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RESEARCHES ON THE DESIGN, MATHEMATICAL MODELING AND NUMERICAL SIMULATION OF THE COUNTERACTION MECHANISMS OF THE WAVES ACTION ON THE HANDLED LOADS BY CRAFT CRANES

Alexandru-Daniel MARINESCU[1], Mihail PETRACHE[2]

[1] Hydraulics and Pneumatics Research Institute INOE 2000-IHP, marinescu.ihp@fluidas.ro

[2] Lyra Hydraulics Consulting SRL, memento2020@gmail.com

Abstract: *Marine gas and oil offshore operations requires the installation of special equipments on the floor sea. This is done by installing of vessel-specific mechanisms operating in a dynamic position at the floor. The authors of the article describe such a mechanism, found in project stage at INOE 2000-IHP, which is oil actuated and has the role of counteracting the action of the waves on the handled loads by craft cranes. In order to deepening, establishing and developing the new lines and directions of research related to this innovative topic, for the first time approached in Romania, in the paper is presented for comparison and performance analysis a typical hydraulic device operating in the wave compensator mode, made by foreign specialists. By mathematical modeling and numerical simulations were analyzed the performance limits and it was evaluated a control strategy*

Keywords*: Marine exploitation, wave compensator, hydraulic mechanism for counteracting the waves action, mathematical modeling, numerical simulations*

1. Introduction

The lifting and deployment of the marine equipments for the exploitation of oil and gas fields is achieved by the use cranes from ships operating in a dynamic position. Fig.1 shows a marine vessel used in these types of operations. Due to costs, offshore operations and lifting operations should preferably be carried out under special conditions. Depending on the state of agitation of the sea, the lifting movement of the ship relative to the lifting or lowering load may be significant. This generates oscillations that can damage the load when it approaches on the sea floor and endangers people's lives (divers who monitor the operation near the sea floor). One way to alleviate this is make by using a balancing compensation device that tracks the movement dissociation of the load from the motion of the ship [1].

Fig. 1. Marine vessel equipped with load balancing compensation device

In this paper, the authors describe a mechanism designed at **INOE 2000-IHP**, oil acting to counteract the action of waves on the loads handled by craft cranes. This mechanism has been designed to maintain at fixed point of a load regardless of the up and down movements of the vessel (floating platform) on which the crane is mounted. Since at present, internationally, the researches in the field is much more advanced, the authors present for comparison and performance analysis a typical hydraulic device made **abroad** operating in the wave compensator mode.

2. Description of the principle scheme and operation of a crane with a compensator waves (project of INOE 2000-IHP)

The compensating mechanism is interleaved on the cable route between the crane winch and load, as you can see în Fig. 2, in which is schematically shown a crane mounted on a floating platform.

