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*Research Article*

# Optimization of Dynamic Parameters for Stability of Double-Pontoon Tension Leg Platforms

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**Abstract**

Tension Leg Platforms (TLPs) are compliant offshore structures widely used for deep-water resource extraction due to their high stability. However, extreme wave conditions can still induce significant oscillations. This study proposes a double-pontoon configuration, in which a secondary pontoon is flexibly connected to the main pontoon via springs. The system is modeled as a two-degree-of-freedom oscillator, and key dynamic parameters—pontoon mass distribution and spring stiffness—are optimized through numerical analysis. Laboratory-scale experiments are conducted to validate the analytical predictions. Results show that the double-pontoon arrangement effectively reduces the motion amplitude of the main pontoon by distributing wave-induced energy between the pontoons. Spring stiffness is found to be directly proportional to the main pontoon's amplitude, while variations in secondary pontoon mass cause only minor fluctuations in its response. Compared to a single-pontoon TLP, the proposed configuration achieves lower primary pontoon amplitudes under equivalent wave loading.

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## 1. Introduction

Tension Leg Platforms (TLPs) are a type of compliant floating structure widely employed for deep-water exploration and exploitation of oil, gas, and other marine resources (Muehlner, 2017), (Amaechi et al., 2022). The

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TLP design combines buoyancy forces and vertically tensioned mooring systems, enabling the platform to maintain vertical stability even under significant environmental loads (Li et al., 2025), (Boo et al., 2024). This characteristic makes TLPs more stable compared to other floating platforms, such as semi-submersibles or spar platforms (El Beshbichi et al., 2022). However, their stability can still be compromised under extreme environmental conditions, particularly high sea waves that induce dynamic oscillations in the structure (Prastianto & Sutha, 2003). Figure 1 shows a TLP typically used in deep waters.

Figure 1 Tension Leg Platform (TLP) used for deep-water offshore operations.

(https://www.marinetechnologynews.com/news/floating-production-system-projected-476651)

Numerous studies have been conducted to enhance the dynamic performance of TLPs, particularly in reducing wave-induced motion amplitudes. Previous researchers have developed equations to estimate wave forces on cylindrical structures, which serve as a fundamental basis for wave load analysis on TLPs (Morison et al., 1950). Subsequent numerical and experimental studies have shown that modifications in pontoon geometry (Yang et al., 2023), (Akmal, 2022), mass distribution, and mooring configurations can influence the dynamic response of TLPs (Zhang et al., 2022). Nevertheless, most approaches have focused on single-pontoon designs, leaving the potential of double-pontoon configurations largely unexplored in the literature.

The double-pontoon concept offers the potential for more effective vibration energy distribution, allowing part of the wave-induced load on the main pontoon to be transferred to a secondary pontoon (Williams et al., 2000). By introducing an elastic connector such as a spring between the pontoons, the wave energy can be redistributed through relative oscillations between the two pontoons (Mursid et al., 2025). This approach is analogous to the tuned mass damper concept commonly applied in high-rise buildings (Teplyshev et al., 2018), yet its application to TLP structures remains rare. This represents a significant research gap worthy of investigation.

Beyond energy distribution, two key parameters govern the performance of a double-pontoon system: spring stiffness and the relative mass of the pontoons. Spring stiffness influences the system's natural frequency and determines the extent to which wave energy is absorbed or transmitted, while the mass of the secondary pontoon affects the magnitude of inertial forces and the relative stability of the system. The relationship between these parameters and the dynamic response of a double-pontoon TLP has not been comprehensively studied or experimentally validated.

In this context, the present study aims to investigate the influence of spring stiffness and secondary pontoon mass variations on the dynamic response of a double-pontoon TLP. Numerical analysis is performed using MATLAB to model the system as a two-degree-of-freedom oscillator, and laboratory-scale experiments are conducted to verify the analytical results. The research focuses on parameter optimization to minimize the motion amplitude of the main pontoon under wave loading.

The findings of this study are expected to contribute to the development of more stable TLP designs, particularly for operations in deep-water regions with extreme wave conditions. Furthermore, the results may serve as a reference for the offshore oil and gas industry, marine structural designers, and offshore engineering researchers in applying the double-pontoon concept as a strategy for improving dynamic stability.

