

STUDY ON THE APPLICATION OF U-SHAPE ANTI-ROLL TANK FOR A WATER HYACINTH PROCESSING MACHINE

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(Received: March 23, 2026; Revised: June 11, 2026; Accepted: July 31, 2026)

DOI: 10.31130/ud-jst.2026.24(9).172

Abstract - This study focuses on developing a stabilization system suitable for the actual operating conditions of a water hyacinth shredding machine on inland rivers and canals, with the aim of maintaining roll stability throughout operation. The paper proposes the application of a U-shaped anti-roll tank, or U-tank, to the water hyacinth shredding machine. Based on a survey of the vessel's operating conditions, a two-degree-of-freedom mathematical model was developed to describe the interaction between the vessel's roll motion and liquid oscillation inside the U-tank. In addition, several stabilization methods were reviewed and compared to support the selection of the U-shaped balancing tank as a suitable solution. Simulation results show that the balancing tank significantly reduces the roll response: the frequency response function decreases from 6.3 rad/rad without the tank to 2.3 rad/rad when active control is applied with $G = 8 \times 10^3$ Nm.s/rad. These results confirm the feasibility of the balancing tank and provide a basis for subsequent experimental studies.

Key words - Ship stabilization; anti-roll tank; water-hyacinth process machine; roll motion; U-tube anti-roll tank.

1. Introduction

Among the degrees of freedom of ship motion, roll motion generally has the largest amplitude and the lowest self-damping capability. It therefore becomes the motion most sensitive to stability, especially for small vessels with shallow draft and rapidly varying loads, such as water hyacinth shredding machines. Field surveys conducted in Tay Ninh Province and several provinces in the Mekong Delta show that water hyacinth shredding machines must operate in narrow and winding canal systems, while being simultaneously affected by currents, waves generated by larger vessels, crosswinds, and changes in the distribution of water hyacinth biomass during cutting and shredding operations. In addition, vibrations generated during the working process may cause imbalance phenomena. Under such operating conditions, the vessel's center of gravity tends to shift continuously in the direction of the acting forces. This, combined with vibrations caused by the shredding unit acting on water hyacinths as well as unidentified objects, leads to a periodically repeated heeling state. This process accumulates until it becomes sufficiently influential to cause instability; therefore, an appropriate balancing solution is required [1].

The object of this study is the MPLB-01 water hyacinth shredding machine, a specialized watercraft used for water hyacinth shredding that is currently operating in Tay Ninh Province and neighboring areas, as shown in Figure 1. Structurally, the vessel has a flat-bottom design, which is

suitable for operation in inland canal systems characterized by narrow waterways and water levels that vary seasonally.



Figure 1. The MPLB-01 water hyacinth shredding machine under actual operating conditions

The operating principle is based on a shredding unit mounted at the bow of the vessel, as presented in Figure 2. This unit uses a cylindrical cutting drum equipped with helical blades (1), driven by a diesel engine, in combination with a lifting-cylinder mechanism to adjust the working depth. The propulsion and steering system (2) is controlled through the rudder and propeller, allowing the vessel to maneuver flexibly when processing large aggregated mats of water hyacinth.

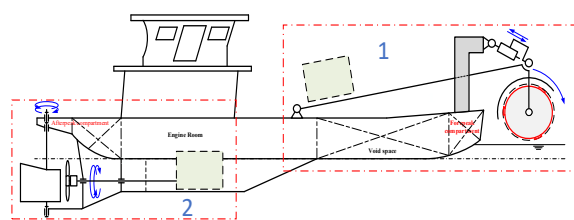


Figure 2. Schematic diagram of the MPLB-01 water hyacinth shredding machine. 1-Cutting and shredding unit; 2-Propulsion unit

According to QCVN 72:2013/BGTVT [2], the vessel is classified for operation in River Zone II, VR-SII, for which the maximum allowable significant wave height is $H_s = 1.2$ m. Waves in the operating environment are generally short waves with relatively small periods.

Table 1. Environmental wave excitation scenarios [1]

No.	Wave height H_s (m)	Wave period T_p (s)
1	0.1	2.0
2	0.4	3.2
3	0.8	4.5
4	1.2	5.6

For a water hyacinth shredding machine operating at low speed, as illustrated in Figure 3, roll motion is the most critical stability issue and should be prioritized for mitigation. This is because roll motion has the largest amplitude and the lowest self-damping capability, while the vessel’s natural roll frequency can easily fall within the resonance range of water waves, resulting in maximum oscillations. In contrast, other motions, such as pitch and yaw, are better controlled by the hull design and steering system [3, 4].

