A Heave Compensation System Prototype for Geotechnical Drilling Vessels*

Aldo G. Arriaga¹, Marcos Arroyo², Ignacio Álvarez³ and Norma Pérez⁴

Abstract

The present work describes the design and initial testing of a scaled physical prototype of the heave compensation systems (HCS) that are employed in offshore geotechnical drilling ships. HCS compensate the vertical movements due to sea waves in vessels (heave), with the goal of maintaining drill bit thrust and depth within the strictly specified limits that are prescribed for geotechnical sampling and testing. Both passive and active HCS can be represented on the prototype.

There is no known precedent of prototyping HCS before their full-scale implementation in geotechnical vessels: this makes the evolution of such systems costly and slow and might explain, for instance, the slow take-up of more advanced active-control designs. The project aims to be a middle step between the design and implementation phases of control schemes for heave compensation and will make possible to try several options before having the actual ship, decreasing the start-up time and becoming another link in the concurrent engineering chain.

The prototype makes possible to modify different operation and environmental parameters, in order to test and simulate diverse working conditions and disturbances. This will allow both to evaluate current control techniques and to develop new, optimized, control strategies.

The prototype conception started from a dynamical model of the system which describes the behavior of the drilling boat, strings and pipes submitted to the forces of sea waves. This model allowed us to understand the different parts the overall system, aiming to emulate their behavior on a smaller scale, in a lab environment. With the goal of emulating the ship’s behavior as well as possible, the prototype has been provided with:

- Electric actuators that simulate sea motion and the behavior of hydraulic elements in active compensation
- Pneumatic springs and dampers to simulate passive compensation, the elasticity of the string and drag/drilling forces.
- Sensors that record the relative displacement of the boat (accelerometers), the compensation crown position (wire potentiometers) and the drag/drilling forces (load cell).

The main design paradigm has been the adjustable parameters. This means that it is possible to test different environmental conditions and different kinds of disturbances, and several control laws. To this purpose, components have been selected and designed in order make possible to modify the model elements’ values from a computer and record the results in real time. Non-linear elements can eventually be added to the system to test the response in difficult conditions.

I. INTRODUCTION

A motion compensator is a device that reduces the undesirable relative motion between two linked objects, in order to counteract its often noxious effects e.g. hysteresis, jerk, sudden changes in link forces and stress, etc. In a sea scenario such artifacts are usually placed between a floating object (a vessel) and another one that is to remain stationary (on the seafloor).

Six parameters describe the position (surge, sway and heave) and orientation (roll, pitch and yaw) of a vessel, each one of which is subject to motion disturbances that must be compensated. This paper focuses on heave compensators since good control of vertical displacement and position is a particular concern for sampling in geotechnical surveys, especially in soft soils.

A. Passive Heave Compensation PHC

The basic principle of PHC consists in storing the energy of the sea waves when displacing the vessel and releasing it afterward. This is achieved through the implementation of oleo-pneumatic springs which effectively reduce the mechanical transmissibility to the drilling mechanisms. A typical example of these mechanisms is depicted in Figure 1. When the piston rod is contracted, the gas is compressed and the pressure increases; when stretched, the opposite occurs.

There are some other techniques based on the principle of vibration absorption in unbalanced mechanisms. These techniques incorporate a mass and a spring in order to modify the system dynamics (Korde, 1998; Liujun, 2009). While relatively less cumbersome they have the drawback of making more difficult the later addition of active compensation elements.

B. Active Heave Compensation AHC

AHC complements the passive solutions (Figure 1). While PHC can deal with large amplitude movements, AHC is used to improve precision for small amplitude movements, hard to compensate with PHC because of the friction effects of the pneumatic pistons. These disturbances, among others in the system, are to be taken into account into a closed-loop system that constitutes the core of the AHCs.

*Research Supported by the Autonomous Government of Catalonia (Generalitat) through its Industrial Doctorate Programme

¹A. Arriaga is with Igeotest, Borrassà s/n 17600 Figueres, Girona, Spain (phone: (34) 972 513 466; fax: (34) 972 513 473; e-mail: aldo@igeotest.com).
²M. Arroyo is with the Department of Geotechnical Engineering and Geosciences, UPC, Barcelona, Spain (e-mail: marcos.arroyo@upc.edu).
³I. Álvarez is with the Systems and Automation Engineering Department, University of Oviedo, Spain (e-mail: ialvarez@hecate.edv.uniovi.es).
The dynamic target for an AHC should be based on precise relative motion measurement, for this purpose inertial measurement units (IMU) are incorporated into the system described in this paper. The difference between the distance measured by the IMU and the cylinder rod position (measured with wire potentiometers or encoders) gives an error signal used to determine actuator incremental displacement.

