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Review of offshore winch drive topologies and control methods

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Winch systems have a critical role in offshore operations as they are widely used in various applications. Hydraulic drives have been preferred for these systems, though currently, there is an increasing trend to replace them with fully electrical systems, mainly due to the higher energy efficiency they offer. However, hydraulic winch drives have several characteristics, such as robustness and the ability to produce large torques for their size, which are attractive to the offshore industry. This paper provides an analytical review of offshore winch drive topologies and their corresponding control techniques. Based on the reviewed articles, some literature gaps are identified, and several suggestions for future research are given.

Keywords: Winch drives, offshore, maritime, review
Target audience: Fluid power community, Offshore industry

1 Introduction

In recent years, an increasing interest in offshore operations has been reported by a plethora of studies. This is driven by the current need to exploit more of the maritime environment. Such processes can include mineral extraction [1–3], offshore constructions [4–8], subsea exploration, and subsea lift and installations [9]. As a result, humanity moves away from fixed structures, such as offshore platforms, and towards vessels that are preferred to carry out these tasks in deep waters [10–12]. Winch systems play a pivotal role in these operations. They are widely used for various applications such as anchoring, hoisting and lifting, launch and recovery systems, towing, chain pulling, and mooring [13–15].

Winches are actuated by the so-called winch drive, which typically consists of an electric or hydraulic system. Following the trending growth of the offshore industry, a considerable amount of research has been focused on improving winch drives.

The focus of this work is to provide a systematic overview of publicly available articles that consider either a new winch drive topology or a control scheme. The review is limited to applications that require lifting or lowering heavy loads in the offshore environment.

This paper is organized in the following manner. Section 2 provides a brief description of the offshore environment concerning how it affects the winch system. Then Section 3 presents the conventional winch drives along with recent developments. Section 4 reviews the various control methods available in the literature. Then, in Section 5, the multiple literature gaps and suggestions for future work are discussed. Finally, the paper contents are summarized in Section 6.

2 Offshore Environment

The offshore environment imposes rigorous requirements on offshore winch systems. A short description is presented here regarding various problems that the driver has to handle. **Figure 1** shows a simplified illustration of an offshore operation for a crane mounted on a vessel with a winch actuating the crane load. Typically, winch systems are characterized by low operation speed and unidirectional loads. The wave-induced motion, which can be described with six different motions, will affect the vessel and the crane throughout the operation. However, the three horizontal movements (yaw, sway, and surge) are usually neglected as dynamic positioning systems can sufficiently compensate them [2, 16, 17]. Therefore, only vertical motions are considered (pitch, roll, and heave) and are typically measured with a motion reference unit (MRU) [17–19]. The vertical motions can be combined to calculate a net heave motion [2], which is then compensated with some kind of heave compensator. Calculating the various motions from the MRU signals requires a considerable amount of time and introduces feedback delay [8].

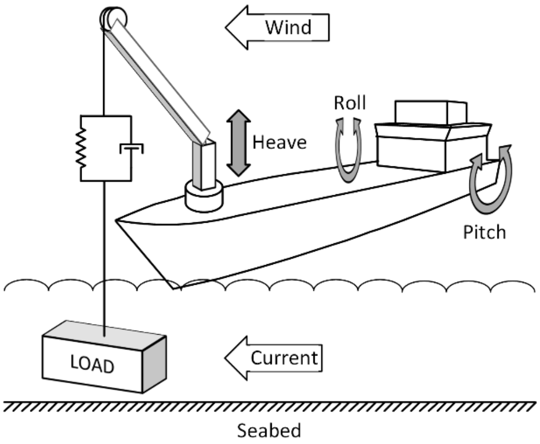

Figure 1: Simplified sketch of an offshore crane mounted on a vessel.

Heave compensation can be achieved either by equipping a passive heave compensation (PHC) system on the rope, which can generally be described as a spring and damper system, or with an active heave compensation system (AHC) [19]. AHC can be realized with linear actuators or by directly controlling the winch drum, though this paper focuses on the latter [19]. The waves acting on the vessel are naturally irregular and have varying shape, height, length, and propagation speed [20]. Therefore, it is vital to choose a suitable model to describe them, as they will directly influence the complexity of the motion that needs to be compensated. The crane itself is often assumed rigid and neglected. However, its dynamics can significantly impact the system, and thus methods for modeling these mechanical dynamics are already available in the literature [12]. Neglecting rope dynamics for relatively short lengths, in the order of tens of meters, can be acceptable; however, they should be considered when paying out several hunder meters of wire the drum’s inertia will also be affected [21]. Several external forces act on the load and wire, though these can vary depending on the considered lifting phase. For example, during a lift in the air, the load can be affected by strong winds and while underwater by sea currents [22]. Particularly during water entry, hydrodynamic forces caused by waves can damage the payload [11]. Finally, seabed dynamics should be considered when lowering a load to avoid possible damage to the load itself or any present foundations [20].

3 Offshore winch drives

3.1 Hydraulic drives

Hydraulic drives are the most commonly used for offshore winch applications [18, 23]. Typically hydraulic rotary drives are classified as hydrostatic (primary) or secondary controlled drives [24], where the latter are based on constant pressure line topology. Primary drives can be further characterized as open-circuit or closed-circuit. In the following, a brief description of the available topologies is given, along with recent contributions to them.

3.1.1 Open-circuit

An open-circuit drive system is shown in **Figure 2**. Here the primary unit is regarded as the supply with a directional valve. It is common practice that a Hydraulic Power Unit (HPU) supplies multiple actuators with constant pressure. A directional valve provides each motor with a constant flow; thus, the pressure drop across the motor depends on the produced torque. Often a booster pump will be included to ensure a minimum return line pressure [25].

Open-circuit drives are the most widespread drives in the offshore industry and are generally characterized by low cost, flexibility, and low efficiency [19, 26]. The most significant losses are typically attributed to the inability to recover energy, throttling losses, and selection of fixed-displacement machines [27, 28]. A theoretical investigation for the efficiency of open-circuit drives was conducted by Woodacre et al. [29]. The authors calculated especially low efficiencies for low speed/heavy load operations, and they also noted that the control strategy had a very low impact on the system's efficiency.

