# Integrated Multi-domain System Modelling and Simulation for Offshore Crane Operations

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**ABSTRACT**
Advanced offshore machinery, such as an offshore crane, usually involves several energy domains. Modelling and simulation of multi-domain systems is challenging due to not only the complexity in modelling of the related sub-systems, but also the interfacing of these sub-models in one integrated model and the performance of simulating, especially when real-time simulation is required. This paper introduces a modelling approach for offshore crane operations based on the Bond Graph method. Specifically, the integrated model includes mechanical properties, hydraulic actuators and control algorithms. For the purpose of testing, particularly of advanced control algorithms, it is necessary and crucial to include the response of physical systems. In this paper, a flexible control algorithm for offshore crane operation, including functions of heave compensation and load anti-sway is implemented.

*Keywords: Multi-domain system modelling and simulation; Bond Graph; Offshore hydraulic crane; Heave compensation; Anti-sway.*

## 1 Introduction

With the development of technology, human activities have been reaching further and deeper from coast to offshore and subsea. Today, offshore industry has become one of the most technologically innovative and demanding sectors in the world. In nature, the geography and weather conditions pose harsh challenges for resource exploration and harvesting. Meanwhile, human society has been exerting stricter rules and regulations regarding safety and environment protection. Intensive tests and experiments must be carried out before realization and execution of a new solution. But testing with real full-scale maritime machinery systems is not necessarily the most efficient or cheapest way. Particularly, sea testing of offshore machinery operations is rather costly and time consuming, and often prevented by adverse weather conditions. In this context, modelling and simulation in a virtual environment becomes a vital phase during product and system design, manufacturing, verification, optimization, and stability analysis. With 3D animation, simulations in a virtual environment can also be used for operator training. To date, there is no well-developed simulator for offshore machinery operations. A few simulators exist for dynamic positioning, navigation bridge and crane operation that are designed for ship crew training; however, these simulators do not involve models of physical systems, but only graphical visualization, Offshore Simulation Center AS (2014).

Offshore cranes are the main deck machinery for transportation, lifting, and handling operations onboard of many kinds of vessels and offshore platforms. Due to the inherited issues such as heavy lifting, positioning accuracy, load sway, and security, offshore crane operation is always a demanding task when considering both work efficiency and safety. Nowadays, it is still common to control a crane joint by joint. As well, training operators to adapt to various types of cranes and input devices takes a lot of time and cost. The operation is rather inefficient and, to a large extent, relies on the skills of the crane operators. Many accidents have occurred. Unlike cranes used on land with fixed working platforms, offshore cranes are greatly impacted by the sea movements. The pendulum payload causes many challenges for the operation, such as load sway, dynamic forces to the crane structure, wire fatigues, and slamming forces from the waves. A lot of time and cost are wasted just on waiting for a better weather condition. To tackle these problems and broaden the weather window for operation, intelligent control algorithms were studied for heave compensation and anti-sway.

One of the most widely applied approaches for heave compensation in offshore and subsea operations is using the so-called Active Heave Compensated (AHC) winch. To keep the pendulum load still, the lifting wire is constantly paid in and out, referenced to the heave motion, Talberg (2012). Load sway is more complex to deal with, as the pendulum motion of the load is in three dimensions and hard to estimate. One relatively effective approach is to use a tagline attached to the lifting wire or load to reduce the load sway, Park (2007); Ku N. (2013). Our research group proposed a flexible control algorithm combined with compensation functions for offshore crane operations, using both crane and winch, Chu (2014). The control algorithm and compensation functions are explained further in section II and implemented in the case study model in section III. To test the algorithms and functions, modelling and simulation of the crane operation in a virtual environment is indispensable.

In recent years, more and more interest in multi-domain system modelling and simulation, especially simulation of control of dynamic systems, has arisen. Object-oriented modelling (OOM) is one popular approach for complex engineering system modelling. The newest standardized modelling language, called Modelica, a unified object-oriented language, is increasingly well-developed and accepted as a neutral exchange format between proprietary modelling tools. Another approach starting from the conception that interaction between sub-systems based on energy exchange has been in use since Bond Graph (BG) was devised by Professor H. Paynter at Massachusetts Institute of Technology in the United States as early as 1959, Paynter (1961). The two approaches have much in common, e.g., both allow for domain independent, non-causal hierarchical modelling. In this sense, BG modelling can also be viewed as a special form of OOM, Borutzky (1999).

Offshore cranes are often hydraulically actuated with much capacity redundancy to ensure the operation stability and safety during offshore activities. It is rather delicate to model hydraulic systems because of the nonlinear aspects of fluid dynamics. Moreover, because of the inherited nature of an offshore crane system in heavy lifting and high stiffness, the simulation model for the control algorithm and compensation function must include models of physical properties and run in real time. Related to modelling of engineering systems that involve hydraulic domain, some literatures can be identified using both of the two approaches. Liu (2012) investigated the modelling of the mechanical-electrical-hydraulic coupling system of a special vehicle in weaponry. Modelling and co-simulation was realized in Pro/E, EASY5, MATLAB, and ADAMS. La Hera (2012) studied control of the electro-hydraulic actuators used in forestry cranes. A dSPACE prototyping hardware is used for the control of the crane as well as the implementation of algorithms. BG method uses generalised variables to write the state equations describing the energy interaction between the sub-systems from multiple domains. A few attempts based on BG modelling of a multi-domain system were identified. Aarseth (2014) presented a simulator for a launching and recovery system using 20sim for modelling and Bachmann controller for real time simulation and communication. Several modelling tools and simulation environments were involved in these studies. These software programs provide libraries for various systems. The differences are how well-developed the libraries are, and whether the models can be integrated and interfaced with other simulation tools when more than one are used, and hardware, for example input devices and controllers. In the research work presented here, the simulation model was developed based on the BG method and implemented and simulated using the software tool 20sim, Controllab (2014). Instead of the provided hydraulic library, which lacks certain critical components, the hydraulic models for the offshore crane system were created using basic bond graphs. The 3D model was built in Solidworks and imported into the 20sim 3D Mechanic Editor (3DM), which is a built-in toolbox in 20sim. The model also included a 3D animation simulator, which gives 3D visualization of the operational behaviour of the crane.

The rest of the paper is organized as follows. BG method and the simulation tool 20sim are introduced in Section 2. Mechanical and hydraulic modelling of the crane and winch and the control functions for heave compensation and load anti-sway are described in Section 3. Then the simulation results are presented and discussed in Section 4. Section 5 provides the conclusion and describes future work.

## 2 Methods and tools

The research motivation was to develop a fully functional simulator for offshore crane operations. It was oriented to simulations of control algorithms and functions, but with physical models included, specifically, the hydraulic system and the mechanical dynamics of the crane and winch. Furthermore, it included the interaction at the ship and the hydro-dynamics between ship and wave. The simulator did not only simulate the behavior of the lifting operations, but also the performance of the physical components and system. The simulation results provided beneficial data for product and system design, new control algorithm and components testing, and operator training with 3D animation running in real time.

