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DEVELOPMENT OF AN OSCILLATION CONTROL SYSTEM TEST BENCH FOR RIVERBOAT

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Abstract. *The aim of this article is to present the development of a test bench that simulates oscillations of boat models, their instrumentation and the tests of an active system. The advantage of developing the experimental platform is that it does not require a test tank and the manufacture of a reduced the vessel model to perform experiments. This simplifies testing procedures and reduces costs.*

Keywords: *test bench, boat simulation, oscillation control*

1. INTRODUCTION

In the Amazon region of Brazil, transport is mostly done by waterways, as the Amazon forest hinders and limits transportation by land. Most of this transportation is made by typical boat with a wooden hull, shallow draft and raised deck structure as shown in Fig.1.

Figure 1. Amazonian typical boat (Soares and Vidal Filho, 2014)

The Amazonian typical boat has a specific design and carries both passengers and cargo (Vasconcelos and Nascimento, 2010; Soares and Vidal Filho, 2014). The importance of these boats can be evaluated by the number of passengers and cargo that they carry in the Amazon region annually: about 8.8 million passengers and 4.5 million tons of cargo (ANTAQ, 2014). The nautical accidents with the highest percentage of fatal victims are related to this type of boat, considering that they are responsible for the majority of the transport of the region. According to the data provided by the region's naval authority, this vessels wreck accounted for 31% of fatalities between 2001 and 2003 (Soares and Vidal Filho; 2014). Magalhães et *al*. (2015) observed through a statistical study of the judgments of the Maritime Court that the typical Amazonian boat accounted for 40.8% of fluvial accidents between 2011 and 2012. A study of the type of accident that most victimized people was not done. Tipping occurs less frequently than collisions with other boat or with submerged objects, however, when a sudden tipping occurs, passengers do not have time to put on life jackets and leave the boat safely. In order to contribute to safety against boat tipping due to dynamic factors, some studies have been done. In Soares and Vidal Filho (2014) the dynamic characteristics of the vessels were studied, the static stability curves and the natural frequency of the vessel for transverse roll oscillations were determined. In Rylo et *al*. (2016) it was studied the use of a passive system for these vessels, meanly u-shaped tanks with water, where the fluid oscillates out of phase with the boat, mitigating the boat's oscillations at the natural frequency point. In Magalhães et *al*. (2015) the use of Bilge Keels in a riverboat was studied to increase the the hull hydrodynamic drag during roll oscillations.

The Bilge Keels has the advantage of its low cost and the fact that they do not occupy internal space in the boat. The Paravanes are other passive systems that also use the hydrodynamic drag to damp the oscillations. They are "glider" shaped elements that are dragged submerged on each side of the vessel by cables hanging from side masts (Bass, 1998). Though if one of these cables break, this solution became a hazard to the boat stability. Figure 2 shows the three major passive systems.

