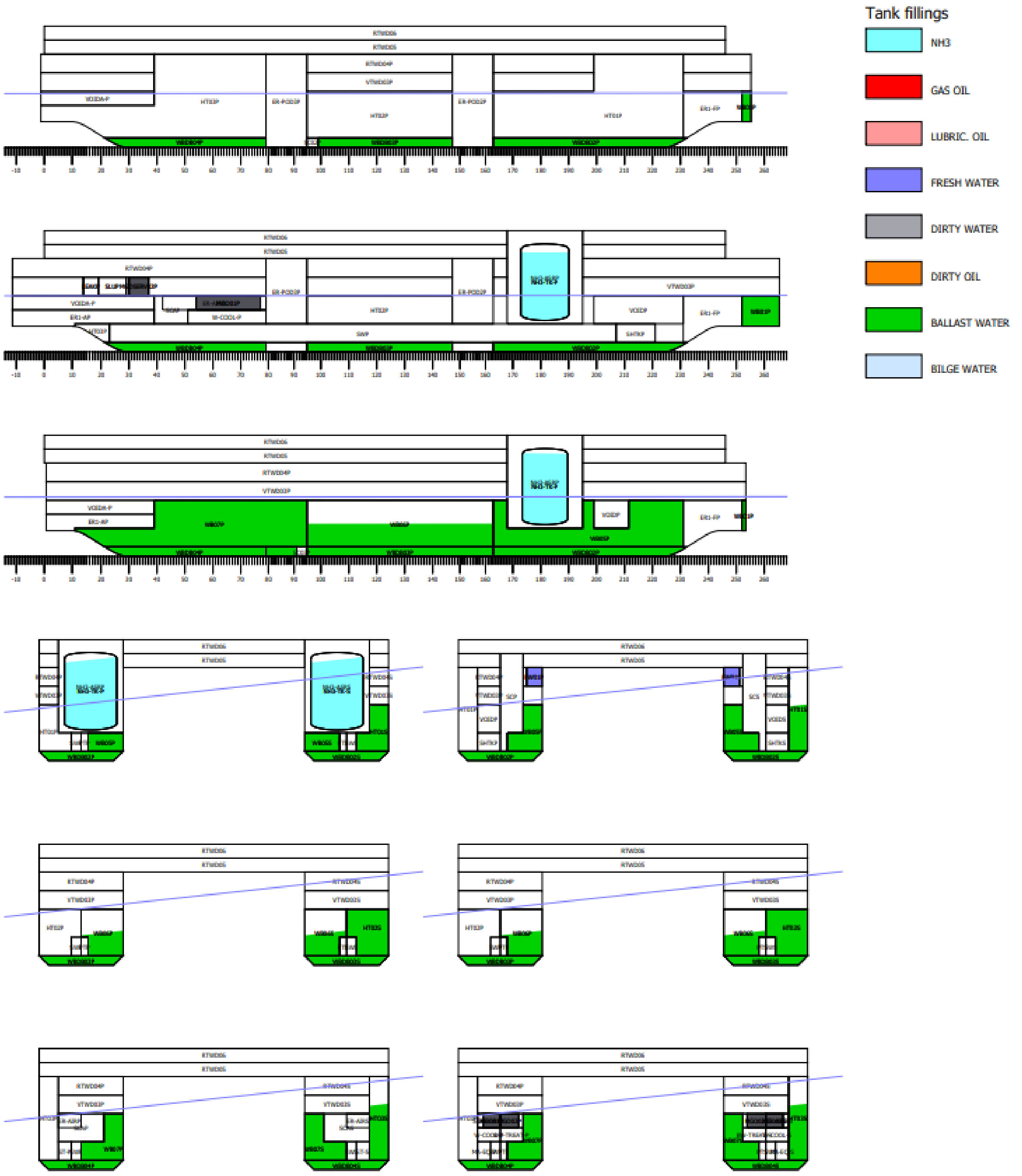

Figure 5. 1 Ballast water arrangement

5.1 Ballast Removal Possibilities

The additional ballast in case of sudden loss of load can be managed by two ways. First when load loss from the port side, the ballast water can be displaced from the starboard side to the port side to reduce or minimize the perturbing moment. However, no such arrangement or mechanism available in current studied ship to transform the ballast from port to starboard side. Hence, this option is withdrawn. The second option is to discharge the water directly to the sea. Currently, the ship has two pumps installed on starboard side and same on the port side. The optimum moment reduction is possible by discharging water through pumps with their full capacity on starboard side while on port side, pumps can be used to intake the water from sea. This intake water will serve the purpose of counter moment to the perturbation. Currently, each pump can operate with flow rate of 3000 m^3/h at its full capacity. The table (5.1) shows the ballast condition in the respective tanks in operating condition of the ship.

