2008 American Control Conference
Westin Seattle Hotel, Seattle, Washington, USA
June 11-13, 2008

A Heave Compensation Approach for Offshore Cranes

Jörg Neupert, Tobias Mahl, Bertrand Haessig, Oliver Sawodny, Klaus Schneider

Abstract—**Offshore installations during harsh sea conditions results in rigorous requirements in terms of safety and efficiency for the involved crane system. Hence a heave compensation system based on heave motion prediction and an inversion based control strategy is proposed. The control objective is to let the rope suspended payload track a desired reference trajectory in an earth fixed frame without being influenced by the heave motion of the ship or vessel. Therefor a combination of a trajectory tracking disturbance decoupling controller and a prediction algorithm is presented and evaluated with simulation and measurement results.**

I. Introduction

Today offshore installations such as underwater conveying systems for oil and gas or wind parks are becoming more and more important. The processing equipment for the development of oil and gas fields is already installed on the seabed. Hence the availability for maintenance, repair and replacement decreases compared to floating or fixed production platforms. Dealing with such installations results in rigorous requirements in terms of safety and efficiency for the involved crane system (see Figure 1). The main goal is to guarantee operation during harsh sea conditions in order to minimize production downtimes. Also essential is the safety of the operators on board. Situations may occur where the control of the payload get lost.

Besides the navigation/positioning problem the wave-induced motions of the ship/vessel lead to critical tension of the rope. The tension should not be less than zero in order to avoid slack rope situations. The peak value must not exceed a safety limit. Therefore heave compensation systems are utilized to improve the operability of offshore installations during harsh sea conditions. Additionally the vertical motion of the payload can be reduced significantly, which makes an exact positioning of the load possible.

The problem of safe and efficient offshore operation in harsh sea conditions was addressed by numerous publications in the past. Basically there are two different approaches for the heave compensation. The fist one is based on passive heave compensation systems. For example Frederick et al. [1] determined the stiffness and damping characteristics of a cage mounted passive heave compensation system by a sequential quadratic programming optimization. Hatleskog and Dunnigan [2] investigated a heave compensation system, which uses pneumatic cylinders to compensate the wave-induced motions of a drilling vessel. The second approach is to derive an active heave compensation system. Korde [3] for example presented a system for drilling ships subjected to irregular-wave excitation. The drilling ship is modeled as coupled mechanical oscillators. By adding an additional actuated oscillator the vertical motion of the load can be damped.

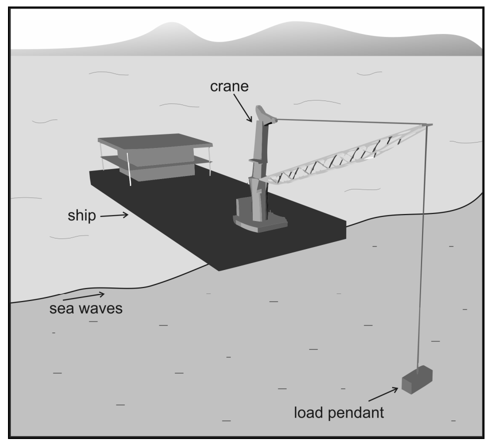

Fig. 1. vessel with offshore crane

Other publications focus not only on the heave compensation above- or underwater but also on the wave synchronization during water entry of the payload (see Sagatun et al. [4]). Skaare and Egeland [5] presented a parallel force/position controller for crane loads in marine operation. A two phase crane control was discussed in Messineo et al. [6]. The first phase is the heave compensation, where the tension of the rope is controlled to a constant value equal to the weight of the load. In the second phase the relative velocity between the sea waves and the load is controlled in order to decrease the hydrodynamic slamming forces that affect the payload at water-entry. Sagatun [7] presented a combination of an augmented impedance control for the water entry phase and a passive heave compensation system. The compensation system consists of pneumatic cylinders in order to obtain constant spring stiffness and hydraulic cylinders to damp the oscillating system.

