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# Considerations regarding the laboratory testing of electro-hydraulic heave compensators

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Abstract. The unpredictable weather and the movements of the floating ships transform the marine environment into one of the most hostile working environments on the planet. Movements of lifting / lowering of vessels / platforms carrying cranes or drilling rigs, caused by the disturbing dynamics of the waves, varying in force, frequency, direction and amplitude, affect the precise positioning of the loads, lead to premature wear of the drilling holes, and endanger the integrity or even the life of the crew members. Due to the resonance in the cable of the load lifting system, caused by the movement of the vessel on a rough sea, the tension in the cable can increase more than 100 times. The safety and protection engineering solution is force compensation, that is the decoupling of the vertical movement of the ship from the one of the floating load. The authors present an experimental model, with the generator and the heave compensator at a small scale, which will simulate in the laboratory the operating modes of heave compensation systems very close to the real ones, existing on ships.

## 1. Introduction

It has been shown that for large suspended loads (weights) at the bottom of the ocean, the suspension cable could theoretically break despite being designed with a reasonable static safety coefficient. The reason is the resonance in the cable of the load system, caused by the vertical movement of the vessel produced by waves, which can increase the tension in the cable more than 100 times. Such a cable presents two potential dangers: *breaking*, causing loss or destruction of the load, and its falling close to the crew can cause serious bodily injury, or *retracting quickly*, with oscillations around the normal trajectory of retreat, causing material damage again and bodily harm.

The resonance in the cable could be avoided by two simple solutions: carrying out the lifting operations only when the sea is calm or the excessive over-sizing of the cable of the lifting system. Both solutions are prohibitively expensive, leading to: the extension of the lifting operations during more days or weeks during the rough sea seasons; increasing the weight of the lifting system and of the ship or platform on which it is mounted.

The engineering solution to avoid the resonance in the cable is the decoupling of the vertical movement of the ship which is floating from the vertical movement of the suspended load. This decoupling of the load from the movement of the ship is commonly known as force compensation.

For the safety of the human crew, working on rough seas and oceans, for increasing the productivity of the floating lifting / drilling installations, respectively the progressive reduction of the dependence of their operation on the agitation state of the waves, compensation systems have been implemented in the structure of the machines in question, which have increasingly evolved: passive compensation systems, active compensation systems and hybrid compensation systems (active-passive or semi-active). All these types of compensators decouple the loading / unloading movement of the load from the lifting / lowering movement of the ship. The control models for the heave compensation systems have evolved into today's predictive semi-active models, with the prediction of the intensity of the wave agitation.

There are four stages of development of the heave compensator [1]:

**Stage I** - passive heave compensators (PHCs), which are vibration isolation systems consisting of a shock absorber and a compression spring, mounted in parallel, or from a hydro-pneumatic accumulator. They function as open loop systems, in which the input is represented by the movement of the ship and the output is the reduced amplitude of the load movement attached to the floating crane hook. Such compensators do not require outside energy for operation and can have an average efficiency of up to 10-35%. Passive compensators are ineffective in applications such as transferring a payload from one ship to another, or in compensating heaves when passing a load from air to water. In these cases, the PHCs are not able to compensate for the relative movement between two vessels with independent movement reference points. For these applications, an active heave compensator must be used.

**Stage II** - active heave compensators (AHCs), which are automated systems that involve closed loop control and require outside energy for operation. If the ship is lifted by the heave, then an active compensating system is controlled by a controller, which acts in the opposite direction to lower the load, with the same displacement. These systems can have an efficiency of at least 80%.

**Stage III** - hybrid or semi-active compensators. These arose because of the high costs of the active compensators, which were abandoned although they had a high efficiency. Hybrid compensation systems have two components: a passive one and an active one. The passive component contains two large pneumatic cylinders, which load at a pressure corresponding to keeping the weight in balance, mid-stroke. Active component, much cheaper than in the case of fully active compensators, contains a small hydraulic servo cylinder, which applies the adjusting forces to the load based on an active control strategy.

**Stage IV** - the most modern control models for hybrid heave compensation systems are predictive semi-active models, with forecasting the intensity of the wave agitation.

Figure 1 shows an example of a small ship, which moves a load vertically and uses a passive heave compensator to reduce the movement of the load, as a result of the vertical movement of the ship, generated by the waves. The compensator isolates the ship from the load, as it is placed between them.