The displacement estimation with the IMU is obtained by double integration of acceleration signals. However, such signals are noisy and this leads to an error that increases quadratically. This problem has been addressed thoroughly in recent years and some good results have been achieved concerning periodic or quasi-periodic motions (Tun Latt, 2011). A solution based on (Madgwick, 2013) is proposed as an inexpensive inertial displacement measurement alternative for both the actual vessels and the simulator.

The actuation system in AHC for drilling usually consists of oleo-hydraulic cylinders controlled with proportional valves for raising and lowering the drilling mechanisms. There are other applications (e.g. materials loading and unloading) which utilize servomotors and winches with pulleys arrangements in order to roll/unroll cable instead of pushing/pulling. In both cases the end result is a smooth predefined movement (or remaining standstill) with the sea-wave originated disturbance being canceled.

C. The need for a physical simulator

The main motivation behind this project is to create a bridge between the conception of AHC systems and their physical implementations. Simplified dynamic models are often used when designing control schemes and performing numerical simulations. However, imperfections and non-linearities are often left out of simulations in order to render the model manageable and the design possible. Posterior experimentation would allow to verify the validity of such simplifications, but in a real-life implementation (i.e. aboard a drilling ship) scenario performing experiments is not only expensive but dangerous. Also, several operation modes might be envisaged (for instance piston sampling vs drilling in coring mode) that would require operator interaction with the system. Training is essential, but also difficult and costly if done aboard the vessel.

Having a physical simulator that allows to perform such experiments before testing them in actual vessels will decrease the risks inherent to the experimentation. The simulator must depict the behavior of the ship and the compensation devices well enough in order to be useful as a validation tool previous to the implementation in the ship. In a more advanced stage of the project a common computational platform and graphical user interface for both the simulator and the controller on board could be developed, reducing even further the start-up time.

Equipped with sensors and actuators similar to those on board of real vessels, the simulator could be used also as a training tool for new personnel. This will allow to reduce the training period and the overall cost of it by reducing as well “unproductive” time offshore.

II. DESIGN OF THE SIMULATOR

A. Dynamic Model and Scaling Down

A simplified dynamic model has been formulated inspired by the models presented by (Albers, 2010) and (Azpiazu, 1983) and (Hatleskog, 2007). Some dynamical models (Hatleskog, 2007) seek to take into account the harmonic effects due to the drill assembly going from the seafloor to the drilling rig. However, the drill assembly can be also modeled as a single elastic
spring with adjustable stiffness (Azpiazu, 1983), this approach simplifies greatly the model for a physical implementation; and appears versatile enough to meet the experiments requirements. Coulomb’s friction effects in cylinders and linear-rail mechanisms are also taken into account; as shown later those effects will prove important for small amplitude movements.

Equation 1 represents an idealized description of the heave movement (Albers, 2010). More realistic descriptions of sea heave motion (Sverdrup-Thygeson, 2007) can be easily applied in the physical simulator. Nevertheless, for initial testing, scaling and design this representation suffices.

\[ X_1 = \frac{H}{2} \sin(\omega t) \]  

(1)

The vertical movement of the compensation block (crown), can be expressed in terms of \( X_1 \) according to the free body diagram in Figure 2:

\[ m X_2 = F_A + F_0 K_1 + K_1 (X_1 - X_2) + C_{v_1} (\dot{X}_1 - \dot{X}_2) + C_{c_1} \cdot \text{sgn}(X_1 - X_2) - K_2 (X_2 - X_3) - mg \]  

(2)

\[ M X_3 = K_2 (X_2 - X_3) - C_{v_2} \dot{X}_3 - \beta Mg \]  

(3)

Where:

\( X_1 \) : Vessel vertical displacement.
\( X_2 \) : Compensation crown’s displacement.
\( X_3 \) : Drill-bit displacement.
\( F_A \) : AHC actuator force.
\( F_0 K_1 \) : Pneumatic spring preload.
\( K_1 \) : Pneumatic spring stiffness coefficient (linear approximation).
\( C_{c_1} \) : Coulomb’s friction coefficient.
\( C_{v_1} \) : Viscous friction in pneumatic spring.
\( K_2 \) : Combined spring coefficient of drilling pipes and cables in drilling tower.
\( m \) : Combined mass of the drilling mechanism and compensation cylinders.
\( M \) : Combined mass of drill bit and pipes.
\( C_{v_2} \) : Viscous friction coefficient representing the drilling force (soil resistance) and sea drag force.
\( mg \) : Weight of \( m \).
\( \beta \) : Buoyancy factor.