This paper presents a heave compensation system based on ship/vessel motion prediction and an inversion based control strategy. There are basically two requirements on the compensation systems for offshore cranes. The first is to let the load track a desired reference trajectory generated out of

Jörg Neupert, Tobias Mahl, Bertrand Haessig, and Prof. Oliver Sawodny are with the Institute for System Dynamics, Universitaet Stuttgart, Stuttgart, Germany (phone: +49-711-68569899; fax:+49-711-68566371; e-mail: joerg.neupert@isys.uni-stuttgart.de).
Dr. Klaus Schneider is divisional director at the Liebherr Werk Nenzing GmbH, Postfach 10, A-6710, Austria, (e-mail: k.schneider@liebherr.com).

the hand lever signals of the operator in an earth fixed reference frame. The load should move at the assigned reference velocity in this frame decoupled from the wave-induced ship motion. The second requirement is a modular heave-compensated crane. That means, the crane systems used for offshore installations can be installed at many different kinds of ships or vessels. Additionally the estimation and prediction algorithm for the vertical ship/crane motion must be independent of the ship/vessel type.

The paper is organized as follows. In section 2 the dynamic model of the system consisting of the hydraulic actuator (winch) and the flexible rope is derived. Based on this model a linearizing control law is formulated in section 3. In order to stabilize the control system a feedback controller is derived. Fig: 2 illustrates the general control structure.

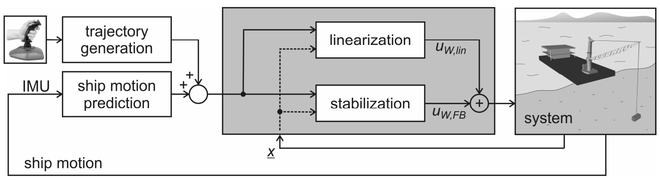

Fig. 2. general structure of the heave compensation

In section 4 the estimation and prediction of the ship/vessel motion is presented. Therefore a model is formulated based on the dominant modes of the heave motion. The modes are obtained by a Fast Fourier Transformation and a peak detection algorithm. The estimation and prediction is done by a Kalman Filter. In section 5 simulation and measurement results are presented. At the end of the paper concluding remarks are given.

II. Dynamic Model

The heave compensation system under consideration basically consists of a hydraulic-driven winch, a crane-like structure and the rope suspended load. For the modeling of the system it is assumed that the crane's structure is a rigid body. The rope suspended payload can be approximated by a spring-mass-damper system (see Fig. 3).

To approximate the flexible rope the equivalent mass m_{eq} and the stiffness of the spring c_{rope} has to be calculated. Utilizing the Hook's law the deformation $\varepsilon(z)$ for a rope at an arbitrary position z can be obtained from

$$\varepsilon(z)=\frac{\sigma(z)}{E}=\frac{F(z)}{EA_{rope}}=\frac{g}{EA_{rope}}\left(\left(depth-z\right)m_{l,rope}+m_{load}\right). \quad (1)$$

$\sigma(z)$ is the tension of the rope, E the Young's modulus, $F(z)$ the static force acting on the rope at position z, A_{rope} the intersectional area of the rope, g the gravitational constant, $depth$ the distance of the load from the sea level, $m_{l,rope}$, and m_{load} the mass of the rope per meter and the mass of the payload respectively.

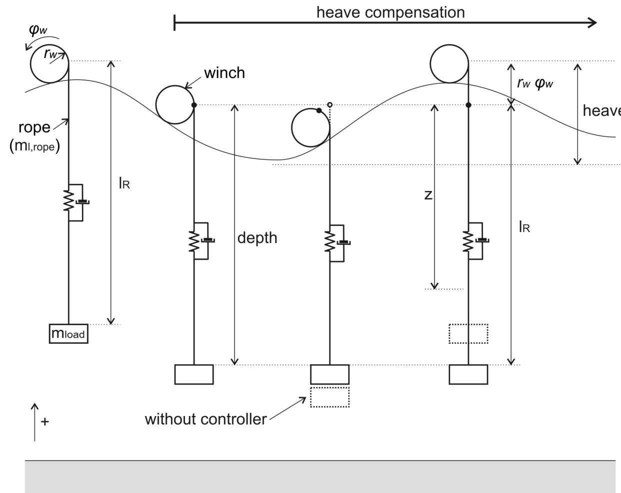

Fig. 3. heave compensation system

The extension of the whole rope Δl_R is obtained by using equation (2).

$$\begin{aligned}\Delta l_R &= \int_0^{depth} \varepsilon(z)\,dz \\ &= \underbrace{g\left(\frac{depth}{2}m_{l,rope}+m_{load}\right)\frac{depth}{EA_{rope}}}_{\text{rope suspended load}} = \underbrace{\frac{g m_{eq}}{c_{rope}}}_{\text{approximation}}\end{aligned} \quad (2)$$