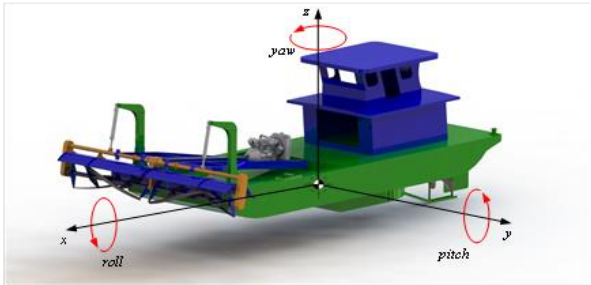


Figure 3. Illustration of the 6 DOF of the MPLB-01 water hyacinth shredding machine

Table 1. Hydrostatic parameters of the MPLB-01 water hyacinth shredding machine

No.	Parameter		Value
1	Maximum length, $L_{max}(m)$		9.27
2	Design waterline length, $L_{dn}(m)$		7.95
3	Maximum breadth, $B_{max}(m)$		2.97
4	Design breadth, $B_{tk}(m)$		2.90
5	Depth, $D(m)$		0.75
6	Draft, $d(m)$		0.233
7	Displacement, $\Delta(T)$		4.76
8	Crew		2
9	Dimensional ratio	L/D	10.6
10		B/D	3.87

The equipment is designed with a reasonable transverse mass distribution; therefore, the vessel’s static heel is insignificant. Accordingly, this study mainly focuses on reducing the dynamic roll amplitude generated under the action of external excitation forces. The OXYZ coordinate system is selected with the origin O located at the intersection of the midship section plane, the centerline plane, and the baseline plane of the vessel.

Table 2. Center-of-gravity parameters of the MPLB-01 water hyacinth shredding machine [1]

No.	Component	Pi (T)	Xi (m)	Yi (m)	Zi (m)
1	Ship hull	2.7	-0.25	0	0.575
2	Permanent stern ballast	1.3	-2.5	0.18	1.2
3	Working module	0.6	3.5	-0.08	0.45
4	Fuel	0.01	-0.12	0.1	0.1
5	Crew	0.15	-0.5	0	1.85
6	Total	4.76	-0.4	0.04	0.77

While stabilizing fins are dominant at high speeds, anti-roll tank systems still play an important role, particularly for vessels operating at low speeds or in stationary conditions, where stabilizing fins are ineffective [6–8].

According to the study by Watts [6] and the theoretical foundation on the influence of waves on roll motion established by Froude [9], the tank has a uniform cross-section, is installed transversely across the vessel, is located above the center of gravity, and is partially filled with water. The mass of water moves back and forth inside the tank with a phase lag relative to the vessel’s roll motion, generating an out-of-phase moment that helps reduce roll amplitude. The roll-reduction mechanism strongly depends on tuning the natural frequency of the water oscillation in the tank close to the vessel’s natural roll frequency, which can be achieved by varying the water depth in the tank [6].

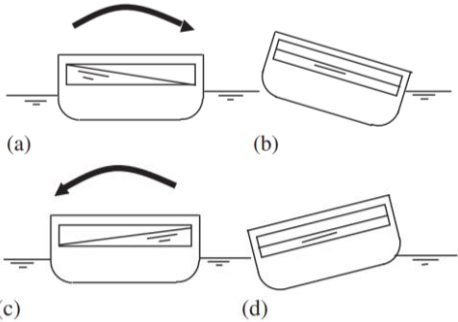


Figure 4. Operational mechanism of a free-surface anti-roll tank [1]

To optimize fluid dynamics in free-surface tanks, various geometric configurations and baffle arrangements have been investigated. According to Ikeda and Yoshiyama, the occurrence of sway motion reduces anti-roll performance and lengthens the natural period of the tank [10]. On the other hand, Lee and Vassalos experimentally demonstrated that modifying the baffle configuration allows flexible adjustment of the amplitude and optimal operating frequency range of the system [11].

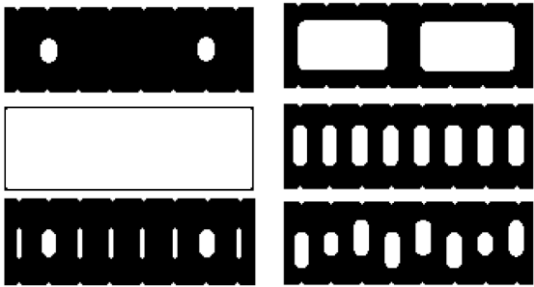


Figure 5. Baffle design for the anti-roll tank according to the study by Lee and Vassalos [11]

However, free-surface tanks exhibit several practical limitations: fluid dynamics are difficult to control under large-wave conditions; the free-surface effect reduces the metacentric height, GM, thereby degrading static stability; and the installation location often occupies critical space on the vessel. To overcome these limitations more thoroughly, the U-tank configuration was proposed by Frahm [12]. With a design consisting of two side compartments connected by a water duct at the bottom and an air duct at the top, the U-tank minimizes the free-surface area, thereby significantly eliminating negative effects on the vessel’s static stability.

