An Effective Heave Compensation and Anti-sway Control Approach for Offshore Hydraulic Crane Operations

Yingguang Chu, Filippo Sanfilippo, Vilmar Æsøy and Houxiang Zhang

Department of Maritime Technology and Operation
Aalesund University College
Larsgårdsveien 2, 6009 Ålesund, Norway

{yich & fisa & ve & hozh}@hials.no

***Abstract* – Offshore hydraulic cranes are difficult to operate safely, accurately and efficiently due to their heavy structure, large inertia, non-intuitive control interface and load sway issues that result from external disturbances. This paper presents an effective heave compensation and anti-sway control approach for offshore crane operations, which is based on robotic arm kinematics and energy dissipation principles. Unlike common operator-based joint-by-joint control procedures, this automated method is more flexible, allowing for more intuitive crane operations and more accurate positioning of the hoisted load. In particular, a unique feature of this approach is that the two control functions of heave compensation and anti-sway are transparently combined and simulated in an integrated modelling environment. The system architecture integrates the control model for crane operations, the hydraulic system model for hydraulics characteristic analysis, the 3D model of the crane to be controlled, the vessel and the environment for visualisation.**

The proposed control algorithm and simulation model can be extended to any type of crane model regardless of its configuration or degree of freedom (DOF) without influencing the effectiveness of the method. The hydraulic model is built by using Bond Graph elements and integrated with the control model in the 20-sim simulation environment. The crane operation can be simulated and controlled by the operator using a 3-axis joystick, which provides a transparent user interface. Related simulations were carried out to validate the efficiency and flexibility of the system architecture. As a case study, a 3-joints knuckle boom crane was implemented and tested. The simulation results prove the presented control algorithm for heave compensation and anti-sway to be a valid and efficient solution.

Index Terms* – *offshore hydraulic crane, heave compensation, anti-sway, modeling and simulation

I. INTRODUCTION

Today, the offshore industry is one of the most innovative and technologically demanding sectors in the world. The working environment poses harsh challenges to offshore operations and thus rigorous requirements to the stability and efficiency of offshore solutions. Norway holds a unique position in the global maritime industry through its concentration of some of the world's leading ship owners, shipping and ship equipment firms, yards and a whole range of specialized maritime service providers. Given by the geographic location, rigorous missions in tough weather and under difficult working conditions have led to the development of what are now the world's most technologically advanced solutions.

Offshore cranes are the major actors on-board of platforms and vessels in transporting and lifting operations (Fig. 1). Under rough sea conditions, offshore activities involving crane operations result in many problems such as load sway, positioning accuracy, collision avoidance and manipulation security, etc. Unlike cranes mounted on fixed bases, offshore crane operations are significantly influenced by the ship motions resulting from currents and waves. The dynamic forces generated from the heave motion of the vessel and the sway movements of the pendulate load have extensive effects to the crane structure and the lifting wire. In the context of such a challenging operating environment in the maritime industry, it is still quite common to use relatively simple control interfaces to perform offshore crane operations. In most cases, the operator has to handle an array of levers and buttons to operate the crane joint by joint. When considering working efficiency and safety, this kind of control is extremely difficult to manage and relies on extensive experience with high operating skill level of the operators. In particular, when a large load sway occurs under extreme sea conditions, reliable control is almost impossible to achieve. Currently, a huge amount of resources were spent on training operators annually and a great deal of cost can be wasted during the downtime waiting for a better weather condition.

Fig. 1 Offshore crane operations.

This paper introduces a more flexible control approach for offshore crane operations that provides heave compensation and anti-sway combined functionalities. The rest of paper is organized as follows. In Section II, an overview of the related research work is given. In Section III, the proposed control algorithm for heave compensation and load anti-sway functions are described. As a case study, a 3-joint knuckle boom crane which implements the proposed method is also presented. In Section IV, modelling and simulation of the proposed approach is discussed. At last, the conclusion and future work are outlined in Section V.

II. RELATED WORK

Much research and many investigations have been done to help reduce the risks in offshore crane operations. As implied in the introduction section, heave compensation and anti-sway control are the two crucial technical challenges in the offshore crane control field.