Figure 2 shows a ship which rises vertically on an ocean wave. The suspended load under water follows the movement of the vessel, indicating the deactivated state of the active hydraulic compensation system (AHC). With AHC enabled, the load (marked in gray) is kept at a constant depth.

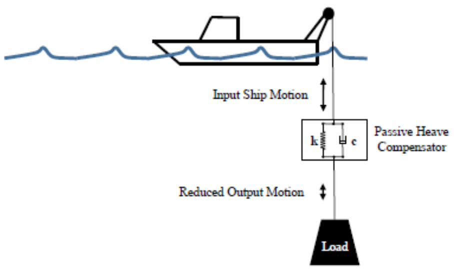


**Figure 1.** Passive heave compensator.

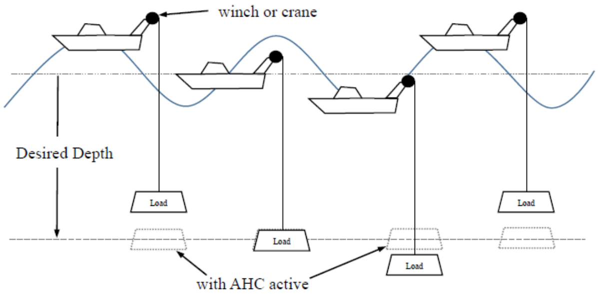


**Figure 2.** Active heave compensator.

Figure 3 shows a hydraulic system of active compensation, which operates in the case of transferring a load from one ship to another, under high turbulent conditions. The ship receiving the load is equipped with a heave amplitude sensor (it measures the tension in the cable and its direction of movement). The cargo transfer vessel is equipped with a winch crane, which contains the active heave compensator. It consists of a double-acting differential hydraulic cylinder, a hydraulic positioner, a spring attachment, attached to the crane, which tensions the transducer cable fixed by the first vessel deck, a proportional hydraulic valve, which regulates the pressure in the hydraulic cylinder to maintain a constant height of the load, suspended by the crane cable, from the deck of the vessel taking over the load.

Figure 4 shows a possible hybrid compensation system. The system contains two passive hydraulic cylinders, each supporting half of the total weight of the $F_L$ load and a third smaller hydraulic cylinder, which is part of an active control loop, which can generate an additional tuning force, called $F_A$. The active cylinder must be capable of moving at maximum load speed, in any situation. Since much lower forces than those loaded on the passive cylinders will generally apply to the active cylinder, it may be physically smaller, requiring a lower feed rate, lower pressure and therefore less hydraulic power compared to the hydraulic cylinder of a strictly active compensation system.

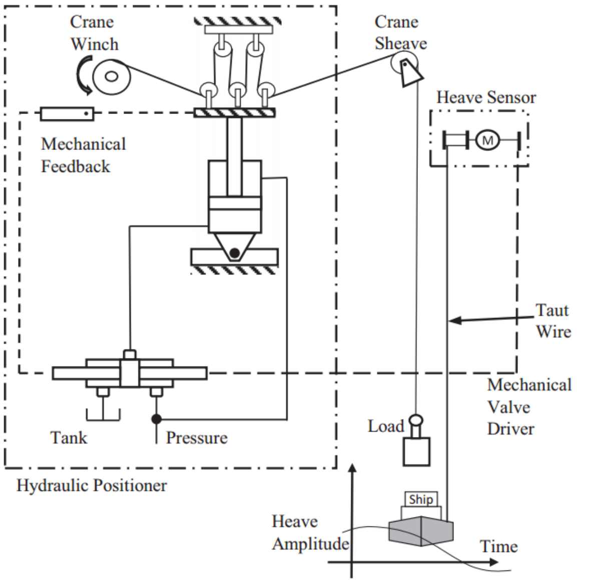


**Figure 3.** Active heave compensator; load transfer from one ship to another (Southerland -1970).

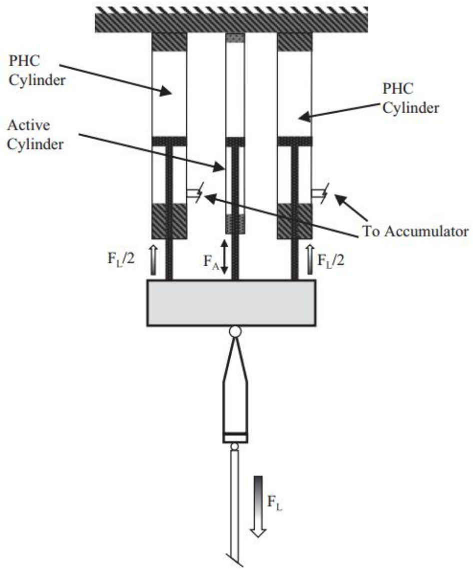


**Figure 4.** Hybrid heave (semi-active) compensator.

## 2. Experimental tests for an active heave compensator

### *2.1 Technological facilities*

The experimental stand in figure 5 [2] is intended for the evaluation of the functional performances, in the laboratory, of the active heave compensators, real but reduced in scale, intended for floating cranes and marine drilling rigs.

