*active heave compensation (AHC),
subsea installation, offshore equipment,
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Marek SZCZOTKA[1]

SIMULATION OF AN AHC SYSTEM DURING OFFSHORE INSTALLATION

An active heave compensation system (AHC) is described in the paper. Active control enables us to compensate vertical vessel motions generated by sea waves. Such systems are used on a special offshore installations and construction vessels, performing the necessary work related to subsea field exploration. The most common AHC systems are based on an application of winches. Among many possible solutions, an electric drive is often used in order to control the speed and the moment, by the application of a frequency converter. The simple control system presented in the paper allows its real-time application on an onboard PC and through a PLC software. The system is implemented and operates currently on an offshore pipelay vessel.

SYMULACJA UKŁADU AHC PODCZAS INSTALACJI OFFSHORE

W pracy przedstawiono aktywny układ kompensacji falowania (AHC). Aktywne sterowanie układu napędowego urządzenia pozwala na kompensację pionowych ruchów jednostki pływającej, spowodowanych falowaniem morza. Systemy te są montowane na specjalnych jednostkach przeznaczonych do instalacji i prac konstrukcyjnych offshore, wymaganych dla uruchomienia produkcji i eksploatacji podwodnych pól naftowych lub gazowych. Najczęściej systemy te są wbudowane w układy wciągarek. Spośród wielu możliwych rozwiązań, w pracy przedstawiono napęd za pomocą silników elektrycznych, które przekazują odpowiedni moment i prędkość, dzięki zastosowaniu falownika. Prosty układ sterujący, zastosowany w modelu pozwala na implementację systemu w komputerze pokładowym na statku za pomocą PLC. Zaproponowany układ kompensacji został wykonany i jest obecnie używany na jednym ze statków do instalacji rurociągów na dnie morza.

1. INTRODUCTION

Natural resources accessible by onshore-based exploration are more and more exhausted. In order to meet global energy demand, the industry is forced to seek different sources of energy well into the seabed. This applies in particular to oil and natural gas, in some extend also to other materials and minerals. However, mining and other exploration activities carried out in offshore conditions are much more complex, dangerous and

[1]University of Bielsko-Biała, ul. Willowa 2, 43-309 Bielsko-Biała, Poland, email: mszczotka@ath.bielsko.pl

expensive than similar operations on land. This is not only due to the need for special units (platforms, mining, drilling, storage ships) and equipping them with specialized, complex machinery. Sometimes the decisive factor (both in the view of cost and technology applied) are environmental conditions, which must be taken into all design stages. Very rich in natural resources marine areas are often characterized by difficult weather conditions, which occur most of the year (eg. the North Atlantic, Barents Sea). Strong wind, ocean currents, insensitive waves and ice generates many issues and challenges. In addition the water depth comes as one of the key factor from financial and technical point of view.

Mostly, the systems designed for compensation of wave motion of the vessel reduce the vertical movement (heave). This is particularly the case when moonpool-based solutions are in use and the wire rope exit sheave is close to the vessel's centre. In certain applications (such as cranes operating in AHC mode at greater working radius), additional compensation for the rotary motion of the ship (roll and pitch) is required. The similar situation takes place in the system described in this article: because of an davit arm, the exit point is located sideways of the vessel, and the roll angle has to be included in the calculation algorithm. Sometimes, the active compensation system may react only to rotational movement of the vessel, such as the reel pipelay system presented in [1] (when pitching of the lay vessel generated high dynamic loads and instability).

