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Experimental Analysis based on Strip Theory Method for Pitch Hydrodynamic Coefficients

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Abstract. In this work, we focus on predicting ship motions using the strip theory optimization method. This method served to determine the hydrodynamic coefficients for pitch motion. Several experimental tests on a tanker ship model have been carried out in the department of maritime engineering, University of Sciences and Technology of Oran towing tank, with a chosen encounter frequency for head waves. Comparative validations of obtained results with experimental data are given showing an acceptable correlation.

Keyword: Marine vehicles, Hydrodynamic coefficients, Pitching motion, Added mass, Strip theory method, Maxsurf.

INTRODUCTION

The study of ship-wave interaction has recently been developed, as it is known that this interaction can generate nonlinear effects on the ship's hull. Nonlinear effects play a crucial role in ship design, encompassing various phenomena such as over rolling (Perez and Blanke, 2010), slamming (Hoque, 2014), water on deck (Spanos et al., 2002), whipping (Tuitman and Malenica, 2008), and wave breaking (Fu et al., 2014) experienced by a ship. These effects have the potential to impact ship performance significantly. Consequently, it is imperative to design ships with a thorough consideration of nonlinear effects to prevent accidents. Predicting ship response is a critical aspect of ship design (Baso et al., 2013). While many ship motion predictions rely on linear and simplistic assumptions, mainly applicable to small-amplitude motions that can be modeled as linearly uncoupled or coupled modes, it is essential to acknowledge that a ship's position and orientation can also be influenced by hydrodynamic, hydrostatic, and propulsion forces and moments.

Stability tests are often conducted on ship models, and sometimes on actual ship hulls, to validate the theoretical results through experimental analysis. Accurate prediction of ship motion is vital as it directly affects the ship's design, operation costs, and overall safety (Suleiman, 2000).
Due to the successful outcomes achieved through model testing, 'experimental' series of hull forms have been devised, incorporating diverse hull parameters. These series were built based on a "favorable" hull form serving as the parent design (Bertram, 2011).
The roots of model tank testing can be traced back to William Froude's influential works in the 1860s, aimed at maximizing ship stability (Peña et al., 2013). Subsequently, the first ship model basin was constructed in 1883 by the Shipbuilding Company William Denny and Brothers, following Froude's successful contributions. This basin enabled various ship models to undergo hydrodynamic tests in a towing tank, leading to improved ship performance and refined designs.
In the 1930s, Ken Davidson at the Stevens Institute of Technology effectively resolved the challenge of using scale model tank testing to forecast the upwind performance of sailing yachts (DeBord et al., 2004). This test primarily aimed to demonstrate advancing resistance due to wind forces, thereby improving sail performances. Ship performance prediction can be categorized into key areas, including resistance, propulsion, seakeeping, and manoeuvring (Bertram, 2011). Various facilities exist for model testing, such as towing tanks for resistance and propulsion tests, open water testing of propellers, manoeuvring, and seakeeping tests. Additionally, cavitation tunnels are used for testing propeller cavitation, hull pressure fluctuations, and propeller noise measurement.

Several methods are available for studying and predicting ship motions, such as time-domain analysis (Liu et al., 2014), strip theory, and predictions based on system identification (Barrass, 2004). Strip theory is widely utilized for determining the parameters of ship motion equations due to its simplicity and computational efficiency. Typically, strip-theory calculations yield satisfactory results for slender body ships with small amplitude motions, where nonlinear and three-dimensional effects are negligible (Salvesen et al., 1970).
This work represents a study predicting ship motions using the strip theory approach to determine the hydrodynamic coefficients of a tanker ship model. These coefficients are then used to derive the ship's motion equations. The focus of this study is specifically on uncoupled pitch motion. The experiment was conducted at the Towing Tank of the Maritime Engineering Department, University of Sciences and Technology of Oran, Mohamed Boudiaf. To conduct hydrodynamic tests with ship models, a modular, FFT-based system for signal and system analysis was utilized. The investigation also explored the time capture measurement mode for acceleration, velocity, and position.
Finally, the obtained results based on strip theory are validated using experimental data.