The experimental stand contains a hydraulic servomechanism with an external loop and two internal position adjustment loops. The first internal position adjustment loop is manifested at the level of a Parker hydraulic servo-cylinder, consisting of cylinder + valve + stroke transducer, figure 6, which simulates the wave agitation and is located at the bottom of an assembly of two identical hydraulic cylinders, coupled together. The second internal position adjustment loop manifests at the level of a Moog hydraulic servo-cylinder, consisting of cylinder + valve + stroke transducer, figure 7, located at the top of the same assembly, which simulates the dynamic behavior of the active component of a hybrid heave compensation system. The external position adjustment loop causes that, irrespective of the excitation signal applied to the first servo cylinder, the second servo cylinder follows the movement of the first, in the opposite direction and at the same speed, so that the end of its rod remains permanently positioned at the same elevation as a fixed reference plan (the floor of the

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laboratory, for example). The two servo-cylinders were connected to a pumping group existing in the INOE 2000-IHP Servo Technique laboratory, with the following characteristics: flow = adjustable, 0 ... 120 l/min; pressure = adjustable, 0 ... 310 bar; oil tank volume = 400l; electric motor power = 55kW; engine speed el. = 1500 rev / min. Also used for the tests: a programmable logic controller from Schneider Electric, code TM221CE16U, which generates current controls for valves and takes information from the transducers; a signal generator; a PC; a control software application dedicated to the test and data acquisition stand, compatible with the programmable logic controller used.

**Figure 5.** Active heave compensator. The experimental stand.

**Figure 6.** Perturbation actuator (heave sim.).

**Figure 7.**Tracking actuator (active heave comp.).

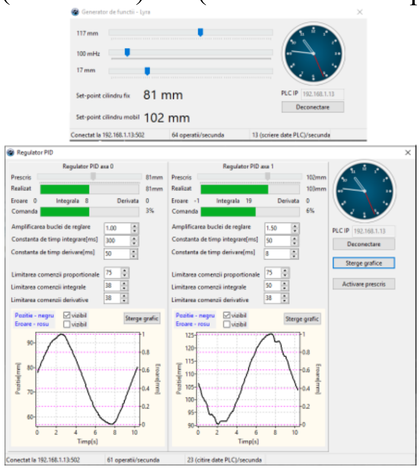



**Figure 8.** The response of the tracking servo-cylinder to the excitation of the disruptive servo-cylinder with a 0.1Hz/17 mm sinusoidal signal.

*2.2 Experimental tests*

In the control window of the software application installed on the PC managing the samples and acquisition of experimental results, figure 8, different sinusoidal excitation signals were prescribed for the disruptive servo-cylinder (PID controller axis 0) and the dynamics of the tracking servo-cylinder was followed (PID controller axis 1). They had the frequency of 0.1 Hz and 0.2 Hz, and the amplitude of 17 mm, 31 mm, 34 mm. Six types of sinusoidal signals were used: 0.1 Hz / 17 mm; 0.2 Hz / 17 mm; 0.1 Hz / 31 mm; 0.2 Hz / 31 mm; 0.1 Hz / 34 mm; 0.2 Hz / 34 mm. The smallest tracking error (1%) was recorded for the combination of 0.1 Hz / 17 mm. In figure 8 one can notice that the tracking servo cylinder has reproduced sufficiently exactly the excitation signal of the disruptive servo-cylinder, for the mentioned combination.