Evaluating (2) results in

$$m_{eq}=\left(\frac{depth}{2}m_{l,rope}+m_{load}\right) \text{ and } c_{rope}=\frac{depth}{EA_{rope}}. \quad (3)$$

Utilizing the method of Newton/Euler, the second order differential equation for the motion of the rope suspended payload is obtained (see (4)). The load oscillations are exited by winch accelerations $\ddot{\varphi}_W$ and the second derivation of the heave motion $\ddot{w}$.

$$m_{eq}\ddot{l}_R+d_{rope}\dot{l}_R+\frac{EA_{rope}}{depth}\left(l_R-depth\right)=m_{eq}\left(r_W\ddot{\varphi}_W+\ddot{w}\right) \quad (4)$$

The actuator for the heave compensation system is the hydraulic-driven winch. The dynamics of this actuator can be approximated with a first order system.

$$\ddot{\varphi}_W=-\frac{1}{T_W}\dot{\varphi}_W+\frac{2\pi K_{V,W}}{i_W V_{mot,W} T_W}u_W \quad (5)$$

$\ddot{\varphi}_W$ and $\dot{\varphi}_W$ are the winch angular acceleration and velocity respectively, T_W the time constant, $V_{mot,W}$ the volume of the hydraulic motor, u_W the input voltage of the servo valve and $K_{V,W}$ the proportional constant of flow rate to u_W.

III. Control strategy

In order to derive a control law, the dynamic model of the system is formulated in the following form. The disturbance d is defined as the 4th derivation of the heave motion. In Neupert et al. [9] a similar model extension by a disturbance model was presented. Hence the system's relative degree is equal to the disturbance's relative degree and a disturbance decoupling by Isidori [8] is possible.

$$\begin{aligned} \dot{\underline{x}} &= f\left(\underline{x}\right)+g\left(\underline{x}\right)u_W+p\left(\underline{x}\right)d \\ y &= h\left(\underline{x}\right) \end{aligned} \tag{6}$$

With the states $\underline{x}=\begin{bmatrix} l_R & \dot{l}_R & \varphi_W & \dot{\varphi}_W & \ddot{\text{w}} & \dddot{\text{w}} \end{bmatrix}^T$, equation (4) and (5), and the model extension, the dynamic equations are obtained as follows

$$\dot{\underline{x}}=\begin{bmatrix} x_2 \\ -\dfrac{EA_{rope}}{m_{eq}depth}\left(x_1-depth\right)-\dfrac{d_{rope}}{m_{eq}}x_2-\dfrac{r_W}{T_W}x_4+x_5 \\ x_4 \\ -\dfrac{1}{T_W}x_4 \\ x_6 \\ 0 \end{bmatrix}+\begin{bmatrix} 0 \\ \dfrac{2\pi r_W K_{V,W}}{i_W V_{mot,W} T_W} \\ 0 \\ \dfrac{2\pi K_{V,W}}{i_W V_{mot,W} T_W} \\ 0 \\ 0 \end{bmatrix}u_W+\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 1 \end{bmatrix}\ddddot{w} \tag{7}$$

$$y = x_1 - r_W x_3 = l_R - r_W \varphi_W$$

In order to investigate the flatness property of the proposed model of the system, the relative degree has to be determined.

A. Relative degree

The relative degree concerning the system's output is defined by the following conditions

$$\begin{aligned} &L_g L_f^i h\left(\underline{x}\right)=0 \qquad \forall i=0,\ldots r-2 \\ &L_g L_f^{r-1} h\left(\underline{x}\right)\neq 0 \qquad \forall x\in R^n \end{aligned} \tag{8}$$

The operator L_f represents the Lie derivative along the vector field f and L_g along the vector field g respectively. With the output y a relative degree $r=4$ is obtained. The disturbance's relative degree is obtained to $r_d=4$ by utilizing (8) with the vector field p in stead of g. Because the order of the system is $n=6$ a second order internal dynamics exists and y is not a flat output. It can be shown, that this internal dynamics is the disturbance model. In our case the internal dynamics consists of a double integrator chain. This means, the internal dynamics is instable. Hence it is impossible to solve the internal dynamics by on-line simulation. But for the here given application case not only the disturbance $d=\ddddot{w}$ but also the states $x_5=\ddot{w}$ and $x_6=\dddot{w}$ can be estimated and predicted by the method discussed in section 4. This makes the simulation of the internal dynamics unnecessary and a trajectory tracking and disturbance decoupling controller can be derived.