One of the most common operations performed with AHC systems are the subsea lifting operations [2]. The weight of objects varies between kilograms and a few hundred tons. Each system demands a special design as the weight of the load would require different energy to be installed and accumulated. Typical operations are lowering different objects on seabed and lifting from seabed. The operation consists of several phases. Firstly, the load is lifted from the deck, lowered through the splash zone and, in normal operation mode, lowered down even for a few kilometers. The load overcomes the largest distance between the sea surface and seabed the in the normal mode, due to lowering time. No compensation is normally required in this phase. When the load is close to the seabed (some ten, twenty meters above), the operator switches the mode from normal to active compensation. Usually, the system requires some time (10-30 seconds) in order to stabilize and work in a fully active mode. When the system is ready, the operator, using a joystick can safely and precisely lower the load down to the seabed. The peak lowering speed is in the range of meters per minute (slow) and the greatest part of the energy is brought to compensate the vessel movement and the dynamic forces associated with the moving actuator elements (winch drum, hydraulic cylinder). Only a small part of it remains available to the operator, which can be used for load raising and lowering. When the load lands on the seabed, the rope is spooled out (still in the heave compensation mode) and the subsea robot can detach the hook. All the time, the system is monitored, including cameras on ROVs (Remote Operated Vehicles). The reverse operation (lifting from seabed to the vessel) is performed in a similar way. AHC systems are also applied to elimination of drilling string movement [3] (compensation of the platform heave).

On a rough sea, when no compensation is used, the load may not follow oscillatory movement of the vessel. This is highly dangerous in the systems with considerable resistance forces due to hydrodynamic drag, where the wire rope or cable may become "slack", Fig 1 a). In unstable motion of the load, the system is uncontrolled by the changing rope length (natural frequency of oscillations may become resonant). As a result, the rope can break. In the contrast, the AHC system eliminates most of the heave movement. Therefore the suspended load becomes practically steady or standstill (Fig. 1 b).

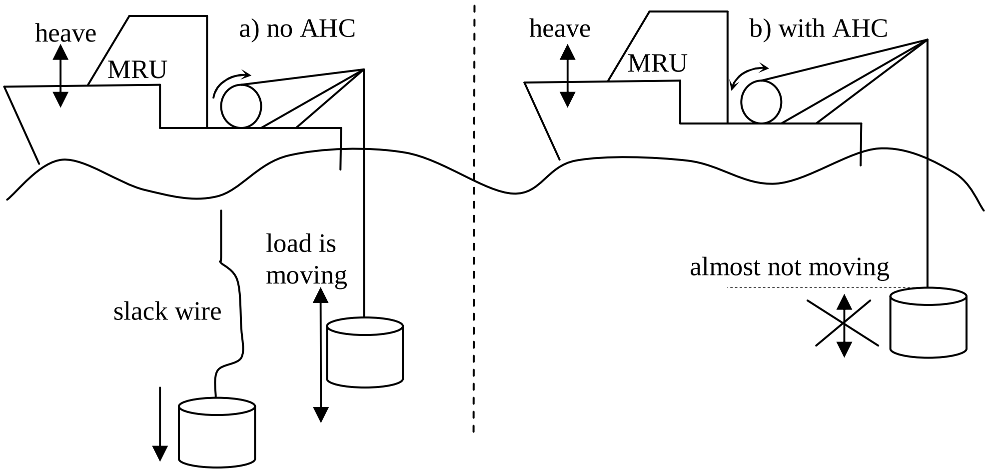

Fig.1. Lowering a load on rough sea. a) slack wire rope when AHC system is deactivated, b) AHC system activated, load oscillation amplitude reduced to minimum

In the following chapters a simple control system is described, based on a digital PID controller. Some assumptions concerning the acceleration and motor response time are included in the control algorithm. The computer simulation tool has been written in order to test the AHC system prior to the installation onboard. A problem similar to presented in this work has been solved using a nonlinear optimization methods (for given heave motion) [4] and in the connection with an A-Frame model, [5].

2. MODEL OF THE SYSTEM

2.1 Assumptions

The model is formulated with following assumptions:

- motion of the vessel is defined by six independent degrees of freedom, but only three of them are involved in the model: the heave, the roll and the pitch,
- the heave, the roll and the pitch (as well as their time derivatives) are known functions of time (obtained from MRU - Motion Reference Unit),
- transformation matrix $\mathbf{T}$ and its derivate $\dot{\mathbf{T}}$ are formulated, and location of any point on the vessel is given by homogenous transformation,
- last sheave location (components of the vector $\mathbf{r}_{ex}^{b}$) is known and remains constant during operation,
- all structural supporting structures are considered as rigid,
- the load has only vertical degree of freedom.

