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Experimental Performance Study of a Lab-Scale Hydrokite System

Shriya Gosavi

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Experimental Performance Study of a Lab-Scale Hydrokite System

by

Shriya Gosavi

A Thesis Submitted in Partial Fulfillment of the Requirements for the Degree of Master of Science in Mechanical Engineering

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Acknowledgments

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Abstract

Experimental Performance Study of a Lab-Scale Hydrokite System

Shriya Gosavi

Supervising Professor: Dr. Mario W. Gomes

Hydropower plants are the main source of renewable energy from moving water. However, traditional dam systems are somewhat controversial since they are large and often require population relocation, disrupt fish migration, and change the natural flow of the river. Hydrokinetic systems are currently being developed which can harness energy from flowing water with less ecological impact. One hydrokinetic system that may be promising, a hydrokite system, consists of an oscillating arm, boom, and a translating hydrofoil. Due to the hydrodynamic forces caused by the water’s velocity, the hydrokite moves back and forth extracting energy from the flow. Previous simulations have shown that power production for this system depends on at least ten different parameters. Little experimental work has been done on hydrokite systems to validate the results of these simulations. This work focused on the experimental testing of a lab-scale hydrokite system. Tests were run to determine the system power trends with respect to hydrofoil angles, boom flip angles, pivot-point location on hydrofoil, tow speed, and hydrofoil submerged depth.

Single dimension parameter tests were done to determine the changes in average cycle power for the system as a function of a given parameter. Power production was highly sensitive to changes in hydrofoil angles (for hydrofoils pivoting at both the quarter and half-chord point). The optimal hydrofoil angle for the tests that were run was approximately $\beta = 60^\circ - 80^\circ$. As predicted, the power production increases with increased tow speed and submerged depth, neglecting energy used to flip the hydrofoil. Changes in boom flip angle did not significantly affect power production for the quarter-chord tests, but appeared to be significant in the half-chord tests. Although the largest power produced in all of this testing was approximately 0.14 Watts, this initial testing of the lab-scale system has given us some insight into the important design decisions that will need to be made in order to scale-up the system.
# Contents

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acknowledgments</td>
<td>iv</td>
</tr>
<tr>
<td>Abstract</td>
<td>vi</td>
</tr>
<tr>
<td>1 Background Information</td>
<td>1</td>
</tr>
<tr>
<td>1.1 Introduction</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Literature Review</td>
<td>2</td>
</tr>
<tr>
<td>1.2.1 Theoretical Models</td>
<td>3</td>
</tr>
<tr>
<td>1.2.2 Investigating the use of Flapping Foils for Power Generation</td>
<td>6</td>
</tr>
<tr>
<td>1.2.3 Experimental Models</td>
<td>14</td>
</tr>
<tr>
<td>2 Experimental Setup</td>
<td>22</td>
</tr>
<tr>
<td>2.1 Hydrokite System Description</td>
<td>22</td>
</tr>
<tr>
<td>2.1.1 Generator and Encoder Setup</td>
<td>22</td>
</tr>
<tr>
<td>2.2 Results and Discussion</td>
<td>27</td>
</tr>
<tr>
<td>2.3 Conclusions</td>
<td>31</td>
</tr>
<tr>
<td>3 Resistive Load Redesign (Friction Brake)</td>
<td>33</td>
</tr>
<tr>
<td>3.1 Test Setup</td>
<td>33</td>
</tr>
<tr>
<td>3.2 Test Description</td>
<td>38</td>
</tr>
<tr>
<td>3.3 Test Plan</td>
<td>40</td>
</tr>
<tr>
<td>3.4 Average Power Calculations</td>
<td>42</td>
</tr>
<tr>
<td>4 Test Results</td>
<td>50</td>
</tr>
<tr>
<td>4.1 Hydrofoil Angle ($\beta$) Test (Quarter-Chord)</td>
<td>50</td>
</tr>
<tr>
<td>4.1.1 Steady State</td>
<td>51</td>
</tr>
<tr>
<td>4.1.2 Results</td>
<td>51</td>
</tr>
<tr>
<td>4.2 Hydrofoil Angle ($\beta$) Test (Half-Chord)</td>
<td>53</td>
</tr>
<tr>
<td>4.2.1 Steady State</td>
<td>54</td>
</tr>
<tr>
<td>4.2.2 Results</td>
<td>54</td>
</tr>
<tr>
<td>4.3 Boom Angle position ($\theta_{\text{flip}}$) Test (Quarter-Chord)</td>
<td>57</td>
</tr>
<tr>
<td>4.3.1 Steady State</td>
<td>58</td>
</tr>
<tr>
<td>Section</td>
<td>Title</td>
</tr>
<tr>
<td>---------</td>
<td>-------</td>
</tr>
<tr>
<td>4.3.2</td>
<td>Results</td>
</tr>
<tr>
<td>4.4</td>
<td>Boom Angle position ($\theta_{flip}$) Test (Half-Chord)</td>
</tr>
<tr>
<td>4.4.1</td>
<td>Steady State</td>
</tr>
<tr>
<td>4.4.2</td>
<td>Results</td>
</tr>
<tr>
<td>4.5</td>
<td>Tow Speed Test ($V_\infty$) Test</td>
</tr>
<tr>
<td>4.5.1</td>
<td>Steady State</td>
</tr>
<tr>
<td>4.5.2</td>
<td>Results</td>
</tr>
<tr>
<td>4.6</td>
<td>Submerged Depth ($D$) Test</td>
</tr>
<tr>
<td>4.6.1</td>
<td>Steady State</td>
</tr>
<tr>
<td>4.6.2</td>
<td>Results</td>
</tr>
<tr>
<td>4.7</td>
<td>Overall Conclusions</td>
</tr>
<tr>
<td>4.8</td>
<td>Main Contributions</td>
</tr>
<tr>
<td>4.9</td>
<td>Future Work</td>
</tr>
</tbody>
</table>

**Bibliography**

**A Data Analysis Code**

A.1 System Repeatability Code | 76
A.2 Power Calculation Code | 76
A.3 Code for entire range of data with repeatability tests | 82
A.4 Microcontroller Code for Hydrofoil Control (Arduino) | 87
A.5 LabVIEW Data Collection Code | 88

**B Calibration**

B.1 Load Cell | 94
B.1.1 Tension Only: Vertical Mount | 94
B.1.2 Tension and Compression: Horizontal mount | 96
B.2 Tow cart position potentiometer | 98
B.3 Boom angle potentiometer | 100
B.4 Hydrofoil angle potentiometer | 103

**C Error Analysis**

C.1 Tow Cart Position Error | 107
C.2 Load Cell Error | 109
C.3 Boom Angular Position | 110
C.4 Hydrofoil Angular Position | 112
C.5 Propagation of Error for Torque Calculation | 114
C.6 Propagation of error for Average Cycle Power calculation | 117
List of Tables

2.1 Test parameter matrix taken from [1] ........................................... 26
2.2 Combinations used for different runs for the hydrokite system producing positive and negative power .................................................. 31
4.1 Parameters held constant during the hydrofoil angle test ............... 50
4.2 Constant test parameters for quarter chord NACA 0015 for hydrofoil angle testing ................................................................. 52
4.3 Constant test parameters for half chord NACA 0015 for hydrofoil angle testing ................................................................. 54
4.4 Average cycle power produced for half chord NACA 0015 hydrofoil for various hydrofoil angles ................................................. 55
4.5 Test parameters that were constant throughout the runs carried out for boom angles for quarter chord NACA 0015 hydrofoil ....... 57
4.6 Average cycle power produced for quarter chord NACA 0015 hydrofoil for various boom angles ............................................. 59
4.7 Test parameters which were constant throughout the runs carried out for boom angles for half chord NACA 0015 hydrofoil .......... 61
4.8 Average cycle power produced for half chord NACA 0015 hydrofoil for various boom angles ................................................. 62
4.9 Test parameters which were constant throughout the runs carried out for tow speeds. The tow speeds were tested only using the quarter chord NACA 0015 hydrofoil ............................................. 64
4.10 Average cycle power produced for quarter chord NACA 0015 hydrofoil for various water speeds ............................................. 66
4.11 Test parameters which were constant throughout the runs carried out for submerged depth of hydrofoil. The depth was tested only using the half chord NACA 0015 hydrofoil ............................................. 68
4.12 Average cycle power produced for half chord NACA 0015 hydrofoil for submerged depth in water ............................................. 69
B.1 Load cell calibration when only tension loading is taken into consideration (vertical orientation) ......................................................... 95
# List of Figures

1.1 Simple Kite Model. Image taken from [18] .............................. 5  
1.2 Short caption for figure 1 .................................................. 5  
1.3 Short caption for figure 1 .................................................. 5  
1.4 Mathematical Model for Flapping Foil. Image taken from [23] .... 7  
1.5 Heaving and Pitching Motion of the Hydrofoil. Image taken from [11] 8  
1.6 Oscillating airfoil in its power extraction regime (χ > 1). Image taken from [11] ................................................................. 9  
1.7 Experimental Setup of the Oscillating Wing. Image taken from [9] . 11  
1.8 Side View of the Experimental Setup of the Oscillating Wing. Image taken from [9] ................................................................. 12  
1.12 Wingmill system: showing experimental setup and simplified 2D model. Image taken from [20] ........................................... 17  
1.13 Schematic of model used in the wind tunnel. Image taken from [10] .... 19  
1.14 Comparison between numerical and experimental results. Image taken from [10] ................................................................. 20  

2.1 Figure shows the generator and encoder assembly used for the hydrokite system ................................................................. 23  
2.2 The boom and hydrofoil angles for the generator-encoder assembly ................................................................. 25  
2.3 Top plot shows the raw position and torque data collected from the sensors along with the desired hydrofoil angle. The two red marks delineate the clipped data which isolates the stable periodic motion of the system and ignores the data from the transient portions of the motion ................................................................. 27  
2.4 The first plot shows the angular position of the boom along with the calculated time derivative of that signal. The right plot shows a phase plane plot of the periodic boom motion ................................................................. 28
2.5 Left plot shows that the dynamic response of the DC generator is significantly different from the linear steady-state response. The right plot shows a standard simplified steady-state generator response for a DC motor.

2.6 The plot show that the average cycle power for this run was 0.6 Watts.

3.1 Figure shows the top view of the friction brake and potentiometer setup.

3.2 Top view of the friction brake made out of wood to provide variable resistance to the boom shaft.

3.3 Top view of the new friction brake designed for better surface contact between the boom shaft and brake.

3.4 The electrical connections for the microcontroller. The boom potentiometer reads the angular position of the boom and the microcontroller changes the servomotor angular position. This data is collected at the data acquisition system for further analysis.

3.5 Figure shows the boom angle ($\theta$) and the hydrofoil angle ($\beta$).

3.6 Figure shows the deploy and return stroke and the deploy and return flip which make 1 cycle stroke.

3.7 Raw data collected from all four signals on hydrokite system.

3.8 Plot shows the selected range for boom angle and hydrofoil angle after removing the transients from Fig. 3.7.

3.9 Figure shows a low-order curve fit to the five data points in the window (2,3,4,5,6). Note that this curve fit will be associated with the middle data point in the window (4). The slope of this curve fit can be calculated from the curve fit at the middle data point and this will be the value of the derivative curve for that data point (4).

3.10 Plot shows the boom angular velocity and boom angular position with respect to time. The boom angular velocity has been more refined by reducing the noise.

3.11 Instantaneous power for the hydrokite system after reducing the noise. Average cycle power of 0.0931 Watts is recorded for this test.

3.12 Plot shows the tow cart velocity over three cycles.

3.13 Left plot shows the earlier system without the encoder and new brake did not reach steady-state operation. The right plot shows that on adding the encoder and the new brake, the system reached steady-state, thus making the data reliable.
3.14 Plot shows the earlier data was not repeatable. During the tests, boom angle amplitude should have been the same. However, certain peaks are smaller than the others.

3.15 Top left plot shows the change in torque with respect to the boom angular velocity when the resistance is provided by the Aluminum brake. Top right plot shows ideal relation between angular velocity and the torque when the kinetic coefficient of friction is equal to the static coefficient of friction. The bottom plot shows the relation between boom angular velocity and the torque for a wood brake.

4.1 Plot shows the phase plane between the boom angle and the boom angular velocity. This overlapping runs show us that the system is in steady state.

4.2 Average cycle power produced by a half chord NACA 0015 over a range of hydrofoil angles.

4.3 Left plot shows the average tow speed at every angle of hydrofoil where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of hydrofoil angles for quarter chord NACA 0015 hydrofoil.

4.4 Phase plane for boom angular position and boom angular velocity for half chord NACA 0015 hydrofoil.

4.5 Average cycle power produced by a half chord NACA 0015 over a range of hydrofoil angles.

4.6 Left plot shows the average tow speed at every angle of hydrofoil where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of hydrofoil angles for half chord NACA 0015 hydrofoil.