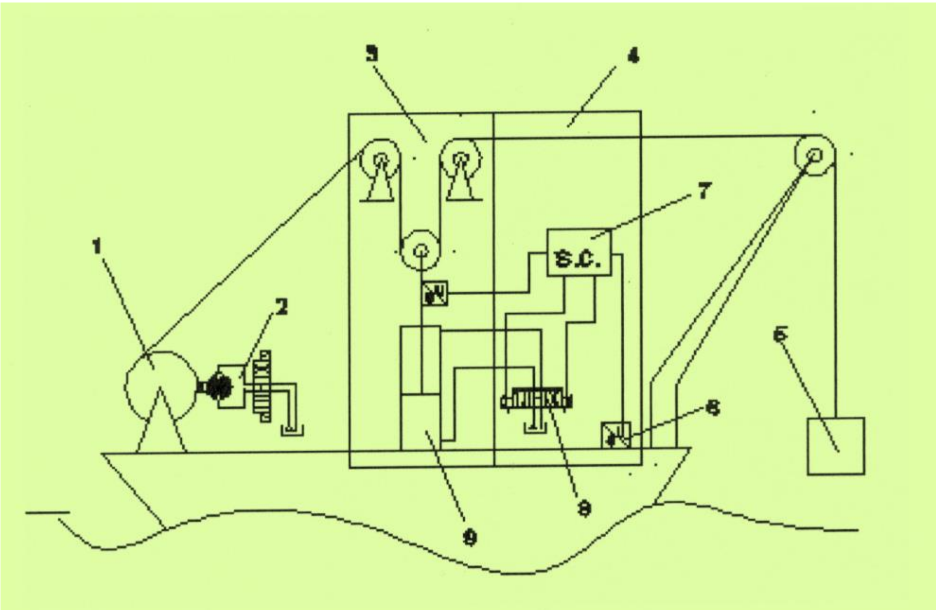

Fig. 2. Wave compensator crane schema

Caption

1-Winch; **2**-Winch action system; **3**-Waves compensator; **4**-Waves compensator action system; **5**-Load; **6**-Position transducer; **7**-Servocontroler; **8**-Proportional hydraulic directional valve ; **9**-Hydraulic cylinder with position transducer;

The operation of this mechanism is as follows: The position sensor 6 senses the up and down of the floating platform and in this moment controls by means of the servocontroller 7 and the proportional hydraulic directional valve 8, the extension and retraction of the hydraulic cylinder 9, maintaining as at fixed point the load 5, regardless of the floating platform movements on which crane stands. The extension or retraction of the hydraulic cylinder is proportional achieved with the amplitude of the floating platform motion, the hydraulic cylinder being actuated in a closed position regulating loop in order to obtain a minimum deviation of the load from the "zero level" position. The described mechanism will be referred to as the "**Wave Compensator**".

3. Wave compensation hydraulic device (international project)

3.1 Description

Some waves compensators use a hydraulic system to change the cable tension that feeds the load or to change the speed at which the cable is inserted and removed. Fig. 3 shows a simplified scheme of a particular model of such a device. The basic components are those using the cables connected to the load, as well as a strip connected to the hydraulic system. The winch normally runs at a constant speed, ie v_{ca1} is constant. The system can usually work in two ways: **passive** and **active**. In passive mode, the proportional valve is bypassed by another valve, so the fluid flows free into the active system; by adjusting the bypass valve the resistance to the fluid flow can be changed. The air pressure in the air accumulator is set to a value so as to offset the weight of the load in the water, and the pulley is positioned at the average stroke. It normally establishes the passive system's compliance and the weight compensation value depends on the weight of the load and the dominant frequency of the vessel's movement. In this way the system behaves like a spring-loaded damper. In active mode, the vessel movement is measured, and the proportional valve is used together with a pump to adjust the fluid flow in the active cylinder. Thus, the active cylinder behaves as a controlled force actuator according to the movement of the vessel, [1]