The transformation matrix, describing the motion of the vessel takes the form:

$$\mathbf{T} = \begin{bmatrix} \cos(\theta) & \sin(\theta)\sin(\varphi) & \sin(\theta)\cos(\varphi) & 0 \\ 0 & \cos(\varphi) & -\sin(\varphi) & 0 \\ -\sin(\theta) & \cos(\theta)\sin(\varphi) & \cos(\theta)\cos(\varphi) & z \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{1}$$

where: $z = z(t)$ is heave displacement,

$\theta = \theta(t)$ is the pitch angle,

$\varphi = \varphi(t)$ is the roll angle.

The location and speed of the wire exit point (last sheave in the wire route path) in global coordinate system can be calculated according to formulae:

$$\mathbf{r}_G = \mathbf{T} \cdot \mathbf{r}_{ex}^b \tag{2.1}$$

$$\dot{\mathbf{r}}_G = \dot{\mathbf{T}} \cdot \mathbf{r}_{ex}^b \tag{2.2}$$

where $\mathbf{r}_{ex}^b = \begin{bmatrix} x_{ex}^b & y_{ex}^b & z_{ex}^b & 1 \end{bmatrix}^T$ is the position of the exit point in the deck's local coordinate system,

$\mathbf{r}_G, \dot{\mathbf{r}}_G$ are position and velocity in the global coordinate system,

$\dot{\mathbf{T}} = \frac{d}{dt}\mathbf{T}$.

Both matrices $\mathbf{T}(z, \varphi, \theta)$ and $\dot{\mathbf{T}}(\dot{z}, \dot{\varphi}, \dot{\theta})$ are generated in every cycle, therefore six MRU signals are required in order to calculate the position and velocity of the rope exit point. Whenever the simplifications are possible (roll or pitch are not necessary), one should remove unnecessary signals from the system, as the cycle time strongly depends on the number of measurements. In some cases the linearization of the transformation matrix is beneficial. Linear form of (1) is often possible, since most vessels are equipped with a roll reduction tanks (used during offshore installation work).

2.2 Digital PID and control scheme

The combination of the PID controller with an additional feed forward loop gives good results, when the disturbance can be measured. Ideal, digital PID regulation algorithm has been applied in the control unit of the winch. It is defined by the know equation:

$$u(t) = K\left\{ e(t) + \frac{1}{T_I}\int_0^t e(t)dt + T_D \frac{d}{dt}e(t) \right\} \tag{3}$$

where K, T_I, T_D are proportional, integral and derivative gain, respectively,

$e(t)$ is the control error.

Applying trapezoid approximation for the integral part and backward differences, discrete form of (3) may be written as:

$$u(k) = u(k-1) + q_0 e(k) + q_1 e(k-1) + q_2 e(k-2) + q_3 e(k-3) \tag{4}$$

where $q_0 = K\left(1 + \frac{T_0}{2T_I} + 3\frac{T_D}{2T_0}\right)$, $q_1 = -K\left(1 - \frac{T_0}{2T_I} + 7\frac{T_D}{2T_0}\right)$, $q_2 = 5K\frac{T_D}{2T_0}$, $q_3 = -K\frac{T_D}{2T_0}$,

T_0 is the sampling time.

A classic control scheme with the feed forward loop used is shown in Fig. 2.

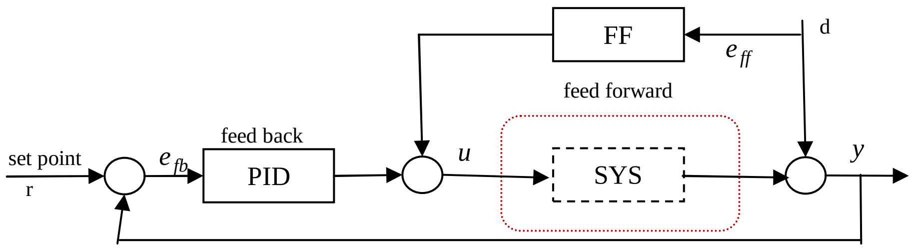

Fig.2. Regulation scheme, d - disturbance, u - calculated motor speed, y - actual drum speed, e_{ff} , e_{fb} control errors