B. Trajectory tracking controller

The trajectory tracking and disturbance decoupling controller can be formulated as follows based on the input/output linearization method.

$$\begin{aligned} u_{W,lin} &= \frac{1}{L_g L_f^{r-1}h\left(\underline{x}\right)}\left(-L_f^r h\left(\underline{x}\right)-L_p L_f^{r-1}h\left(\underline{x}\right)\ddddot{\text{w}}+y_{ref}^{(r)}\right) \\ &= \frac{m_{eq} i_W V_{mot,W} T_W depth}{2\pi EA_{rope} r_W K_{V,W}}.\left(\left(-\frac{EA_{rope}}{m_{eq}depth}\right)^2\left(l_R-depth\right)+\ldots\right. \\ &\qquad \left.\ldots\frac{r_W}{m_{eq}T_W depth}\dot{\varphi}_W-\frac{EA_{rope}}{m_{eq}depth}\ddot{\text{w}}-\ddddot{\text{w}}+\ddddot{y}_{ref}\right) \end{aligned} \tag{9}$$

To stabilize the resulting controlled system a feedback term is added. The term (equation (10)) compensates the error between the reference trajectories $\underline{y}_{ref}$ and the derivatives of the output y.

$$u_{W,FB}=\frac{\sum_{i=0}^{r-1}k_i\left[L_f^i h\left(\underline{x}\right)-\overset{(i)}{y}_{ref}\right]}{L_g L_f^{r-1}h^*\left(\underline{x}\right)} \tag{10}$$

The feedback gains k_i are obtained by the pole placement technique. The control structure is illustrated in Fig. 2.

IV. Estimation and Prediction of heave Motion

The first part of this section gives a proposal how the whole ships/vessels movement can be estimated by measuring with an **I**nitial **M**easurement **U**nit (IMU). As a crucial claim, any ship specific information should be used for this estimation. The second part discusses a short time prediction problem. Here only the cranes heave motion gets predicted. This reduces the complexity of 6 degrees of freedom to only one without losing information needed. As desired before, the prediction will also be completely independent of a ship model.

A. Measurement of ship movement

The ship/vessel, considered as a rigid body, has 6 degrees of freedom. With an IMU the ship's displacement from steady state can be measured with high accuracy. These low cost stand alone motion sensors have 3 accelerometers for measuring surge, sway and heave and 3 rotation rate sensors for roll, pitch and yaw. To obtain the desired relative position of the ship, a double-integration of the acceleration signals and a single-integration of the rotation signals are needed. To reduce typical errors like sensor noise, bias and misalignment of the accelerometers and ensure a stable integration, the signal conditioning can be realized like presented in [10].

If the IMU isn't fastened at the payload's attachment a simple transformation between the sensor's coordinate system and the payload's attachment coordinate system leads to the desired heave motion.

B. Prediction of the payload's attachment heave

The fact that the payload's attachment heave isn't totally chaotic, but depends on the ship dynamics and the sea condition, makes it possible to calculate a prediction for its movement. A short time prediction can even be done without knowing anything about the ship's properties.

The main idea of this prediction method is to detect the periodic components of the measured heave motion and use them to calculate the future heave progress. Therefore the measured heave motion $w(t)$ between two time points t_0 and T gets decomposed in a set of N sinus waves, the so called modes, and an additional arbitrary term $\upsilon(t)$. This results in a heave motion model described by

$$w(t)=\left(\sum_{i=1}^{N}A_i\sin(2\pi f_i t+\varphi_i)\right)+\upsilon(t)\quad i=1,...,N\quad t_0\le t\le T \tag{11}$$

where A_i is the amplitude, f_i the frequency and φ_i the phase of the i-th mode. The aim of the prediction is to estimate how much modes will be necessary for an accurate forecast of the length T_{Pred} and to fit the three parameters for each mode.

