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The Investigations of Hydraulic Heave Compensation System

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Abstract. In the paper, a review of the current state of the knowledge in the field of heave compensation systems is presented. Three most important types of heave compensation systems such as: passive (PHC), active (AHC) and semi-active (hybrid) are described in more detail. Special attention is given to show the differences between presented in this paper heave compensation systems. In the paper also the developed by authors simulation model and test stand of hydraulic heave compensation solution is described. Its chosen simulation and experimental investigation results are presented.

Keywords: Heave compensation · Active heave · Passive heave Winch system · Heave system

1 Introduction

Different operations made on the ocean can be difficult due to the occurrence of waves. Very often the ocean is rough, which causes that the operations of the vessel on the sea are very difficult and even dangerous. The most important operation performed on such ships are: lifting, lowering and maintaining the load on a constant level i.e., at a constant height relative to the seabed. Just for these reasons heave compensation systems are developed and applied. In the last few decades such systems have been deeply investigated, developed and implemented on many ships. Heave compensation solutions can be divided into three categories: passive heave compensation (PHC), active heave compensation (AHC) and active-passive (semi-active) system. The last one combines advantages of both AHC and PHC systems. The target of heave compensation system application is to reduce the load fluctuations and to improve the safety [1].

First active heave compensation system was presented by Southerland in 1970 and described in paper [2]. In this paper Authors presented that the proposed AHC system can be successfully used in handling operation made on sea vessels. A few years later, i.e. in 1973, the prototype of the passive heave compensation system for offshore drilling is tested. The results of the tests of these and other similar solutions allowed

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AHC and PHC systems to become extensively accepted in offshore industry. In 2008 Do and Pan studied for the first time a nonlinear controller for the AHC system [3]. They used an electro-hydraulic system with a double acting cylinder. The use of the disturbance observer in the system made it possible to reach a success.

In active heave compensation system hydraulic and electric drives are most commonly used, but in passive compensators a pneumo-hydraulic drives are most commonly used. According to Nespoli et al. [4], hydraulic actuators (cylinders, motors) ensure the highest power to weight ratio of any other actuator. Moreover, the hydraulic drives applied in offshore application are well tested showing high reliability.

The paper is organized as follows. After the introduction, a review of the state of the art in the area of passive, active and hybrid system are given in Sect. 2, in which examples of these systems are described. Their advantages and disadvantages have been mentioned. In Sect. 3 the solution proposed by authors are presented. The experimental and simulation results are included. Conclusion and future work is presented in Sect. 4.

In this paper the Authors described built simulation model of the electrohydraulic servo drive. The Authors of this paper performed basic experimental and simulation investigations of an active heavy compensation system. Simulation model is built using Simulink SimHydraulics tool. The basic simulation research are shown for different parameters of the sine signal. Position of the hydraulic cylinder and hydraulic motor are measured.

2 Heave Compensation

2.1 Passive Heave Compensation

A passive heave compensation solution is used to reduce the influence of waves upon lifting and drilling operations (Fig. 1). In a simple solution a spring which utilizes spring isolation to reduce transmissibility is used. The more complicated system is based on a spring and a damper which were connected in parallel. These both elements are passive components. The PHC system is used to control of the cargo-carrying line. The passive compensator is mounted directly in this line. The PHC system does not consume any external power for their operation.

Nowadays, the passive systems consist of a hydraulic cylinder and gas accumulator, in which the gas is compressed by the hydraulic cylinder while piston rod extends, thereupon the pressure upon the piston grows. Example of the AHC system with accumulator is shown in Fig. 2. In the paper [5] the research results of passive heave compensation system are presented in detail. The stiffness and damping characteristics of the PHC system are measured and discussed. Passive heave compensation system with accumulators is also presented in paper [6].

