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Review

A review of vertical motion heave compensation systems

J.K. Woodacre [a,*], R.J. Bauer [a], R.A. Irani [b,a]

[a] Department of Mechanical Engineering, Dalhousie University, Halifax, Nova Scotia, Canada B3H 4R2
[b] Rolls-Royce Canada Limited - Naval Marine, 461 Windmill Road, Halifax, Nova Scotia, Canada B3A 1J9

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ABSTRACT

This paper provides a comprehensive review of vertical heave motion compensation systems used on ocean vessels from the early 1970s up to, and including, modern systems. Specifically, this review provides details on passive heave compensation, active heave compensation, hybrid active–passive heave compensation systems, and wave synchronization systems along with detailed explanations of the most common motion actuation methods, control schemes, and heave motion decoupling potential found with each. Based on the results of this review, it is recommended that more experimental work be carried out on real-world systems to experimentally validate the active heave compensation controllers being designed and simulated in literature. It is also suggested that future work involving model-predictive control may be used to further improve upon the performance of the current active heave compensation systems.

Contents

1. Introduction

Equipment handling on the ocean can be a difficult task, especially during rough seas. When lifting, lowering, or holding a load at sea, heave compensation is used to remove vessel heave motion from the load, resulting in the decoupling of load motion from ship motion and, therefore, reduced variation in cable tension. The past 40 years have seen heave compensation systems to become commonplace in many maritime operations. Fig. 1 provides a timeline of the major developments within the field of heave compensation.

Southerland (1970) presented a paper outlining the difficulties in payload handling at sea. Focusing on sub-sea salvage, recovery, and rescue operations, Southerland states that the most significant hurdle to these operations comes from surface ship motion in rough seas. He goes on to present examples of both passive and active heave compensation systems to alleviate the issue. The

* Corresponding author.
E-mail addresses: jwoodacr@dal.ca (J.K. Woodacre), robert.bauer@dal.ca (R.J. Bauer), rishad.irani@rolls-royce.com, rirani@dal.ca (R.A. Irani).

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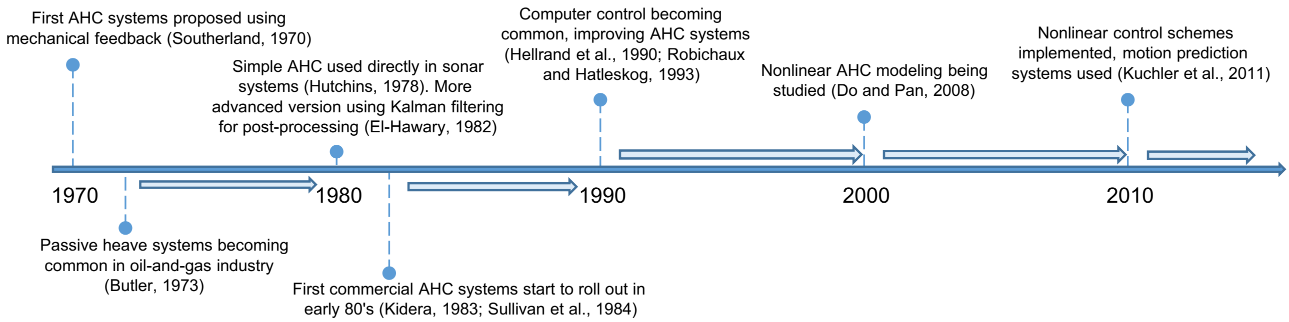

Fig. 1. An approximate timeline of heave compensation development (Hellrand et al., 1990; Sullivan et al., 1984).

passive system is designed to maintain a constant line tension, while the active system uses a simple mechanical feedback system to adjust for the ship heave amplitude.

Not long after Southerland (1970) was suggesting that heave compensation be used in handling operations, a study by Butler (1973) demonstrated a heave compensated drill string prototype being tested for offshore drilling. These tests showed successful isolation of the drill string from ship heave motion, resulting in longer operational windows and increased profits. The success of these and other similar tests allowed heave compensation to become widely accepted in the drilling industry, leading to further research and development.

Since the 1970s heave compensators have been benefited from computational advances allowing advanced sensor integration and better system modeling, hydraulic advances allowing faster and more accurate control, and control system advances allowing the application of more evolved control algorithms. These developments have largely been applied to heave compensation systems related to the oil and gas industry; however, both active and passive heave compensation are also prevalent in remotely operated vehicle (ROV) operations such as is seen in the work by Nicoll et al. (2008), as well as payload transfer between vessels as shown in an early patent by Blanchet and Reynolds (1977).

The current authors have found a great deal of literature on the subject of heave compensation systems; however, the works are spread through multiple sources such as journals, conference proceedings, theses, and patents with no extensive review of this increasingly important field being published. It is therefore the major contribution of this paper to provide a review of vertical heave compensation systems within a single, comprehensive study. First, in Section 2, a detailed explanation and comparison of active and passive compensation techniques will be provided including a brief history of each technique, current applications, as well as a discussion of their advantages. Following heave compensation, a discussion of wave synchronization methods is provided in Section 3, as wave synchronization is a closely related field. Next, Section 4 examines methods of actuation as they apply to heave compensation. Section 5 of this review paper looks at control theory as it is applied to heave compensation and what issues exist in current systems. Finally, the present authors conclude by summarizing the current state of the art and by proposing a new control method for use in heave compensated systems.

2. Heave compensation

Heave compensation can be divided into two main categories: passive heave compensation (PHC) and active heave compensation (AHC). Additionally, hybrid active–passive systems exist which combine features of both passive and active systems. Regardless of the compensator type, the goal of heave compensation is to decouple load motion from ship heave motion. In Sections 2.1–2.3 a basic functional description of different heave compensator implementations will be given.

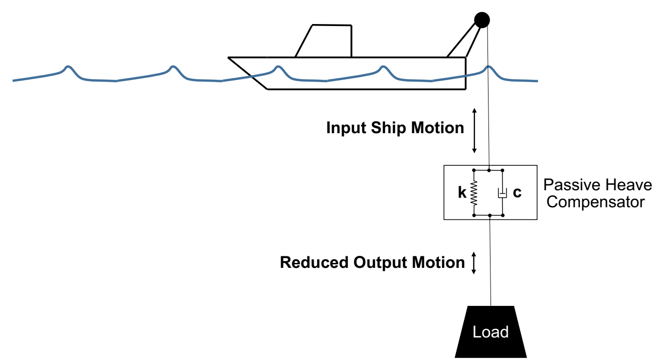

Fig. 2. This schematic shows an example of a small vessel hauling a load using a passive heave compensator in line between the load and the vessel.

2.1. Passive heave compensation (PHC)

At their simplest, PHCs are vibration isolators; open-loop systems, where the input is ship motion and the output is a reduced amplitude motion of the attached object, partially decoupling the load from the vessel. PHCs require no input energy to function. In Fig. 2 a simplified PHC is represented as a parallel spring–damper system placed at the center between crane and load — although the compensator can be placed anywhere on the load-carrying line, including on the deck of the ship.

The theory of vibration isolation is well established in many textbooks and the reader may refer to the literature by Inman (2001), Rao (2010), and Wow (1991) for a few such examples. In most vibration isolation systems, a parallel spring–damper is placed in series before the load which the designer wishes to isolate. The parallel spring–damper acts as a mechanical low-pass filter in which different values of spring-constant k, and damping c, produce a different low-pass filter corner frequency. Consider the system in Fig. 2 which shows a small surface vessel using a PHC, consisting of a parallel spring–damper, to help isolate the load motion from the vessel motion. The following differential equation can be written to describe the load motion:

$$m_L\ddot{x}_L = -k(x_L - x_H) - c(\dot{x}_L - \dot{x}_H), \tag{1}$$

where x_H is the ship heave, x_L is the load displacement and m_L is the load mass. Taking the Laplace transform of Eq. (1) results in

$$m_L s^2 X_L(s) = -k(X_L(s) - X_H(s)) \\ - c(sX_L(s) - sX_H(s)), \tag{2}$$

which, after rearranging Eq. (2) for X_L/X_H, becomes

$$\frac{X_L}{X_H} = \frac{cs+k}{m_L s^2 + cs + k} \tag{3}$$

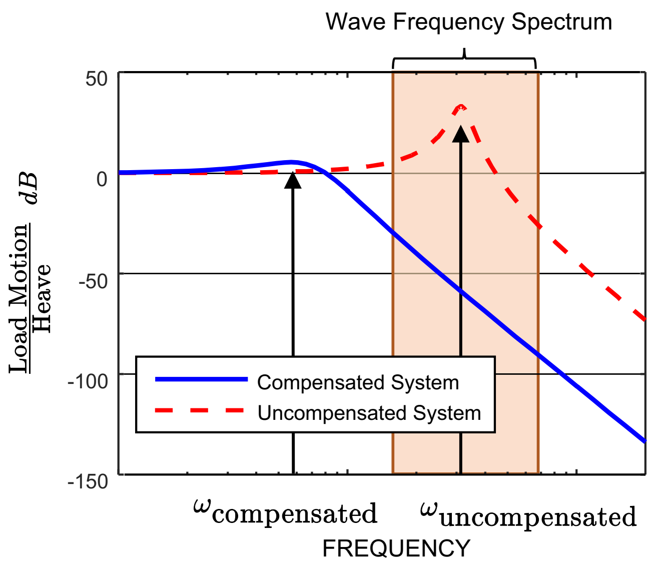

Fig. 3. These Bode diagrams show an uncompensated (or poorly compensated) system operating within the wave spectrum, with a compensated system attenuating motion in the ocean wave spectrum.