The prediction method's structure is displayed in Fig. 4

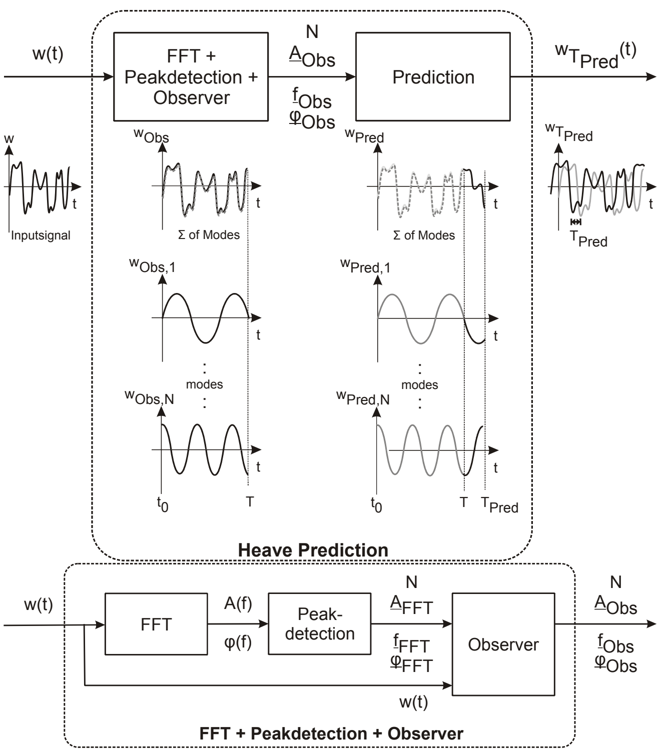

Fig. 4. structure of the prediction method

At first a Fast Fourier Transformation (FFT) is applied to the measured heave motion $w(t)$. The analyzed length and sample time of the input signal are to be chosen in a way that the maximal heave motion's frequency can be detected and the desired frequencies resolution is reached. The peaks of the resulting amplitude-response over frequency A(f) will then be extracted by a peak detector. This leads to a first estimation of the mode's amplitudes and frequencies which will be stored in the respective parameter vectors $\underline{A}_{\mathrm{FFT}}$ and $\underline{f}_{\mathrm{FFT}}$. The modes quantity N is equal to the number of detected peaks. By taking the phase-response φ(f) into account the mode's phases $\underline{\varphi}_{\mathrm{FFT}}$ can be defined too. With these parameters, which will be updated online, the heave motion model described in (11) can be parameterized. Evaluation of real measured heave motion data shows the need for a permanently updated model (see Fig. 5).

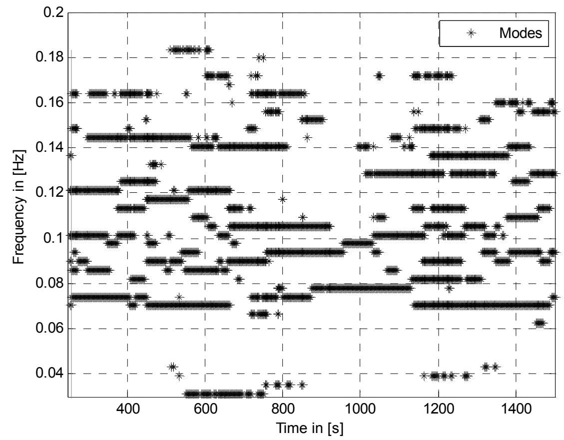

Fig. 5. mode detection for real measured data

Here the detected peaks of a ship's heave under harsh sea conditions are displayed. It can clearly be seen, that modes change during measurement.

In the next step an observer adapts the parameter vectors by comparing the measured heave motion $w(t)$ with the modeled one. This is necessary because the FFT only detects mean values of a long period whereas the observer is able to take latest modifications into account. With these new parameter vectors denoted by $\underline{A}_{\mathrm{Obs}}$, $\underline{f}_{\mathrm{Obs}}$ and $\underline{\varphi}_{\mathrm{Obs}}$ the heave motion prediction can be performed by using (11) again.

1) Observer

The observer design depends on a heave motion model described by a set of ordinary differential equations (ODEs). To transform model (11) to a set of ODEs, there are two opportunities possible. On the one hand, the heave motion can be modeled as a nonlinear system which enables the observer to estimate all parameters needed for the heave prediction. However due to the claim to receive an online prediction this method isn't applicable on recent computers. On the other hand a linear model can be used instead. Here only the mode's frequencies will not be fitted again. But these are estimated with high accuracy by the FFT, anyway. Choosing the linear method, a Kalman Filter can be used. This yields to an observer equation displayed below.

$$\begin{aligned}\dot{\hat{\underline{x}}}&=A\hat{\underline{x}}+L\left(w-\hat{w}\right) & \hat{\underline{x}}(t_0)&=\underline{x}_0\\ \hat{w}&=C\hat{\underline{x}} & t_0&\le t\le T\end{aligned} \tag{12}$$

The system matrices A and C result from the heave motion model described as follows, whereas the prediction results also depend on defining the correction matrix L properly.