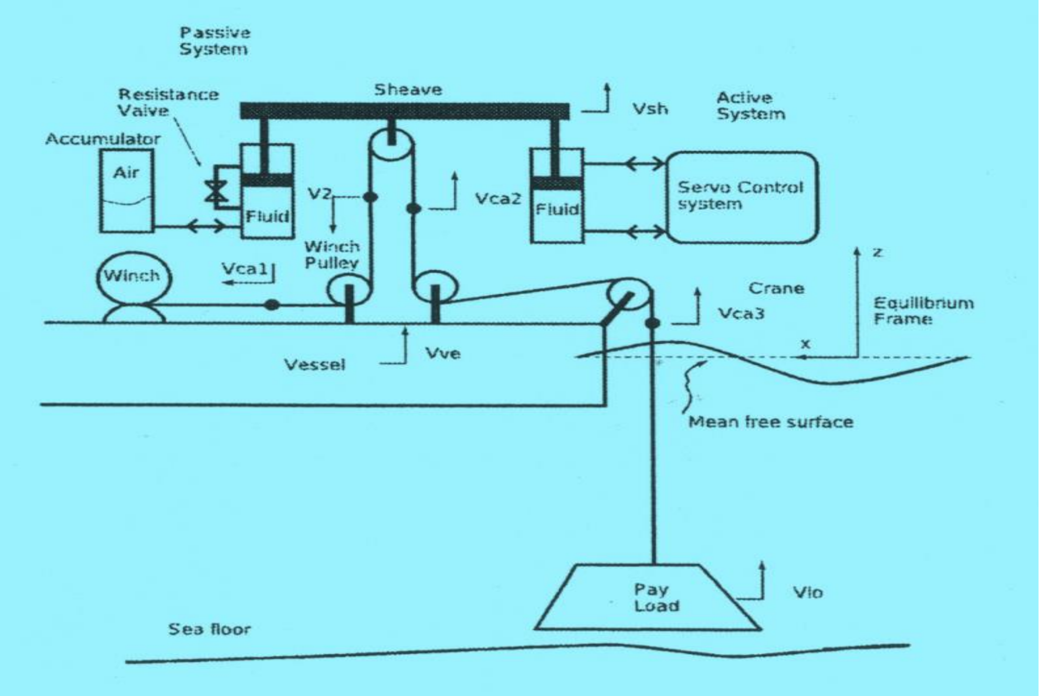

Fig. 3. Crane schema with wave compensator, [1]

3.2 Control objective and motion variables

The objective of the heave compensation control is to prevent the heave motion of the vessel reaching to the load. Tab. 1 shows the main motion variables associated with the wave compensator shown in the scheme in Fig. 3. This control objective(CO) can be achieved by obtaining a cable velocity from the crane as close as possible to the speed at which the winch feeds the cable , ie:

$$CO: v_{ca\,3} \approx v_{ca\,1} \tag{1}$$

Table 1: The relevant motion variables refer to the equilibrium frame on the average free floor, [1]

Description variable / + vessel's position	
$V_{ca\,1}$	Cable velocity at the winch / +vessel releed in
$V_{ca\,2}$	Cable velocity Crane side of sheave/ +vessel up
$V_{ca\,3}$	Cable velocity after crane /+vessel up
V_{sh}	Sheave velocity / +sheave up
V_{ve} $(=\hat{z}_{ve})$	Vessel heave velocity / +vessel up
Z_{lo}	Load positiom/ +vessel up
Z_{vc}	Vessel heave displacement / +vessel up

It is assumed that the pulley is located near the shaft by which the load is lifted. This means that the vertical speed (due to the combined force,of the roller and the vessel) is the same at the pulley and the lifting point; and therefore, the movement of the vessel in these points can be simulated with only one degree of freedom, that is the lifting movement. The vertical speed of the vessel in these two positions will be called *Vve*. In addition, it is assumed that most of the wave compensator compliance is concentrated on the air accumulator so that the following constraints can be applied, [1]

$$V_{ca2} = V_{ca3} \tag{2}$$

From the diagram of the freebody of the pulley close to the winch (see Fig. 2), we have:

$$V_2 = V_{ca1} - V_{ve} \tag{3}$$

And from the moving pulley attached to the sheave, we have::

$$V_{sh} = ½\,(V_{ca2} - V_{2)} \tag{4}$$

Therefore,

$$V_{ca2} = V_2 + 2V_{sh} = V_{ca1} - V_{ve} + 2V_{sh} \tag{5}$$

The above, and (1) and (2) translate to the next *control objective for the motion of the sheave:*

$$V_{sh} - ½\,V_{ve} \approx 0 \tag{6}$$

That is, when viewed from a reference frame fixed to the average free floor, the sheave moves at half of the vertical vessel velocity. Alternatively, (6) can be write, as

$$V_{ve} - V_{sh} \approx ½\,V_{ve} \tag{7}$$

Which specifies the control objective in terms of the sheave velocityof the sheave, in relation of the vessel, [1]

3.3 . Performance limitations

In this section, the conditions under which the system can achieve good compensation are analyzed. From the control objective and the constraints outlined above, we can have an optimal compensation in the situation that the next relations are satisfied, [1]

$$v_{sh} \approx \tfrac{1}{2} v_{ve} \tag{8}$$

$$|z_{ve} - z_{sh}| \leq \frac{1}{2}\ \textit{Stroke},$$

where *Stroke is the maximum operational cylinders extension.* .
If consider a sinusoidal movement:

$$z_{ve} = \bar{z}_{ve} \sin(\omega t)$$
$$v_{ve} = \omega \bar{z}_{ve} \cos(\omega t) \tag{9}$$