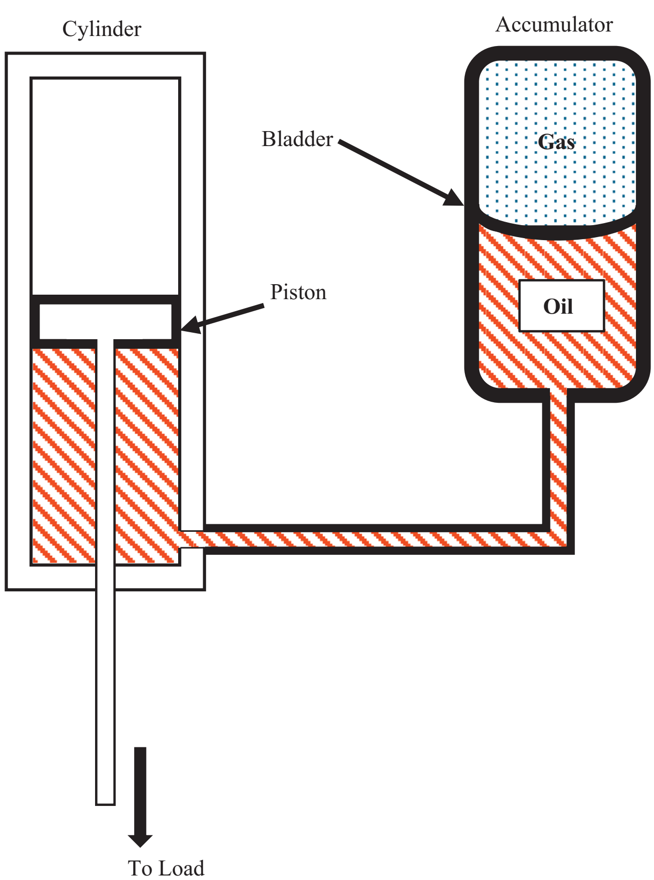

Fig. 4. This schematic shows a gas-backed, hydraulic piston passive-heave compensator. The gas is pressurized to hold a desired load and the bladder separates oil from gas while equalizing gas pressure and oil pressure. The pressurized oil holds the load by pressing on the piston.

For the second order system described by Eq. (3), the corner frequency (damped natural frequency) ω_d will occur at

$$\omega_d = \omega_n \sqrt{1 - \left(\frac{c}{2\omega_n}\right)^2} \tag{4}$$

where the undamped natural frequency ω_n is given by

$$\omega_n = \sqrt{\frac{k}{m_L}} \tag{5}$$

Heave amplitudes occurring with a frequency above ω_d will begin to be attenuated at the load, suggesting the goal should be to design a compensator such that ω_d occurs well below the expected frequency range of the ocean waves (and, therefore the ship motion).

Fig. 3 plots the Bode diagram of the transfer function given in Eq. (3) for two systems: an uncompensated system and a compensated system. In the uncompensated system the spring constant is dominated by cable properties. In Fig. 3, the uncompensated natural frequency $\omega_{uncompensated}$ occurs within the input wave spectrum meaning that, for an uncompensated system, heave motion would be amplified at the load. For a compensated system, it is desired that ω_d occurs below the expected range of input frequencies, meaning that the designer should choose k such that ω_d occurs at position $\omega_{compensated}$ as shown in Fig. 3, which successfully attenuates motion at the load. Tuning of the compensator is mainly performed by adjusting the spring-constant k, as damping tends to be difficult to control.

It is common to use some variation of a gas-backed accumulator driven hydraulic piston as a spring for passive compensation. Examples of this gas-backed accumulator design can be found in a PHC system design by Huster et al. (2009), as well as patents by Bolding and Person (1976), Ormond (2011), and Kammerer (1964). Fig. 4 shows a simplified schematic of a gas-backed hydraulic piston accumulator. The accumulator is charged with pressurized gas on one side of a bladder. Gas pressure is set to hold the load at steady state while a bladder separates the gas from hydraulic oil. The hydraulic oil is at the same pressure as the gas, holding the load by pushing on the piston in the cylinder shown.

A strictly pneumatic passive compensator was fully treated mathematically by Stricker in his 1975 thesis (Stricker, 1975). Jordan (1987) suggests that strictly pneumatic systems are not commonly used because a cable break would result in rapid cylinder motion potentially causing damage, whereas in a hydraulic system cylinder motion is limited by the oil flow-rate into the cylinder. In 1976, Woodall-Mason

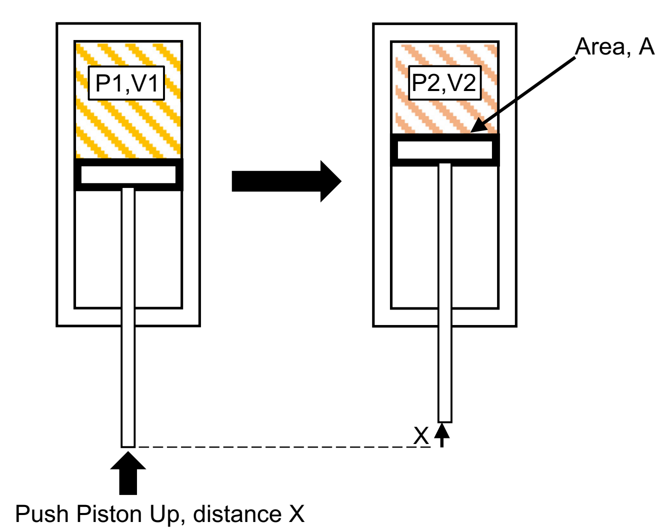

Fig. 5. This Figure shows a gas at pressure P_1 and volume V_1 being compressed to pressure P_2 and volume V_2, by moving the piston at a distance x.

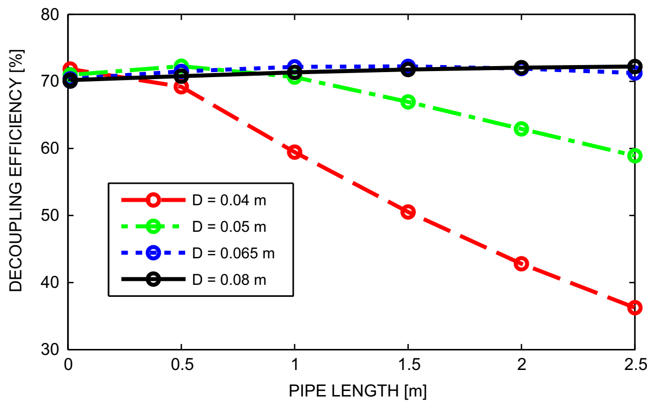

Fig. 6. A plot summarizing the pipe sizing data by Ni et al. (2009).

and Tilbe (1976) published work reviewing the use of compensation systems and found pneumatic systems were in fact used, and fast-closing valves were used to limit actuator motion in case of a cable break. Woodall-Mason and Tilbe (1976) instead suggest that pneumatic systems are at a disadvantage, since, they must be mechanically locked into place while a hydraulic system can close a valve leading to the actuator, allowing fluid lock to hold the actuator in place.

In systems using a passive compensator, total damping is defined by the system components such as submerged cable length, drag due to load geometry, and mechanical friction. The spring constant k for the passive compensator is set based on gas accumulator volume. To illustrate the k dependance on volume, start by looking at the isothermal process in Fig. 5, where P is pressure, V is volume, x is displacement, and A is piston area. Making the assumption that $P_1V_1^n = P_2V_2^n$, with n being the associated gas constant, and $V_2 = V_1 - \Delta V$, where ΔV is the volume change, then

$$\begin{aligned} P_2 &= P_1\left(\frac{V_1}{V_2}\right)^n \\ &= P_1\left(\frac{V_1}{V_1-\Delta V}\right)^n \\ &= P_1\frac{1}{\left(1-\frac{\Delta V}{V_1}\right)^n} \end{aligned} \tag{6}$$

Subtracting P_1 from both sides of Eq. (6) and using the identities, $\frac{1}{(1-z)^n} \doteq 1+nz$ for small z, $P_2 - P_1 = \Delta P = \Delta FA$, and $\Delta V = xA$, yields

$$\begin{aligned} P_2 - P_1 &= P_1\left(1+n\frac{\Delta V}{V_1}-1\right) \\ \Delta P &= P_1\frac{n\Delta V}{V_1} \\ &= P_1\frac{nxA}{V_1} \end{aligned} \tag{7}$$

Multiplying both sides of Eq. (7) by A gives the final result

$$F = \frac{nP_1A^2}{V_1}x \tag{8}$$

where F is the force due to the pressure change created by the piston motion x. In Eq. (8), the force is not defined in a particular direction and simply pushes out on all sides of the volume, V_2. Comparing Eq. (8) to Hooke's Law $F = -kx$ and following the convention whereby the force must oppose the motion, x, results in a value of k, such that

$$k = \frac{nP_1A^2}{V_1} \tag{9}$$

which shows increasing V_1 softens the spring.

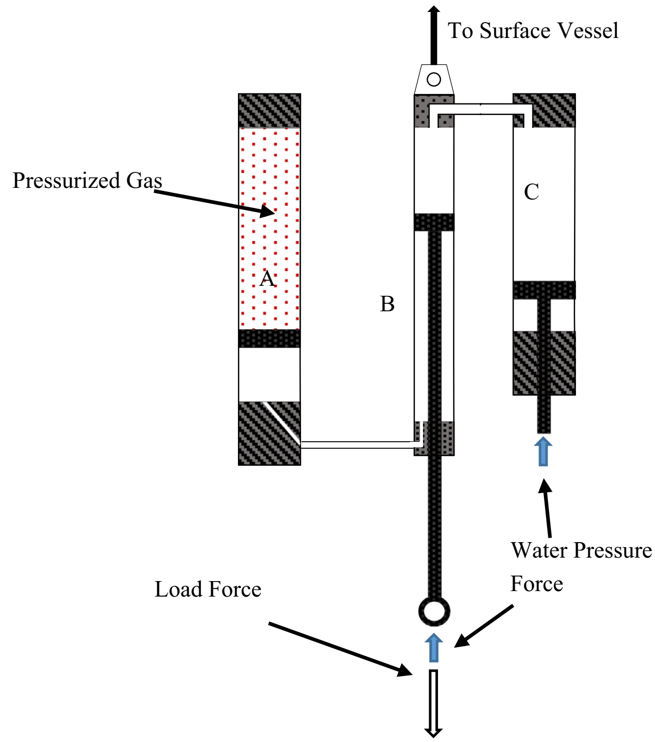

Fig. 7. A depth-compensated passive heave system. Cylinder sizes are not to scale for a working PHC.

