Coupling a Standard Hydraulic Valve and Advanced Control to Achieve a Motion Compensation System

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This research highlights how a standard hydraulic valve can be coupled with a model-predictive controller (MPC) to achieve motion compensation. The system tested in this research uses a common hydraulic proportional valve where the inherent hysteresis, dead-band and non-linear properties of the valve have been successfully overcome to control a radial piston motor that emulates an unloaded winch. Additionally, the MPC controller uses a set-point prediction algorithm and the results are compared to a classical Proportional-Integral-Derivative (PID) controller. Furthermore, a MATLAB Simulink model of the hardware arrangement is developed to further examine the control performance as well as predict the theoretical system efficiency. It was found that, with constant parameters for the MPC and PID systems, the MPC controller performed better than the PID controller over the range of operating conditions tested. As a result of this work, it is foreseen that the MPC controller and the corresponding signal modifiers have the potential to replace complex hydraulic circuits or components for a wide range of marine applications. The implementation could be used as an add-on to a standard winch to achieve active heave compensation and even help to mitigate anti-pendulum/sway in future systems.

Index Terms—**Active Heave Compensation, Modelling and Control, Predictive Systems**

I. Introduction

A great deal of research has focused on advancing vertical motion compensation systems for marine operations [1]. Many of the modern motion or active heave compensation (AHC) systems use specialized hydraulic components and circuits. The focus of the work presented in this research began with the idea of utilizing common hydraulic components with an advanced control system such that any valve-based actuator or winch could be used as an AHC unit. By using common hydraulic components: a proportional valve, radial piston motor and a swash-plate pump, it is not expected that the system would be efficient but it would be economical from a materials/cost standpoint.

The idea started with the 2007 statement of Hatleskog and Dunnigan [2] who mentioned that a predictive controller may be helpful for an AHC system to achieve 100% effectiveness. The exact rational of how the prediction system should be implemented is not stated; however, it is conceivable that a predictive controller could be useful when lags exist between measurements and commanded motion, such as a marine AHC system. Furthermore, Hastlekog and Dunnigan also stated that the heave motion of a vessel is "...essentially unpredictable with a high probability of significant predictive error". With these statements in mind a focused review of the literature was conducted by the present authors to see how one could construct a predictive controller with common hydraulic components for the purpose of an active heave compensation system. The current paper examines a possible option to help correct for large phase lag using model-predictive control (MPC) while preserving system flexibility with a simple 4-way, 3-position proportional valve on a hydraulic open-loop circuit.

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In 2006 Halliday *et al.* [3] provided a method of using Fast Fourier Transforms (FFT) to accurately predict wave motion within 10%, 10 seconds into the future when the measurements were taken up to 50 metres away from the point of interest. The original intended use of Halliday *et al.*'s short-term wave prediction method was to increase the efficiency of wave-energy collectors; however, their work could also be useful to predict short-term ship motion data from an inertial measurement unit (IMU) commonly used in AHC systems. Using a linearized model of the crane dynamics with the pole-placement control method, Neupert *et al.* [4] used a short-term prediction algorithm as part of a control methodology for an AHC crane. Neupert *et al.*'s [4] simulation research show that their controller can track a step-input to within $\pm$ 3 cm with a ship heave motion of approximately 0.5 m. Kuchler *et al.* [5] carried this work further and found that one could reduce the energy in the load by 83% when the predictive controller was utilized.

Upon a review of AHC literature, it was revealed that an MPC controller in combination with wave prediction has yet to be implemented. MPC solves a quadratic optimization problem at every control action while relying on a system model. Until recently, MPC was typically used for larger slower systems, such as process plants and HVAC systems; however, with the improvement of computer control, MPC can now be used for embedded real-time control applications such as AHC. Known as 'previewing', MPC can use a prediction algorithm to react to upcoming reference changes before they occur.

