# A NOVEL APPROACH TO ANTI-SWAY CONTROL FOR MARINE SHIPBOARD CRANES

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## ABSTRACT

This paper addresses the development of a novel anti-sway control approach for marine shipboard cranes, offering stability, safety, and efficiency during lifting, handling, transportation, and other manipulation. The proposed idea consists of the development of an integrated system with control strategies to both reduce the effect of three-dimensional payload pendulation and to minimize wave impact during shipboard crane manipulation. We propose to use a control mechanism based on energy dissipation. The simulation results confirm the principle and effectiveness of the proposed methods for damping out pendulation. In future work, we aim to minimize wave impact on the payload by reducing the dynamic forces through controlling the length of the hoist cable while adapting to the lateral wave velocity. During the final phase of this project, the proposed control strategy will be implemented as a real physical prototype for controlling different kinds of shipboard cranes.

## I. INTRODUCTION

Shipboard cranes are widely used to handle and transfer objects from large container ships to smaller lighters or to the quays of the harbours. The control of cranes is always a challenging task which involves many problems such as load sway, positioning accuracy, suppression, collision avoidance, and manipulation security. Generally, shipboard cranes are relatively big, heavy, stiff, and rely on complex kinematic models of their system as well as an equally complex model of the environment with which they interact. Typical shipboard crane operations are shown in Figure 1. There are two big challenges during crane operation. First, unlike cranes mounted on solid bases, load sway is affected by the ship’s motion when the load is hoisted by a shipboard crane. This motion produces large pendulations of the hoisted cargo and causes operations to be suspended. Second, when the payload is hit by waves on the surface of the sea, it is subject to an impulsive hydrodynamic slamming force, which, in harsh sea conditions, can damage the payload.

The pendulation caused by the payload and the sway caused by the waves not only limits the functionality of marine facilities in adverse weather conditions, but also poses a threat to the safety of personnel and equipment during off-shore operations. The pendulation is often induced by a combination of the vessel's motions and the crane operator's normal actions. In general, the payload acts as a spherical pendulum whose attachment point is manoeuvred using the crane's degrees of freedom. As the operator commands the various axes of the crane to affect rigid body payload translation and rotation, the payload's sway degrees of freedom can be excited. This payload pendulation problem becomes much more complicated if the crane is mounted to a moving vessel. An experienced operator can often generate crane inputs correctly, in such a way that the payload is sway-free at the end of the manoeuvre. However, training an operator to use a crane requires significant resources and poses potential hazards. Imagine the challenge of installing a 200 ton sub-sea module at 2000 meters sea depth with an accuracy of centimetres! When considering both working efficiency and safety, quality control is impossible to achieve.

Figure 1 Typical shipboard Crane operation.

We propose a combined control strategy that both reduces the effects of payload pendulations and minimizes wave impact on shipboard crane manipulation. The new control approach will improve the safety of demanding marine operations. In this paper, we only present the first phase and current work of this project. The research result will be integrated with the current “crane/winch simulator” developed by

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the Offshore Simulator Centre. Finally, a real prototype will be built and tested at the Rolls-Royce marine AS.

## II. RELATED WORK

There are two challenging problems of handling shipboard cranes. To date, cargo loading operations at sea are often paused in case of unfavourable weather conditions. Without an efficient control mechanism, a modest movement of the ship can result in a large sway motion of the cargo. This can lead to dangerous situations. As a consequence, time and money are often wasted on waiting for better weather conditions, or worse, the risk may be taken to load the cargo in dangerous conditions. Once sea conditions build to a low sea state 3 (as defined by the Piersion-Moskowitz Sea Spectrum, with significant wave heights in the range of 1.0 - 1.6 m), hoisted payload pendulations on crane ships become dangerously large and operations must be suspended [1]. An analysis of worldwide weather and sea-condition data show that more than 35% of all potential joint logistics over the shore operations (JLOTS) sites have sea conditions of sea state 3 or higher 50% of the time during which operations would be suspended [1], [2]. Nojiri and Sasaki [3] calculated that payload pendulations due to excitation frequencies near the resonance frequency of the cable-payload assembly also have a pronounced effect on the rolling and pitching motion of the crane vessel under the influence of both regular and irregular waves.

