Controller Design and Motion Compensation for Marine Towed Bodies

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***Abstract*— At sea, a vessel is subjected to waves. If the ship is towing a submerged body containing sensory equipment, then any ship motion at the surface can impart unwanted disturbances on the towed body via the tow line. To help compensate for these disturbances, a winch on-board the host vessel can be operated in response to the surface ship's motion by reeling in or reeling out the tow line. A corresponding method, however, is needed to determine how much cable the winch controller should pay in or out to effectively attenuate unwanted towed body motion. This paper, therefore, proposes and explores four different approaches that, using various combinations of sensor measurements such as ship inertial measurement unit data and measured tow-line angle, can be used to try to establish appropriate winch control actions for motion compensation of marine towed bodies. Small-scale towed body experiments using a spherical tow body as well as computer simulations are carried out to test the control approaches and, by analyzing the corresponding motion reduction achieved by each of these winch control strategies, the most effective method is identified.**

Keywords—Motion Compensation; Towed Bodies; Winch Dynamics; Control Systems; Multi-Body Dynamics; Multi-Domain Modelling

I. Introduction and Problem Definition

While at sea, a research vessel on the ocean surface is subjected to wave actions. The resulting ship motion can be described in terms of ship displacements in six degrees-of-freedom about the vessel's centre of gravity, namely heave, surge, sway, roll, pitch and yaw. If the surface ship is towing a submerged body containing oceanographic sensory equipment, then any ship motion at the water surface can impart disturbances on the submerged towed body via the tether. A compensation method is, therefore, needed to effectively attenuate unwanted towed body motion caused by wave motion at the surface. Motion compensation research tends to focus on vertical heave compensation [1]. Some of the most common applications of vertical heave motion compensation are for offshore drill operation and to stabilize Remotely Operated Vehicles (ROVs). In these scenarios vertical heave tends to be the most dominant disturbance acting on the system and the tow cable is primarily oriented in the vertical direction resulting in a one degree-of-freedom system. As the surface vessel heaves up, the winch must let out line equal to the heave displacement to effectively cancel the motion. Similarly, as the surface vessel lowers, the winch must reel line in equal to the displacement. A corresponding control loop is depicted in Figure **1** which outlines how one can use on-board measurements of wave disturbances with an Inertial Measurement Unit (IMU). In the case of the one degree-of-freedom heave compensation system, minimal processing is required since measurement of vertical motion of the host vessel is directly used as the winch controller's reference tracking signal. The winch controller and an appropriate feedback sensor, such as a winch encoder (to measure the actual length of tow line which has been reeled in or out) completes the control loop.

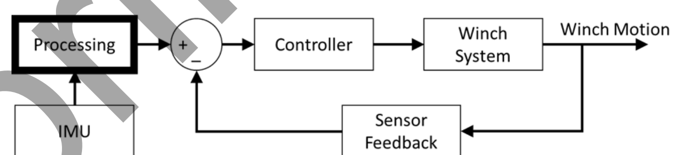

Figure 1: Control loop directing the response of an on-board winch using processed IMU data for a reference signal

As the name suggests, however, towed bodies operate when their host ship is underway and, as waves interact with the surface vessel, the sheave is not limited to simple vertical heave motion. Furthermore, the tow line exits from the sheave tow point at an angle. Figure **2** shows a schematic of a hypothetical research vessel and its towed body experiencing perturbations from wave motion where the cable connecting the towed body to the active heave compensation winch passes over a sheave mounted at the surface vessel's stern.

While a winch can be used to reel in and reel out tow cable, the present authors could not find any published liturature that explains, for a known generalized ship motion, how much cable should be reeled in or out by the winch controller for towed body active motion compensation system. To extend current motion compensation and modelling efforts, this paper, therefore, proposes and investigates four different strategies that can be used with a winch controller to try to effectively decouple ship motion from a towed body. This research focuses on surface disturbances which cause unwanted vertical and forward motion of the towed body and, at this time, does not investigate any lateral motion that may be experienced by the towed body. Also, within this paper the host ship has sustained forward motion and any considerations which might arise due to yaw and sway of the surface vessel are assumed to be negligible. Thus, the surface

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vessel's motions to be compensated for are heave, surge, roll and pitch.

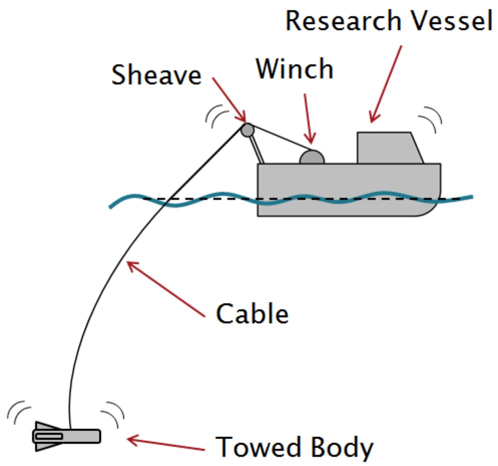


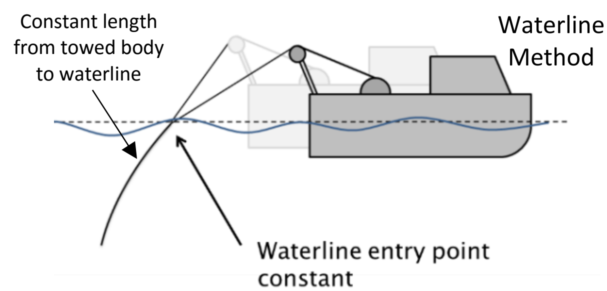
Figure 2: Schematic of a research vessel transferring motion to a towed body via its tether. The tow line passes over the vessel's sheave before crossing the waterline and is attached to the underwater towed body

The following section describes the four different strategies used to determine the winch controller's reference tracking signal while Section III presents preliminary simulation results for these different control signal reference schemes to help evaluate their effectiveness. Section IV outlines the experimental apparatus and tests used to validate the simulator, Section V discusses the tuned simulator results, while Section VI draws conclusions and makes recommendations.