Regarding the heave compensation problem, two main different approaches have been extensively investigated. Passive Heave Compensation (PHC) is the simplest and chronologically most dated of these two approaches. A PHC system can simply be modeled as a spring damping system by means of hydraulic cylinders and compressors. The second method, Active Heave Compensation (AHC), differs from PHC by having controlled actuators that actively try to compensate for the heave movements. To monitor the ship movements, commercial offshore cranes usually adopt some motion detection units, e.g. Inertial Measurement Unit (IMU) and Motion Reference Unit (MRU). Then, according to this data input, a control system calculates how the actuators have to react to the movements. The actuators can be electric or hydraulic winch systems or hydraulic cylinders. For example, Jörg Neupert et al. [1] proposed an approach based on heave motion prediction and an inversion based control strategy to let the suspended payload track a desired reference trajectory in an earth fixed frame by using a hydraulic-driven winch. Bjørn Skaar [2] and Saverio Messineo [3] presented heave compensation controllers for moonpool operations also utilizing winch systems. Svein I. Sagatun [4] proposed an augmented impedance control scheme combined with a spring–damper based PHC strategy for offshore underwater installation. The U.S. navy developed heave compensated ramps that can move between two vessels using hydraulic cylinders [5]. Other investigations using heave compensated cylinders are found for offshore drilling operations [6].

One of the major issues of using AHC winch systems is the wire fatigue during the heave compensation phase. Repeatedly releasing and retracting under high frequency causes a small part of wire to suffer repeated cyclic bending and loading so as to consequently deteriorate much more rapidly than other parts. To minimize this issue, Shinichi Takagawa et al. [7], and Rolls-Royce Marine AS [8], proposed new design solutions to lengthen the life of the cable. However, the problem is still inevitable. Another disadvantage of these approaches is the high-energy consumption, which is needed to keep running the driven heave compensated systems. For instance, a 400-ton, all-electrical driven AHC winch system has 7720kw power consumption while an electrical-hydraulic system has 9400kw [9]. However, no research in existing literatures was found using hydraulic cylinders for heave compensation in crane operations.

Compared to heave compensation, the anti-sway control function is even more difficult to achieve. Load sway motion has more than one DOF, and is affected by more complex external disturbances. Generally, two main approaches can be found in previous research. The first one uses input shaping and filter techniques to smooth the input command and suppress load oscillations [10]. This approach is mainly designed and tested for land-based cranes with one DOF, for example gantry cranes. By filtering the input-shaped command signals, crane operators can position the crane more efficiently without generating significant load sway energy. From the operating point of view, this approach does help to reduce generating load sway. However, it is not able to reduce the sway generated by the continuous external disturbances during offshore crane operations, e.g. the currents, waves, winds and even human operational errors. The other approach uses a so-called Active Rider Block Tagline Control System (ARBTCS) to suppress the load swing, which includes a driven winch system too [11]. This ARBTCS winch system is configured with the crane in a way that cannot be integrated with the AHC winch system. In fact, in existing literature, not much work has been done to incorporate both the heave compensation and anti-sway control procedures in a combined control method. For regular above-surface crane operations like goods transporting and ROV lunching, it would be much simpler if the crane could compensate the heave and sway by using only the crane motor and cylinders.

Our research group investigated the possibility of implementing a flexible architecture for maneuvering different maritime cranes from a control point view [12]. Based on the control algorithm, two control functions of heave compensation and anti-sway are transparently combined by only using the crane's hydraulic motor and cylinders. The concepts of the compensation algorithms are simple but efficient: the crane end tip should always maintain a constant position in order to compensate the heave motion of the vessel, and follow the load movements to damp out the sway energy.

When choosing a modeling technique to simulate the presented system, several aspects were considered, including the requirement for an energy-based approach due to the interaction with human operators, the multi-domain nature of the problem and the need for a modular approach. For all these reasons, the so-called Bond Graph (BG) method [13] becomes a natural choice to model the system. BG method is a highly modular energy-based approach for modeling and simulation of multi-domain dynamic systems. In [14], our research group presented the modular prototyping system architecture that allows for modeling, simulation and control of different kinematic models of cranes by using BG method. Due to the fact that offshore cranes are usually much heavier, stiffer and mostly hydraulically actuated, classic robotic control doesn't assure the same effectiveness to offshore crane control. Hence, in the simulation model, a hydraulic system was also included.