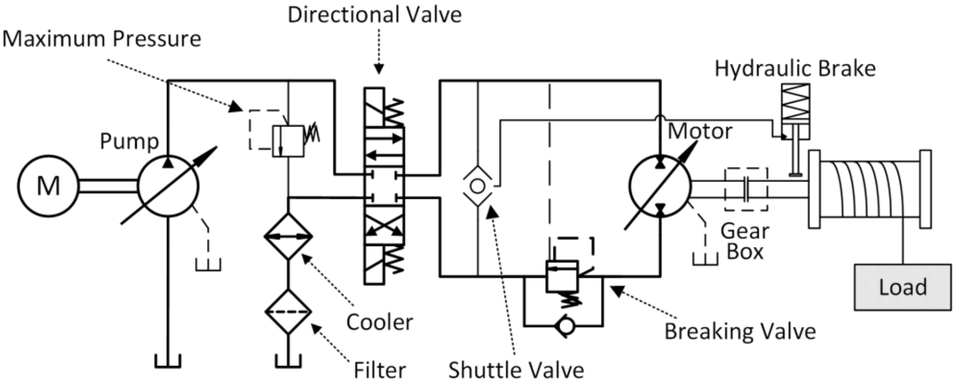

Figure 2: *Simplified open-circuit hydraulic winch driver.*

A slightly different topology was preferred by Skjong and Pedersen [21, 30, 31], Gu et al. [32], and Walid et al. [33], respectively. Here the 4/3 directional valve was replaced with a pair of 3/3 directional valves, which were actuated opposite of each other. The resulting control method was similar to using a conventional 4/3 valve but provided more control options. Specifically, in [31], these valves were mechanically driven by a minor hydraulic system with a double rod cylinder driven by a smaller solenoid 4/3 directional valve. The reason given by the authors for this approach was that large valves are not able to be operated using magnetic fields. Another topology variation can be found in Than et al. [12], where an open-circuit drive for a constant tension winch was considered. A compensator valve was installed prior to the directional valve and a short cut valve between the motor outlets to ensure a specified pressure when in constant tension mode. The short-cut valve opened only when the load torque was greater than the selected motor torque. As the load drove the motor, the generated flow was sent to the other outlet and stabilized the motor in a new equilibrium, therefore, keeping the rope tension constant.

3.1.2 Closed-circuit

The directional valve is absent in a closed-circuit configuration as a dedicated variable-displacement pump is directly coupled with a hydraulic motor, as shown in **Figure 3**. This setup enables flow in both directions, allowing the motor to operate in all four quadrants, thus enabling energy regeneration. Closed-circuit drives are distinguished for their very high energy efficiency [34], attributed to the absence of throttling losses and the fact that the pump will only supply the required flow. Unfortunately, they are penalized for low response time and more expensive components [24, 28]. Costs can be dramatically increased depending on the number of required motors, as each motor requires a dedicated pump.

Modern closed-circuit systems can also store potential energy [35] to increase their efficiency further. Such a system was presented by Xiao et al. [36]. By developing a suitable flow controller, the system can recover energy by storing it in the accumulator through a hydraulic transformer for later usage. This method of energy recovery is potentially superior to sending the energy back to the grid, as hydraulic accumulators are better suited for energy storage than battery banks [37, 38]. Simulation results showed that the scheme could meet the performance requirements by containing the heave motion below 5 mm. However, faster dynamics for the heave-induced motion were not considered. Unfortunately, the work was focused on proving the concept of using such a topology, so the authors did not investigate the system's efficiency.

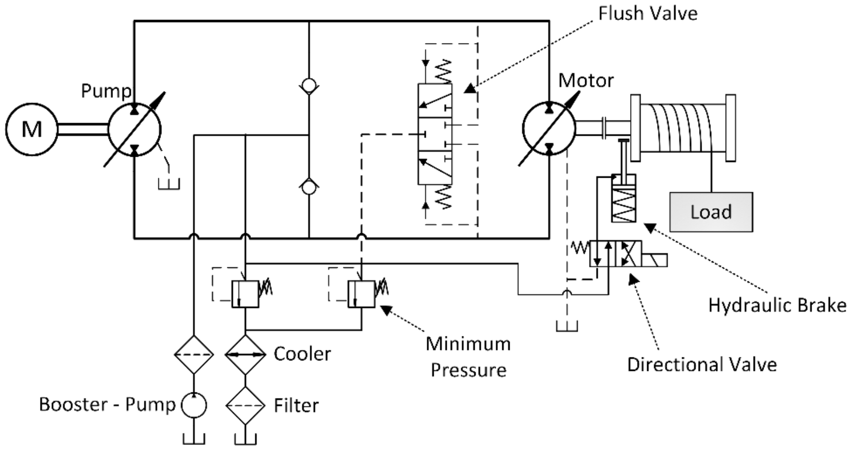

Figure 3: Simplified closed-circuit hydraulic winch driver.

The drive considered by Mostlått et al. [16, 18] combined both secondary control as a passive system and primary control as an active system. These operate in parallel and are therefore commonly referred to as an active/passive hydraulic winch system. This type of driver has been in use since the 90s and is still widely used on cranes that lift over 100 tons due to its low cost [18]. The drive presented in the paper simulated a winch drive used by a National Oilwell Varco crane. A planetary gear connects multiple motors to the winch drum. The majority is used by the passive system, which operates with secondary control, and its only purpose is to counterbalance a predefined load. The rest of the motors are speed controlled by the active system and are tasked with compensating for the other forces e.g. friction, heave motion, and operator input. Even though, the hydraulic actuators have variable displacement, they are only occusionaly adjusted based on the current winch layer and depedning on whether the crane is lifting a heavy or light load to simplify the control task. Hence the motors always act like having a fixed-displacement.

Recently, a novel topology was presented by Schmidt et al. [25] that shares some similarities with an open-circuit topology. The primary system consists of a dedicated variable-displacement pump equipped with a boot-strap reservoir and a proportional valve. Similar to a conventional closed-circuit system, the footprint of the winch driver on deck was significantly reduced as there was no need for an HPU or a booster pump. The system introduced some strong flow couplings between the valve, pump, and reservoir, which needed to be addressed by a compensator. A linear decoupling compensator was shown to provide sufficient reduction of these couplings. It should be noted that even though the removal of the boost pump is theoretically possible, there has been no consideration for the effects that this would have on the quality of the oil.