The overall structure of the offshore crane simulator is depicted as in Figure 1. The crane is controlled via a universal haptic device with force feedback. By universal, here we mean the same joystick applies to different types of cranes with various configurations and

Degrees of Freedom (DOFs). The control box contains the control algorithm, solving the kinematics and heave compensation and anti-sway functions. The computed signals are then sent to the hydraulic controllers to actuate the crane and winch. The hydraulic sub-models simulate the dynamics of the hydraulic systems of the crane and winch. The 3D model includes the mechanical dynamics of the crane, winch, wire, and payload. In addition, a hydrodynamic sub-model representing the interaction between the ship and wave, and thermal sub-models for the cooling system and energy consumption may be included.

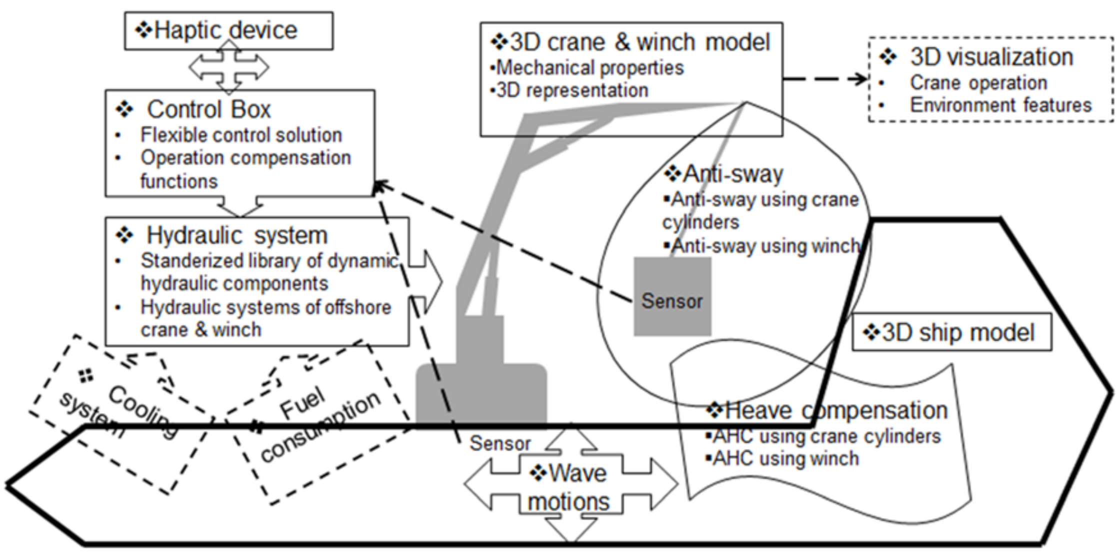



Fig. 1: Offshore hydraulic crane modelling structure

To represent a physical system mathematically, the model must be close to the original system, and simplified, keeping only the necessary components, depending on the focal points of the simulation purposes. BG method is a modelling technique for multi-domain systems based on identifying the energetic structure in the physical system, Karnopp (2012). A system can be decomposed into a few basic physical properties and described by the interrelated idealized elements representing these properties. The energy or power interaction between two elements is called a “power bond”, represented by a half arrow, Fig. 2 (a), (b). A power bond is defined by two variables with generalized names of “effort” and “flow”, of which the product is power. For example, in the mechanical domain, “effort” and “flow” are, respectively, “force” and “velocity” or “torque” and “angular velocity”. In the hydraulic domain, “effort” and “flow” are respectively “pressure” and “volumetric flow rate”. Computational causality is represented by a perpendicular short stroke attached to either end of a power bond, indicating where the “flow” variable is computed, given the “effort” variable, which is determined by the other end of the bond. Two kinds of junctions are used in the BG method, Fig. 2 (c), (d). Around a 0-junction the bonds share equal “effort” and the sum of “flow” is zero. A 1-junction is the opposite, i.e., the bonds share equal “flow” and the sum of “effort” is zero.

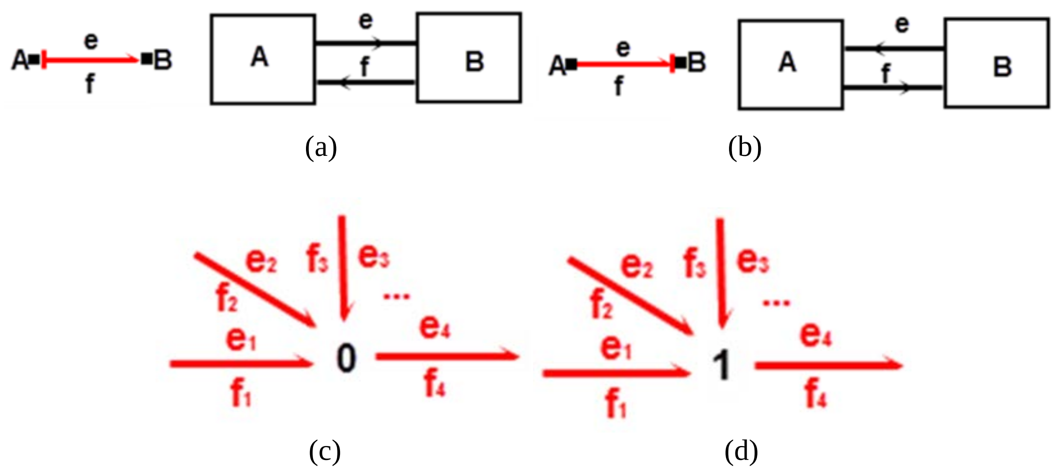



Fig. 2: Bond Graph representations and junctions – (a). “effort” is determined in A and flow is determined in B; (b). “flow” is determined in A and effort is determined in B; (c). $e_1 = e_2 = e_3 = e_4 =\dots$ and $f_1 + f_2 + f_3 + \dots = f_4$; (d). $f_1 = f_2 = f_3 = f_4 =\dots$ and $e_1 + e_2 + e_3 + \dots = e_4$.

For implementation and simulation of the BG models, a software tool called 20sim was adopted. The commercial program now includes standardized libraries and packages for modelling of multi-domain dynamic systems using not only bond graphs but also block diagrams, physical components, and equations.

# 3 Modelling and simulation of offshore crane operation

Compared to robotic arms, offshore cranes usually have fewer joints, i.e., fewer DOFs, but are much larger in size and slower in movement. Figure 3 shows a common type of offshore hydraulic crane called a knuckle boom crane, which consists of three revolute joints actuated by hydraulic motor and cylinders. The assignment of reference frames and notations followed the Denavit-Hartenberg (D-H) method. The global reference frame 0 was attached to the base of the crane.