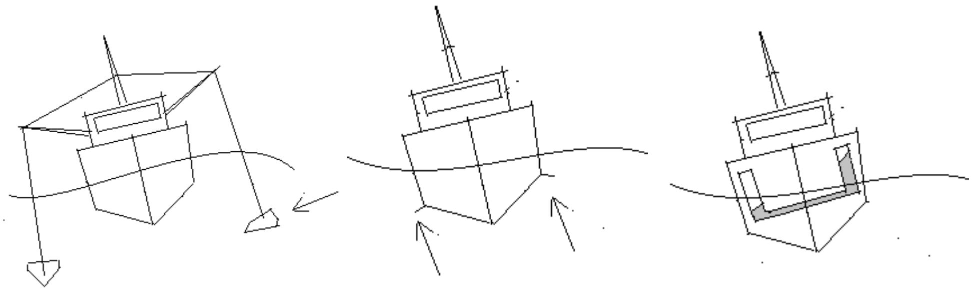

Figure 2. (a) Paravanes (b) BilgeKeels (c) U-Tank stabilization

In Bass (1998) the stabilization tanks and Paravanes were compared in small fishing vessels. The Bilge Keels provide a reduction of 30 ~ 40% of the amplitude of the ship oscillations (Windén, 2009). Stabilizing tanks have two main models, the U-tube and the free surface tank (Windén, 2009). U-tube tanks are more commonly used because they promote better cargo accommodation on the vessel and a mitigation of up to 75% of the oscillation (Windén, 2009; Taskar, Dasgrupt, et *al*., 2014). A tank volume equivalent to 1.5 ~ 5% of the water displacement provided by the hull of the vessel is required for minimum effectiveness (Husick, 2008). A passive system employed in tall buildings that could be employed in the naval area is the mass displacement passive system, known in the academic literature as Tuned Mass Damper (TMD). This type of device consists of a mass, a spring and a damper. In this system the mass and spring are adjusted to remove the maximum oscillating energy of the structure, and this energy is dissipated by the inertia damper (Connor and Laflamme, 2014).

There are active stabilization systems which use electronic system to measure the oscillations and send signals to the elements that act against these oscillations. An example is the Active Fin Stabilizers which are pairs of fins coupled to the side of the vessel hull, acting as a small inverted wing, generating hydrodynamic forces downwards. The angle of attack in relation to the water flow is controllable to generate forces in opposition to the oscillations. It is recognized as the most effective system in its class, with a performance of 70 ~ 90% of attenuation of roll oscillations (Weng, 1995). However it only work with the boat in motion and generate energy consumption by dragging in the water.

In this paper, the development of an experimental assembly to test an active system to mitigate boat transversal oscillations is proposed. A boat dynamics are represented by the torsional pendulum mathematical model which was used for data validation of the test bench. Some experimental results are shown in order to emphasize the efficiency of the proposed platform.

2. MATHEMATICAL MODELING

The simplified mathematical model of a boat oscillating in the water in roll can be approximated to a torsional pendulum representation, as shown in Fig. 3. Therefore, a test bench was conceived using a torsional pendulum modeling.

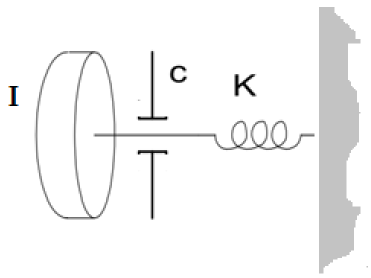

Figure 3. Torsion pendulum representation

The boat oscillating in roll has a similar behavior compared to a torsion pendulum whose model is given according to equation (1):

$$J = \frac{d^2}{dt^2}\varphi + c\frac{d}{dt}\varphi + k\varphi = 0 \tag{1}$$

Correlating equation (1) with the boat parameters results in equation (2):

$$I_{xx}\frac{d^2}{dt^2}\varphi + B\frac{d}{dt}\varphi + \Delta \cdot GZ(\varphi(t)) = 0 \tag{2}$$

Where I_{xx}, B, Δ and the GZ function correspond respectively to the inertia moment of the vessel around the x-axis, the roll-damping coefficient, the weight of the vessel, equivalent to the weight of the volume of water displaced by its hull, and the distance from the point of actuation of the thrust force in relation to the point of application of the force weight, the so-called lever arm. To adjust the bench to the model, the inertia, the friction, and the spring constant must be adjusted to have similarity with the model boat to be simulated.

$$\omega_d = \sqrt{\frac{\Delta \cdot GM_0}{I_{xx}}} \tag{3}$$

$$B = \zeta \cdot B_{CR} \tag{4}$$

$$B_{CR} = 2 \cdot \omega_R \cdot I_{xx} \tag{5}$$

Where ω_d, ω_R, GM_0, ζ and B_{CR} correspond respectively to the angular roll frequency, which can be inferred from the oscillation period of the boat, to the natural angular roll frequency of the boat, to the metacentre height of the boat; to damping factor, approximately 2 ~ 3% at low speed and 5 ~ 6% at high speed, being able to reach 10% with large oscillations (Sheikh, 2008); and the critical damping coefficient of the vessel.