II. Motion Compensation – Control Signal Reference

The four active heave compensation approaches that are presented in this work were selected assuming that the only sensors available at sea are inertial measurement sensors on the surface vessel to measure the ship's roll, pitch, yaw, surge, heave, and sway, a winch encoder to measure the length of cable that has been reeled in or out, and possibly a tow-angle sensor to measure the angle of the tow line as it leaves the sheave to enter the water. While it is possible to equip the towed body with inertial measurement sensors to report its location underwater for additional controller feedback, this towed body motion information is generally not available or feasible to acquire.

Figure **3** illustrates the different methods proposed in this paper to determine the amount of cable the winch motor should reel in or out for active heave compensation. The upper diagram in Figure **3** depicts what the present authors call the "Waterline" methods. These methods attempt to compensate for unwanted towed body motion by trying to ensure that the same point along the cable always enters the water. The lower diagram in Figure **3** depicts what the authors call the "Sheave" methods which determine the desired cable adjustment length based on the motion of the vessel's sheave tow point projected along the tow-line angle. These methods can be implemented with real-time knowledge of the actual tow angle at the sheave – referred to in this paper as "Rigorous Waterline" and "Rigorous Sheave", or without real-time knowledge of the actual tow-line angle at the sheave where a nominal tow-line angle is used and assumed to be constant – referred to as "Simplified Waterline" and "Simplified Sheave".



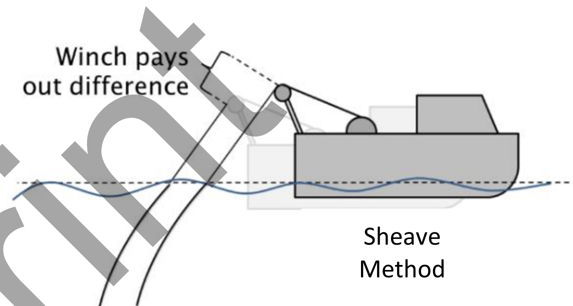


Figure 3: TOP) Schematic of the "Waterline" methods attempting to keep the water entry point of the tow line constant; BOTTOM) Schematic of the "Sheave" methods, which determine the desired cable adjustment based on the motion of the vessel's sheave projected along the tow-line angle

This Simplified Waterline method is depicted in the upper diagram in Figure **4**. With this compensation method, the effects of the surface vessel's heave, surge, roll and pitch on the sheave's vertical motion can be calculated; however, with the tow-line angle unknown, the actual exposed tow-line length cannot be determined. For this case a nominal constant tow-line angle is used to roughly approximate the amount of tow cable exposed so that the winch controller can ensure that the same point along the cable enters the water. The lower diagram in Figure **4** illustrates the Rigorous Waterline method in which the actual tow-line angle is measured in real-time. In this case, both the sheave height and tow-line angle are then known which enables the exposed line length to be fully defined and used by the winch controller to ensure that the same point along the cable enters the water.

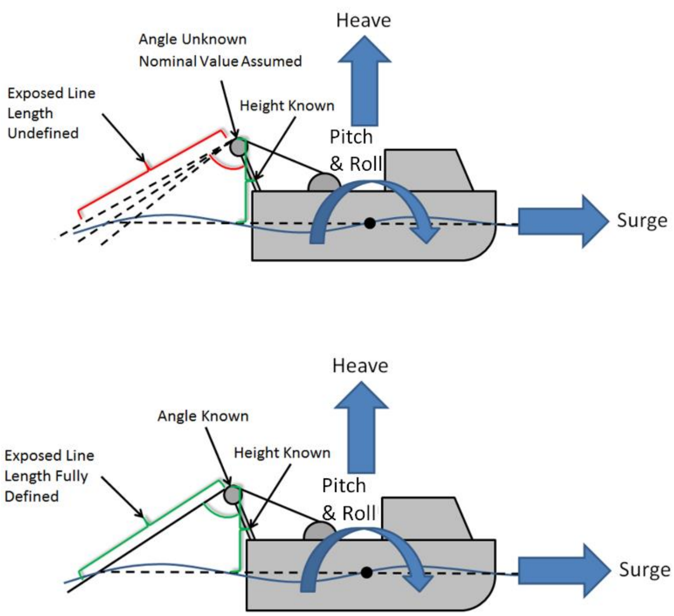


Figure 4: TOP) Schematic of the "Simplified Waterline" method and the approximate value for exposed line length; BOTTOM) Schematic of the "Rigorous Waterline" and the more accurate calculation of exposed line length

The upper diagram in Figure **5** depicts the Simplified Sheave method where the effects of the surface vessel's heave, surge, roll and pitch on both the sheave's vertical and horizontal motion can be calculated. For this case, the tow-line angle is unknown; therefore, a nominal constant tow-line angle is assumed. The resulting displacement of the sheave tow point in the vertical and horizontal directions measured relative to the nominal undisturbed position of the sheave can then be projected onto the assumed tow-line angle to determine the amount of cable that needs to be reeled in or out by the winch controller. The lower diagram in Figure **5** illustrates the Rigorous Sheave method in which, similar to the Rigorous Waterline method, the tow-line angle is measured in real-time. As a result, the displacement of the sheave tow point can be projected onto the actual tow-line angle to determine the reference that the winch controller needs to track.