To transform the heave model described in (11) suitable for an observer, one single mode can be defined by the ODE

$$\underline{\dot{x}} = A_i \underline{x} = \begin{pmatrix} 0 & 1 \\ -(2\pi f_i)^2 & 0 \end{pmatrix} \underline{x} \quad \underline{x}(t_0) = \underline{x}_{0,i} = \begin{pmatrix} A_i \sin(\varphi_i) \\ 2\pi A_i f_i \cos(\varphi_i) \end{pmatrix} \qquad (13)$$
$$w_i = C_i \underline{x} = \begin{pmatrix} 1 & 0 \end{pmatrix} \underline{x} \qquad i = 1, \ldots, N.$$

Applying the parameter vectors obtained by the FFT, summating all modes together and introducing an offset-state representing the non-periodic term $\upsilon(t)$ leads to the observer heave model

$$\underline{\dot{x}} = A\underline{x} = \begin{bmatrix} A_1 & 0 & \cdots & \cdots & 0 \\ 0 & A_2 & \ddots & & \vdots \\ \vdots & \ddots & \ddots & \ddots & \vdots \\ \vdots & & \ddots & A_N & 0 \\ 0 & \cdots & \cdots & 0 & 0 \end{bmatrix} \underline{x} \qquad \underline{x}(t_0) = \underline{x}_0 = \begin{bmatrix} \underline{x}_{0,1} \\ \underline{x}_{0,2} \\ \vdots \\ \underline{x}_{0,N} \\ 0 \end{bmatrix} \qquad (14)$$
$$w(t) = C\underline{x} = \begin{bmatrix} C_1 & C_2 & \cdots & C_N & 1 \end{bmatrix} \underline{x}.$$

The choice of the L matrix elements can be achieved by using the Filter design of Kalman and Bucy. This requires solving the Riccati equation (solution is P) and computing the gain-matrix L as described in (15).

$$PC^T R^{-1} CP - AP - PA^T - Q = 0$$
$$L = PC^T R^{-1} \qquad (15)$$

Here Q, used as a design parameter, is chosen as a diagonal matrix penalizing fast modes harder than slow ones, whereas R influences all modes equally.

The parameters fitted by the observer can be extracted from its states. Based on the equations of one single mode

$$\hat{x}_{1,i}(t) = A_{Obs,i} \sin\left(2\pi f_{FFT,i} t + \varphi_{Obs,i}\right)$$
$$\hat{x}_{2,i}(t) = 2\pi A_{Obs,i} f_{FFT,i} \cos\left(2\pi f_{FFT,i} t + \varphi_{Obs,i}\right) \qquad (16)$$

the new parameters can be calculated by

$$f_{Obs,i} = f_{FFT,i}$$
$$\varphi_{Obs,i} = \arctan\left(\frac{\hat{x}_{1,i}(T)}{\hat{x}_{2,i}(T)}\right) - 2\pi f_{FFT,i} T \qquad (17)$$
$$A_{Obs,i} = \frac{\hat{x}_{1,i}(T)}{\sin\left(2\pi f_{FFT,i} T + \varphi_{Obs,i}\right)} \qquad i = 1, \ldots, N.$$

2) Prediction

The last part of the prediction method is the prediction calculation itself. Therefore (11) can be used taking the parameter vectors adapted by the observer which results to:

$$w_{\mathrm{Pred}}(t) = \left(\sum_{i=1}^{N} A_{Obs,i} \sin(2\pi f_{Obs,i} t + \varphi_{Obs,i})\right) + \upsilon(t) \quad i = 1, \ldots, N, \; T \le t \le T_{\mathrm{Pred}} \qquad (18)$$

The progress of the non-periodic term $\upsilon(t)$ cannot be predicted. As it is equal with the observer's offset state it should be defined as a constant with

$$\upsilon(t) = const. = \hat{x}_{2N+1}(T) \qquad T \le t \le T_{\mathrm{Pred}}. \qquad (19)$$

To give a short impression of the heave prediction's performance, simulation results are presented below. Therefore real IMU signals of a ship under harsh sea conditions were used to reproduce the heave motion. Fig. 6 shows the predicted and measured heave motion over time. The prediction interval T_{Pred} was chosen to 1 second. For a better illustration, the predicted heave motion was shifted back in time afterwards. So an error free, predicted signal would be congruent with the measured one.