In cases where ship motion is larger than the compensator stroke, it can become necessary to increase k, stiffening the system which can reduce the amplitude of compensator motion, or lock the compensator entirely (Driscoll et al., 1998). Stiffening or locking the system ensures that snap loading does not occur when the compensator suddenly hits a hard-stop, as could happen in situations when heave motion is larger than the compensator range of motion.

Generally, increasing PHC accumulator gas volume will improve the ability to decouple the motion; however, simulations by Ni et al. (2009) show diminishing returns on increasing compensator gas volume indefinitely. They found that eventually system performance becomes dominated by the size and length of the pipe attaching the accumulator to the compensation cylinder. In Fig. 6 the reader is shown a plot of passive compensator decoupling efficiency versus length of pipe between the accumulator and the compensator cylinder, where decoupling efficiency is defined as the normalized reduction in motion between ship and load. Decoupling efficiency is plotted for four different diameters D of pipe. For the smallest pipe diameter, compensator effectiveness is reduced as much as 36% as length increases since, in a smaller diameter pipe, fluid drag becomes very significant for higher flow rates. It is interesting to note that in the 0.08 m diameter pipe, increasing pipe length actually increases effectiveness. The reason for increased effectiveness was not given and could be the focus of further work.

In a PHC system initial accumulator pressure is user set to hold the steady-state load. For a crane or winch, the total load also includes the weight of cable holding the payload. At sufficient depth, the cable mass can dominate a load and cable resonance may also become involved, creating motions larger than that of the ship. Driscoll et al. (1998) treat cable resonance effects in simulation and determine that a compensator mounted at depth, near the load, provides more effective motion decoupling. The downside to a compensator near the load is that operating depth and load need to be known in advance to tune the compensator, and tuning cannot be changed in situ. If the load changes significantly during operation, then a ship based passive compensator should be used. The problem of tuning for depth may be solved by using a depth-compensated passive heave module (Ormond, 2011).

A schematic of a depth compensated system is shown in Fig. 7. As depth is increased, water pressure pushes on the bottom of the rod extending from cylinder B. This water pressure directly opposes the load force, which effectively reduces the load for which the system was tuned (where the system was tuned by pressurizing gas in cylinder A). Cylinder C is added to the system to compensate for the force due to the increased water pressure. As water pressure increases, a force develops on the rod extending from cylinder C. This force pressurizes the fluid in cylinder C, directly pushing on the top of the rod in cylinder B, opposing the force developed at the bottom of cylinder B's rod. Thus, the contribution of forces due to water pressure cancels each other out.

It is an advantage of passive compensators to be capable of being added to existing uncompensated systems allowing increased operational capabilities in rougher seas. Huster et al. (2009), for example, designed a system to retrofit into an existing ROV launch-and-recovery system. Lab testing of the Huster et al. (2009) system shows a motion reduction of 68%. In operation, the authors claim 90% motion reduction in 2 m seas; however, no corroborating data is shown. Furthermore, a paper by Hatleskog and Dunnigan (2006) concludes by stating that a passive compensator can be no more than roughly 80% effective which the authors support through field experience and simulation. It is important to note that the present authors were unable to find published experimental data on the effectiveness of passive compensators, with the exception of Huster et al. (2009). A possible explanation is the high cost of hardware involved. Companies who have the capital to test and produce passive compensators likely keep these results internal to the company and their customers.

According to Kidera (1983) many early passive systems suffered from the problem of cylinder stick, where static friction was too large for the load to overcome easily. Kidera does, however, go on to report that one system he had surveyed at the time had a static friction break-away force of approximately 15 lbs while being able to carry 4000 lbs. In the current authors' survey of published works, no studies could be found which analyze the effects of nonlinear friction in hydraulic cylinders with respect to PHCs. Breaking the initial “stiction” to start moving the cylinder would require some amount of force, depending on the system size. If a system is improperly sized, the load may not be large enough to break this friction force.

Hatleskog and Dunnigan (2006) considered the passive compensator dynamics for an oil drilling platform and one of the key conclusions they made was that, in the real-world, the only way to reduce heave motion coupling to the load by over 80% is by using an active compensator. Hatleskog and Dunnigan mention that the desire to reduce heave motion coupling further was one of the driving forces behind the development of active heave compensation in the 1990s. Additionally, passive compensators are ineffective in applications such as payload transfer from ship-to-ship or in wave matching when transitioning a load from air to water. In the cases of payload transfer and wave matching, PHCs are unable to compensate for relative motion between two independently moving references. For these applications, an active heave compensator must be used. In the next section we will discuss active heave compensation and how it can improve motion decoupling when compared to passive systems.

2.2. Active heave compensation

Contrasting the open-loop passive systems, active heave systems involve closed-loop control and require energy input. In an active system, ship heave motion is measured and relayed to a controller, which then moves an actuator to oppose the heave motion. So, if a ship heaves upward, the controller commands the load to move downward that same amount. For an active system, one of the greatest advantages is that the feedback variable is not limited to ship heave motion. Feedback can, for example, be based on the separation between two ships such as is used during payload transfer, or it can be a measured force from a load cell used to maintain a constant tension in the cable at all times. Feedback can also be based on wave height which is most often used when a load transitions from air to water. Wave height feedback is discussed in Section 3 which specifically covers wave synchronization.

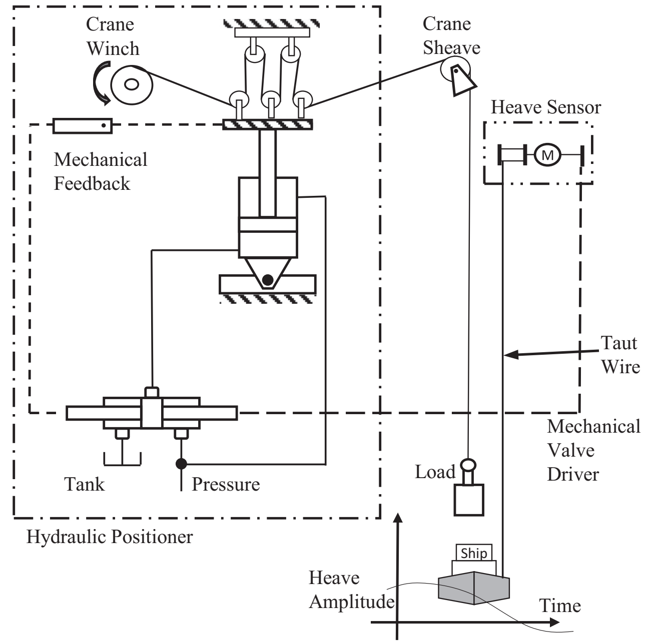

Fig. 8. This system was presented by Southerland in 1970 as a method to transfer payload from ship-to-ship in the presence of significant waves. Figure reproduced from Southerland (1970).

One of the first active heave systems was shown by Southerland (1970) where a spring-loaded tether was attached from a crane-boom on one ship to the deck of a second ship. A schematic of this system can be seen in Fig. 8. As the tether was pulled in and out, it moved a hydraulic proportional valve which adjusted the load, maintaining a constant height from the deck. The system shown in Fig. 8 was fully integrated into the crane operation. A similar mechanically actuated system was patented in 1977 (Blanchet and Reynolds, 1977) but the system was packaged for retrofit onto cranes which were not heave compensated and could be hung from the crane, between the crane and the load.

Little published work is found between 1980 and 1990 on mechanical AHC systems – likely because this time period occurred before real-time computer control was mature enough to integrate into a complicated system. Furthermore, in the 1980s passive systems were generally sufficient for the oil and gas industry, which were one of the main driving forces for initial heave compensation research. A patent by Barber (1982) does show a circuit based AHC system where heave motion was sensed and a fixed circuit design was implemented to control heave motion, but a downside of the fixed circuit is that it cannot be changed. If control scheme changes need to be implemented it would require rework of the circuit board. So, although published works were sparse in this time period with respect to mechanical AHC systems, work on heave compensation theory and algorithm development did continue in the sonar field.

A patent by Hutchins (1978) shows how a simple double-integrator circuit was used to convert accelerometer data into vertical motion data as part of a towed sonar array control circuit. In this case, the sonar array was used for mapping the ocean bottom. Having vertical position data allowed the sonar array to adjust the sonar pulse timing, effectively correcting for vertical motion on-board and demonstrating an early example of transitioning from mechanical feedback to electronic feedback in an AHC system (before computer control became dominant).

An improved method of correcting heave in sonar data was presented by El-Hawary (1982). The author analyzed sonar data using Fast Fourier Transform (FFT) analysis to determine the frequency components of ship heave and, through application of an optimized Kalman filter, was able to selectively remove heave motion in post-processing while retaining the ocean bottom profile. Due to the computation power required, analysis could not be applied in real-time at the time of publication.

A patent granted to Jones and Cherbonnier (1990) is one of the first examples the present authors could find of a microprocessor controlled AHC system. As it is a patent, details on the control method are limited; however, a patent by Robichaux and Hatleskog (1993) does suggest that the benefits of a microprocessor come mainly from adaptability. With mechanical hardware in place, the control parameters or control method can be changed by uploading new software to the controller. Operators could easily adjust control parameters on-the-fly, accounting for a wide range of loads or ocean conditions. The ability to modify software would be significantly less expensive than hardware changes, while also broadening the use of the control system so that it could potentially be used on large oil rigs, or adapted for smaller vessels which may want to use AHC for remotely operated vehicles. Software could also be written for accepting different sensor inputs depending on the AHC application which is appealing to users who may have multiple uses for an AHC system.