III. Control Algorithms for Heave Compensation and Anti-sway

Instead of controlling each joint separately, the control of the crane is realized by maneuvering the crane end tip through solving its kinematics model to obtain the needed angle for each joint. This method is widely applied in industrial robotic arm control systems. The advantages regarding to offshore crane operations include the following aspects: firstly, it offers more flexibility in controlling different types of cranes regardless of their configurations; secondly, it is more intuitive for the operator to position the load; and last, specifically for offshore crane operations, heave compensation and anti-sway control can be achieved easily.

A. A flexible control algorithm for offshore crane operations

A relevant feature of the proposed approach is that the crane model can be separated from the control functions, thus giving it the flexibility to control different types of cranes. To illustrate the method, a general offshore hydraulic crane was implemented as an example (Fig. 2). The crane has three rotational joints that are actuated by hydraulic motor and cylinders. The coordinate systems, link dimensions and joint angles are shown and corresponding to the notations in D-H method.

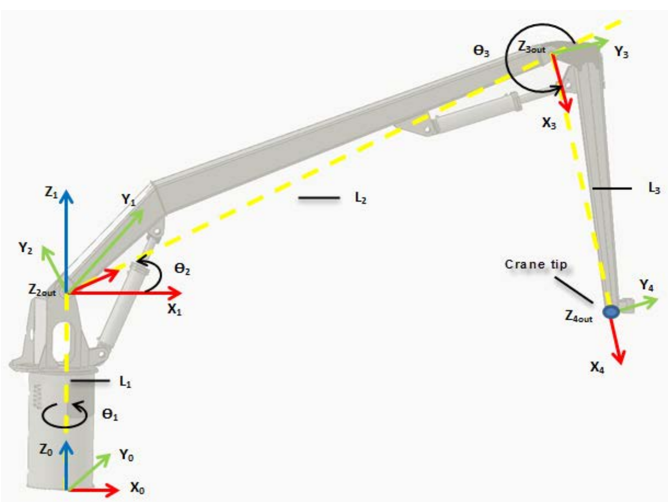

Fig. 2. Knuckle boom crane configuration and coordinate system setup.

Denavit-Hartenberg (D-H) method is a classic way to solve kinematic chains. Similarly, the crane can be regarded as a kinematic chain with three joints and four links. It can then be solved by defining its corresponding standard D-H parameters and variables. The forward transformation matrix from the crane foundation to the crane tip can be derived (1).

$$
{}^0_4T = \begin{bmatrix} c\theta_1 c(\theta_2+\theta_3) & -c\theta_1 s(\theta_2+\theta_3) & s\theta_1 & c\theta_1(L_2 c\theta_2 + L_3 c(\theta_2+\theta_3)) \\ s\theta_1 c(\theta_2+\theta_3) & -s\theta_1 s(\theta_2+\theta_3) & -c\theta_1 & s\theta_1(L_2 c\theta_2 + L_3 c(\theta_2+\theta_3)) \\ s(\theta_2+\theta_3) & c(\theta_2+\theta_3) & 0 & L_1 + L_2 s\theta_2 + L_3 s(\theta_2+\theta_3) \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{1}
$$

Where $c\theta_i = \cos(\theta_i)$, $s\theta_i = \sin(\theta_i)$, $c\alpha_i = \cos(\alpha_i)$, $s\alpha_i = \sin(\alpha_i)$ (i = 1,2,3).

From the forward transformation matrix (1), we get the crane tip Cartesian positions as a function of joint angles (2) in world coordinate system:

$$
\begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{bmatrix} c\theta_1(L_2 c\theta_2 + L_3 c(\theta_2+\theta_3)) \\ s\theta_1(L_2 c\theta_2 + L_3 c(\theta_2+\theta_3)) \\ L_1 + L_2 s\theta_2 + L_3 s(\theta_2+\theta_3) \end{bmatrix} \tag{2}
$$

The Jacobian is a multidimensional form of the derivative. In the field of robotics, the Jacobian is used to relate joint angular velocities to Cartesian velocities of the arm tip. For example,

$$
{}^0v = {}^0J(\theta)\dot{\theta} \tag{3}
$$

Where $\dot{\theta}$ is the vector of joint angular velocities and ${}^0v$ is the vector of arm tip Cartesian velocities. The leading superscript 0 indicates that the calculated Cartesian velocities are in coordinate system 0 which is usually the global coordinate system.