The prescribed values for the displacements of the two servo-cylinders are given by the relations (1) and (2), namely:

$$Sp_0 = 75 + \mathrm{a} \cdot \sin\left(\frac{2\pi \cdot f}{1000} \cdot (t - t_0)\right) \tag{1}$$

$$Sp_1 = 300 - \mathrm{h} - Sp_0 \tag{2}$$

Where,

$Sp_0$ = Set point for heave simulator [mm]; *75*= 50% of the maximum stroke of the servo-cylinder [mm]; a = the amplitude of the sinusoidal excitation signal [mm]; *f* = the frequency of the sinusoidal excitation signal [Hz]; $\pi$ = 3.14 [-]; *1000* = scale factor [-]; $t\text{-}t_0$ = variable time [s].
$Sp_1$ = Set point for active heave compensator [mm]; *300* = the maximum stroke of the servo-cylinder 0 + the maximum stroke of the servo-cylinder 1 [mm]; *h*= const. = positioning of the vertical load [mm].

For the sinusoidal excitation signal, with the frequency of 0.1 Hz and amplitude of 65 mm, applied to the wave generator, we followed the dynamics of the active heave compensator and the positioning of the vertical load at a constant level relative to the floor of the laboratory. For the two adjustment loops the following were prescribed: Amplification 1.5, Limit 100; Integration time constant 500 mms, Limit 100 mms; Derivative time constant 50 mms, Limit 50 mms.

The acquired experimental data were exported to an Excel file and processed graphically, figure 9.

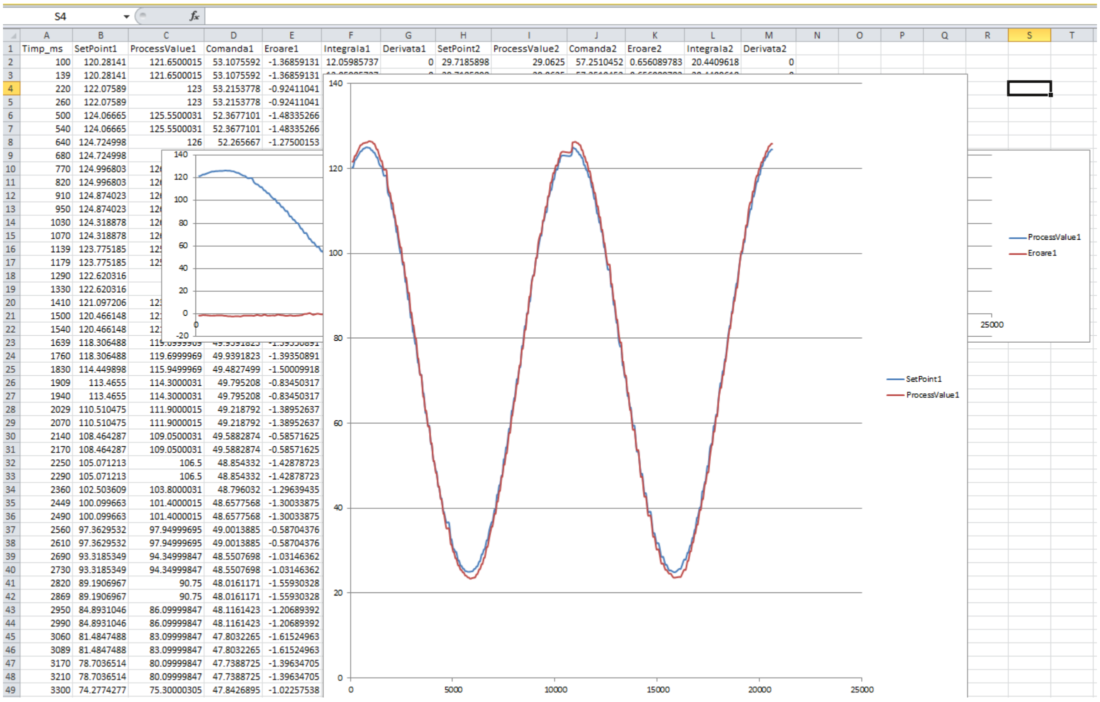
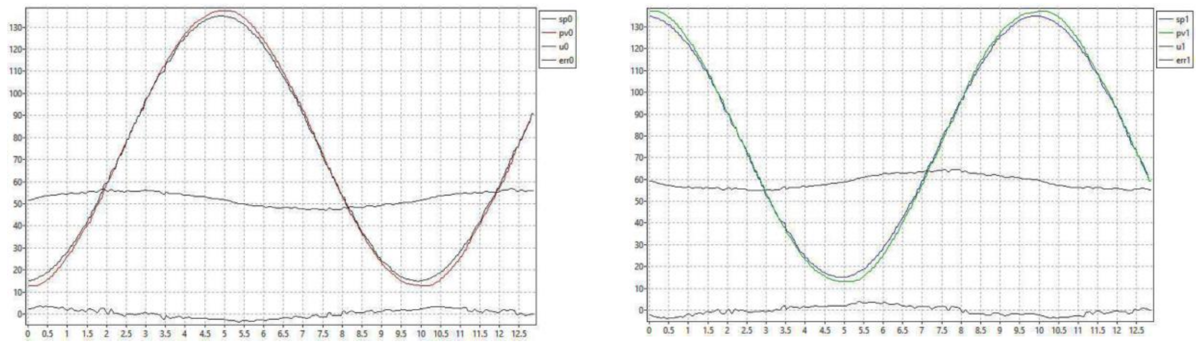