When drilling at sea, there are a number of drilling vessel types — either floating or fixed in place — performing drilling operations at a range of depths. In the case where a vessel is floating, it is important to remove vessel heave motion from the entire drill string, where drill string is a term which often describes the entire drilling system from the ship down to the drill bit. Removal of heave motion from the drill string extends operational time and reduces fatigue on the drill and riser (Korde, 1998). Korde (1998) performed an in-depth mathematical treatment of an AHC system used to stabilize the drill string for a drill ship. In his system, accelerometer data was used for position and force feedback in an active position control system as well as an active vibration absorber. A more in-depth discussion of the system by Korde (1998) will be performed in Section 5; however, note that simulation results show that the system is able to fully decouple motion using a linear model. Do and Pan (2008) applied a nonlinear model and control scheme to actively compensate for heave motion in a similar drill string system to that which was examined previously by Korde (1998). In using a nonlinear model, Do and Pan (2008) were unable to fully decouple ship heave from the drill string suggesting that using a linear system model may be too simplified to capture the full system dynamics.

Requiring more than simple acceleration measurements, modern systems often use an inertial measurement unit (IMU), also called a motion reference unit (MRU), to determine ship motion in real-time. Using 3-axis accelerometers and gyroscopes an IMU determines ship motion based on algorithms similar to those presented by Godhaven (1998). Marine IMUs tend to be expensive to purchase, thus a promising low-cost GPS based alternative for measuring heave was presented in a paper by Blake et al. (2008). Preliminary results show that heave measurements with their device are comparable to those obtained from an IMU; however, the sampling rate of the GPS is limited to below 4 Hz which could be a concern when implementing high-speed control algorithms.

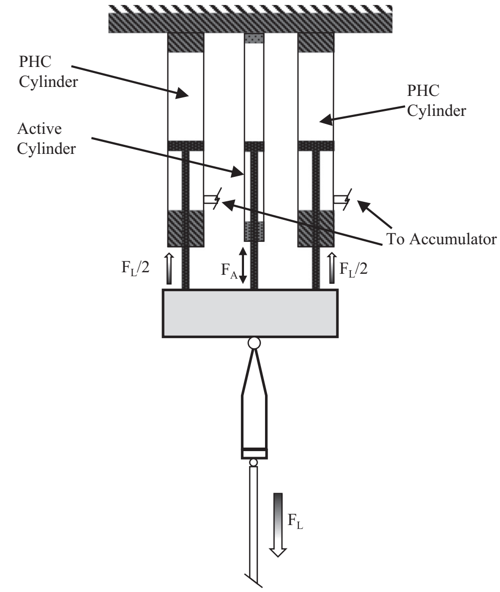

Fig. 9. This schematic illustrates one possible example of a hybrid heave compensator. The larger, passive cylinders hold the load weight while the active cylinder applies adjustment forces based on an active control strategy.

Such control algorithms in an active heave system can be as simple as basic PID and pole-placement control, or as advanced as systems using Kalman filtering and observers to include complicated features like tether dynamics as part of the control scheme. In any control system, corrections for the inherent lag, perhaps introduced by the hydraulic system or through slow communication between the IMU and the control system, must be made to ensure ideal control. A system by Kyllingstad (2012), for example, applied transfer function filters to correct for time/phase lag in their overall system. Alternatively, Kuchler et al. (2011) used heave-prediction algorithms to predict vessel heave motion based on previous measurements and then applied control action based on these predicted motions. Now, as more advanced algorithms and better sensors are included in AHC systems, control quality improves; however, there are disadvantages to the inclusion of more advanced components.

For an active system, electronics, sensors, and controlled actuators are all involved, increasing design and production cost as well as potentially introducing the need for specialized training for troubleshooting and repair. In a passive heave compensator, feedback and control systems are not necessary, making troubleshooting a relatively easy task due to the simple nature of the system. With a strictly active system, not only can the system be difficult to troubleshoot, but additionally the potentially significant power requirements must be considered. An active system requires actuators powered either hydraulically or electrically and requires maximum power to be available to the actuator at all times to ensure that the system operates as expected. If power delivery is a limiting factor, an active–passive hybrid system may be an option as it allows active compensation without the need to actively hold the full load.

2.3. Active–passive hybrid system

A hybrid system, such as that shown schematically in Fig. 9, has both active and passive cylinders. Fig. 9 illustrates a system with two passive cylinders each holding half of the total load weight F_L, with a third, smaller cylinder being part of an active control loop which can apply an additional adjustment force, labelled F_A. The active cylinder needs to be capable of moving at the maximum load speed; however, since the active cylinder will generally apply much smaller forces than those experienced by the passive cylinders, it can be physically smaller requiring less flow, less pressure and, therefore, less power than compared to a strictly active system.

A hybrid compensator design for a drill string presented by Hatleskog and Dunnigan (2007) combines a passive system to hold the bulk of the load, and an active system to assist in further load motion decoupling from vessel heave. In their report, a hybrid system designed to passively hold a 1,000,000 lbf load required an actuator capable of providing only 100,000 lbf for the active compensation portion. Robichaux and Hatleskog (1993) also patented a very similar system in 1993.

Nicoll et al. (2008) simulate attaching a passive heave compensator near the load, with an active system operating at the surface. Although their results show reduced load motion and cable tension compared to the active or passive systems separately, this system requires the active system to hold the entire load and, if adjustments are required to the passive system, it must return to the surface.

In much of the previously mentioned work a controller is used to compensate for ship motion, holding the load steady with respect to a heaving ship or platform. In Section 3, wave synchronization is examined, where the systems allow the load to follow surface wave motion as the load transitions from air to water. In essence, a wave synchronization system operates in a very similar way to an AHC system.

3. Wave synchronization

Driven by the Oil and Gas Industry attempting to increase operability in harsh ocean conditions, wave synchronization became the research focus for a number of groups starting in the early 2000s. In wave synchronization operations, consideration is given to load interaction at the air–water interface. During the transition from air to water the load is subject to potentially large hydrodynamic forces, or “slamming” forces, which can cause serious damage to the load or can lead to a cable break (Messineo and Serrani, 2009). These hydrodynamic forces are not directly accounted for in a standard AHC system. In 2002, Sagatun (2002) presented a controller to minimize the dynamic forces acting on the load as it transitions from fully in the air to fully submerged. Sagatun's (2002) controller used position and velocity feedback coupled with mechanical and hydrodynamic models to create a time-variant trajectory for the load to follow while transitioning from air to water. In simulation, the controller was able to reduce the largest acceleration seen by the load by 50%, which occurred when contact was first made with the water. Despite this reduction, acceleration felt by the load when first contacting the water still exceeded the maximum acceleration felt over the rest of the operation.

Johansen et al. (2003) published work implementing a feed forward controller which utilized wave height measurements to estimate a control trajectory that would minimize the hydrodynamic effects on the load. Also included in their controller was an AHC system which attained heave compensation through double-integration of an accelerometer attached to the ship. Using a scale model of an at-sea crane, the authors were able to experimentally achieve a reduction in the cable tension standard deviation of 22% in 1.8 cm waves, and up to 54% in 6.8 cm waves. Johansen et al. (2003) state that their controller performance could be increased further by using a short-horizon predictive controller to reduce their 37° phase error which was caused by filtering and the motor itself.

A publication by Skaare and Egeland (2006) proposed a parallel force/position controller for wave synchronization which did not directly measure wave height. Mirroring a control scheme which is often used in robotics, Skaare and Egeland employed a controller in which position control dominates for high frequency motion, while force control dominates for low frequency motion. Skaare and Egeland compared their control scheme to that used by Johansen et al. (2003), finding that the parallel force/position controller showed improved performance in all cases with respect to ensuring the cable did not lose tension and become slack. The authors also performed water entry simulations with their parallel force/position controller, an AHC controller, and a wave synchronization controller. In both the AHC and wave synchronization simulations the authors found that some operations resulted in zero cable tension or slack line conditions. Slack line conditions are very dangerous at sea as they can lead to the cable catching on equipment or personnel, causing damage to equipment and potentially life threatening injuries. Additionally, tension could be suddenly reestablished when the load drops faster than the compensator provides cable, resulting in a snap load and potentially breaking the cable completely.

Inspired by the work of Johansen et al. (2003), Messineo et al. (2008) designed a combined wave synchronization and AHC controller using feedback control instead of feed forward control as was used by Johansen et al. (2003). When compared to the feed forward controller, the feedback controller leads to a smoother cable tension change when transitioning from air to water as well as a reduction in cable tension standard deviation from 0.23 to 0.15 N once submerged. The feedback controller by Messineo et al. (2008) was further improved by Messineo and Serrani (2009) through the inclusion of an adaptive external disturbance estimator as well as an adaptive observer to estimate uncertain model parameters. Compared to the controller by Messineo et al. (2008), a 50% reduction in the standard deviation of cable tension and a 14% decrease in hydrodynamic forces acting on the load were realized by the adaptive controller.

Coupled with superior performance and an overall smoother air–water transition, the adaptive controller by Messineo and Serrani (2009) seems to be an ideal candidate for a full-scale design; however, when scaling up a design, consideration must be given to the components to be used. As an example, Messineo and Serrani's (2009) small-scale system used an electric motor to actuate the load vertically, but in a large system the load may be several tonnes, and a hydraulic actuator may be preferred. In Section 4, consideration is given to the different actuators which could be used in heave compensation systems.

4. Actuation

Primary actuation of most heave compensation systems is delivered by either hydraulic or electric drive systems. Although passive systems use a pneumohydraulic system, they are not strictly pneumatic due to the need of an additional brake to hold a pneumatic system in place as well as the increased damping introduced by the hydraulic fluid to smooth out the resulting motion.