In the current work, an MPC controller with a wave prediction algorithm is implemented to actuate an unloaded hydraulic testbed and the results are compared to a PID controller commanding the same experimental test arrangement. Section II outlines the hydraulic system used and how the limitations of a common 4-way, 3-position proportional hydraulic valve were overcome. This section also highlights a computer simulator of the hardware that was used to investigate the system efficiency. Section III outlines the control methodology throughout the research, while Section IV shows the Benchmark motion reference data used in this research along with the additional Test Case data. Section V compares the performance of MPC and PID controllers on the experimental setup and the computer simulations. Section V, additionally, assesses the theoretical mechanical efficiency of the system using the computer simulator. Section VI concludes the paper and presents some future avenues of work.

II. SYSTEM HARDWARE

Figure 1 presents a schematic of the open-loop hydraulic AHC testbed used in this work. The pump is an Eaton PVM141ER load-sensing pump that provides flow to the Danfoss PVG-120 4-way, 3-position proportional valve. An analog voltage signal controls the valve that directs the flow to and from the Black Bruin BB4-800 hydraulic motor that is instrumented with an encoder for the controller feedback.

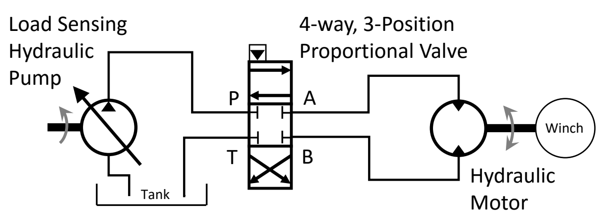

Fig. 1: A hydraulic schematic showing the experimental AHC testbed layout.

Often proportional valves such as the Danfoss PVG-120 exhibit a hysteresis which results from flow forces acting on the spool, residual magnetism of the solenoid armatures, and the inertia of the spool itself [6]. To combat the hysteresis, a dither signal was superimposed on the valve's analog control signal. The sinusoidal dither signal had a peak voltage V_{pk} of 0.3 V and a frequency of 50 Hz. This dither signal was found to effectively reduce the valve's hysteresis while allowing the spool to travel further, thereby maximizing the flow to the motor and allowing for a 15% increase in the maximum angular velocity.

Unlike linear servo valves, proportional valves also exhibit deadband and non-linear gain. Physically, the deadband corresponds to a range of spool travel over which fluid flow is blocked from travelling from the pump to the valve outputs and the motor. For the proportional valve tested in this research, the deadband corresponded to control voltages from 5.37 V to 7.12 V, with control voltages less than 5.37 V opening the valve in one direction allowing for a clockwise rotation, and voltages above 7.12 V opening the valve in the other direction producing a counterclockwise rotation. To circumvent the deadband a suitable offset was added to the valve control voltage to effectively skip the voltages corresponding to this deadband region enabling the controller to function as if the deadband did not exist.

Figure 2 presents a schematic of the resulting motor's angular velocity in revolutions per second (rps) as a function of control voltage after the deadband has been corrected. The linear response transfer function $(\omega[rps]/V[voltage])$ was found to be a first order system with a corner frequency close to 24 rad/s. The valve's non-linear gain is demonstrated in this schematic at ± 0.68 V by the deviation between the linear transfer response and actual system response.

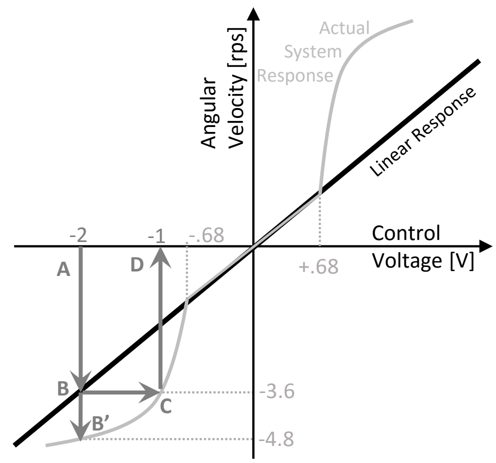

Fig. 2: The motor angular velocity response as a function of valve control voltage. The valve deadband has been removed. The labels A to D show a simple method of correcting the nonlinear gain.