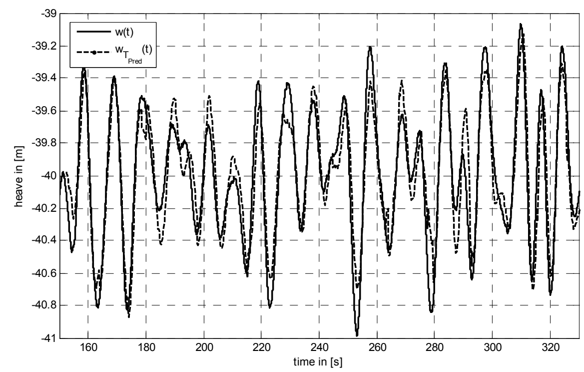

Fig. 6. prediction of 1s over time

V. Simulation and Measurement Results

In Fig. 7 the simulated tracking behavior of the heave compensation system can be seen. The reference trajectory is generated by a hand lever signal and the crane is subjected to a heave motion. For this simulation only a linearizing controller without stabilization was used. With this setup the excitation of the payload's attachment heave, which is displayed in the first graph of Fig. 7 can be reduced by factor 5. The reason why these oscillations aren't rejected completely is that the pump-motor system was simulated with a dead time, which is not considered in the controller design.

Applying the observer and closing the loop of the control system improves the tracking behavior tremendously. As it can be seen in Fig. 8, the simulated position displacement never grows beyond ± 3cm.

For the first two simulations, the heave prediction was switched of. Fig. 9 illustrates the open loop tracking behavior of the payload position with a heave prediction in the range of the actuator's dead time (0.2 seconds). It can be seen, that as soon as the linearizing controller gets activated, which is at time 250s, good heave compensation results are achieved. Comparing the tracking with and without heave prediction, a clear improvement can be seen.

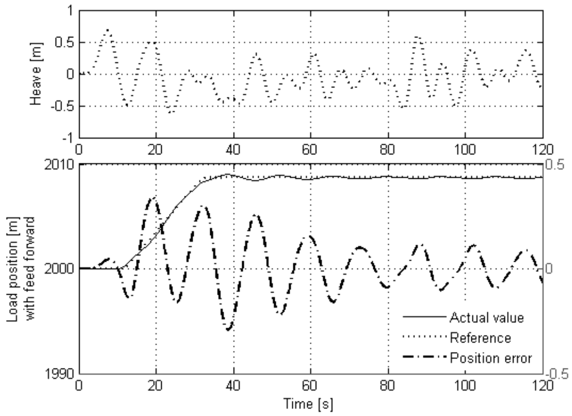

Fig. 7. simulation results with linearizing controller

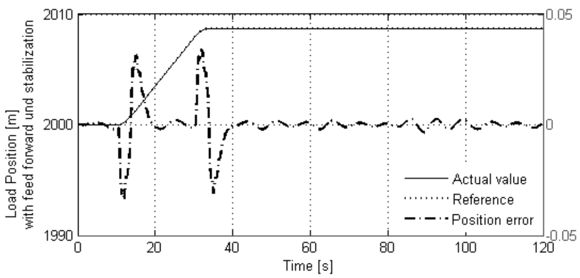

Fig. 8. simulation results with trajectory tracking controller

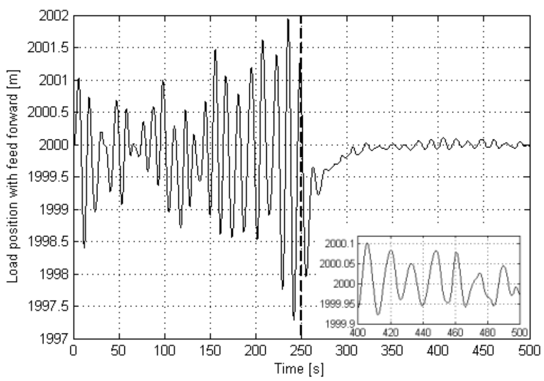

Fig. 9. simulation results with heave motion prediction and linearizing controller

To prove the simulation results, measurements with an experimental setup were performed. Fig. 10 shows this setup. The ship's heave is imitated by the winch HW whereas the second winch (SW) realizes the heave compensation such that the payload's vertical position should remain constant. The heave signal used for the active heave compensation is based on the rotation angle of the winch HW.