STRIP THEORY

Strip theory, rooted in fluid mechanics, was initially applied to floating bodies by Krylov in 1896 (McCormick, 2009), and later revised by Korvin Kroukovsky in 1955 (McCormick, 2009). Due to its simplicity and computational efficiency, it has become the most widely used approach for determining ship motion equation parameters (Suleiman, 2000). Further refinements were made by Korvin-Kroukovsky and Jacob in 1957 (McCormick, 2009; Journée, 2001), and the relationship between diffraction and radiation potentials was established by Haskind in 1957, confirmed by Timman and Newman in 1962 (Journée, 2001). Motora made subsequent modifications to this work in 1964, focusing on applying strip theory to planar ship motions in regular seas. Salvesen, Tuck, and Faltinsen expanded the theory in 1970, and a comprehensive review of the linear strip theory was provided by Bishop

and Price in 1979. Nonlinear strip theories have also been developed, including the "quadratic theory" proposed by Jensen and Pedersen in 1978 (McCormick, 2009).
Strip theory calculations generally yield satisfactory results for slender body ships with small amplitude motions, where nonlinear and three-dimensional effects are negligible (Salvesen et al., 1970). However, it is important to recognize the limitations and disadvantages of strip theory, including its inherent 2D approach, reliance on linear assumptions, and simplifications of boundary conditions. In strip theory, the 3D ship motion coefficients are expressed in terms of integrals of 2D sectional coefficients (Suleiman, 2000).
The strip theory treats the ship as a finite number of cross-sectional sections rigidly connected together, although their shapes may not closely resemble the actual ship segments. Each section is treated as if it were part of a floating infinitely long cylinder, neglecting the complex wave systems around high-speed vessel hulls, which can be divergent and unstable. This method is based on the assumption of linearity, meaning that the vessel's movements are expected to be small compared to its size. It focuses on the hydrodynamic effects of the hull below the waterline when the vessel is motionless, simplifying the analysis by considering forces acting on slender strips of the hull and ignoring complex nonlinearities and wave systems that arise during motion.
Furthermore, strip theory does not differentiate between different hull shapes in varying surface conditions, neglecting additional resistance caused by waves, which is proportional to the square of relative movements. Inaccuracies in the relative movements between cross-sections, often overlooked, can significantly impact result accuracy. Strip theory is valid only for long, slender bodies, but despite this limitation, experiments have shown that it can be successfully applied to floating bodies where $L/B > 3$.

SHIP MOTION FORMULATION

The motion of a ship on the surface of the sea can be described by six distinct movements: three translational motions (surge, sway, and heave) and three rotational motions (roll, pitch, and yaw) (Fig.1).
The heave, pitch, and roll motions of a ship are characterized by oscillations primarily influenced by the restoring force resulting from changes in buoyancy (Seakeeper User Manual, 2007). When responding to waves, the ship's movements can be likened to a forced damped spring-mass system.

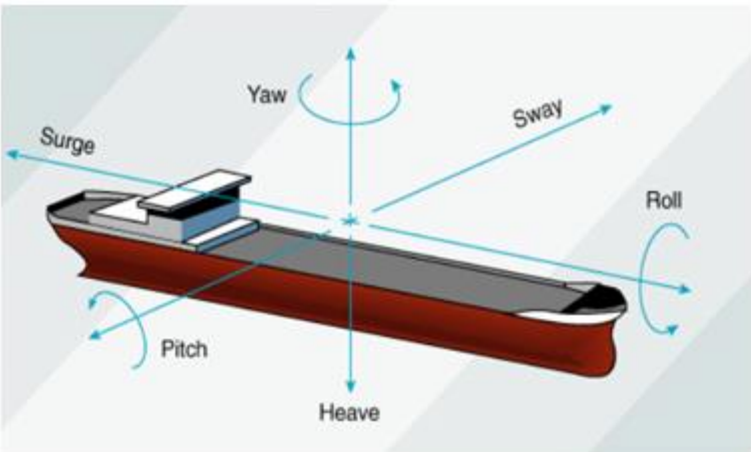

Fig.1. Ship motions.