4.7 Plot shows that the data is repeatable for larger boom angles for a quarter chord NACA 0015 hydrofoil.

4.8 Average cycle power produced by a quarter chord NACA 0015 over a range of boom angles.

4.9 Left plot shows the average tow speed at every boom angle where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of boom angles for quarter chord NACA 0015 hydrofoil.
<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.10</td>
<td>Plot shows the phase plane for the boom angular position and boom angular velocity for the half chord NACA 0015 hydrofoil. As the closed loops are nearly right on top of each other, the system is said to be in steady state.</td>
</tr>
<tr>
<td>4.11</td>
<td>Average cycle power produced by a half chord NACA 0015 over a range of boom angles</td>
</tr>
<tr>
<td>4.12</td>
<td>Left plot shows the average tow speed at every boom angle where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of boom angles for half chord NACA 0015 hydrofoil.</td>
</tr>
<tr>
<td>4.13</td>
<td>Phase plane for boom angular position and boom angular velocity to check if the system is in steady state for this set of experiments.</td>
</tr>
<tr>
<td>4.14</td>
<td>Average cycle power produced by a quarter chord NACA 0015 over a range of tow speeds</td>
</tr>
<tr>
<td>4.15</td>
<td>Standard deviation in tow speed for quarter chord NACA 0015 hydrofoil for tow speed experiments</td>
</tr>
<tr>
<td>4.16</td>
<td>Plot shows that the system is at steady state for this experiment</td>
</tr>
<tr>
<td>4.17</td>
<td>Average cycle power produced by a quarter chord NACA 0015 over a range of submerged depth of hydrofoil</td>
</tr>
<tr>
<td>4.18</td>
<td>Left plot shows the average tow speed at every test conducted for submerged depth of hydrofoil. Right plot shows the standard deviation in the tow speed over this range of depth for half chord NACA 0015 hydrofoil.</td>
</tr>
<tr>
<td>4.19</td>
<td>Plot shows the results from McConnaghy's [19] work. The results show cycle power as a function of hydrofoil angles.</td>
</tr>
<tr>
<td>A.1</td>
<td>LabVIEW code used to collect data from all four analog signals</td>
</tr>
<tr>
<td>B.1</td>
<td>Load cell calibration</td>
</tr>
<tr>
<td>B.2</td>
<td>Load cell calibration setup to test load cell in both tension and compression</td>
</tr>
<tr>
<td>B.3</td>
<td>Calibration of load cell in tension and compression</td>
</tr>
<tr>
<td>B.4</td>
<td>Tow cart Calibration</td>
</tr>
<tr>
<td>B.5</td>
<td>Method used to measure the boom angle to calibrate the potentiometer</td>
</tr>
<tr>
<td>B.6</td>
<td>Boom calibration</td>
</tr>
<tr>
<td>B.7</td>
<td>Figure shows the boom potentiometer assembly on the tow cart. A wooden bracket and a set screw has been used to ensure that the potentiometer provides true positioning of the boom</td>
</tr>
</tbody>
</table>
B.8 Hydrofoil calibration

B.9 The hydrofoil potentiometer assembly is shown in this figure. Epoxy has been used between the wood and the pinion to ensure that there was no more slipping of the potentiometer.

C.1 Histogram of the noise level for a constant reading of the tow-cart position.

C.2 Histogram of load cell data for a constant applied load (data collected using the DAQ).

C.3 Method used to measure the boom angle to calibrate the potentiometer.

C.4 Histogram of boom angle analog signal for a constant boom position (data collected using the DAQ).

C.5 Method used to measure the error while determining the hydrofoil angle.

C.6 Histogram of hydrofoil angle analog signal for a constant hydrofoil position (data collected using the DAQ).

C.7 Exaggerated view of the load cell if it was attached at an angle $\phi$ instead of being perpendicular to the arm.
### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>Angle of attack</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Relative angle between hydrofoil and boom</td>
</tr>
<tr>
<td>$\beta_d$</td>
<td>Deploy flip angle</td>
</tr>
<tr>
<td>$\beta_r$</td>
<td>Return flip angle</td>
</tr>
<tr>
<td>$\theta_{flip}$</td>
<td>Boom angle where hydrofoil flips</td>
</tr>
<tr>
<td>$\theta_d$</td>
<td>Deploy angle</td>
</tr>
<tr>
<td>$\theta_r$</td>
<td>Return angle</td>
</tr>
<tr>
<td>$\tau$</td>
<td>Torque from boom arm on resistive load</td>
</tr>
<tr>
<td>$\omega_{Boom}$</td>
<td>Angular velocity of boom</td>
</tr>
<tr>
<td>$BL$</td>
<td>Boom length</td>
</tr>
<tr>
<td>$CL$</td>
<td>Chord length of airfoil</td>
</tr>
<tr>
<td>$D$</td>
<td>Submerged depth of hydrofoil in water</td>
</tr>
<tr>
<td>$HF_S$</td>
<td>Hydrofoil shape</td>
</tr>
<tr>
<td>$\bar{P}$</td>
<td>Average cycle velocity</td>
</tr>
<tr>
<td>$P_{inst}$</td>
<td>Instantaneous power produced by hydrokite</td>
</tr>
<tr>
<td>$V_{\infty}$</td>
<td>Tow cart velocity</td>
</tr>
<tr>
<td>$V_o$</td>
<td>Voltage reading of the analog signal</td>
</tr>
</tbody>
</table>
Chapter 1

Background Information

1.1 Introduction

Most of the energy production systems used today are fossil fuel dependent. Due to the fast depleting nature of fossil fuels and their hazardous impact on the environment, there is a need to look at alternative methods to obtain energy. Renewable energy resources such as wind energy, solar energy and hydro power are being used to reduce our dependency on fossil fuels. Currently, 83% of the total power used in the USA is provided by fossil fuels whereas only 2.48% is provided by hydro power [7].

For traditional large scale hydro power plant systems, expensive infrastructure is required. These power plants require very specific locations, most of which have already been exploited in the USA. Similar to existing centralized power plants, dams generate power far away from where the energy is needed which leads to high transmission losses. In addition, ecological controversies surround dams due to the negative ecological impacts on the surrounding environment such as disruption of fish migration routes and increased sedimentation behind dams [4]. Conventional hydro power cannot economically exploit low pressure head water bodies which are widely distributed.

There are other systems, both existing and in development, which can exploit low-pressure head fluid flows for energy production. One such system, is a translating
air/hydrofoil system. For her thesis work, Kelsey McConnaghy [19] made a model for a translating hydro power system, which she called a hydrokite. It consists of a hydrofoil attached at the end of a boom. This system was allowed to travel between the two sides (banks) of a river. McConnaghy created a simulation to estimate the amount of power extracted. She predicted that significant power could be produced using this system. However, there were several simplifications made when creating the model, probably the most significant simplification was that she assumed the system was operating at steady-state (i.e. that the system could accelerate instantaneously). McConnaghy found that among other parameters, parameters such as the hydrofoil flipping angle, boom rotation angle and velocity of water influence the power production.

The work proposed for this thesis would provide the data to validate McConnaghy’s simulation results. We propose to experimentally study this hydrokite system on a smaller scale. Through our experiments, we will determine the relationship between power extracted and a certain set of parameters such as boom angle, deploy and return angle of the hydrofoil (pitching angles), torque on friction brake and tow cart velocity. These experiments will help determine the range of validity of the assumptions made in McConnaghy’s thesis. The results should also help us improve our understanding of the impact of design parameters on the novel hydro power system and help improve the design and implementation of this system that is able to harness a previously untapped energy source in a hopefully, ecologically benign way.

1.2 Literature Review

Some work has been done on systems similar to the hydrokite system. Two systems which are related to our work are the high altitude air kite systems and the oscillating wing systems. These systems operate in fluids with low pressure heads, where the flow of the fluid is approximately perpendicular to the translational velocity of the
kite/wing. These systems which operate in water do not require dam construction. The theoretical results related to power production from these other research studies will be relevant to our experimental results.

1.2.1 Theoretical Models

Some theoretical models have been published for both, the high altitude air kite systems and the oscillating wing system for their ability to extract energy. Some of this work is being examined due to its similarity with our system.

Theoretical Models: Air Kite Models

The study of high altitude air kites have been around for over a century now. One of the earliest articles that we could find on this topic was provided by Pocock [22]. The high altitude air kite models consist of a kite connected to a massless tether that is fixed to the ground.

Lyod [18] presented a design concept which can produce wind power using kites. Calculations were done for three simple kite models discussed in this paper. The three models were the simple kite, crosswind motion kite and the drag power kite as shown in figures 1.1 and 1.2. In the simple kite model, the kite holds its position as long as the tether does not unwind. However, if the tether unwinds, then power is generated at the ground. The crosswind motion kite model is a special case in which he examines if the tether is parallel to the wind. Power is generated by allowing the tether to lengthen. It is assumed that lift to drag \((L/D_K)\) ratio is very large and so the crosswind velocity and the relative velocity are equivalent. In the drag power model, the motion of the kite is similar to the crosswind kite model but the tether length is constant. Small wind turbines on the kite produce an additional drag which
results in power production. The power produced by the models is given by:

\[ P = P_W A C_L F \]  \hspace{1cm} (1.1)

\[ P_W = \frac{1}{2} \rho V_W^3 \]  \hspace{1cm} (1.2)

where, \( P_W \) is the power density, \( V_W \) is the wind velocity, \( \rho \) is the air density, \( A \) is the wing area, \( C_L \) and is the lift coefficient. \( F \) is a parameter which varies with the model. The calculations are based on 2-D, steady state models [18].

The crosswind kite model is very similar to the system which will be the focus of this thesis. The power produced is given by

\[ P = LV_L \]  \hspace{1cm} (1.3)

where \( L \) is the lift and \( V_L \) is the load velocity by which the tether pulls in the load,

\[ L = \frac{1}{2} \rho AC_L (V_W - V_L)^2 (L/D_K)^2 \]  \hspace{1cm} (1.4)

As the lift to drag ratio \((L/D_K)\) is large, the magnitudes of the crosswind velocity \( (V_c) \) and the relative velocity \( (V_A) \) through the air are equal. Therefore,

\[ V_A = V_C = (V_W - V_L)L/D_K \]  \hspace{1cm} (1.5)

By using these equations, \( F \) becomes

\[ F_C = (L/D_K)^2 (V_L/V_W)(1 - V_L/V_W)^2 \]  \hspace{1cm} (1.6)

At

\[ V_L/V_W = 1/3 \]  \hspace{1cm} (1.7)
the maximum value of $F_C$ occurs and is given by

$$F_{C_{max}} = 4/27(L/D_K)^2 \quad (1.8)$$

Similarly, for the drag kite model the equations are

$$P = D_P V_A \quad (1.9)$$

$$V_A = V_W L / (D_P + D_K) \quad (1.10)$$

$F_D$ for drag kite model is given by:

$$F_D = (L/D_K)^2 (D_P/D_K) (1 + D_P/D_K)^3 \quad (1.11)$$

$$F_{D_{max}} = 4/27(L/D_K)^2 \quad (1.12)$$

The paper shows that more power can be obtained from the crosswind kite model and drag power model compared to the simple kite model. The calculation in the paper estimates that 6.7MW at 10m/s wind speed can be produced for a kite with wing area of 576m$^2$.

Though the kite system is an air based system and our proposed project is a water
based system, the basic concept remains the same. Both systems are used to extract energy from low pressure head fluid and the functioning of both systems depend on lift and drag forces acting on them. Though the efficiencies may be different, both systems have a maximum theoretical limit to their efficiencies, the Betz limit.

1.2.2 Investigating the use of Flapping Foils for Power Generation

Theoretical Models: Oscillating Wing Models

Similar to the high altitude kite concept, oscillating wings can be used to extract power from water. In his preliminary thesis report, Mark D Ripper [23] examined a system which consists of a hydrofoil (oscillating wing) that can be used to extract power from a flowing fluid. The foil is assumed to have two degrees of freedom, a vertical motion (heaving motion) and a rotational motion (pitching motion). He examined the effects of three varying parameters, the heaving amplitude, the pitching amplitude and the Reynolds number on the power extraction efficiency. Three methods of simulation, the quasi method, the wake interaction method and the computational fluid dynamics method were used to investigate the ability to extract power from a single hydrofoil. The mathematical model is shown in figure 1.4 and the lift force, \( L \) and the pitching moment \( M \), for the model are given by:

\[
L = m\ddot{y} + c_1\dot{y} + ky \\
M = Ig\ddot{\theta} + c_2\dot{\theta} + k_2\theta
\]

The instantaneous power output is given by

\[
P = La + M\Omega,
\]
and the power extraction efficiency is given by

\[ \eta = \frac{\bar{P}}{\frac{1}{2} \rho U_\infty^3 d} \]  

(1.16)

where \( \bar{P} \) is the mean power output over one cycle, \( \Omega \) is the angular velocity, \( U_\infty \) is the free stream velocity, \( a \) is the pivot point parameter and \( d \) is the swept area. The range of the heaving amplitude, \( h \) is restricted as it is assumed that the airfoil is at the maximum angle of attack before any vertical motion. The minimum heaving amplitude \( h_{\text{min}} \) is given by

\[ h_{\text{min}} = c \sin \theta_{\text{max}} \]  

(1.17)

where \( c \) is the chord length and \( \theta \) is the angle of attack.

Of the three methods studied, the quasi-steady method is the simplest. The lift and moment forces are a function of the angle of attack and all other aerodynamic effects are ignored.\[23\] This project proved "the concept of a simulating mechanical stop to force a foil rotation at the end of its vertical travel."\[23\]

The wake interaction model deals with the interaction of the airfoil with its wake.
The major disadvantage of this method is that it depends upon knowing the velocity and acceleration for the current time step and the operating frequency. Due to this, the results were poor. The third method used the computational fluid dynamics (Fluent) software. The results of this method were plotted and it showed a sudden spike in the lift force. Adjustments were made to locate the source of error. The inconsistencies have not yet been identified.