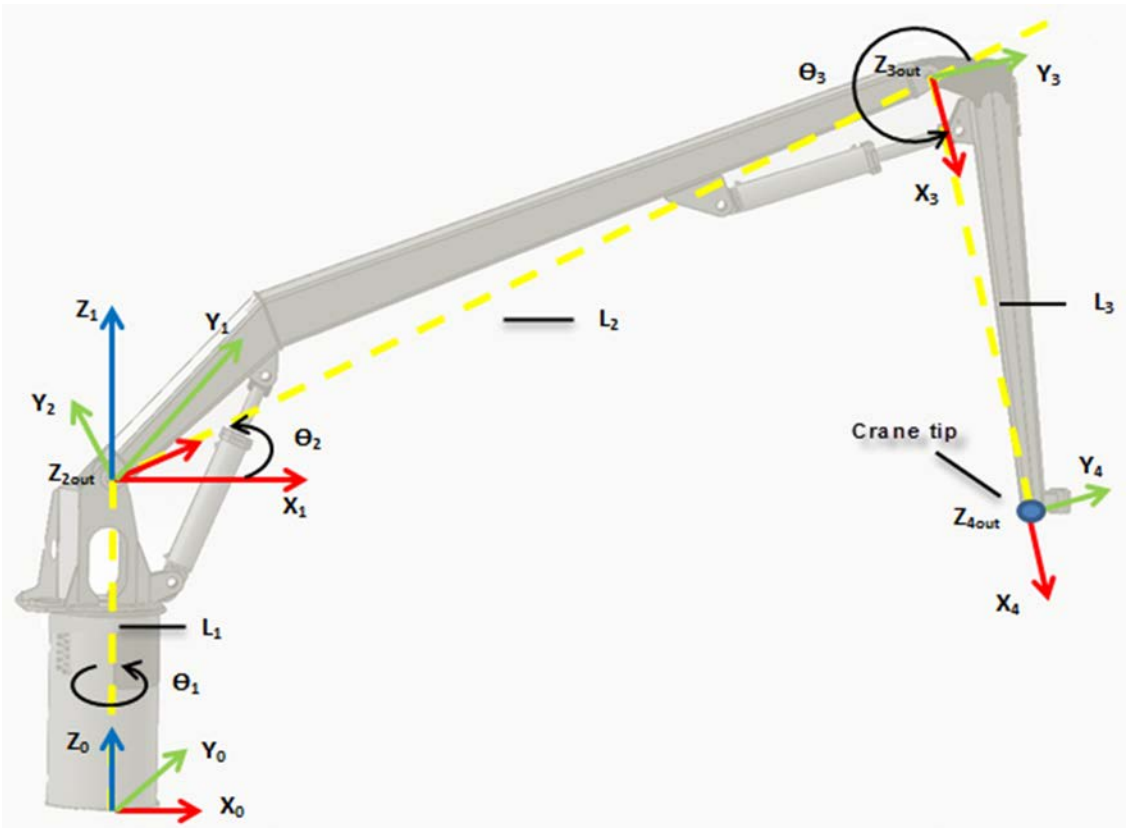



Fig. 3: Knuckle boom crane and D-H method frame assignment

## 3.1 Control algorithm and compensation functions

Inspired by the control of industrial robotic arms, a flexible control algorithm for offshore crane operations was proposed by our research group at Mechatronics Lab, Aalesund University College. Rather than moving the crane joint by joint, the control of the crane was realized by moving its end tip and through solving its kinematics to establish the relationships to the joint actuators. The intention was to control various types of cranes using one universal input device. This control algorithm gave a more intuitive feeling to the crane operators in positioning the load. Dependent on the DOF and configuration of the crane, the algorithm can be easily adapted through changing the kinematic solutions.

The D-H method is popularly applied in solving the kinematic chains, Siegwart (2004). The velocity Jacobian matrix from the crane tip to the reference frame 0 is presented, eq. (1).

$$ {}^{0}J(\theta)=\begin{bmatrix} -s\theta_1\left(L_2c\theta_2+L_3c(\theta_2+\theta_3)\right) & -c\theta_1\left(L_2s\theta_2+L_3s(\theta_2+\theta_3)\right) & -L_3c\theta_1s(\theta_2+\theta_3) \\ c\theta_1\left(L_2c\theta_2+L_3c(\theta_2+\theta_3)\right) & -s\theta_1\left(L_2s\theta_2+L_3s(\theta_2+\theta_3)\right) & -L_3s\theta_1s(\theta_2+\theta_3) \\ 0 & L_2c\theta_2+L_3c(\theta_2+\theta_3) & L_3c(\theta_2+\theta_3) \end{bmatrix} \tag{1} $$

Given the end tip velocity, the angular velocities of the joints are calculated by eq. (2), where $\dot{\Theta}$ is the vector of joint angular velocities and $v$ is the vector of the arm tip Cartesian velocities. $J(\Theta)^{-1}$ is the inverse of the Jacobian matrix from the crane tip velocity to the joint angular velocity:

$$\dot{\Theta} = J(\Theta)^{-1} v \tag{2}$$

The cylinder velocities can be computed geometrically as functions of the joint angular velocities, eq. (3) and eq. (4), where $v_1$, $v_2$ are the velocities of the cylinders, and $\dot{\theta}_2$, $\dot{\theta}_3$ are the angular velocities of the second and third joints:

$$v_1 = f(\dot{\theta}_2) \tag{3}$$

$$v_2 = f(\dot{\theta}_3) \tag{4}$$

The proposed control algorithm increased the efficiency of crane operation, giving more flexibility to control different types of cranes. Compensation functions were proposed utilizing the algorithm. The following two alternatives were presented for AHC using the crane or the winch.

For heave compensation and anti-sway functions using the crane, the idea is to move the crane tip in the opposite direction of the heave and following the movements of the load. Using crane cylinders for heave compensation, the required joint velocities are given by eq. (2), where $v_{heave}$ is the wave heave velocity in a vertical direction.

$$\dot{\Theta} = J(\Theta)^{-1} \begin{bmatrix} 0 & 0 & -v_{heave} \end{bmatrix}^T \tag{5}$$

Using winch for heave compensation, the required wire speed is equal to the heave speed, but in the opposite direction. The winch drum speed is calculated by eq. (6), where $d$ is the winch drum diameter.

$$\omega = -v_{heave} / d \tag{6}$$

One of the common challenges of AHC solutions in industry is the performance of the system in responding to the signal variation received from sensors. Heave compensation using the crane cylinders is limited by the dimensions of the crane links, and the movements require coupled movements of the two or more cylinders. The advantage in using a winch system for heave compensation is that it is relatively easy to realize since there is only one DOF in the vertical direction. Other problems include wear of the lifting wire due to repeatedly launching and recovering the payload and significant an energy consumption in driving the system.

The load sway problem is even harder to deal with as the motion of sway is in three dimensions and difficult to predict. The proposed algorithm for anti-way in crane operations is developed based on the energy dissipating principle, i.e., to dissipate the load sway energy by steering the movements of the crane tip and varying the wire length according to the instantaneous position and movements of the load. In effect, the crane tip is always moving towards the load, and the wire length is reduced when the load moves 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, the effect of adjusting wire length accounts for 10%–15% of the total reduced time, Van Albada (2013).