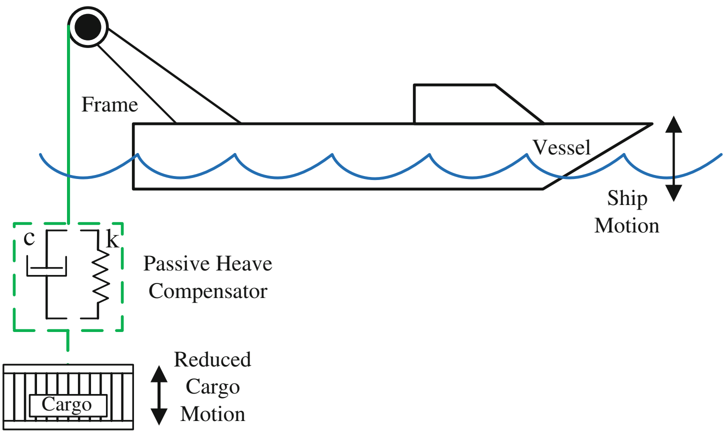

Fig. 1. Passive heave compensation example

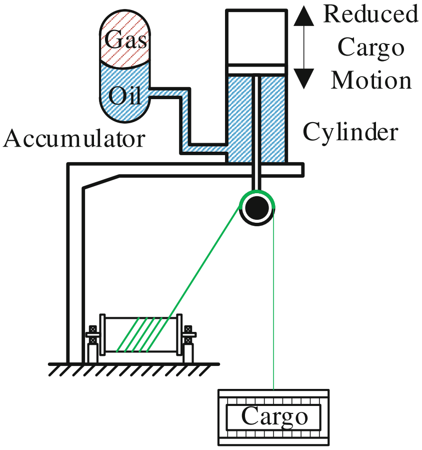

Fig. 2. Passive heave compensation system with accumulator

One of the passive compensation system disadvantage is the change of the efficiency for different vessel heave conditions. On the other hand it's not necessary to provide any energy to passive compensator, which is the most important advantage of PHC systems. Authors of the article [7] presented the modeling, simulation and optimal designing of an active heave compensation system for a draw-works on a hoisting rig.

A passive flying-sheave heave compensation system is presented in [8]. Their work is divided into three parts: design, tests and operational performance. In this paper a big passive cylinder was used to reduce a tether management system motion. A simple dynamic system model, which can predict the performance of the PHC is described. The model described only mechanical part of the passive compensator. A factory tests of the passive heave compensator on the ship are carried out. They allowed to create a hybrid active-over-passive heave system.

The simulation research on the PHC system is presented in paper [6]. In this case the PHC system with cylinders and accumulators is investigated. Authors of this paper built a simulation model of the system using MATLAB/Simulink environment. Parameters describing the model are calculated using the same software. Under the impact of random sea wave the compensation rate of the presented passive heave system was at 80%. The proposed PHC system worked much better for high sea wave frequency than for low wave frequencies.

2.2 Active Heave Compensation

In order to improve the compensation instead of passive system, active ones with closed-loop control can be used. The disadvantage of every AHC is the need to provide energy for supplying. In most cases the active heave systems are assembled on the board of the vessel. The scheme of the active heave compensation system is shown in Fig. 2. In this case the role of the compensator is made by an active winch, which a steel rope is wounded. The most important component of this system is the hydraulic motor that is responsible for rotating of the winch drum (Fig. 3).

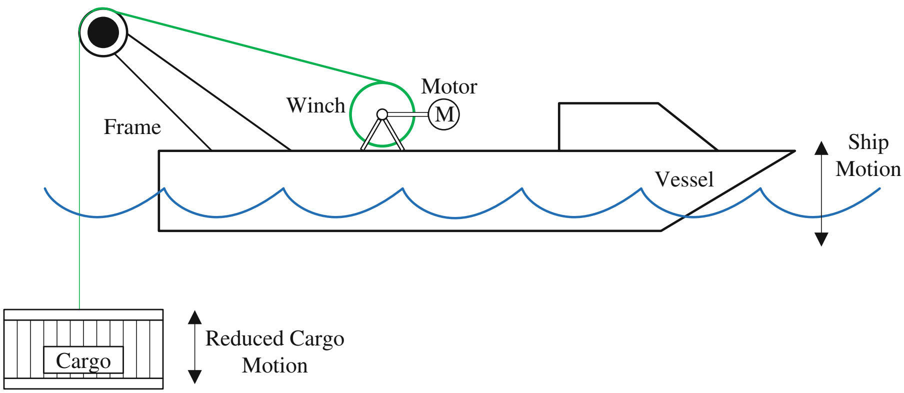

Fig. 3. Scheme of active heave compensation system

In the AHC system, the measuring of the vessel heave motion is needed. The ship current vertical position is measured and relayed to a main controller. The hydraulic motor is controlled by such a way, that the motor moves in the opposed direction of the heave motion of the ship. If a ship floats up, the controller sends signal to hydraulic actuator to move the cargo below. When the ship lowers the hydraulic actuator should move the cargo up. In this case the feedback signal is based on wave height, i.e. ship vertical position. As the feedback signal the force between the load and the cargo can be used. In this case the force on a rope should be measured.