Kinsey and Dumas [11] discuss the ability of a wing to extract energy from a fluid flow while undergoing a simultaneous pitching and heaving motion. The pitching and heaving motions and their respective velocities are given by:

\[
\theta(t) = \theta_0 \sin(\gamma t) \rightarrow \Omega(t) = \theta_0 \gamma \cos(\gamma t) \tag{1.18}
\]

\[
h(t) = H_0 \sin(\gamma t + \phi) \rightarrow V_y(t) = H_0 \gamma \cos(\gamma t + \phi) \tag{1.19}
\]

Where \( \phi \) is the phase difference between the pitching and heaving motion, \( \gamma \) is the angular frequency \((2\pi f)\), \( \theta_0 \) and \( H_0 \) is the pitching and heaving amplitudes and \( \omega \) and \( V_y \) are their respective velocities as shown in figure 1.5.

The feathering parameter is defined to "qualify the effect of the imposed motion
Fig. 1.6: Oscillating airfoil in its power extraction regime ($\chi > 1$). Image taken from [11].

on the flow regime” [11]. It is given by:

$$\chi = \frac{\theta_0}{\arctan(H_0 \gamma / U_\infty)}$$

(1.20)

If $\chi < 1$, then it belongs to the propulsion regime and if $\chi > 1$, then it belongs to the power extraction regime (refer Fig. 1.6). $\chi = 1$ is a special case in which the angle of attack, $\alpha(t) = 0$, which takes place at every $t = (nT/4)$. This is known as the feathering limit and the pitching amplitude is given by:

$$\theta_0 = \arctan(H_0 \gamma / U_\infty)$$

(1.21)

Considering the power extraction regime ($\chi > 1$), mean power extracted per cycle is given by

$$C_p \equiv \frac{P}{\frac{1}{2} \rho U_\infty^3 c}$$

(1.22)

And efficiency is given by:

$$\eta \equiv \frac{\bar{P}}{\bar{P}_a} = \frac{\bar{P}_y + \bar{P}_\theta}{\frac{1}{2} \rho U_\infty^3 c} = \bar{C}_p \frac{c}{d}$$

(1.23)

Where $\bar{P}_a$ is the power present in the fluid, $\bar{P}$ is the total power extracted from the flow, $d$ is the vertical travel, $c$ is the chord length, $P$ is the instantaneous total
extracted power and $U_\infty$ is the free stream velocity $[11]$. Initially, parameters such as NACA 0015 airfoil, $H_0/c = 1$, $x_p/c = 1/3$ and $Re = U_\infty c/v = 1100$ were kept fixed with respect to varying frequency and pitching amplitude. A maximum efficiency of 34% was achieved for $\theta_0 \approx 70 - 80^\circ$ and $f^* \approx 0.12 - 0.18$, where $f^* \equiv f c/U_\infty$. The maximum theoretical efficiency is 59% (Betz efficiency). Next, the above mentioned parameters were divided into three groups, motion parameters, geometric parameters and viscosity parameters. It was observed that a change in motion parameters such as the heaving amplitude, frequency and maximum angle of attack have an impact on the performance whereas the geometric parameters and viscosity parameters such as the relative thickness of airfoil and the Reynolds number, respectively, did not have a significant impact.

Jones, Davids and Platzer $[9]$ investigated power extraction efficiency of an oscillating wing generator undergoing pitching and plunging motions. The plunging motion of the wing generates thrust whereas the combined pitch and plunge motion extracts power. The motion of the airfoil is sinusoidal and is given by the following equation:

$$\alpha(\tau) = \Delta \alpha \sin(k\tau) \tag{1.24}$$

$$y(\tau) = h \sin(k\tau + \phi) \tag{1.25}$$

The investigation is carried out to obtain the optimum combination of parameters (pitch and plunge amplitude) to extract power from low speed water. In the theoretical approach, the wing is divided into a finite number of panels, with a global, uniform vorticity strength and a local uniform source strength. According to the Helmholtz theorem, the total vorticity in a flow is constant. At the end of each step, the vorticity is released into the flow. The numerical approach is an iterative method "since the magnitude and direction of the velocity of the wake panel are not initially known" $[9]$. In the experimental hydrofoil approach, the setup had a wing, a swing
arm, pitch arm, bell crank and two vertical rails with grooves as shown in Fig. 1.7.

A wing of 350mm span, 62mm chord length and 0.0217m² wing area was allowed to slide freely in the grooves. The model was forced to have pitch amplitude of ±65° and plunge amplitude of ±125mm. Due to the plunging of the wing, the swing arm rocks. This motion is conveyed to a gear which rotates and drives the pitch arm. Through a linkage, the pitch arm moves the bell crank back and forth and this provides the pitching motion in the wing as shown in Fig. 1.8.

The numerical results were compared with McKinney and DeLauriers wingmill [20]. The results were similar at low angle of attack and flow speed but not for higher angles of attack. For a maximum angle of attack of 15°, the power coefficient was 0.58 and the efficiency was 26% for a plunge amplitude of 0.95 and a reduced frequency of 1.6. Here efficiency is defined as the ratio of average power for one cycle to the absolute power present in the fluid. The experimental wingmill was tested at flow speeds as low as 0.3m/s. Due to natural causes, they were not able to collect
Fig. 1.8: Side View of the Experimental Setup of the Oscillating Wing. Image taken from [9]
significant amount of experimental data and hence, the author believes that further investigation needs to be done to check its accuracy.

Along with the above mentioned system, Platzer and others [21] also studied a second power generator system, the tandem power generator, which consists of two hydrofoils in tandem arrangement. These hydrofoils are arranged such that ”the null spot (where the translation of the hydrofoil changes direction) of one hydrofoil coincides with the power stroke of the other, thereby making the generator self starting” [21]. For this result, the two hydrofoils oscillate with a phase difference of 90°. The mechanical design of the previous test bed was changed to reduce the friction problems associated with translation motion of the hydrofoil. The performance of this system highly depends on the distance between the two pivot points (X) and the phase (ψ) between the leading and trailing foil. The results show that for ψ =45°, the net power output is more than that produced for ψ=90°. However, the latter is preferred as the net power output is uniform throughout the cycle and the operation is smooth.

**Conclusion for Numerical Models**

As seen from the above mentioned papers, there is a possibility of extracting power from a flowing fluid with the help of an airfoil/hydrofoil. This fluid may be air or
water. The papers mention a system which consists of a hydrofoil with plunging and pitching motion and various parameters of this system are varied to find the maximum mechanical efficiency. The Kinsey and Dumas paper \[11\] also talks about the motion parameters being the most significant parameters, and these parameters will be studied in detail. Efficiencies as high as 34% have been reported in the theoretical models.

The oscillating wing models are similar to the hydrokite system. The basic principle remains the same, power is extracted from the fluid (water) by using the water forces acting on the wing/hydrokite. The difference in the proposed system is that the hydrofoil would have a horizontal movement as opposed to the vertical plunge mentioned in the above papers. Another difference is that the hydrofoil will not be flipping during its travel but will be flipped at both end points near the rails of the towtank. Validation of the maximum efficiency will be done by carrying out runs with varying parameters such as the hydrofoil flip angles and the boom angles.

### 1.2.3 Experimental Models

Not only have several theoretical models been created and studied, several smaller scale experimental systems have been built and tested. These experiments were designed to test the validity of the simplified theoretical models and to determine if the simplified models could be used to make accurate predictions on larger scale systems.

**Experimental Models: Kite Models**

In their paper, Lansdorp, et al. \[13\], \[14\], \[15\] and \[16\] built a system to test the concept of high altitude energy extraction using the Laddermill concept, Fig. 1.10. The Laddermill concept employs a tether "which stretches into the higher regions of the atmosphere" \[14\]. The top end of the tether is attached to a kite or a wing and the other end is wound around a drum. The generator is powered when the tension in
the tether pulls the tether off the drum. To obtain a good efficiency of the Laddermill concept, the ratio of the ascending cable tension" and the "descending cable tension" should be high [14]. To achieve this, a high lift is achieved in the ascending kite by increasing the angle of attack or by using crosswind power to increase the apparent wind as mentioned in the Lyod paper [18]. During the descent, the angle of attack should be reduced or the use of crosswind power should be avoided so as to reduce the lift of the kite, thereby decreasing the tension in the tether [13].

Currently, surfkites are used to test the Laddermill concept. They are cheap, easily available and give a satisfactory response. The test rig consisted of a kite linked by a single line to the ground. Increasing number of lines would increase the drag and
hence this is avoided. The kite is also provided control mechanisms such as the drag flaps to steer the kite. Two servos are located under the kite and move the wingtip inwards so as to change the angle of attack. The last control mechanism is used to change the ”tether attachment point position” \[13\]. Sensors were also attached to measure the line tension and to locate the position of the kite. The authors insist on carrying out more experimental work in order to understand the properties of better kite designing. Some companies like \[2\] and \[3\] are looking into power generation from high altitude kites.

**Experimental Models: Oscillating Wing Models**

Like the experimental work of high altitude wind kites \[13\], some investigation was also carried out to extract power through water. McKinney and DeLaurier \[20\] describe a harmonically oscillating wingmill which can be used to extract energy from the wind. The wingmill operates with a vertical plunge motion and a pitching motion about its axis as shown in figure 1.12 The harmonic motion is given by:

\[ h = h_0 \sin \omega t \]  \hspace{1cm} (1.26)

\[ \alpha = \alpha_0 \sin(\omega t + \phi) \]  \hspace{1cm} (1.27)

Where, \( h \) is the vertical motion, \( \alpha \) is the pitch angle, \( \omega \) is the frequency of oscillation and \( \phi \) is the phase difference between the pitching and plunging motion.

The experimental setup consisted of a wind tunnel and a wingmill with a scotch yoke mechanism which provided the pitch and plunge motion to the wingmill for a given frequency of oscillation and a phase angle. Experiments were conducted for a fixed plunge amplitude (\( h_0 \)) of 6 cm but pitch amplitude (\( \alpha_0 \)) was varied to 25° or 30°. Also, the experiments were conducted for two wind speeds(\( U_0 \)), 6.2m/s and
Fig. 1.12: Wingmill system: showing experimental setup and simplified 2D model. Image taken from [20]

8.0m/s. The net power extracted by the wingmill was given by:

\[ P \approx N \dot{h} + M \dot{\alpha} \quad (1.28) \]

where, \( N \) is the normal force and \( M \) is the moment.

The power extraction efficiency is given by:

\[ \eta = \frac{\bar{P}}{\bar{P}_{\text{ideal}}} \quad (1.29) \]

where,

\[ \bar{P} = \frac{2\pi}{\omega} \int_{\text{cycle}} P \, dt \quad (1.30) \]

and

\[ \bar{P}_{\text{ideal}} = \frac{16}{27} \left( \frac{1}{2} \rho \bar{A} U_0^3 \right) \quad (1.31) \]

\( \rho \) is the air density, \( \bar{A} \) is the rectangular swept area and \( U_0 \) is the velocity of air.

\( \bar{P}_{\text{ideal}} \) is the Betz efficiency, which is the maximum theoretical efficiency that can be obtained (59.3%). The maximum efficiency obtained from the experiment was
28.3% for $\alpha_0=30^\circ$ and $U_0=8$ m/s.

The experiments carried out by McKinney and Platzer [20], [21] are in good agreement with the Kinsey and Dumas numerical model [11].

From the information provided by McKinney and Delaurier, Jones, Lindsey and Platzer [10] created a computational and experimental model. For their computational model, they used equations provided by McKinney and DeLaurier. It consists of an airfoil having two degrees of freedom, the pitching motion and the plunging motion given by:

\[
\alpha(\tau) = \Delta \alpha \sin(k\tau + \phi) \quad (1.32)
\]

\[
z(\tau) = h\sin(k\tau) \quad (1.33)
\]

\[
k = \frac{2\pi f c}{U_\infty} \quad (1.34)
\]

where $\Delta \alpha$ is the pitching amplitude, $h$ is the plunge amplitude, $\tau$ is the non-dimensional time, $\phi$ is the phase angle between the pitch and plunge and $k$ is the reduced frequency [10]. The instantaneous non-dimensional power coefficient and average power output are given by:

\[
P = q_\infty U_\infty \bar{C}_p S \quad (1.36)
\]

where, '$q_\infty$ is the free stream dynamic pressure and $S$ is the wing area' [10].

Figure 1.13 shows the schematic of the test setup used to conduct this investigation. The main difference this setup and McKinney and DeLaurier’s wingmill setup.
was that Jones’ model employed two wings which were 90° out of phase. The test was carried out in a water tunnel with velocity no more than 16 inches per second. A load cell connected to a Prony brake was used to measure the torque and an encoder was used to measure the rotational speed. From figure 1.14 it can be seen that there is a large difference between the theoretical and experimental results. This may be because the numerical model ignores ‘mechanical friction, the acceleration of mechanical mass and the added mass for the submerged components, buoyancy and three dimensional losses at the wing tips and the gap between wing sections’ [10].

Lindsey’s thesis [17] tried to explain the reason for these errors between the theoretical model and the experimental model. The main reasons for error provided by this thesis was that friction of the system was neglected and wake effects were not considered.