The set point r is the joystick value controlling the lowering speed of the load in AHC mode. When no joystick signal is sent (joystick in neutral position), the load positioning is obtained. The control error is defined by the simple equation:

$$e_{fb} = v_{AHC} - \left([0 \quad 0 \quad 1 \quad 0] \cdot \dot{\mathbf{T}} \cdot \mathbf{r}_{ex}^{b} + \omega_d \cdot R(\varphi_{dA})\right) \tag{5}$$

where ω_d is actual drum speed,

φ_{dA} is actual rotation angle of the drum (measured by the encoder),

$R(\varphi_{dA})$ is active drum radius (depends on the actual number of rope layers, drum design, and a groove system used, eg. Lebus®).

The FF controller receives as the input the signal, which comes directly from the MRU:

$$e_{ff} = \dot{z} \tag{6}$$

The actual load position is calculated from the formula:

$$z_{load} = w_{out} - L_o + [0 \quad 0 \quad 1 \quad 0] \cdot \mathbf{T} \cdot \mathbf{r}_{ex}^{b} \tag{7}$$

where $w_{out} = \int_0^{\varphi_{dA}} R(\varphi_d) d\varphi_d$ is the unspooled rope length,

$R(\varphi_d)$ is a function describing actual rope wrap radius on drum,

L_0 is the wire route length (constant, depending on the system layout).

2.3 Operation modes of the AHC winch

Typical AHC winch control system has the following operation modes:

- Start and stop
- Normal mode
- Phase in
- Phase out
- AHC mode
- CT mode (constant tension)
- Empty hook mode

Normal mode is used in a normal lifting/rising operation, where large depth requires maximum speed of winch. The typical speed in normal mode is approximately 50-70m/min, which requires for 2500m sea depth 50min-36min of "waiting" time[2]. The system cannot be switched directly from normal mode to the AHC mode, and similarly, from AHC to normal mode. A smooth transition is performed in phase-in and phase-out modes. Phase-in and phase-out are in principle similar to the AHC mode, with exception to filtered controller "activity".

Let us assume the factor κ as:

$$\kappa = \frac{t - t_{pin}^{0}}{T_{pin}}\text{, for phase-in mode} \tag{8.1}$$

$$\kappa = \frac{t - t_{pout}^{0}}{T_{pout}}\text{, for phase-out mode} \tag{8.2}$$

where t is the current time,

t_{pin}^{0} is the time of activation of phase-in mode (operator switches to AHC mode),

t_{pout}^{0} is the time of activation of phase-out mode (operator switches to normal mode)

T_{pin}, T_{pout} is the duration of phase-in and phase-out modes (system constant)

Then the proportional coefficients defined in (3) are taken as:

$$K'_{FF} = \kappa \cdot K_{FF} \tag{9.1}$$

$$K'_{PID} = \kappa \cdot K_{PID} \tag{9.2}$$

where K_{FF}, K_{PID} are proportional gains of feed forward and PID controller.

The system calculates:

$$\varphi_d^j = \varphi_d^N (1-\kappa) + \kappa\left(E_{PID}\Delta t + \varphi_d^{j-1}\right) + \kappa E_{FF}\text{, for phasing-in} \tag{10.1}$$

$$\varphi_d^j = \left((1-\kappa)E_{FF} + \omega_C\right)\Delta t + \varphi_d^{j-1} + (1-\kappa)E_{FF}\Delta t\text{, for phasing-out} \tag{10.2}$$

$$\dot{\varphi}_d = \frac{\varphi_d^j - \varphi_d^{j-1}}{\Delta t} \tag{10.3}$$

where $E_{FF}(t, \dot{z})$ and $E_{PID}(t, v_{AHC} - v_{load})$ are feed forward and feedback regulator outputs defined in (4),

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[2] Lowering speed may vary significantly between applications. The requirement for high speed in normal mode has to comply with other winch modes, which could be challenging.