In the paper [9], the prediction of the vessel's motion is used to activate control for offshore crane. A feed forward controller connected to the prediction system is used for compensation of the vertical-motion of the load (Fig. 4). In this article measurement and simulation results are included and compared.

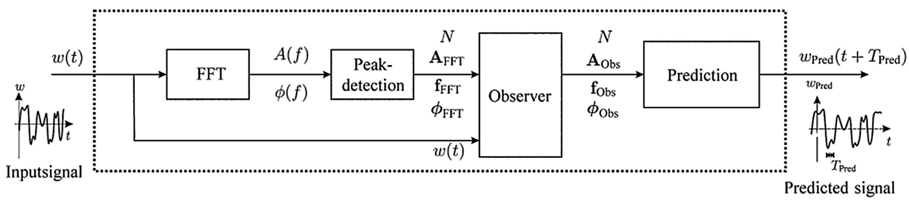

Fig. 4. Diagram of the prediction method for the vertical motion [9]

In the article [10] the design of a motion platform which was used to simulate the wave effects (Fig. 5) is presents. The task was to reproduce an operational scenario of controlling of the offshore cranes using a laboratory setup. A test stand, which consisted of an industrial robot, a joystick and a motion platform is built. The robot was controlled by the user via the joystick, while the motion platform simulated the sea waves. Accelerometer was mounted on the platform and used to measure its position. Information about platform acceleration is send to the main robot controller.

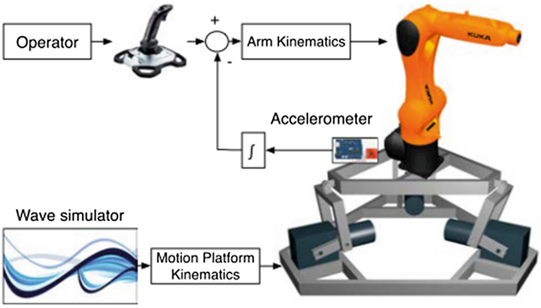

Fig. 5. The design of wave simulator [10]

In the paper [11] the experimental design, dedicated to analysis of the performance of the marine crane, is demonstrated. Proposed experimental test rig consisted of mechanical and driving system, control system and detecting system. The two types of control method implemented on DSP controller and in MATLAB/Simulink platform are tested. The experimental results confirmed the reliability and usability of the proposed experimental system.

In the article [12] modeling and simulation results of an active heave compensation system for draw-works is presented. Authors modeled the main components of the draw-works and hoisting rig. The simulation model included three main parts: mechanical, control and hydraulic. The model was tested for its dynamics in two cases. In the first case a vertical stabilization of the simulation model is tested and in the second case, a soft lowering of the load to the seabed is tested. In both cases, the sea wave effect has been reduced. In this paper, the simulation results showed that the proposed system parameters are in established limits.

2.3 Semi-active Heave Compensation

Semi-active heave compensation system is also often called a hybrid system. One of the options of the semi-active heave compensation is the system shown in Fig. 6. This system consists of one hydraulic cylinder which is a part of the active control loop, and two passive cylinders. The active cylinder is smaller than passive cylinders and therefore it can generate much smaller force than the passive cylinder. However the active cylinder is able to move with higher speed than the passive cylinders. So, the hybrid system is smaller in dimensions than strictly active heave system.

In the paper [13] a modeling and displacement controller design of a new type of semi-active heave compensation system is presented. The task was to verify a new type of a hybrid heave compensation system in a simulation and in experimental study. Authors used the accumulator and hydraulic motor as passive components. The active compensation motor is used, which compensated the cargo motion. The simulation and experimental research indicated that compensation rate of proposed system is above 90%.