Another alternative closed-circuit winch driver was proposed by Schmidt et al. [39] to improve the energy efficiency of winch drivers. Here, the fluid supply was provided by three pumps, two connected to the inlet of the motor and one at its outlet. Both outlets were also connected to the return tank through a 2/2 proportional valve. The pumps had variable-speed/ fixed-displacement and were actuated by a single electric servo drive. It was noted that this topology is only suitable for unidirectional loads driven by symmetrical motors. Even though no significant benefits were provided for tracking performance during heave compensation, the proposed method was suggested to have twice the energy efficiency of a conventional open-circuit driver. The authors suggested that further energy savings could be achieved if the electric motor could recover energy.

3.1.3 Secondary-controlled drives

Secondary-controlled hydraulic drives resemble the behavior of electrical systems. They were developed as an alternative to primary controlled systems that would not suffer from many of the disadvantages that come with them, such as low response time and throttling losses. The concept was presented first by H. Nikolaus [40], and it has been widely used since 1980 [27]. As illustrated in **Figure 4**, the motor has to be able to alter its displacement as it is used to control the torque output. The motor speed then follows due to the difference between the produced torque and the load torque. The hydraulic pumps produce flow only in one direction and must keep the pressure constant, regardless of flow, thus resembling an electric circuit's high- and low-voltage lines.

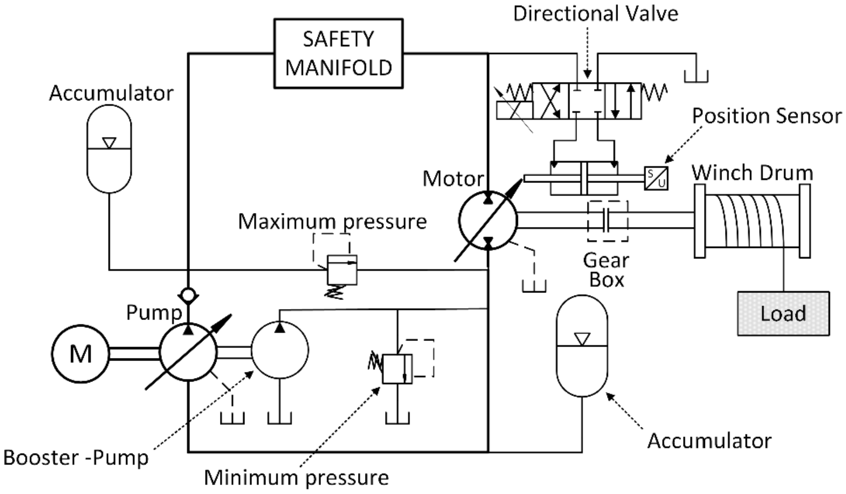

Figure 4: Simplified closed-circuit system with secondary control.

The variable-displacement motor shown in Figure 4 could be realized as a digital displacement motor (DDM). The primary motivation for the DDM development was its superior energy efficiency, but other benefits include scalability, redundancy, and four-quadrant operation [41]. As a technology, DDMs are relatively new as they were first introduced in the early 80s, while the first application on the offshore environment seems to be for a wind turbine in 2013 [42]. The first DDM was developed by Artemis Intelligent Power which is currently owned by Danfoss [43]. However, there are also other companies that currently work with DDM technology, such as Diinef A/S and Chapdrive A/S [41].

3.2 Electric Drives

Electrically driven winches have been in use since 1970 [44]. The first were DC-driven, whereas AC-driven drives first came into use in 1995 [45]. A typical electric winch drive system consists of a Variable Frequency Driver (VFD) that controls an induction motor with scalar or field-oriented control. A position sensor is used to provide feedback for the controller [46], as shown in **Figure 5**.

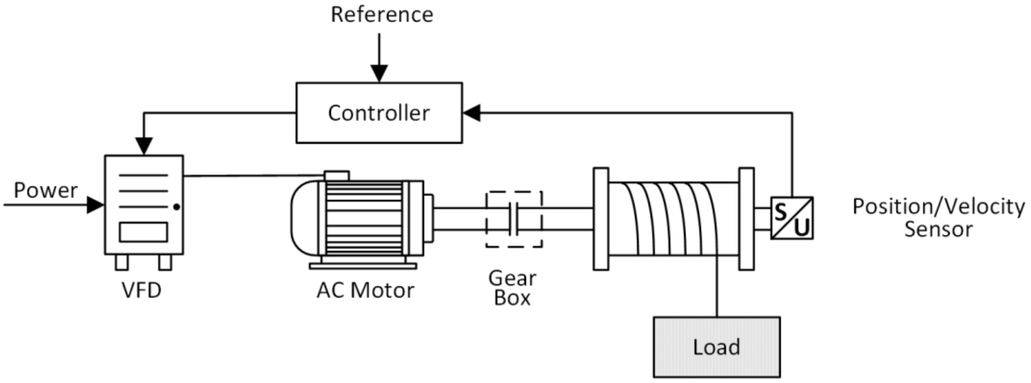

Figure 5: Simplified electric driver topology.

This topology has to be repeated for every motor in the system, thus increasing the overall cost. Furthermore, an electric motor will be equipped with a braking system and its own air cooling system [24], which is designed based on the operation of the motor. In general, increased cooling will be demanded in applications that require low-speed and high torque, while even water cooling can be necessary in exceptional cases. Regardless of the chosen cooling method, an electric motor can operate at maximum torque only for a short amount of time.

Induction motors naturally operate in all four quadrants, meaning that they can also act as generators when driven by the load, as is often the case in e.g. lowering of a load. This excess energy can either be burned off using dynamic braking resistors or recovered through the power grid [47]. Energy recovery is an available option in the industry [48]; however, a present challenge is that this additional energy can also disrupt other systems connected to the grid if it cannot be used immediately [19]. Therefore, this requires the installation of battery or capacitor banks, leading to increased costs [38, 49].

3.3 Overview of Electric and Hydraulic Drives

Hydraulic drives are widespread in offshore winch applications [18, 23], though fully electrical drives have become prominent in recent years. This trend has been mainly attributed to electric drives' overall high energy efficiency, reaching 85 to 90% [37], though more conservative estimates report 70 to 85% [23]. The energy efficiency will generally be much lower for hydraulic drives and greatly depend on the application and topology. For example, closed-circuit and secondary-controlled systems will typically have higher efficiency compared to open-circuit systems. However, multiple parameters can affect the selection of an offshore winch driver. Therefore, this section provides some insight by presenting a general overview of electric and hydraulic drive systems.