The preload in the pneumatic spring of the compensator is set to cancel the weights \( mg \) and \( \beta Mg \) so that the cylinders remain at mid-travel when the system is at rest. This preload is usually adjusted manually by the drill operator and often the effective stiffness coefficient has an offset from its ideal value, making the spring more or less stiff that it should be. This is a source of problems when drilling soft soils since the viscous coefficient is not as high as in harder soils, making easier for the drill bit to move and affecting the soil sample.

Using typical parameter values for a drilling vessel (Halkeskog, 2007) numerical simulations under several conditions were carried out. These data allowed to initially select and later adjust different scaling factors for the system components.
and compare the behavior of the original and the scaled system. The scaling factor for the components was selected taking into account the dimensions foreseen for a small lab equipment and the availability of the components used to emulate the dynamic parameters, as well as adjustments that would make the behavior of the simulator resemble even further the vessel. Table I shows the values used for each parameter for both the real scale and the scaled down simulations, as well as the resulting scaling factor used for each component.

The results of the simulation for sea waves with a frequency of \( \frac{2\pi}{8} \text{ rad/s} \) and amplitude equal to the maximum allowed heave are shown in the Figure 3. The real scale and downscaled models were simulated as open-loop (PHC) and closed-loop (AHC) systems (Albers, 2010). The closed-loop results behave in a similar fashion in both models, effectively canceling in both cases the disturbances due to the sea waves. However, the open-loop results show that while the friction effects in the real scale model are almost unnoticeable for these operation conditions, in the downscaled model the friction effects cannot be completely compensated and affect more the output. Friction coefficients for lineal-railing mechanisms can be adjusted only within a range: they can be increased with brake mechanisms but they can only be decreased to a minimum value inherent to the properties of the railing system. Hence in the physical simulator these non-linearities will be more important and, if they are compensated correctly in small scale, they will be presumably also compensated in real scale.

### B. Adjustable Components as Elements of the Model

Figure 4 shows a comparison between the dynamic model formulated and the design proposed for the simulator. The components of the simulator were chosen taking into account four factors: appropriate scaling, adjustability, availability and price.

#### a) Vessel Heave:
A stepper servomotor and a transmission screw are used as a linear actuator to produce periodic or quasi-periodic specified heave motions. The motion characteristics can be adjusted in real-time from the PC, realistic (Sverdrup-Thygeson, 2007) or random waveforms can be input of the control algorithms implemented.
b) Pneumatic Springs and Dampers: The oleo-pneumatic spring $K_1$ is simulated with a simple pneumatic cylinder with an air tank. The viscous effects of the hydraulic components of the real spring $C_v$ are represented using with throttle valves which add resistance to the air flow, this principle will be used also to represent $C_v$. Equation 4 is used to establish a pressure control on the stiffness that represents the air reservoir in PHC given the rest of the parameters. Table II presents the values for the real and small scale models. The value of $K_2$ depends on the pipes material (steel) Young modulus $E$, cross section area $A$ and length $L$ (100m) (equation 5). In the physical simulator the mass and stiffness of the drill pipes are separately represented to simplify scalability and control. Electronic pressure regulators allow to control the pressure in the air tanks for the pneumatic springs whereas actuated throttle valves make possible to control remotely the viscous coefficient, making the parameters adjustable while running experiments.

$$K_1 = P \cdot A \frac{V_0(0.127 - 0.029)}{2H_v} - \frac{V_0}{(V_0 + 0.25H_vA)}K_a$$ \hspace{1cm} (4)

$$K_2 = \frac{AE}{L} = \frac{\pi((0.127)^2(0.127-0.029))}{2} \times 200 \times 10^9 = 1.02494 \times 10^7 \frac{N}{m}$$ \hspace{1cm} (5)

**TABLE II**

<table>
<thead>
<tr>
<th>Values for Maximum Load (100m pipe ×10)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>Elasticity constant $K_1$ [N/m]</td>
</tr>
<tr>
<td>Cylinder piston rod area $A$ [m^2]</td>
</tr>
<tr>
<td>Gas Volume $V_0$ [m^3]</td>
</tr>
<tr>
<td>Maximum Heave $H_v$ [m]</td>
</tr>
<tr>
<td>Adiabatic Constant $K_a$</td>
</tr>
<tr>
<td>Pressure $P$ [Pa]</td>
</tr>
</tbody>
</table>
The system will not be subject to high speed linear movements, and therefore it is reasonable to consider an incompressible laminar flow in the damper (with Mach and Reynolds numbers $M < 0.3$ and $Re < 2300$, respectively). This assumption allows to simplify the damper model, so the viscous friction force $F_v$ can be represented in terms of piston rod speed $V_r$, area $A_r$, throttle valve area $A_t$ and air density $\rho$ (equation 6). This scheme enables easy control of the viscous friction forces that represents the soil resistance and the drag forces of sea water against the pipes ($C_{v2}$).

