Active heave compensation of a floating crane using electric drive

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Abstract—Floating cranes have shown their benefits in port operations and in offshore tasks. With the increase of offshore wind farms, installation and maintenance of the wind turbines require crane vessels for multiple tasks. In order to operate in open sea regardless the ocean waves, active heave compensation system is necessary to be installed. In this work an electric drive is proposed for active heave compensation of a floating crane, as more sustainable solution among other benefits. Dynamic model of a floating crane was derived for model based control. Different control algorithms were developed and numerically tested for efficiency.

Index Terms—floating crane, active heave compensation, dynamic modeling, controller design

I. INTRODUCTION

Nowadays there is an increased activity in ocean and offshore engineering. The capacity of offshore wind farms has increased from 4,117 MW in 2011 to 18,814 MW in 2017 [1], and the trend is to install more and bigger wind turbines. Installing submarine power cables require quite an amount of resources [2] and sometimes remote-operated vehicles are used for installation. The wind farms rely on regular maintenance to maintain their high efficiency. The operation and maintenance costs are estimated to be 14%-30% of the total life-cycle costs of wind farms [3] [4]. For all these offshore and ocean engineering operations and their maintenance, there is increased demand for vessels to provide the technicians, components, spare parts and other resources. The variable costs of a crane vessel can be up to 40,000 USD/day [5]. To increase the workability and efficiency of offshore crane vessels, it is necessary to design these systems to handle weather conditions accordingly [17]. Ocean waves are impeding the handling of payloads when offshore crane vessels are operating in open sea. When the payload is held, lowered or lifted at sea, active heave compensation (AHC) is necessary to disconnect the heave motion the vessel endures with respect to the payload. This results in the decoupling of the payload motion from the vessel motion. Currently, there are companies trying to find alternatives for hydraulic actuators used for passive heave compensation (PHC) and AHC. Instead of using hydraulic systems, it may be possible to use electric drives. The reasons for using electric drives are mainly the lack of an oil reservoir (which can also cause leaks) and lower motor noise. With the increased attention to sustainability, choosing electric actuators instead of hydraulic may be appealing for consumers [6]. However electric actuators used for PHC and AHC is a field that is not significantly explored yet [7].

The design of a system for AHC relies on the choice of actuation, in this case an electric drive, and a control strategy. Two commonly applied controllers are the well established PID controller and LQR and the more advanced model predictive controller (MPC) [8]. The main advantages of MPC over PID are the abilities to handle processes with large time delays, non-minimum phase and unstable processes. Also, PID controllers tend to have a large phase lag which might be another reason for preferring a MPC. Furthermore a MPC is easy to tune and it excels in handling constraints with respect to a PID controller. Finally, MPC is also able to handle systematic changes such as the failing of sensors and actuators. The combination of using an electric drive in combination with a MPC is a novel approach in AHC [7]. In this work AHC system is explored by analysing the dynamical system and setting up a control design strategy. Dynamic model had been derived and used for controller design. Thereafter, numerical simulation results are presented and discussed using Matlab as modeling platform. This work explores the benefits of AHC based on electric drive and traditional controllers like PID and LQR, as well as model based MPC.
II. DYNAMIC MODELING

The design of an AHC system began with a dynamic model derivation of a floating crane. The system shown in Fig. 2a consists of a shipboard crane that is driven by an AC induction motor in combination with a gearbox and a drum. The drum holds the steel cable which is loaded by the payload. When operating at sea, waves are inevitable and have to be accounted for. The handling of the payload is important in these circumstances as waves can be dangerous for maintaining a predictable motion of the payload. Hence, the control objective is to decouple the upward wave motion of the ship with respect to its payload by using AHC. Fig. 2b shows how the payload is held at a constant position since the drum, together with a gearbox and the AC induction motor, compensates for the heave. As the vessel sustains an upward heave motion, the drum unwinds to retain the payload at a constant position. On the other hand, when the vessel makes a downward have motion, the drum winds up in order to maintain the payload at the same position. With the aim to reach the control objective, the following assumptions are made:

- The crane structure is a rigid body.
- The steel cable will be considered as a mass-spring-damper system.
- The mass of the steel cable is neglected.
- The mass and the friction of the sheave is neglected.