By computing the derivations of the position vector of the crane tip (2), the Jacobian matrix relating joint angular velocities and crane tip Cartesian velocities can be written as (4):

$$
{}^0J(\theta) = \begin{bmatrix} -s\theta_1(L_2c\theta_2 + L_3c(\theta_2+\theta_3)) & -c\theta_1(L_2s\theta_2 + L_3s(\theta_2+\theta_3)) & -L_3c\theta_1s(\theta_2+\theta_3) \\ c\theta_1(L_2c\theta_2 + L_3c(\theta_2+\theta_3)) & -s\theta_1(L_2s\theta_2 + L_3s(\theta_2+\theta_3)) & -L_3s\theta_1s(\theta_2+\theta_3) \\ 0 & L_2c\theta_2 + L_3c(\theta_2+\theta_3) & L_3c(\theta_2+\theta_3) \end{bmatrix} \tag{4}
$$

Hence given the crane tip velocity, the joint angular velocities can be calculated with the following equation:

$$
\dot{\theta} = {}^0J(\theta)^{-1}\,{}^0v \tag{5}
$$

Based on this equation, heave compensation and anti-sway control algorithms can be implemented. In brief, to compensate the ship's heave motion, the crane tip will always try to maintain at a still position, while in order to damp out the load sway, the crane tip will always try to follow the load movements so as to eliminate the sway energy.

B. Heave compensation

Since the crane end tip movements can be controlled in every direction, to compensate the vertical heave motion of the wave, i.e. to keep the crane tip in a steady position, the needed angular velocity for each joint can be calculated with the following equation:

$$\dot{\theta} = {}^{0}J(\theta)^{-1} \begin{bmatrix} 0 \\ 0 \\ -V_{heave} \end{bmatrix} \tag{6}$$

In order to avoid the singularity problem in calculating the inverse matrix, the following equation from the Damped Least Squares (DLS) method, also known as the Levenberg–Marquardt algorithm (LMA), was applied to find the inverse of the Jacobian (7):

$${}^{0}J(\theta)^{-1} = {}^{0}J(\theta)^{T} ({}^{0}J(\theta)^{T}\, {}^{0}J(\theta) + \lambda^{2} I)^{-1} \tag{7}$$

Where ${}^{0}J(\theta)^{T}$ the transpose of the Jacobian matrix, λ is is the damping constant, and I is the unit matrix.

For simulation, the heave motion from the ship is assumed to be a simple sine wave function (8). In real applications, a more complex wave function can be applied and obtained by sensors, e.g. Motion Reference Unit (MRU).

$$V_{heave} = A \sin\left(\frac{2\pi}{T} t\right) \tag{8}$$

Where A is the wave amplitude, T is the wave period.

C. Anti-sway control

The principle for the anti-sway control algorithm is to dissipate the load sway energy by steering the movements of the crane tip and varying the wire length according to the movements of the load. In effect, the crane tip is always moving towards the load, and the wire length is reduced when the load moving away from the resting point and extended when it moves towards the resting point. Both of the two parts aim to reduce the sway kinetic and potential energy. According to preliminary simulation results in Matlab, the effect by adjusting wire length accounts for 10%-15% of the total reduced time [15]. Since a winch system is not considered in the paper, the algorithm for load anti-sway control is only implemented the first part, i.e. moving the crane tip.

In one DOF (Fig. 3) the total energy E of the system is given by the Lagrangian formula:

$$E = K + V = \frac{1}{2} m\dot{x}^{2} + \frac{1}{2} m\dot{z}^{2} + mgz \tag{9}$$

Where K is the kinetic energy, V is the potential energy, $\dot{x}$ and $\dot{z}$ are the load velocity in x and z directions.

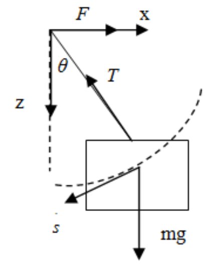

Fig. 3 Load pendulum in x-z plane.