Reliability

As previously mentioned, induction motors and VFDs are susceptible to failures due to overheating, especially in high-torque/low-speed operation, high levels of humidity, and contaminations [23, 38]. A water cooling system may be required for the motors, while the VFD needs to be kept in an enclosed space to accommodate these problems, which increases costs [24]. Hydraulic drives offer robustness as the oil can compress to absorb impacts. They also have reduced friction as various drivetrain components are lubricated, and the fluid removes excess heat. Hydraulic actuators do not suffer from fatigue during intermittent, discontinuous, or low-speed/high-torque operations [38]. Electric winch driver reliability can also be high when considering the total lifetime of the driver due to having fewer moving parts [23]. However, in some instances, various electrical system components, such as VFDs, power supply, and controllers, have been reported to have failures [23].

Maintenance

Fewer moving parts also means that electric drives do not require regular maintenance, though problems are significantly harder to locate, and system restoration requires highly trained personnel [23]. Their reparation time is often prolonged, as component replacement requires multiple spare parts limited to specific suppliers. In contrast, hydraulic drives require regular maintenance; however, replacement components are widely available, and even components from other hydraulic systems on the vessel can be used [23]. Additionally, they are more intuitive as technology and technicians are more familiar with them, making reparation easier. Often, electric drives are regarded as the most environmentally friendly option for eliminating the danger of oil spillage and offering higher energy efficiency and significantly lower noise levels [38].

Performance

Regarding performance, electric systems have a significant advantage as they allow for highly accurate torque and speed control. This characteristic allows electrically driven winches to lift different loads with a higher velocity compared to hydraulic drivers [24]. Additionally, electric actuators can operate in temperatures ranging from -40 to +45°C [50], while the hydraulic system performance depends highly on oil conditions [38]. There are rigorous requirements for the oil, such as acceptable temperature and contamination. An excessive or low temperature changes the physical characteristics of the oil and, in turn, the overall system performance. As such, immersion heaters or cooling systems may be needed while the HPU may be required to be relocated inside the ship [37]. In the case of moving the HPU, additional pipe systems need to be installed to carry the fluid to the actuators, resulting in increased costs as well as installation and commissioning time. The oil needs to have a certain level of decontamination from water and debris particles to prevent failures in the system; thus, the oil needs to be carefully filtered before entering the system [37].

Costs

Hydraulic drives are suggested as having a low initial cost, while electric drives are expected to pay off their higher price over time [51]. Overall costs will also be affected by the weight and volume of the equipment [52]. Therefore manufacturers aim for lighter solutions with a small footprint on deck [22]. As such, several claims estimate that electric drives occupy less space when compared to hydraulic solutions [23, 38]. However, the driver size will depend on multiple parameters and vary depending on the application. Electric drives generally require additional systems for proper operation, such as mechanical brakes, cooling systems, harmonic reduction systems, transformers, and battery banks that occupy a lot of space [37, 38].

Additionally, electric motors tend to be physically large and have high inertia. For this reason, there is considerable interest in developing methods to reduce the size of induction motors [46, 53]. Hydraulic actuators can provide dynamic breaking if pressure relief valves are included for that purpose [38]. Hydraulic accumulators offer a better alternative to battery banks for storing recovered energy as they require less space and are significantly cheaper, especially for storing energy temporarily [37, 38]. Hydraulic motors also produce the largest torques for their inertia and thus require less space at the actuation point. This also allows faster acceleration which is necessary for following trajectories with fast dynamics [19]. However, it should be noted that HPUs used by open-circuit systems typically occupy a significant deck space and have a large mass. As a result, costs increase, especially since pipe installation is more expensive than wires that offer minimal testing time [37].

4 Control Methods

This section presents the various control schemes found in the literature used to control an offshore winch. The articles are presented with respect to the topology that they considered.

4.1 Open-circuit

One of the earliest articles to consider the water-entry phase was Svein I. Sagatun [54], who developed an AHC controller capable of reducing the wave forces acting on the load. Soon after, Johansen et al. [54] developed a feedforward compensator for a moonpool crane drive to reduce these forces. Moonpool cranes lower the load through an opening in the ship's hull called a moonpool. Force reduction was achieved by matching the vertical load speed with the wave speed and was referred to as wave-synchronization. Experimental results were obtained with a scaled-down model of a moonpool crane with a single electric servo motor driving the winch. The setup was validated with experimental data and used extensively to investigate several other controllers that considered the water-entry phase of lift operations [11, 55–57]. The results showed that the wire tension standard variation and peak values were significantly reduced across all tests.

Skaare and Egeland [58] proposed a parallel force/position controller to reduce the hydrodynamic forces as the load passes the water surface. A passive heave compensation system reduced the bulk of the vertical movement while the controller handled the resulting differences. The scheme was compared with the wave-synchronization technique developed in [54] using simulations. Results here show a further reduction of oscillations in wire tension while maintaining similar performance for reducing hydrodynamic forces on the load, though certain large tension spikes were observed. The controllers were later tested in [56] on the scaled-down setup of Johansen et al. [54] and compared with the AHC controller of [56]. The suggested parallel force/position controller seemed to avoid negative wire forces altogether and generally improved the minimum wire tension, which is vital for offshore operations as a slack wire can lead to dangerous situations.

Later, Messineo and Serrani [11] suggested that a model-based feedback compensator can offer better performance when compared to the feedforward controller proposed by [54]. For that purpose, they developed two separate controllers. One was tasked with reducing the wire tension variation while the load is on air, and the other aimed to reduce the slamming forces on the load during water entry. As both controllers required feedback that a sensor could not directly provide, the necessary information was estimated by two observers. Experimental results showed that the feedback controller reduced the most significant hydrodynamic force on the load by 11.25% and the standard deviation by 8%. Similar results were shown for the tension control; the wire-tension standard variation dropped by approximately 18%. Similarly, the most prominent peak value was reduced by 4%, and the lowest was increased by about 60%. The control scheme was further developed in [57] by allowing the controller to adapt to system parameter variations and estimate external disturbances. The suggested technique improved overall performance compared to their past work. Specifically, the maximum absolute value of the hydrodynamic force was reduced by almost 14%, while the standard deviation was reduced to 50%.