In order to linearize the system response outside this ± 0.68 V range, an algorithm was developed to make the needed adjustments to the valve control voltage signal. For example, referring to Figure 2, imagine the case where the intent is to apply a control signal of -2V to the valve such that the valve behaves linearly and yields a corresponding motor angular velocity of -3.6 rps at point B. In reality, however, this -2V control signal would produce an angular velocity of -4.8 rps as shown by point B′ instead of the desired -3.6 rps. For this case, the algorithm determines that the desired -3.6 rps actually corresponds to a control signal of -1V and generates this voltage signal instead. [7]

The addition of the 50 Hz dither, ± 0.68 V deadband correction and non-linear gain adjustment effectively helps enable the standard proportional valve to imitate a more sophisticated linear servo-valve.

A. Simulator

Given that the experimental setup cannot operate under loaded conditions, experimental power results could not be obtained nor could the controllers be tested in the presence of a load. To help further characterize and quantify the performance of the controllers, a MATLAB Simulink model of the experimental testbed was first validated for the experimental no-load case and then used to provide insight to the corresponding power demands under loaded conditions. A Simscape/SimHydraulic fixed-displacement hydraulic motor is used to model the BB4-800 motor and additional orifices are added to match the manufacturer-specified leakage characteristics [8]. The 4-way, 3-Position Proportional Valve in Figure 1 is a PVG-120 proportional valve and is modelled as a set of four orifices, corresponding to output ports A and B, the pressure port P, and the tank port T. The valve orifice to port T was modelled using the manufacturer specification sheet [9]; however, fluid flow path from P to A/B was modelled from experimental data of the motor velocity as a function of the valve control signal. The Eaton PVM141ER pump is represented as a PID control system which maintains the pressure drop across the load-sense PVG120 valve as a function of the control signal to the valve. The pressure-voltage relationship used to tune and emulate the pump was measured experimentally. A complete analysis of the tuning, construction and validation of the sophisticated Matlab Simulink simulator and the corresponding hardware can be found in [7]. The system's controller logic is described in the following section.

III. Control Methodology

The block diagram in Figure 3 outlines the controller logic used to implement the MPC and PID controllers. The Controller block (MPC or PID) outputs a control signal based on the error signal from the desired setpoint and the actual motor angle. The current work does not make any recommendations or analysis of how to obtain the appropriate setpoint signal. Rather, the current work only acts on the prescribed signal with a delay of 0.25 seconds. This research uses the heave prediction algorithm developed by Kuchler *et al.* [5] that acquires data from the prescribed set-point signal. The PID controller does not use the heave prediction algorithm; however, the MPC controller uses the output of the heave prediction algorithm as a form of previewing. The previewing allows for the controller to respond preemptively based on future setpoint changes and a system model.

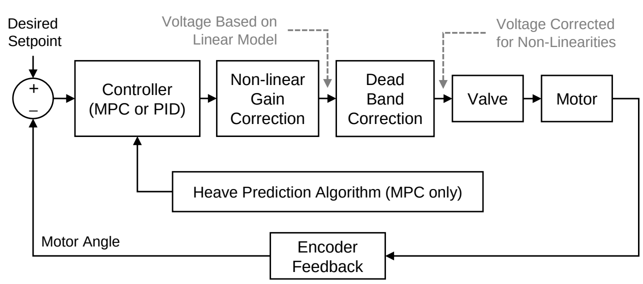

Fig. 3: The valve and motor system are nonlinear. The nonlinear gain correction and deadband correction blocks adjust the controller output such that the valve and motor system appears linear to the controller.

The PID and MPC controllers assume that the system has a linear response; however, the system is not linear. In Figure 3 the 'non-linear gain correction' and the 'dead band correction' blocks adjust the controller output voltage so that the valve and motor system emulate a linear servo system and follow the linear response shown in Figure 2. An encoder provides the hydraulic motor angular position for the feedback signal.