F is the force applied to the crane tip, T is the wire tension, mg is the gravity force of the load, θ is the load sway angle and $\dot{s}$ is the instant velocity of the load.

The energy dissipation is realized through the work done by the force at the crane tip (10):

$$W = \int F \cos\theta \cdot s dt \tag{10}$$

The system's energy will be reduced when W is negative, e.g. at the position in Fig. 3 when load velocity $\dot{s}$ is in negative x direction, the force applied at the crane tip F should be in the positive x direction which means the crane tip moves towards the load.

Instead of feeding acceleration to the crane end tip, we send velocity signals of the crane tip directly to the anti-sway controller. Under this assumption, we assume the crane end tip can always move at a constant maximum velocity towards the load regardless its responding time. The angular velocity of each joint can be calculated with (6).

The anti-sway control algorithm includes two DOFs in both ship longitude and transverse directions. For one direction, the velocity of the crane end tip is given by (11):

$$v_{tip} = \begin{cases} -v_{\max} & x_{tip} > x_{load} \\ v_{\max} & x_{tip} > x_{load} \end{cases} \tag{11}$$

Where $v_{\max}$ is the predefined constant velocity applied to the crane tip, x_{tip} and x_{load} are crane tip position and load position in x direction (ship transverse direction).

IV. SYSTEM MODELING AND SIMULATION

To test the proposed control algorithm and operation functions, a model was developed including the hydraulic system of the crane and 3D model for representing the mechanical properties and visualization. The modelling tool and simulation environment is 20-sim, which is a modeling and simulation program for multi-domain system simulation. The modeling process is not presented in this paper. Modeling of crane hydraulic system using BG method refers to [16]. The following Fig. 7 shows the block model in 20-sim. The model is grouped in several submodels: joystick signals, control box, crane hydraulic system, wave generator, and crane 3D model. A common 3DOF gaming joystick is applied in this case for sending the speed signals and trigging the control functions.

The feedback signals of positions and angles are obtained through sensors included in 20-sim. In real applications, to obtain these data is more complicated in terms of where the sensors should be attached, etc., however, this is not what this paper focuses on.

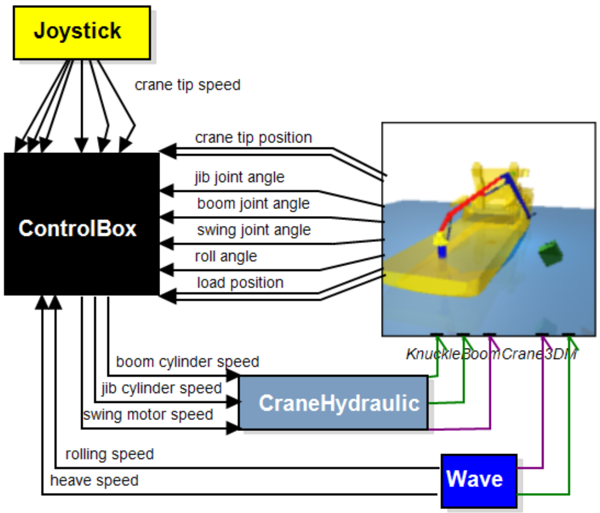

Fig. 7 Offshore hydraulic crane model.

Through simulation, the load position and crane tip position can be plotted. Figure 8 shows the results without hydraulic system included but only the crane with its mechanical properties kept.

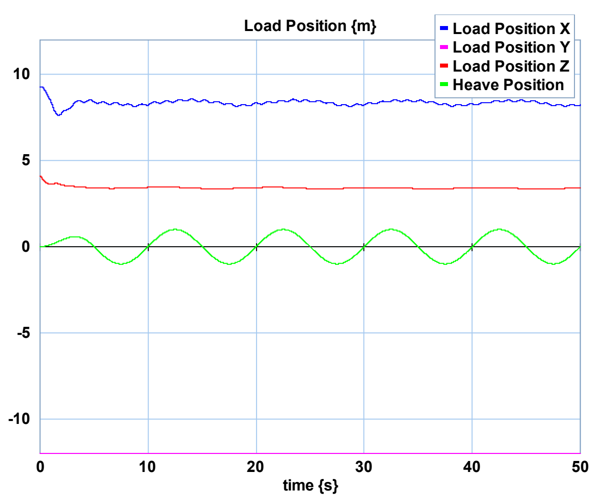

Fig. 8 Plotting of load position under heave compensation and anti-sway - without hydraulic model.