To implement the crane movement in one dimension, the required crane tip velocity in the most simplified form is given by eq. (7), where $v_{\max}$ is a predefined constant velocity applied to the crane tip, $x_{tip}$, and $x_{load}$ are the positions of the crane tip and load in the ship transverse direction. Joint velocities can then be calculated using eq. (2), as well as the cylinder velocities.

$$v_{tip} = \begin{cases} -v_{\max} & x_{tip} > x_{load} \\ v_{\max} & x_{tip} > x_{load} \end{cases} \tag{7}$$

Both the heave compensation and load anti-sway functions were implemented in simplified forms. In reality, the situations are expected to be more complicated, e.g., the algorithms should consider time delay, and signal sampling and processing.

## 3.2 Mechanical modelling and animation

A mechanical model was built using 3DM, which is one of the built-in toolboxes in 20sim for 3D mechanical modelling. In 3DM, a multi-body mechanism is interconnected using constrained joints. The representations of the rigid bodies can be simple blocks like sphere, block, cylinder, or imported CAD files. A rigid body is defined by its inertia and mass, which can be obtained from the CAD model and must be assigned manually. The knuckle boom crane was created in Solidworks and imported into 3DM for assembly via proper joint connections, Fig. 4. An alternative is to import an assembly model directly from Solidworks into 3DM; however, different rules must apply when mating the parts in Solidworks. Specifically for the crane case, the actuators include cylinders that form closed kinematic loops. It is much easier to set up the reference frames following the D-H method for solving the kinematics in developing the control algorithm. To simulate the ship movements, a translation joint and a rotational joint are added, representing heave and rolling movements. For the wire, a translation joint is added in addition to a sphere joint connecting the load and crane tip. In other words, the wire was assumed to be always tensioned.

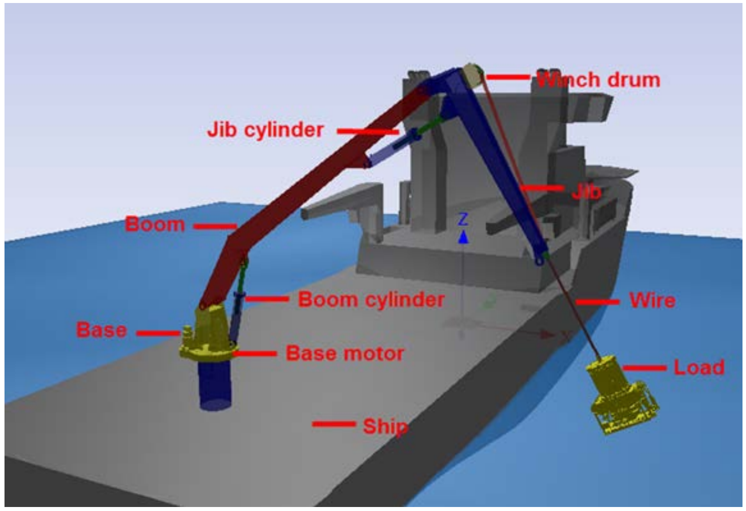



Fig. 4: 3DM crane model with ship, wire and load

From the 3DM model, a 20sim model and animation scene was generated with the 3DM model as a sub-model, Fig. 5. The input device was a joystick for sending velocities of the crane tip and winch. The velocity mapping to the joystick and function buttons was depicted as in the figure. The control algorithm and compensation functions were coded in the control sub-model, with inputs from the joysticks and outputs to the crane and winch actuators. The actuators were represented by MSe-elements in PID-controlled loops. MSe-element in BG method is Modulated Effort Source which, outputs "effort" according to "flow" signal, i.e., force or torque according to velocity. Feedback signals, including the joint angles and positions, were read from the 3DM sub-model via sensors. The "wave" block contained the wave signals represented simply by their velocities as sine functions.

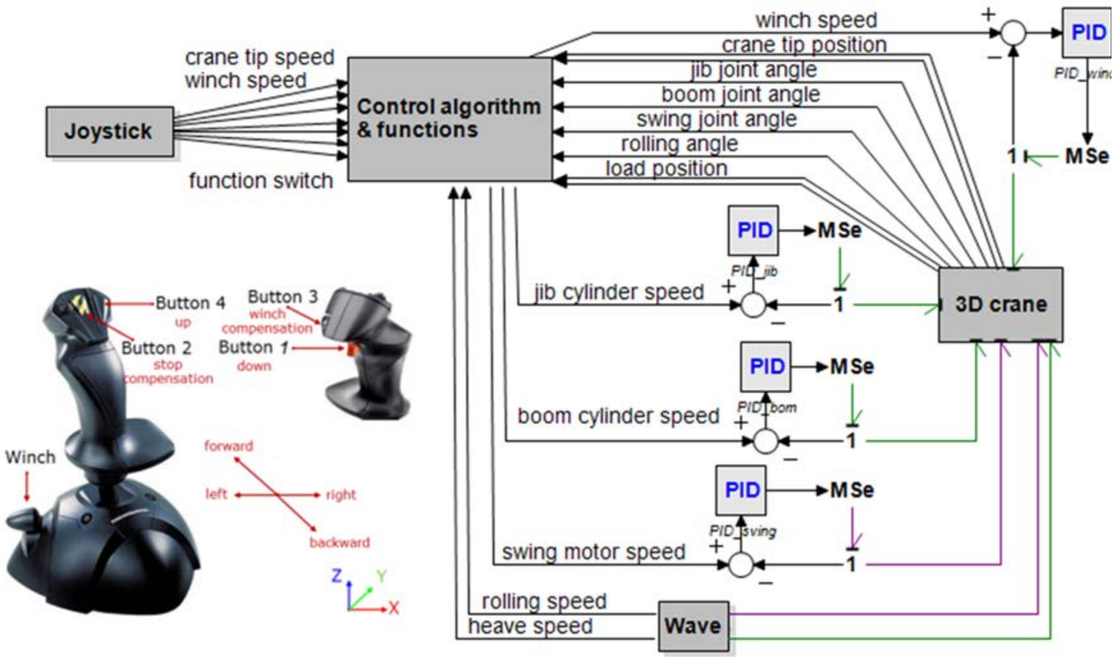



Fig. 5: Crane control model – with compensation functions – without hydraulics

## 3.3 Hydraulic system and BG modelling

The model shown in Figure 5 was developed to validate the proposed control algorithm and compensation functions. The hypothesis behind using MSe-elements to replace the hydraulic systems was that the capacity and performance was always sufficient. Because of the 3D model with physical properties, PID-controllers were adopted to regulate errors. As stated previously, the hydraulic systems were not negligible to achieve a realistic simulation. First, a model with hydraulic sub-models was developed, Fig. 6. The MSe-elements were replaced by the hydraulic models of the crane and winch. The global parameters, for example the fluid density, viscosity and bulk modulus, are grouped in one sub-module “Globals”. Each joint was connected to one hydraulic actuator signaled directly via the joystick. The control algorithm here only included the kinematic computation without heave compensation and anti-sway functions.