A modified proportional integral derivative (PID) controller was presented by Sun et al. [4, 59], where the proportional gain was updated regularly with a genetic algorithm. A simulation model was developed for test purposes. This model combined ADAMS, AMESim, and MATLAB models for the ship and crane structure and included the rope dynamics and coulomb friction between the cable and the winch drum. The genetic-PID step response was compared to that of a conventional PID, and it was shown that it could reduce the overshoot and oscillations in steady-state. Experimental tests were provided for the controllers. However, the results are difficult to evaluate quantitatively as a small electric motor was used to drive a load with a mass of 10 kg. Additionally, the authors did not provide evidence to show how the setup dynamics corresponded with a real system. Additionally, a position sensor provided direct measurements for the load, which is not typically feasible [11].

Another PID controller with a variable proportional gain based on fuzzy logic was developed by Chen et al. [60] to achieve tension control. The proportional gain was regulated based on a tension sensor. The performance was compared to that of a regular PID-controller using a simulation model verified experimentally in previous work [5]. The results showed a reduction of the highest force peaks by more than 50%, while the RMS value was reduced by approximately 20% and 35% for two different set-point values.

Walid et al. [33] and later Gu et al. [32] presented a methodology for modeling hydraulic draw-works with AHC to optimize the design before starting production. For that purpose, several parameters were considered for the mechanical system, such as friction on the sheave axis, wire dynamics, and the drum's inertia for the amount of released wire. The system was controlled using a proportional controller to regulate the position while a second loop velocity PI controller compensated for the heave motion of the crane. Two test cases were considered to verify that the rig could reject heave motion while the load remained fixed at a specified height and while being lowered to seabed. In both cases, the performance was satisfying as the load oscillated about 1cm from the desired position while the wire tension remained within the specified limits.

Sanders et al. [61] suggested that the AHC of a winch system could be improved by supplementing a PD controller with a velocity feedforward and tension feedback control using measurements from an MRU and a tension sensor, respectively. The hypothesis was tested using simulations where the rope dynamics and drum variable inertia were considered. Results showed a reduction of about 75% in the produced velocity error; however, they await experimental validation.

Zhao et al. [17] considered developing a simulation model for an offshore crane hardware-in-the-loop test system. The purpose was to test an actual controller using a simulated model and debug it prior to implementation in a real system. The system model was linearized and modeled using state-space equations, while a conventional PID controller was used to control the load position and achieve AHC. The controller showed good performance by reducing heave motion by about 90.3% while maintaining constant tension and 95% when lowering the load to the seabed. Here it should be mentioned that in real applications, the load states are usually unmeasurable, and therefore some kind of observer would be required to provide that information [62].

In [63], Entao and Wenlin presented an adaptive feedforward controller that utilized least-square parameter identification to estimate the system's parameters and adapt the control output accordingly. Simulation tests showed that this method could reduce the velocity tracking error by 60% compared to a conventional feedforward control.

The effects of the wire dynamics on the system were investigated by Skjong and Pedersen [21]. By modeling the wire as a series of spring/damper systems, they showed that the natural frequency of the system shifts for extensive wire lengths. This model was later used to test a model-based controller that was suggested to be better suited for handling the various nonlinearities of hydraulic systems [31]. The final scheme consisted of a speed and a torque controller for AHC and tension control, respectively, based on backstepping and sliding mode methods. Simulation tests were conducted for frequencies up to 1.3 Hz and included hydrodynamic forces acting on the load and wire. The results showed that the robust speed controller accurately tracked the reference with a slightly reduced performance in higher frequencies. The performance of the tension controller seemed to be satisfying though it was somewhat sensitive to the load direction, which caused spikes in the order of approximately 500 N/m to appear. Still, they dissipated after 0.5 s.

Woodacre et al. [29] proposed a model-predictive controller (MPC) for AHC based on the results from their review paper [19]. In addition to the controller itself, three supplementary techniques were applied to improve the valve's hysteresis, dead-band, and nonlinearity. Specifically, a small sine-wave signal was added to the valve control signal with a frequency of 50 Hz. This signal was suggested to reduce the valve's hysteresis and allow the spool to move further, thus increasing the motor's maximum speed by 15%. Additionally, the valve's input signal was adjusted to skip the dead-band region. Further adjustments to the signal included an algorithm to linearize the valve's response. The sensor delay was compensated using the motion prediction algorithm developed by Kuchler et al. [8]. A comparison with a conventional PID controller was carried out, using an experimental setup without load for two different heave-induced motions. The controller reduced the standard deviation of the heave motion error by 8.6 mm or 14.8 mm, depending on the considered heave motion. Later, the authors tested the controller with additional experimental tests and simulations [65]. Two more heave motion trajectories were used that contained higher frequencies. Here the MPC showed superior performance by reducing the heave motion by 13 cm and 26.4 cm for each case. The loaded simulation tests showed that the MPC produced a lag in its response, attributed to inaccurate modeling of the motor leakage, and a PI controller was added to reduce it. New simulations showed that the proposed MPC-PI controller significantly outperforms the MPC and PID controllers in all test cases.

Recently, Li et al. [65] proposed a method for addressing the dead-band and input saturation of the valve while also rejecting external disturbances in the system. They suggested a cascade controller where an MPC strategy ensured that the motor followed the reference. An active disturbance rejection controller (ADRC) generated the MPC input and compensated for external disturbances and model uncertainties. The scheme was compared with a PID, an ADRC, and a Sliding Mode Controller (SMC) developed in past work [66, 67], and also the MPC presented by Woodacre et al. [64]. Simulation results showed that the ADRC and PID were very sensitive under harsh sea conditions. The load peak-to-peak oscillation reached 2 m, while the MPC seemed to require a significant amount of time (5 s) to recover after the appearance of disturbances. However, the results were based on simulation results only and did seem high compared to results reported by other researchers [17, 29, 32, 33]. After the 5 s period, the MPC performance deteriorated with peak-to-peak oscillations of almost 40 cm. The SMC was compared with the proposed scheme only for irregular waves and showed a similar behavior though with larger fluctuations. The proposed controller showed superior performance across all tests, and the authors reported load heave motion reduction in the order of 94.4%, 92.4%, and 85.6% for three different sea states.