The other problem during the lifting and transportation of shipboard cranes with long hoist cables is wave impact. When the payload is hit by the waves on the surface of the sea, it is subject to an impulsive hydrodynamic slamming force which, in harsh sea conditions, can damage the payload. Also in this case, if the sea conditions become prohibitive, the operations have to be suspended.

Regarding the load pendulation, many researchers investigated the problem for fixed-base cranes. As the rolling motion of a ship is dominating, and the cranes usually work from the sides of the ship, the swaying of the payload is mainly confined to two dimensions. Some controllers were originally designed for boom cranes, while others were modifications of earlier work on gantry cranes. Two main approaches can be identified among this research: one targets pendulation suppression throughout the whole transport manoeuvre, while the other is more concerned with end-of-manoeuvre pendulation suppression, the so-called "elimination of residual pendulation". In both approaches, limited research included the operator as a part of the model plan. Lewis et al. [4] and Parker et al. [5] presented a three-dimensional linear model of a boom crane. A controller applies quasi-static filters to the operator's input commands to avoid exciting the natural frequency of the cable-payload assembly. Experimental results showed a significant reduction in both the in-plane and out-of-plane payload pendulations.

Balachandran et al. [6, 7] modified the common boom-crane configuration in order to suspend the payload from a pivot that in turn was suspended under the boom. The pivot acted as a non-linear vibration absorber, a mechanical filter, to absorb the cargo oscillations. They derived two-dimensional and three-dimensional models of this new configuration. Their simulations showed that the absorber can suppress sub-critical bifurcations and shift the bifurcation points arising from the non-linear dynamics of the cable-payload assembly. Current ship cranes use a Rider Block Tagline System (RBTS). In this method, the intention is to change the natural frequency of the pendulum. The current RBTS has many deficiencies. As a result, Hunt et al. [8] proposed a new crane structure to suppress the pendulation using adjustable damping to dissipate the pendulation energy. In 1999, Kimiaghalam et al. [9] studied a crane with an additional tagline connecting the rope and the boom. They used the length of the tagline as the single control parameter, and designed a fuzzy controller in order to keep the load in place. However, they found that load oscillations were decreased by only about 50 %. In 2000, Kimiaghalam et al.[10] considered a crane with a rope suspended from two points on the boom, with the cargo hanging from a pulley in the middle. They showed that it is possible to almost eliminate sway motion by applying a combination of feedback and feed-forward control to the rope length and luffing angle of the boom. Thus, an implementation of this system seems to solve the problem of in-plane payload-pendulation. In their approach, Kimiaghalam et al. focused on keeping the equilibrium position of the pulley in place, which they approached by using feedback and feed-forward control on the angle of the boom with the horizon. Although this approach has proven to lead to good results for small rolling angles, we expect it to quickly break down under more extreme conditions. We state that real-time tracking of the load position is essential in order to cope with heavy weather conditions. The main reason for this is that the distance from the suspension point to the actual cargo can be very large, and as such, a tiny movement of the suspension point can lead to large oscillations of the cargo. In addition, not only the angle of the boom is of importance, but also the position and velocity of the payload should be considered at the same time. This becomes more important at larger rolling amplitudes of the ship.

In the rest of the paper, we will present our anti-sway approach. Firstly, in Section III, we outline the main control methods, which includes three important cases. In addition, we show simulation results for the different control algorithms. A series of simulation will be introduced to confirm the strategy. Finally, in section IV, we discuss the results and draw our conclusions as well as describe our ideas for future work.

## III. INTEGRATED ANTI-SWAY APPROACH : METHODS AND RESULTS

We model the crane/winch system as a frictionless, non-elastic spherical pendulum. The motion of the pendulum is fully determined by algorithms steering the position of the support point and the length $L$ of the cable via the winch, and external noise acting upon the position of the support point. Both the support and the wire are modelled as massless objects. The mass $m$ of the load is in this approximation irrelevant for the motion of the pendulum. First, consider the simple pendulum moving in a plane, as shown in Figure 2, consisting of a mass $m$ attached to a cable of lenght $L$.