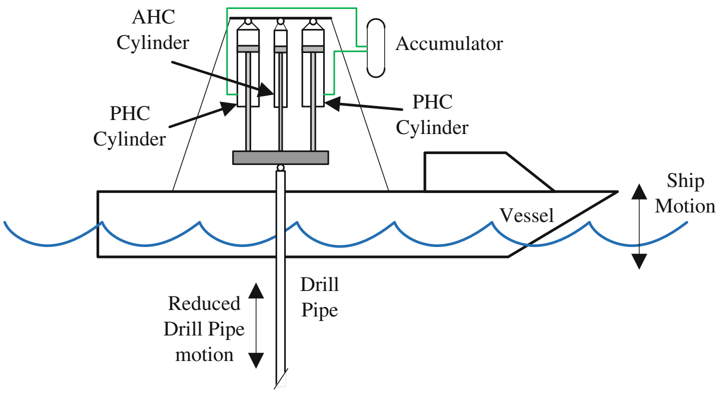

Fig. 6. One possible example of a semi-active heave compensation system

This results suggested that the new type of the semi-active heave system is very good. The passive heave compensator is used to hold the most of the load and the active system is used to assist in further load motion.

Hatleskog and Dunnigan in publication [14] proposed a hybrid heave compensator solution for a drill string. The task is to combine a PHC and AHC system. The passive heave compensator was used to hold the most of the load and an active system to assist in further load motion separate from ship heave. The Authors pointed out that ideal compensation is not possible because of sensor inaccuracies and limitations of the drive however it is possible to heave reduction from 90% to 95%. A similar system was patented by Robichaux and Hatleskog in 1993.

3 Experimental and Simulation Research

Authors of this paper performed also basic experimental and simulation investigations of an active heavy compensation system. At first a simulation model is built using Simulink SimHydraulics tool. Following equations describe the hydraulic motor dynamics [15, 16]:

$$\begin{aligned} \dot{\omega}_k = J_m^{-1}[q_m f(q)\left(p_i - p_j\right) - (M_l/u_{mech} \\ + b_\omega \omega_k + b_p|p_i - p_j| \cdot sign\omega_k + b \cdot sign\omega_k)] \end{aligned} \tag{1}$$

$$\dot{\varphi}_k = \omega_k \tag{2}$$

$$Q_{i,j} = q_m f(q)\omega_k \pm k_{lea} p_{i,j} \tag{3}$$

where: ω_k – angular speed of the hydraulic motor shaft, J_m – moment of inertia, q_m – maximal geometric volume, $f(q)$ – parameter of regulation: $-1 \le f(q) \le 1$, M_l – loading moment, b_ω, b_p, b – coefficients of the hydraulic motor hydro mechanical losses, u_{mech} – transfer number of the working mechanism gear, k_{lea} – coefficient of the hydraulic motor volumetric losses.