Then,from the control objective

$$v_{sh} = \frac{1}{2}\ \omega \bar{z}_{ve} \cos(\omega t) \Rightarrow z_{sh} = \frac{1}{2} \bar{z}_{ve} \sin(\omega t) \tag{10}$$

and

$$z_{ve} - z_{sh} = \frac{1}{2} \bar{z}_{ve} \sin(\omega t) \tag{11}$$

From the last expression , it follows that:

$$z_{ve} \leq Cursă \tag{12}$$

However, the movement of the vessel, is almost sinusoidal and can be more precisely modeled as a narrow Gaussian stochastic process. Under this association, the peaks of the process have a Rayleigh distribution (as Lloyd showed in 1989),[1, 2]. The amplitude $\bar{z}_P$, which is expected to exceeded with a probability P_r is,[1]:

$$\bar{z}_{Pr} = z_{RMS}\sqrt{-2 log P_r}$$

Then, for example, we can consider the value P_r = 0,001 to be constant, and this would result a probability that peaks not exceeding Stroke of 0,999 m, value with condition:

$$z_{RMS} \leq \frac{1}{3{,}7}\ Cursă \tag{13}$$

This last formula establishes a fundamental conditioning between the cylinders stroke and the amplitude of the vessel's vertical movement. This conditioning must be met in order to obtain a good lift compensation. For a stroke of 3.0 m,for example, there is a vertical movement limit of 0.8 m RMS, [1].

3.4. Mathematical modelling

3.4.1 Modelling Hypothesis

Crane wave-induced motion

The vertical movement of the crane will be considered a series of time. This will be obtained as a sum of random phases sinusoids and amplitudes and frequencies calculated from the spectral density of the vertical motion computed from the wave spectrum and the vessel response amplitude operators (Perez 2005),[1, 3]. Also, the boom will be considered rigid compared to hydrostatic system compliance, [1]

Deployed load

Asssociated with the load, there are: Mass: (Mt), Displacement (ρV_{dl}), Added mass (A)

Viscous damping force is limited, using Morisson's Equation:
$Bv(u)\frac{\rho}{2}C_d A_p|u|u$,
where C_d is the drag coefficient and A_p is the projected transversal aria on the movement direction (Faltinsen,1990), [1, 4]

Dynamics of cable

The cable is considered a spring with a constant K_c [N/m], which varies with the deployed cable length. The upper end moves at the v_{ca3} velocity and the bottom end is connected to the payload, so it moves at the speed viteza v_{l0} – see Fig. 3,[1]

Sheave

The sheave will be considered have a mass of M_{sh}. The inertia of all the sheaves will be neglected , [1].

Winch

The winch will be controlled such that the velocity v_{ca1} în Fig. 3 is not affected by the load motion.. This is a good assumption because the active heave compensator control reduces the tension variations in the cable, [1].

Passive System

The passive system will be assimilated to a spring damper. The damper quantifies some small losses associated with the movement of the cylinder and of the fluid in the pipe. Referring to Fig. 3, we can assume that all compression is taken over by the pressurized air. For an isentropic system, the rate between the initial and the final pressure is, [1]:

$$PV^{\upsilon} = P_0 {V_0}^{\upsilon} \tag{14}$$

where υ = 1,4 (Karnopp and others, 2006), [1, 5]. The initial pressure is determined by the weight of the load in water plus the weight of the sheave:

$$P_0 = 2(M_l g - \rho V_{dl}) + M_{sh} g A_{pc},$$

where A_{pc}, is the transversal area of the passive cylinder.

The instantaneous volume of compressed air is:

$$V = V_0 - \int \dot{V}\mathrm{dt}$$

Therefore, results from (14) that:

$$\left[\frac{V_0}{V_0 - \int \dot{V}\mathrm{dt}}\right]^{\upsilon} P_0 \tag{15}$$

The total force of the passive system incorporating the hydraulic losses is:, [1]:

$$F_P = A_{pc}\left[\frac{V_0}{V_{0} - A_{pc}\int(v_{sh}-v_{ve})\,dt}\right]^{\upsilon} P_0 + R_P|V_{sh}-V_{ve}|(V_{sh}-V_{ve}), \qquad (16)$$