Table 5.1 Ballast arrangement during with 1500T load mounted

Tank	Description	Weight (t)	Fill%	TCG (m)
HT01S	Heeling Tank 01 SB	1967.4	75.00	34.91
HT02S	Heeling Tank 02 SB	2557.4	100.00	33.36
HT03S	Heeling Tank 03 SB	1737.1	74.30	35.20
WB01P	Forepeak PS	421.6	100.00	-28.50
WB01S	Forepeak SB	421.6	100.00	28.50
WB05P	Water Ballast 05 PS	1914.1	100.00	-22.83
WB05S	Water Ballast 05 SB	1914.1	100.00	22.83
WB06P	Water Ballast 06 PS	1535.7	45.50	-23.41
WB06S	Water Ballast 06 SB	1535.7	45.50	23.41
WB07P	Water Ballast 07 PS	2321.4	100.00	-22.88

WB07S	Water Ballast 07 SB	2321.4	100.00	22.88
WBDB02P	Double Bottom 02 PS	1258.4	100.00	-28.52
WBDB02S	Double Bottom 02 SB	1258.4	100.00	28.52
WBDB03P	Double Bottom 03 PS	1257.1	100.00	-27.59
WBDB03S	Double Bottom 03 SB	1257.1	100.00	27.59
WBDB04P	Double Bottom 04 PS	1051.8	100.00	-28.50
WBDB04S	Double Bottom 04 SB	1051.8	100.00	28.50

Considering the Ballast water filling arrangement at the upright condition with load hanging, the ballast water tanks on the port side are already filled to their maximum level and cannot be used for water intake to make counter moment except tank WB06P which is 45.5% fill. The only feasible option left is to discharge the water through starboard side pumps at their full capacity.

5.2 Ballast Water Discharge Analysis

To determine the moment loss due to discharge, it is necessary to estimate the transversal point force position with respect to the coordinates. The transversal point force is estimated by using table below (5.2) that contains water ballast tanks on the port side of the ship.

Table 5.2 Transversal force point estimation for Ballast water on starboard side of ship

Tank	Description	Weight (t)	Fill%	TCG (m)
HT01S	Heeling Tank 01 SB	1967.4	75.00	34.91
HT02S	Heeling Tank 02 SB	2557.4	100.00	33.36

HT03S	Heeling Tank 03 SB	1737.1	74.30	35.20
WB01S	Forepeak SB	421.6	100.00	28.50
WB05S	Water Ballast 05 SB	1914.1	100.00	22.83
WB06S	Water Ballast 06 SB	1535.7	45.50	23.41
WB07S	Water Ballast 07 SB	2321.4	100.00	22.88
WBDB02S	Double Bottom 02 SB	1258.4	100.00	28.52
WBDB03S	Double Bottom 03 SB	1257.1	100.00	27.59
WBDB04S	Double Bottom 04 SB	1051.8	100.00	28.50
Weighted TCG Value				**28.74**

It is important to make a correction on TCG force point distance because due to loss of 1500 ton load, an offset of 2.83m to the centre of gravity must be considered. Therefore, the corrected TCG value comes out to be 25.91 m.

Corrected TCG will act as a moment arm of the water to be discharged and now, weight of the water that must be discharged to achieve the equilibrium with zero roll angle can be calculated from equation (5.1).

$$M_T = Water\ weight * Corrected\ TCG \qquad (5.1)$$

That means 4197 tons of water must be discharged to make the ship upright at 0-degree heel angle. To discharge this water, two pumps of each capacity 3000 m^3/hr can be used, and simulated results are shown in figure (5.2) and (5.3).