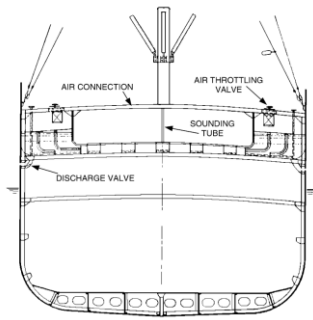


Figure 6. U-tank model according to the study by Frahm [12]

The study successfully tested the U-tank on a physical model, achieving a significant reduction in roll with a water mass accounting for approximately 1.3–1.5% of the vessel displacement. Studies on the modeling and control of U-tanks are diverse, ranging from linear models using Lagrange's method and energy-based approaches by Goodrich, Kagawa, and Shyu [13–15], to more complex nonlinear models by Youssef, confirming their effectiveness in reducing roll even in nonlinear frequency ranges and under various external-force conditions [16].

In recent years, studies on anti-roll tanks have focused on parameter optimization, active control, and simulation. Shaik et al. [17] analyzed the nonlinear dynamics and stability of a vessel equipped with an active anti-roll tank, considering control-process delay and pump-driven liquid transfer. Subramanian et al. [18] optimized the design of a passive anti-roll tank. Kapsenberg and Carette [19] proposed a consistent design–evaluation method. Chen et al. [20] used computational fluid dynamics to simulate roll decay of a vessel equipped with an anti-roll tank. Wawrzynski [21] indicated the influence of damping models on roll response, especially near resonance. Zhang et al. [22] further developed anti-roll control for low-speed or zero-speed conditions [17–23]. Hung et al. [24] applied adaptive fuzzy backstepping control to ship motion regulation under wave, wind, and current disturbances, demonstrating the potential of adaptive control strategies for marine systems operating under varying environmental conditions. These results show that the linear model remains suitable for analysis around the equilibrium state; however, the assumptions and application limits of the equivalent damping coefficient must be clearly stated [21, 24].

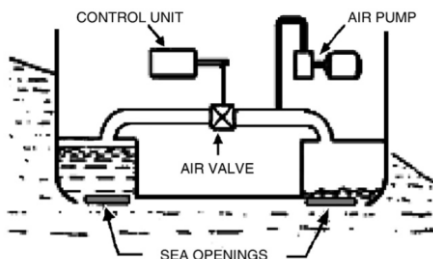


Figure 7. Free-flooding anti-roll tank configuration [25]

Another variant is the free-flooding anti-roll tank, in which the transverse water duct at the bottom is replaced by openings directly connected to the surrounding water environment. The main advantage of this type of tank is

that it does not require an internal water duct inside the hull [26]. Nevertheless, this configuration presents several disadvantages: reduced performance when the vessel changes speed, increased hydrodynamic resistance of the hull, and complex maintenance issues caused by corrosion and biofouling [25].

2. Problem formulation

2.1. Research methodology

This study employs mathematical modeling and frequency-domain simulation in MATLAB/Simulink to evaluate and design a U-shaped anti-roll tank for a specialized vessel. First, a linearized two-degree-of-freedom coupled model is developed to describe the interaction between the vessel roll angle, ϕ , and the liquid oscillation in the tank, represented by the free-surface angle δ . Based on this model, the governing dynamic equations are established, and the frequency response function between wave-slope excitation and roll response is derived. The response is expressed in rad/rad to enable an objective comparison over the frequency range [4]. On this modeling basis, active control is implemented using roll-rate feedback: the pump installed in the connecting duct of the anti-roll tank generates a pressure difference proportional to $\dot{\phi}$, which is equivalent to producing an active damping moment with a gain coefficient G .

The vessel–anti-roll tank system is modeled as a linear two-degree-of-freedom coupled system, including the vessel roll angle and the oscillation angle of the liquid free surface. The model is developed under the assumptions of small roll angles, incompressible liquid, nearly one-dimensional flow in the connecting duct, neglect of higher-order free-surface oscillations, and conversion of hydraulic losses into an equivalent linear damping coefficient. The system of differential equations is established. Then, assuming harmonic excitation, the system is transformed into the frequency domain to determine the amplitude response function and investigate resonance by means of frequency sweeping.

The amplitude response function, H [3, 4, 6], is the ratio between the amplitude of the vessel roll angle and the slope angle of the exciting wave. It depends on the frequency, ω , and represents the relationship between the vessel motion amplitude ϕ , as the output, and the amplitude of the wave excitation θ , as the input. In this study, the amplitude response function is used to evaluate the effectiveness of each roll-reduction configuration.

$$|H_{\text{roll}}(\omega)| = \left| \frac{\phi(\omega)}{\theta(\omega)} \right|$$

The frequency-domain simulation method is selected because the main objective of this study is to determine the steady-state response characteristics of the vessel–balancing tank system as a function of excitation frequency, including the amplitude response function, phase difference, resonance regions, and the effects of design, damping, and control parameters. Within the small-roll-angle range, the vessel–tank system is linearized around the equilibrium state. When the excitation moment is expressed in harmonic form, the responses of both the

vessel and the liquid have the same frequency. Therefore, the differential equations in the time domain can be converted into a system of complex algebraic equations and solved directly at each value of ω . This approach eliminates the influence of transient responses and initial conditions, while also facilitating the sweep over the entire frequency range to identify frequencies at which the tank may either reduce or amplify the roll response.