Fig. 2c summarises these assumptions where \( k_c \) is the spring constant, \( d_c \) the damping coefficient, \( A_c \) the surface area and \( E \) the Young’s modulus of the steel cable. Moreover, \( r_d \) and \( \theta_d \) are the radius of the drum and the angular acceleration of the drum, respectively. Furthermore, \( \ddot{w} \) is the wave acceleration and \( g \) the gravitational constant. The mass of the payload \( m_p \) has its position measured by using parameter \( z \) whereas \( z_0 \) is the length of the cable when loaded.

\[
R = \frac{1}{2} d_c \frac{\dot{z}}{2} (4)
\]

Also, the potential energy is given by (5).

\[
U = \frac{1}{2} k_c z^2 (5)
\]

Fig. 3: Electro-mechanical system

A. Electro-mechanical system

The mechanical system contains an AC induction motor which is connected to the drum through a reduction gearbox. The steel cable around the drum is suspended by the payload using also a sheave which is found at the tip of the crane. The electro-mechanical system is shown in Fig. 3 where \( \theta_m \) is the motor angle, \( J_m \) the motor shaft’s mass moment of inertia. Furthermore, \( J_d \) is the mass moment of inertia of the drum and \( \theta_d \) the drum angle and \( i \) is the gear ratio of the gearbox. Last but not least, the motor torque is defined as \( T_m \) and whereas the effective torque is \( T_{eff} \). The motor torque of the AC induction motor is defined by (1).

\[
T_m = \frac{p}{2 \omega_r (\frac{R_2}{s})^2 + (\omega_r (L_1 + L_2))^2} \frac{R_1}{s} (1)
\]

Where \( p \) is the number of poles, \( \omega_r \) the angular velocity of the rotor, \( E_q \) the rated phase voltage and \( s \) the slip. Furthermore, \( R_1 \) is the rotor resistance, \( R_2 \) the stator resistance whereas \( L_1 \) and \( L_2 \) are the stator inductance and the rotor inductance, respectively. In order to control the speed and the torque of the AC induction motor, a variable-frequency drive (VFD) is connected to the AC induction motor. The motor input frequency and voltage could be varied to control the latter. The efficient mass moment of inertia \( J_{eff} \), seen from the AC induction motor, is presented by (2).

\[
J_{eff} = J_m + \frac{J_d}{i^2} + \frac{m_p \cdot r_d^2}{i^2} \quad \text{with} \quad i = \frac{T_{eff}}{T_m} (2)
\]

For obtaining the differential equation for the system, Lagrange’s method is applied as shown in (6) where \( T \) is the kinetic energy, \( R \) the energy dissipation, \( U \) the potential energy, \( q_j \) the independent coordinate of the system its motion and \( G_j \) the corresponding load in the independent coordinate. The independent coordinate \( q_j \) can be substituted by \( z \). The kinetic energy of the system is shown in (3).

\[
T = \frac{1}{2} m_p \dot{z}^2 (3)
\]

Moreover, the energy dissipation is presented in (4).

\[
R = \frac{1}{2} d_c \dot{z}^2 (4)
\]

Also, the potential energy is given by (5).

\[
U = \frac{1}{2} k_c z^2 (5)
\]
The three previous mentioned equations are now defined for (6).
\[
\frac{d}{dt} \left( \frac{\delta T}{\delta q_j} \right) - \frac{\delta T}{\delta q_j} + \frac{\delta R}{\delta \dot{q}_j} + \frac{\delta U}{\delta q_j} = G_j
\]  
(6)
Rewriting (6) results in the second order differential equation, seen in (7).
\[
m_p \ddot{z} + d_c \dot{z} + \frac{E A_c}{L_0} \dot{z} = m_p (\ddot{\bar{w}} + \frac{T_{eff}}{J_{eff}} \tau_d)
\]  
(7)
The aforementioned equation shows the motion of the cable suspended payload. The vibrations of the load are compensated by wave accelerations \(\ddot{\bar{w}}\) and the effective torque of the drum \(T_{eff}\). \(L_0\) represents the steel cable in unloaded state.