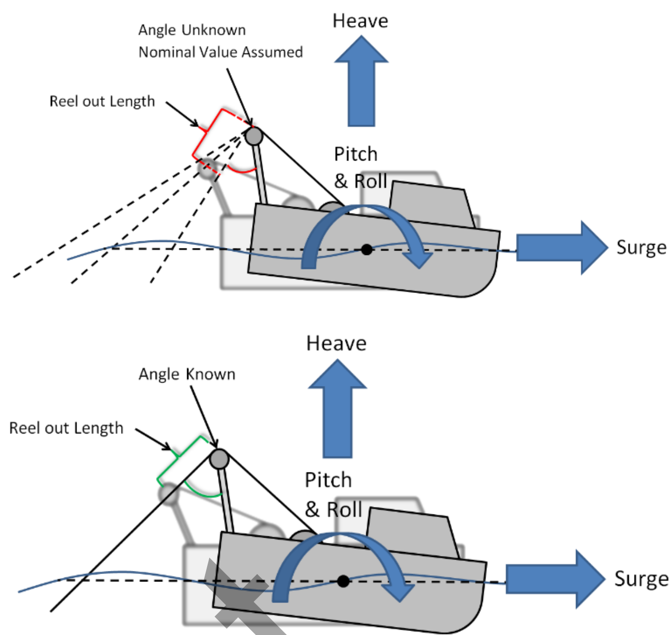


Figure 5: TOP) Schematic of the "Simplified Sheave" method and the approximate value for the reel-out length; BOTTOM) Schematic of the "Rigorous Sheave" method and the more accurate calculation of the reel-out length

III. Preliminary Modelling & Controllers

To investigate the performance of these four different compensation methods, a preliminary computer simulator was developed. To model the flexible tow cable several researchers have used a lumped-mass model. For example, Buckham et al. [2] developed a lumped-mass cable model formulation which was solved using a Runge-Kutta integrator for the DOLPHIN semi-submersible towing vehicle which pulled the AURORA Towfish. Their work pertained to the optimization of the system's design and operation. Driscoll and Nahon **Error! Reference source not found.** also developed a lumped mass cable model for an ocean mooring. Their system was solved using a fourth-fifth order Runge-Kutta technique with an adaptive step size. Sun et al. **Error! Reference source not found.** used a lumped mass cable model to examine directional stability, maneuverability, safety and control characteristics of the towed body. Park et al. **Error! Reference source not found.** used a low-tension cable system with a rotational stiffness to describe the cable.

Similar to the present authors' previous work **Error! Reference source not found.**, the lumped-mass cable model developed for this research was constructed with Matlab's SimMechanics toolbox using a series of SimMechanics body elements – except that, instead of revolute joints, universal joints were used between each body element where rotational stiffness and damping were applied, as in [5]. The system was solved using Simulink's ode45 solver and Figure **6** shows a graphical representation of the cable elements used within SimMechancis.

Building on the work of previous lumped mass cable models [2-5] [2], [3], [4], [5], the drag force on the cable F_{DRAG} was calculated in the x, y and z directions by

$$F_{DRAG_{xyz}} = \frac{1}{2}\rho A C_D v_{xyz}|v_{xyz}|$$

where ρ is the density of the water, A is the projected area of the cable, C_D is the drag coefficient and v is the relative velocity between the cable link and the fluid flow in the x, y and z directions. The net body forces of the cable weight and buoyancy were also added to each link. Currently, the linear cable stiffness and damping are not accounted for in the model; however, future iterations of this research could include these parameters.

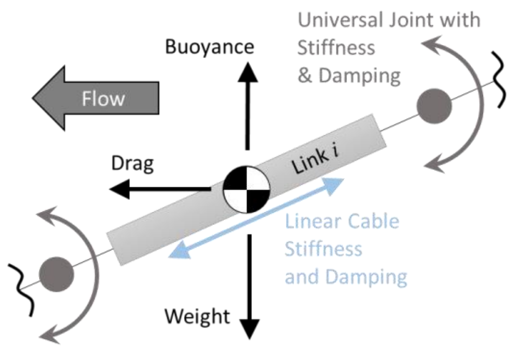


Figure 6: Schematic of individual cable link

The cable model was connected to the towing sheave via a one-dimensional prismatic joint within SimMechanics (with a spherical joint or Uni at the tow point? The prismatic is the actuation). This joint mimicked the cable being reeled in or out by the winch following one of the four compensation methodologies described earlier. It should be pointed out that this preliminary simulator model does not account for any delays or dynamics associated with the winch itself – instead, the desired cable displacements predicted by each of the four methods were directly applied through kinematic inputs to the prismatic joint.

Simulator parameters were selected to reflect a small vessel towing a streamlined, cylindrical body with an un-fared tow line. Twenty cable links were used in the simulation and the key simulator parameters are presented in Table 1.