4.1. Electric

An article in Offshore Magazine (1999) mentions that alternating current (AC) driven heave compensation systems were introduced in the early 1990s. Electric heave compensation systems have increased in popularity due to their relatively high efficiency (estimated between 70% and 80% peaks) (Angelis, 2009) attributed to efficient control and motor systems as well as regenerative

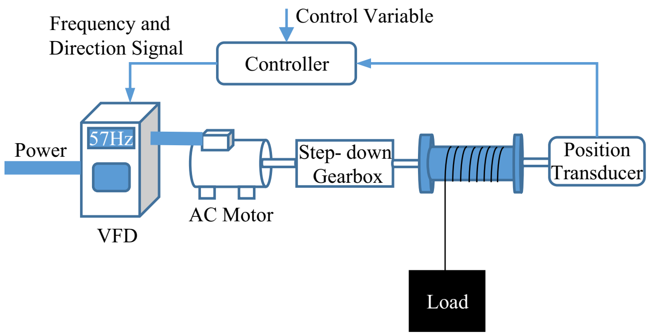

Fig. 10. This diagram shows a simple AC drive winch system with feedback control.

techniques used during braking (Kang, 2013). Lack of an oil reservoir and low motor noise when compared to hydraulic systems is also appealing to consumers (Angelis, 2009) who may not want to deal with oil replacement or potential leaks.

High power electric AC motors tend to be physically large, having a correspondingly large moment of inertia. A large inertia means large torques are needed to change motor speed when responding to transient behavior. In some situations it could be that, when changing speed, it is the motor inertia which dominates the required power and not the load itself.

The active heave system shown in Fig. 10 uses an AC electric variable frequency drive (VFD), AC induction motor or motors, gearbox, sensor feedback and control system, as well as a braking system and potentially a cooling system. In an AC induction motor the motor speed is directly proportional to the supplied AC voltage frequency as described by the equation

$$\omega_m = \frac{120f}{p} \tag{10}$$

with ω_m being motor speed in revolutions per minute (RPM), f being the AC voltage frequency in Hz, and p being the number of motor poles. A VFD creates an AC voltage signal where the user may adjust the output frequency to drive the AC motor at an angular velocity as described in Eq. (10).

If multiple actuators are needed or multiple winches are to be installed, then the entire system must be replicated in full for each actuator as shown in Fig. 11 where the system from Fig. 10 has been replicated three times to create a multiple winch system. Replication of the full system is not ideal because the AC motors are large when compared to an equivalent power hydraulic motor. As an example, the Marathon Electric E213 100 horsepower electric motor weighs 1220 lbs (Marathon motors product catalog, 2013), while the hydraulic Bosch-Rexroth MCR20 110 horsepower motor weighs 167 lbs (Radial piston motor, 2012).

The first alternating current electric AHC systems were likely powered by a VFD known as a scalar VFD. A scalar VFD maintains a constant voltage to frequency ratio to correct for reduced motor impedance at lower frequencies. A reduced impedance means that a lower voltage is required to maintain equivalent current and, therefore, torque. Scalar VFDs could lose torque during rapid speed changes forcing designers to oversize both the physical system and the power system (Godbole, 2006). Systems using a scalar VFD can provide their designed torque at a constant low speed (Parekh, 2003); however, for high-torque low-speed applications additional cooling is generally required for the motor since most AC motors rely on a fan directly connected to themselves to provide cooling. Additional cooling can be achieved by the addition of an externally driven fan or through fitting of the AC motor with an encasement and providing a water cooling system — both of which increase the total cost.

Modern VFD systems can now use vector control, also called field-oriented control, which more efficiently controls power delivery to the motors, resulting in better control and reducing the need to oversize motors (Godbole, 2006). Vector control also integrates regeneration into the electronics, allowing energy capture when decelerating, thereby increasing system efficiency. A current issue with energy capture in VFDs is storing the energy because if power is pushed into a ship's electrical grid when it cannot be used, this excess power may disrupt other systems. Battery or capacitor bank storage is, therefore, needed which increases cost due to increased weight as well as additional storage space requirements.

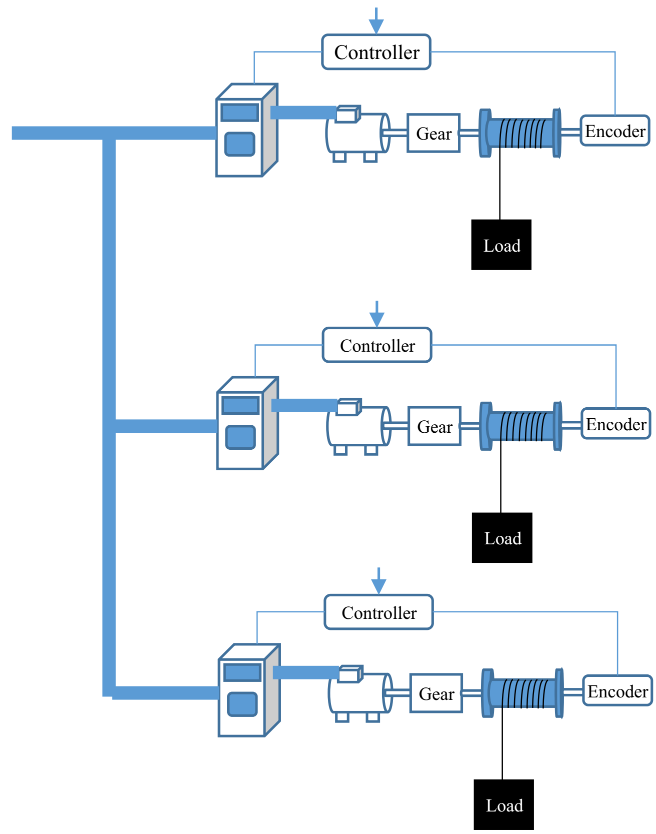

Fig. 11. This diagram shows how multiple AC electric winches would require full duplication of the system in Fig. 10.

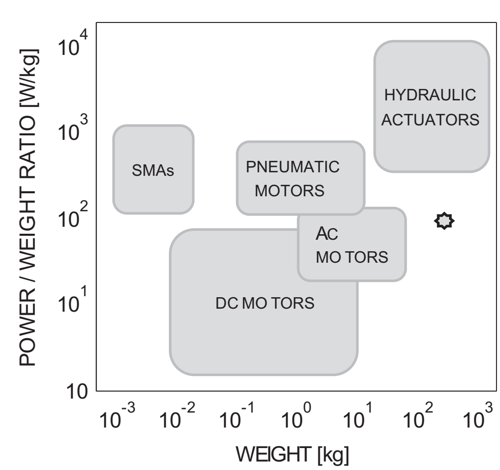

Fig. 12. This figure, reproduced from Nespoli et al. (2010), shows that hydraulic systems can provide a higher actuator power density than AC drives. Here, SMA is short for shape-memory alloy.

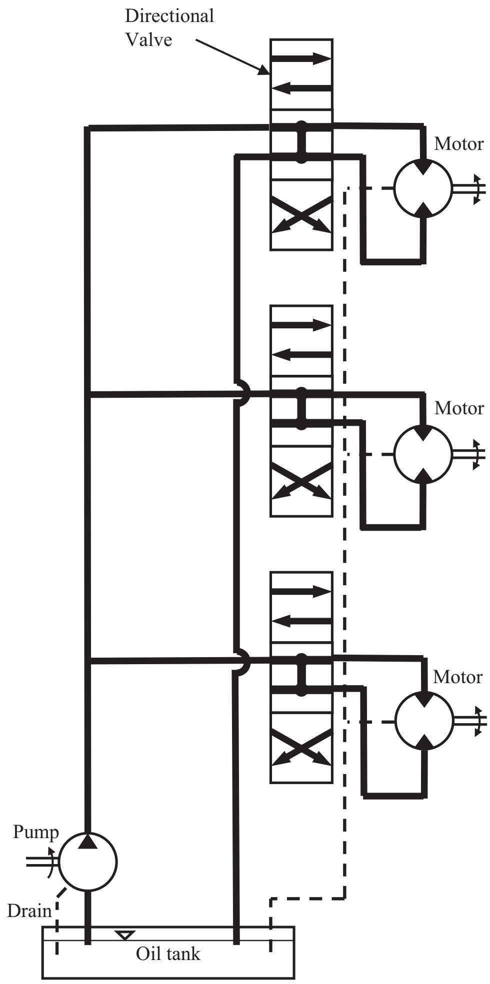

Fig. 13. A single hydraulic pump can operate multiple motors; however, care must be taken if trying to operate each motor at the same time.

Computer models of hydraulic driven AHC systems exist (Gu et al., 2013; Ayman, 2012) and are used when evaluating hydraulic AHC performance; yet equivalent AHC systems modeled using an electric drive could not be found in the literature. This lack of full system modeling constitutes a gap in AHC research that should be examined further.

4.2. Hydraulic

Hydraulic systems are well established in the marine industry. Hydraulic systems can be used for anything from opening large doors on a marine vessel to a simple winch on a fishing boat. As shown in Fig. 12, hydraulic actuators provide the highest power to weight ratio of any actuator currently on the market (Nespoli et al., 2010). This figure is incomplete, however, because larger weight AC motors are not included. For example, the Marathon Electric E213 100 HP motor mentioned previously would appear at the star to the right of the AC motor block in Fig. 12.

The high power to weight ratio of hydraulic motors allows the actuator to maintain a small footprint at the point of actuation which can be appealing when deck space is limited. The downside to using hydraulic actuators is that a hydraulic power unit (HPU) must be placed somewhere aboard the ship. These HPUs can be large depending on the loads in question; however, it should be noted that one HPU can operate multiple actuators as shown in Fig. 13. In Fig. 13 each motor can be operated independently by operating their respective directional valves.