A. Controller Implementations

The MPC controller was designed using the identified system and the control output was determined through quadratic minimization of the following cost function J:

$$J = \sum_{i=0}^{N_p} x_i^T Q x_i + \sum_{i=0}^{N_c} u_i^T P u_i + \sum_{i=0}^{N_c} \Delta u_i^T R \Delta u_i \qquad (1)$$
$$x_{min} \leq x_i \leq x_{max}$$
$$u_{min} \leq u_i \leq u_{max}$$
$$N_c \leq N_p$$

where Q, P, and R are unitless weighting parameters, x is the model state error, u is the controller output, and Δu represents the rate of change of the controller output. The prediction horizon N_p allows the model to evolve N_p time steps into the future, while the control horizon N_c determines how many time steps into the future the control action is calculated and, in this work, a 50 msec time-step is used. When $i \geq N_c$, the controller output u_i and rate of change of the controller output Δu_i are held constant within the cost function. The values of the MPC parameters used throughout this work can be found in Table I.

Following the approach taken by Kuchler *et al.* [5], applying an FFT on previous ship motion data can be used to help predict heave motion into the future. The FFT provides the dominant wave modes while a state estimator is applied to determine the amplitude and phase of each mode. The resulting future heave action is then used in conjunction with the MPC controller to try to improve the control action.

The upper plot of Figure 4 shows sample heave data from [5] with a dotted line, and the heave prediction data from the implementation of the heave prediction algorithm as a solid line where the prediction algorithm begins at 32 s. Notice that, as time progresses, the prediction data deviates from the actual test case data. Figure 4 shows this prediction at a single time-step of the heave data and, as expected, moving forward in time from the start of the prediction reduces the prediction accuracy as indicated by the curves diverging. This divergence occurs because the system is not completely predictable. The lower plot of Figure 4 shows the error between the heave prediction and the actual heave data at 0.25 s (5 time steps) in the future at each time-step, with the average error being 0.0120 motor revolutions. The large initial error in lower plot of Figure 4 is due to the controller observer states not having converged until 3 seconds.

The MPC controller was then compared against a common PID controller. The corresponding PID controller was implemented using the following equations:

$$\begin{aligned}
e(k) &= SP(k) - PV(k) \\
u_p(k) &= K_p e(k) \\
u_i(k) &= u_i(k-1) + \frac{K_p}{T_i}\Big(\frac{e(k)+e(k-1)}{2}\Big)\Delta T \\
u_d(k) &= -K_p \frac{T_d}{\Delta T}\Big(PV(k) - PV(k-1)\Big) \\
u(k) &= u_p(k) + u_i(k) + u_d(k)
\end{aligned} \qquad (2)$$

where

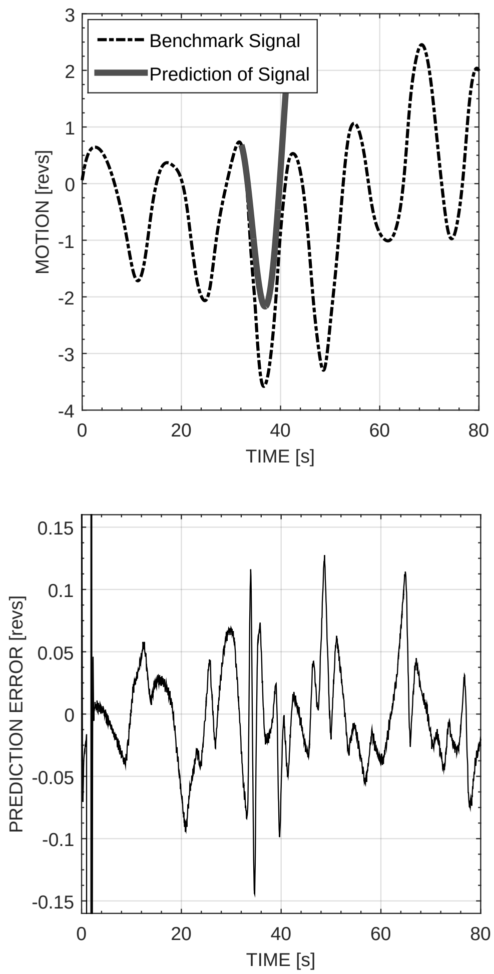

Fig. 4: TOP: Heave Data with a representative wave prediction curve at t=32s plotted for 10 seconds in the future. BOTTOM: Error between actual Benchmark Data and wave prediction values for the Heave Data at 0.25 s in the future (5 time steps).