The $GZ(\varphi)$ function can be obtained by:

$$GZ(\varphi) = GM_0 \cdot \sin(\varphi) \tag{6}$$

Considering that for most boats, for the inclination up to 10 degrees, the metacentre height gradient is constant, the GZ function can be approximated by a first-order function as a function of slope and initial metacentre height (Ibrahim and Grace, 2010).

$$GZ(\varphi) \approx GM_0 \cdot \varphi \tag{7}$$

Simplifying equation (2) the equation (7) can be obtained:

$$\frac{d^2}{dt^2}\varphi + 2\zeta\omega_R\frac{d}{dt}\varphi + {\omega_d}^2\varphi = 0 \tag{8}$$

The parameters for simulation were obtained from Soares and Vidal (2014), where the dynamic characterization of a regional boat was made using a reduced model in 1:25 scale. Using the experimental GZ curve of Fig. 4 and equation (8), supposing small inclinations, the metacentre height can be estimated.

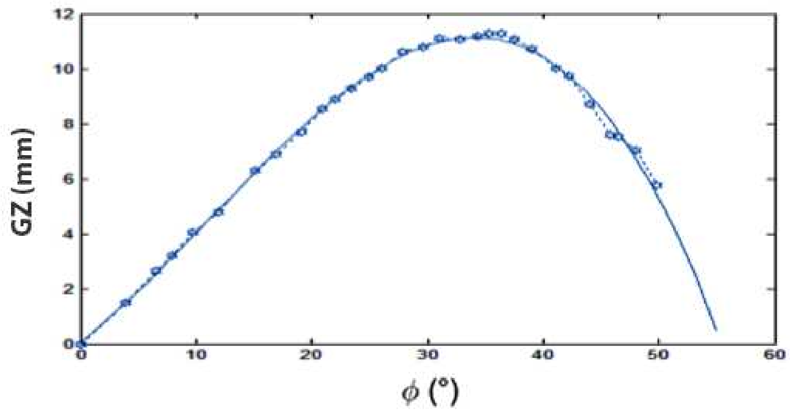

Figure 4. *GZ* experimental curve (Soares and Vidal Filho, 2014)

$$GM_0 \approx \frac{GZ(10°)}{0.174533\, rad} = 0.0229183\, m \tag{9}$$

Replacing the remaining experimental data in the equation described above:

$$\Delta = 6.45 kg \cdot 9.81 \frac{m}{s^2} = 63.2745 \frac{kg \cdot m}{s^2} \tag{10}$$

$$\omega_d = \frac{2\pi}{1.1\, s} = 5.7120\, rad/s \tag{11}$$

$$I_{xx} = 0.0444\, kg \cdot m^2 \tag{12}$$

$$B_{CR} = 0.5084 \frac{kg \cdot m^2}{s} \tag{13}$$

The mechanical bench system can be designed with the above parameters. The stabilization system chosen for the analysis was modeled. It is intended to test a mass displacement stabilization system, Active Mass Damper (AMD), as shown in Fig. 5. A mass is moved on rails from one side of the vessel to the other, generating a contrary torque to the oscillation movement, reducing oscillation.

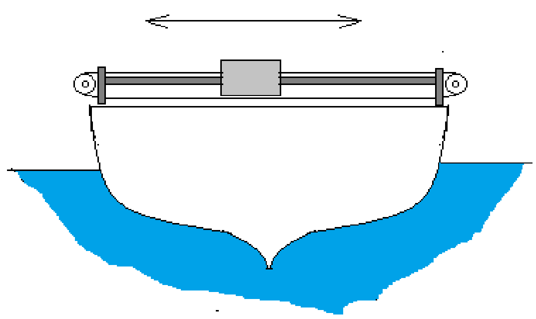

Figure 5. Proposed mitigation system

The AMD system is commonly utilized in civil engineering applications, acting in the suppression of vibration caused by earthquakes and wind loadings (Ikeda, et *al.*, 2001).The system modeling can be achieved from the sum of the forces (Ikeda, et *al.*, 2001):

$$m_a \frac{d^2}{dt^2} x_a(t) = u(t) \tag{14}$$

Where x_a, m_a, and u correspond respectively to the car linear displacement above the rails, the car weight with the coupled load, and the control force imposed by the control system. A more advanced model is proposed in Ikeda, et *al*., 2001, in which a Columb friction is also evaluated between the car and its supporting rails, however, due to the project scale and the magnitude of that friction, such model was not considered.