There are companies that are looking at low pressure head (hydro-kinetic) systems. Minesto [8], a Swedish company and BioPower Systems [5], [12], an Australian company are looking at harvesting energy from water.
Conclusion for Experimental Models

The experiments carried out by McKinney and Platzer [20], [21] are in good agreement with the Kinsey and Dumas numerical model [11]. Instead of the vertical rails and bell crank that provides the plunging and pitching motion as mentioned in McKinney’s work, my project will use a horizontal swinging arm (boom), one end of which is connected to the tow tank platform and the other end has a hydrofoil attached to it. In addition, my supporting arm will not be subjected to fluid forces, and the motions of my systems are not prescribed to be sinusoidal, unlike the the system studied by McKinney, Kinsey, and Dumas. Similar to McKinney’s wingmill, a friction brake would be used as a dump load for the energy produced by the system and I will be measuring the mechanical power produced by the system.

Gaps in the literature

Although there have been some experimental studies of oscillating hydrofoils, none of the experimental systems are configured similarly to the one that is examined.
in this thesis. Most of the previous experimental studies specified the pitch and plunge motion. Also, these systems are vertically oriented system with relatively high velocity. This research tries to look at horizontally oriented system working in low water speeds and does not specify the motions, but, looks at the trends of each independent parameter.

Research Goals

- To determine, experimentally, how power production varies with changes in hydrofoil angles, boom flip angle, submerged depth, and tow-cart speed
- Develop an understanding of the repeatability limits for the testbed
- Determine effect of a different class of resistive load on system behavior (electric generator vs. mechanical friction brake)
Chapter 2

Experimental Setup

Parts of this chapter were published in [6]:


2.1 Hydrokite System Description

2.1.1 Generator and Encoder Setup

Test Setup

Figure 2.1 shows the experimental setup for the hydrokite system. While one senior design team built the hydrokite system, a sister team built the tow tank. It consists of a 16 ft long tank with a width of 2 ft and depth of 2.5 ft. The tow tank has an attached tow cart which can be pulled at a given constant velocity by a motor controlled by a Dart Controls DC speed controller (MD 30E). A HP DC motor with a 12.5:1 gear reduction is used to pull this cart. Though this tank is 16 ft long, only 10.5 ft is used for testing, leaving some distance on both ends of the tank to ensure that the tow cart and the hydrofoil do not bang into the ends. Limit switches are provided on both ends to ensure that the tow cart comes to a halt.

For the tests done on this system, an Eulerian frame of reference is used. In this
Fig. 2.1: Figure shows the generator and encoder assembly used for the hydrokite system

method, the focus is on specific locations in space as the fluid flows over time. This means that instead of having a stationary system with fluid moving past it, we have stationary fluid with system moving past it. The hydrokite consists of a boom and a hydrofoil. The boom is made out of Aluminum for a high strength to weight ratio and due to ease of machining. The hydrofoil is made of extruded polystyrene, reinforced with by a wooden ribs and then covered with fiberglass. The base of this hydrokite is located at the center of the tow cart and a generator is mounted to the base. Through a shaft, the generator is coupled to one end of a rigid beam, which will be referred as the boom. Another shaft is used for coupling the other end of the boom to the hydrofoil. Due to the motion of the hydrokite, lift and drag forces and their respective moments are created. Therefore, roller bearings are used to avoid their involvement in the system without affecting the boom performance.

A dump load was selected and attached at the base. It consisted of a 12V DC Motor with a 100:1 gear ratio and an intrinsic magnetic encoder (Pololu 100:1 Metal Gear motor 37Dx57L mm with 64 CPR Encoder). Apart from the boom position
data measured by the encoder, the resistive torque on the dump load also needed to be measured. A loadcell (Phidgets 0-5kg) and a signal conditioner (Logos Electromechanical, Bridge Amplifier v1) was selected to measure the resistive torque.

The hydrofoil is also connected to an RC digital servo (HobbyKing HK47011MG) with the help of a gear assembly. The hydrofoil angle is controlled by this servo to set it at a desired angle. This servo is a digital servo with metal gears and uses a magnetic induction encoder to control the position of the hydrofoil. The servo is specified at a rotational velocity is 0.07seconds/60° and a torque of 1.16Nm. This servo is accurate when set at a given value, as it is controlled by a Pulse Width Modulation (PWM) at a high frequency of 300 Hz. However, it has a limited angular range of 180°, due to which, a gear reduction of 3:1 is used so that the hydrofoil can rotate more than 180°. This hydrofoil angle measured is a function of the boom position. In order to obtain accurate hydrofoil angle, the boom position needed to be accurate. This boom angle is measured with the help of a quadrature encoder. The encoder is at a 100:1 gear ratio with the motor shaft and counts 64 times per revolution.

The encoder collects the position data of the boom, its angular velocity can be calculated and the resistive torque is measured by the load cell. Thus, the instantaneous mechanical power produced by the system can then be calculated as the dot product of generator torque and boom arm angular velocity.

\[ \vec{\tau} = \vec{r}_{p/o} \times \vec{F} \]  

(2.1)

\[ P = \vec{\tau} \cdot \vec{\omega}_{\text{boom}} \]  

(2.2)

The energy produced per cycle can then be calculated by integrating the instantaneous power over that cycle. The average cycle power can then be determined via Eqn. 2.3.
\[ \bar{P} = \frac{\text{Energy per cycle}}{\text{cycle time}} = \frac{\int P \, dt}{T} \]  

(2.3)

LabVIEW was used for control and data acquisition of the system. Excel spreadsheets containing all required data could be easily achieved. To ensure that the data collected by LabVIEW was uncontaminated by any external factors, careful considerations were given to the wiring of the system. As this system deals with a wet environment, water resistant fixtures and connectors were used. Shielded cables were used to avoid electromagnetic interference. This shield was grounded at the NI data acquisition system (NI PCI-6229) where all the signals of the sensors are measured.

Test Parameters

![Diagram of boom and hydrofoil angles for the generator-encoder assembly](image)

Fig. 2.2: The boom and hydrofoil angles for the generator-encoder assembly

Several tests were carried out during the Generator-Encoder setup. These tests were carried out by varying the following parameters.

- Motor Speed \( (V_\infty) \)
- Boom Length \( (BL) \)
- Deploy Angle \( (\theta_d) \)
- Return Angle \( (\theta_r) \)
• Deploy Flip Angle ($\beta_d$)
• Return Flip Angle ($\beta_r$)
• Hydrofoil Depth in Water (NACA 0015 [Symmetric] and NACA 4412 [Asymmetric])
• Chord Length (CL)
• Hydrofoil Shape ($HF_S$)

Table 2.1 shows the values of the parameters that were varied.

Table 2.1: Test parameter matrix taken from [1]

<table>
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<tr>
<th>Run</th>
<th>MotorSpeed [m/s]</th>
<th>BoomLength [in]</th>
<th>$\theta_d$</th>
<th>$\theta_r$</th>
<th>$\beta_d$</th>
<th>$\beta_r$</th>
<th>$D$</th>
<th>CL</th>
<th>$HF_S$</th>
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<td>15</td>
<td>50</td>
<td>75</td>
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<td>50</td>
<td>15</td>
<td>145</td>
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<td>13.5</td>
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2.2 Results and Discussion

Figs 2.3 − 2.6 give the results, in detail, for a single representative test, run #33. Fig 2.3 shows the two sensor signals for the entire test along with the desired hydrofoil angles. Fig 2.3 also shows two marks which delineate the data which is later analyzed further in Figs. 2.4 and 2.6. Note that the dataset is clipped to remove the data transients from later analysis.

Fig. 2.3: Top plot shows the raw position and torque data collected from the sensors along with the desired hydrofoil angle. The two red marks delineate the clipped data which isolates the stable periodic motion of the system and ignores the data from the transient portions of the motion.

Fig 2.4 shows the results of the numerical differentiation of the angular position
data to obtain the boom angular velocity curve. Note that you can clearly see the slight fluctuations in the boom’s angular velocity both on the deploy and return stroke. Fig 2.4 also shows a phase plane plot of the system’s motion. Once the transients have been removed from the dataset, it is clear that the motion of the system is periodic and stable.

![Image of angular position and calculated time derivative](image1)

![Image of phase plane plot](image2)

**Fig. 2.4:** The first plot shows the angular position of the boom along with the calculated time derivative of that signal. The right plot shows a phase plane plot of the periodic boom motion.

The relationship between the torque applied to the DC generator and the angular speed of the boom (as sensed by the encoder on the back of the DC generator) is shown in fig 2.5. Note that the relationship is significantly different from the idealized
steady-state DC generator response which is also shown in fig 2.5. Note that the two small loops shown in fig 2.5 occur when the boom is moving across the width of the tank on the deploy and return strokes. These oscillations in boom velocity are due to, we conjecture, a combination of boom vibration and gearbox backlash. Note that the boom position is measured using an encoder which is mounted on the DC generator. In between the DC generator and the boom arm is a gearbox which is used primarily to increase the amount of torque applied to the boom arm. Thus any backlash between the input and output shafts of the gearbox will lead to errors in determining the true boom angular position. In addition, since the system’s rotational motion changes direction each half cycle of operation, gearbox backlash can also affect the dynamics of the system itself.

![Graph showing torque vs. angular velocity](image)

**Fig. 2.5:** Left plot shows that the dynamic response of the DC generator is significantly different from the linear steady-state response. The right plot shows a standard simplified steady-state generator response for a DC motor.

The instantaneous power produced by the system is calculated using Eqn. 2.2 and is shown in fig 2.6. One can clearly see that the instantaneous power fluctuates on both the deploy and return strokes. It is also clear that there are moments of negative power, where the system is doing work on the water instead of the water...
doing work on the system. Note that these negative power spikes occur during flipping of the hydrofoil. It is likely that the power used to flip the hydrofoil, which we have neglected, is contributing to these negative power spikes as measured at the generator.

Fig. 2.6: The plot show that the average cycle power for this run was 0.6 Watts.

Numerically integrating the instantaneous power for Test run #33 results in an average cycle power of approximately 0.6 W. However, we ran several other tests with different operating parameter values, some of which had a negative average cycle power. Table 2.2 shows the results of some of those tests. Note that all the tests shown in Table 2.2 use the following parameters.

It is clear that for all the runs which produced negative average cycle power, that power must be coming from the power used to flip the hydrofoil at each end of the stroke. Thus, the power used for flipping the hydrofoil is not negligible. In addition, we note that the asymmetric hydrofoil appears to correlate exactly with negative power production. This somewhat odd result is, we believe, is due not to the shape of the hydrofoil, but instead due to placement of the pivot point on the hydrofoil.
Table 2.2: Combinations used for different runs for the hydrokite system producing positive and negative power

<table>
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<th>$\beta_d$</th>
<th>$\beta_r$</th>
<th>$\theta_d$</th>
<th>$\theta_r$</th>
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<th>Average Cycle Power (W)</th>
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itself. Our test hydrofoils were manufactured, due to strength and size requirements, with different pivot point locations. The symmetric NACA0015 hydrofoil pivot at the quarter-chord point, but the asymmetric hydrofoil pivots at the half-chord point. This change in pivot point significantly alters the flipping response of the two hydrofoils. Flipping a hydrofoil at the quarter-chord point can provide a force which has a large component that is perpendicular to the boom arm. This force can help accelerate the boom in it’s motion back across the tank. On the other hand, this force component is significantly small for a hydrofoil which flips at the half-chord point. It appears that this change in pivot location is the dominant reason for the difference in power production between the symmetric and asymmetric hydrofoils, and not the change in shape between them.

2.3 Conclusions

The results show that an average cycle power of approximately 1.9 W can be produced by our small-scale hydrokite system in a flow velocity of 0.5m/s. However, this power production neglects the amount of energy required to flip the hydrofoil at the ends of the motion. Although the hydrofoil flipping power was not measured, we can infer from the negative average power production tests that this power is
not negligible. We have also shown that this system is capable of stable periodic motions with a very simple control system. The power generated is sensitive to changes in operating parameters, with hydrofoil and boom flip angles being the largest influences on performance. Further testing is needed, with parameter optimization and additional sensing, before the system’s maximum power production potential can be determined.
Chapter 3

Resistive Load Redesign (Friction Brake)

The presence of unexpected generator dynamics initiated a redesign of the testbed’s resistive load. In the original system, the resistive load for the boom arm was provided by an electrical generator and a electrical resistance load. Although the exact mechanism which explains the complex generator dynamics observed in the experiments (see Fig. 2.5) is not known, the transient electrical dynamics of the generator, gear reduction backlash, or inertia of the gear train, were not modeled and might be the cause for the complex dynamics. To simplify the system, a simple friction brake was designed to serve as the resistive load.

3.1 Test Setup

For the initial resistive load (using a generator and encoder), a gearbox was mounted between the generator and encoder, and variable electrical resistances could be used to vary the torque applied on the boom arm. The backlash between the input and output shafts on the gearbox led to errors when determining the boom angular position. Also, the system moves back and forth, changing direction every half cycle which leads to the gear backlash, affecting the dynamics of the system itself. There were also problems with gearbox lifetimes. The applied loads caused premature failure of the gearboxes. In addition it’s likely that the electrical transients of the generator were large which could also affect the data collected. For these reasons, the resistive
load was replaced with an alternate system. It was decided that a friction brake would be the simplest way to solve these problems.

Since it was desired to have the torque applied to the boom shaft be adjusted by the user, a friction brake was designed with an adjustable torque setting. Figure 3.2 shows a sketch of the designed friction brake. Note that with this setup it is not possible to specify a specific slipping torque, but instead the torque can be adjusted and then measured precisely. Since friction coefficients can vary from day to day based on many variables, \textit{i.e.} temperature, humidity levels, etc., we chose to simply measure the torque that is applied and allow for gross manual adjustments from test to test.