### III. CONTROL STRATEGY DESIGN

The differential equation for the load vibrations can be rewritten in the form of a state space shown in (8).
\[
\dot{x} = A \bar{x} + B \bar{u}
\]
\[
y = C \bar{x} + D \bar{u}
\]  
(8)
with the states \(\bar{x} = [\dot{z} \ z \ \dot{\theta} \ \theta]^T\). The wave acceleration will be considered as an unmeasured disturbance in this work. However, it is possible to predict waves on the sea 2.5 hours in advance [9] and use a prediction model that will act as a measured disturbance.

\[
\dot{x} = \begin{bmatrix}
-\frac{d_c}{m_p} & -\frac{E A_c}{m_p L_0} & 0 & 0 \\
\dot{z} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & -\frac{\dot{\theta}}{\theta} & 0
\end{bmatrix}
+ \begin{bmatrix}
\frac{r_d T_{eff}}{J_{eff}} & + \ddot{\bar{w}} \\
0 & 0 \\
\frac{r_d T_{eff}}{J_{eff}} & 0
\end{bmatrix}
\]  
(9)
\[
y = \begin{bmatrix}
0 \\
\bar{z} \\
0 \\
0 \\
\bar{\theta}
\end{bmatrix}
\]

As shown in the C matrix, the position of the payload is measured and the motor angle is measured. The position of the payload is measured with a motion reference unit (MRU) and the drum angle is measured with a rotary encoder. It is also possible to use an inertial measurement unit (IMU) for measuring the position of the payload. The block diagram of the system, presented in (9), is presented in Fig. 4.

![Fig. 4: Block diagram of the system](image)

It shows that the angle of the drum and the position of the payload are processed in the reference model. The difference between the setpoint and the sensor measurements result in a certain error. The model predictive controller will stimulate actuator input in order to compensate for the error. The wave prediction model is the unmeasured disturbance that acts on the mechanical system of the offshore crane vessel. By utilizing MATLAB, the system is tested and is proven to be stable, controllable and observable. Then a discretization process is performed in order to receive the discrete state space of the system as shown in (10).

\[
\dot{x}(k+1) = A_d \bar{x}(k) + B_1 \bar{u}(k) + B_2 \ddot{\bar{w}}(k)
\]
\[
y(k) = C \bar{x}(k)
\]  
(10)
For obtaining the optimal input control sequence at a given time step \(k\), a cost function \(V_N\) is derived in the form of (11).

\[
\min_{\bar{u}_N} V_N(x_0, \bar{u}_N) = \min_{\bar{u}_N} \left( \sum_{k=0}^{N-1} \left\{ \frac{1}{2} x(k)^T Q x(k) + \frac{1}{2} u(k)^T R u(k) \right\} + \text{min stage cost} \right)
\]
\[
\text{subject to constraint } \begin{cases}
V_f(N) = \text{terminal cost} \\
V_f(0) = \text{stage cost}
\end{cases}
\]  
(11)
Where both \(Q\) and \(P\) are positive definite weight matrices whereas \(R\) is a semi positive definite weight matrix. The \(Q\) and the \(R\) matrix are chosen, then the \(P\) matrix is the algebraic solution of the Ricatti equation [10]. Also, \(\bar{u}_N\) is the optimization variable whereas \(N\) is the horizon length. The cost function, which is minimised, is subject to certain constraints shown in (12), where \(N_c\) is the control and \(N_p\) the prediction horizon.