A load is connected to a massless cable that is suspended from a support point. The position of the pendulum is determined by its angle $\theta$ with its rest position (green) or the distance $s$ from the rest position along its circular trajectory. The forces acting upon the load are the gravitational force $F_g$ (pointing vertically down, not shown in the figure) and the centrifugal force $F_{cf}$ (pointing outward along the cable). $F_g$ and $F_{cf}$ are given by (1),

$$F_g = mg, F_{cf} = m\omega^2 L \qquad (1)$$

where $g$ is the gravitational acceleration and $\omega$ is the angular velocity of the load. If the lateral dispacement $d$ of the support point is in the direction of the load, as shown in Figure 2, the work done is negative, such that energy is taken out of the system.

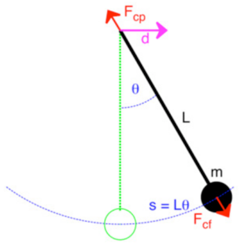


Figure 2 Simple Pendulum

At a given time $t$, the cable is under an angle $\theta$ with the vertical axis. We describe the position of the payload by the arc length $s$ measured from the rest point along the trajectory of the payload. The equation of motion follows from Newton's second law in the direction of the payload motion (2),

$$g \sin\theta = \frac{\mathrm{F}}{\mathrm{m}} = a = \frac{d^2 s}{dt^2} = \frac{d^2(L\theta)}{dt^2} = L\frac{d^2\theta}{dt^2} \qquad (2)$$

where $F$ and $a$ are the force on the payload and its acceleration, both in the direction of motion. We find the well-known differential equation (3), which is most easily solved by numerical methods.

$$\frac{d^2\theta}{dt^2} = \frac{g}{L}\sin\theta \qquad (3)$$

In the present work, we propagate the motion of the pendulum in three dimensions, while we in addition freely move the support position which determines the motion of the pendulum. We derive the equations of motion from the Lagrangian $L = K–V$, in which $K$ is the kinetic energy, as given in (4),

$$\frac{\boldsymbol{K}}{\boldsymbol{m}} = \frac{1}{2}[(\dot{\boldsymbol{x}} + \dot{\boldsymbol{x}}_0)^2 + (\dot{\boldsymbol{y}} + \dot{\boldsymbol{y}}_0)^2 + (\dot{\boldsymbol{z}} + \dot{\boldsymbol{z}}_0)^2] \qquad (4)$$

where $\boldsymbol{r}$=($\boldsymbol{x}$, $\boldsymbol{y}$, $z$) is the relative position of the load with respect to the support position $r_0$ = ($\boldsymbol{x}_0$, $\boldsymbol{y}_0$, $z_0$), and $V$ is the potential energy (5),

$$\frac{\boldsymbol{V}}{\boldsymbol{m}} = \boldsymbol{g}(\boldsymbol{z} + \boldsymbol{z}_0) \qquad (5)$$

We use spherical coordinates to find the equations of motion as shown in (6).

$$\left.\begin{array}{c} r = |r|, \theta = atan\left(\frac{\sqrt{x^2+y^2}}{z}\right), \phi = atan2\left(\frac{y}{x}\right) \\ \Phi = \dot{\phi} \\ \Theta = \dot{\theta} \\ \dot{\Phi} = \frac{-2L\Theta\Phi\cos\theta + \ddot{x}_0 \sin\phi - \ddot{y}_0 \cos\phi}{L\sin\theta} \\ \dot{\Theta} = \frac{1}{L}(-\ddot{x}_0 \cos\phi\cos\theta\sin\phi - \ddot{y}_0 \sin\phi\cos\theta \\ -\ddot{z}_0 \sin\theta - g\sin\theta + L\Phi^2 \sin\theta\cos\theta) \end{array}\right\} \quad (6)$$

These equations have been implemented in a C program that propagates the system in time using Runge Kutta 4. The input of the system is given by the acceleration of the support point as a function of time. These equations are valid as long as the hoist cable is fully stretched, which was verified by testing for positive tension on the wire.