The original brake was constructed out of wood with a compliant cantilever beam used to adjust the normal force between the brake surfaces and the pivot shaft. A screw was used to manually adjust the normal force. Initially, this brake worked well, but over time the shaft-brake connection was not smooth and had uneven surfaces providing uneven resistance when the hydrofoil changed direction after a half cycle. The lack of repeatability of the experimental results led to the design of a brake with tighter dimensional tolerances. A new brake was made out of aluminum, as shown
Fig. 3.2: Top view of the friction brake made out of wood to provide variable resistance to the boom shaft

in Fig. 3.3. The metal brake had two bolts right next to the hole provided for the shaft. Each bolt was provided with two compression springs connected in parallel to provide a manually variable normal force on the boom pivot shaft.

A small shear-type load cell (Phidgets 760 grams load cell) was used to measure the force applied to the brake to keep it from rotating. A signal conditioner (Logos Electromechanical 1X3 Bridge Instrumentation Amplifier) was used to amplify the signal from the load cell so that it could be measured by the data acquisition system.

To replace the magnetic encoder used on the original generator load, the angular position of the boom was measured using a potentiometer. A 3 turn, 1KΩ, wire wound potentiometer (BOURNS 3547S-1AA-102A) was used. A rigid coupler was used to connect the potentiometer to the boom. The initial design consisted only of the potentiometer, however, difficulties in achieving steady state caused further changes. Since the boom position is used by the microcontroller to control the position of the hydrofoil, sensor noise caused problems with the control algorithm and the
system appeared to not fall into a steady periodic motion. To reduce sensor noise, a digital measure of boom position was used, an incremental optical encoder (US Digital E6). The hydrofoil controller reads the current boom angle and then controls the hydrofoil servo to flip the hydrofoil (change the hydrofoil angle) when the boom angle reaches the specified flip angles. The microcontroller uses the digital signal to control the position of the hydrofoil, but a potentiometer was used to measure the hydrofoil angle for data collection purposes. Data collection is done using a LabVIEW virtual instrument which samples all of the sensors at 1000 Hz. Figure 3.4 shows the connections for the microcontroller and the data acquisition system [NI-DAQ (PCI-6229)].

Apart from the boom position, the hydrofoil position and the tow cart position were recorded. A 3 turn, 1KΩ wire wound potentiometer was used to measure the hydrofoil angle and a 10 turn, 10 KΩ potentiometer was used to measure the position
Fig. 3.4: The electrical connections for the microcontroller. The boom potentiometer reads the angular position of the boom and the microcontroller changes the servomotor angular position. This data is collected at the data acquisition system for further analysis.
of the tow cart. In summary, data from four sensors (3 potentiometers and a load cell) are acquired by the DAQ and recorded.

### 3.2 Test Description

Fig. 3.5 shows a schematic of the hydrokite system. The boom angle, \( \theta \), is the angle between the free stream velocity (or towcart velocity) and the center line of the boom. The hydrofoil angle, \( \beta \), is defined to be the relative angle between the center line of the boom and the chord line of the hydrofoil.

All tests are started with the tow cart pulled to the far end of the tank. The hydrokite is set at a given deploy angle and a desired boom angle. An approximately 7-10 minute delay occurs between tests to allow the water in the tank to stop moving. Once the water is stationary, as determined by visual inspection, the tow cart is pulled towards the other end of the tank at a constant speed. The hydrodynamic forces acting on the hydrofoil cause the boom to rotate towards the other side of the tank. The motion is called the deploy stroke. At a specified boom angle, \( \theta = -\theta_{\text{flip}} \), the hydrofoil angle is rapidly changed, flipped. Due to the change in the angle, the hydrofoil now moves back towards its initial position and completes the return stroke.
Fig. 3.6: Figure shows the deploy and return stroke and the deploy and return flip which make 1 cycle stroke
When $\theta = \theta_{flip}$, the microcontroller flips the hydrofoil back to its original relative angle. The cycle then repeats over and over. Hence, one cycle of the hydrokite system consists of deploy stroke, deploy flip, return stroke and return flip of the hydrofoil. This cycle is shown in Fig. 3.6. Each run consists of a number of cycles before the tow cart travels from one end to the other end of the tank. Once the run is complete, the tow cart is pulled back to the starting position. The next run is done only after the water is stationary again.

Hydrofoils with symmetic profiles (NACA 0015) were chosen because they were had been previously fabricated specifically for these tests. The two hydrofoils used (pivots at quarter and half chord) had slightly different chord lengths due to the need for a specific hydrofoil thickness when they were built.

### 3.3 Test Plan

A number of tests were carried out for the hydrokite system. These tests examined how changes in a single parameter affected the average cycle power of the system when it reached steady-state operation. The parameters that were varied were the Hydrofoil angle, pivot point for the hydrofoil (e.g. quarter-chord and half-chord), boom angle position for flip ($\theta_{flip}$), tow-speed ($V_\infty$), and hydrofoil submerged depth ($d$).

**1a) Hydrofoil Angle ($\beta$) Test (quarter-chord)**

This experiment will study how changes in the hydrofoil angle affect power production ($0^\circ \leq \beta \leq 84^\circ$).

$\beta = 0^\circ, 10^\circ, 20^\circ, 30^\circ, 40^\circ, 50^\circ, 60^\circ, 70^\circ, 80^\circ, 81^\circ, 82^\circ, 83^\circ, 84^\circ$. Note that a finer grid for hydrofoil angle was chosen nearby an area of rapid change in power production ($\beta \approx 80^\circ$).

**1b) Hydrofoil Angle ($\beta$) Test (half-chord)** One of the main concerns in this thesis is the loss of momentum of the system when the boom velocity changes sign. It was
noted during the previous tests using the quarter-chord pivot point that the flipping action of the hydrofoil tended to reaccelerate the boom in the other direction. This test, which uses a half-chord pivot point was chosen to reduce this effect. Hydrofoil angles from \((0^\circ \leq \beta \leq 70^\circ)\) were tested. \(\beta = 0^\circ, 10^\circ, 20^\circ, 30^\circ, 40^\circ, 50^\circ, 60^\circ, 70^\circ\). These tests stop at 70° because beyond this point, the forces are not large enough to move the hydrokite system to the other side of the tank. Note that both hydrofoil angles tests use similarly shaped hydrofoils (NACA 0015) but that the hydrofoils in the two tests have different chord lengths \((c = 2.3in, c = 2.8in)\).

2a) **Boom Angle position \((\theta_{flip})\) Test (quarter-chord)**

Experiments were carried out to determine the change in power production as the boom angle changes. Seven tests were carried out with varying boom angles.

\[ \theta = 10^\circ, 15^\circ, 17^\circ, 20^\circ, 23^\circ, 25^\circ, 28^\circ \]

The testing of higher boom angles might have yielded certain results but due to the restriction of the width of the tank, 28° was the largest angle that could be tested before the hydrofoil touched the sides.

2b) **Boom Angle position \((\theta_{flip})\) Test (half-chord)**

The influence of boom angles on the power production of a half chord NACA 0015 hydrofoil was tested. Five tests were carried out for this parameter.

\[ \theta = 10^\circ, 15^\circ, 20^\circ, 23^\circ, 25^\circ \]

At a higher boom angle, a decrease in power is noticed. This is because the hydrodynamic forces are not large enough at higher boom angles for a constant speed.

3) **Tow Speed \((V_\infty)\) Test**

The tow speed is an independent parameter that influences the power production of the hydrokite system. To test this, ten tests were carried out.

\[ V_\infty = 0.12, 0.14, 0.16, 0.19, 0.24, 0.27, 0.33, 0.39, 0.44, 0.50, 0.54[m/s] \]

As the length of tow tank is \(\sim 5m\), tow speed was restricted to below 0.54 m/s, because the system moves fast and the total time is less than 7 seconds. These tests
were carried out using the quarter chord NACA 0015 hydrofoil.

4) Submerged Depth \((d)\) Test

The last parameter that was tested was the submerged depth of the hydrofoil in water. The tests carried out were at: \(Depth = 7, 8, 9, 10, 11, 12, 13\) [in]

### 3.4 Average Power Calculations

For the calculation of average cycle power, two data-series are needed, the torque on the friction brake and the angular velocity of the boom arm. In our case, we use the load cell to measure the force acting on the brake along with a fixed distance measurement of the perpendicular distance from the load cell line of action and the center of the friction brake to calculate the torque on the friction brake. Since the angular position of the boom is measured, we can filter and numerically differentiate that signal to calculate the angular velocity of the boom. In addition, the hydrofoil position and the tow cart speed were measured to determine their variability.

The analysis of a single test is shown in Fig. 3.7 as an example of how average
cycle power is determined from the measured signals. The four signals, in their raw state are imported in MATLAB. All four signals have been calibrated and the details of these calibrations are shown in Appendix B. Fig. 3.7 shows the tow position, boom angular position, hydrofoil angular position and the torque data after calibration.

An initial angular velocity for the boom arm was provided at the beginning of the tests to help the system fall more quickly into a periodic motion. Two points are manually chosen to limit the data set so that the system will be in steady state and all transients have been removed. These two points are manually selected based on visual appearance of graph. The two chosen points are denoted by the “x”s in Fig 3.7. Fig. 3.8 shows the selected data.

To calculate the power produced for one cycle, angular velocity of the boom is calculated by numerical differentiation. This is because the signal is noisy and this leads the points to be above and below an estimated fit. Therefore, some points have positive slope while the others have negative slope. In order to reduce the effect of
these noisy data points, we use the polynomial curve fit method. For eg., consider an example shown in Fig. 3.9 The given data consists of seven points. A window of 5 points is selected and a polynomial curve fit is used for these points. After this, the first point is dropped and the next 5 points are selected for curve fitting. This method helps reduce the noisy signal.

To calculate the power produced for one cycle, the angular velocity of the boom is calculated by numerical differentiation. Numerical differentiation of experimental data is a poor approach if it is done without additional filtering. This is because the experimental signals have noise and differentiation will amplify that noise. In order to reduce the effect of noise, we use a filtering approach which breaks the data up into a series of overlapping windows each with the same odd number of data points. For each window, a lower order polynomial is fit to the data within the window. This analytic polynomial is associated with the middle data point of the window. This polynomial can then be analytically differentiated to determine the derivative at the
Fig. 3.10: Plot shows the boom angular velocity and boom angular position with respect to time. The boom angular velocity has been more refined by reducing the noise central data point.

To numerically differentiate the boom angle position data, a second order polynomial was chosen with a window size of 107 points. Fig. 3.10 shows the result of applying this differentiation to the boom angular position data.

The instantaneous power for the hydrokite can now be calculated. Instantaneous power is given by:

\[ P_{\text{instantaneous}} = \tau \omega_{\text{Boom}} \]  \hspace{1cm} (3.1)

where \( \tau \) is the torque acting on the friction brake and \( \omega_{\text{Boom}} \) is the boom angular velocity. Average cycle power of the system was calculated by approximating the area under the curve by the trapezoidal rule (using Matlab’s TRAPZ function) and dividing by the time duration of the signal. Figure 3.11 shows the instantaneous power and the average cycle power for the system.
From the tow cart position data, tow cart velocity can be calculated by numerical differentiation. Using the same procedure that we used to calculate the angular velocity of the boom from the angular position signal. To calculate the tow car velocity, we used a second order polynomial and a window size of 301 points. Figure 3.12 shows the plot for cart velocity. Average cart velocity was found by integrating the signal and dividing by the time duration.

The last plot that needed to be analyzed was the boom angular velocity versus the torque. Fig. 3.15 shows the boom angular velocity versus the torque. Notice the difference between Fig. 2.4 and Fig. 3.15. It was hard to interpret Fig. 2.4. We believe, this was due to the generator dynamics playing a big role in interfering with our data. However, Fig. 3.15 shows that the torque increases with boom angular velocity. In theory, the angular velocity and the torque relation should look like the right plot in Fig. 3.15. However, this is only when the kinetic co-efficient of friction
Fig. 3.12: Plot shows the tow cart velocity over three cycles

$$(\mu_k)$$ is equal to the static co-efficient of friction $$(\mu_s)$$. Initially, repeatability was a major concern for this system. In order to make claims about how power production is affected by parameter changes, we needed to know how variable our power production measurements were if tests with identical parameters were conducted.

For the data to be reliable, it was necessary that the system reached steady state. One way to check if the system was at steady state was to draw a phase plane between the boom angular position and the boom angular velocity. Fig. 3.13 shows that the system was not repeatable as the closed loop should have been right on top of each other. However a difference of about 0.06 radians was noted between two loops on one side. This meant that the hydrofoil was not flipping at the same $$\theta$$ values. On further investigation shown in Fig. 3.14, we can see that some of the cycle strokes for the boom data have different amplitudes resulting in some strokes being longer whereas the others being shorter.

The non-repeatable boom data was a result of a number of factors. There was
Fig. 3.13: Left plot shows the earlier system without the encoder and new brake did not reach steady-state operation. The right plot shows that on adding the encoder and the new brake, the system reached steady-state, thus making the data reliable.