## 2. Models and Analysis

The model used in this study is a Tension Leg Platform (TLP) with a double pontoon configuration, consisting of a main pontoon and a trailing pontoon connected by spring elements. The system is modelled as a two-degree-of-freedom vibration system, where each pontoon is represented as a concentrated mass connected in series through springs and subjected to excitation from wave forces. Figure 2 illustrates the model along with the coordinates used. The theoretical analysis begins with the determination of the wave forces acting on the structure using Morison's equation (Morison et al., 1950). This equation combines the inertia and drag force components, as shown in Equation (1).

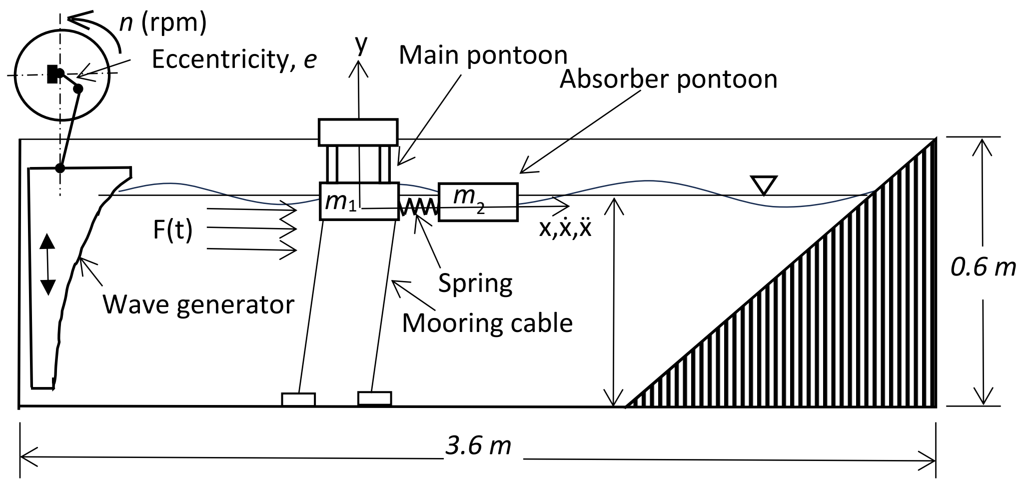


Figure 2 Illustrates the model

Equation (1) expresses the total wave force ($F$) acting on a cylindrical structure in a fluid, which consists of the inertia force ($F_{inertia}$) and the drag force ($F_{drag}$). The fluid particle acceleration determines the inertia force, while the drag force is governed by the relative velocity of the fluid particles for the structure. In this equation, $\rho$ denotes the fluid density (kg/m³), $C_I$ is the inertia coefficient, $A$ is the cross-sectional area of the structure (m²), and $\dot{u}$ represents the horizontal acceleration of the wave particle (m/s²). Meanwhile, $C_D$ is the drag coefficient, $D$ is the characteristic diameter of the structure (m), and $u$ is the horizontal velocity of the wave particle (m/s). The absolute value $|u|$ ensures that the drag force always acts in the direction of the fluid flow, while the factor ½ is the conventional correction constant used in the drag force formulation.

$$F = F_{inertia} + F_{drag} = \rho C_I A\dot{u} + \frac{1}{2}\rho C_D D u|u| \quad (1)$$

Dynamic analysis is illustrated through a two-degree-of-freedom vibration system model, as shown in Figure 2. If the vertical movement of masses $m_1$ and $m_2$ is constrained, then at least one coordinate $x(t)$ is needed to describe the position of the masses at various times. Based on Newton's second law of motion, the force equilibrium of the system can be expressed in Equation (2).

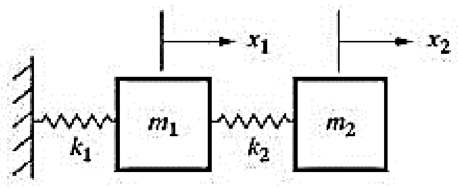


Figure 3 Two-Degree-of-Freedom Vibration System

$$\Sigma F = mg = m\frac{d^2x}{dt^2} \quad (2)$$

The harmonic vibration equations for $m_1$ and $m_2$ can be seen in Equation (3).

$$m_1\left[\frac{d^2x}{dt^2}\right] + (k_1+k_2)x_1 - k_2x_2 = 0$$
$$m_2\left[\frac{d^2x}{dt^2}\right] + k_2x_2 - k_2x_1 = 0 \quad (3)$$