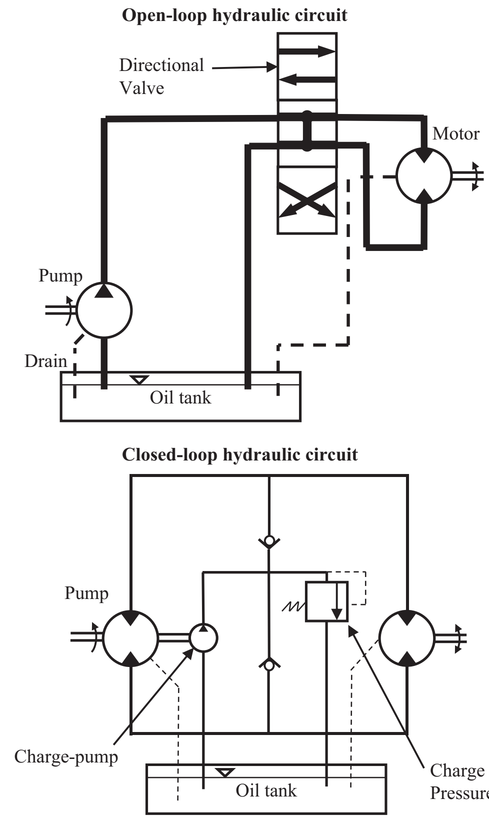

Fig. 14. This figure shows a simple open-loop hydraulic system (top), and a closed-loop hydraulic system (bottom) operating what could be a winch motor.

As mentioned, hydraulic systems are a well known and widely-used technology in the marine industry. Parts can be readily available so troubleshooting and repair of a hydraulic system can often be done quickly. In contrast, troubleshooting of electric systems can be more difficult and require specialized electrical training (Angelis, 2009).

Fig. 14 demonstrates two simple hydraulic circuits operating a motor. The upper circuit is an open-loop circuit, where fluid from the pump is regulated by a directional-valve as it travels to a motor, performs work, and returns to the open-air reservoir. The lower circuit in Fig. 14 is known as a closed-loop circuit as fluid is regulated by the pump itself, travelling directly to the actuator, then returning to the pump. In a closed-loop system, the pump is able to provide flow in both directions, whereas an open-loop pump only provides flow in one direction.

In an open-loop system, and hydraulic systems in general, the most significant downside is low efficiency. Depending on the design and operation, some open-loop systems can have an average efficiency as low as 10–35% (Virvalo and Liang, 2001); however, efficiencies as low as these generally occur when operating a system far from maximum load. The lowest efficiency systems use a fixed displacement hydraulic pump delivering constant flow. Unused flow is diverted away from the load at significant energy cost, and a proportional valve controls how much useful flow is delivered to the motor. For a system which will only operate for short periods of time, a fixed displacement pump may be acceptable — trading efficiency for simplicity, low initial cost of hardware, and ease of maintenance. In larger systems or systems which may run for extended periods of time, inefficiency can be very costly; therefore, a variable displacement hydraulic pump is preferred. Variable displacement pumps only deliver fluid when needed — better matching the process requirements and avoiding losses from dumping excess flow away from the load. A

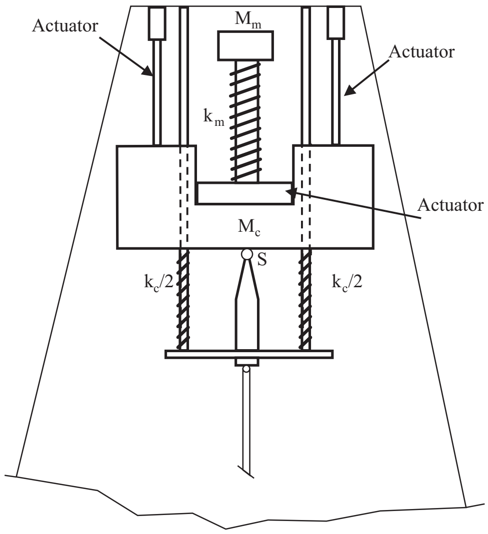

Fig. 15. Here, the actuated portion of Korde's (1998) system is shown. The harmonic absorber M_m and the support block M_c are shown with their respective actuators. Figure reproduced from Korde (1998).

proportional control valve is used to moderate flow delivered to the load. In systems using a variable displacement pump the most significant energy losses come from metering across the proportional control valve, and from pump and motor inefficiencies. These losses will be system specific and dependent on the stand-by pressure of the pump (where stand-by pressure is the pressure a variable displacement pump maintains when flow is not demanded). It would not be unreasonable to see efficiency numbers between 50% and 80% for a system using a proportional valve and a variable displacement pump. An alternative to having a proportional control valve is to use a closed-loop hydraulic system.

An efficiency of at least 80% can be realized in closed-loop systems (Jones, 2012). Further efficiency increases can be realized when variable speed control is included on the closed-loop pump — reducing mechanical losses when flow is not required. Increased efficiency is enticing for designers; however, a closed-loop system has increased cost as a dedicated pump and motor are both needed for each actuator to operate independently at high efficiency.

In closed-loop cases, actuator speed is linearly controlled by pump output instead of the nonlinear response found in most proportional control valves which simplifies the control system for AHC. Increased cost for a closed-loop system, however, means that proportional control valves are still commonly used and, as such, it is important to be able to model and control these valves and their systems accurately. In the next section, various control methodologies for active heave systems are examined.

5. Control

Using an AHC system, the goal is to actively remove as much of the ship heave motion as possible from the load or, in other words, to decouple ship motion from load motion using controllers and actuators. In 1970, one of the first AHC systems was presented by Southerland (1970) using proportional control with mechanical feedback in a payload transfer situation. Fig. 8 shows the mechanical feedback consisting of a tether attached from a crane tip on one ship to the deck of a second ship. Motion of the second ship resulted in the tether pulling in, or letting out, moving a hydraulic valve either pulling the load up or letting it down. The work did not give experimental results on how effective the system was.

A report by Bennett (1997) mentions that a system used in the North Sea was able to reduce motion of 6 to 7 ft swells down to less than a 2 in. motion based on visual inspection — which is a 95% reduction. They do not, however, mention the type of control used, simply labelling the controller as a "computer". The report by Bennett (1997) presented results of implementing an AHC system which was purchased from a supplier, so it is reasonable that they would not know or be able to present the type of control used. In this case, the company supplying their AHC system would be unlikely to reveal the control algorithm.

As mentioned, the work by Southerland (1970) presents a system idea and the work by Bennett (1997) presents final results of a system without details of the system itself. Often, if a group has funding to construct or purchase the experimental apparatus they may not want to fully reveal the design to protect their intellectual property. Due to the prohibitive cost in the construction of an experimental apparatus, much of the work found in the literature presents a design, or a design with simulated results only.

In a 1998 paper, Korde (1998) presented a full linear drill-string model and developed a control system using accelerometers and an actuated harmonic absorber. Fig. 15 shows the actuated part of Korde's system with the central actuator acting on M_m (the vibration absorber) while the other two actuators act on M_c (where M_c combines the mass of the drill string and the block holding the string to the actuators). Korde's system applies feed-forward control based on direct accelerometer measurements to control the vibration absorber, as well as double-integrating the accelerometer data for position control of both sets of actuators. This type of vibration absorber is similar to that used in multistory buildings to reduce seismic and wind vibration (Lee-Glauser et al., 1997). Theoretical results show that this system can fully decouple load motion from ship motion; however, the theoretical full decoupling results are based on idealized calculations and the author mentions that a real-world system may require online estimates of system parameter changes to obtain ideal controller performance.

Time domain simulations of a similar vibration absorber system were presented by Li and Liu (2009), where the authors used a linear quadratic regulator (LQR) to actuate the vibration absorber and the block holding their drill string. An LQR controller is a state feedback controller which optimizes controller gains by solving a quadratic minimization problem. The optimization is based on weighting parameters. Li and Liu (2009) were able to show a heave motion decoupling of up to 84% with the potential to achieve further decoupling with additional iterations of weighting parameters in the LQR system.

Built upon a similar linear drill-string model as used by Korde (1998), Hatleskog and Dunnigan (2007) derive a linear transfer-function model for an active–passive hybrid system using feedfoward control on displacement (as opposed to Korde who used acceleration) as well as a PD feedback loop with respect to actuator position. The Hatleskog and Dunnigan system is mechanically simpler as a vibration absorber is not used in this case. The design and considerations for Hatleskog and Dunnigan's (2007) system are presented, but not simulated or implemented in their paper. Hatleskog and Dunnigan expect the system to be 90–95% effective, attributing any deviations from 100% to potential sensor error. It should be noted that Hatleskog and Dunnigan discuss using a closed-loop hydraulic system, as mentioned in Section 4.2 of this paper, to ensure a linear system response. A linear response, meaning that the actuator motion is directly proportional to the control signal, makes control design much less complicated.

In both the papers by Korde (1998) and the paper by Hatleskog and Dunnigan (2007), friction is considered linear. This assumption is rarely accurate in real-world applications, but is often used for simplicity. Do and Pan (2008) correct any linearized friction inaccuracies by modeling the total force on their hydraulic

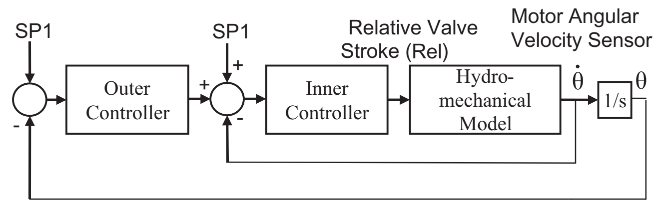

Fig. 16. The control scheme used by Gu et al. (2013) is shown here, with an inner velocity control loop and an outer position control loop.

actuator as

$$m_H\ddot{x}_H = A_H P_H - b_h\dot{x}_H + \tilde{\Delta},$$

where $m_H\ddot{x}_H$ represents the total force on their actuator, $A_H P_H$ models the force due to hydraulic pressure, $b_h\dot{x}_H$ models linear friction, and $\tilde{\Delta}$ is a state dependent disturbance term meant to account for nonlinear friction and other unmodeled forces. In this case the disturbance is not measurable so an observer is used. Additionally, Pan and Do build their system model to include a proportional control valve which, due to a flow across the valve being proportional to $\sqrt{\Delta P}$ where ΔP is the pressure drop across the valve, the system is inherently nonlinear (Eryilmaz and Wilson, 2006). Although the system could be linearized, Pan and Do chose to apply a nonlinear control scheme using Lyapunov's direct method. In using nonlinear control they were able to maintain the model's accuracy. For their simulation, Pan and Do obtained system parameters from Korde (1998) and the simulations show a load motion of less than 0.1 m deviation for a significant wave height of 4 m or an approximately 97.5% motion decoupling.