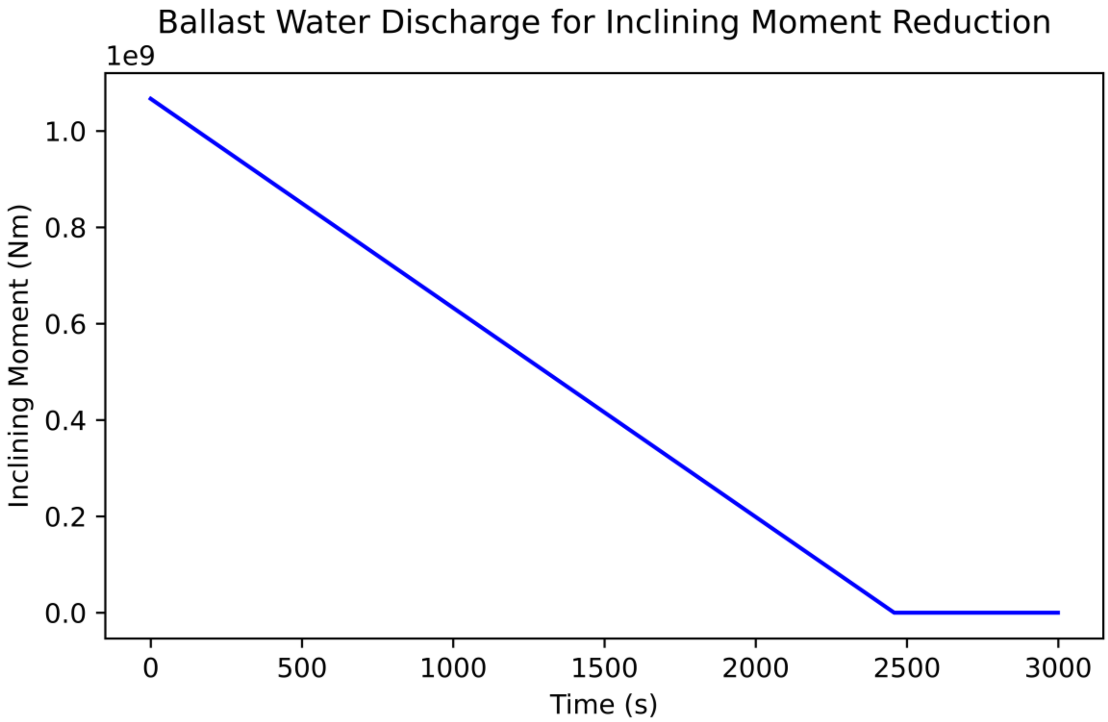

Figure 5.2 Ballast water discharge for inclining moment reduction while using starboard side pumps