Through the angle of rotation of the motor θ_m measured by the encoder sensor, and the transmission ratio of the gear set, represented by the constant λ, it is possible to determinate the linear displacement x_a of the car and its load.

$$x_a(t) = \theta_m(t) \cdot \lambda \tag{15}$$

Replacing the equation (15) into equation (14), can be obtained:

$$u(t) = m_a \cdot \lambda \frac{d^2}{dt^2}\theta_m(t) \tag{16}$$

For equating the mechanical torque, it is considered that the DC motor has a rotating shaft with an inertial moment J_m, and internal friction characterized by a constant B_m. The external torque component due to the car coupling is described by the relation of the control force $u(t)$ and one of the gears from the transmission gear set of diameter D, in which the belt connected to the car makes contact.

$$\tau_m(t) = u(t) \cdot \frac{D}{2} + J_m \frac{d^2}{dt^2}\theta_m(t) + B_m \frac{d}{dt}\theta_m(t) \tag{17}$$

As can be seen in the Eq. (17), there wasn't any torque component of the transmission gear set, this occur due to the utilization of a small and low weight gears. As such, replacing the Eq. (16) in the Eq. (17), the information below can be obtained:

$$\tau_m(t) = (J_s + J_m)\frac{d^2}{dt^2}\theta_m(t) + B_m \frac{d}{dt}\theta_m(t) \tag{18}$$

$$J_s = \frac{D \cdot m_a \cdot \lambda}{2} \tag{19}$$

The term J_s in the equation (18) can be interpreted as the perceived inertial moment of the AMD system coupling to a DC motor. The same equating of the torque can be achieved utilizing the electrical model of the motor DC. Therefore, utilizing the conventional equation with the Laplace Transformation, Eq. (20) can be obtained:

$$\tau_m(s) = K_\tau \frac{E_a(s) - K_m \cdot \theta_m \cdot s}{L_m \cdot s + R_m} \tag{20}$$

Where the parameters K_τ, K_m, E_a, L_m and R_m of the Eq. (20) are respectively the torque constant, the motor constant, the armature voltage inductance and the resistance of the armature circuit of the Motor DC. Correlating Eq. (18) and Eq. (19), the transfer function of this system can be obtained:

$$\frac{\theta_m}{E_a}(s) = \frac{K_\tau}{L_m(J_m + J_s)s^3 + \big(B_m L_m + R_m(J_m + J_s)\big)s^2 + (B_m R_m + K_\tau K_m)s} \tag{21}$$

Thus, it remains only perform the mitigation system's influence equating on the boat oscillations. The influence can be understood as an external torque and described by the resulting torque due to the positioning of the actuating load on the rails and the system's structure weighting relation to the center of rotation of the oscillation. However, in order to simplify the equation, weights were added to cancel out the torque component due to the structure's weight. Thus, the oscillation of the coupled AMD system can be obtained from the geometric analysis of the installation position of the mitigation system:

$$I_{MS}(t)\frac{d^2}{dt^2}\varphi = -m_a \cdot g\big(x_a(t) \cdot cos(\varphi) + d \cdot sen(\varphi)\big) \tag{22}$$

$$I_{MS}(t) = I_{EST} + I_{LOAD}(m_a) + m_a(x_a(t)^2 + d^2) \tag{23}$$

Where I_{MS}, I_{EST}, and I_{LOAD} in the equation 23 are respectively the inertial moment of the mitigation system, its structure component, and the load component. The remain unexplained component in the equation 23 comes from the utilization of Steiner's theorem to determine the inertial moment due to the load linear dislocation in the rails.The parameter d corresponds to the distance between the center of rotation of the boat and the car's rail.

Mixing the Eq. (2) and the Eq. (23), the final control equation can be obtained:

$$I(t)\frac{d^2}{dt^2}\varphi + B\frac{d}{dt}\varphi + \Delta \cdot GZ(\varphi(t)) = -m_a \cdot g\big(\theta_m(t) \cdot \lambda \cdot cos(\varphi) + d \cdot sen(\varphi)\big) \tag{24}$$

$$I(t) = I_{xx} + I_{MS}(t) \tag{25}$$

3. BENCH DESIGN

The idea to build up a test bench arises from the necessity to develop a dynamic system model similar to the Amazonian boat without the costs of building reduced boat models and an instrumented testing water tank. Naval applications havealways used instrumented test tanks, with mechanisms to produce waves and sensors to measure the behavior of the reduced models of the boats (Clayton and Bishop, 1982). The Fig. 6 shows a reduced boat model being tested in a test tank (Rylo, et *al*., 2016), this model was used as a reference for the bench design.