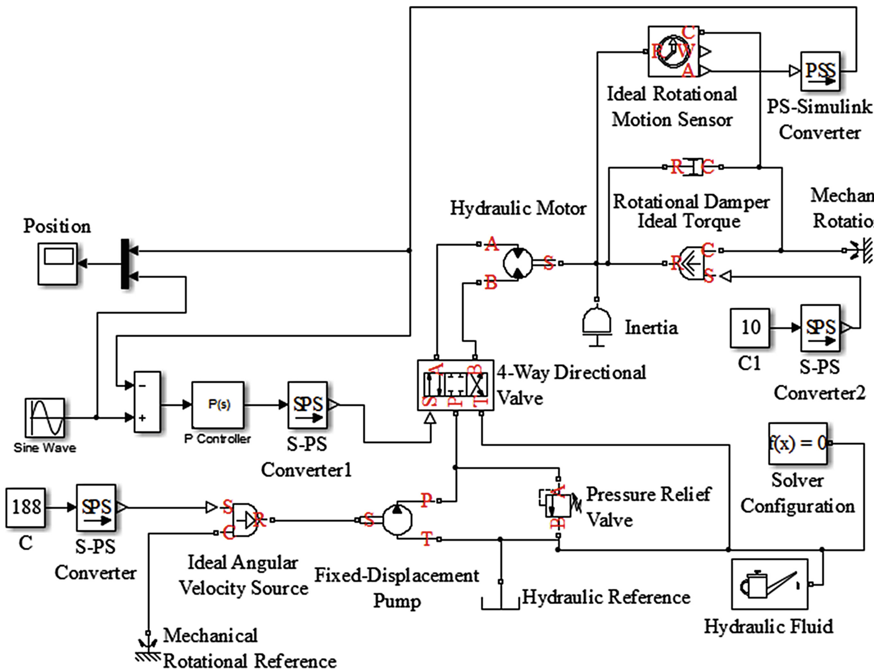

Fig. 7. Model of the electrohydraulic servo drive

The simulation model made in MATLAB/Simulink environment, is shown in Fig. 7. The test stand consist of: a hydraulic cylinder with a stroke range of 400 mm. At the end of the hydraulic cylinder piston rod, a hydraulic motor is fixed. The hydraulic motor max rotary speed is 810 rpm. Max output power of the hydraulic motor is equal to 16 kW. The hydraulic cylinder and hydraulic motor are controlled by two servo valves. The hydraulic cylinder is equipped with displacement sensor, which is used to measure the actual position of the hydraulic cylinder. Actual position of the hydraulic motor is measured by incremental encoder. The used power supply is characterized by the following parameters: maximum flow rate = 100 dm^3/min, maximum pressure p_0 = 40 MPa. The supply pressure is equal to 10 MPa. The hydraulic cylinder is used to generate the sinusoidal movement with different frequencies and amplitudes. The task is to measure the real object position (steel load) and to control of the hydraulic motor in such a way, that the steel load vertical position remains unchanged. As a result, the hydraulic motor followed the hydraulic cylinder. The double acting hydraulic actuator generated the sine signal with amplitude of 100 mm. The P-type controller is used to control of the motor. The controller gain value was kp = 80. In Fig. 8 the simulation results are presented. The simulation was performed for different desired signal. For the first simulation the signal time period was equal to T = 4 s and for the second simulation was equal to T = 2 s. In Fig. 9 the experimental test stand is presented.