$v_{load} = (\dot{\mathbf{r}}_G)_3$ is the load vertical speed, given in (2.2),

v_{AHC} is desired lowering speed in AHC mode (joystick input),

$\varphi_d^j, \varphi_d^{j-1}$ are actual and previous rotations of the drum,

φ_d^N is the drum position if the winch would work in normal mode,

$$\omega_C = \frac{\omega_E - \omega_0}{T_{pout}} t + \omega_E - \frac{\omega_E - \omega_0}{T_{pout}} \left(t_{pout}^0 + T_{pout}\right)$$

$$\omega_E = \frac{v_N}{\max(R(\varphi_d))},\quad \omega_0 = \frac{v_{AHC}}{\max(R(\varphi_d))},$$

v_N is nominal lowering speed in normal mode.

In the so-called regular AHC mode, the angle of winch rotation is calculated according to formula:

$$\varphi_d^j = \varphi_d^{j-1} + E_{PID}\Delta t + E_{FF}\Delta t \tag{11}$$

and the drum speed from (10.3).

An important property of the control module is consideration of the response time of the electric motor. Time delay is included in the simulation program and PLC software uploaded to the vessel onboard PC. In addition, a module responsible for an overload check has been implemented, using the information from motor characteristics and parameters provided by the manufacturer. These details are not considered here.

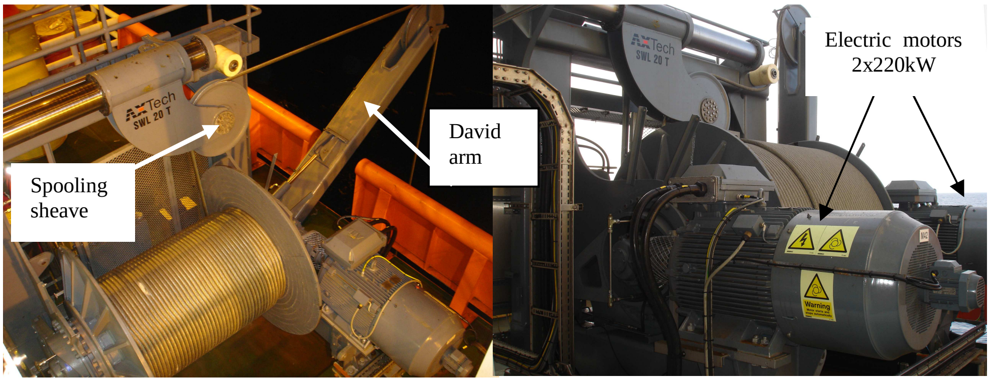

Fig.3. SWL 20T AHC electric winch: (left)- the system in working configurations, (right)-winch in a out of service position - design: AXTech AS

3. AHC WINCH - EXAMPLE

An example active heave compensated winch is presented in Fig. 3. It has the maximum capacity of 20 metric tons and operational depth up to 2500m. As shown in Fig.3, the winch has a special jib (kind of a davit arm) that allows the wire to move directly down to the sea. This is very beneficial for both efficiency and rope life time, since the wire has to be routed only via two sheaves. Two large electric motors (specially designed for the purpose) are the power source. The torques from motors are transferred to the drum by two angled, planetary gear boxes, incorporated inside the drum space (compact design).

The winch is used during installation and recovery of different subsea modules, valves, pumps and compressors and other equipment required on the seabed.

4. SIMULATION EXAMPLE

The simulation results are presented in Fig.4. "*HeaveSpeed*" series represent measured vessel motion (during trial tests), and "*WireSpeed*" is the wire rope speed (calculated by the control system) with respect to the vessel coordinate system. At the beginning, the winch is started-up (a few seconds of acceleration time from 0 to 10m/min) and in first 30-32sec constant lowering speed is maintained. Next, the operator activates the AHC mode. During 15sec, the system synchronizes with heave motion, and both time series are coincident. At simulation time 70sec one could observe a difference in both lines. It is generated due to system overload, where the control allows for less compensation precision. Some misalignment in AHC mode is generated because of joystick input (lowering with a small speed between simulation time 80sec-130sec). Phase-out is initiated at approximately 155sec and the system returns to previous settings used in normal mode (lowering with constant speed 10m/sec).