**Figure 9.** Data export to Excel file; graphic processing.

Following the processing of the acquired data over an interval of 12.5 sec., the graphs from figure 10 and figure 11 were obtained.




(a) (b)

**Figure 10. (a) Heave simulator**; **(b) Active heave compensator**; Set point values, sp = f(t); Process values, pv = f(t); Command, u = -50…+50 mA; Error, err = pv - sp; **x: Time [s]**; **y: Stroke [mm]**.

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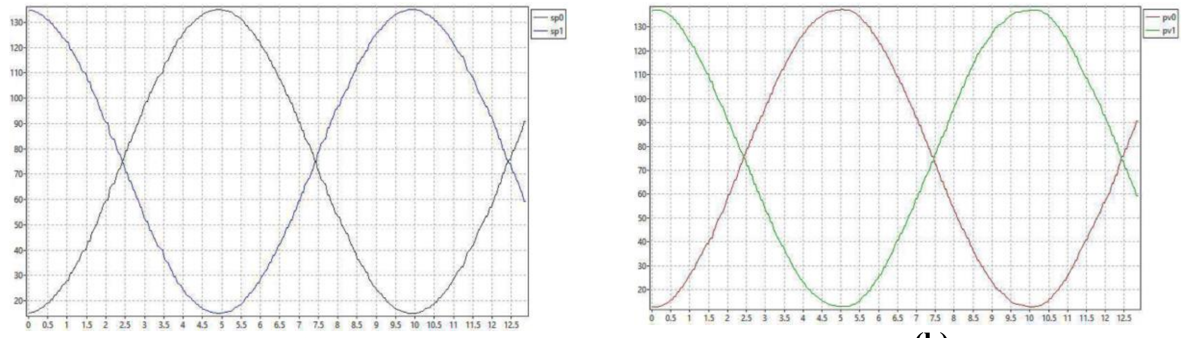
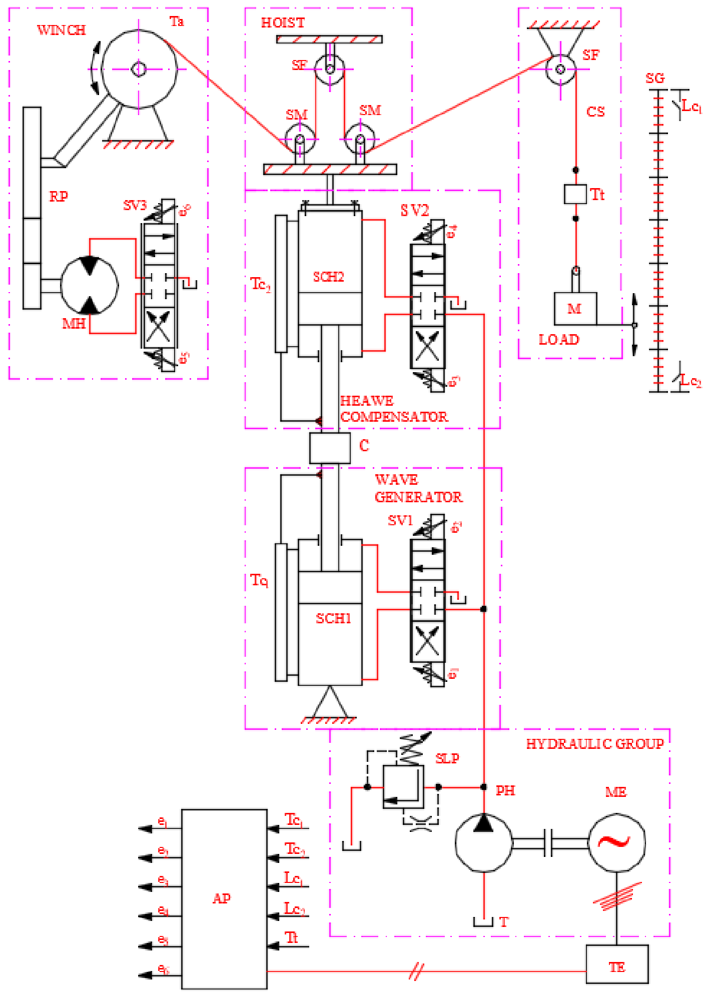