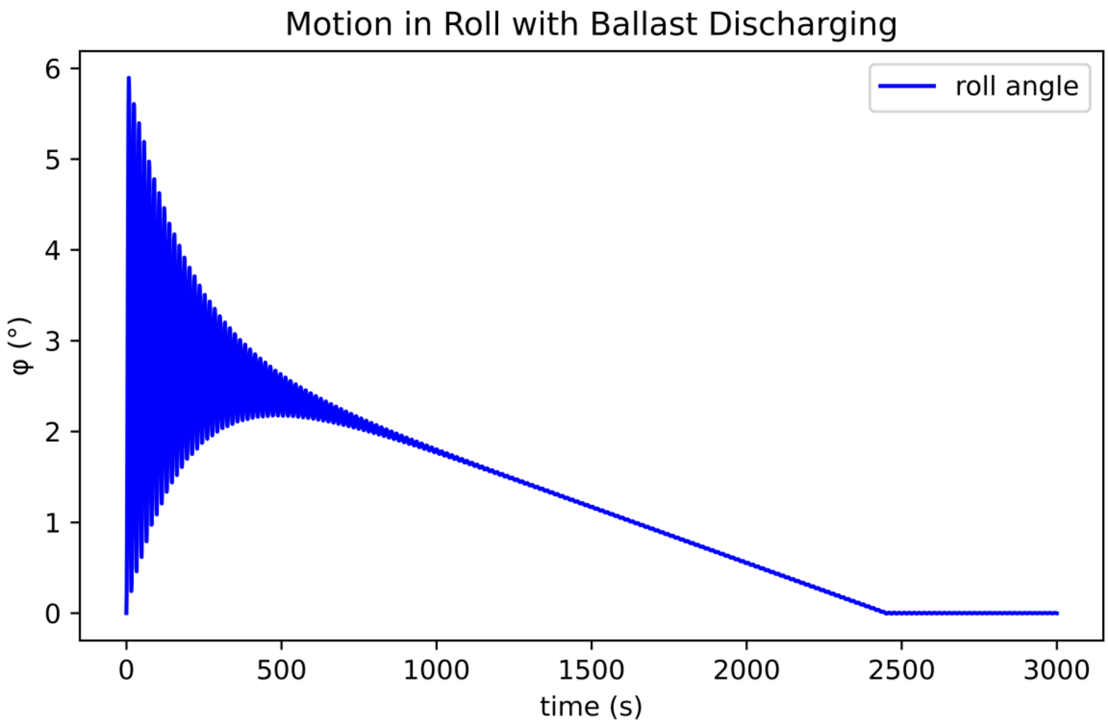

Figure 5.3 Depiction of roll angle reduction due to ballast discharge while using starboard side pumps

In case of tank WB06P on port side utilized as intake water from the sea, in that case the time required for full discharge is reduced from 2502 to 2255 sec as shown in figure (5.4).

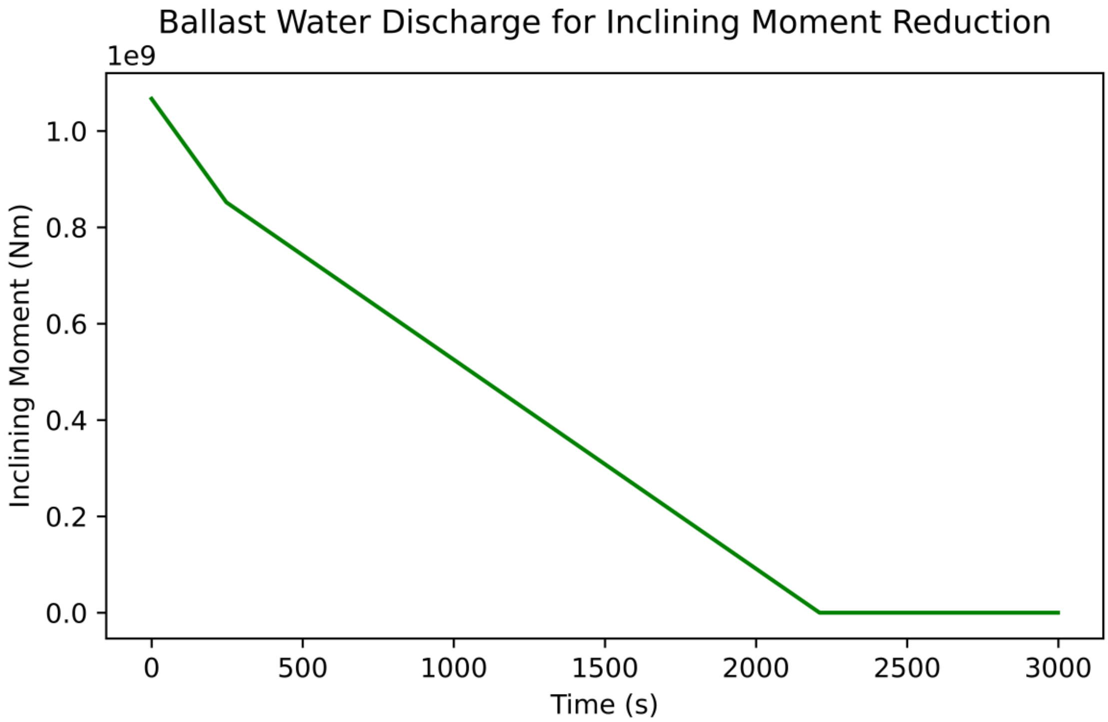

Figure 5.4 Ballast discharge for inclination moment reduction while using port and starboard side pumps

With current installation pumps the maximum roll angle is reduced from 5.904^{o} to 5.885^{o} that is a negligible effect on the maximum roll angle as shown in figure (5.5). The maximum roll angle occurrence during the simulation is on the first oscillation i.e., it must be minimized within the first oscillation before peak value. The analysis for reducing maximum roll angle to 25%, 50% and 75% of the original value is shown in table (5.3).