In addition, the length $L$ of the wire is controlled via the winch. Although the results presented in this paper are in 2 dimensions, all simulations were done in 3 dimensions, and the methods presented here also are valid for 3-dimensional motion.

In order to control the motion of the pendulum, we will subsequently consider the following cases:
1. Damping out a large non-driven in-plane swaying motion.
2. Transport of the load from one position to another.
3. Damping out a driven planar oscillation.

### III.I DAMPING A NON-DRIVEN OSCILLATION

We will start with the first case, damping out a non-driven planar swaying motion. This is the main method that we propose to use in order to damp out swaying motion. The approach is based on reducing the energy of the system. In the current section, we use the method to damp out a non-driven oscillation. We will show that the method also works to damp out driven oscillations. In addition, we have tested the method in three dimensions, both for driven and non-driven oscillations, with similar results. Most importantly, we expect similar methods to work in combination with operator-driven actions.

When the amplitude of the pendulum is large, our approach is to dissipate the energy from the swaying motion as quickly as possible. We can do this by making the work performed by the support on the pendulum maximally negative. This, in turn is done by moving the support point into the direction of the displacement of the pendulum, as shown in Figure 2.

The work performed by the support point on the pendulum can be obtained by (7):

$$dW = \boldsymbol{F}_{\text{sup}} \cdot d\boldsymbol{r}_0 \qquad (7)$$

where $\boldsymbol{F}_{\text{sup}}$ is the force applied by the support point on the cable. The displacement $\boldsymbol{r}_0$ is in the horizontal direction whereas the force applied by the support point is directed along the direction of the hoist cable. The force applied by the support on the pendulum consists of three parts: a gravitational component $F_g \cos\theta = mg\cos\theta$, a centrifugal component $F_{cp} = m\omega^2 L$ and, if the support point in addition moves horizontally, a component depending on its acceleration:

$$F_{sup} = F_{cp} + F_g \cos\theta - m\ddot{x}_0 \sin\theta \qquad (8)$$

First assume that the support point remains at its initial position. As the displacement is zero, we see from (7) that no work is performed by the support. In addition, we see from (8) that the applied force vanishes if the acceleration of the support is given by (9)

$$a = \frac{d^2x_0}{dt^2} = \frac{(\omega^2 L + g\cos\theta)}{\sin\theta} \qquad (9)$$

This may happen for large accelerations of the support of the order of the gravitational constant, and large angles $\theta$ near 45 degrees. In this case, the hoist cable is no longer stretched. Also in this case the support point does not perform any work (8), this time because the applied force is zero, rather than the displacement. We see that the larger the instantaneous acceleration of the support is, the less negative is the work d$W$. As we wish to make the applied work as negative as possible, it might seem beneficial to apply a low acceleration. However, it is not only the instantaneous work that determines the energy dissipation from the pendulum, but its integrated value over time. Since the applied work is proportional to the displacement of the support point, it is beneficial to initially accelerate the support point maximally in the lateral direction towards the load until the maximal velocity has been reached. This is the method we apply in order to quickly dissipate energy from the pendulum at large swaying amplitudes. The results can be seen in Figure 3. The figure shows that whenever the $x$ position of the load crosses the $x$ position of the support point, the latter starts accelerating at maximum acceleration $a_{\max}$ into the new direction of the load, until its maximum velocity $v_{\max}$ has been reached. This method effectively dissipates energy from the pendulum, and eventually results in a situation where the support and load move together at a final velocity smaller than $v_{\max}$ (see Figure 3).

We also propose a general method for dissipating energy from the system: While applying the Lagrange method to integrate the equations of motion, we choose a trial step for the acceleration of the support point during the next time step. We then compute and record the total kinetic plus potential energy $K+V$ after this trial step. We repeat this for all possible accelerations $\frac{d^2\boldsymbol{r}_0}{dt^2}$ in all directions of the support point at the current position, and then choose the direction that leads to the state of minimal total energy. We then proceed to the next time step. In this way, energy is removed from the system as quickly as possible at all moments in time.