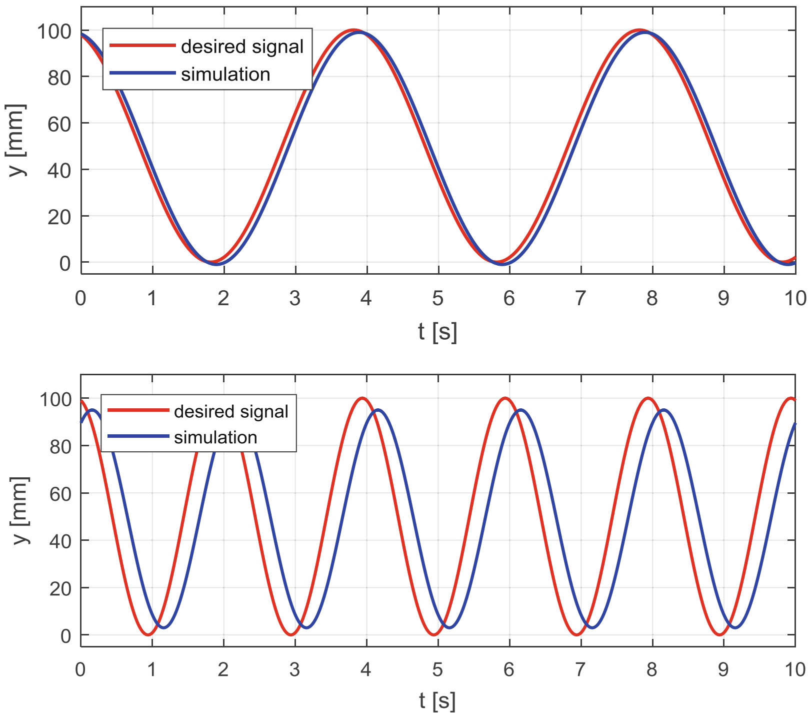

Fig. 8. Simulation results of the hydraulic motor model

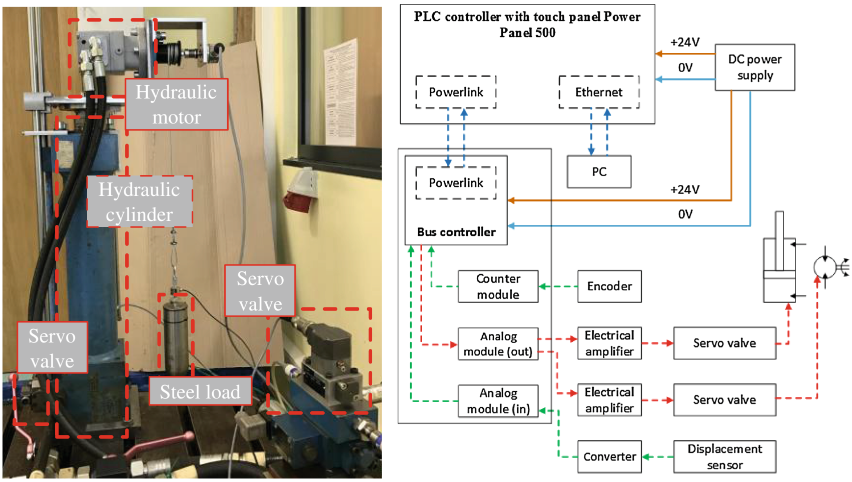

Fig. 9. The experimental test stand: (a) view, (b) scheme block

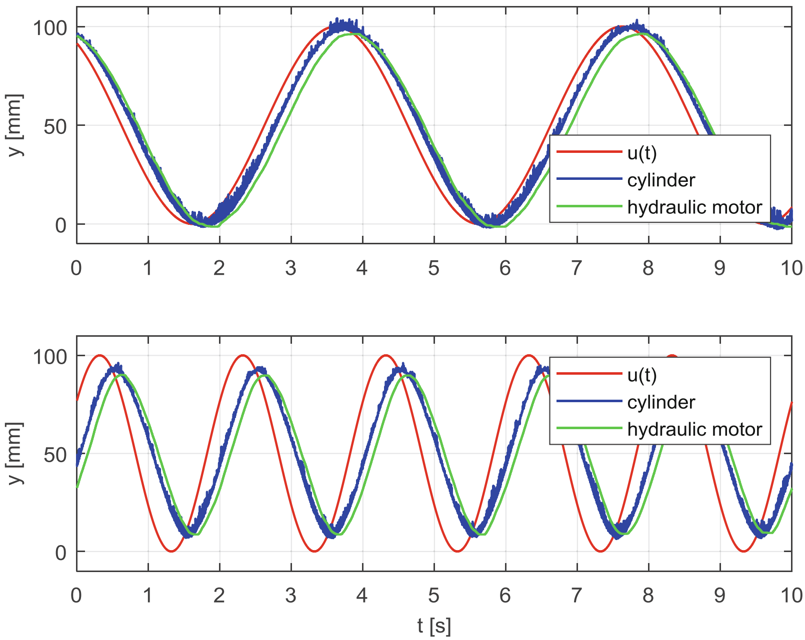

Fig. 10. Position of the hydraulic motor and hydraulic cylinder

In the next step of the presented here investigations the displacement of the hydraulic cylinder and hydraulic motor is measured. The desired signal was sine signal with amplitude of 100 mm. This signal time period was equal to $T = 4$ s and $T = 2$ s. The collected data are presented in Fig. 10.

In the last step of this paper a motion of the steel load was measured via AHRS sensor manufactured by AISENS company. The sensor was placed on the top of the steel load as in Fig. 11. Roll, Pitch and Yaw were measured during movement of the hydraulic cylinder. The sensor is put in dedicated housing made by AISENS company. The collected data are shown in Fig. 12. This sensor used acceleration, gyroscope and magnetic field to estimate a roll, pitch and yaw angle. A resolution of this sensor for roll and pitch angle is under 1 degree. In this research the most important data from sensor were for angle roll and pitch. Communication between sensor and PC is made by WI-FI. In the sensor is equipped in Wi-fi module. This module starts as AP (access point). PC is connected to the module with WI-FI name AISENS_07. The name of WI-FI is different for every module, but second module can work as Client. Data from module to computer are streamed by UDP protocol. All data are send with time domain. Application installed on computer is made in Unity 3D software for fast visualization data. This SDK include UDP protocol with reading data from sensor, saving files with data and allow to prepare own visualization. All data are streamed real time with frequency up to 200 Hz. Data from sensor are send after calibration correctness process. The output data from sensor are very stable.