Assuming that periodic motion with harmonic behavior at certain amplitude and frequency variations, the components can be expressed as follows:

$$x_1 = A\sin\omega\, t \ ; \ x_2 = B\sin\omega\, t \quad (4)$$

The second derivative of equation (4) is as shown in Equation (5)

$$\frac{d^2x_1}{dt^2} = -\omega^2 A\sin\omega\, t \ ; \frac{d^2x_1}{dt^2} = -\omega^2 B\sin\omega\, t \quad (5)$$

By substituting Equations (3) and (4) into Equation (5), the vibration equation can be written as Equation (6) and Equation (7)

$$m_1\left[\frac{d^2x}{dt^2}\right] + (k_1+k_2)x_1 - k_2x_2 = 0$$
$$-m_1\omega^2 A\sin\omega\, t + (k_1+k_2)A\sin\omega\, t - k_2B\sin\omega\, t = 0 \quad (6)$$
$$A\sin\omega\, t(k_1+k_2-m_1\omega^2) - B\sin\omega\, t(k_2) = 0$$

$$m_1\left[\frac{d^2x}{dt^2}\right] + k_2x_2 - k_2x_1 = 0$$
$$-m_2\omega^2 B\sin\omega\, t + k_2B\sin\omega\, t - k_2A\sin\omega\, t = 0 \quad (7)$$
$$B\sin\omega\, t(k_2-m_2\omega^2) - A\sin\omega\, t(k_2) = 0$$

Thus, the equation of motion becomes Equation (8).

$$A \sin \omega\, t(k_1 + k_2 - m_1\omega^2) - B \sin \omega\, t(k_2) = 0$$
$$B \sin \omega\, t(k_2 - m_2\omega^2) - A \sin \omega\, t(k_2) = 0 \tag{8}$$

$A$ and $B$ are the amplitudes of the two masses, the ratio must always be constant and dependent on time. Furthermore, Equation (8) can be written without including the function $\sin \omega\, t$, as seen in Equation (9).

$$A(k_1 + k_2 - m_1\omega^2) - Bk_2 = 0$$
$$-Ak_2 + B(k_2 - m_2\omega^2) = 0 \tag{9}$$

Equation (9) can be rearranged into a matrix, as shown in Equation (10).

$$\begin{bmatrix} k_1 + k_2 - m_1\omega^2 & -k_2 \\ -k_2 & k_2 - m_2\omega^2 \end{bmatrix} \begin{bmatrix} A \\ B \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{10}$$

The matrix shows a linear algebraic equation with the solution $A = B = 0$ as the definition of system equilibrium. To obtain the values of $A$ and $B$, the determinant of the equation = 0.

$$det. \begin{bmatrix} k_1 + k_2 - m_1\omega^2 & -k_2 \\ -k_2 & k_2 - m_2\omega^2 \end{bmatrix} = 0$$
$$\begin{vmatrix} k_1 + k_2 - m_1\omega^2 & -k_2 \\ -k_2 & k_2 - m_2\omega^2 \end{vmatrix} = 0 \tag{11}$$
$$(k_1 + k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - (-k_2)(-k_2) = 0$$
$$m_1 m_2 \omega^4 + \big((-k_1 - k_2)m_2 - k_2 m_1\big)\omega^2 + k_1 k_2 = 0$$

The similarities in their characteristics are:

$$\omega^4 - \left(\frac{(k_1 + k_2)}{m_1} + \frac{k_2}{m_2}\right)\omega^2 + \frac{k_1 k_2}{m_1 m_2} = 0 \tag{12}$$

The general solution to the vibration equation is harmonic motion in the form of a wave, as shown in Equation (12).

$$x_1 = A_1 \sin \omega_1 t + A_2 \sin \omega_2\, t$$
$$x_2 = B_1 \sin \omega_1 t + B_2 \sin \omega_2\, t \tag{13}$$

Finally, a general solution was obtained for the motion equation of a two-degree-of-freedom system, as shown in Equation (14).

$$x_1 = A_1 \sin \omega_1 t + A_2 \sin \omega_2 t$$
$$x_2 = A_1\lambda_1 \sin \omega_1 t + A_2\lambda_2 \sin \omega_2 t \tag{13}$$

## 3. Laboratory-scale experiments

The experimental study was carried out using a wave testing device, namely a wave maker, as shown in Figure 4. Data were obtained from the motion of the Tension Leg Platform (TLP) double-pontoon structure between the main pontoon and the trailing pontoon. In this study, several parameter variations were applied, specifically the spring stiffness between the main pontoon and the trailing pontoon, as well as the additional mass on the absorber pontoon. The spring stiffness values tested were 10 N/m, 20 N/m, 30 N/m, 40 N/m, and 50 N/m. Meanwhile, the mass of the main pontoon (m1) = 800 g, while the mass of the absorber pontoon (m2) varies: 400 g, 500 g, and 600 g.