Where R_P is hydraulic resistance representing the losses. Tab. 2 resume all the system's parameters, [1]:

Table 2: System parameters, [1]

Parameter	Description
M_l	Load mass
A	Added load mass
V_{dl}	Load volume displaced
C_d	Load drag coefficient
A_p	Load projected area
K_c	Cable stiffness at a given depth
M_{sh}	Sheave mass
P_0	Passive system initial pressure
V_0	Passive system initial volume
A_{pc}	Passive cylinder transversal area
R_p	Passive cylinder losses

Active Heave Compensator System

Consider the fluid in the incompressible active cylinder also in the ideal cylinder. Therefore, the active cylinder and the hydraulic energy that feeds the liquid into the cylinder will be collectively modeled as a source of effort (force) modulated by the amplifier of the controller. A first order system will be used to quantify the hydraulic and controller gap. Therefore, the active transfer model of the system is, [1]:

$$F_a(s) = \frac{1}{TS+1} K\left[v_{sh(s)} - \frac{1}{2} v_{e(s)}\right] \qquad (17)$$

where K is the gain of the controller, and τ = 0,6s is the constant time that characterizes the hydraulic plus the response of the control system. The above control law dissipates energy - this ensures the stability of the system. The amount of dissipated energy depends on the amount of control gain, which will be determined according by the meteorological conditions and payload [1].

3.4.2 Bond Graph Model

Due to the physical domain interference in which the components of the system model are: mechanical, hydrodynamic and hydraulics, a Graph-Bond (GB) modeling approach is chosen. With this approach, a small set of components is used to obtain a model based on the modeling hypothesis mentioned above. Once GB is done, the computational models such as block diagrams and state space equations are obtained by applying simple rules. Fig. 4 shows the Bond Chart of the system. For more details on GB modeling see, for example, Karnopp et al (2006), [1,5].

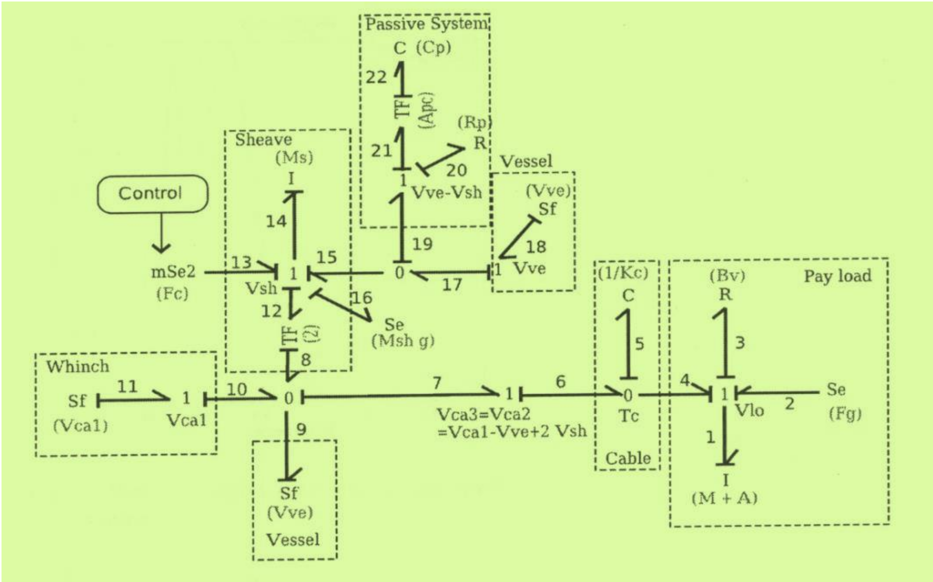

Fig. 4. Bond Graph System, [1]