$SP(k)$	=	Setpoint at the current time step, k [revs]
$PV(k)$	=	Process variable at the current time step, k [revs]
$e(k)$	=	Error between setpoint and process variable [revs]
K_p	=	Controller gain [V/rev]
T_i	=	Integral time constant [min]
T_d	=	Derivative Time Constant [min]
ΔT	=	Control Loop Time [min]
$u_{p,i,d}(k)$	=	Proportional, Integral, and Derivative control terms [V]
$u(k)$	=	Controller output [V]

and the PID gains and values presented in Table I were manually tuned to minimize the integral of the error of a step response [7]. All of the tuning parameters for both PID and MPC were held constant throughout all experimental and simulated tests.

TABLE I: Controller Parameters

MPC Parameters:	Value	Units
Q	10	[–]
P	0	[–]
R	0.5	[–]
N_p	4	[–]
N_c	1	[–]
PID Parameters:	Value	Units
K_p	8	[V/rev]
T_i	5×10^{-3}	[min]
T_d	5×10^{-4}	[min]
ΔT	8.33×10^{-4}	[min]

IV. Test Cases

Figure 5 shows the two heave motions (a Benchmark and a Test Case) that were used to compare the MPC and PID controllers within this work. For each data set there are two axes: the left axis indicates motor rotation in revolutions (revs), while the right axis is the equivalent heave motion in meters, assuming that a 16 inch (0.4046 m) diameter winch drum is attached to the hydraulic motor.

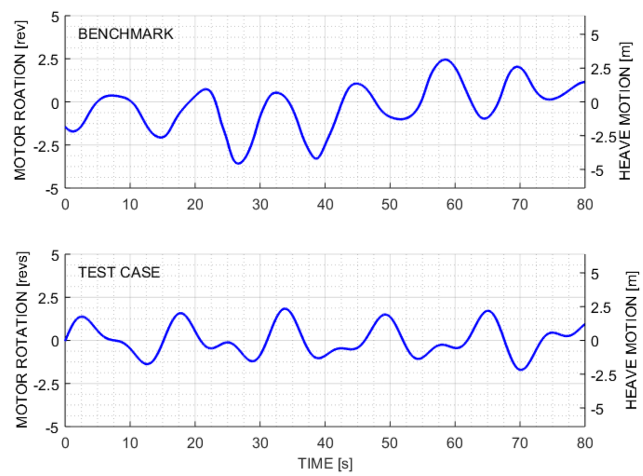

Fig. 5: Two test cases shown here were used to compare MPC and PID controllers tracking a moving heave motion reference. The top plot is digitized from Kuchler *et al.* [5]. The bottom plot was artificially generated for this work.

A Benchmark heave motion data set was digitized from Kuchler *et al.* [5] so that publicly-available data could be used to try to help standardize AHC comparisons and various controller implementations between researchers. The Test Case data was artificially generated to represent motions a vessel might experience while at sea. The Test Case data has prominent frequency modes at 6.67×10^{-2} Hz, 1.00×10^{-1} Hz, and 1.22×10^{-1} Hz. The controller parameters used in the experiments were tuned by minimizing the tracking error for the Benchmark Case and the resulting tuned parameters are summarized in Table I. The MPC controller gains that were established using the Benchmark data were then held constant when applying the Test Case data for both the simulation and experimental work. The PID controller gains were tuned using a standard step-response methodology.

V. RESULTS

To compare the PID controller to the heave predictive MPC controller, each controller was used to experimentally track the two motions from Figure 5 using the AHC testbed described in Section II. LabVIEW software was used to implement the PID and MPC controllers for the experiments. The corresponding standard deviation results in units of revolutions for both controllers are tabulated in Table II.