A simple P-PI controller is used by Gu et al. (2013) for control of a hydraulic hoisting rig meant to lower heavy loads to the seafloor. In their controller design shown in Fig. 16, Gu et al. (2013) use PI control as part of a closed-loop velocity control scheme for heave compensation, while P control is used in the outer control loop as position control to lower the load. In simulations, the controller was able to reduce a 1 m, 0.1 Hz sinusoidal heave motion input to approximately 1 cm or a 99% decoupling. Although this simulation predicts excellent performance, it should be noted that a pure sinusoidal input is an idealized heave signal, and it would be preferential to provide the system response for a full spectrum of ocean waves. Additionally, when moving from simulation to a real-world implementation, time-delay in system components may become a concern.

In the work by Hatleskog and Dunnigan (2007), it is briefly mentioned that a predictive controller may be helpful in creating an AHC system that approaches 100% effectiveness in heave motion decoupling. Reasoning is not given as to how prediction may improve performance; nevertheless, it is possible that a predictive controller could be useful in systems where a significant but consistent and known time-lag exists between heave measurement and actual motion. Prediction could also be used to partially correct for a large phase lag within the controller structure. Hastlekog and Dunnigan go on to say that heave motion of a vessel is "…essentially unpredictable with a high probability of significant predictive error". Halliday et al. (2006) published work providing a method for using Fast Fourier Transforms (FFTs) to accurately predict wave motion within 10% approximately 10 s into the future and up to 50 m away from the point of measurement. Although Halliday et al. intended to use short-term wave prediction to increase efficiency of wave-energy collectors, their work is easily adaptable to predicting short-term ship motion using IMU data. Neupert et al. (2008), at a conference in 2008, presented work to this effect.

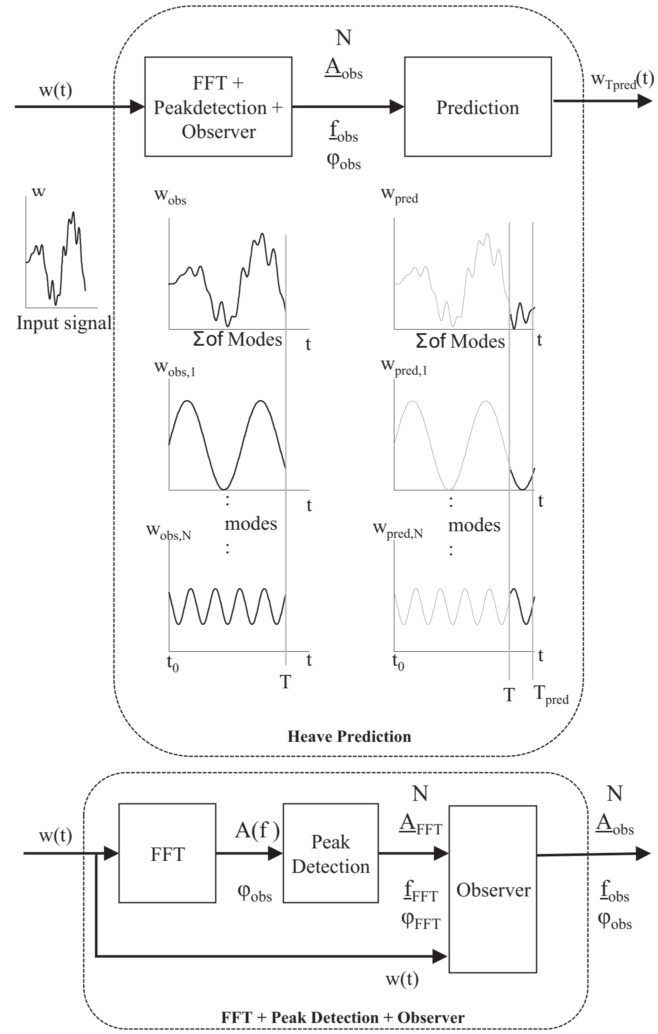

Fig. 17. A simplified schematic of the Neupert et al. method for heave prediction is shown here. Figure taken from Neupert et al. (2008).

Neupert et al. (2008) present a system using heave motion prediction as part of the control methodology for an AHC crane. Fig. 17 shows a simplified schematic of their heave prediction system.

To predict ship motion, ship heave data from an IMU data ($w(t)$ in Fig. 17) is collected for a set amount of time and an FFT is performed. Peak detection is performed on the FFT and the dominant peaks are determined, initializing an observer with the peak height A_{obs}, frequency f_{obs}, and phase ϕ_{obs}. A Kalman filter updates the value of A_{obs} in real-time while the other values are held constant until the next FFT is performed. Using a Kalman filter to update dominant peaks instead of performing an FFT every time step saves considerable computing power. When the FFT is performed again, some peaks may be removed or added to the observer depending on the data. The values for amplitude A_{obs}, frequency f_{obs}, and phase ϕ_{obs} are used by the prediction algorithm to predict future heave motion. The primary purpose of prediction in this controller is to help in dealing with known time delays between sensors and actuators which is important in systems with long delays as delay will introduce phase lag in a system, hindering a controller's ability to respond quickly.

Neupert et al. (2008) use a linearized model of crane dynamics along with the pole-placement control method to set load position. The authors apply a simple observer using a mass–spring–damper model to calculate actual load position during operation.

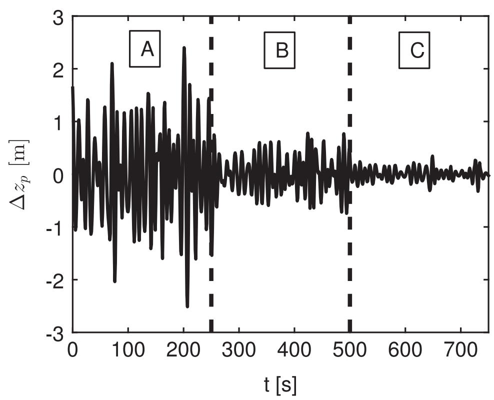

Fig. 18. Simulation results showing load motion split into three sections: no controller, controller without prediction, and controller with prediction. Figure is taken from Kuchler et al. (2011).

For a relatively stiff cable, this observer is likely unnecessary as the cable will not stretch appreciably and load motion will match actuator motion. Considering Eq. (3), if k is dominant in the numerator and denominator, then Eq. (3) can be simplified to $X_L/X_H = 1$ — meaning that load motion X_L matches actuator motion X_H. A dominant k would be representative of a load held at a shallow depth since cable mass, length, and damping would be relatively small. For considerable depth an observer becomes useful as k is no longer dominant in the transfer function and the load motion will not match ship motion.

Neupert et al. (2008) perform simulations showing that their state feedback controller can track a step-input to within ± 3 cm with a ship heave motion of approximately 0.5 m. In the follow-up work of Neupert et al. (2008), Kuchler et al. (2011) present the data seen in Fig. 18 showing that, with a larger heave motion, a load motion of less than ± 3 cm is no longer attainable. In region A of Fig. 18, from $t=0$ to 250 s, the controller is inactive. Region B of Fig. 18 shows the state-feedback control active, but heave prediction is unused. Region C of Fig. 18 shows state-feedback and heave prediction being used together. Based on a performance factor that the authors introduced, namely $\int_{t_0}^{t_0+250} \Delta z_p^2 \, dt$, energy in the load is reduced by 83% for the nonpredictive controller and energy is reduced by 98.2% for the predictive controller — showing a clear improvement when using heave prediction. Similar results are shown for experimental results; however, the same performance factor cannot be used as values are not reported for the heave motion.

In their experimental system, Kuchler et al. (2011) report a delay of approximately 0.7 s between sensor measurements and actuator response. It is possible that the inability of their controller to completely decouple load from heave motion is caused by the prediction algorithm error when trying to predict 0.7 s into the future. Reducing the system delay may further increase the ability of the system to reject heave from the load motion. Additionally, the use of state-feedback can be thought of as applying a filter to the system. When applying a filter, it is not always possible to completely decouple the output from the input. Feed-forward control is often applied to complement state-feedback controllers where it can lead to zero-error moving reference tracking in ideal circumstances (Lewis, 1992).

It is the present authors' opinion that the inability for many controllers to totally compensate for heave motion may not only be due to sensor lag, as mentioned by Kuchler et al. (2011), but also inherent phase lag in a system. It is well known that simple PID and pole-placement based controllers cannot perfectly track a sinusoidal moving reference because the controller's inherent phase lag ensures some delay in the system. While the addition

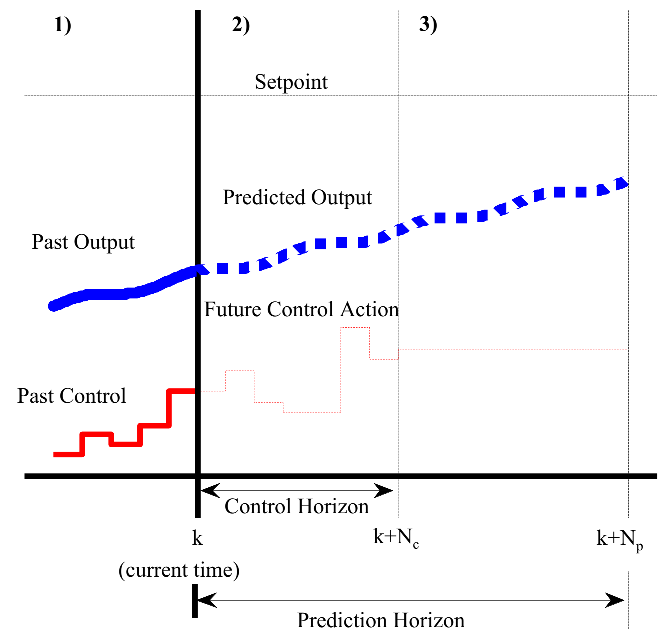

Fig. 19. The behavior of an MPC system is seen with the control and prediction horizons clearly labelled. The future action is calculated based on the current state at time t_0.

of a feed-forward component to both PID and pole-placement controllers can overcome delay due to system phase lag allowing perfect tracking of a sinusoidal reference (Lewis, 1992), coupling the inherent system phase lag with additional time delay can lead to significant system delays in the phase diagram which cannot be easily compensated for. A possible option to correct for large phase lag is the use of model-predictive control (MPC). A model-predictive controller relies on a system model to determine optimal controller output by solving a quadratic optimization problem.