4.2 Closed-circuit

Huang et al. [68] considered a closed-circuit drive with a variable-speed/variable-displacement pump to develop a Bang-Bang and PID control scheme. The authors evaluated the pump type's effect on the system performance using AMESim and Simulink by comparing the system step response for three different types of pumps. Specifically, the considered pumps could vary their speed, displacement, or both of these parameters. Utilizing a variable-speed/variable-displacement pump resulted in a system with a significantly reduced response time. However, some oscillations appeared in steady-state, which corresponded with the presented experimental verification results. The proposed scheme switched between the two controllers depending on the produced error.

A significant deviation from the velocity reference would activate the Bang-Bang, while the PID-controller took over for more minor errors. Simulation results showed a reduction of 50% in the response time and the elimination of overshoot. Experimental tests revealed that the controller could successfully track step-wise inputs and a sinusoidal signal of 0.1 Hz frequency. However, faster dynamics were not considered while no information was given regarding the motor's power rating or the actuated load.

Zinage and Somayajula [2] presented an extensive comparison between linear quadratic integrator (LQI), PD, MPC, and SMC controllers to investigate their performance for a closed-circuit system. The authors calculated a heave net motion based on a KCS container ship which resulted in a heave motion with complex dynamics. Three different sea states were considered to investigate the performance of the controllers under the presence of sensor noise and external disturbances. Simulation results revealed that the nonlinear methods had superior performance only under specific conditions. Specifically, the LQI controller seemed unable to compensate for harsher sea states, while the SMC and MPC seemed susceptible to disturbances and sensor noise, respectively. Therefore, the PD controller was distinguished as the most reliable option because it could maintain an acceptable performance in all kinds of conditions. Based on this conclusion, Zinage and Somayajula compared a PD controller with a novel reinforcement learning controller (RL) [69]. This type of control did not require a model for the system. Instead, it used a deep deterministic policy gradient algorithm "trained" through simulations to calculate the desired control law. Similar to their previous work [2], the controllers were tested and compared under various sea conditions and the presence of external disturbances and sensor noise. The RL controller showed superior performance under all sea states and with high feedback noise, while the PD controller seemed better suited for low noise levels. Both controllers handled disturbance rejection satisfactorily though the RL controller was more capable of dealing with saturation dynamics.

A semi-secondary control method for active/passive hydraulic winches was proposed by Moslått et al. [18] that adjusts the motor displacement during the operation. Two main issues were identified in the currently used scheme concerning the motor displacement control and the production of errors when changing direction at low speeds. The proposed method controlled the pumps with a feedforward signal and a linear feedback controller while adjusting the motor displacement based on the attached payload and the feedforward signal from the pumps. During low-speed operation, the motor displacement was kept at its maximum value. Simulation tests showed that the proposed strategy could increase the operation speed for specific loads up to 30% while reducing pressure peaks by 20% and regulation errors by 30%. The proposed scheme was further considered in later work in Moslått et al. [16], where its performance was verified experimentally in a full-scale crane. The experimental results corresponded with those shown in simulations with and without a load, though only unloaded results were presented analytically. When operating without a load, the proposed scheme was tested both for a low and a high speed as well as AHC. In all the tests, the controller achieved a smoother and more accurate control for position and velocity, while it was also able to reduce pressure peaks significantly. Specifically for the high-speed test, it was noted that the controller reduced the maximum pressure peak by over 120 bar and the control error by 50%.

Studies done at the Institute for System Dynamics from the University of Stuttgart have focused on addressing the sensor feedback delay [1, 8, 70, 71]. This was achieved with controllers that rejected the wave-induced motion based on trajectories provided by heave motion prediction algorithms. The prediction algorithm was initially developed by Neupert et al. [70], later expanded upon by Küchler et al. [8], and consisted of three parts. First, the heave motion signal from the MRU was analyzed with FFT, and the most dominant sine waves were extracted using a peak-detection algorithm and parameterized a heave motion model, which was essentially a sum of the said sine waves. Until the next FFT was performed, a Kalman filter adapted the parameters based on online measurements. Based on these estimations, the future heave motion was calculated with excellent results, as it accurately predicted the future heave motion for a period of one second. Without using the prediction algorithm, the controller reduces heave motion peaks by 50%, while a 75% reduction was achieved with motion prediction. This improvement was also reflected by the position error RMS value, which was reduced from 34 cm to 11 cm. Experimental results were included that closely resembled the presented simulations.

Richter et al. [71] further expanded the algorithm to provide optimal trajectories based on velocity and acceleration constraints for the winch. A repetitive polynomial-based trajectory planner could also offer a trajectory if an optimum were not found in time. The method was verified with experimental data, where the proposed scheme generated a reference that fulfilled all the requirements. This work was extended in Richter et al. [1] to produce more experimental results for the AHC. The prediction algorithm was based on the superposition of sine waves whose parameters were estimated using the Variable Order Levinson RLS algorithm. This was suggested to improve accuracy during sea state rapid changes. The control scheme combined a feedforward controller with a PID controller while also rejecting external disturbances. Experimental results on a full-scale setup showed that the prediction algorithm gave excellent results for the short-term prediction of 0.5 s, while the accuracy was slightly reduced for a 2 s prediction. The control scheme seemed to accurately track the operator input while compensating for heave motion and rejecting disturbances. Though, as the authors did not include specific numbers for the accuracy of the prediction algorithm or the control error, the results are hard to evaluate and compare with other methods.

4.3 Secondary-Control

One of the earliest works that considered secondary control for AHC at offshore cranes was Dabing et al. [72], where a cascade controller combining conventional PID and fuzzy PID was presented. The simulation model was built with Matlab/Simulink, where the motor was assumed ideal and the pressure supply constant. At the same time, a single spring/damper system was used to model the rope dynamics. Specific details for the controller were not provided; however, from the given figure, it could be concluded that the fuzzy PID adjusted the position of the motor swash plate while the conventional PID controlled the motor position. The oscillations were reduced when the controller operated at constant tension mode but remained significantly large as a maximum peak of 1 m could be observed. The AHC performed better, as the maximum position error remained below 30 cm at all times, which could still be considered significant for an offshore lifting application [22].