TABLE 1: TEST SHIP AND TOW PARAMETERS

Parameter	Value	Units
Sheave height above ship CG (z-axis)	2.5	m
Sheave distance behind ship CG (x-axis)	5.5	m
Towed body mass	192	kg
Towed body length	2.36	m
Towed body frontal drag coefficient	0.05	
Tow line length	40	m
Tow line drag coefficient	1.00	
Tow line radius	0.006	m
Tow line density	7850	kg/m^3
Nominal tow speed	3	m/s

For the purposes of an initial investigation into the performance of the four compensation methodologies, simple sinusoidal motion was applied to the sheave tow point and the resulting towed body motion was then assessed. Table 2 provides the corresponding amplitudes and frequencies of the surge, heave, roll and pitch sinusoidal inputs. These motions were applied relative to the vessel's Centre of Gravity (CG). Figure **7** plots the planar x-z view of the resulting motion of the sheave tow point as it responds to the sinusoidal inputs, where x corresponds to the horizontal axis and z corresponds to the vertical axis. The total vertical displacement of the sheave is 1.5 m while the horizontal displacement is 0.75 m.

TABLE 2: SHIP MOTION FOR PRELIMINARY SIMULATION

Motion	Value	Units
Surge amplitude	0.25	m
Heave amplitude	0.50	m
Roll amplitude	7.5	deg
Pitch amplitude	2.5	deg
Surge frequency	0.5	Hz
Heave frequency	0.7	Hz
Roll frequency	0.6	Hz
Pitch frequency	0.5	Hz

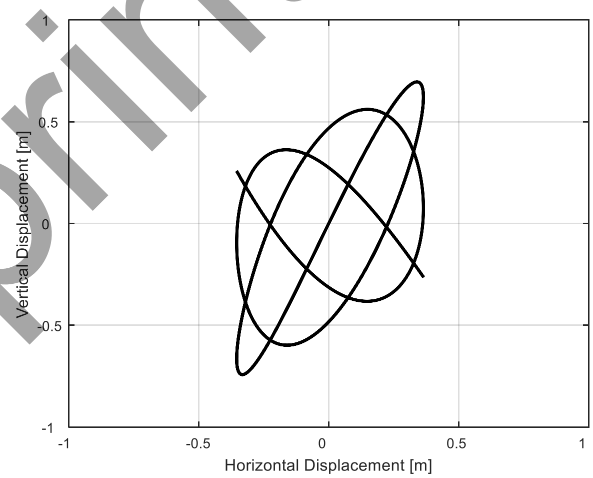


Figure 7: Displacement of the simulated sheave tow point in planar x-z view

Figure **8** plots the planar x-z view of the corresponding towed body motion when no compensator is acting. It can be seen in this figure that the motion of the towed body is amplified and erratic when compared to the two-point input. The uncompensated towed body motion had a vertical displacement of 1.7 m and horizontal displacement of 1 m.

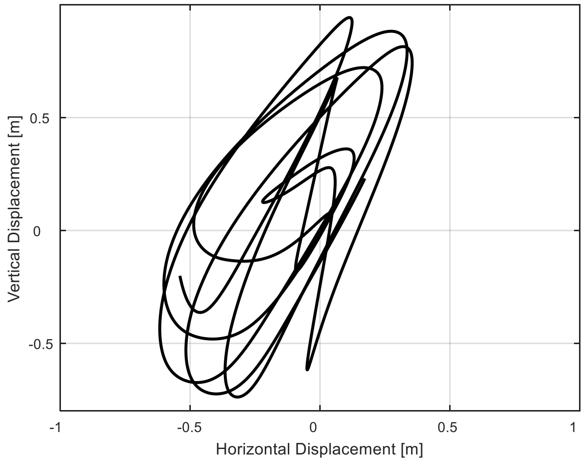


Figure 8: Displacement of towed body without motion compensation in planar *x-z* view

Figure **9** plots a sample of the results of the streamlined cylindrical towed body motion when the Rigorous Sheave compensation technique was implemented. The compensated towed body motion in Figure **8** had a vertical displacement of 0.15 m and horizontal displacement of 0.20 m.

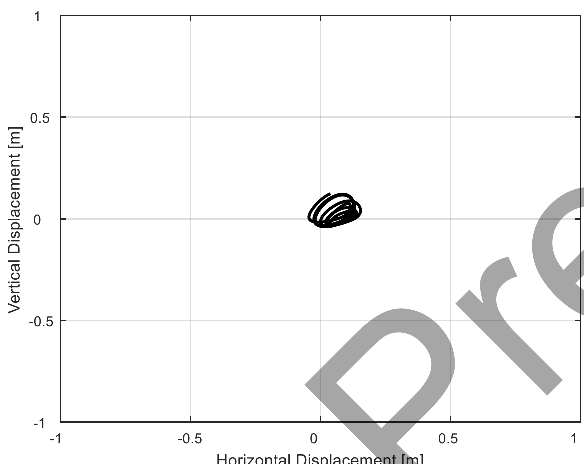


Figure 9: Displacement of towed body under Rigorous Sheave-based motion compensation control in planar *x-z* view

To assess the performance of the various compensation methods, an ellipse was fit around the motion path of the towed body for each case. The ellipse's boundary was positioned such that the area was minimized while still containing 95% of the data. The resulting area of each ellipse was then calculated to compare the motion reduction achieved with each of the four compensation methods as shown in Figure **10**. As seen in this figure, for the uncontrolled case it was found that 95% of the data point fit within an ellipse having an area of 1.203 m^2. The method that reduced the motion the greatest was the Rigorous Sheave method having an ellipse area of only 0.031 m^2 – a reduction of 97% from the uncompensated system. It can also be seen from Figure **10** that the Rigorous Sheave method reduced the ellipse area by 45% when compared to the Simplified Sheave method, while the Rigorous Waterline method reduced the ellipse area by 75% when compared to the Simplified Waterline method.