3.4.3 State space equations
In accordance with the GB methodology, the following state variables are chosen:

$$x = [p_1, q_5, q_{22}, p_{14}]^T \tag{18}$$

Which correspond to the moment of the load $p_1 = (M_l + A)\dot{z_{l0}}$, cable deformation ($z_{ve}$ - z_{l0}), the instantaneous air volume in the passive system accumulator V, the sheave momentum, [1]:

$$p_{14} = M_{shvsh}$$

From Bond Graph BG, results:

$$\begin{aligned} \dot{p_1} &= e_2 - R_3(p_1/I_1) + q_5/C_5 \\ \dot{q_5} &= f_{11} + 2p_{14}/I_{14} - f_9 - p_1/I_1 \\ \dot{q_{22}} &= A_{PC}\,(f_{18} - p_{14}/I_{14}) \\ \dot{p_{14}} &= e_{13} - 2q_5/C_5 + R_{20}(f_{18} - p_{14}/I_{14}) + ApC_{22}^{-1}\,(q_{22}) - e_{16} \end{aligned} \tag{19}$$

Using variables x_i

$$\begin{aligned} \dot{x_1} &= g\rho V_{dl} - R_3(x_1/(M_l + A)) + K_c x_2, \\ \dot{x_2} &= V_{Ca1} + 2x_4/M_{sh} - Vve - x_1/(M_l + A), \\ \dot{x_3} &= A_{PC} + (V_{ve} - x_4/M_{sh}) \\ \dot{x_4} &= F_a - 2K_c x_2 + R_{20}\,(V_{ve} - 4/(M_{sh}) + + ApC_{22}^{-1}\,(x_3) - gM_{sh} \end{aligned} \tag{20}$$

where nonlinear relationship are R_3(u) and C_{22}^{-1} (V):

$$\begin{aligned} R_3(u) &= \frac{\rho}{2} C_d A_P |u| u \\ C_{22}^{-1}(V) &= \left[\frac{V_0}{V}\right]^{\upsilon} P_0 \end{aligned} \tag{21}$$

and F_a is the force of the active system modeled as a modulated effort source – as shown în Fig. 4, with a constitutive law expressed in (17). The only two nonlinearities in system are the passive system compliance (isentropic process) and the viscous damping of the load, [1].

4. Numerical simulations

For the simulation scenarios, a marine environment representative of Australia's northwest reef was chosen for a summer period. The sea is composed of a crest (a significant height of 1 m and a dominant peak of 13 seconds) and waves created by the wind (a significant wave height is 2 m for a peak period of 9 seconds). Both components are represented with a **JONSWAP** spectrum. The ricrest reaches from the sea, while the waves form a 30 degree circle arc [1]