Figure 6 shows a sample of the results for the PID system tracking the Test Case data while Figure 7 shows the results of the MPC system for the same scenario. The upper plot in these two figures shows the reference set-point data as black circles, the simulator predictions as blue lines and the experimental results as red dashed lines plotted as a function of time. The left axis is the testbed motor rotation in revolutions [revs], while the right axis shows the equivalent vertical motion of the ship assuming the same 16 inch (0.4046 m) diameter winch drum. Since the set-point data, predicted response and actual system response are difficult to distinguish, the lower plot in each figure shows the corresponding differences between the simulation and experimental reference tracking as a function of time.

The metric used in this work to compare the results is the standard deviation of the error plots between 5 and 80 seconds (to omit any initial transients). Smaller values of standard deviation are an indication of better controller performance. For the experimental results using the Test Case data, the PID error standard deviation shows that the tracking error falls within $\pm1.65\times10^{-2}$ revs (±2.11 cm) 68.2% of the time, while the experimental results of the MPC error standard deviation shows that the tracking error falls within $\pm1.31\times10^{-2}$ revs (±1.67 cm) 68.2% of the time. The difference between experimental PID and MPC error standard deviation for the Test Case data is 3.40×10^{-3} revs which corresponds to 4.3 mm (or a reduction in load motion of 8.6 mm when using MPC instead of PID control).

Using the standard deviation as the comparison metric for the experimental Benchmark PID controller case, the tracking error falls within $\pm1.88\times10^{-2}$ revs (±2.40 cm) 68.2% of the time while, for MPC control, the tracking error falls within $\pm1.30\times10^{-2}$ revs (±1.66 cm) 68.2% of the time. This $\pm5.8\times10^{-3}$ revs (±7.4 mm) difference actually represents a reduction of 14.8 mm of motion. These experimentally-observed improvements when using MPC compared to PID for the Benchmark and Test Case data could suggest that the MPC controller is more robust to changing conditions when compared to the PID controller. This robustness is a desirable trait for a controller as it allows the same controller to be used under multiple heave motion conditions without needing to modify any controller tuning parameters. Future work will examine the robustness of the MPC system and how it relates to the bandwidth of the system.

In Figures 6 & 7, the plots also demonstrate how well the simulator was able to emulate the experimental results. Examining the PID controller simulations using the Test Case data (Figure 6), the simulator error signal standard deviation is 1.41×10^{-2} revs (±1.80 cm) which is very similar to the experimental error signal standard deviation of 1.65×10^{-2} revs (±2.11 cm) — representing a difference of 6.2 mm (±3.1 mm). It is interesting that the simulator is not only able to accurately emulate the motion and tracking of the system for this case but also capture the generalized trends of the error between the reference signal and the system response as a function of time for the PID system in Figure 6. Examining the corresponding MPC results in Figure 7, however, it can be seen by the lower plot that the simulator does not predict the trends in the error nearly as well as the PID implementation. The MPC simulator error signal standard deviation is 2.16×10^{-2} revs while the experimental error standard deviation is 1.31×10^{-2} revs. With the excellent agreement between simulation and experiment of the PID controller, the observed discrepancy for MPC is likely due to the experimental implementation of MPC in LabVIEW differing from the simulated MPC implementation in MATLAB

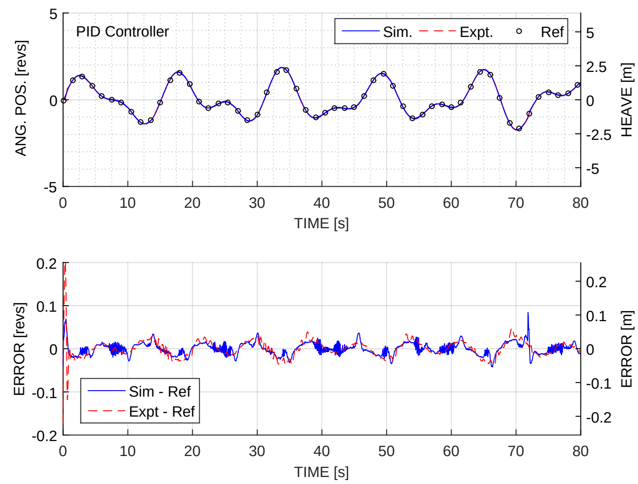

Fig. 6: PID system; TOP: Results of the simulation and experimental tracking reference signal of the Test Case ; BOTTOM: Error, $Sim - Ref$ and $Expt - Ref$