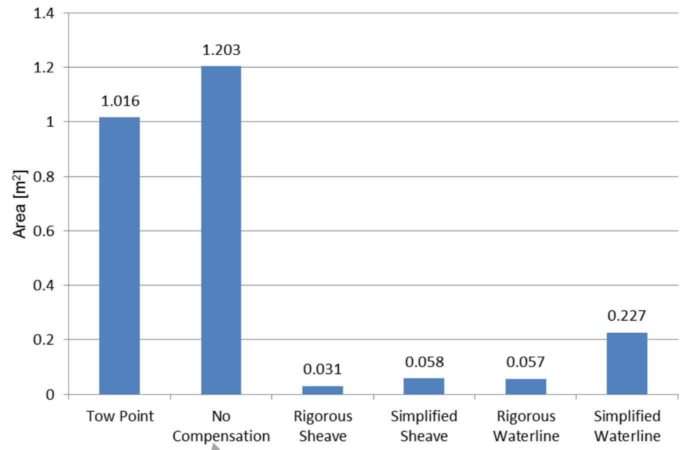


Figure 10: Summary of ellipse areas for the various control methods using preliminary simulations

IV. Flume Tank Experimental Results

To help validate the simulation results, a small-scale test apparatus was developed and used, as shown in Figure **11**, in a recirculating flume water tank. The apparatus was designed using rack and pinion mechanisms to produce repeatable tow-point motion. The resulting test rig had a maximum vertical motion (*z*-axis heave), and maximum horizontal motion (*x*-axis surge) motion of ±4 cm and ±4 cm, respectively. A small winch was attached to the actuated platform with a length of 20 lb fishing line spooled around it to act as a tow line. To measure the tow-line angle, a balanced cable follower with a non-contact absolute encoder was mounted beside the winch. This cable angle measurement was used for controller feedback in the Rigorous motion compensation methods. For the towed body, a sphere was used so that a simple and classical solution could be benchmarked without the additional complexities associated with towed body dynamics arising from more complex shapes.

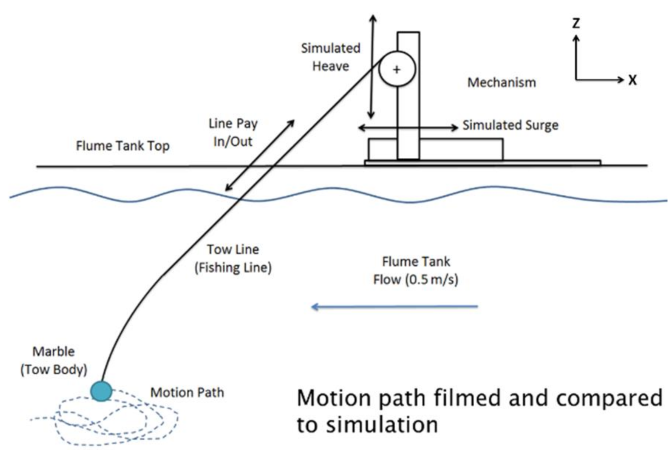


Figure 11: Diagram of test apparatus

The test apparatus was mounted above a recirculating flume tank and the flow profile of the flume tank was measured using a Vectrino Doppler velocimeter **Error! Reference source not found.** and the average fluid flow was 0.5 m/s.

Ship motion data was digitized from an Australian Defence Science and Technology Organisation (DSTO) report **Error! Reference source not found.**. The data was then resolved into three translational degrees of freedom for a towed body winch located at the back of a ship. This motion was then scaled down to allow for appropriate motion of the mechanism as shown in Figure **12**. The resulting vertical (*z*-axis) and horizontal (*x*-axis) motion of the sheave tow point is presented in Figure **11**. The resulting horizontal and vertical motion data include effects from heave, surge, roll and pitch.

For each trial, the test apparatus' actuators tracked this motion path.

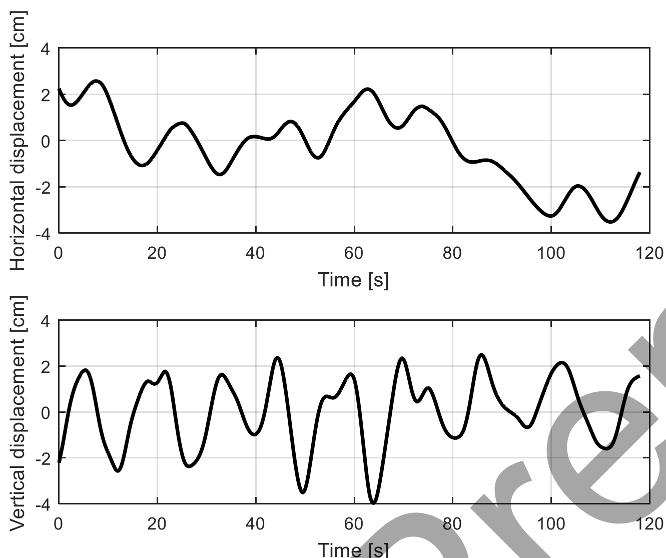


Figure 12: Vertical and horizontal motion of the sheave tow point

Motion of the towed body was captured by a digital camera positioned beside the flume tank's acrylic wall and perpendicular to the flume tank flow. The video footage was decomposed into image frames and image analysis was carried out to track and record the towed body motion over time.