Table 5.3 Pump requirement analysis for maximum roll angle reduction

Sr. No	Percentage Maximum angle reduction (%)	New maximum angle (^{o})	Pump Capacity (m^3/hr)	Total Pump Required (qty)
1	25	4.428	3000	336
2	50	2.952	3000	808
3	75	1.476	3000	1840

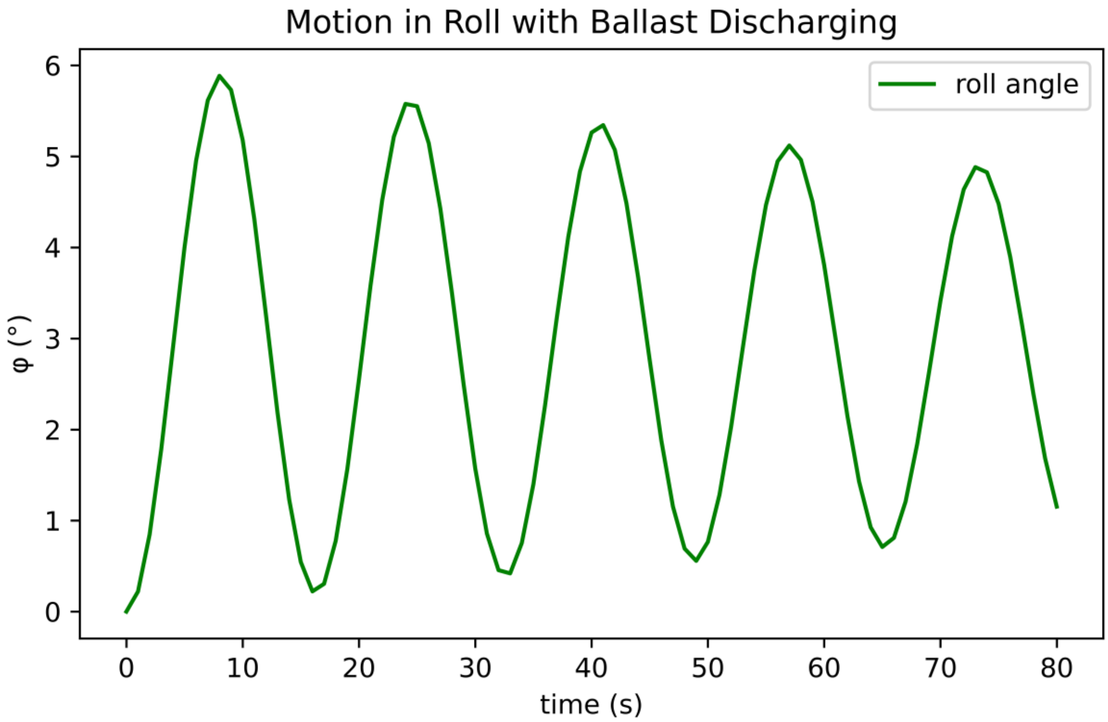

Figure 5.5 Zoomed view of roll angle reduction at initial stage of simulation with ballast discharging

The analysis above shows an impractical solution to the problem. Therefore, it can be concluded that required ballast discharge rate to lessen the maximum peak angle is practically impossible.