(a) (b)

**Figure 11. (a) Set point values; (b) Process values;** The overlap of the set point values and the process values of the displacements of the two servo-cylinders; **x: Time [s]; y: Stroke [mm]**.

## 2. Considerations regarding the experimental testing of hybrid heave compensators

### *3.1. Technological facilities*




**Scheme composition:**

**a) Hydraulic group:** ME = electric motor of drive pump; PH = fixed volume hydraulic pump; T = hydraulic oil tank; SLP = pressure limiting valve.

**b) Heave generator:** SCH1 = hydraulic servo-cylinder; SV1 = electro-hydraulic servo valve; e1, e2 = prop. electromagnets SV1; Tc1 = stroke translator SCH1; C = actuator rod coupling.

**c) Heave compensator:** SCH2 = hydraulic servo-cylinder; SV2 = electro-hydraulic servo valve; e3, e4 = prop. electromagnets SV2;
Tc2 = stroke transducer SCH2.

**d) Winch:** SV3 = electro-hydraulic valve; e5, e6 = prop. electromagnets SV3; MH = fixed volume hydraulic motor; RP = planetary reducer; Ta = trolley drum.

**e) Hoist:** SF, SM = fixed / mobile pulley.

**f) Load:** CS = load cable; Tt = cable tension transducer; M = mass of load; SG = graded scale;
Lc1, Lc2 = stroke limiters.

**g) TE:** Electrical and control panel;

**h) AP:** automatically programmable (servo-controller) with signals from transducers, on inputs, and controls to electromagnets, on outputs.

**Figure 12.** The experimental stand for testing hybrid heave compensators.

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The experimental stand, figure 12, is characterized by: modification of the structure of the mobile assembly and structure of the fixed assembly, so that they allow also the mounting of the passive component of the compensator (variant of the solution with special hydraulic cylinder with pneumatic accumulator, according to the patent [3], or variant with two pneumatic cylinders, or variant with a hydraulic damper and a spring, connected in parallel); the introduction on the experimental stand of the system for lifting-lowering with cable of a 20 kg -load, consisting of a hydraulic motor, controlled by a hydraulic directional valve with electric control, a planetary gearbox, a hoist with two fixed pulleys and a mobile one, another fixed pulley, a cable winding drum, fitted with bearings; introducing on the experimental stand a system for constant maintenance of the tension in the cable, when raising and lowering the load, regardless of the disturbances introduced by the heave simulator; replacement of the electrical and control cabinet; replacement of the control software application (for two servo valves and a hydraulic directional valve) with electric control and data acquisition (from the two stroke transducers, from the two load-limiting cable displacement limiters and from the cable tension transducer); hydraulic connection of the experimental stand to its own mobile pumping group; transforming the product into a mobile assembly, which can be carried to fairs, exhibitions and workshops for the following demonstrations: vertical, controlled movement, of a mass suspended by a cable, precise positioning of the mass, keeping the cable tension relatively constant (tolerance ± 2%), under conditions of simulation of wave vertical movements.

*3.2. Numerical simulations*

Numerical simulations have been carried out in Amesim Simcenter, [4], and they show that the efficiency of an active or hybrid compensation system can be identified by means of a test stand, composed of two mechanically connected systems: a heave generator and the compensation system (figure 10).

The heave generator is an electro-hydraulic servo system, composed mainly of a hydraulic cylinder, an electro-hydraulic servo valve, a linear displacement transducer, a PID controller and a constant pressure supply system. It can generate different types of "heaves", which are applied to a heave compensator.