\[
T_{eff,min} \leq T_{eff}(k) \leq T_{eff,max} \quad \forall k = 0, 1, \ldots, N_c - 1
\]
\[
\bar{w}_{min} \leq \bar{w}(k) \leq \bar{w}_{max} \quad \forall k = 0, 1, \ldots, N_c - 1
\]
\[
\bar{z}_{min} \leq \bar{z}(k) \leq \bar{z}_{max} \quad \forall k = 1, 2, \ldots, N_p
\]
\[
\bar{\theta}_{min} \leq \bar{\theta}(k) \leq \bar{\theta}_{max} \quad \forall k = 1, 2, \ldots, N_p
\]  
(12)

### IV. NUMERICAL SIMULATION

The numerical simulations are carried out in Matlab where the system is analysed with ocean wave disturbances and different control strategies. The list of parameters that are kept constant for every simulation are shown in Table I. An analysis is made between MPC, LQR-controller, PID-controller and open-loop. In total five simulations are carried out. Table II shows the parameters that are tuned during the simulations, where \(K_P\) is the proportional term, \(K_I\) the integral term and \(K_D\) the derivative term of the PID-controller. The system has to compensate for the wave acceleration \(\ddot{\bar{w}}(t)\). In order to operate the offshore crane vessel in a safe way, it is recommended that the wind speed should not exceed above 9.0 \(m/s\) (32.4 \(km/h\) or 17.49 knots) [11]. Wind speeds of around 9.0 \(m/s\) result in sea waves with an approximate amplitude of 1.5m and a period of about 5s [12]. The function that is used as the disturbance is shown in 13 after taking the second derivative of \(w(t)\).
TABLE I: Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$J_{eff}$</td>
<td>$4.0 \cdot 10^{-3} \ [kg \cdot m^2]$</td>
<td></td>
</tr>
<tr>
<td>$m_p$</td>
<td>$1.0 \cdot 10^{-3} \ [kg]$</td>
<td></td>
</tr>
<tr>
<td>$E$</td>
<td>$8.5 \cdot 10^{10} \ [Pa]$</td>
<td></td>
</tr>
<tr>
<td>$A_c$</td>
<td>$2.0 \cdot 10^{-4} \ [m^2]$</td>
<td></td>
</tr>
<tr>
<td>$L_d$</td>
<td>$1.0 \cdot 10^2 \ [m]$</td>
<td></td>
</tr>
<tr>
<td>$d_c$</td>
<td>$1.8 \cdot 10^3 \ [kg/s]$</td>
<td></td>
</tr>
<tr>
<td>$N_c$</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>$t_s$</td>
<td>0.5</td>
<td>[s]</td>
</tr>
</tbody>
</table>

\[
\ddot{w}(t) = - \left( \frac{2\pi}{T} \right)^2 A \sin \left( \frac{2\pi}{T} t_s \right) \tag{13}
\]

The first simulation, presented in Fig. 5 and Fig. 6, shows that all controllers are stable. During the time the wave acceleration is active the PID-controller is outperforming both the LQR-controller and the open-loop controller. Even the MPC is outperformed by the PID-controller. However, when the wave accelerations stop, the settling time $t_{set}$ becomes an important characteristic of the controller. For the simulation the settling time has to be within the $\pm 0.01 m$ steady state band. The LQR-controller shows it has the fastest settling time compared to the other controllers although it has more overshoot and more oscillations than the PID-controller and the MPC. Table III summarises the motion decoupling percentages and the settling time of the controllers. The motion decoupling is calculated by (14).

\[
\text{Peak}_{\text{OpenLoop}} - \text{Peak}_{\text{Controller}} \cdot 100\% \tag{14}
\]

Fig. 6: Performance of the four controllers PID, LQR, open-loop seen from a narrowed intervall