Figure 6. Boat model in test tank (Rylo, et *al*., 2016)

As mentioned in the previous section, a torsional model was used to represent the roll dynamics of the boat. The moment of restitution can be achieved by springs and the rotational inertia of the model boat can be obtained by the disc of the pendulum. In this scenario, it can be avoided to put together in the same environment water and electronic devices, which drastically reduces the cost and complexity of the experiments. An active vibration absorber system, also called active mass damper, which consists of a mass system with uniaxial displacement, in other words, a mechanism that moves a mass, was also developed and tested on the bench. This mass was moved on a pair of guides through a servomotor. An electronic system was specified to measure the movements and communicate with a computer.

3.1 Mechanical system and prototype

After the mathematical modeling of the boat, a test bench can be designed that physically simulates the behavior of the boat, similar to a torsion pendulum form, allowing experiments with oscillating damping devices. The bench is composed of a disc fixed to an axis, where springs are placed to represent the torques of thrust restitution. With the values calculated from (8) to (12), it is possible to specify the device parameters.

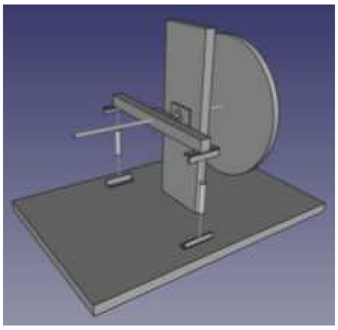

Figure 7. Bench with springs drawing

An initial drawing is seen in Fig. 7, showing where springs are displaced. The mathematical model of Fig. 7 is given in Eq. (26) assuming initially that the friction in the bearings are negligible in relation to the forces involved:

$$I_{Disc}\frac{d^2}{dt^2}\varphi + F_k sen(\varphi) = 0 \qquad (26)$$

Using Eq.(5) it is obtained:

$$\Delta \cdot GM_0 = \frac{K \cdot L^2}{2} \qquad (27)$$

The bench can be adjusted by changing the K of the spring or lengths of the lever arms to obtain the value that satisfies the equality eq. (6). The same simplification done in Eq. (7) can be repeated here.

To adjust the viscous friction it is possible to add a small reservoir of water and a rod that is dragged when the pendulum oscillates. After that, the figure drawing and the model are complete. To adjust the viscous friction, the rod is more or less sunk in the reservoir.

$$I_{Disco}\frac{d^2}{dt^2}\varphi + c\frac{d}{dt}\varphi + \frac{K \cdot L^2}{2}\varphi = 0 \qquad (28)$$

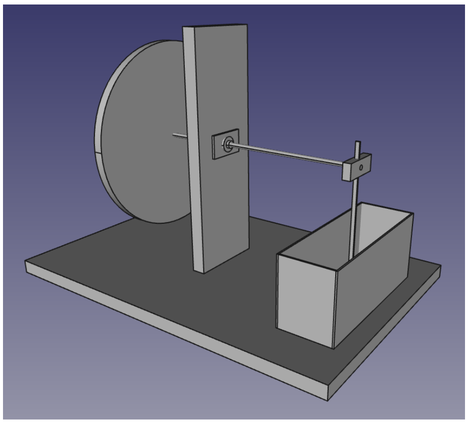

Figure 8. Bench design with viscous friction

Radial holes were made in the disc to adjust the inertia of the disc. Weights were screwed, adjusting the inertia of rotation. Rails and a mobile mass driven by servo motor were set up to represent the stabilization system that was tested.