Figure 4 Eksperimental di Laboratorium

## 4. Results and Discussion

Berdasarkan penelitian yang telah dilakukan data didapatkan dari pergerakan posisi struktur Tension Leg Platform (TLP) double pontoon antara ponton utama dan ponton gandengan. Pada penelitian ini digunakan berbagai variasi parameter yaitu kekakuan pegas antara ponton utama dan ponton gandengan, kemudian tambahan massa pada ponton gandengan. Kekakuan pegas memiliki beberapa variasi, yaitu 10 N/m, 20 N/m, 30 N/m, 40 N/m dan 50 N/m. Sedangkan untuk variasi massa tambahan yang dipakai ialah 100 gram, 200 gram dan 300 gram.

### *4.1. Effect of Spring Stiffness on Displacement Amplitude*

Figure 5 presents the displacement response of the double-pontoon Tension Leg Platform (TLP) under different spring stiffness values of 10 N/m, 20 N/m, 30 N/m, 40 N/m, and 50 N/m. The graphs compare the motion of the main pontoon ($m_1$ = 200 g) and the absorber pontoon ($m_2$ = 100 g) over a simulation period of 60 seconds.

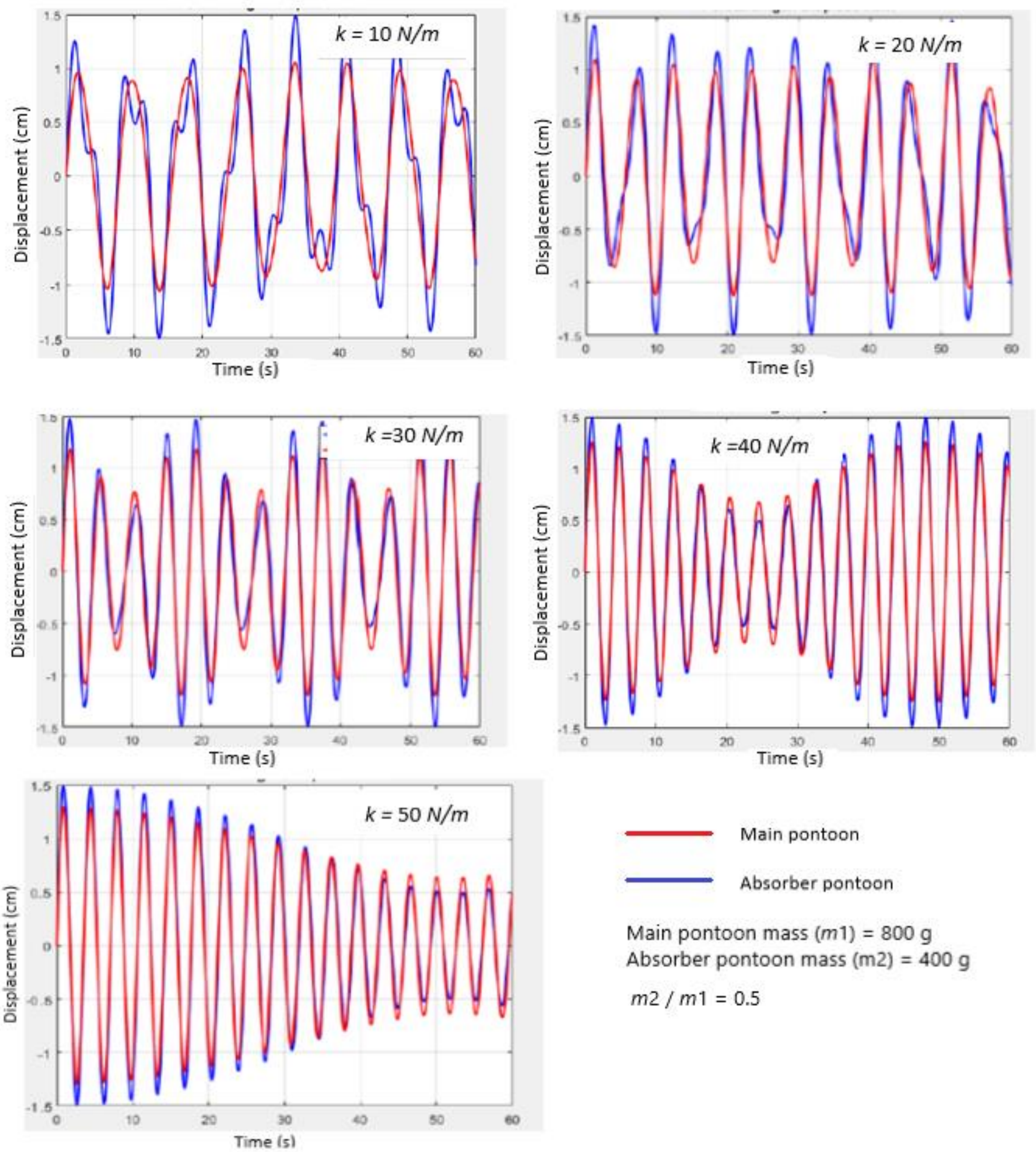


Figure 5 Displacement of the double-pontoon TLP with spring stiffness variations of k=10 N/m, 20 N/m, 30 N/m, 40 N/m, and 50 N/m.

In general, the displacement amplitude of the main pontoon is lower than that of the absorber pontoon. This indicates that part of the wave energy received by the system is transferred to the absorber pontoon through the spring mechanism, thereby reducing the dynamic load acting on the main pontoon. This phenomenon is consistent with the principle of a tuned mass absorber, in which the additional mass serves to redistribute vibration energy.

At low stiffness values (10 N/m), both pontoons exhibit relatively large amplitudes with less stable oscillation patterns. As the stiffness increases to 20–30 N/m, the displacement response becomes more regular, although the absorber pontoon still exhibits higher amplitudes compared to the main pontoon. This confirms the role of the absorber pontoon in accommodating additional energy absorbed from wave excitation.

### 4.2. The effect of mass ratio (m2/m1) on displacement

Figure 6 illustrates the effect of varying the mass ratio between the absorber pontoon and the main pontoon ($m_2/m_1$) on the displacement response of the double-pontoon TLP. The mass ratios considered were 0.5, 0.625, and 0.75, with a constant spring stiffness of k=10 N/mk. The graphs compare the displacement responses of the **main pontoon ($m_1$)** and the **absorber pontoon ($m_2$)** over a period of 60 seconds.