The MPC and the PID-controller perform outstanding compared to open-loop control and the LQR-controller when looking at motion decoupling. However, the settling time of the LQR-controller is outperforming the other three controllers. For constant waves at sea, motion decoupling is the decisive factor for choosing the right controller. Table IV shows all the results of the five simulations. The MPC has a very consistent motion decoupling (MD) even when the prediction horizon and the weight matrices $Q$ and $R$ are varied. However, when the wave accelerations stop, the settling time of the MPC depends on the ratio between $Q$ and $R$. The bigger this ratio is, the larger the settling time of the MPC is. Also note that the ratio between $Q$ and $R$ is the same in both simulation 4 and simulation 5. This has a minimum effect on the motion decoupling. Considering the results of the MPC, a final simulation will be done by halving the sampling time $t_s = 0.25$ instead of $t_s = 0.50$ and utilising the same parameters in simulation 4. The PID-controller performs

TABLE III: Motion decoupling and settling time of the controllers in the first simulation

<table>
<thead>
<tr>
<th>Simulation</th>
<th>Motion decoupling [%]</th>
<th>Settling time [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>LQR-controller</td>
<td>33.3</td>
<td>10.49</td>
</tr>
<tr>
<td>PID-controller</td>
<td>97.0</td>
<td>11.75</td>
</tr>
<tr>
<td>MPC</td>
<td>92.3</td>
<td>12.32</td>
</tr>
<tr>
<td>Open-loop control</td>
<td>0</td>
<td>11.34</td>
</tr>
</tbody>
</table>

The second simulation, presented in Fig. 7 and Fig. 8, shows that the MPC is outperforming the PID-controller, even though the PID-controller is still outperforming the LQR-controller. The simulation is performed with the same sampling time $t_s = 0.50$ but with the parameters from simulation 4. Even though the controller parameters are the same, the MPC performs better. The reason for this is that the MPC uses a horizon of 4 samples, which is equal to $50$ ms, instead of one sample, which is equal to $20$ ms. This results in a more consistent motion decoupling (MD) compared to the other controllers. For the simulation the motion decoupling has to be within the $\pm 0.1 m$ steady state band. The PID-controller shows it has the fastest motion decoupling compared to the other controllers although it has more overshoot and more oscillations than the MPC. Table IV summarises the motion decoupling percentages and the settling time of the controllers. The motion decoupling is calculated by (14).

\[
\text{Peak}_{\text{OpenLoop}} - \text{Peak}_{\text{Controller}} \cdot 100\% \tag{14}
\]

The MPC and the PID-controller perform outstanding compared to open-loop control and the LQR-controller when looking at motion decoupling. However, the settling time of the LQR-controller is outperforming the other three controllers. For constant waves at sea, motion decoupling is the decisive factor for choosing the right controller. Table IV shows all the results of the five simulations. The MPC has a very consistent motion decoupling (MD) even when the prediction horizon and the weight matrices $Q$ and $R$ are varied. However, when the wave accelerations stop, the settling time of the MPC depends on the ratio between $Q$ and $R$. The bigger this ratio is, the larger the settling time of the MPC is. Also note that the ratio between $Q$ and $R$ is the same in both simulation 4 and simulation 5. This has a minimum effect on the motion decoupling. Considering the results of the MPC, a final simulation will be done by halving the sampling time $t_s = 0.25$ instead of $t_s = 0.50$ and utilising the same parameters in simulation 4. The PID-controller performs