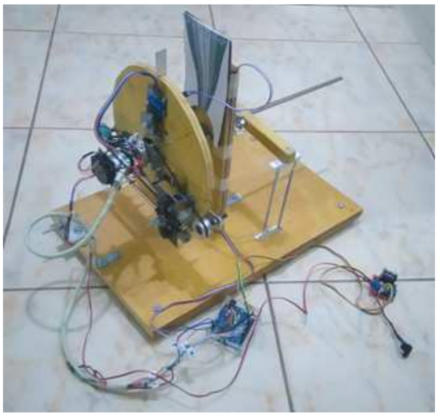

Figure 9. Prototype assembled

The Fig. 9 shows the constructed prototype. The AMD system is assembled on the disc. The inertial sensor can be seen above the linear guides. The servomotor is in the center of the disc.

3.2 Instrumentation

The instrumentation project required four sensors to identify the movement of the actuation system and the pendulum swing for the test bench. The uniaxial displacement of the mass system was determined by the use of an incremental encoder coupled to the motor rotation axis of this system (RH7-1531) with a resolution of 200 pulses per revolution. The DC motor was actuated by a driver board with the IC L298N, allowing the change of direction of the motor rotation and the control of the average voltage supplied, realized by the switching of the input voltage. The positioning calibration was performed by IC F-35 infrared sensors, positioned at the ends of the linear guides and in the central position of the apparatus. The central sensors help to preserve the calibration considering that its activation is used to reset the pulse counter and consequently the identification of the car position. The movement of the pendulum was measured using inertial sensors, a gyroscope and an accelerometer in the InvenSense MPU-6050, implemented in the IC GY-521 board. However, due to the noisy behavior of such sensors, the values of Roll and Pitch angles used in the control were identified from a Kalman filter software implementation. The sensors and DC motor supply cabling had to be assembled separately in order to reduce the electromagnetic influence. Figure 10 illustrates the test bench instrumentation assembly.

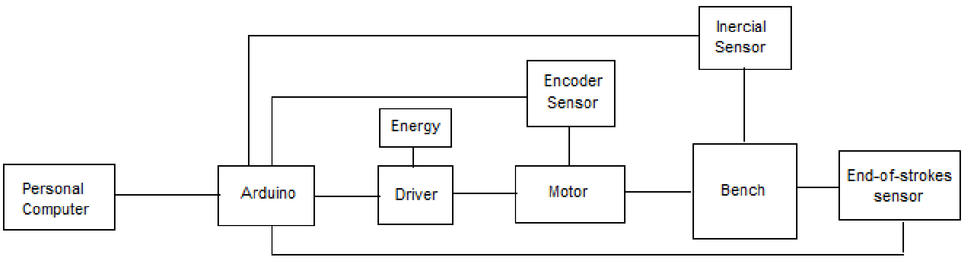

Figure 10. Instrumentation diagram

4. RESULTS ANALYSIS

The scale model data of the boat developed in Soares and Vidal (2014) were used for the experimental validation. The response of this model was compared with the response of the simulated system without the stabilization system, and the obtained results were quite similar, as shown in Fig. 11.The most damped response of the experiment is related to the nonlinearities present in the characterization of the damping moment, where the damping factor ζ used for the estimation of B varies as a function of the slope and angular velocity of the vessel. The discussion of different nonlinear models can be found in Taylan (2000).

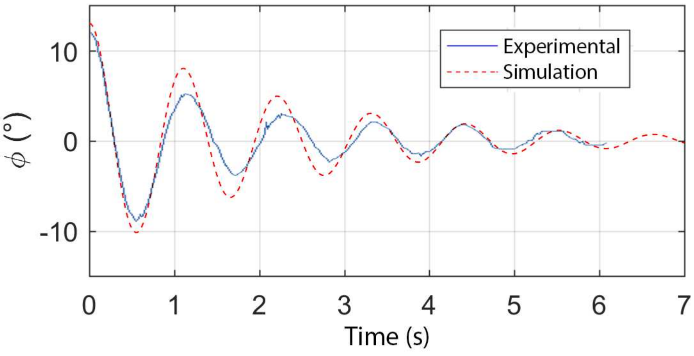

Figure 11. Model validation

Once the model was validated, two simple PID control were implemented to test the effectiveness of the proposed mitigation system. The first one was design to move the car once the boat was rolling and the second was responsible to restore the car to its neutral position. In Fig. 12, some simulations were made by varying the mass of the active mass damper to analyze the effect in the damping of the boat oscillations, where the load was a certain percentage of the boat weight.