With an initial angle of 45 degrees in X direction, the load sway was damped out in about 3s. The heave (green line) and rolling (black line) applitude is set at 1m and 10 degrees. The load position in z direction (red line) is compensated within 0.1m while in x direction (blue line) at about 0.4m. Considering the dimensions of the crane, the work scope and the heavy load lifting in offshore operations, this deviation is neglectable.

When including the hydraulic model, which is not designed for the compensation functions, due to the lag of its responding time the results are not as good. But the control algorithm and functions are still valid. The hydraulic system need to be adjusted to adapt to the compensation functions. In Fig. 9 and 10, the load position is shown simulating the model with hydraulic system included.

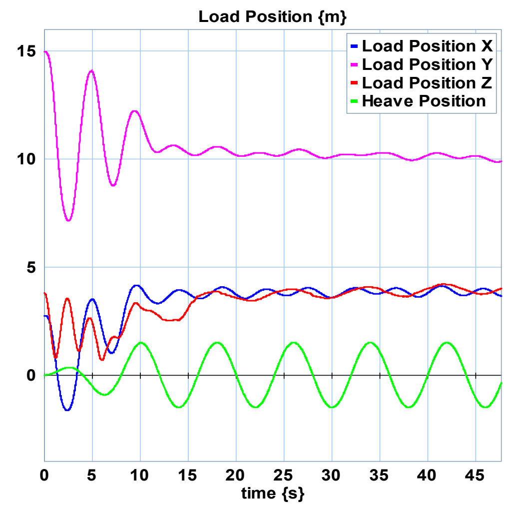

Fig. 9 Plotting of load position under heave compensation and anti-sway - with hydraulic model.

With initial angles of 45 degrees in both X and Y directions (blue and pink lines), the load sway was damped out in about 9s. Load displacement in Z direction (red line) was suppressed within a range of 0.5m with a wave height of 3m and wave period of 8s (green line).

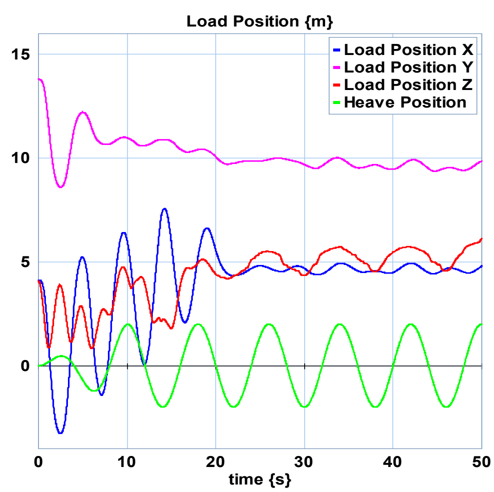

Fig.10 Plotting of load position under heave compensation and anti-sway - with hydraulic model.

With initial angles of 60 degrees in both X, Y directions (blue and pink line), the load sway was damped out in about 19s. Load displacement devation in Z direction (red line) was suppressed within a range of 1.2m with wave height of 4m and wave period of 8s (green line).

The crane can damp out the sway in a few seconds, while heave compensation using the crane cylinders is unable to compensate all the vertical heave motion, especially under large wave height and short wave period conditions. The heave compensation using crane cylinders is limited by the crane's dimensions. As can be seen from Fig. 10 above, heave compensation for waves with 4m in height is difficult for this crane with a cylinder stoke of only 1m and the main boom length of 7m.

V. CONCLUSION

In previous sections, a flexible control algorithm for offshore crane operations was presented and tested in a simulation environment. A simulation model for the offshore hydraulic crane was designed and performed. The simulation results validated the control algorithm as well as the heave compensation and anti-sway functions.

Regardless of fuel consumption, including a winch system will result in better heave compensation, especially in sub-sea lifting operations. In addition with optimized hydraulic system, anti-sway time can be further reduced as well. As future work this part will improve the effectiveness of the whole system. Tests on a physical offshore hydraulic crane system will be carried out. As field testing in offshore industry is confined in many ways, a hardware-in-the-loop simulator will be developed for offshore crane system.

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