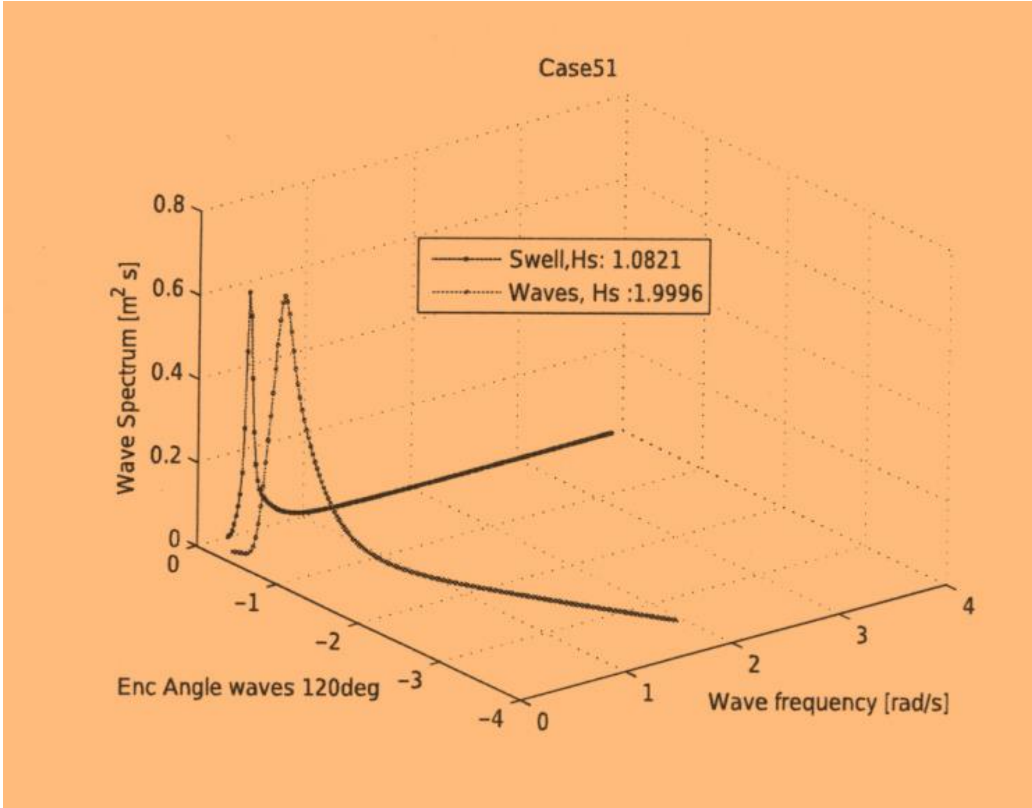

Fig. 5. Wave and crest spectrum, [1]

Fig.5 shows the corresponding spectra adopted. By combining the wave spectra with the vessels's response amplitude operators, the vertical motion spectrum was obtained where the crane is located This is shown in Fig. 6, [1]

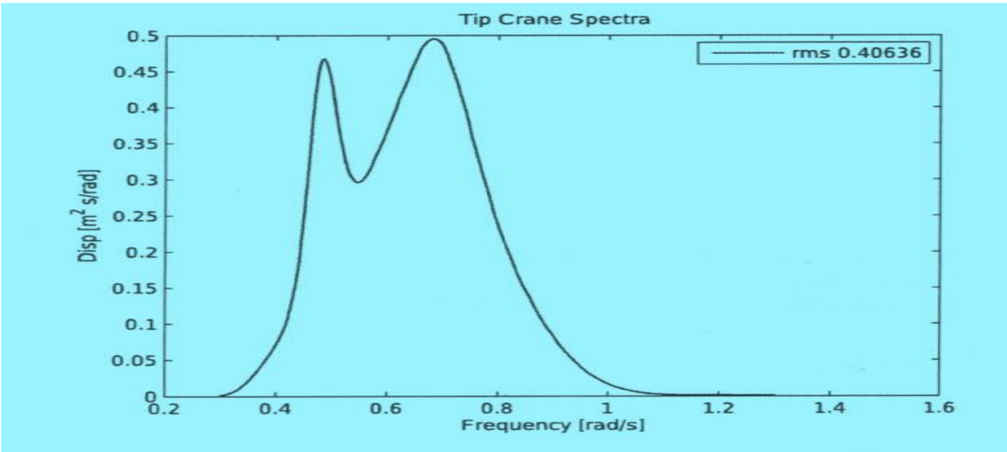

Fig. 6. The vertical motion spectrum where the crane is located, [1]

Timing simulations were conducted in a special environment chosen for various payloads commonly used in marine gas operations. Fig.7 shows an example of the vessel's lifting, cable tension and a 26000kg load displacement in relation to a depth of 150 m. The reduction in the **RMS (Root Mean Square)** value of the load displacement relative to the RMS of the vessel's displacement obtained in this the case was 99.1%. [1]