However, as we have mentioned before, minimizing the instantaneous work at all time does not lead to the desired result of minimizing the integrated work over time. On the other hand, when we instead minimize the kinetic energy of the system at all moments in time, rather than the kinetic plus potential energy, the system does exactly what we want: The support point moves at its maximal velocity into the direction of the load, such that the latter loses its kinetic energy as quickly as possible. This is the algorithm we use to dissipate energy from the system and in this way damp out large swaying motions. The results can be seen in Figure 3. The methods have been verified to work for three-dimensional oscillations as well (results not shown).

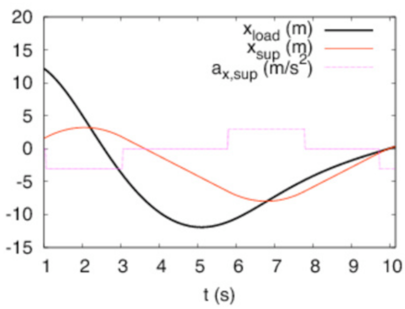


Figure 3 Basic control method: Damping by crane motion only. Positions of the load (black line) and support point (red line) as well as the acceleration of the support point (pink line) are shown as a function of time.

In Figure 3, the length of the hoist cable is 20 meter, the maximum speed of the crane $v_{\max}$ is 3 $m/s$ and its maximum acceleration $a_{max} = 3\, m/s^2$ . The initial displacement of the load is 45 degrees. The support point always moves at maximum speed towards the load position. The support point starts decelerating in advance when it approaches the lateral position of the load, in order to prevent an overshoot. When the maximum velocity of the support $v_{\max}$ has been reached, the acceleration is set to zero. In the final situation, the support point and the load move together at constant velocity.

In order to assure that the load is not accelerated into the wrong direction, the time of the next crossing between the x positions of the support and the load is calculated by extrapolation, in order to let the acceleration vanish exactly at this point in time. The lateral distance between the position of the load and the support at which the support point has to start its acceleration into the new direction is given by (10):

$$\Delta x = x_{load} - x_{sup} = \left(\frac{1}{2} v_{sup} - v_{load}\right) \Delta t \qquad (10)$$

where $\Delta t = \left|\frac{v_{sup}}{a_{max}}\right|$ is the time it takes to decelerate the support point to a standstill. In this way, motion of the the load and the support is always in opposite directions. Since the positions of the support point and the load are close, we neglect the acceleration of the load in the above computation.

Another method to dissipate energy from the pendulum in order to damp out a large swaying motion is by using the winch to vary the length of the hoist cable over time. We can study how the length of the cable should be varied in order to minimize the work performed by the winch. As before, the force applied by the load is always directed downward, along the direction of the cable, as long as there is tension on the wire. In this case, that means that the downward acceleration of the winch should not be larger than the component of the gravitational acceleration of the load along the cable, plus the acceleration caused by the centripetal force $\frac{F_{cp}}{m}$. We see that the force applied by the winch at the position of the support point is directed along the direction of the cable, i.e. pointing away from the direction of the load, under all normal circumstances. As a consequence, in order to minimize the instantaneous work performed by the winch on the pendulum, the cable should at all times be extended, and the acceleration should at the same time be small enough to keep a positive tension on the cable. To indefinitely extend the length of the cable is, of course, undesirable, although it is possible to damp out the swaying motion in this way. Instead, it is preferable to reduce the cable length at times when the force on the cable is minimal, and extend it when the force is largest. In this way, the applied work is smaller than zero. We can achieve this by using the winch to lift the load when it is at its highest points (with maximum potential energy; the force on the cable is minimal), and lowering it when it is at its lowest point (with maximum kinetic energy; the force on the cable is maximal). In effect, this means shortening the wire at constant speed when the load moves away from the support and lengthening it at constant speed when the load moves towards the support.

Figure 4 and Figure 5 show the simulation results for this method. It can be seen that this method uses a long time to dissipate energy from the pendulum, especially for longer cable lengths.