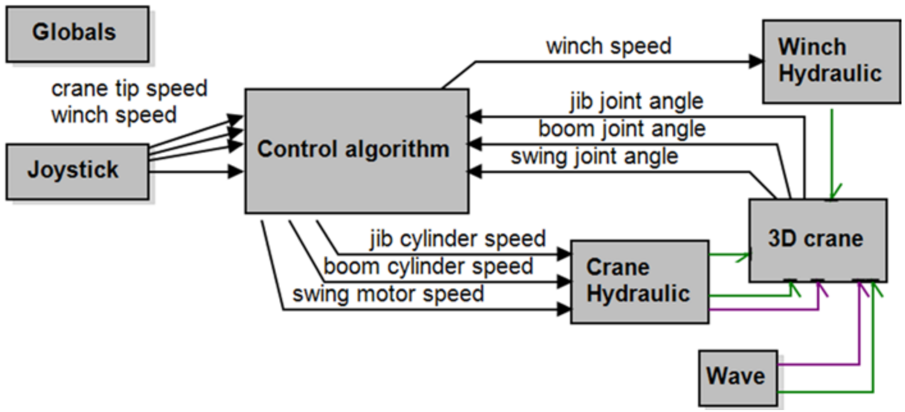
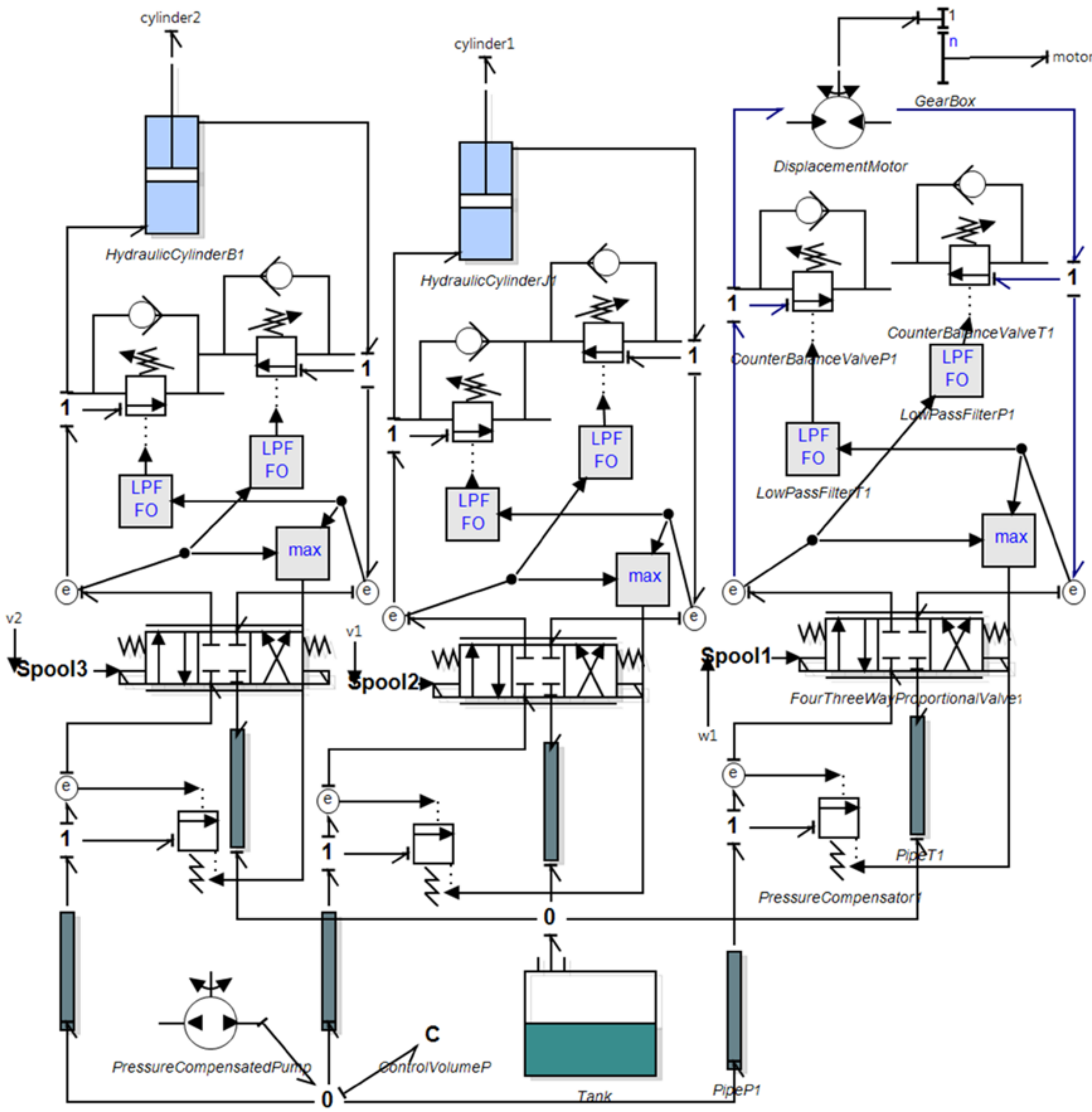



Fig. 6: Crane control model – with hydraulics – without compensation

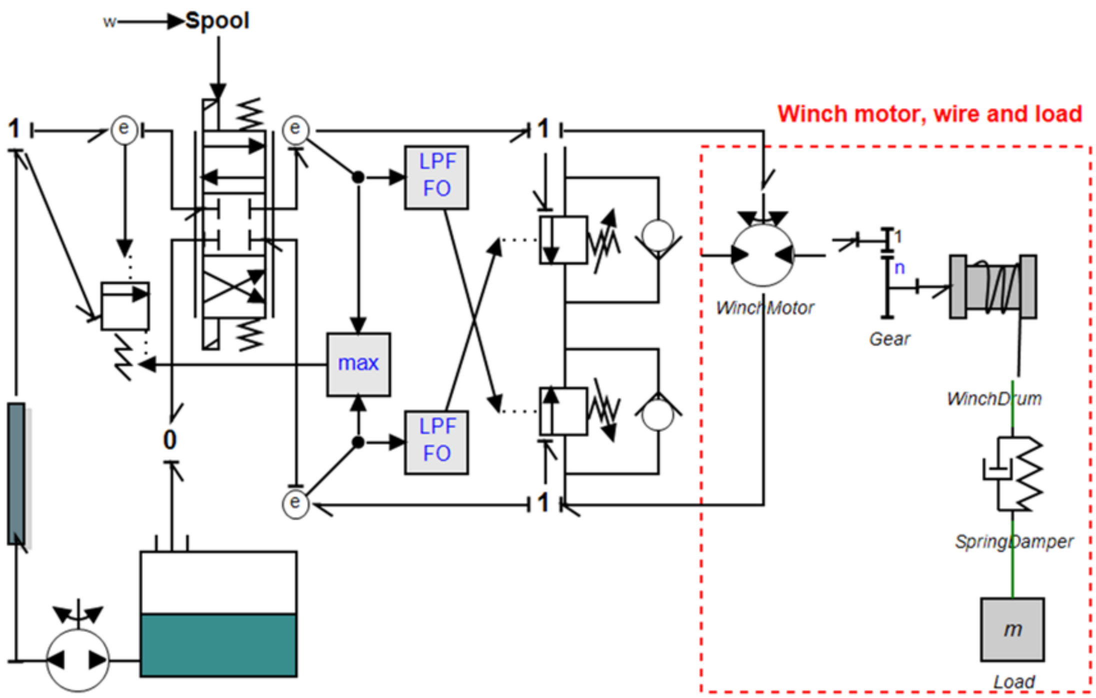
To illustrate the simplified hydraulic schematic of the crane hydraulic system, the BG models of the hydraulic components were represented using block icon diagrams, Fig. 7. The development of the BG models of the hydraulic components was documented in Chu (2014).