Fig. 3.14: Plot shows the earlier data was not repeatable. During the tests, boom angle amplitude should have been the same, however, certain peaks are smaller than the others.
Fig. 3.15: Top left plot shows the change in torque with respect to the boom angular velocity when the resistance is provided by the Aluminum brake. Top right plot shows ideal relation between angular velocity and the torque when the kinetic coefficient of friction is equal to the static coefficient of friction. The bottom plot shows the relation between boom angular velocity and the torque for a wood brake.

Some mechanical slipping between the shaft running through the hydrofoil and the hydrofoil body. Also, the potentiometer attached to measure the angular position of the boom was slipping. Even after these issues were managed, we still had repeatability problems. We believe this was due to the noisy signal of the potentiometer, which leads us to make a few changes on the system. A quadrature encoder is now attached to read the boom position accurately and reduce noise. Note that this change in the system has made the system repeatable as can be seen from Fig 4.1. With this data now being reliable, we can conduct our experiments.
Chapter 4

Test Results

4.1 Hydrofoil Angle ($\beta$) Test (Quarter-Chord)

McConnaghy [?] predicted, from a quasi-static simulation, that the power produced by a simple hydrokite system is extremely dependent on the hydrofoil angle and boom angles. To determine if a similar trend would be seen in an experimental test, a one dimensional study was carried out by keeping all parameters constant, except for the hydrofoil angle ($\beta$). The test parameters that are kept constant are listed in Table 4.1.

An initial study for determining how the average cycle power changes with hydrofoil angle was done by taking a step size of $10^\circ$. The preliminary tests done are shown in Table 4.2. A maximum of 0.13 Watts of power was recorded for a hydrofoil

Table 4.1: Parameters held constant during the hydrofoil angle test

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tow Cart speed</td>
<td>0.3</td>
<td>m/s</td>
</tr>
<tr>
<td>Boom Length</td>
<td>20</td>
<td>in</td>
</tr>
<tr>
<td>Boom Deploy Angle</td>
<td>10</td>
<td>deg</td>
</tr>
<tr>
<td>Boom Return Angle</td>
<td>-10</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Depth in Water</td>
<td>7.0</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Chord Length</td>
<td>2.3</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Shape</td>
<td>NACA 0015</td>
<td></td>
</tr>
<tr>
<td>Hydrofoil Pivot Point Location</td>
<td>Quarter Chord point</td>
<td></td>
</tr>
<tr>
<td>Braking Torque</td>
<td>not changed, but also measured</td>
<td></td>
</tr>
</tbody>
</table>
angle of 70°. Tests from 80° to 84° were done with 1° increment to see the change in power production even with just 1° change.

4.1.1 Steady State

Fig. 4.1 shows that the experiment is in steady state. The closed loops are nearly on top of each other in the phase plane, thus showing steady state.

4.1.2 Results

Figure 4.2 shows us the average power produced for a quarter chord NACA 0015. To ensure that the system was repeatable, a number of tests were done without changing any parameters. These tests are shown by the red ‘*’.

From Fig. 4.3 we can see that the error in the results changes depending on the operating point of the system. For example, the mean and standard deviation for
Table 4.2: Constant test parameters for quarter chord NACA 0015 for hydrofoil angle testing

<table>
<thead>
<tr>
<th>Run</th>
<th>$\beta_d$ [°]</th>
<th>$\beta_r$ [°]</th>
<th>Average Cycle Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>-10</td>
<td>0.00041</td>
</tr>
<tr>
<td>2</td>
<td>20</td>
<td>-20</td>
<td>0.01515</td>
</tr>
<tr>
<td>3</td>
<td>30</td>
<td>-30</td>
<td>0.03642</td>
</tr>
<tr>
<td>4</td>
<td>40</td>
<td>-40</td>
<td>0.05675</td>
</tr>
<tr>
<td>5</td>
<td>50</td>
<td>-50</td>
<td>0.09495</td>
</tr>
<tr>
<td>6</td>
<td>60</td>
<td>-60</td>
<td>0.10493</td>
</tr>
<tr>
<td>7</td>
<td>70</td>
<td>-70</td>
<td>0.13007</td>
</tr>
<tr>
<td>8</td>
<td>80</td>
<td>-80</td>
<td>0.11347</td>
</tr>
<tr>
<td>9</td>
<td>82</td>
<td>-82</td>
<td>0.10298</td>
</tr>
<tr>
<td>10</td>
<td>83</td>
<td>-83</td>
<td>0.10318</td>
</tr>
<tr>
<td>11</td>
<td>84</td>
<td>-84</td>
<td>0.09632</td>
</tr>
<tr>
<td>12</td>
<td>85</td>
<td>-85</td>
<td>0.09537</td>
</tr>
</tbody>
</table>

Fig. 4.2: Average cycle power produced by a half chord NACA 0015 over a range of hydrofoil angles
Fig. 4.3: Left plot shows the average tow speed at every angle of hydrofoil where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of hydrofoil angles for quarter chord NACA 0015 hydrofoil.

beta at $80^\circ$ is 0.1098 Watts and 0.00669 Watts whereas for a beta of $84^\circ$ the mean and standard deviation is 0.0946 Watts and 0.0022 Watts. The standard deviation for $80^\circ$ is thrice as much as $84^\circ$. The results show a maximum error of $\pm 11\%$ for repeatability of the experiments done at a hydrofoil angle of $84^\circ$.

To further our understanding as to why the error margin is as large as it is, we looked at the standard deviation of the tow speed of the cart. This is where we notice a possible reason for issues of repeatability. We see that there is a slight increase in the tow speed as we move towards testing higher angles. What is even more interesting is that the tow cart velocity drops after the hydrofoil angle where maximum average cycle power is produced. A maximum standard deviation of 0.1 m/s is recorded. Fig. 4.3 shows the standard deviation in tow speed for all the quarter chord NACA 0015 hydrofoil angle experiments and also shows the average tow speed for the same experiments.

4.2 Hydrofoil Angle ($\beta$) Test (Half-Chord)

Similar to the quarter chord NACA 0015 hydrofoil tests, a new set of tests were carried out for a NACA 0015 hydrofoil with its pivot point located at half of its chord
length. The test parameters for these runs are listed in Table 4.3. Note that the chord length for the half chord hydrofoil is 2.8 inches whereas for the quarter chord hydrofoil it is 2.3 inches. This is because we wished to test the existing hydrofoils and gain an understanding of their performance before designing new ones.

Table 4.3: Constant test parameters for half chord NACA 0015 for hydrofoil angle testing

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tow Cart speed</td>
<td>0.31</td>
<td>m/s</td>
</tr>
<tr>
<td>Boom Length</td>
<td>20</td>
<td>in</td>
</tr>
<tr>
<td>Boom Deploy Angle</td>
<td>10</td>
<td>deg</td>
</tr>
<tr>
<td>Boom Return Angle</td>
<td>-10</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Depth in Water</td>
<td>7.0</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Chord Length</td>
<td>2.8</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Shape</td>
<td>NACA 0015</td>
<td></td>
</tr>
<tr>
<td>Hydrofoil Pivot Point Location</td>
<td>Half Chord point</td>
<td></td>
</tr>
<tr>
<td>Braking Torque</td>
<td>Not changed, but also measured</td>
<td></td>
</tr>
</tbody>
</table>

4.2.1 Steady State

To see if the system has reached steady state, the phase plane was plotted. A visual inspection of Fig. 4.4 shows that a majority of the initial transients in the system have decayed.

4.2.2 Results

For the half chord NACA 0015 hydrofoil, a maximum average cycle power of 0.137 Watts was determined at $\beta = 60^\circ$. Figure 4.5 shows the average cycle power trend for the half chord NACA 0015 hydrofoil. The figure shows that there is no power production at 70$^\circ$ hydrofoil angle. However, there weren’t enough forces acting on the hydrofoil to ensure that it moved across the width of the tank. This may be due to the low speed of tow cart and the low submerged depth of hydrofoil in water. Repeatability tests were done for two angles as shown in the figure. Table 4.4 shows the average cycle power produced for every 10$^\circ$. A maximum error of $\pm 14\%$ in average
Fig. 4.4: Phase plane for boom angular position and boom angular velocity for half chord NACA 0015 hydrofoil

cycle power was found in average cycle power during the number of identical tests done at $\beta = 60^\circ$ was recorded for a hydrofoil angle of $60^\circ$. Standard deviation of 0.007 Watts was calculated at $\beta = 50^\circ$ and 0.0094 Watts at $\beta = 60^\circ$.

Next, we wanted to see if the average tow cart velocity remained constant throughout this set of experiments. However, we notice the same trend in tow velocity as we noticed with the quarter chord NACA 0015 hydrofoil; a slightly higher tow velocity at

Table 4.4: Average cycle power produced for half chord NACA 0015 hydrofoil for various hydrofoil angles

<table>
<thead>
<tr>
<th>Run</th>
<th>$\beta_d$ [$^\circ$]</th>
<th>$\beta_r$ [$^\circ$]</th>
<th>Average Cycle Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>-10</td>
<td>0.00409</td>
</tr>
<tr>
<td>2</td>
<td>20</td>
<td>-20</td>
<td>0.01776</td>
</tr>
<tr>
<td>3</td>
<td>30</td>
<td>-30</td>
<td>0.03591</td>
</tr>
<tr>
<td>4</td>
<td>40</td>
<td>-40</td>
<td>0.07295</td>
</tr>
<tr>
<td>5</td>
<td>50</td>
<td>-50</td>
<td>0.12936</td>
</tr>
<tr>
<td>6</td>
<td>60</td>
<td>-60</td>
<td>0.13773</td>
</tr>
<tr>
<td>7</td>
<td>70</td>
<td>-70</td>
<td>0.00007</td>
</tr>
</tbody>
</table>
higher boom angles and the maximum tow velocity at the boom angle which produces maximum power.

The results of the tow cart velocity are quite intriguing. It is interesting to note that the tow cart velocity is maximum for both quarter and half chord NACA 0015 hydrofoil at their respective maximum power producing hydrofoil angles. However, we are unsure if the higher tow cart speed leads us to higher average cycle power in the system or that the higher hydrofoil angles are somehow causing an increase in tow speed. Figure 4.6 shows the maximum standard deviation of 0.08 m/s for the entire range of hydrofoil angle experiments and Fig. 4.6 shows the average tow speed for the same experiments.
Fig. 4.6: Left plot shows the average tow speed at every angle of hydrofoil where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of hydrofoil angles for half chord NACA 0015 hydrofoil.

4.3 Boom Angle position ($\theta_{flip}$) Test (Quarter-Chord)

Next, the influence of the boom angles on the system was tested. A hydrofoil angle of $50^\circ$ was chosen, and keeping this constant, the boom angles were varied. As the tank width is a constraint, higher boom angles could not be used but a range of boom angles from $0^\circ$ to $30^\circ$ could be used. Parameters used for these test are shown in Table 4.5.

Table 4.5: Test parameters that were constant throughout the runs carried out for boom angles for quarter chord NACA 0015 hydrofoil

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tow Cart speed</td>
<td>0.3</td>
<td>m/s</td>
</tr>
<tr>
<td>Boom Length</td>
<td>20</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Deploy Angle</td>
<td>50</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Return Angle</td>
<td>-50</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Depth in Water</td>
<td>7.0</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Chord Length</td>
<td>2.3</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Shape</td>
<td>NACA 0015</td>
<td></td>
</tr>
<tr>
<td>Hydrofoil Pivot Point Location</td>
<td>Quarter Chord point</td>
<td></td>
</tr>
<tr>
<td>Braking Torque</td>
<td>Not changed, but also measured</td>
<td></td>
</tr>
</tbody>
</table>
Fig. 4.7: Plot shows that the data is repeatable for larger boom angles for a quarter chord NACA 0015 hydrofoil

4.3.1 Steady State

For every set of experiments carried out, repeatability tests were done to ensure that our system always reaches steady state. Figure 4.7 shows the phase plane with the closed loop lying on top of each other which implies repeatability with less than a difference of 0.01 radians. This, we believe, might be just due to the noisy signal of the potentiometer.

4.3.2 Results

Figure 4.8 shows the average cycle power produced by the hydrofoil for change in boom angles. However, there isn’t a significant change in power with change in boom angle. This, we believe, may be due to the pivot point location of the hydrofoil. Due to this, every time the hydrofoil flips, it has enough momentum to swing across the width of the tank. A maximum power of 0.101 Watts was recorded with an error of
Table 4.6: Average cycle power produced for quarter chord NACA 0015 hydrofoil for various boom angles

<table>
<thead>
<tr>
<th>Run</th>
<th>$\theta_d$ $^\circ$</th>
<th>$\theta_r$ $^\circ$</th>
<th>Average Cycle Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>-10</td>
<td>0.09288</td>
</tr>
<tr>
<td>2</td>
<td>15</td>
<td>-15</td>
<td>0.09322</td>
</tr>
<tr>
<td>3</td>
<td>17</td>
<td>-17</td>
<td>0.09144</td>
</tr>
<tr>
<td>4</td>
<td>20</td>
<td>-20</td>
<td>0.09576</td>
</tr>
<tr>
<td>5</td>
<td>23</td>
<td>-23</td>
<td>0.08979</td>
</tr>
<tr>
<td>6</td>
<td>25</td>
<td>-25</td>
<td>0.10166</td>
</tr>
<tr>
<td>7</td>
<td>28</td>
<td>-28</td>
<td>0.09705</td>
</tr>
</tbody>
</table>

Fig. 4.8: Average cycle power produced by a quarter chord NACA 0015 over a range of boom angles
±5% at a boom angle of 25°. Table 4.6 shows the power produced at various boom angles. From the set of identical tests done at boom angles shown by the red ‘*’, the standard deviation in average cycle power is 0.007 Watts at θ = 10°, 0.00234 Watts at θ = 20° and 0.00272 Watts at θ = 25°.