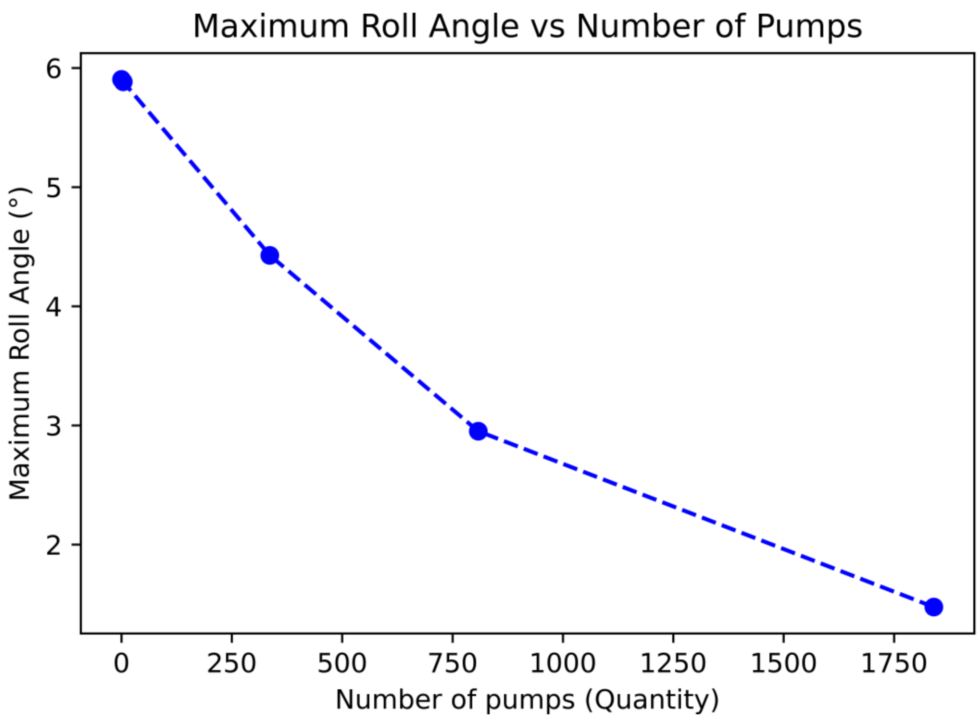

Figure 5.6 maximum roll angle desired value vs number of pumps required

6. CONCLUSION

1. Literature review and classification societies have shown focus on roll motion of the ship in case of sudden load loss. Therefore, single degree of freedom analysis in roll motion is sufficient to check the stability of ship design. Maximum roll angle during the simulation is quite low as compared to the DNV classification rule and based on the results, it can be said that ship complies with the stability criteria of DNV classification rule. Other classification rule can also be validated if GZ curve along with the angle of down flooding angle is available.
2. Pitch motion in single degree of freedom has shown a quite low inclination angle -0.0032^o. Therefore, pitch motion can be neglected. Moreover, due to less acceleration and velocity component, the effect of pitch to the other motion is minute. However, Heave motion has shown reasonable displacement during single degree of motion. Therefore, the effect of heave motion to the pitch is seen quite significant.
3. Sensitivity analysis of roll motion has shown 0.0237% of increase in maximum roll angle with respect to maximum roll angle at estimated added mass. Similarly, pitch and heave motion have shown 0.285% and 4.48% increase in maximum pitch angle and heave displacement with respect to maximum pitch angle and heave displacement at estimated added mass.
4. Pitch and heave motion are coupled strongly with each other. However, the roll motion has no or negligible effect of heave and pitch motion contribution. Moreover, changing the coupled coefficient from 30% to 100% of diagonal values of mass, damping and restoring coefficients has shown an increase in heave damping time duration and maximum inclination angle in pitch motion (from -0.0032^o to -0.05^o). Roll motion remains the same because of no contribution of coupled coefficients.
5. Concept of ballast discharging to reduce the first peak or maximum roll angle is impractical. To get the significant results due to ballast discharge, huge number of pumps and space are required. It will also change the design and operating draft of the ship due to increase weight. Discharging completely or a significant proportion of 4197 tons of ballast water is impractical before getting the first peak in roll (i.e., within 8.293 sec). However, from the analysis ship complies with the classification society requirement. Therefore, ship is stable even with ballast water discharging with an equilibrium state at roll angle of 2.952^o. To further reduce the maximum roll angle, an idea of installation of butterfly valves in the hull

of ship can be considered. These valves would be submerged in the water during design and operating draft and can be opened automatically in case of sudden loss of load and subsequently, intake water from the sea would work as a counter moment for the inclining moment. This idea can be accessed as a continuation of further work. A suitable size and quantity of valves along with attached piping strength calculation, sway motion due to water impact and maintenance or check-up possibilities can be accessed.

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