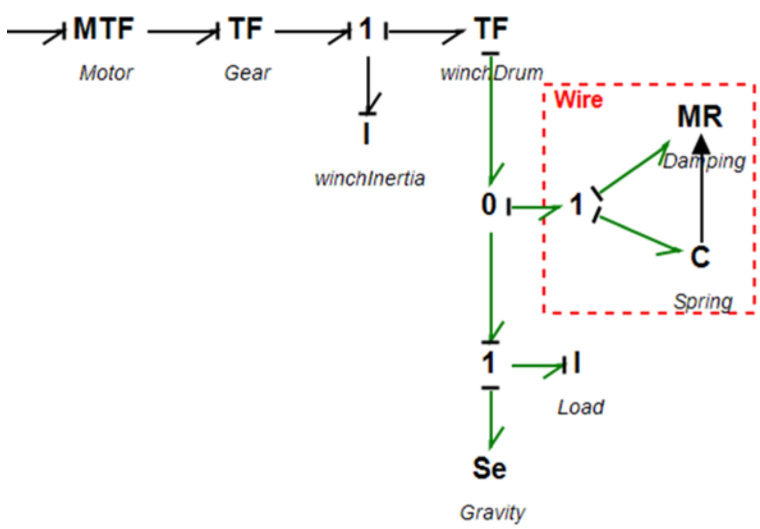

Three signal input ports "w1, v1 and v2" and three power output ports "motor, cylinder1, and cylinder2" were defined. The flow distribution was adjusted via the spool signals to the directional valves. Low pass signal filters were inserted to adjust the signal frequency to obtain a "smoother" signal flow. The effort sensors were inserted to read the pressures at certain points in the hydraulic system. The "max" block gave the maximum value of the input signals as the reference pressure for the pressure compensator. The C-element named "ControlVolume" was inserted to obtain the correct causality in simulation. In formulating the dynamic equations that describe the system, causality defines, for each modelling element, which variable (effort or flow) is dependent and which is independent. The C-element was defined as a small volume which contains a negligible capacity, inertia and compressibility.

Fig. 7: Knuckle boom crane hydraulic sub-model

Similarly, a block diagram representation of the winch hydraulic system is presented, Fig. 8 (a). The development of the BG model for the following components is described, including winch motor, gear, winch drum, wire, and load, Fig. 8 (b).




(a)




(b)

Fig. 8: (a) Winch hydraulic sub-model. (b) BG of winch motor, wire and load

The hydraulic motor transforms hydraulic energy to mechanical energy represented by TF-element. The physical laws, eq. (8), represented using BG variables, is given in eq. (9), where $\forall$ is the displacement of the motor, $e_1$ is hydraulic pressure at motor, $e_2$ is output torque of motor, $f_1$ is flow through motor, and $f_2$ is motor rotational speed.

$$\dot{V} = \frac{\forall}{2\pi}\omega \qquad T = \frac{\forall}{2\pi}P \tag{8}$$

$$f_1 = \frac{\forall}{2\pi} f_2 \qquad e_2 = \frac{\forall}{2\pi} e_1 \tag{9}$$

Gear transmission changes the output speed and torque of the motor and can be represented by TF-element in BG. The transmission equation, eq. (10), is given in eq. (11), where n is gear ratio, $e_1$ is output torque of the motor, $e_2$ is the torque to the winch shaft (drum), $f_1$ is the motor rotational speed and $f_2$ is the shaft (drum) speed.

$$T_1 = \frac{1}{n}T_2 \qquad \omega_2 = \frac{1}{n}\omega_1 \tag{10}$$

$$e_1 = \frac{1}{n}e_2 \qquad f_2 = \frac{1}{n} f_1 \tag{11}$$

The inertia of motor, gear, and shaft (drum) are represented in one I-element. The momentum *L* is the integration of torque, eq. (12), and interpreted as in eq. (13), where *I* is the sum of the moment of inertia, $f$ is the angular velocity of the winch shaft (drum), and *e* is the shaft torque.

$$L = I\omega \tag{12}$$

$$f = \frac{1}{I}\int e \tag{13}$$

Winch drum transforms rotational torque from the drum shaft to translational power of the wire, and is represented by TF-element. The following equations apply: eq. (14), eq. (15), where *d* is drum diameter, $e_1$ is output torque of the winch drum (shaft), $e_2$ is the force to the wire, $f_1$ is the winch shaft (drum) rotational speed and $f_2$ is the wire speed.

$$T = \frac{d}{2}F \qquad v = \frac{d}{2}\omega \tag{14}$$

$$e_1 = \frac{d}{2}e_2 \qquad f_2 = \frac{d}{2} f_1 \tag{15}$$

The lifting wire is represented by its elasticity and stiffness. For the spring represented by a C-element in BG, the physical law eq. (16) is described by eq. (17), w here $k = \frac{EA}{L}$ is the stiffness of the wire as a function of the wire length, $E$ is wire elastic modulus and $A$ is wire cross section area.

$$F = k \cdot \Delta x \tag{16}$$

$$e = k \cdot \int f \tag{17}$$

For the damping represented by an MR-element in BG, the physical law eq. (18) is described by eq. (19), where $c = \xi \cdot 2\sqrt{k \cdot m}$ is the damping factor of the wire, $k$ is the wire stiffness, $m$ is the load mass, and $\xi$ is the ratio to critical damping.

$$F = c \cdot v \tag{18}$$

$$e = c \cdot f \tag{19}$$

The inertia of the load represented by an I-element, the momentum $p$ is the integration of the force given by eq. (20) and interpreted as in eq. (21), where $I = m$, $m$ is the load mass.

$$p = mv \tag{20}$$

$$f = \frac{1}{m}\int e \tag{21}$$

The gravity force, eq. (22), is represented by an Se-element, and interpreted as in eq. (23) as an “effort” source.