The model is validated indirectly by setting the system to passive mode so that it degenerates into the same two-degree-of-freedom model as that used in the experimental study by Stigter [27]. Harmonic wave excitation is then simulated over a frequency sweep covering the natural resonance range of the ship model, and the roll response of the present model is compared with the data and characteristics reported by Stigter [27]. Model validation is evaluated through statistical comparison.

2.2. Research contents

This study focuses on addressing the transverse imbalance of the vessel by using a U-shaped anti-roll tank system. The resonant oscillation mechanism of the liquid column in the tank enables passive absorption and dissipation of roll energy [16, 28], without requiring a complex mechanical structure. The system has high reliability and is suitable for the limited space available on small vessels. The arrangement of the anti-roll tank is shown in Figure 8.

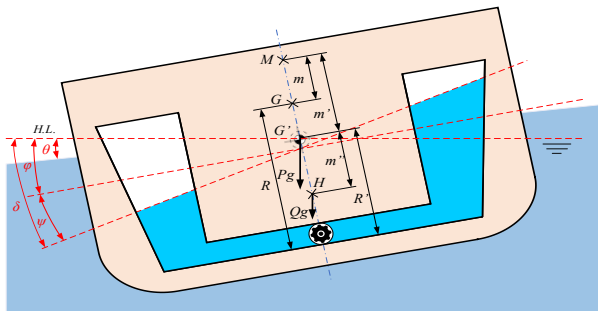


Figure 8. Schematic arrangement of the anti-roll tank on the water hyacinth shredding machine

When pump control is not activated, the system operates in passive mode. The water mass naturally moves from the higher side to the lower side under the action of gravity when the vessel heels due to waves. In active mode, the system integrates sensors that measure the roll state in real time to control the pump, thereby generating a forced counteracting moment. The resultant moment produced by the active anti-roll tank directly cancels the heeling moment caused by waves or moving loads, maintaining the dynamic stability of the vessel around the roll angle position $\varphi \approx 0^\circ$.

The system under consideration is a two-degree-of-freedom mechanical system subjected to random excitation generated by waves in the operating environment. The state variables include:

+ $\varphi(t)$: the roll angle of the vessel relative to the vertical direction;

+ $\psi(t)$: the angle of the liquid surface in the anti-roll tank relative to the vessel.

Based on Stigter's linear model, the equations of motion are written as follows [27]:

$$\{a_1\ddot{\phi} + a_2\dot{\phi} + a_3\phi + c_1\ddot{\psi} + c_3\psi = m_1 \quad (1)$$

$$l c_1 \ddot{\phi} + c_3 \phi + b_1 \ddot{\psi} + b_2 \dot{\psi} + b_3 \psi = m_2 \quad (2)$$

where, m_1 is the external moment acting on the vessel, namely the wave excitation, and m_2 is the external moment acting on the tank, namely the pump-induced moment.

The elements a_i , b_i , and c_i in the matrices represent the physical characteristics of the system:

+ $a_1 = P j_s^2 + I_{TG}$: mass moment of inertia of the vessel and anti-roll tank system (kg.m^2);

+ $a_2 = d_s$: linear roll damping coefficient of the hull (Nm/s);

+ $a_3 = (P+Q)m'g$: restoring moment of the vessel and anti-roll tank system (Nm);

+ $b_1 = \frac{1}{2} \rho w^2 w_3^2 I E_1$: mass moment of inertia of the liquid contained in the tank (kg.m²);

+ $b_2 = \frac{1}{2} \rho w_2^2 w_3^2 l E_2 d_T$: total linear damping coefficient between the liquid flow in the tank and the tank walls (Nm/s);

+ $b_3 = \frac{1}{2} \rho g w_2^2 w_3^2$: restoring moment of the liquid in the tank (Nm);

+ $c_1 = \rho w_2 w_3 I_{E_3}$: mass moment of inertia of the liquid contained in the tank (kg.m^2);

$$+ c_3 = \rho g w_2 w_3^2 l = b_3.$$

Here, P is the vessel mass; Q is the mass of the liquid contained in the anti-roll tank; $I_{TG'}$ is the moment of inertia of the liquid in the tank with respect to the center of gravity of the system; j_s is the radius of gyration of the vessel; d_s is the linear roll damping coefficient of the hull; m' is the distance between the metacenter M and the center of gravity of the system G' ; g is the gravitational acceleration; ρ is the density of the water in the tank; l is the length of the anti-roll tank; and d_T is the total linear damping coefficient of the tank.