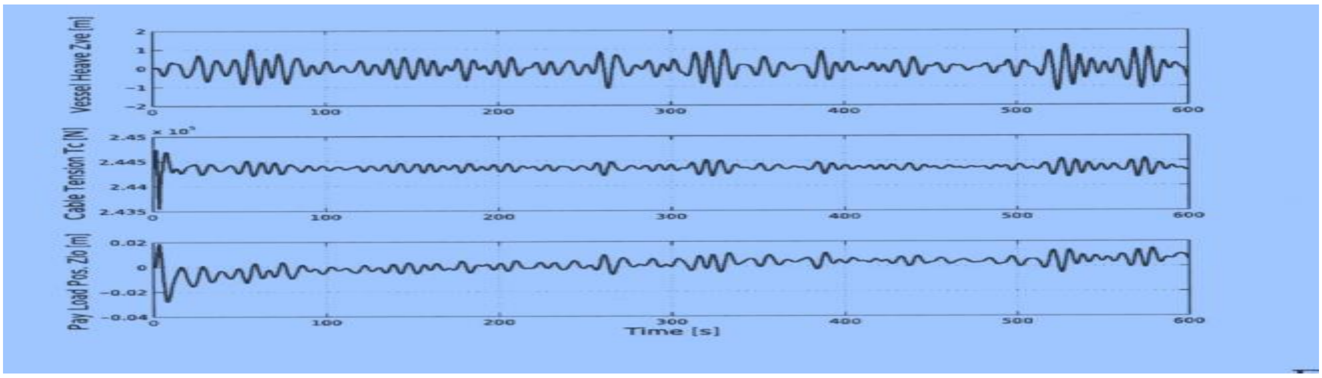

Fig. 7. Vessel movement, cable tension and load movement at a depth of 150 m, [1]

Fig. 8 shows the corresponding variables associated with the wave compensator, satisfying the constraints of +/- 1.5 x (half stroke)

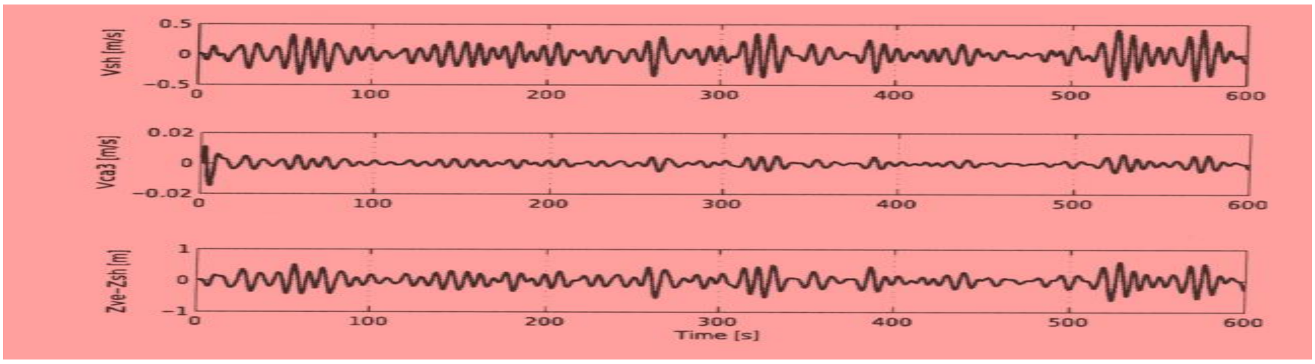

Fig. 8. Pulley speed, Cable speed and Relative displacement of the pulley relative to the vessel [1]

5. Discussions

In Chapters 3 and 4 the authors presented the modeling and performance analysis of a typical wave compensator for the handling of loads by cranes on seagoing vessels. An analysis of the operability and limitations of the vessel's movement and the capacity of the wave compensator stroke was made. These limitations represent a fundamental boundary that must not be overcome in order to achieve good performance. A mathematical model based on a Bond Graph approach has been described and all modeling hypotheses have been established. This model was then used to obtain a space state model of the system in terms of energy variables. Also, a simple control law has been proposed with an increase that can be used by the operator so as to obtain maximum performance without exceeding the limitations of the wave compensator. The system was tested in simulations with 8 different t loads, and the wave-effect mitigation (expressed in RMS) was over 95% in all the studied cases.

6. Conclusions

-The ongoing project at INOE 2000-IHP is in a field of smart specialization that belongs to energy, environment and climate change and wishes to contribute to energy efficiency.
-The innovative solution for achieving a wave compensator proposed by the authors of the article to be carried out in Romania is to use the hydraulic force in a closed-loop automatic regulation system to stabilize the loads handled by cranes boarded on floating platforms.
-Because the topic is for the first time approached in the field of research in our country, the authors considered it appropriate to deepen, establish and develop the necessary lines and directions to follow in order to achieve the proposed objective.
-When studying a similar project by specialists from abroad, the authors of the paper have proposed the realization and testing of a hydraulic mechanism to counteract the action of waves actually found in the project stage, and through mathematical modeling and repeated numerical simulations, to study its performances.
-By inventing and analyzing of these achievements in the field, INOE 2000-IHP is going to determine the optimal solution to compensate the disturbing effect of the waves on the floating cranes.

Acknowledgments

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