$$G = mg \tag{22}$$

$$e = mg \tag{23}$$

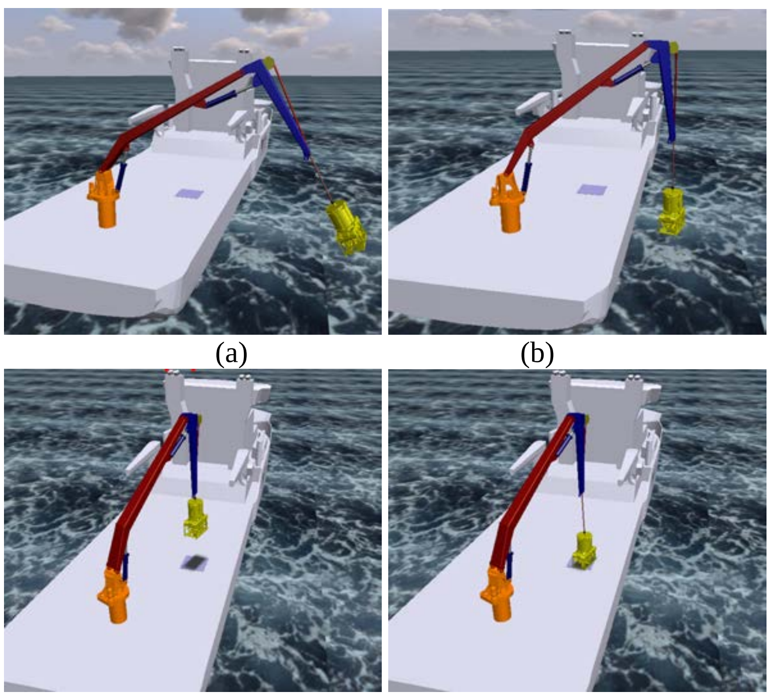
The rest of the sub-models, such as the pump, pipe, accumulator and gear, are not presented here, and similarly to those in the crane cylinder and motor models. The implementation in 20sim may not be exactly the same, but the applied physical laws and principles in formulating the BG equations are the same.

# 4 Simulation results and discussion

To validate the modelling and the effectiveness of the control algorithms and compensation functions, several tests based on the models in the previous chapters were performed. Simulation of heave compensation and load anti-sway control was performed using the model in Figure 5. Simulation of the hydraulic properties of the crane and winch was performed using the model in Figure 6.

## 4.1 Simulation of heave compensation and load anti-sway control

The crane was operated by moving the crane tip instead of each joint, which was much easier for the crane operator to position the load. Two working modes were designed for heave compensation: using the crane cylinders or using the winch. With heave compensation and anti-sway control, the load was maintained in a relatively steady position. The screenshots of the 3D animation scenarios of the operation in recovering an object from overboard position to the deck (a-b-c-d) is shown, Fig. 9.



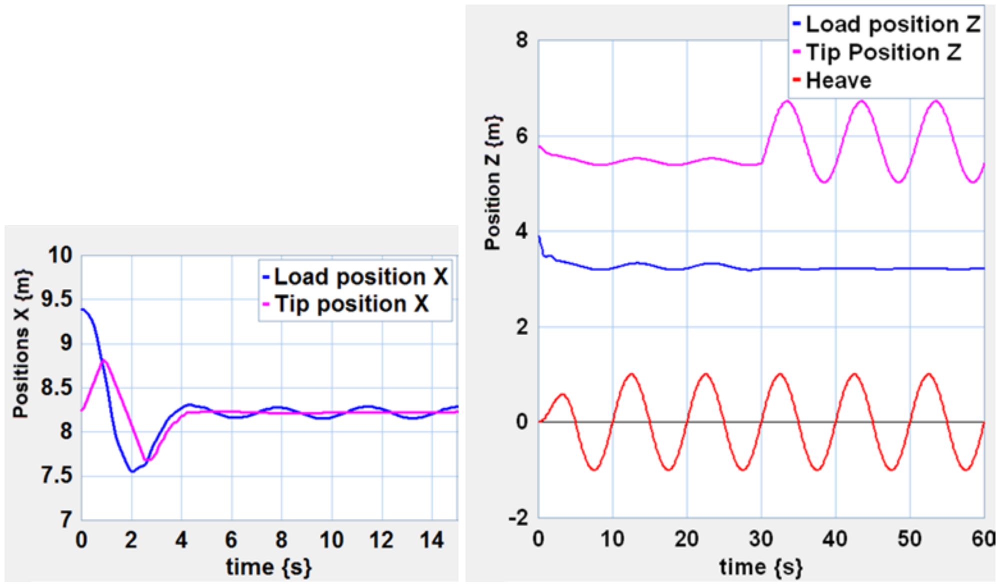
(a) (b)

(c) (d)

The operation started from the load recovering from the sea with a 45 degree load sway in the ship transverse direction (a). Anti-sway control (b) and heave compensation applied before and during maneuvering the load to the ship board (c). Then the load was lifted up via the winch and lowered on the deck (d).

Fig. 9: Knuckle boom crane 3D animation scenario

The 3D simulator provided visual animation to the crane operator. Critical parameters and variables were plotted for analysis. Figure 10 shows separately the crane tip and load position during the operation with anti-sway and heave compensation functions applied.




(a) (b)

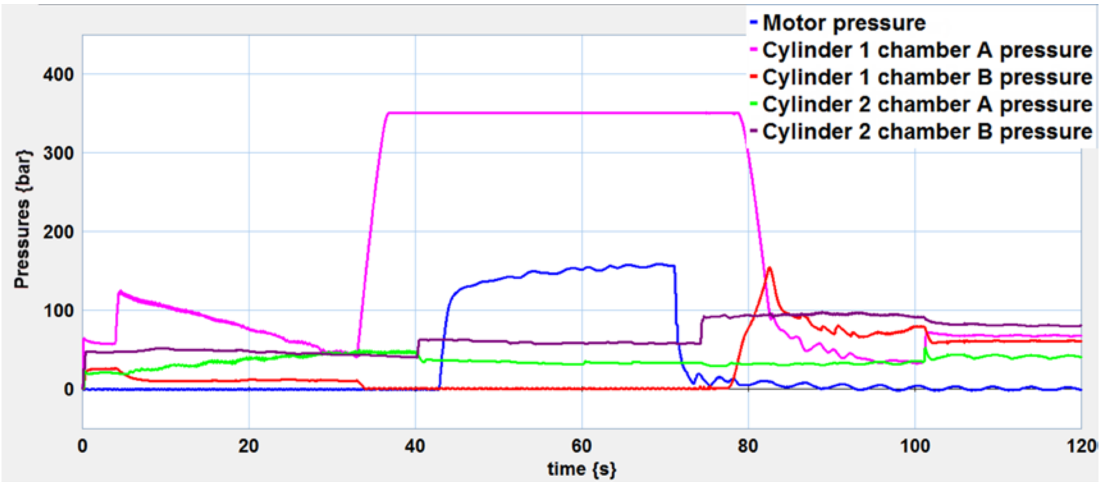
The initial conditions set a 45 degree angle at x direction and 5s wave ramp period. Heave amplitude was defined at 1m with 10s period. Despite the performance of the hydraulic systems, when arbitrary velocities were applied, the load sway was damped out after two crossings of the load and crane tip, i.e. one sway cycle. The effectiveness of heave compensation was obvious and using winch for heave compensation was better than using the cylinders.