As shown in Fig.12, the stabilization process behavior is heavily influenced by the stabilization mass of the system, yielding a quick stabilization. Such as discussed in Smith and Thomas III (1990), it was estimated that the load ratio for this kind of application should be around 1-2% and no more than 5% of the ship weigh.

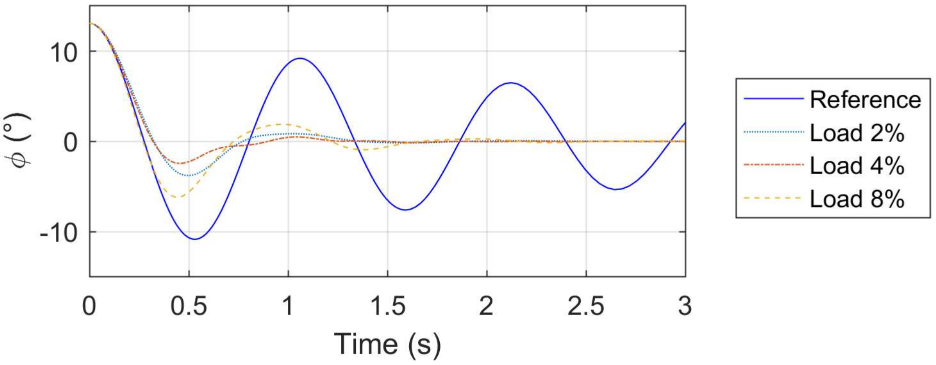

Figure 12. Mitigation system response

Almost 30 years later, it's observed that excessive increase of the stabilization mass do not provides an improvement on the stabilization process that worth's the decrement of the ship's payload capability and power efficiency.

A similar experiment was done utilizing the test bench, the results are available in Fig. 13. Having the system at rest, its disc was impulsed and then the response was observed and sent to the computer. In the following experiment there was 105.7g as the load for the mitigation system.

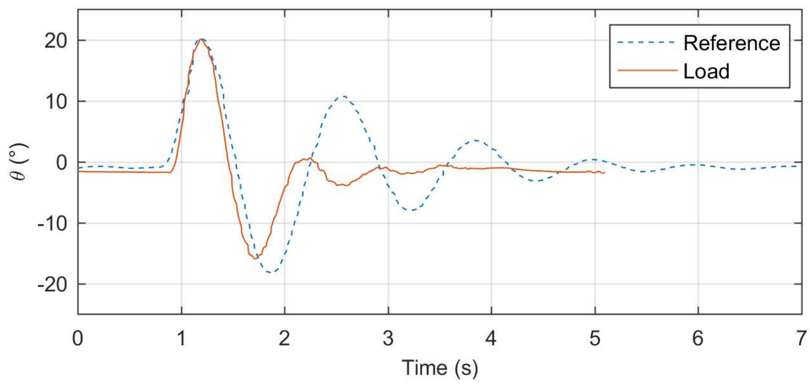

Figure 13. Experimental response

The execution of the experiment in the test bench presents some restrictions on its performance, for example, the systematic introduction of the perturbations and peculiarities of the mechanism itself are highlighted. Due to its construction, the cabling used for data transmission and the DC motor power supply adds a restorer moment that is not represented in mathematical models. For the same reason the initial slope is hardly null, observed in the initial part of Fig. 13. However, despite all constraints of the mechanical model, as can be seen in Fig. 13, its response still achieves a mitigation of the external perturbation with greater efficiency than the passive dynamics of the mechanism.

5. CONCLUSIONS AND FUTURE WORKS

Testing on the developed bench showed that the idea of a "dry" bench is very useful for experimental trials. Similarly to all experimental systems, there were implementation difficulties, need to deal with noise and communication difficulties between the sensors and the computer.

The influence of the actuation load ratio on the effectiveness of the stabilization system was verified. Thus, the system load definition procedure consists of the weighting of the safety factor and comfort provided by the mitigating system to the ship's cargo transportation, mobility and economic capacity. Theses results reaffirm that the proposed solution needs to be as effective as it is economically viable. Thus, considering the constraints inherent to the test bench and ignored nonlinearities in the model development, as shown by the experimental results, such characteristics did not prevent the demonstration of the desired oscillatory phenomena.

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