Wu and Wu [62] considered a control scheme for secondary control with energy recovery capabilities optimized for improved performance and energy efficiency. Three separate controllers were proposed to control the system. A PI controller regulated the rotational speed of the actuator based on the combined output of a proportional load position controller and a feedforward controller. The latter was also designed to reject the heave motion based on MRU measurements. It should be noted that since the load position could not be directly measured, an observer was designed to estimate its position. The algorithm used particle swarm optimization to select the pump and motor models based on their power output. The same technique was used to optimize the values for the control parameters, gearbox reduction ratio, and the accumulator pre-charge pressure and gas volume. The authors used the integral square of the position error to evaluate the system performance while two indexes were used for the energy efficiency. One index represented the power output of the supply pump, while the other index was calculated by dividing the amount of energy that the motor consumed by the amount of power that the accumulator provided. Simulations results showed that the proposed controller achieved a satisfying performance, which further improved by 57.8% after parameter optimization. A significant improvement could also be observed concerning energy efficiency, as the proposed method reduced the power consumption by 6% and 15.6% for the supply and actuator, respectively.

The water-entry phase of a subsea lift was recently considered by Wu et al. [9]. They used the same topology as [62] and a cascade controller to reduce the hydrodynamic forces acting on the payload. Here, an indirect adaptive robust backstepping controller was utilized to adjust the motor's displacement while compensating for model uncertainties and disturbances. The input for that controller was provided by an impedance controller, which generated a trajectory based on the current motor displacement and a load motion estimation. The proposed scheme was compared with the parallel force/position control of Skaare and Egeland [56] in a simulated environment. Shortly after passing through the water surface, the load seemed to oscillate more with the impedance controller than the force/position scheme. However, after approximately 5 s, the position error was reduced and closely followed the trajectory, while the force/position control had a significant steady-state error. For the hydrodynamic forces, the authors reported an 11.1% improvement when the maximum values were considered. Similarly, the force standard deviation error was reduced by 23.9%, while the load passed the surface and by 48.4% after submersion. A similar pattern was seen for the rope tension, with the minimum tension improving by 22.1%, while the standard deviation numbers showed an improvement of 9.1% and 61.4% during and after the payload was submerged, respectively.

Marien and Wiig [73] investigated a secondary controlled driver with a digital displacement motor (DDM) that could directly drive the winch drum without a gearbox. A constant pressure supply was assumed to simplify the system, though several mechanical dynamics were considered, such as rope elasticity and a variable winch drum inertia. The proposed scheme was a cascade controller with three levels. A PID controller and a feedforward velocity element controlled the displacement of the DDM by adjusting the cylinder stroke volume. Then, an operation mode controller selected the operation quadrant for the motor, which would then activate the corresponding mode that generated a valve actuation sequence. Simulation results showed that the winch accurately tracked the reference trajectory even in the presence of heave motion though the performance considerably deteriorated at near-zero velocities. Therefore, the authors proposed the addition of an extra mode that closed all the valves when the motor had a very low speed instead of relying on the control scheme to keep the actuator immobile.

A unique control method for DDMs was presented by Larsen et al. [74] that achieved excellent positioning at low speeds. This "creep-mode" controller activated the cylinders so that the motor moved from one equilibrium position to the next in a step-wise fashion. The controller was validated in an experimental setup where six DDMs actuated a 25-ton load while two electric motors provided dynamic loading. The installed HPU had a large energy capacity, allowing the motors to operate with constant pressure throughout the tests. Results with the winch running with half of its torque capacity showed that the controller could accurately control the load position as the shaft position rotated approximately 0.005 radians every 0.1 seconds. The authors outlined that this control method could also introduce various issues as it could fatigue the valves or other electrical components. Applications with fast dynamics using creep-mode could also suffer from excessive vibrations that would further increase noise levels or could potentially resonate with various materials on the crane, thus leading to critical situations.

Nordås et al. [75] investigated the performance of a winch driver with digital hydraulic motors for both an AHC and a constant tension controller. Several parameters of the offshore environment that can affect the system performance were considered such as, buoyancy and drag forces that affect the load underwater. Multiple mechanical dynamics were also included, such as variable winch drum inertia and rope and seabed dynamics. The control scheme consisted of a PID for active heave motion control and a PI for constant tension control. Similarly to [73], the controllers generated a displacement reference that, depending on which quadrant the motor operated at the moment, activated the corresponding valve actuation sequence to create an equivalent displacement for the DDM. Various simulation scenarios were presented to evaluate the performance of the driver. When the load was lowered to the seabed under the effects of heave motion, the tracking accuracy seemed to be very good. The maximum position error was 1 cm and occurred when the AHC controller was activated. At the same time, the error was contained within ±3 mm afterward, even during landing. Similar behavior was noted when the load was lowered with tension control. Here, the maximum force peaks remained below 0.5% of the maximum safe working load at all times. When lifting through the air, the heave motion was neglected, and a constant disturbance force was added that affected the load periodically. Here, the controller achieved zero tracking error with occasional 5 mm peaks appearing when the external force was applied or diminishing and when the motor direction changed. In general, the controller seemed to have an exceptional performance; however, several assumptions were made, such as zero feedback delay and constant pressure supply, that could significantly affect performance.

In a later work, Nordås et al. [22] used the simulation model developed in [75] to evaluate and compare various control methods with respect to performance and robustness while also identifying implementation challenges. Three control methods were designed to be tested for lifting from the deck and lowering to the seabed, a conventional PD controller, a sliding mode controller (SMC), and an adaptive controller. All presented controllers seemed to fulfill the preset requirements for position reference tracking regardless of the case considered. The

SMC and adaptive controllers had a significantly smaller error when compared to the PD, which produced some small constant errors. These minor errors could be attributed to the absence of an integral action, as the PID controller presented in previous work [75] showed better performance that closely resembled that of the nonlinear controllers. However, the improved performance for the SMC and adaptive controller came at a high cost of control effort as the cylinders needed to be activated more times. The PD and SMC were concluded to be the most straightforward methods for implementation as they only required tuning for two and three parameters, respectively. However, the SMC parameters were suggested to be adjusted carefully to achieve the required performance to energy consumption ratio. In the case of the adaptive controller, it was noted that, depending on the operation, it could cause the system to become unstable and a safety function should be included to ensure parameter convergence.