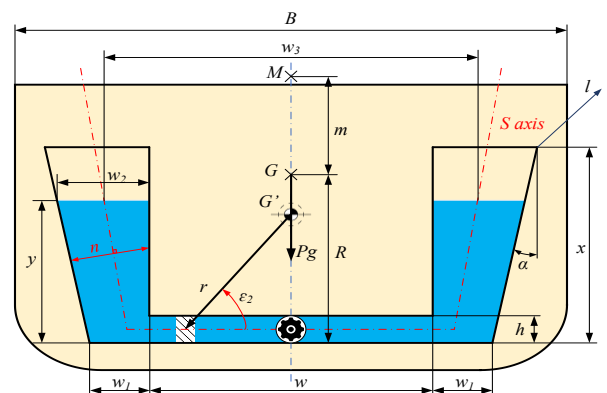


Figure 9. Geometric dimensions of the anti-roll tank

The geometric coefficients E_1 , E_2 , and E_3 depend on the shape and dimensions of the tank and are determined through line integrals along the tank profile, namely the S axis, where n is the width perpendicular to the S axis [27]:

$$E_1 = \frac{1}{2} \int_{S,1}^{S,r} \frac{ds}{n} = \frac{w+w_1}{2h} + \frac{2,31}{a} \log \left(1 + \frac{y-1/2h}{h/2+w_1/a} \right) \quad (3)$$

$$E_2 = \frac{1}{2} \int_{S,l} \frac{S_r ds}{n^2} = \frac{w+w_1}{2h^2} + \frac{y-1/2h}{\alpha^2(y+w_1/\alpha)(h/2+w_1/\alpha)} \quad (4)$$

$$E_3 = -\frac{1}{2} \int_0^{S,r} r \cdot \cos\left(\varepsilon_2 + \frac{\pi}{2}\right) ds \quad (5)$$

$$= (R+y-h) \left(\frac{w+w_1}{2}\right) + \frac{\alpha R}{2} \left(y - \frac{h}{2}\right)$$

Let: $\delta(t) = \varphi(t) + \psi(t)$

$$\rightarrow \begin{cases} \dot{\delta}(t) = \dot{\varphi}(t) + \dot{\psi}(t) \\ \ddot{\delta}(t) = \ddot{\varphi}(t) + \ddot{\psi}(t) \end{cases} \quad (6)$$

$$\rightarrow \begin{cases} \dot{\delta}(t) = \dot{\varphi}(t) + \dot{\psi}(t) \\ \ddot{\delta}(t) = \ddot{\varphi}(t) + \ddot{\psi}(t) \end{cases} \quad (7)$$

Substituting into the equations of motion yields:

$$(a_1 - c_1)\ddot{\varphi} + a_2\dot{\varphi} + (a_3 - b_3)\varphi + c_1\ddot{\delta} + b_3\delta = m_1 \quad (8)$$

$$(c_1 - b_1)\ddot{\varphi} - b_2\dot{\varphi} + b_1\ddot{\delta} + b_2\dot{\delta} + b_3\delta = m_2 \quad (9)$$

From the moment equilibrium equation of the vessel, the moment generated by the relative motion of the liquid in the tank and acting back on the vessel is determined as:

$$M_{U_{\text{tank}}}(t) = -[c_1\ddot{\psi}(t) + c_3\psi(t)] \quad (10)$$

$$= -[c_1[\ddot{\delta}(t) - \ddot{\varphi}(t)] + c_3[\delta(t) - \varphi(t)]]$$

where, $c_1\ddot{\psi}(t)$ represents the coupled inertial moment component due to the acceleration of the liquid, while $c_3\delta(t)$ represents the restoring moment component due to the liquid-level difference between the two branches of the tank. The damping coefficient b_2 does not appear directly in the above moment expression, but it affects the amplitude and phase of ψ through the equation of motion of the liquid.

To design and develop the integrated control algorithm, the differential equations of motion of the vessel-anti-roll tank system are mapped into the matrix form of a linear structural dynamic system:

$$M\ddot{x} + B\dot{x} + Kx = F \quad (11)$$

Where:

+ The state-variable vector is:

$$x = [\varphi(t), \delta(t)] \quad (12)$$

+ The excitation-force vector is:

$$F = [m_1(t), m_2(t)] \quad (13)$$

+ The system matrices M , B , and K , corresponding to the mass, damping, and stiffness matrices, respectively, are defined as follows:

$$M = \begin{bmatrix} a_1 - c_1 & c_1 \\ c_1 - b_1 & b_1 \end{bmatrix}, B = \begin{bmatrix} a_2 & 0 \\ -b_2 & b_2 \end{bmatrix}, K = \begin{bmatrix} a_3 - c_3 & b_3 \\ 0 & b_3 \end{bmatrix} \quad (14)$$

The formulation of these matrices provides the mathematical basis for implementing system identification algorithms, constructing the system frequency response function, and designing the active control law for the water hyacinth shredding machine under practical operating conditions.