Fig. 10: Crane tip position and load position during recovering: (a). Anti-sway; (b). Heave compensation using crane and winch

## 4.2 Simulation of hydraulic dynamics

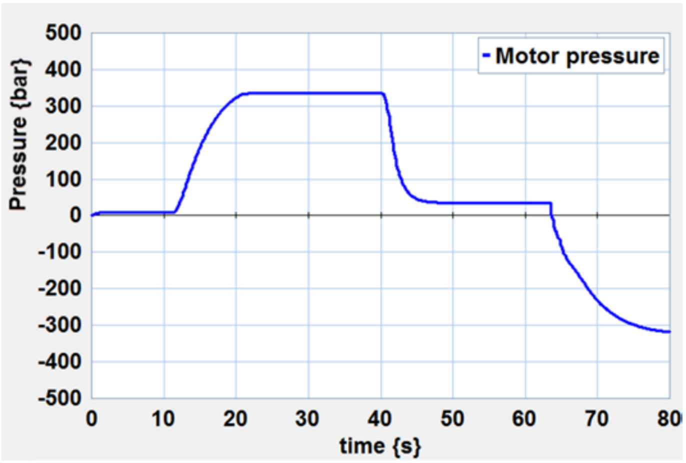
As stated previously, one of the contributions of this project work was that the model included not only the control algorithms and animation, but also the physical properties of the system. The following results from simulation of the model in Figure 6 are presented. The parameters assigned in the model are not necessarily suitable, but can be adjusted easily.

Hydraulic pressure is a critical index for the performance of hydraulic systems. Hydraulic fluid flows from high pressure to low pressure and thus creates power. On one hand, pressure must not increase over the safe working pressure of one component. On the other hand, pressure must not be negative in case of a cavitation problem, Manring (2005). Figure 11 (a) shows the hydraulic pressures at the crane motor and cylinders. Figure 11 (b) shows the hydraulic pressure at winch motor. The results provided useful data for system optimization and fault tracing. None extreme pressure over the allowed limits (350bar in this case) was found during the simulation.




(a)

The initial conditions set the the load at the recovering position without sway. First, the load was lifted up from 5s, and the pressures at the cylinder chambers varied along with the cylinder movements. Cylinder 1 reached maximum length at 33s and the pressure built up until the maximum pressure at 350 bar. At 42s the load was moved to an over board position. The motor pressure (blue) increased and stopped at 72s. From 80s the load was lowered down to the deck.

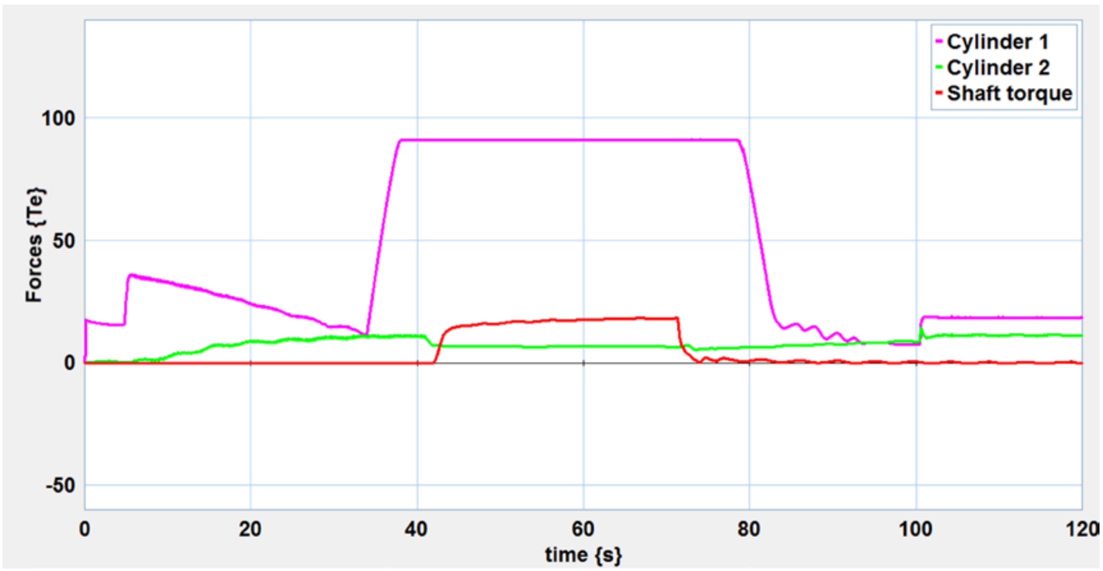



(b)

The initial conditions was set at the motor pressure at 0bar. At 12s the winch wound in to lift up the load. The pressure at the motor increased until the pump reached the maximum pressure at 350bar, giving maximum output flow. From 64s the winch wound out to lower down the load. The pressure was negative, which indicated the sign of the pressure difference on both sides of the motor.

Fig. 11: Pressures of: (a) the crane motor and cylinders and (b) winch motor

Structural failures of the crane booms and joints are one of the common failures during crane operation due to the heavy lifting of the pendulum load. The forces applied on the cylinder pistons were obtained, Fig. 12.




Referring to the operations in Figure 10 and 11, the load was firstly lifted up and then moved to the over board position and lowered down on the deck. The unit of the force was given in Te.

Fig. 12: Cylinder forces and motor torque

## 5 Conclusion and future work

In this paper, modelling of an offshore hydraulic crane based on BG method was described. A flexible control algorithm for offshore operations, including control functions for heave compensation and load anti-sway, was integrated and simulated. The presented modelling and simulation architecture using the BG method offers much potential for simulating the control of complex multi-disciplinary machinery systems. The BG models are the building blocks of the proposed offshore crane prototyping simulator. Other sub-modules mentioned include the haptic device with force feedback, hydro-dynamic sub-model of the wave and ship, and thermal sub-models for the cooling and energy consumption.

Modelling and simulation with integrated models of dynamic systems combines product and system design, control algorithm development, operational behavior, and system performance testing and analysis. The results of the research work provided a virtual prototyping environment to the offshore industry with pre-testing, fault finding, error investigating, and operation verification functions. Virtual simulation with 3D animation can also be used for training operators.

One of the challenges was the real-time performance of running the model when several sub-modules are integrated. Each sub-model had its own computation step size and frequency. Using filters helped to reduce the computation frequency bandwidth but, nevertheless, the computation power of the PC was one cause. The model was implemented into different levels that oriented to different simulation purposes. Another possible alternative solution is to distribute the model into several sub-units, and export and compute as co-simulation units running on multiple threads of one or several machines.

As continuity, a meta module library for offshore crane related systems needs to be defined and developed. To achieve a generic solution for modelling and real-time simulation of complex multi-domain systems, an interfacing framework for integration of the sub-modules needs to be developed. One possibility is to use the Functional Mock-up Interface (FMI) standard for co-simulation of dynamic models from various simulation tools exported as a Functional Mock-up Unit (FMU), which describes the interfaces and variables of the sub-unit model using a combination of xml-files and compiled C-code, Modelica Association (2014)

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