4.4 Electrical-Systems

An offshore riser tensioning system for offshore drilling was considered in [10] by Yin Wu. Here, the system compensated for heave motion directly with an electric winch driven by a permanent magnet synchronous motor (PMSM) and a passive heave compensation system. A linear-quadratic Gaussian controller was proposed to regulate the motor's position, which was controlled using field-oriented control. The scheme was tested with simulations under the presence of heave motion, caused by irregular waves with an amplitude of 4 m and an unmodeled disturbance force with a peak of 300 kN. The controller's performance was hard to evaluate, as no graphs were provided to show the resulting wire tension or the tension error. Instead, graphs showing the load tension and the corresponding tension delivered by the hydro-pneumatic and the electric motors were offered. It could be observed that the PHC could not wholly compensate for heave motion, which resulted in tension variation on the wire that deviated as much as 1800 kN from the required value of 15.33 MN. The electric motors could therefore reduce the maximum tension variation by approximately 80%. The scheme was validated on a test bench with a single PMSM and a DC motor acting as a load.

Chupina and Usoltsev [50] developed a fuzzy control scheme for vessel descent-rise devices. Here heave compensation was achieved by an electric motor that adjusted a derrick nodding boom. As vector control was utilized to control the induction motor, a PID controller ensured constant flux. At the same time, the fuzzy controller compensated the heave motion by regulating the motor torque. Simulation tests compared the step response of the proposed controller with that of a modal controller. The results illustrated an overshoot reduction from 18% to 0%, while the settling time was reduced by 33%.

Huang et al. [76] presented a control scheme for achieving AHC with a PMSM. The suggested strategy was a cascade controller for the speed, position, and current of the actuator. As the motor torque depends directly on motor current, a current controller with a fast response time was required. For that purpose, a PI controller was chosen with feedback from a hall sensor. The speed controller provided the reference and was also realized as a PI. A proportional position controller was included in the outer loop to ensure that the load accurately followed the trajectory. The control scheme was tested with simulations on three cases where the load was lifted, lowered, or kept fixed. Throughout all the tests, it was shown that the controller was able to reduce the load oscillations to near-zero values. However, it is difficult to evaluate the control performance, as there is no comparison with similar work. Additionally, it is unknown if several other parameters were considered for the simulation, as no information was given regarding wire dynamics, loading conditions, and external disturbances.

5 Discussion

An investigation for specific electric and hydraulic drives characteristics concerning reliability, maintenance, performance, and cost was presented. The results indicated that electric drives are becoming more prominent due to their high energy efficiency and accurate torque/speed control. At the same time, they may require less maintenance, but system restoration will be challenging and expensive. In contrast, hydraulic drives are still widely preferred for their robustness, low maintenance costs, and ability to produce large torques. When considering the overall cost of a winch drive, there seems to be no consensus. Hydraulic drives are often suggested as having a lower initial price, while electric drives are expected to offer reduced costs over time due to energy savings. However, several parameters will affect the overall cost, which will significantly depend on the application. An analytical study that would provide an overview of the expected costs of winch drive systems is missing from the literature.

Since the shift towards electric winch drives seems to be mainly motivated by the energy savings they offer, it could be concluded that DDMs are a strong candidate for improving the energy efficiency of hydraulic drives. Currently, digital displacement motors are regarded as expensive [16] and challenging to control [41]. As such, there has been limited research that considered them as winch actuators [22, 73–75].

In the reviewed literature, it was evident that the contribution towards improving winch drive topologies is limited. In contrast, several contributions are made towards improving the control methods, with most articles focusing on AHC. It was observed that hydraulic systems attracted more attention as very few studies considered electric winch drives. Multiple reports presented conventional PID controller alterations by adjusting the proportional gain or supplementing the controller with other elements such as a feedforward signal. This approach has the added benefit of requiring little to no changes in the systems currently used by the industry, as PID methods are widely in use. Several nonlinear controllers were suggested, especially for the hydraulic drives. This was attributed to the hydraulic systems' multiple nonlinear characteristics, which would benefit from a nonlinear controller. Even though numerous studies compared PID methods with more complex controllers, this was mainly limited to simulation tests [2, 22, 29, 65], and the results are not in agreement. Some experimental comparisons were provided by [64], but they were unfortunately carried out on a test bench that did not allow for load placement. A comparison with an advanced testing facility could provide further insight into the possible benefits of using advanced controllers.

In general, experimental results were limited in the reviewed articles. Certain studies preferred scaled-down models such as the one used in [11, 55–57], while other experimental setups consisted of a hydraulic motor directly connected to a load motor [74]. Even though tests like these are an essential intermediate stage to the realization of a controller, they do not provide sufficient insight for control performance. On the other hand, full-scale experiments require a lot of space and resources. As such, full-scale results were very few [1, 16, 71], and in all the cases, the setup was provided by an industrial partner. It is, therefore, suggested that the development of an elaborate simulation model that has been verified experimentally and available publicly could offer a viable, affordable, and effective solution for testing winch drive controllers.

6 Conclusions

This paper has presented a short review of various articles found in the literature that proposed new topologies or control methods for offshore winch drives. The review was focused on publicly available academic papers, and it did not consider research conducted by the industry. A brief description of the offshore environment and its effects on winch drives was given, followed by a short presentation of the conventional winch drive topologies along with recent topology improvements. Hydraulic and electric drives were compared concerning reliability, maintenance, performance, and cost. Several control schemes were presented with respect to the topology that they considered. It was concluded that there is limited available work on winch drive topology improvement while there is a significant research interest on developing control methods. Suggestions for future work included an overall winch drive cost evaluation, experimental controller comparison, development of an elaborate simulation winch drive model, and further work on actuation with digital displacement machines.

Nomenclature

Abbreviation	***Full Name***

ADRC	Active Disturbance Rejection Controller
AHC	Active Heave Compensation
DDM	Digital Displacement Machine
HPU	Hydraulic Power Unit
LQI	Linear Quadratic Integrator
MPC	Model Predictive Controller
MRU	Motion Reference Unit
PHC	Passive Heave Compensation
PID	Proportional Integral Derivative
PMSM	Permanent magnet synchronous motor
RL	Reinforcement Learning
SMC	Sliding Mode Controller
VFD	Variable Frequency Driver

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