A maximum standard deviation of 0.085 m/s was noticed for the tow speed for the quarter chord NACA 0015 boom angle experiments. We have already seen that there isn’t a significant difference in the power produced during these experiments. However, it is worthy to note the average cycle power produced at 15° is slightly more than the power produced at 10°, but the tow speed at 10° is greater than the tow speed at 15°. However, as there is no significant change in power, we cannot really say how the tow speed affect this data. Fig. 4.9 shows the deviation in tow speed and the average tow speed for each experiment of the quarter chord boom angle tests.

### 4.4 Boom Angle position (θ_{flip}) Test (Half-Chord)

To compare our results of the quarter chord hydrofoil data, the same tests were done with a half chord point. Boom angle is a parameter that shows that location of a the pivot point strongly influences the power trends of a hydrokite system. Now that the
Table 4.7: Test parameters which were constant throughout the runs carried out for boom angles for half chord NACA 0015 hydrofoil

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tow Cart speed</td>
<td>0.31</td>
<td>m/s</td>
</tr>
<tr>
<td>Boom Length</td>
<td>20</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Deploy Angle</td>
<td>50</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Return Angle</td>
<td>-50</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Depth in Water</td>
<td>7.0</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Chord Length</td>
<td>2.8</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Shape</td>
<td>NACA 0015</td>
<td></td>
</tr>
<tr>
<td>Hydrofoil Pivot Point Location</td>
<td>Half Chord point</td>
<td></td>
</tr>
<tr>
<td>Braking Torque</td>
<td>Not changed, but also measured</td>
<td></td>
</tr>
</tbody>
</table>

swing angles are larger, the momentum reduces as the boom moves from one side to the other. However, unlike the quarter chord hydrofoil, the half chord hydrofoil does not have a large force component perpendicular to itself. Due this, we can now see the real power trend for the boom angle. Table 4.6 shows the parameters that were held constant throughout the test.

4.4.1 Steady State

The repeatability test was carried out to see if the data was periodic. Figure 4.10 shows the phase plane for the boom angle and boom angular velocity. The closed loops are nearly on top of each other. There is a small gap of about 0.01 radians between two loops. This, we believe, is due to the noisy signal of the potentiometers.

4.4.2 Results

A general increasing trend of power production was seen with increasing boom angles. A maximum of 0.134 Watts of average cycle power was produced at a boom angle of 26°. Table 4.8 shows the power produced for a range of boom angles. A maximum error of ±5% and a standard deviation of 0.0031 Watts in average cycle power was seen for a boom angle of 20°. Fig. 4.11 shows the average cycle produced by a half chord NACA 0015 hydrofoil.
Fig. 4.10: Plot shows the phase plane for the boom angular position and boom angular velocity for the half chord NACA 0015 hydrofoil. As the closed loops are nearly right on top of each other, the system is said to be in steady state.

Table 4.8: Average cycle power produced for half chord NACA 0015 hydrofoil for various boom angles

<table>
<thead>
<tr>
<th>Run</th>
<th>$\theta_d$ [°]</th>
<th>$\theta_r$ [°]</th>
<th>Average Cycle Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>-10</td>
<td>0.10071</td>
</tr>
<tr>
<td>2</td>
<td>15</td>
<td>-15</td>
<td>0.11613</td>
</tr>
<tr>
<td>3</td>
<td>20</td>
<td>-20</td>
<td>0.12575</td>
</tr>
<tr>
<td>4</td>
<td>25</td>
<td>-25</td>
<td>0.13477</td>
</tr>
<tr>
<td>5</td>
<td>28</td>
<td>-28</td>
<td>0.12541</td>
</tr>
</tbody>
</table>
Fig. 4.11: Average cycle power produced by a half chord NACA 0015 over a range of boom angles.

Fig. 4.12: Left plot shows the average tow speed at every boom angle where tests were conducted. Right plot shows the standard deviation in the tow speed over this range of boom angles for half chord NACA 0015 hydrofoil.
Fig. 4.12 shows the standard deviation of the tow speed for the half chord NACA 0015 hydrofoil. A maximum deviation of 0.026 is noted. From Fig. 4.12 we see that the tow speed fluctuates a little, but is mostly constant throughout the tests. This tells us that the change in average cycle power production is only because of the change in boom angles. However, one of our hypothesis is that the tow speed changes due to higher angles. If this is the case, we wonder why the tow speed does not increase over this range. One reason for this could be that due to the loss in momentum as the boom angle being is large and the NACA 0015 hydrofoil has its pivot location at half chord point.

4.5 Tow Speed Test \((V_\infty)\) Test

We predicted that the power production should increase with increasing water speeds. Considering this, we began our tests with a speed slow enough to just produce positive average cycle power. Tests were carried out in the range of 0.12 m/s to 0.54 m/s. The highest tow speed was 0.54 m/s only because we have a length constraint for the tank and going any higher would only shorten the testing time. Table 4.9 shows the parameters used for this test.

Table 4.9: Test parameters which were constant throughout the runs carried out for tow speeds. The tow speeds were tested only using the quarter chord NACA 0015 hydrofoil

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom Length</td>
<td>20</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Deploy Angle</td>
<td>50</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Return Angle</td>
<td>50</td>
<td>deg</td>
</tr>
<tr>
<td>Boom Deploy Angle</td>
<td>10</td>
<td>deg</td>
</tr>
<tr>
<td>Boom Return Angle</td>
<td>-10</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Depth in Water</td>
<td>7.0</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Chord Length</td>
<td>2.3</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Shape</td>
<td>NACA 0015</td>
<td></td>
</tr>
<tr>
<td>Hydrofoil Pivot Point Location</td>
<td>Quarter Chord point</td>
<td></td>
</tr>
<tr>
<td>Braking Torque</td>
<td>Not changed, but also measured</td>
<td></td>
</tr>
</tbody>
</table>
4.5.1 Steady State

A repeatability test was performed for the tow test speed experiments. Figure 4.13 shows the phase plan for boom angle and boom angular velocity.

4.5.2 Results

Maximum power of 0.1341 Watts was produced at 0.54 m/s speed. However, figure 4.14 shows the general increasing trend of power which leads us to believe that more power is generated at higher speeds. Table 4.10 shows the power produced for a range of water speeds. After conducting a number of tests, without changing the parameters, an error margin of ±9% was recorded. The standard deviation is 0.0022 Watts at 0.19 m/s and 0.0022 Watts at 0.24 m/s.

The standard deviation in tow speed for these experiments is ~ 0.033 m/s. This is quite low in comparison to the standard deviation of 0.15 m/s over the entire range of
Table 4.10: Average cycle power produced for quarter chord NACA 0015 hydrofoil for various water speeds

<table>
<thead>
<tr>
<th>Run</th>
<th>Water Speed [m/s]</th>
<th>Average Cycle Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.12</td>
<td>0.0433</td>
</tr>
<tr>
<td>2</td>
<td>0.14</td>
<td>0.0515</td>
</tr>
<tr>
<td>3</td>
<td>0.16</td>
<td>0.0511</td>
</tr>
<tr>
<td>4</td>
<td>0.19</td>
<td>0.0577</td>
</tr>
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<td>5</td>
<td>0.24</td>
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<td>0.33</td>
<td>0.0841</td>
</tr>
<tr>
<td>8</td>
<td>0.39</td>
<td>0.0944</td>
</tr>
<tr>
<td>9</td>
<td>0.44</td>
<td>0.1111</td>
</tr>
<tr>
<td>10</td>
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<td>0.1288</td>
</tr>
<tr>
<td>11</td>
<td>0.54</td>
<td>0.1341</td>
</tr>
</tbody>
</table>

Fig. 4.14: Average cycle power produced by a quarter chord NACA 0015 over a range of tow speeds
tow speeds. There is a small change in speed through these experiments. However, the average cycle power keeps increasing with increase in tow speed. Fig. 4.15 shows the standard deviation for these tests.

4.6 Submerged Depth ($D$) Test

All previous tests had been done with a submerged depth of 7 inches. To determine how power produced changes with submerged depth, tests were done with varying depths from 7 inches to 13 inches with increments of 1 inch. Table 4.11 shows the parameters used for this test and Fig. 4.17 and table 4.12 show the power produced for different depths of hydrofoil.

4.6.1 Steady State

Figure 4.16 shows the phase plane for the depth tests done on the half chord NACA 0015 hydrofoil. Repeatability tests were done for the very last reading. Also, the individual tests repeatability shows that the data collected after adding the encoder in the system is definitely more repeatable than the old data. Through all the phase planes done so far, we see that the repeatability of the individual tests are comparable,
Table 4.11: Test parameters which were constant throughout the runs carried out for submerged depth of hydrofoil. The depth was tested only using the half chord NACA 0015 hydrofoil

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tow Cart speed</td>
<td>0.31</td>
<td>m/s</td>
</tr>
<tr>
<td>Boom Length</td>
<td>20</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Deploy Angle</td>
<td>50</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Return Angle</td>
<td>50</td>
<td>deg</td>
</tr>
<tr>
<td>Boom Deploy Angle</td>
<td>10</td>
<td>deg</td>
</tr>
<tr>
<td>Boom Return Angle</td>
<td>-10</td>
<td>deg</td>
</tr>
<tr>
<td>Hydrofoil Chord Length</td>
<td>2.8</td>
<td>in</td>
</tr>
<tr>
<td>Hydrofoil Shape</td>
<td>NACA 0015</td>
<td></td>
</tr>
<tr>
<td>Hydrofoil Pivot Point Location</td>
<td>Half Chord point</td>
<td></td>
</tr>
<tr>
<td>Braking Torque</td>
<td>Not changed, but also measured</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 4.16: Plot shows that the system is at steady state for this experiment
Table 4.12: Average cycle power produced for half chord NACA 0015 hydrofoil for submerged depth in water

<table>
<thead>
<tr>
<th>Run</th>
<th>Depth [in]</th>
<th>Average Cycle Power [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7</td>
<td>0.1004</td>
</tr>
<tr>
<td>2</td>
<td>8</td>
<td>0.1241</td>
</tr>
<tr>
<td>3</td>
<td>9</td>
<td>0.1296</td>
</tr>
<tr>
<td>4</td>
<td>10</td>
<td>0.1358</td>
</tr>
<tr>
<td>5</td>
<td>11</td>
<td>0.1429</td>
</tr>
<tr>
<td>6</td>
<td>12</td>
<td>0.1473</td>
</tr>
<tr>
<td>7</td>
<td>13</td>
<td>0.1551</td>
</tr>
</tbody>
</table>

in the sense that, the noise level through all these tests are low and consistent and there is no drastic change. For the identical tests done at 13 inches, the standard deviation in average cycle power is 0.008 Watts.

### 4.6.2 Results

A maximum of 0.155 Watts of power was produced for a depth of 13 inches. The error margin for this parameter is approximately ±10%.

Standard deviation for tow speed for the depth experiments is shown in Fig. 4.18.

A maximum standard deviation of 0.05 was noted. Fig. 4.18 shows the actual tow speeds for the depth experiments. It should be noted that even though there are slight changes in the tow speed for these experiments, the average cycle power generated keeps increases with every inch added to the submerged depth.

### 4.7 Overall Conclusions

All of the above tests examined the power trends of a hydrokite system. These experiments have also determined the sensitivity of the system’s power production with regard to parameter variations.

On comparing the quarter-chord NACA 0015 and the half-chord NACA 0015 hydrofoil with respect to the hydrofoil angles (β) experiments, we see that the power
Fig. 4.17: Average cycle power produced by a quarter chord NACA 0015 over a range of submerged depth of hydrofoil.

Fig. 4.18: Left plot shows the average tow speed at every test conducted for submerged depth of hydrofoil. Right plot shows the standard deviation in the tow speed over this range of depth for half chord NACA 0015 hydrofoil.
trend is quite similar (Fig. 4.2, Fig. 4.5). There is an increasing trend till a certain peak, and then the power drops. However, the peak for quarter-chord hydrofoil is at 70° whereas the peak for the half-chord hydrofoil is at 60°.

Comparing these experimental results to McConnaghy’s steady-state simulation results [19], there are similar trends when varying the hydrofoil angle. Fig. 4.19 shows the region of peak cycle power production to be at a deploy angle of 92.5° and a return angle of −92.5°. Note that McConnaghy examined different hydrofoil angles on the deploy and return, and the work presented in this thesis examined hydrofoil angles which were symmetric only. By looking at only symmetric combinations of hydrofoil angles in Fig. 4.19 one can see that this would follow a 45° line on the plot. Along this line one can see the power increase with increasing hydrofoil angle until a sharp peak is reached and then the power drops off dramatically. Although the total power production values are drastically different than the power our system produces, the two systems are quite different. This difference is likely due to McConnaghy considering a much larger system. Another reason for the difference is because McConnaghy’s system is at steady state (acceleration at every instance is zero) due to which the power production is at its upper bound. However, the trend of increasing power production with increasing hydrofoil angle followed by a sharp drop off in power at the hydrofoil angle if increased further was seen in the experimental data presented in this thesis. Given the differences in assumptions used in McConnaghy’s simulation work and the experimental work as well as the size differences, the existence of similar trends in power production for varying hydrofoil angles is promising.