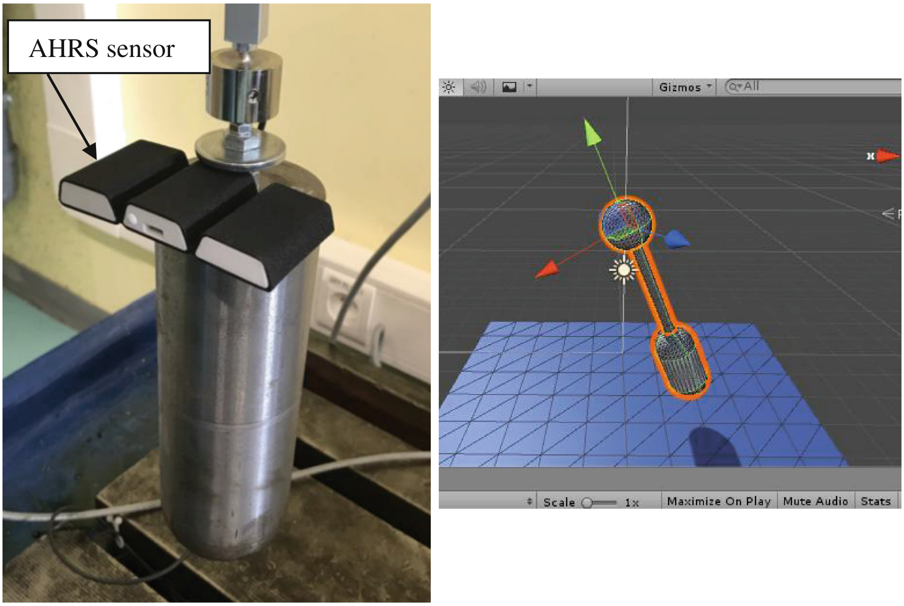

Fig. 11. View of AHRS sensor by AISENS company and visualization [17]

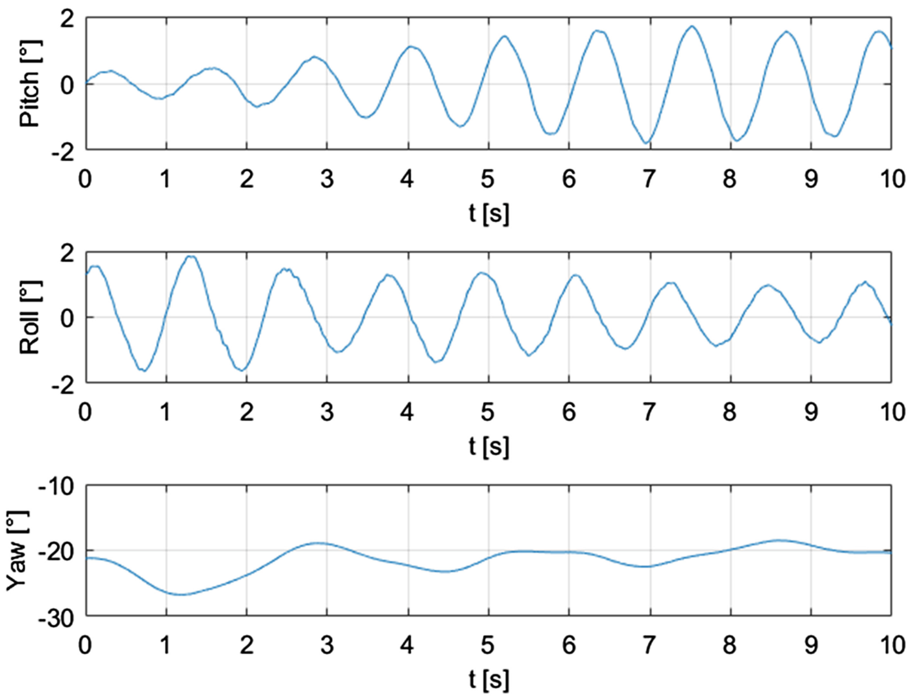

Fig. 12. Roll, pitch and yaw angle results

4 Conclusion

The article describes a chosen heave compensation systems. In this paper Authors described a three mine types of heave compensation systems such as: passive, semi-active and active systems. The basic equations which describe the active heave compensation system are proposed in Sect. 3. Simulation model is built in MATLAB/Simulink software. In this section the test stand built by the Authors is also described. The main components of the test stand are: hydraulic motor and hydraulic cylinder. Two electrohydraulic servo valves were used to control of the hydraulic motor and hydraulic cylinder. The control system was based on PLC type Power Panel 500. In the Fig. 8 and in Fig. 10, the simulation and experimental results are presented. These results confirmed that the proposed solution can successfully compensate floor fluctuations like sea waves.

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