To test the four different compensation methodologies, a fast-responding PD controller was designed to control the winch on the experimental apparatus, while a National Instruments MyRIO controller was used to execute the compensation methodologies at a sampling rate of 1 kHz. The controller gains were tuned to a 90% rise time of 0.12s and the corresponding motor parameters and controller gains are listed in Table 3. The controller output is provided as a PWM signal ranging from 0 to 1, while the winch system output was measured in encoder counts with a resolution of 720 counts per revolution.

TABLE 3: WINCH MOTOR PARAMETERS AND CONTROLLER GAINS

Parameter	Value	Units
Back EMF constant	1.4×10^{-3}	Vs/rad
Coil resistance	5.0	Ω
Rotational inertia	1.0×10^{-5}	kgm^2
Motor inductance	5.0×10^{-3}	H
Rotational friction	1.4×10^{-4}	Nms/rad
Proportional gain	0.0067	
Derivative gain	0.00049	

The ellipse-fitting approach described in the previous section was again used to assess the experimental test results. Figure **13** shows the resulting cluster of towed body positions that result when there is no motion compensation along with the ellipse that encompasses 95% of the data. The cluster of points has been centered about the mean and rotated such that the major axis of the ellipse is in line with the horizontal axis. Figure **14** shows a sample of the motion compensated results using the Rigorous Sheave method. It can be seen in Figure **14** that the ellipse is significantly smaller than the uncompensated case.

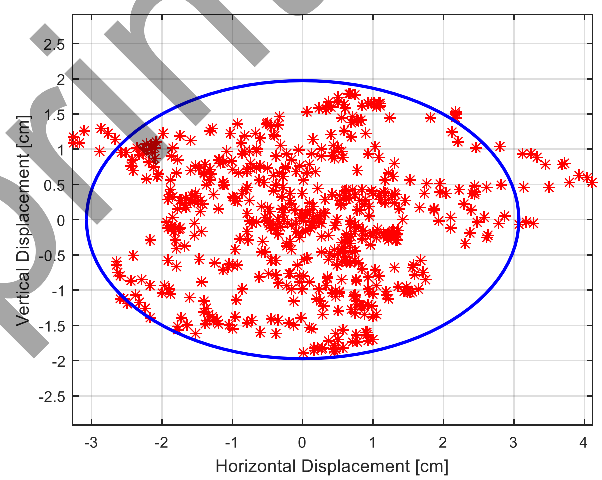


Figure 13: Experimental test result for the case of uncompensated motion

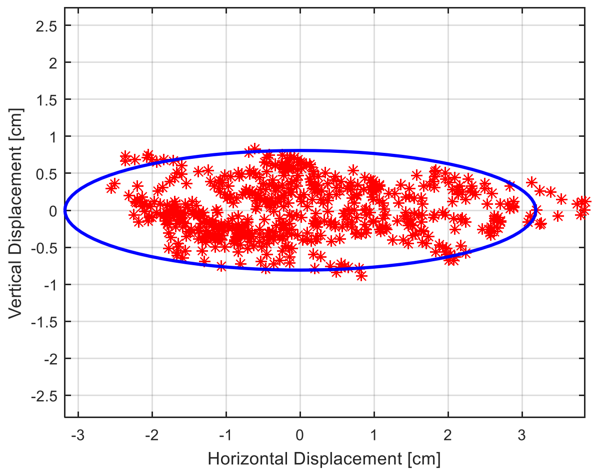


Figure 14: Experimental test result for the case of compensated motion using the Rigorous Sheave method

Figure **15** summarizes the corresponding experimental results showing the ellipse areas for the stationary tow-point mechanism case, the uncompensated case, and each of the compensation methods – except for the Rigorous Waterline method which produced unstable results. For the uncompensated case it was found that the 95% of the data points corresponding to the towed body positions fit within an ellipse area of 19.04 cm^2. The compensation method that reduced the motion the greatest was, again, the Rigorous Sheave method. For this case it was found that 95% of the data fit within an ellipse area of 8.08 cm^2, indicating an overall motion reduction of 58% from the uncompensated system. For these tests, real-time measurement of the tow-line angle offered little improvement. It should be noted that a perfect compensation strategy would ideally keep the towed body to within the same ellipse area as the stationary mechanism case. While not unexpected in physical experiments, none of the compensation methodologies could achieve the stationary mechanism results having an ellipse area of only 2.79 cm^2.

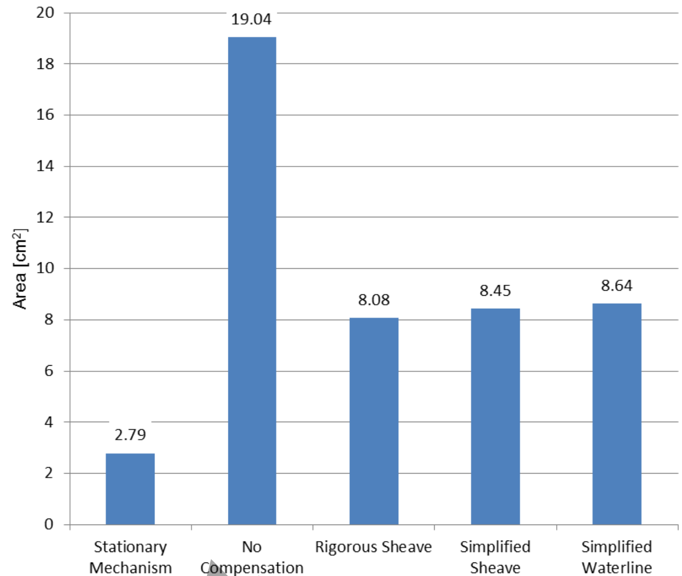


Figure 15: Summary of ellipse areas for the various compensation methods from flume tank experiments