Assuming that the excitation moments and responses are harmonic, they are expressed as: $F(t) = \text{Re}[\hat{F}e^{i\omega t}]$, $x(t) = \text{Re}[\hat{x}e^{i\omega t}]$. The differential equation system is then transformed into the Laplace frequency domain, with $(s=i\omega)$, $\dot{x} \rightarrow i\omega\hat{x}$; $\ddot{x} \rightarrow -\omega^2\hat{x}$:

$$(-\omega^2 M + i\omega B + K) \cdot \hat{x}(i\omega) = \hat{F}(i\omega)$$

$$\Leftrightarrow S(i\omega) \hat{x}(i\omega) = \hat{F}(i\omega)$$

$$\Leftrightarrow \hat{x}(i\omega) = S^{-1}(i\omega) \cdot \hat{F}(i\omega) \quad (15)$$

Where:

+ The complex amplitude vector of the angles φ and δ is:

$$x = [\hat{\varphi}, \hat{\delta}] \quad (16)$$

+ The complex amplitude vector of the two excitation moments m_1 and m_2 is:

$$F = [m_1(t), m_2(t)] \quad (17)$$

+ The dynamic stiffness matrix is:

$$S(i\omega) = -\omega^2 M + i\omega B + K \quad (18)$$

The system response in the form of an angular-velocity vector is:

$$\dot{x}(i\omega) = i\omega \cdot x(i\omega)$$

$$\Leftrightarrow \dot{x}(i\omega) = i\omega S^{-1}(i\omega) \cdot F(i\omega)$$

$$\Leftrightarrow \dot{x}(i\omega) = Y(i\omega) \cdot F(i\omega) \quad (19)$$

where the matrix $Y(i\omega) = i\omega S^{-1}(i\omega)$ contains the frequency response functions between the angular-velocity outputs and the excitation-moment inputs:

$$Y(i\omega) = \begin{bmatrix} Y_{\varphi, m1} & Y_{\varphi, m2} \\ Y_{\delta, m1} & Y_{\delta, m2} \end{bmatrix} \quad (20)$$

The active anti-roll system employs the roll angular-velocity feedback control method. The forced pump installed in the duct generates a pressure difference Δp , which is proportional to, but opposite in direction to, the vessel roll velocity $\dot{\varphi}$:

$$\Delta p = -g_r \dot{\varphi} \quad (21)$$

The pressure difference generated by the pump exerts a force on the liquid flow and produces a moment opposite to the roll velocity of the vessel. On this basis, the control moment M_b generated by the pump is formulated as follows:

$$M_b = -G \cdot \dot{\varphi} \quad (22)$$

The control moment component M_b acts as an external force applied to the tank, namely m_2 , and is included in the excitation-force vector F .

In this expression:

- + The roll angular velocity of the vessel is $\dot{\varphi}$ (rad/s);
- + The moment gain coefficient in the active balancing tank control system is $G = g_r h l R$ (Nms/rad);
- + The pressure gain coefficient is g_r .

The coefficient G serves to generate an active damping moment, M_b . When the vessel rolls with an angular velocity $\dot{\varphi}$, the system generates an opposing moment to attenuate the vessel's roll motion.

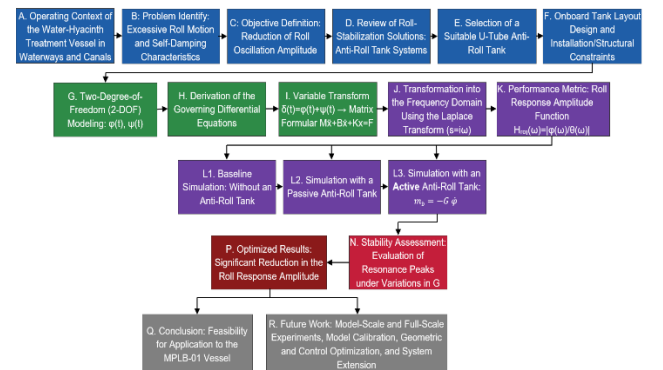


Figure 10. Research implementation flowchart

3. Results and discussion

This study uses the vessel and passive anti-roll tank parameters from the experimental model of Stigter [27] for comparison and validation.

Table 3. Validation parameters according to Stigter [27]

No.	Parameter	Value
1	Vessel mass, P (kg)	4700×10^3
2	Overall length, (m)	103.5
3	Width, (m)	18.8
4	Draft, (m)	3.9
5	Water mass, Q (kg)	22.8×10^3
6	Balancing tank length, l (m)	1
7	Bottom width of the water compartment, w_1 (m)	2.85
8	Duct width, w (m)	12.04
9	Balancing tank height, x (m)	0.66
10	Duct height, h (m)	2.7
11	Static water level in the tank, y (m)	0.05
12	Tank wall inclination angle, α (°)	22.8×10^3

The validation results show that the proposed model represents the roll response very well compared with the reference experimental data of Stigter [26], with a coefficient of determination $R^2 > 0.9$, specifically $R^2 = 0.9393$ without the tank and $R^2 = 0.9045$ with the passive anti-roll tank. The mean absolute percentage error is less than 17%. These results confirm that the model is sufficiently reliable for predicting the response around the resonance region.