Fig. 7: Performance of the four controllers PID, LQR, open-loop
very well considering the motion decoupling percentages. The first simulation is executed by setting all three PID-controller terms on 35. The second simulation is done by holding $K_P$ on 35 and the other two terms on 10. One can see that the motion decoupling decreases in the second simulation with respect to the first simulation. However, the settling time of the second simulation is higher than of the first simulation. Hence, for the PID-controller it is a trade-off between motion decoupling percentage and the settling time when picking the PID-controller terms. The LQR-controller is not performing very well considering the motion decoupling. The LQR-controller performs the best when the ratio between $Q$ and $R$ is very high. A large gain is required for the LQR-controller to compete with the MPC and the PID-controller. The settling however is consistent. Now a final comparison is made between the PID-controller of the first simulation and the MPC of the fourth simulation as they have the best performance looking at motion decoupling percentage. A third MPC (also with parameters used of the fourth simulation) is also added but its sampling time is not 0.50 s but 0.25 s. Fig. 7 shows the results of the three controllers. The motion decoupling of the MPC has improved significantly when the sampling time is halved. With a smaller sampling time of 0.25 s, the MPC is able to optimise twice as many intervals than when using a sampling time of 0.50. However, the computational time increases. With a prediction horizon of $N_p = 7$, the MPC is relatively aggressive. This is necessary as for incoming waves, control action have to be taken. On the other hand, if the wave frequency is low, one may opt for a more robust controller and therefore increase the prediction horizon. Furthermore, with a control horizon of $N_c = 5$, the computational effort is reasonably low. Also increasing the control horizon results in a more aggressive controller. In this scenario the PID-controller performs better. However, the PID-controller has no knowledge of constraints. Also it does not have the ability to coop with constraints. Due to the fact MPC has the ability to anticipate in the future, it is an excellent choice when the waves are predicted in advance. Replacing hydraulic actuation by electric actuation will be a challenge for the future. The power density of hydraulic actuation systems is higher than AC drives [7]. For example, [13] shows that only hydraulic actuation provides the required power density. The final MPC with $t_s = 0.25$ provides maximum load position of 0.091 m which is comparable and sometimes performing better than [14] and [15] in which hydraulic actuation is used. Considering all the results and literature, the combination of MPC and AC induction motors is promising. The motion decoupling is consistent and may achieve even higher values than approximately 94%. However, the main downside of AC drives is the power density. The power density is defined as a power-to-weight ratio of the actuator. Hydraulic actuation is unmatched for heavy lifting. Furthermore, [16] shows that the hydraulic actuators provide a higher power density than AC drives. AHC may be achieved using a VFD connected to the AC induction motor. Also, AC drives should first be considered in offshore support ships for bringing supplies and utilising maintenance. When AC drives are used in combination with wide diameter winches, a large torque is required in order to adjust the motor speed. For example, In a case of a winch with a high inertia, the motor

![Fig. 7: Performance of the three controllers in the final comparison](image-url)
torque might not be able to surmount the load torque enough in order to achieve an appropriate acceleration. Design of floating structures for different purposes (living, aqua-farming, energy, etc.) has raised a lot of attention in the last years. This comes from the need of more space on land, which is not always possible, and sustainability in utilizing the open sea resources. Floating terminals [17] can be beneficial in different scenarios, commercially and for disaster relief. All these concepts would benefit from sustainable AHC system based on electric drive and model based control.

V. CONCLUSION
This work explored the combination of MPC and AC induction motors for AHC of floating crane. The initial results showed that the feasibility, potential and benefits of such systems. Even though the main downside of AC drives is the power density, other benefits coming from sustainability trends and energy transition might encourage the maritime industry to look into this solution. AHC could be accomplished with a variable-frequency drive connected to the AC induction motor. For further investigation it is also recommended looking into multiple, smaller AC drives which work together in order to achieve AHC. However, the main problem stays that AC drives take a lot of space. Large AC drives might also result in distortions of the power grid if the drive uses too much power than the grid can handle. One should look into realistic offshore vessel designs which meet the needs of the current market. For example, the offshore wind turbines are becoming larger and therefore bigger offshore (support) vessels are needed. This also results in the need of actuators which provide enough power for maintaining these structures. There is however an optimal combination of offshore crane vessels with a corresponding AC drive. This is an aspect one should look into more extensively. There is also unknown territory when using accurate wave predictions in combination with MPC and AC drives. Further research is required to obtain more in-depth knowledge of this subject. On top of this, a physical model should be made in combination with a towing tank. Both regular and irregular waves can be used as input in order to achieve a variety of results.

REFERENCES