It is also interesting to note that the Rigorous Waterline method exhibited a stability issue that led to erratic behavior. One of the reasons for this instability is that this compensation method may produce large responses to small changes in the tow-line angle. For example, with the winch in its nominal position 46 cm above the waterline, an error of 1º in tow-line angle measurement in its expected range corresponds to an error of approximately 1 cm in tow line length. Furthermore, low line tension allowed for a discontinuity in the line as depicted in Figure **16**, which aggravated the issue by introducing a source of measurement error – especially when the tow line was being reeled in and out quickly. The Rigorous Sheave method was more robust against sensor error, since the compensation method calculates the winch command based on the winch location, not on the difference in tow-line length. For the same error of 1º in tow-line angle measurement near its nominal range, the maximum expected error for the Rigorous Sheave method is approximately 0.13 mm.

As seen in Figure **15**, the performance of the three successful compensation methods was nearly identical. This observation can be explained because of the geometry of the experimental setup. Since the nominal height of the sheave tow point was 46 cm above the flume tank waterline and the vertical motion was limited to only ±4 cm, the calculated winch commands are very similar, regardless of which compensation method is implemented. This geometry, however, is not proportionally representative of a real-life towed sensor system. Ideally, the test apparatus would be mounted closer to the waterline, such that there is greater relative vertical displacement of the sheave tow point. Due to physical limitations of the flume tank and equipment available it was not possible to modify the experiment's geometry; therefore, the computer simulator was modified and validated to match the experimental conditions. Once validated, the geometry within the simulator could easily be adjusted to examine the performance of alternative configurations.

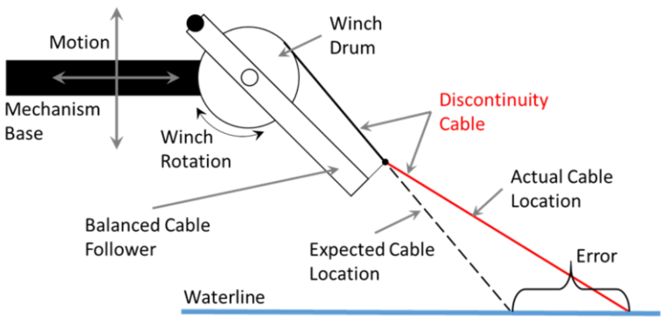


Figure 16: Diagram depicting the source of Rigorous Waterline method erratic behaviour

V. Small-Scale Simulation

The simulator discussed in Section III was adjusted to reflect the conditions which were experienced during the flume tank tests.

The fluid velocity profile and turbulence within the flume tank was measured and replicated in simulation. Periodic buffeting forces were also added to the spherical tow body following the approaches described by Jones and Clarke **Error! Reference source not found.** and Sakamoto and Haniu [9] which outline the appropriate turbulent drag force coefficient and buffeting frequency for a sphere.

The tow-line rotational stiffness and damping values were identified by observing the impulse response of a length of cable and fitting the response frequency and logarithmic decrement rate to a simple second order mass stiffness damper model. The resulting tow-line density, stiffness and damping parameters are summarized in Table 4.

Table 4: Simulation tow-line parameters

Parameter	Value	Units
Tow line density	1221	kg/m^3
Rotational stiffness	6.231×10^{-6}	Nm/deg
Rotational damping	1.323×10^{-8}	Nms/deg

To validate the simulator, the simulation and experimental results were compared for the stationary tow-point mechanism case. Figure **17** compares the simulation and experimental results for this stationary mechanism case with the sheave tow point and flume tank waterline superimposed for scale. The origin for this plot is taken as the centre of the field of view of the experimental test setup's digital camera. One can see from this figure that there is good agreement between the experimental and simulation results.

Figure **18** summarizes the corresponding simulation results showing the ellipse areas for the stationary tow-point mechanism case, the uncompensated case, and each of the compensation methods – except, again, for the Rigorous Waterline method which produced unstable results similar to that observed with the experiments.

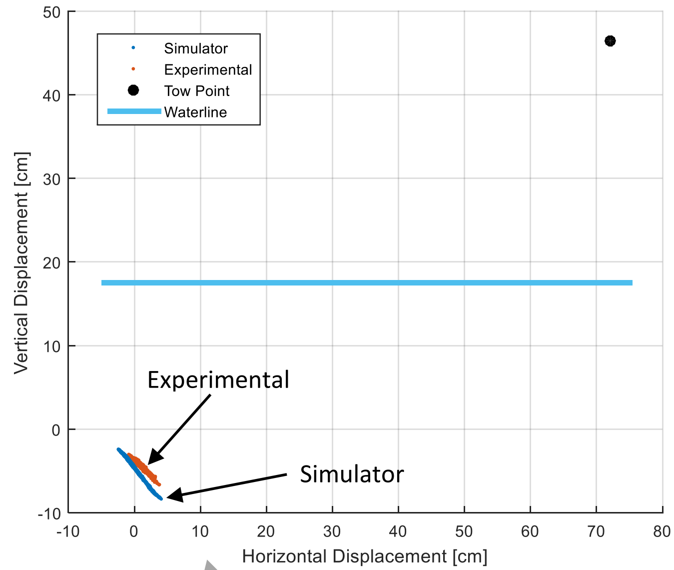


Figure 17: Superposition of experimental and simulator results for a stationary tow-point mechanism test

Referring to Figure **18**, for the uncontrolled case it was found that the 95% of the data point fit within an ellipse area of 33.02 cm^2. Although the actual ellipse areas obtained from the simulations are different than those of the experiments shown in Figure **15**, the trends are similar. For example, the compensation method that reduced the motion the greatest in both simulation and experiment was the Rigorous Sheave method. Furthermore, the Rigorous Waterline method displayed erratic and unstable behaviour in simulation in a manner very similar to that observed in experimentation.