Given a system model, MPC minimizes a cost function J where

$$J = \sum_{i=0}^{N_p} x_i^T Q x_i + \sum_{i=0}^{N_c} u_i^T P u_i + \sum_{i=0}^{N_c} \Delta u_i^T R \Delta u_i$$
$$x_{min} \le x_i \le x_{max}$$
$$u_{min} \le u_i \le u_{max}$$

Here, Q, P, and R are weighting parameters for the model states x, the controller output u, and the rate of change for the controller output Δu, respectively. N_p is the prediction horizon over which the controller allows the model to evolve and N_c is the control horizon, or how many time-steps forward the ideal control action is calculated. It is required that $N_p \ge N_c$ and for $i \ge N_c$, u_i and Δu_i are held constant. The choice of N_c and N_p will depend on the system sampling time.

To minimize the cost function, a type of mathematical optimization problem called Quadratic programming (QP) is used. The simple explanation is that the algorithm considers the set of all possible values of control action u over the control horizon and determines which values for u will minimize the cost function over the prediction horizon.

The function of a model predictive controller can be best explained by examining Fig. 19. In this figure there are three regions of time to consider:

1. The past, where the previous system output and the past controller action are found;
2. The current time t_0;

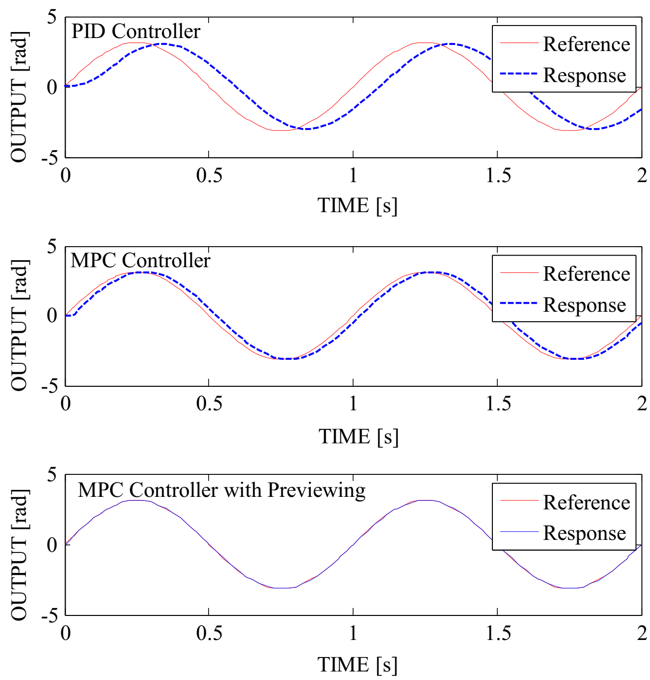

Fig. 20. The upper plot here shows PID being used to track a sinusoidal moving reference for a generic first-order system. The middle plot uses MPC to track the same reference, while the bottom uses MPC with previewing to track the reference with zero error.

3. The future, where the predicted system output is calculated based on the future optimal control action.

Before time t_0, we have the previous system response and controller action. At time t_0, the controller solves the QP problem of minimizing J and determines the optimal values of u from u_{t_0} to $u_{t_0+N_c}$ needed to reach the set-point. These control actions are shown between time t_0 and t_0+N_c. The optimal system reaction to these control actions is also seen between time t_0 and t_0+N_p.

Fig. 20 shows sample results of three controllers tracking a moving reference for a simple motor modeled as

$$\frac{\omega(s)}{V(s)}=\frac{K}{Is+B}$$

where ω is the angular velocity, V is the input voltage, I is the motor inertia, B is the motor damping, and K is a constant relating steady-state angular velocity to input voltage.

The top plot in Fig. 20 shows a tuned PID controller tracking a sinusoidal moving reference. Tuning was performed using the auto-tune function within MATLAB Simulink version 2013b. In Fig. 20, the PID controller lags behind the reference noticeably. The middle plot of Fig. 20 shows an MPC controller tracking the reference much more closely when compared to the PID system. The bottom plot shows a technique known as previewing being used in conjunction with the MPC controller, where knowledge of the moving set-point is used by the controller to improve tracking control action. In this scenario, we can see that the controller tracks a moving reference without error. Note that it is important to provide an accurate model when performing such simulations. In work by Ayman (2012) a linearized hydraulic valve model is shown in simulation to react to a step input faster and with less oscillation than the nonlinearized valve model. Since MPC uses the model to determine controller action, having a different response from a linear model compared to the nonlinear equivalent could lead to errors in a model based controller response.

Use of MPC is less common than PID or state-feedback controllers; however, there are examples of MPC being used in hydraulic applications, at-sea applications, and AHC systems. In 2001, for example, Kimiaghalam et al. (2001) simulated the application of MPC in parallel with a feed-forward controller to reduce swinging motion of a crane-suspended load at sea. A reduction of swing motion by a factor of approximately 200 compared to a free-swinging load was shown. Entao et al. (2009) used MPC to improve upon a simple inversion based feed-forward controller operating on a nonlinear hydraulic winch model. Entao et al. (2009) implement a finite difference model with real-time parameter estimation in their MPC controller to reduce a 20% error in sinusoidal reference tracking from their feed-forward controller down to a 3% tracking error using the MPC controller. Results by Deppen et al. (2011) in a 2011 conference proceeding show that MPC has been successfully implemented in the control of an electro-hydraulic system. In their work, the model is linearized and MPC is used off-line to produce a table of control objectives governing the controller output. Using an MPC controller off-line is often necessary when the model is too complex to run in real-time. Ideally, an online method using a nonlinear model would be preferred so as to allow the controller to be more accurate and to be used in a wider variety of configurations. A variation of MPC called a Model Predictive Trajectory Planner (MPTP) was successfully implemented by Richter et al. (2014) to generate smooth reference trajectories for a two degree-of-freedom controller to act upon. MPTP mathematically acts very similarly to MPC; however, MPTP acts open-loop to generate smooth, continuous trajectories for position and its derivatives which are then used as the inputs to a controller. Using the MPTP output, the two degree-of-freedom controller action results in smooth motion of the load, reducing potential for snap-loading. Additionally, MPTP can be used to account for physical limitations of the system such as velocity or acceleration limits.

6. Real-world controller validation

As most control engineers know, a controller designed using only simulated results can sometimes under-perform or fail to work entirely when first implemented using real-world hardware. The causes for this reduced performance may be numerous but often include unforeseen issues such as excessive sensor noise, unaccounted for system dynamics, nonlinearities, or even simple miscalculations. For these reasons, a new controller design should be validated in situ once it has been simulated successfully. Full scale sea trials are the ideal method to test an AHC control system; however, these trials are both costly to implement and potentially dangerous if a system failure were to occur. Due to the high cost and potential dangers, alternatives to full-scale testing have been explored by a number of groups.

For example, Johansen et al. (2003) and Messineo and Serrani (2009) used a scale-model floating crane in a small wave-pool to validate their controller designs meant for heavy-lift offshore marine operations. The mass used in the work by Messineo and Serrani (2009) was approximately 600 g, the motor was a small DC motor, and the wave height was reduced to scale. Although these are significant changes from the system which their controller was originally designed to operate, proving that the controller operates with sensor feedback within the frequency range of interest is an important step towards full-scale implementation.

Kjelland and Hansen (2015) were able to generate realistic ship motions using a Stewart platform, performing payload-transfer operations both to and from the Stewart platform using a hydraulic crane. The crane used a combined position feedback and velocity feed-forward AHC controller designed by Kjelland and Hansen (2015). The use of a Stewart platform not only allows the simulation of ship-deck motion but also allows payload transfer operations of at least 400 kg as shown by Kjelland and Hansen

(2015). Based upon the Stewart platforms large payload capacity, it follows that a Stewart platform could also support the installation of a winch and an IMU directly, allowing for simulation of an AHC controlled winch used in undersea towing operations. An alternative option to the Stewart platform for full actuation of a winch and IMU was employed by Richter et al. (2014) who performed real-world winch testing using a suspended platform capable of actuating to simulate full pitch, roll, and heave motions of a ship at sea. The platform was suspended at three points by three cables attached to an overhead crane and pulley system for actuation. The platform itself had a winch and an IMU mounted on-board, reacting to the platform motion and operating identically to an AHC system at sea.

7. Conclusion

For low-cost heave compensation where the best possible performance is not critical, passive heave compensator systems are an excellent solution due to their simplicity and ability to hold very large loads with minimal additional hardware requirements. In general, passive systems will not achieve higher than 80% heave decoupling. If further heave motion decoupling is required, it is recommended that an active or a hybrid passive–active system is used. A hybrid system will have reduced power requirements compared to a strictly active system; however, the increased complexity and infrastructure may not be worth the additional cost when compared to a strictly passive or active system.

Several controllers have been presented for active systems with load motion decoupling often depending on how accurately the researchers were able to model their system. Even with accurate models, the inclusion of time delay between sensor measurement and controller action can lead to poor response and, although it is not shown directly in this review, it is well known that time delay can also lead to closed-loop system instability. A method for removing system delays was shown by Kuchler et al. (2011), and the present authors believe that further improvement can be made by utilizing not only wave prediction but also MPC with previewing to attain optimal heave decoupling.

With respect to published works, further modeling and experimentation in the area of electric AHC systems are recommended as this area has not been significantly explored. Additionally, it would be beneficial to apply many of the controllers presented in the literature to a real-world system for validation on physical hardware as there are often significant differences between physical hardware and simulated hardware.

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