In this study, our focus is exclusively on the non-coupled pitch motion, which refers to the rotational motion around the transverse axis. During pitch motion, the ship undergoes alternating trims, tilting either by the bow or by the stern. The four main moments at play in this motion are the inertial moment, damping moment, restoring moment, and exciting moment. The equation of motion for pitch is as follows:

$$a\frac{d^2\theta}{dt^2}+b\frac{d\theta}{dt}+c\theta=M_0\sin\omega_e t \quad (1)$$

Where:

- $\left(a\frac{d^2\theta}{dt^2}\right)$ represents the Inertial moment with "a" denoting the virtual mass moment of inertia, and $\frac{d^2\theta}{dt^2}$ representing the angular acceleration of pitching.
- $\left(b\frac{d\theta}{dt}\right)$ denotes the Damping moment with "b" is the damping moment coefficient, and $\frac{d\theta}{dt}$ represents the angular velocity.
- $(c\theta)$ The Restoring moment is represented by "c" as the restoring moment coefficient, and "θ" is the angular displacement of pitching.
- $(M_0\sin\omega_e t)$ The exciting moment, were "M_0"represents the amplitude of the exciting moment, "ω_e" is the encounter frequency and "t" is the time.

EXPERIMENTAL ANALYSIS

The experimental work was conducted in the tank of the Maritime Engineering Department (Fig.2). In the following section, a concise presentation and description of the experiment will be provided. This will include an overview of the experimental setup, the objectives of the experiment, and the methods employed.

Fig.2. The Model used in the experiment.

Ship model

In this experiment an oil tanker model used, which has the following characteristics:

Table 1. Model characteristics.

Symbol (unit)	Signification	Value
L_{pp}(m)	Model length	1.43
B_{max} (m)	Breadth max	0.206
T_{max}(m)	Draught	0.066
Δ(Kg)	Displacement	15.73

C_b	Bloc coefficient	0.786
C_w	Prismatic coefficient	0.799
K_B	Height of hull center	0.0343
K_G	Height of gravity center	0.060
S “$s = L_{pp} / (n-1)$”	Interval between sections	0.143

The ship model tank (Fig.3)

The main dimensions of the tank are:

- Length: 20 m.
- Breadth: 2 m.
- Depth: 1 m.

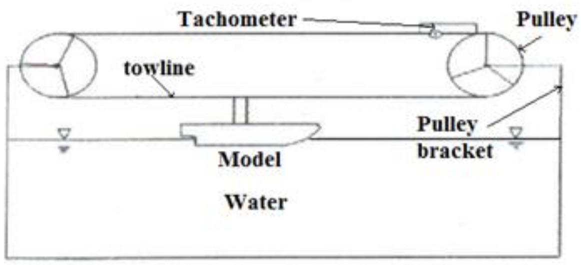

Fig.3. General view of the towing tank.

Wave generator

The wave generator is of the triangular profile plunger type, which performs vertical oscillations. In the experiment, the plunger arm is driven by an electric motor rated at 1.5 kW. The stroke of the plunger is 140 mm maximum.

Signal Analyser Unit–Type 2035 (Fig.4)

The Signal Analyser Unit Type 2035 is a central mainframe unit that includes a 12" raster scan screen, a disc drive, and a keyboard. It serves as the housing for the signal and display processors, as well as the required memories and hardware essential for analysis and system control. This unit plays a crucial role in signal analysis by providing the necessary tools and functionality for data processing and visualization.

This unit, in our case, showcases the frequency of the model and the wave generator using two impedance heads of type 8000. One of these heads detects the vertical motion, while the other detects the horizontal motion.

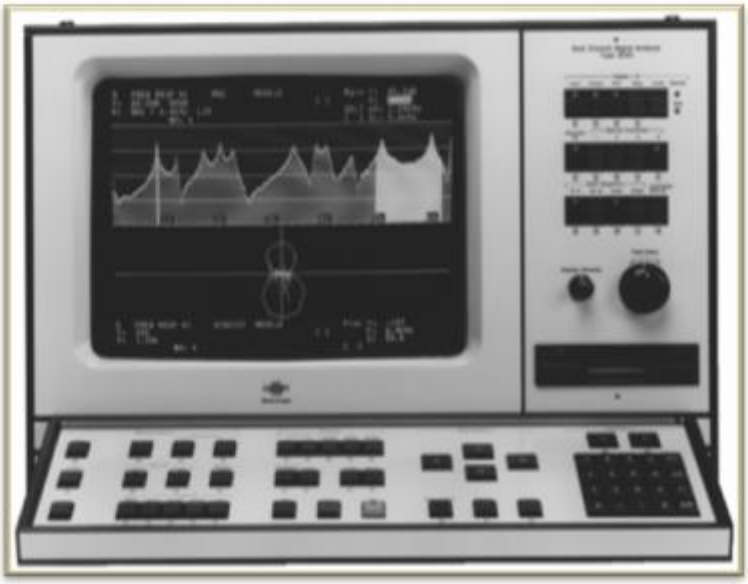

Fig.4. Signal Analyzer Unit–Type 2035 Brüel&Kjær.