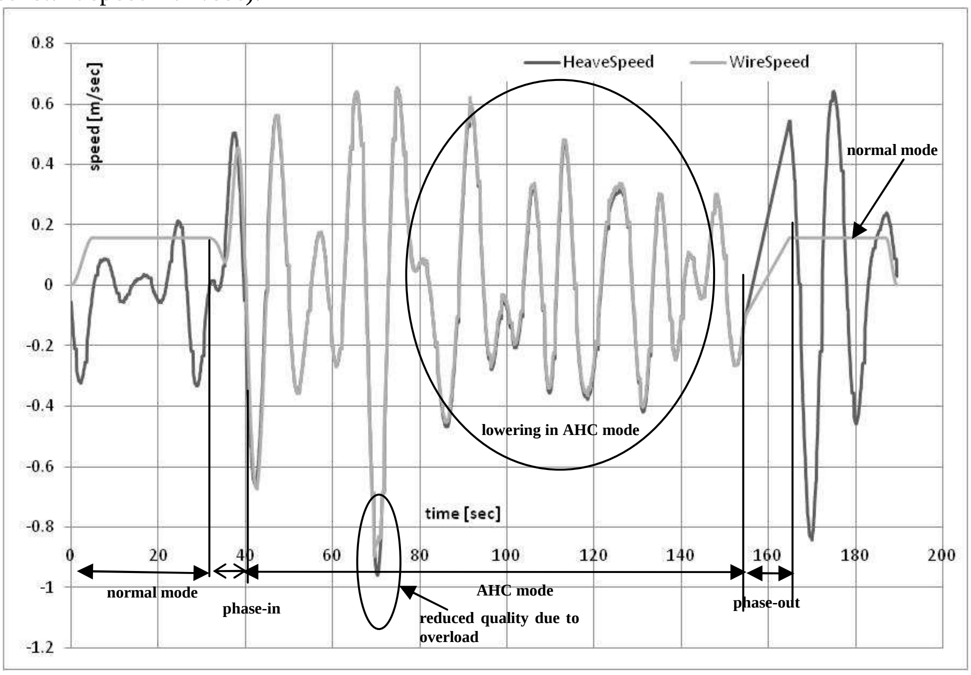

Fig.4. Load speed (WireSpeed) vs. vessel heave speed (HeaveSpeed)- winch response to heave motion in different operation modes

Fig.5. presents the vertical speed and movement of a load as well as the signal from the joystick (desired speed while lowering the load during compensation). Active heave compensation period can be recognized by limited oscillations, and regular load movement.