$$F_v = V_r^3 \rho \cdot \frac{A_r^3}{2 \cdot A_t^2} \quad (6)$$

With equation 6 and the throttle area provided by the valves supplier, ideal curves were generated; an example is presented in Figure 6. A full range calibration of the pneumatic damper was performed (e.g. Figure 7). This process was repeated for several turns of the throttle valve (which changes the effective throttle area) in order to obtain an experimental function relating the number of turns in the valve to the viscous coefficient.

c) Electrical Linear Actuator emulating the AHC cylinder: A stepper servomotor and a linear actuator are being used to represent the actuator that, in real AHC systems, is usually a hydraulic cylinder by applying a transfer function before the
control signal. Using this kind of actuator instead of a hydraulic had the advantage of reducing the overall volume of the machine (since it does not require a pump) and facilitate cleanliness and transportability. The same servomotor controller used for the sea wave emulator motor was used for sake of standardization.

Fig. 9. Signal acquisition and generation diagram.

d) Sensors, Acquisition and User Interface: Different kind of sensors are included in the prototype to measure important parameters. A load cell provides feedback data concerning the drilling forces, a wire potentiometer measures the distance displaced by the cylinder. An IMU gives real time data regarding position and orientation (even though at this stage orientation changes are not foreseen). Matlab/Simulink is used as processing platform and to provide a graphical user interface (GUI). Data is acquired through the USB port thanks to a microcontroller and then processed for implementing the control laws and plotting data in real time. A simplified scheme of the signal acquisition system is shown in Figure 9.

III. CURRENT STATE OF THE SIMULATOR

A. Assembly and Initial Tests.

Fig. 10. The simulator in Oceanology, London. March, 2014.

All the mechanical and electrical components have been verified and basic motion profiles have been tested. The mechanical and electrical designs of the machine have been completed and the simulator has been assembled. Certain posterior modifications may be pertinent in order to increase its functionality; however, its modular design allows to add or remove components with ease.
B. Low cost Inertial Displacement Estimation

The first tests showed exactly what was expected about the position estimation through accelerometers. Even the slightest disturbance makes the error values grow rapidly, making this approach unsuitable for our application. Additional algorithms to the integration have been developed recently. (Madgwick, 2013) presents a working algorithm that prevents drift when retrieving position of periodic motions. Such algorithm works offline using recorded data from the IMU and currently a real-time algorithm is being developed.

The IMU selected (Figure 11) is equipped with accelerometers, magnetometers and gyroscopes, it sends the data via a serial port and in the PC the real-time version of the following algorithm is being implemented in a program that will link the data to Matlab/Simulink:

1) Retrieve gyroscope and accelerometer data.
2) Calculate orientation relative to the Earth (AHRS).
3) Calculate angular compensation for accelerometer readings.
4) Calculate linear accelerations
5) Calculate linear velocities (integration).
6) High-pass filter velocities.
7) Calculate linear displacements (integration).
8) High-pass filter displacements.
9) Send/Print the result and store it for next iteration.

C. Full Charaterization of Components and Instrumentation of a Vessel

Apart from single component calibration and validation such as that in Figure 7 it is essential to calibrate / verify the whole system. The overall scaling factor should be put to test and adjusted if required, for which it is imperative to instrument an actual vessel and retrieve data during drilling works. The Figure 12 shows where the IMU and potentiometer could be fixed.
in order to obtain inertial data about the heave and the relative position of the compensation platform from the cylinder base in a real vessel with PHC. The IMU is equipped with a digital signal processor and analog input pins, so the potentiometer signals could be read and stored along with the inertial data. The data can be stored in SD cards and it is designed to work with batteries.

IV. CONCLUSION

In this work we have described the conception and development of a physical emulator for the heave compensation systems applied in geotechnical drilling ships. Of course many of the concepts pertaining to these systems are used more extensively. The modular approach followed in design should facilitate adaptation to other heave compensation applications. Similarly, the implementation of advanced control strategies proposed for other HC systems (neuro-fuzzy, Zheng et al., 2012; non-linear, Do, 2008) is being developed. To facilitate user-training and data acquisition a unified control unit and GUI for simulator and real systems seems also interesting.

REFERENCES