Impedance Head type 8000 (Fig.5)

The Impedance Head Type 8000 comprises a piezoelectric accelerometer mounted on a force gauge, enabling the simultaneous measurement of both force and acceleration at a single point.

Impedance heads present a straightforward approach to measuring point mechanical mobilities and impedances.

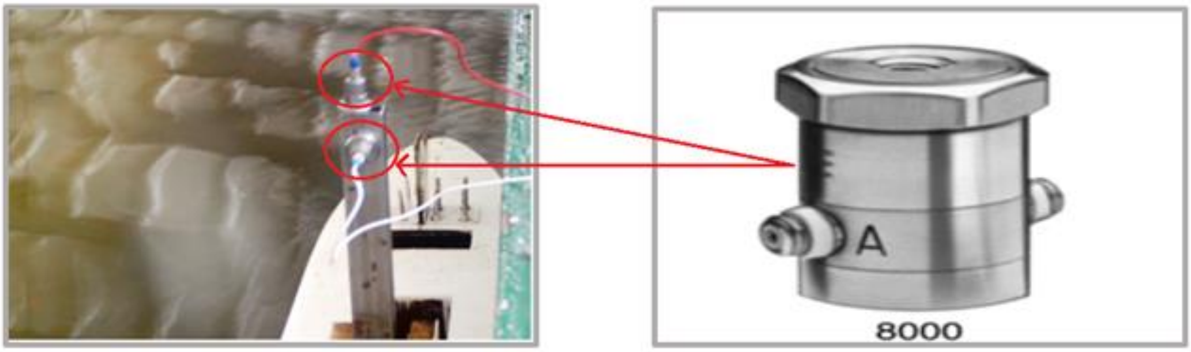

Fig.5. Impedance head on the ship model.

In this experimental investigation, two impedance heads are mounted on the ship model, one in the vertical direction and the other in the horizontal direction. Both are connected via wires to the Signal Analyser Unit.

DETERMINATION OF PITCHING MOTION COEFFICIENTS USING STRIP THEORY APPROACH

Strip theory, also known as 2D potential theory, is employed in ship hydrodynamics to calculate potential coefficients, such as added mass and potential damping, and wave loads, including Froude-Krylov and diffraction forces. In this method, the ship is divided into 20 cross-sections, and its two-dimensional nature allows for the consideration of three degrees of freedom: vertical or heave, horizontal or sway, and rotational about a horizontal axis or roll. Fossen and Smogeli (2004) have developed a computationally efficient nonlinear time-domain strip theory formulation, specifically for dynamic positioning (DP) and low-speed maneuvering. The following assumptions are made:

- Potential Flow: In ship hydrodynamics, the fluid flow around ships is commonly assumed to be potential, indicating that it is both inviscid and irrotational. This assumption is justified by the fact that the impact of viscous damping on ship motions is generally considered to be negligible.
- Negligible Surface Tension: The effects of surface tension on the fluid flow are considered to be negligible and can be disregarded.
- Irrotational Flow: The fluid is assumed to be irrotational, implying that there are no vortices or rotational effects present.
- Small Amplitudes and Velocities: The motion amplitudes and velocities are assumed to be sufficiently small, allowing for the consideration of linear terms exclusively. Nonlinear terms present in the free-surface condition, kinematic boundary condition on the cylinder, and Bernoulli equation can be disregarded.

This method consists of determining a coefficient for each section, then integrate it over the entire length of the ship.

For ship model design, the Maxsurf packages are utilized, which employ a strip theory algorithm. Maxsurf is a highly robust three-dimensional surface modeling software extensively utilized in marine design applications. It enables the creation of multiple surfaces to model various aspects of the design. With its integrated hydrostatic calculations, designers can experiment with different shapes and explore various design parameters. For specific details and visual representations, please refer to table 2, figures 6 and 7.