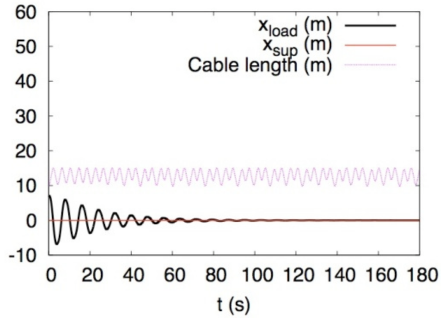


Figure 4 Winch damping.

In Figure 4, an initial displacement of 45 of the pendulum is damped out by adjusting the length of the cable by operating the winch. The maximum acceleration and velocity of the winch are 5 m/s$^2$ and 3 m/s, respectively, the cable length is varied between 10 and 15 meters.

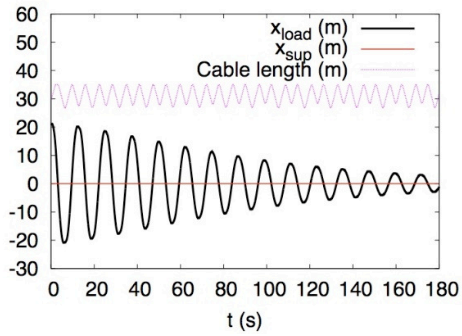


Figure 5 Winch damping for a long cable.

Figure 5 shows the same simulation results as those in Figure 4, the only difference being that the length L of the cable now equals 30 meters and is varied between 25 and 35 meters. It is seen that the damping process takes very much longer in this case.

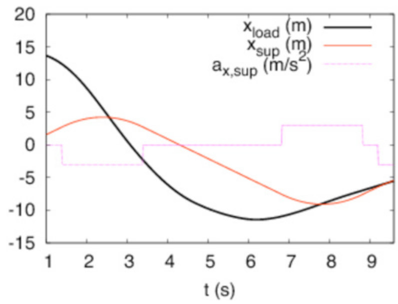


Figure 6 Damping by the combined effect of crane and winch.

In Figure 6, an initial displacement of the pendulum is damped out by a combination of the above methods for energy dissipation by crane motion and winch control. The positions of the load (black line) and support point (red line) as well as the acceleration of the support point (pink line) are shown as a function of time. Parameters are the same as in Figure 3; in addition the maximum velocity of the winch is set to 3 m/s, its maximum acceleration is set to 5 $ms^{-2}$. As can be seen by comparison to Figure 3, the additional effect of damping by using the winch is small: a steady state is reached about 10 % earlier than by only crane motion.

## III.II TRANSLATION

Besides damping out oscillatory motion (section III.I), another important method is to transport the load from one position to another, in the absence of external noise. This is a typical operation which a crane operator is trained at. Nevertheless, we implemented and tested the method for two reasons. Firstly, the method is ideal to damp out any lateral motion that is left after having damped out a motion by means of the methods in the previous section. Secondly, the method is an ideal test case for our simulations as a virtual operation which one can perform without needing an operator. Namely, the operation is able to perfectly move the load from one position to another, without any final oscillation. In the presence of noise, we can combine the translational motion with the methods for damping out oscillations of section III.I and III.III.

The method is exact when the load is initially hanging still, and there are no perturbations. Importantly, the method can also be combined with the above-mentioned method for energy dissipation to damp out sway motion, both initially, and during the phase where the crane moves at constant velocity.

In order to explain the method, it is easiest to first consider how the load can be accelerated from a standstill to a final velocity $v_f$ without causing it to sway. This is what is done during the first 5 seconds in Figure 7. In the second half of the translation in Figure 7 (5 s < t < 10 s), the reverse process is applied, in which the load is decelerated back to a standstill.