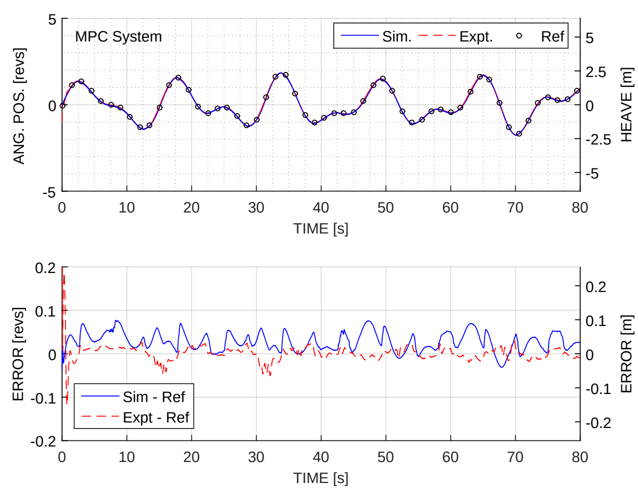

Fig. 7: MPC system; TOP: Results of the simulation and experimental tracking reference signal of the Test Case; BOTTOM: Error, $Sim - Ref$ and $Expt - Ref$

TABLE II: Standard Deviation Results [revs]

	SIMULATION	EXPERIMENTAL
PID		
BENCHMARK	1.60×10^{-02}	1.88×10^{-02}
TEST CASE	1.41×10^{-02}	1.65×10^{-02}
MPC		
BENCHMARK	2.83×10^{-02}	1.30×10^{-02}
TEST CASE	2.16×10^{-02}	1.31×10^{-02}

Simulink.

Often the power requirements or usage is reported for hydraulic systems; however, in this research the experimental system was unloaded and, therefore, the power usage was not experimentally obtained. The computer simulator was, therefore, used to theoretically determine the power consumption and efficiency for different loaded scenarios.

A. Power & Efficiency

Many believe that an open-loop hydraulic system always has a very low efficiency — between 10 and 35% [10]. However, these stereotypical low efficiencies are more commonly associated with fix-displacement pumps rather than the variable displacement swash-plate pump used in this work. When using a variable displacement pump and load-sense valve it would not be unreasonable for an open-loop hydraulic system to see efficiency numbers as high as 80%. To achieve efficiencies greater than 80% one must normally use a dedicated closed-loop hydraulic circuit [11] or specialized systems such as the Bosch Rexroth secondary control drives [12]. These specialized hydraulic systems are typically more efficient (as they may use less power and energy) but often have a high initial cost. The more simplistic open-loop hydraulic circuit used in this research does not use any highly-specialized components, manifolds or circuity, thus, the cost is significantly lower than other active heave compensation systems. Unlike some hydraulic closed-loop systems the circuit used in this work retains the full manual lever control which is often a desirable feature for emergency recoveries and procedures. Furthermore, the digital control system developed in this work could be applied to any electrically-actuated proportional valve as an upgrade or retro-fit.

The previously-discussed controllers and the experimentally-validated simulator was used to investigate the efficiency of the system under various operating conditions – load, speed and direction. Figure 8 plots the simulated efficiency of the system as a function of line speed for the following constant load cases: 8900, 4450, 2225 and 1112 N (2000, 1000, 500 and 250 lbs). A negative line speed indicates that the winch is reeling out cable while a positive line speed indicates the winch is reeling cable onboard. Within the simulator the motor and valve leakage has been accounted for. The pump has been modeled as the identified load and pressure-compensated flow source [7]. In Figure 8, the system input power for the efficiency calculation was computed from flow and pressure exiting the pump and the system output power was computed by the line tension and velocity.