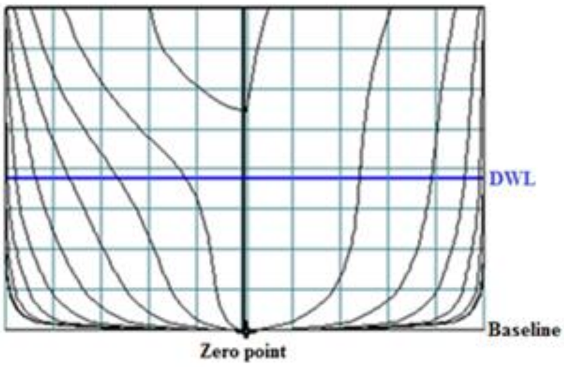

Fig.6. Transversal side body plan.

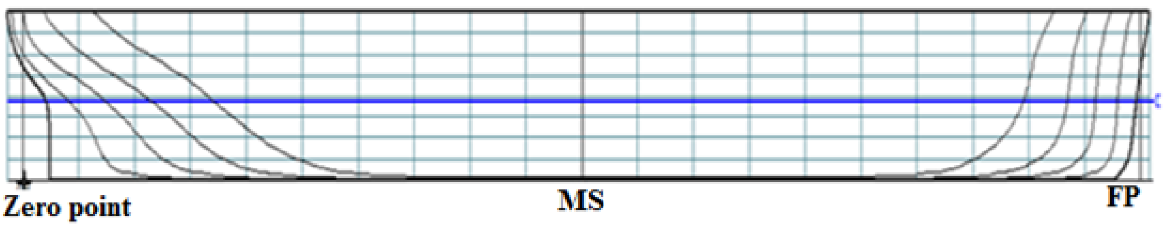

Fig.7. Longitudinal side body plan.

Table 2. Hull characteristic from Maxsurf.

Displacement	15,60	kg
Volume (displaced)	0,015	m^3
DraftAmidships	0,066	m
Immerseddepth	0,066	m
WL, Length	1,441	m
Beam max extents on WL	0,208	m
Wetted Area	0,395	m^2
Max sect. Area	0,013	m^2

Waterpl. Area	0,253	m^2
Prismaticcoeff. (Cp)	0,793	
Block coeff. (Cb)	0,770	
Max Sect. Area coeff. (Cm)	0,970	
Waterpl. Areacoeff. (Cwp)	0,844	
LCB length	0,030	from zero pt. (+vefwd) m
LCF length	0,018	from zero pt. (+vefwd) m
LCB %	2,114	from zero pt. (+vefwd) % Lbp
LCF %	1,228	from zero pt. (+vefwd) % Lbp
KB	0,035	m
BMt	0,052	m
BML	2,178	m
GML	2,147	m
Immersion (TPc)	0,003	Tonne/cm

The virtual mass moment of inertia for pitching is considered as:

$$a = I'_{yy} = I_{yy} + \delta I_{yy} = \frac{\Delta}{g} K_{yy}^2 + \delta I_{yy} \quad (2)$$

Where δI_{yy} is the added mass moment of inertia for pitching and K_{yy} is radius of gyration for pitching

- The inertial moment of the added mass for pitching movement is denoted as a_{55} which is defined as:

$$a_{55} = \int_{-l/2}^{l/2} a_n \xi^2 d\xi (3)$$

Where:

"a_n" Is the added mass for each section and "ξ" is the distance of the considered section from the longitudinal center of gravity.

- Damping coefficient

$$b_{55} = \int_{-l/2}^{l/2} b_n \xi^2 d\xi (4)$$

Where:

"b_n" Is denote the damping coefficient for each strip along the ship's length and "ξ" the distance of the individual strip from the longitudinal center of gravity.

- Restoring coefficient

$$c_{55} = \int_{-l/2}^{l/2} c_n \xi^2 d\xi (5)$$

Where:

"c_n" is the restoring moment coefficient, and "ξ" the distance of the individual strip from the longitudinal center of gravity.

Encounter frequency

The velocity "V_m" of the wave, which is defined in a direction at an angle (referred to as the wave direction) relative to the ship's speed vector, can be represented by the following equation:

$V_m = \frac{\omega_w}{k}$ (6)

Or:

$V_m = \frac{\lambda}{t}$ (7)

Where:

"ω_w "Characterizes the circular frequency of the wave *(rad/sec),* "λ" corresponds the wave length, "k" represents the wave number and "t" is time.