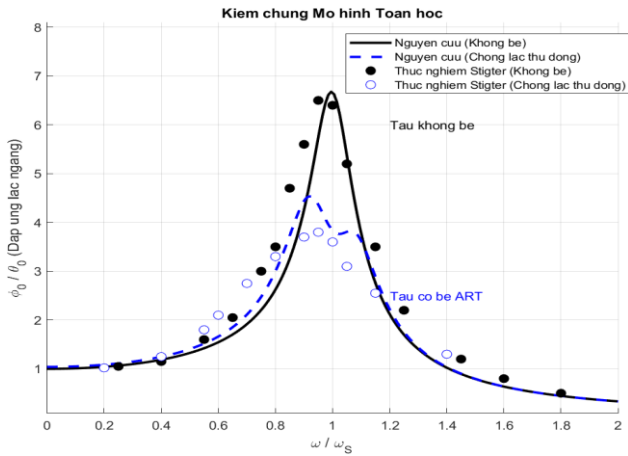


Figure 1. Comparison of the roll motion response amplitude with experimental results [27]

Table 4. Parameters of the anti-roll tank installed on the MPLB-01 water hyacinth breaking machine

No.	Parameter	Value
1	Water mass, Q (kg)	331.93
2	Anti-roll tank length, l (m)	0.5
3	Bottom width of the water compartment, w_1 (m)	0.28
4	Duct width, w (m)	1.6
5	Anti-roll tank height, x (m)	0.7
6	Duct height, h (m)	0.3
7	Static water level in the tank, y (m)	0.33
8	Tank-wall inclination angle, α (°)	10

The geometric parameters of the anti-roll tank are determined based on the existing structure and arrangement of the water hyacinth shredding machine, in order to ensure practical installation feasibility and avoid adverse effects on structural strength. At the same time, these parameters are selected so that the natural roll frequency of the vessel, ω_s , is approximately equal to the natural frequency of the liquid oscillation in the tank, ω_t , namely $\omega_s \approx \omega_t$. This arrangement optimizes the reduction of roll amplitude within the characteristic operating frequency range of the vessel [27].

The dynamic parameters of the vessel and anti-roll tank system were calculated and determined as follows.

Table 5. Dynamic parameters of the vessel–anti-roll tank system on the MPLB-01 water hyacinth shredding machine

No.	Parameter	Value
1	Mass moment of inertia of the vessel and anti-roll tank system, a_1 (kg.m ²)	15109.2
2	Linear roll damping coefficient of the hull, a_2 (Nm/s)	6893.3
3	Restoring moment of the vessel and anti-roll tank system, a_3 (Nm)	122849.4
4	Mass moment of inertia of the liquid contained in the tank, b_1 (kg.m ²)	372.9
5	Total linear damping coefficient between the liquid flow in the tank and the tank walls, b_2 (Nm/s)	88.0
6	Restoring moment of the liquid in the tank, b_3 (Nm)	3050.3
7	Mass moment of inertia of the liquid contained in the tank, c_1 (kg.m ²)	240.3

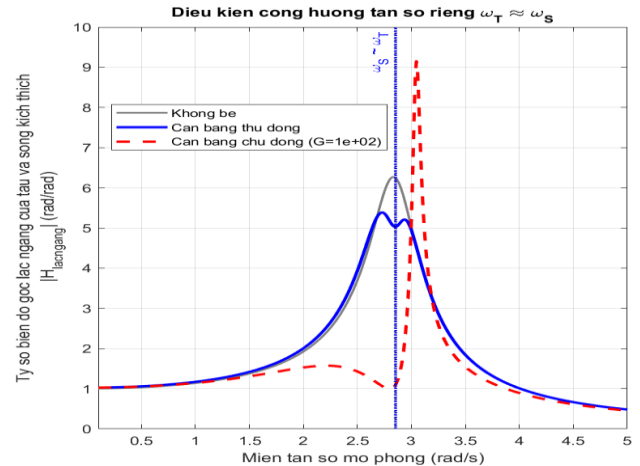


Figure 12. Stability and amplitude response of the anti-roll tank under natural-frequency resonance conditions ($\omega_T \approx \omega_S$)

The simulation results of the frequency response function under the condition $\omega_T \approx \omega_S$, shown in Figure 12, indicate that although the passive mode exhibits good damping characteristics through resonance-peak splitting, reducing the response from 6.3 to 5.4 rad/rad, the application of active control with $G > 1000 \text{ Nm} \cdot \frac{s}{\text{rad}}$ leads to a more unstable condition than that before the anti-roll tank is installed. At the excitation frequency $\omega = \omega_s$, the control moment generated by the pump not

only fails to suppress the oscillation but also resonates in phase with the wave excitation. Under large wave excitation, the excessive resonant response of the liquid mass generates two secondary oscillation peaks with high amplitudes near the natural frequency, causing instability of the vessel. To overcome this issue, the study proposes a tank design configuration with $f < 1$ combined with active control. Instead of relying on the natural frequency, the pump generates a forced moment to compensate for the phase difference, thereby completely eliminating the secondary roll peaks and ensuring optimal roll-reduction performance with tank dimensions compatible with the vessel.