In order to first accelerate the support and load from standstill to a simultaneous movement at a velocity $v_f$ (at $t$=5 sec in Figure 5) we first accelerate the support to $\frac{v_f}{2}$ by a constant acceleration $a_{const}$ during a time $t_{acc}$ (although the method works as well for irregular accelerations over time). This is done during the first $t_{acc} \approx 0.5$ seconds in Figure 7. Next, the support point keeps this exact velocity during a time $t_{1/2}$, until the payload has exactly the same velocity ½ $v_f$ (slightly after 2 secconds in Figure 7). Note that the position of the load at that moment in time still lags behind the support position, although the velocities $v_{sup}$ and $v_{load}$ are exactly equal. From here on, we let the support point keep the same velocity $\frac{v_f}{2}$ for another time $t_{1/2}$ (until the start of the second acceleration pulse, shortly after 4 seconds in Figure 7), and finally we accelerate it to $v_f$ using the same constant acceleration $a_{const}$ during a time $t_{acc}$ (until the end of the second acceleration pulse, shortly after 4 seconds). At the end ($t$=$t_{end}$, shortly after 4 seconds in Figure 7), the support and the payload move together at a speed $v_f$, without any sway. We can see this by using the fact that the system is time-reversible, and by applying the following Galilean transformation (11),

$$\boldsymbol{v}_{\mathrm{sup}}(\boldsymbol{t}) = \boldsymbol{v}_{\mathbf{sup}}(\boldsymbol{t}_{\mathbf{end}}) - \boldsymbol{v}_{\mathrm{sup}}(\boldsymbol{t}_{\mathbf{end}} - \boldsymbol{t}) \quad (11)$$

where the subscript "end" indicates the place and time directly after the second acceleration pulse (shortly after 4 seconds in Figure 7). Since both velocities at $\frac{1}{2}t_{end} = t_{acc} + t_{1/2}$ are invariant under the above transformation, the transformation has to apply also for the rest of the trajectory:

$$v_{\mathrm{load}}(t) = v_{\mathrm{sup}}(t_{\mathrm{end}}) - v_{\mathrm{load}}(t_{\mathrm{end}} - t) \quad (12)$$

But then, it follows from symmetry that

$$x_{load}(t_{end}) = \int v_{load} dt = x_{sup}(t_{end}) = \frac{1}{2} t_{end} v_{sup}(t_{end}) \quad (13)$$

In conclusion, at $t$ = $t_{end}$ the support and load move together at the same $x$ position and velocity, such that we are able to accelerate the system to any desired velocity, starting from the system at rest. In the same fashion, we can decelerate suspension and payload from any given initial velocity to its rest position (see section III.II). This process is shown in the second half in Figure 7 (5 s < t < 10 s), where the payload is decelerated from a velocity of 3 m/s towards a standstill.

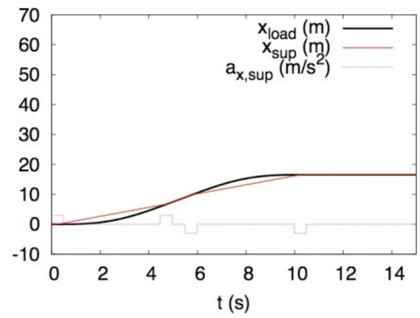


Figure 7 Translational motion by the double pulse method.

In Figure 7, the length $L$ of the wire is 20 meters; the load is translated from a standstill at an intial position $x$ = 0 m to a final position $x$ = 16.5 m in 10 seconds. Maximum lateral acceleration and velocity of the support are 3 $ms^{-2}$ and 3 $ms^{-1}$, respectively.

## III.III DRIVEN OSCILLATION

We here consider the response of the pendulum to a lateral periodic disturbance. A cosine signal is fed to the $x$ position of the support point. Without the presence of damping, the system is unstable. If the feeding frequency is similar to the natural frequency of the system, the oscillations grow significantly in amplitude already after few oscillation periods.