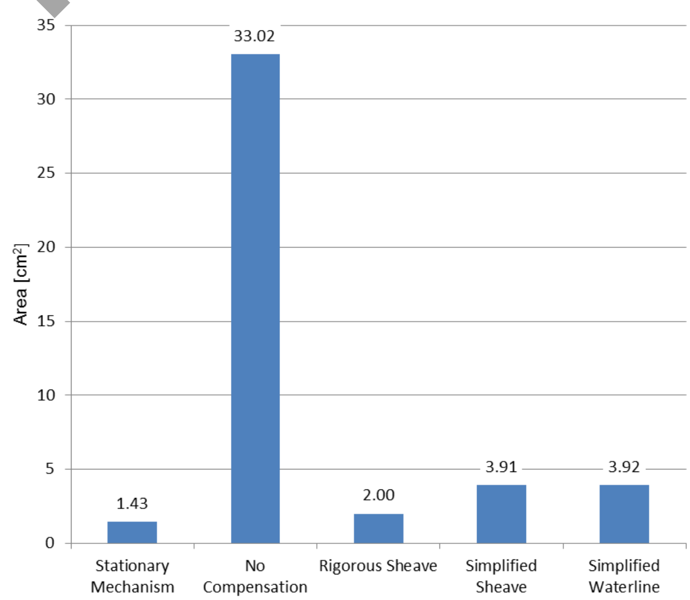


Figure 18: Summary of ellipse areas for the various compensation methods from computer simulations

Having validated the simulator, the nominal sheave tow point was lowered within the simulation from 46 cm above the waterline to 13 cm above the waterline. The resulting ratio of vertical motion experienced by the sheave tow point to vertical offset of the nominal position is more representative of actual ship motion and geometry [7]. Figure **19** summarizes the corresponding ellipse areas from these simulation results for the no compensation case as well as the four compensation methods. It should be noted that, for these simulations, fluid turbulence was neglected causing the Rigorous Waterline method to exhibit improved stability albeit still performing poorly.

For the uncompensated case it was found that the 95% of the data points fit within an ellipse area of 26.99 cm^2. Again it can be seen that the compensation method that reduced the towed body motion the greatest was the Rigorous Sheave method. For this case, the ellipse area was 2.08 cm^2, suggesting an overall towed body motion reduction of 92% from the uncompensated case. The Simplified Sheave method had an ellipse size of 2.85 cm^2 corresponding to an overall towed body motion reduction of 89%, while the Simplified Waterline method had an ellipse size of 3.33 corresponding to an overall towed body motion reduction of 88% for this set of simulations.

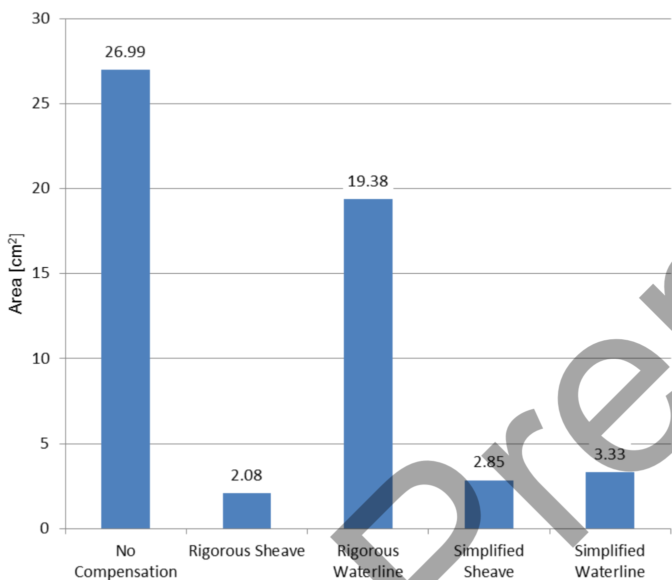


Figure 19: Summary of ellipse areas for the various compensation methods from flume tank experimentation

VI. Conculsions and Future Work

Of the compensation methods which were explored in this paper, the Rigorous Sheave method proved to be most effective in both simulation and experimentation, reducing towed body motion by 92% in the final set of simulation results. The Rigorous Waterline method, while somewhat effective in preliminary simulations, displayed a susceptibility to angle measurement error which led to erratic and unstable behaviour in both simulation and experiments. Future work should be carried out to further investigate the stability of this compensation method.

In the final simulation tests, both Simplified methods performed reasonably well providing an overall towed body motion reduction of between 88% and 89%. These results suggest that, with a modest reduction performance, the Simplified methods might offer a cost-saving opportunity since the extra sensory equipment needed to measure the tow-line angle presents an added expense.

It is suggested that future work be undertaken to determine the effects of different towed body geometry, towed body mass, tow-line drag and flow profiles.

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