The coordinate system O *(x, y, z)* advances to the speed of the vessel, which can be given in the form of the following equation:

$x_0 = \upsilon t \cos\mu + x\cos\mu + y\sin\mu$(13)

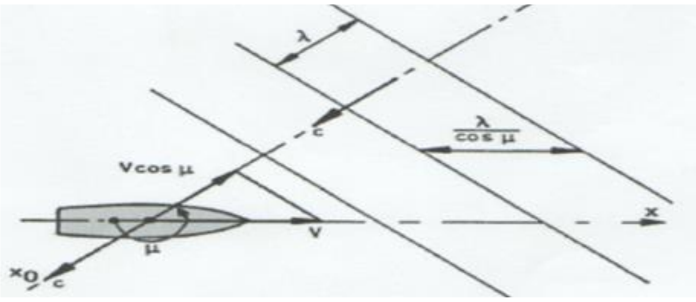

Fig.8. Encounter frequency.

When the ship moves forward, the frequency with whic.h it meets waves ω_e becomes significant, its period can be determined using the following expression:

$$T_e = \frac{\gamma}{c + \upsilon\cos(\mu - \pi)} = \frac{\gamma}{c - \upsilon\cos\mu} \text{(8)}$$

$$\omega_e = \frac{2\pi}{T_e} = k(c - \upsilon\cos\mu)\text{(9)}$$

The encounter frequency and wave frequency are related to each other in the following expression:

$\omega_e = \omega_w - k\upsilon\cos\mu$(10)

Results and discussion

The following figures (Fig.9) represent the hydrodynamic coefficients for the pitching motion, namely the Added Mass Coefficient, the Damping Coefficient using strip theory, and the Restoring Moment for the pitching motion using the strip theory method:

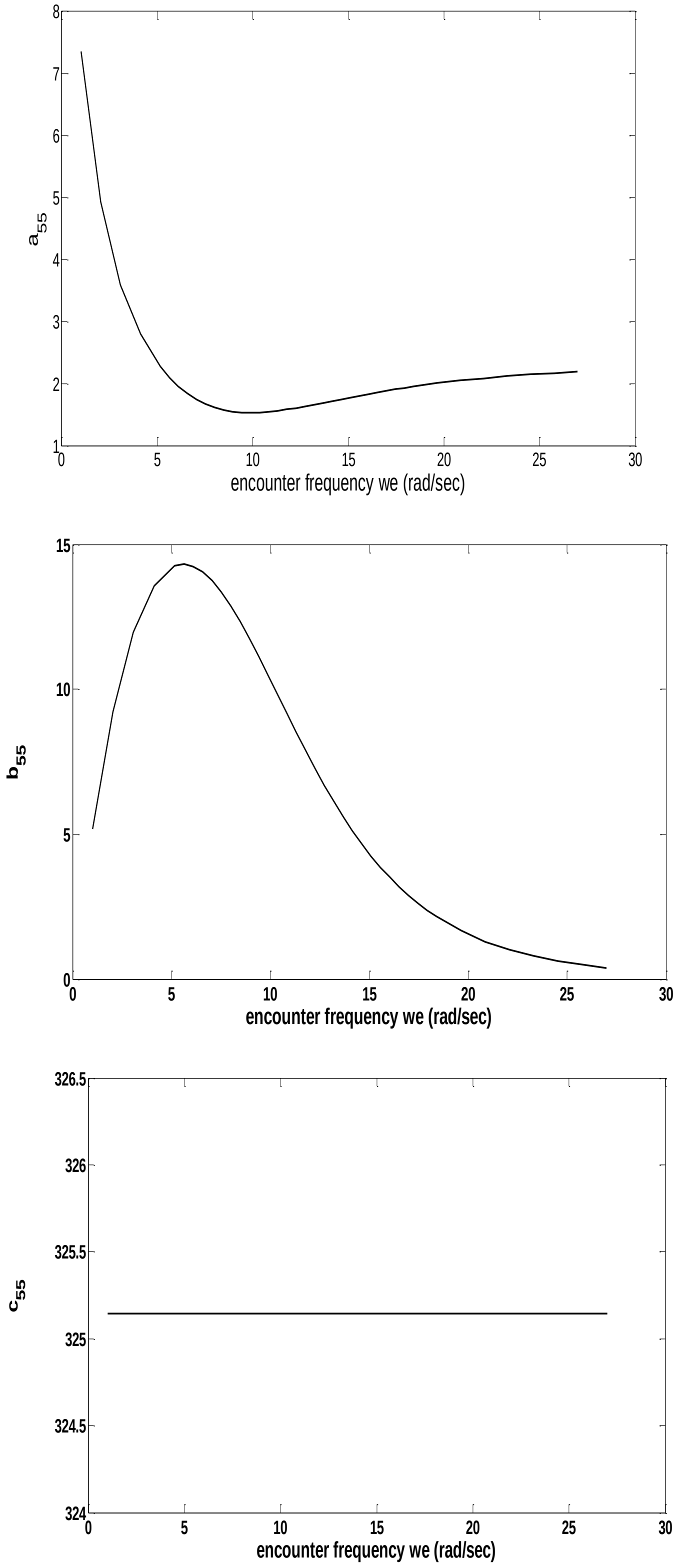

Fig.9. Hydrodynamic coefficients (a55, b55, c55).

For the case of head seas μ =180° and an encounter frequency ωe = 6.565 rad/sec, the hydrodynamic coefficients obtained from Strip theory method are: a_{55} = 1.842, b_{55} = 14.259, c_{55} = 320.244.

The initial conditions for strip theory were taken null while the experimental data begin with $[\theta_0; \dot{\theta}_0] = [0.0192; -0.011]$, μ = 180°, ωe= 6.565Hz and Iyy= 0.1902. The results are given in figure 10 and 11, and there is an indication where the pitch motion and velocity show a good correlation.

The position is presented in Figure 10 and the velocity is presented in figure 11. It is noticed that there is a phasing difference between experimental signal and strip theory results; this is due to the coupling pitch and heave motion which are not taken into consideration in strip theory.

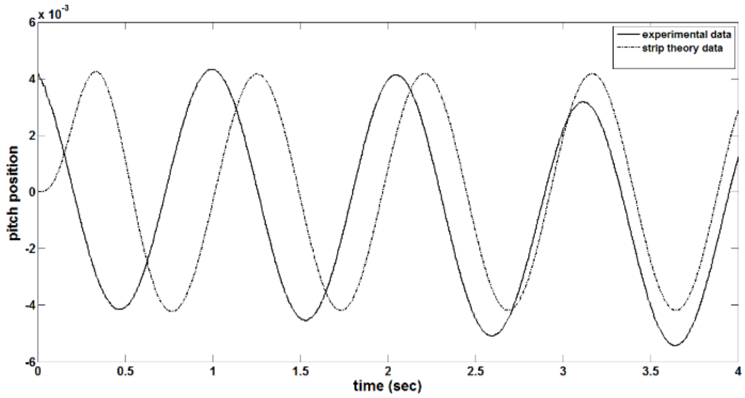

Fig.10. Position for pitch motion.

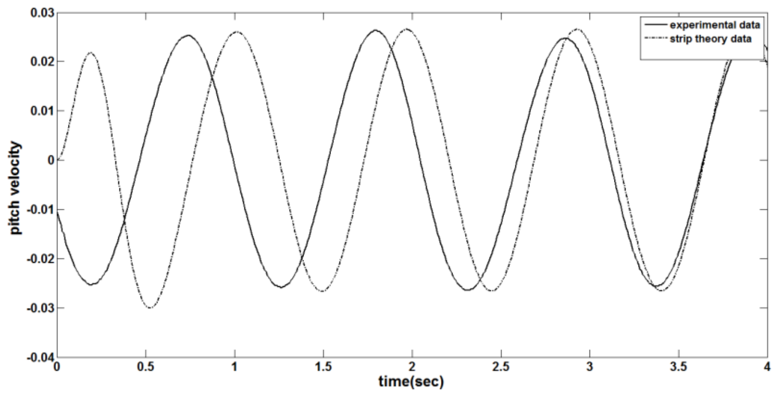

Fig.11. Velocity for pitch motion.

CONCLUSION

This article presents a strip theory formulation that utilizes experimental data to calculate hydrodynamic coefficients for the uncoupled pitching motion at zero forward speed. The technique shows a satisfactory agreement with the experimental data when appropriate constraints are applied, indicating its effectiveness and reliability in capturing the behavior of the system.

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