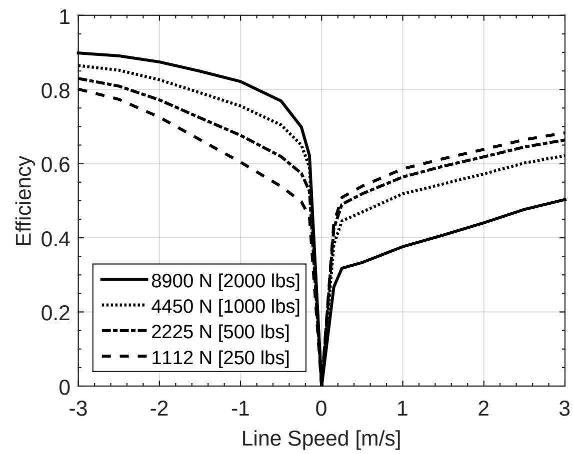

Fig. 8: Theoretical mechanical efficiency as a function of line speed

During the reel-out operations when the load is assisting the winch with active heave compensation operations, the system reached a simulated efficiency of 90% for the highest load case of 8900 N (2000 lbf). This peak efficiency decreased as load decreased. The lowest load case of 1112 N (250 lbs) still had a peak efficiency of 80%. While these efficiencies are high for an open-loop hydraulic circuit, it is not unexpected as this high efficiency is only achieved when the load end of the cable is assisting the winch in the reel-out direction.

During the reel-in operations of the highest load case of 8900 N (2000 lbf) (when the winch direction is opposing the cable load) the efficiency drops to a low of 26% at low speed before climbing to 50% at the system's maximum speed. For the lightest simulated load case of 1112 N (250 lbs) the efficiency drops to a low of 44% at low speed before climbing up to 68% at the system maximum speed. Over the wide range of line speeds ±3 m/s (±9.8 ft/s) and load cases of 8900, 4450, 2225 and 1112 N (2000, 1000, 500 and 250 lbs) simulated, the overall average efficiency is estimated at approximately 60%. It should be pointed out that the system was never designed or optimized for efficiency. Rather, the primary focus was to investigate the feasibility of using a common and inexpensive hydraulic system with an advanced control system to replace or retrofit active heave compensation systems at the expense of the power requirements and efficiency. A loading mechanism will need to be constructed and quantified to properly validate these theoretical efficiencies within the simulator. Furthermore, the simulator can also be used to examine the strengths and weaknesses of the control methodology for a loaded system.

VI. Conclusion and Future Work

The presented work is of an AHC experimental testbed that uses a common hydraulic 4-way, 3-position proportional valve and drives a hydraulic motor with an open-loop hydraulic circuit. Using a 50 Hz dither signal, the inherent hysteresis of the valve was corrected, while the ±0.68 V deadband and non-linear gain associated with this valve were overcome in the control logic such that the nonlinear valve and motor system appeared linear to the PID and MPC systems.

MPC and PID control was implemented on a computer simulator of an AHC testbed. Corresponding experiments were carried out with the physical system to validate the simulator for two unloaded scenarios. Future work could include validating loaded cases using the simulator as well as experimentally determining the power consumption of the system. Overall, it was found that the MPC system outperformed the traditional PID system for the conditions tested in this research. The PID controller simulation results of the error signal estimates were found to be on par with the experimental results as measured by the standard deviation metric used to quantify the results in this research. The discrepancies observed when comparing the MPC controller simulation results with experimental data are likely due to different implementations of MPC within the different software packages used in this research.

The computer simulator was used to evaluate the system's efficiency. It was found that the MPC and PID controller had minimal effect on the simulated efficiency. The system had a peak efficiency of 90% when a 2000 lbs load was assisting the reel-out of the winch. During real-in operation the efficiency dropped to as low as 25% and as high as 50%. When the efficiency was averaged over all of the speeds, loads and directions tested in this research, the system had a mean efficiency of 60%. The strength of using this system, however, is not for efficiency, but rather for the overall cost effectiveness of using standard hydraulic equipment. Continuing work will examine the accuracy of the theoretical efficiencies obtained from the simulator.

Acknowledgment

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