In order to damp out the oscillations, we apply the same methods as in section III.I. As shown in Figure 8, an initial displacement of the load of 45 and a driven lateral oscillation of the support are quickly damped out by the method considered under section III.I, in which the support at all times moves at its maximum speed into the lateral direction of the load. The used maximum acceleration and velocity of the support are 9 $ms^{-2}$ and 3 $ms^{-1}$, respectively. The lateral oscillation of the support position in Figure 8 is given by $f(t) = 5m\cos(\frac{2\pi t}{10s})$ while the wire length $L$ is 20 meters. The pendulum's resonance frequency is therefore (for moderate displacements) $\frac{1}{2\pi}\sqrt{\frac{g}{L}} = 0.11s^{-1}$, close to the driving frequency of 0.1 $s^{-1}$. Without a control algorithm, the oscillation would therefore quickly gain amplitude. The amplitude of the periodic wave is as large as 5 meter, but still the algorithm is able to fully damp out the oscillations, resulting in a linear motion that simply can be compensated for by the method discussed under section III.II.

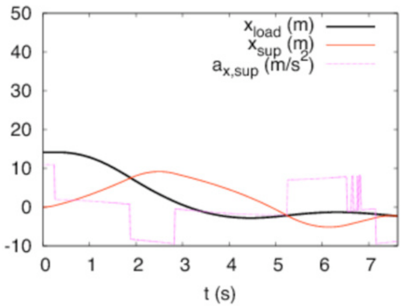


Figure 8 Damping out a periodic input signal.

## IV. DISCUSSION AND CONCLUSION

Our proposed idea consists of developing an integrated system with control strategies that reduce the effect of payload pendulation and that minimize wave impact on shipboard crane manipulation. We here present simulation results confirming the principle and effectiveness of the methods for damping out pendulation.

An important advantage of the proposed methods for dissipating energy is that they are expected to be relatively easily combined with operator-induced actions. When a human operator is involved, it is unwanted and dangerous if the control algorithm without a warning performs actions opposite to those of the crane operator. With our proposed method, we expect to be able to effectively damp out oscillations by using values of $a_{max}$ and $v_{max}$ that are small as compared to the accelerations and velocities used by the crane operator. In addition, the values of $a_{max}$ and $v_{max}$ can easily be adapted to the the current sea state: Without the presence of external noise both values may be set to zero, at intermediate sea states the values are set to fractions of the maximum velocity and acceleration of the crane, while at extreme sea states they may be set to their maximal values.

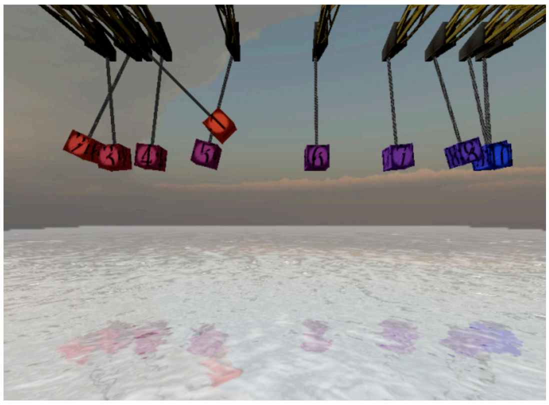

Figure 9 Combined crane-winch simulation.

A simulation of a combined crane-winch operation is shown in Figure 9. An initial 45 displacement (position 1 (pos. 1)) is damped out (pos. 1-3) by the combined crane and winch methods described under section III.I. Directly after this, the payload is translated over a (very) large distance of 62 meters. In order to do so, the support point is first accelerated (pos. 4-5) until it moves at fixed speed (pos. 6) and finally decelerated (pos. 7-9) until the prescribed position has been reached (pos. 10). The wire length is 20 meters and the total duration of the operation is 23 seconds. The simulation shows some of the working capabilities and efficiency of our integrated anti-sway approach. The simulation results were visualized by using Unity3D game engine. Although not shown in the figure, the same methods also apply when the system is under the influence of a driven planar perturbation during the operation, as well as a three-dimensional perturbation.

There is still a great amount of work left for future research. Considering the difficulties of integrating features of hydraulic cranes into the control method, we are currently focusing on investigating the nonlinear dynamic models of all related hydraulic components in the crane system. The work will be integrated with the "crane/winch simulator" developed by the Offshore Simulator Centre. Finally, a real prototype will be built and tested at Rolls-Royce Marine AS.

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