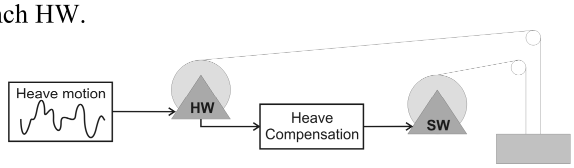

Fig. 10. experimental setup

The measurement results displayed in Fig. 11 illustrate the tracking behavior with constant reference value. The resulting position error is equal with the simulation results.

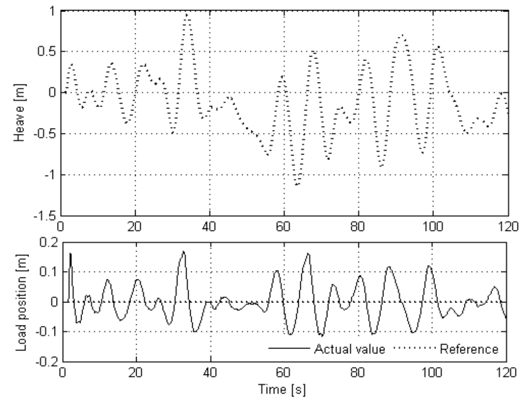

Fig. 11. measurement results with linearizing controller

VI. Conclusion

In this paper a heave compensation approach for offshore cranes is presented. The dynamic model of the compensating actuator (hydraulic driven winch) and the rope suspended load are derived. Based on this model a trajectory tracking controller is designed. In order to compensate the wave-induced ship/vessel motion, the heave motion is defined as a time-varying disturbance and analyzed concerning decoupling conditions. With a model extension these conditions are met and a inversion based decoupling control law is formulated. In order to stabilize the system, a observer is utilized to reconstruct the unknown state from a force measurement. Further the compensation performance can be increased by predicting the heave motion. A prediction method is proposed, where no ship/vessel models or properties are needed. Simulation and measurement results validate the heave compensation method.

References

[1] F. R. Driscoll, B. Buckham, Meyer Nahon, "Numerical Optimization of a Cage-Mounted Passive Heave Compensation System," *in OCEANS 2000,* vol. 2, Sept. 2000, pp. 1121-1127.

[2] J.T. Haltleskog, M.W. Dunnigan, "Heave Compensation Simulation for Non-Contact Operation in Deep Water," *OCEANS 2006,* Sept. 2006, pp. 1-6.

[3] U. A. Korde, "Active heave compensation on drill-ships in deep water," *Ocean Engng.,* vol. 25, no. 7, pp. 541-561, 1998.

[4] S. I. Sagatun, T. A. Johansen, T. I. Fossen, Finn G. Nielsen, " Wave Synchronizing Crane Control during Water Entry in Offshore Moonpool Operations," *in Proc. o f the IEEE Conference on Control Applications,* Glasgow, Scotland, U.K., pp. 174-179, 2002.

[5] B. Skaare, O. Egeland, "Parallel Force/Position Crane Control in Marine Operations, *IEEE Journal of Oceanic Engineering,* vol. 31, no. 3, pp. 599-613, 2006.

[6] S. Messineo, F. Celani, O. Egeland, "Crane feedback control in offshore moonpool operations," *Control Engineering Practice (2007),* doi: 10.1016/j.conengprac.2007.05.003

[7] S. I. Sagatun, "Active Control of Underwater Installation", *IEEE Trans. Control Systems Technology,* vol. 10, no 5, pp. 743-748, 2002.

[8] A. Isidori, "*Nonlinear Control Systems*", Berlin Springer Verlag, 1995

[9] J. Neupert, T. Mahl, O. Sawodny, K. Schneider, "A Nonlinear Control Strategy for Boom Cranes in Radial Direction," *Proc. of the IEEE American Control Conference,* New York, USA, pp. 25-30, 2007

[10] John-Morten Godhavn, "Adaptive tuning of heave filter in motion sensor," *Proc. of OCEANs 1998,* vol. 1, 1998 pp. 174-178.