The simulation results in Figure 13, where $\omega_t = 0.7\omega_s$, show that lowering the natural frequency of the anti-roll tank is a key factor in improving roll-reduction effectiveness. Without the anti-roll tank, the resonance peak around $\omega_s \approx 2.8$ rad/s reaches approximately 6.3 rad/rad, while the optimized passive balancing tank only reduces it to 5.4 rad/rad. In Figure 13, with the active anti-roll tank and an increase in the gain coefficient from $G = 5 \cdot 10^2$ to $5 \cdot 10^3 \div 8 \cdot 10^3$, the resonance peak decreases to approximately 2.7 rad/rad.

The maximum resonance peak of the system occurs at the wave excitation frequency that resonates with the natural frequency of the vessel, $\omega = \omega_s = 2.85$ rad/s. At this frequency, the roll amplitude of the system is amplified to its maximum value. Therefore, the effectiveness of the anti-roll tank is evaluated through its ability to suppress this resonance peak and reduce the roll amplitude to a safe threshold.

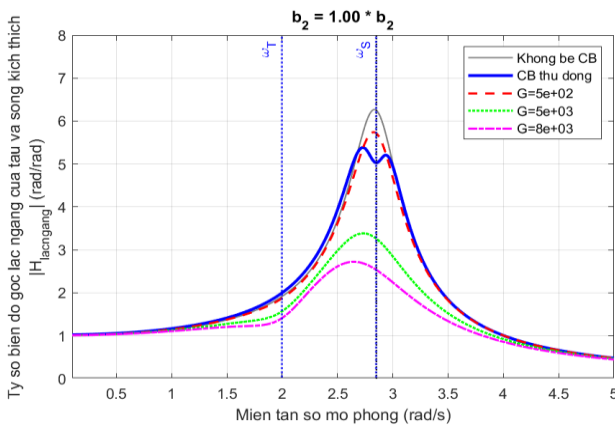


Figure 13. Evaluation of the stabilizing performance of the anti-roll tank under the configuration $\omega_t = 0.7\omega_s$

The deep notch at the frequency $\omega = \omega_t$, with a value of 1.995 rad/s in Figure 13, is characteristic of the anti-resonance phenomenon at the natural frequency of the tank. At this point, the inertial moment of the water flow in the tank oscillates out of phase and almost cancels the wave excitation moment, causing the vessel roll amplitude to decrease to a minimum. The depth of this notch indicates that the flow damping coefficient in the tank, b_2 , has been minimized, allowing the tank to operate as a vibration absorber.

The value of the gain coefficient G plays a direct and decisive role in the energy problem of the pump system.

Power-flow analysis shows that if the system operates with a high G value while the inherent passive damping coefficient b_2 remains large, the pump is forced to consume a large amount of energy merely to overcome friction in order to force the flow, leading to unnecessary energy loss. However, this scenario changes completely if friction in the tank is minimized, that is, if b_2 is reduced to a low level. In that case, combining low friction with a high G value allows the pump to shift from an energy-consuming state to an energy-dissipating state, operating similarly to a device that recovers energy from the vessel's roll motion rather than consuming electrical power from the source. Therefore, the optimal design strategy for ensuring energy efficiency is to combine the selection of a high G value, while still being compatible with the response capability of the pump mechanism, with the maximum possible reduction of the passive damping coefficient b_2 of the balancing tank.

4. Conclusion

Based on the above results, it can be concluded that the balancing tank configuration combined with active control is a feasible and effective solution for improving transverse stability of the water hyacinth shredding machine operating at low speed. This solution is suitable for the spatial constraints and working conditions in inland canal and river systems.

In subsequent studies, the authors will focus on experiments using a scaled model and tests on the actual vessel to calibrate the model, simultaneously optimize the tank geometry and control law according to stability–energy criteria, and extend the analysis to the nonlinear domain and more complex sea states.

Acknowledgements:

This study was conducted within the framework of the project entitled “Research on completing pilot production technology and transferring the water hyacinth processing machine system for canals, waterways, the Vam Co River, and neighboring areas”, project code: ĐTCNCT.02/26.

The authors would like to thank Nguyen Hong Barge Building One Member Co., Ltd. for facilitating the field tests for data collection and providing the facilities and manpower required to support the research implementation process.

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