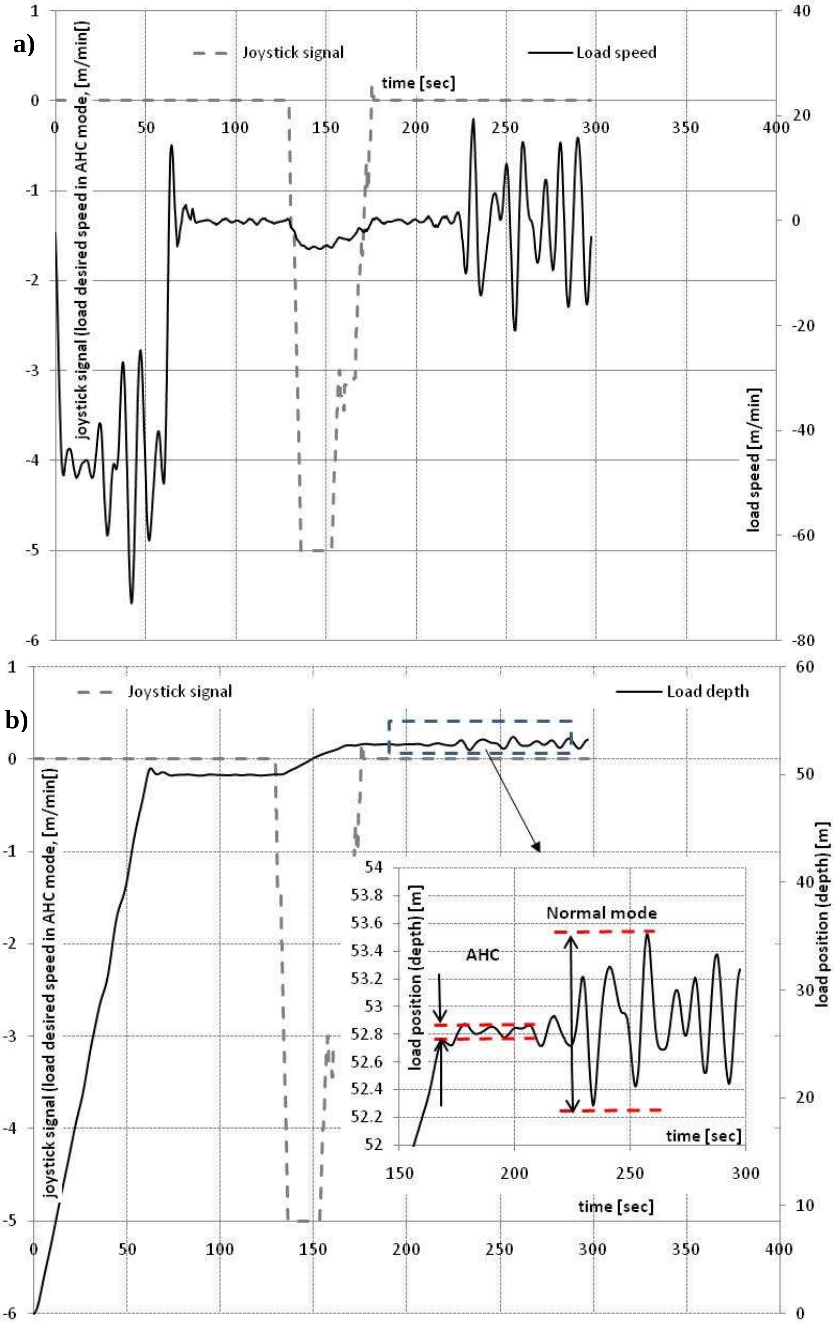

Fig.5. a) The course of load speed during lowering operation, b) load position and the difference in compensated and not compensated load oscillations

As it can be observed from Fig.5. b), the heave compensation module limits the load motion amplitude to a few centimeters. Irregular vessel motion, used in this work, consists of sudden changes in MRU heave signal. Additionally, the system was not ideally calibrated during the sea trial, which has particular influence on the results and compensation quality.

5. CONCLUSIONS

Without an AHC system, many operations performed on rough sea would not be possible. For significant wave height of 3 meters, typical AHC system provides sufficient reduction: the load moves only approximately 10-20cm. The quality of 92-96% of vessel's heave compensation is enough to perform an offshore installation as it would be done almost on a calm sea. Therefore, without AHC, transportation of a module worth sometimes a few millions of euro (sophisticated subsea machines, robots) would not be possible without serious damage. Then the only solution would be to wait for good weather, by which performance of the entire project would suffer a lot.

The quality of heave compensation in the physical system on the vessel achieved is approximately 94%. Generally, it is very difficult to obtain more than 96% compensation. Special drive components and hardware has to be used, which significantly rises the price of the equipment. It is important in technical realization that the model is stable, response and cycle time are within limits. Due to irregular nature of waves, the system has to be equipped with an overload protection system. Such a system, in the case when higher wave comes and required speed is out of limits, allows for decreasing AHC performance in a short period of time. When conditions stabilizes, nominal AHC settings are recovered. The winch presented in this work has been equipped in such algorithms.

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