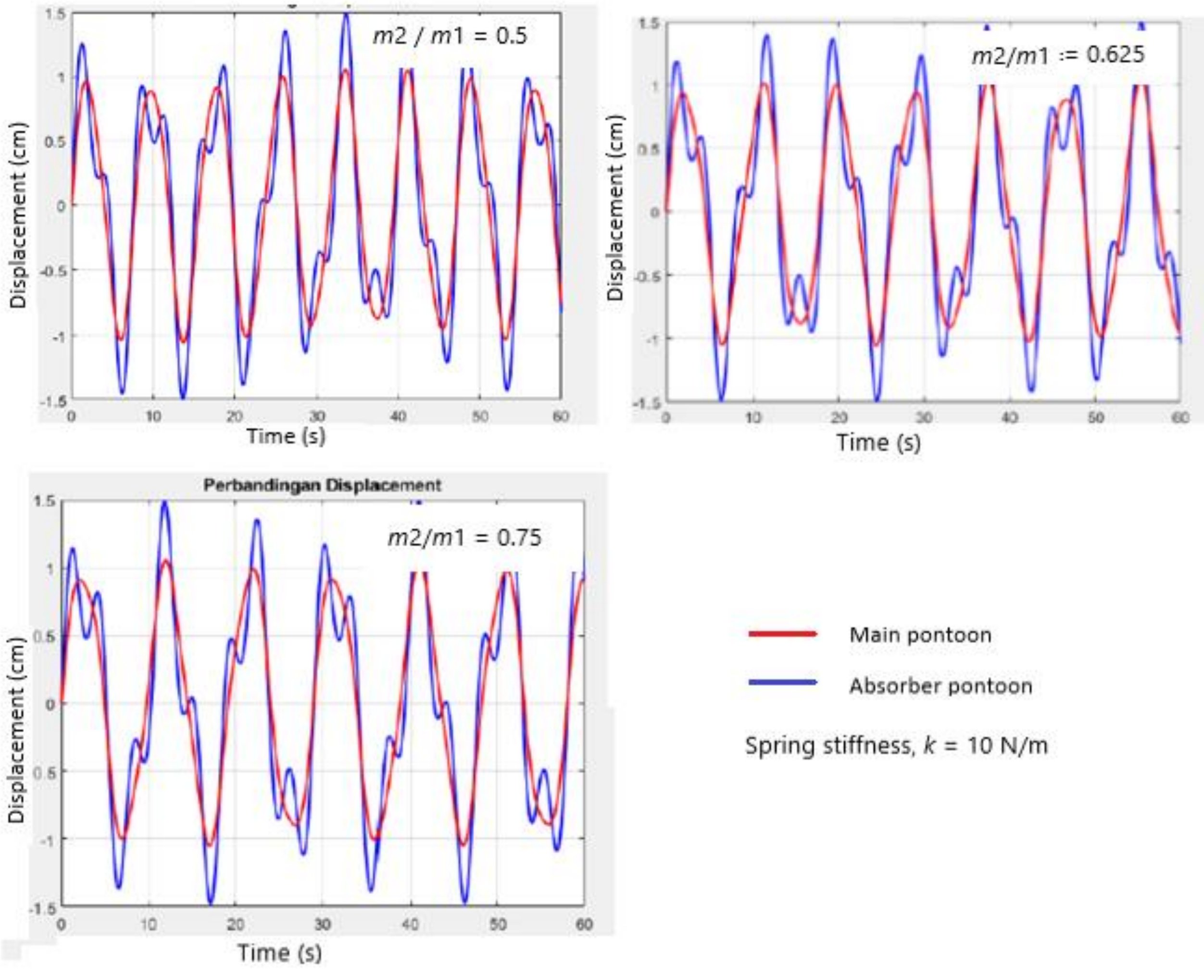


Figure 6 The effect of mass ratio (m2/m1) on displacement

In general, the mass ratio has a significant influence on the displacement amplitude. At a lower ratio (m2/m1=0.5), the main pontoon exhibits relatively higher amplitudes compared to the other cases. This is because the absorber pontoon, with smaller mass, is less effective in absorbing wave energy, leaving a larger portion of the excitation to be carried by the main pontoon.

When the mass ratio is increased to ($m_2/m_1$=0.625), the amplitude of the main pontoon decreases. This indicates that the larger absorber pontoon mass is able to act more effectively as an energy absorber, leading to a more balanced distribution of wave-induced energy between the pontoons.

At the highest ratio ($m_2/m_1$=0.75), the displacement amplitude of the main pontoon is further reduced, while the amplitude of the absorber pontoon increases. This highlights the trade-off in a two-degree-of-freedom system, where a higher absorber mass reduces the response of the primary structure but increases the response of the absorber itself.

These results confirm that the mass ratio ($m_2/m_1$) is a critical parameter in the design of double-pontoon TLPs. A larger ratio enhances the effectiveness of the absorber pontoon in reducing the motion of the main pontoon. However, determining the optimum mass ratio requires careful consideration of the balance between reducing the main pontoon's displacement and maintaining the stability of the absorber pontoon.

## 5. Conclusion

Based on the numerical analysis and experimental validation of the double-pontoon Tension Leg Platform (TLP), the following conclusions can be drawn:

1. The double-pontoon configuration is effective in reducing the displacement amplitude of the main pontoon compared to a single-pontoon TLP, due to the redistribution of wave-induced energy between the pontoons.

2. Spring stiffness (k) has a significant influence on the dynamic response. Increasing the stiffness reduces irregular oscillations, but excessively high stiffness leads to a higher amplitude of the main pontoon. This indicates the existence of an optimum stiffness range for effective vibration mitigation.

3. The mass ratio between the absorber pontoon and the main pontoon (m2/m1) is a critical parameter for stability. A larger ratio enhances the absorber's effectiveness in reducing the motion of the main pontoon, but also increases the displacement response of the absorber pontoon itself.

4. The dynamic interaction between the main and absorber pontoons demonstrates a trade-off mechanism similar to tuned mass absorbers, where optimum performance is achieved through proper tuning of both spring stiffness and mass ratio.

5. These findings provide new insights into the design of double-pontoon TLPs for offshore applications, particularly in enhancing stability under wave excitation. Further research may explore optimization methods and experimental validation under irregular and extreme wave conditions.

6. Future work will include validation under irregular and multi-directional wave conditions to confirm the robustness of the double-pontoon configuration

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