The compensation system includes a hydraulic motor, controlled by an electro-hydraulic servo valve, a planetary gearbox, a pulley that supports the load through a cable and an incremental encoder, which measures the rotation angle of the pulley. Another dedicated controller generates the correction input. Figure 13 shows the simulation network of Simcenter Amesim, which includes all the components and super-components used to generate the "heave" and compensate for the cable movement. The efficiency of this type of compensation system depends on the load mass and the amplitude of the heave. For the nominal mass of the load (250 kg), the amplitude of the movement of the load is reduced by 92%.

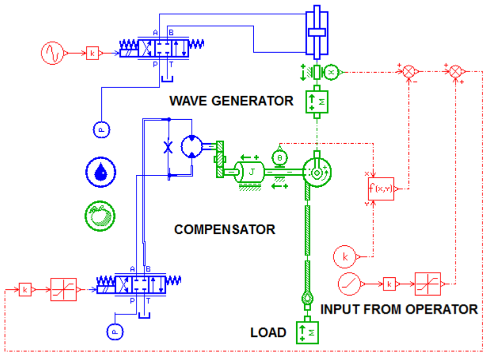


**Figure 13.** Simulation network of the heave generator and the compensation system.

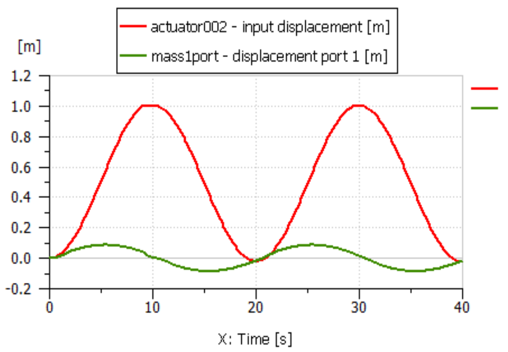


**Figure 14.** Compensation efficiency for a typical sine heave (red curve).

The easiest way to avoid control stability problems is to use a sharp calibration hole (0.7 mm), placed between the hydraulic motor ports, which consumes a large part of the valve supply flow. The load (load weight) has the most important influence on the compensation error. For a mass of 250 kg, the maximum error is approximately 85 mm (figure 14). For a double load, the same error reaches 190 mm (figure 15). In a period of sinusoidal heave, produced by the heave generator, the range of variation of the piston speed of the cylinder is three times greater than the range of variation of the speed of the load (figure16).

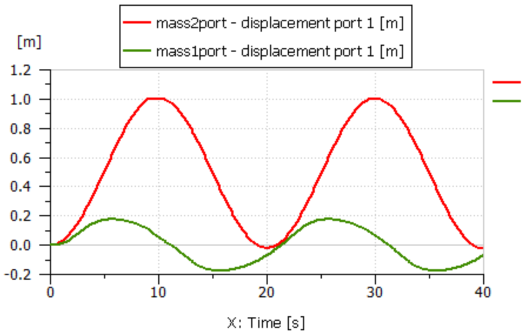


**Figure 15.** Compensation efficiency for a load of 500 kg.

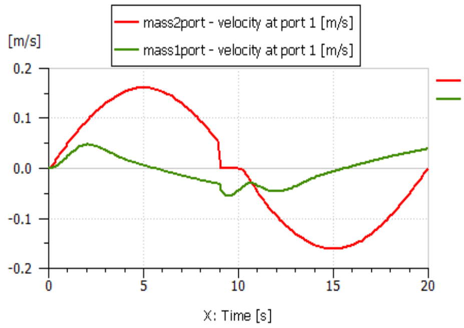


**Figure 16.** Speed and load variation of the piston in a sinusoidal heave period.

## 4. Conclusions

The degree of novelty and relevance of these preliminary results, relative to the national and international state-of-the-art, is given by solving the following problems of the field:

- Testing under laboratory conditions the parts of the active component within the hybrid heave compensation systems intended for floating cranes and drilling installations (active hydraulic servo cylinder, proportional hydraulic directional valve or servo valve, position transducer, controller);
- Checking under laboratory conditions the methods of adjustment and control dedicated to the active component of the hybrid systems of heave compensation;
- Simulating in the laboratory of heave lifting / lowering motion, with variable amplitudes and frequencies, that is of the actual operating conditions of the heave compensators;
- Simulating in the laboratory a forecasting program for the wave agitation state, in relation to which it is possible to assess the dynamic performances of the active component of the hybrid heave compensation systems (response time, stability, positioning error).

A future challenge for authors is the experimental validation of a hybrid heave compensator, on a small-scale model of demonstrator type, on which an interested witnessing public can watch the dynamic performances of the system live.

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