Next step was to compare and see if a similar power trend was observed for the quarter-chord hydrofoil NACA 0015 and the half-chord NACA 015 hydrofoil with respect to change in the boom angles (θ). However, we see that there isn’t much change in the power produced by the quarter-chord NACA 0015 hydrofoil over the range of the boom angles 4.8. This, we believe, may be due to the large force
Fig. 4.19: Plot shows the results from McConnaghy’s work. The results show cycle power as a function of hydrofoil angles.

component which is perpendicular to the hydrofoil. This force is created due to the quarter-chord pivot location. On the other hand, the half-chord NACA 0015 hydrofoil shows an increasing trend with increasing boom angle.

Lastly, two independent parameters, the tow speed of the cart and the hydrofoil submerged depth, were tested. For both of these parameters, power will only increase with an increase in the tow speed and submerged hydrofoil depth.

4.8 Main Contributions

- Completed an initial systematic study of power production, as system parameters are varied (hydrofoil angle, boom flip angle, flow speed, submerged depth, hydrofoil flip location[quarter-chord vs half-chord])
- Quantified the variability in the testbed’s hydrofoil angle and towcart speed
- Improved the resistive load design to better match a simple theoretical model. This will facilitate future model validation studies.
4.9 Future Work

From the testing that was done, we realized that significant power could be produced if the resistive load could be further increased. However, due to time constraints this could not be done, since it would require a new friction brake design. Testing changes in power production due to increased resistive load would be a promising area of exploration.

In addition, determining the performance changes due to system scaling in a realistic environment, such as the Genesee river, would be of significant interest. For simplicity, the first step would be to have the base on a bank of the river and then put the hydrokite in it. As these tests advance, we can move the system to have a base in the middle of the river. Also, the system is currently using a rigid boom, but there are options to explore flexible boom. This would be similar to the concept of high altitude variable length tethered air kites.
Bibliography


Appendix A

Data Analysis Code

A.1 System Repeatability Code

```matlab
% This program will read in the data files from the tow-tank

clear; %clear stored data
close all; %close all figure windows

clc;
format long; %Set display format for output in 15-digit (long) ...
scaled fixed point
format compact; %Set display format for output, basically rounds off ...
the number

run_number_start = 10; % Trial selected for testing repeatability
run_number_end = 10;
color = ['b','g','r','k','y','b','g','r','k','r'];
save_plots = 'yes'; %or 'yes' or 'noo'

for i = run_number_start:run_number_end,
    file_name= sprintf('b%d.txt',i);
    file=fopen(file_name);
    Data = textscan(file,'%f %f %f %f','headerlines',14); % Data input from ...
text file containing 4 analog signals
    Load=Data{1}; % Input load cell data per milli second from text file
    Cart=Data{2}; % Input tow cart position data per ms
    Boom=Data{3}; % Input boom's angular position data per ms
    Hydrofoil=Data{4}; % Input hydrofoil angular position data per ms
    time = [0.001:0.001:length(Load(:,1))/1000]; % data every 1 ms
    time = time';
```
% calibration for tow-cart position

cart.meters = -0.8402.*Cart + 4.521; % Calibration for tow cart position....
    Position is in meters.

% calibration for load cell
loadcell_eqn= -2.2169*Load - 0.2719; % Calibration for load cell in N
loadcell_eqn1=(loadcell_eqn);

momentarm= 2.75; % in inches
moment.metr= 0.0254*momentarm; % in meters
torque = loadcell_eqn1*moment.metr; % torque in N*m

% calibration for boom
boom_eqn1= (-207.65*Boom) + 658.76; % Equation for boom calibration ...
    curve in degrees
boom_eqn=boom_eqn1*(pi/180); % Rewriting and converting from ...
    degrees to radians

% hydrofoil calibration
hydrofoil_eqn1= (191.37*Hydrofoil) - 445.2; % Calibration of hydrofoil in ...
    degrees
hydrofoil_eqn=hydrofoil_eqn1*(pi/180); % Converting from degrees to ... 
    radians

figure(1); % Input data from text file and plot all 4 analog ...
    signals after calibration
clf;
hold on;
subplot(4,1,1);
plot(time,cart.meters); % Tow cart data
title(title.name);
xlabel('time [sec]');
ylabel('cart position [meters]');
grid on
subplot(4,1,2);
plot(time,boom_eqn1); % Boom position data
xlabel('time [sec]');
ylabel('boom angle [deg]');
grid on
subplot(4,1,3);
plot(time,hydrofoil_eqn1); % Hydrofoil position data
xlabel('time [sec]');

figure(1); % Input data from text file and plot all 4 analog ...
    signals after calibration
clf;
hold on;
subplot(4,1,1);
plot(time,cart.meters); % Tow cart data
title(title.name);
xlabel('time [sec]');
ylabel('cart position [meters]');
grid on
subplot(4,1,2);
plot(time,boom_eqn1); % Boom position data
xlabel('time [sec]');
ylabel('boom angle [deg]');
grid on
subplot(4,1,3);
plot(time,hydrofoil_eqn1); % Hydrofoil position data
xlabel('time [sec]');
78 ylabel('hydro angle [deg]');
79 grid on
80 subplot(4,1,4);
81 plot(time,torque); % Load cell data
82 xlabel('time [sec]');
83 ylabel('torque [N.m]');
84 grid on
85
86 %---------------------------------------
87 % Choose data window manually. Selection of two points to choose data.
88 % This metod helps us eliminate the transients in the system.
89 [x,y] = ginput(2);
90 [il] = find(x(1)<time,1);
91 il = il-1;
92 [ir] = find(x(2)<time,1);
93 hold on
94 subplot(4,1,2);
95 plot(time(il),boom_eqn(il),'rx');
96 plot(time(ir),boom_eqn(ir),'rx');
97
98 %---------------------------------------
99 % Modify data sets between the two selected data points for all 4
100 % signals
101 time_mod = time(il:ir);
102 cart_mod = cart_meters(il:ir);
103 boom_mod = boom_eqn(il:ir);
104 hydro_mod = hydrofoil_eqn(il:ir);
105 load_mod = torque(il:ir);
106
107 if (save_plots == 'yes'), % Saving plots in pdf format
108    fn = sprintf('a_raw%d',1);
109    print(gcf,'-dpdf', fn);
110    fn = sprintf('%d_raw',1);
111    print(gcf,'-dpdf', fn);
112 end
113
114 %---------------------------------------
115 % Plot the hydrofoil angle and the torque on top of each other and ...
116    subtract
117 % DC offset
118
119 figure(2);
120 clf;
121 hold on;
plot(time_mod, hydro_mod - mean(hydro_mod), 'r');
ylabel('angle [rad]');
plot(time_mod, load_mod - mean(load_mod), 'b');
xlabel('time [sec]');
legend('hydroang', 'torque');
grid on;

if (save_plots == 'yes'),  % save plot
    fn = sprintf('b_select%d', 1);
    print(gcf, '-dpdf', fn);
    fn = sprintf('%d_select', 1);
    print(gcf, '-dpdf', fn);
end

%-----------------------------------
% plot boom angular position and hydrofoil angular position with the ... new
% selected data
figure(3)
subplot(2,1,1);
plot(time_mod, boom_mod);
title(title_name);
xlabel('time [sec]');
ylabel('boom angle [rad]');
grid on;
 subplot(2,1,2);
plot(time_mod, hydro_mod);
xlabel('time [sec]');
ylabel('hydrofoil angle [rad]');
grid on;
if (save_plots == 'yes'),  % save plot
    fn = sprintf('c_hydro_angle%d', 1);
    print(gcf, '-dpdf', fn);
    fn = sprintf('%d_hydro_angle', 1);
    print(gcf, '-dpdf', fn);
end

%-----------------------------------
% calculate boom angular velocity so as to achieve power. Used to
% smooth out the noisy signal
figure(4);
win = 107; % number of data points to include in the data window
order = 2; % order of polynomial to fit data
[xnew, ang_vel] = deriv_poly(time_mod, boom_mod, win, order);
 subplot(2,1,1)
plot(xnew,ang_vel); % plot for time versus angular velocity of boom in...
  rad/sec
title(title_name);
xlabel('time (s)');
ylabel('boom angular velocity (rad/sec)');
grid on;
subplot(2,1,2)
plot(time_mod(ceil(win/2):end-floor(win/2)),boom_mod(ceil(win/2):end...
  floor(win/2)));
xlabel('time (s)'); % plot for boom angular position after reducing ...
  noise
ylabel('boom angular position [rad/sec]');
grid on;
if (save_plots == 'yes'), % save plot
  fn = sprintf('d_derivative%d',1);
  print(gcf,'-dpdf', fn);
  fn = sprintf('%d_derivative',1);
  print(gcf,'-dpdf', fn);
end

%---------------------------
%Phase plane plots for boom angular position and boom angular velocity
%to ensure repeatability. This phase plane shows that there is no slip
figure(5);
hold on;
plot(boom_mod(ceil(win/2):end-floor(win/2)),ang_vel,color(i));
title(title_name);
xlabel('boom angle (rad)');
ylabel('boom angular velocity (rad/sec)');
grid on;
if (save_plots == 'yes'), % save plot
  fn = sprintf('e_phaseplane%d',1);
  print(gcf,'-dpdf', fn);
  fn = sprintf('%d_phaseplane',1);
  print(gcf,'-dpdf', fn);
end

%---------------------------
%Plot for boom angular velocity versus torque
figure(6);
plot(ang_vel, load_mod(ceil(win/2):end-floor(win/2)));
title(title_name);
xlabel('boom angular velocity (rad/sec)');
ylabel('torque (N.m)');
grid on;
if (save_plots == 'yes'),
fn = sprintf('f_torque%d',1);
print(gcf,'-dpdf', fn);
fn = sprintf('%d_torque',1);
print(gcf,'-dpdf', fn);

end

%--------------------------------------------------------
% Plot to find average tow cart velocity. Using a polynomial fit, the
% noise is reduced
figure(7)
cart_win = 301; %window uses odd numbers only
[time_c_vel,cart_vel] = deriv_poly(time_mod,cart_mod,cart_win,2);
plot(time_c_vel,cart_vel);
grid on
xlabel('time [sec]')
ylabel ('velocity of tow cart [m/s]')

avg_cartvel= mean(cart_vel); %average tow cart velocity for the run
hold on;
plot([xnew(1),xnew(end)],[avg_cartvel,avg_cartvel],'r');
cartvel_text = sprintf('average cart velocity = %3.3f m/s','...
  avg_cartvel);
text(xnew(1),min(cart_vel),cartvel_text);
if (save_plots == 'yes'), % save plot
fn = sprintf('h_cartvelocity%d',1);
print(gcf,'-dpdf', fn);
fn = sprintf('%d_cartvelocity',1);
print(gcf,'-dpdf', fn);
end

%--------------------------------------------------------
% Phase plane for boom position and hydrofoil position to check
% repeatability of data
figure(20);
plot(boom_mod,hydro_mod,color(i));
title(title_name);
xlabel('Boom position (rad)');
ylabel('Hydrofoil position (rad)');
grid on;
if (save_plots == 'yes'), % save plot
  fn = sprintf('f_boomvsfoil%d',1);
  print(gcf,'-dpdf', fn);
  fn = sprintf('%d_boomvsfoil',1);
  print(gcf,'-dpdf', fn);
end
A.2 Power Calculation Code

```matlab
1 clear;
2 close all;
3
4 clc;
5 run_number_start = 1;
6 run_number_end = 12;
7 save_plots = 'yes'; % or 'yes' or 'no'
8 j=1;
9
10 for i = run_number_start:run_number_end,
11    file_name = sprintf('b%d.txt',i);
12    file=fopen(file_name);
13    Data = textscan(file,'%f %f %f %f','headerlines',14); % Data input from ...
14      text file containing 4 analog signals
15    Load=Data{1}; % Input load cell data per milli second from text file
16    Cart=Data{2}; % Input tow cart position data per ms
17    Boom=Data{3}; % Input boom's angular position data per ms
18    Hydrofoil=Data{4}; % Input hydrofoil angular position data per ms
19    time = [0.001:0.001:length(Load(:,1))/1000]; % data every 1 ms
20         time = time';
21
22    title_name = sprintf('trial #%d',i);
23
24    % calibration for tow-cart position
25
26    cart_meters = -0.8402.*Cart + 4.521; % position in meters
27
28    % calibration for load cell
29    loadcell_eqn= -2.2169*Load - 0.2719; % Calibration of load cell in N
30    loadcell_eqn1=(loadcell_eqn);
31    momentarm=2.75; % in inches
32    moment_metr=0.0254*momentarm; % in meters
33    torque = loadcell_eqn1*moment_metr; % torque in N*m
34
35    % boom calibration
36    boom_eqn1= (-207.65*Boom) + 658.76; % equation from boom calibration ...
37      curve in degrees
```
boom_eqn=boom_eqn1*(pi/180); % converting from deg to rad

% hydrofoil calibration
hydrofoil_eqn1= (191.37*Hydrofoil) - 445.2; % calibration of hydrofoil in deg
hydrofoil_eqn=hydrofoil_eqn1*(pi/180); % calibration of hydrofoil in rad

figure(1); % Input data from text file and plot all 4 analog signals after calibration
clf;
hold on;
subplot(4,1,1);
plot(time,cart_meters); % Tow cart data after calibration
xlabel('time [sec]');
ylabel('cart position [m]');
grid on
subplot(4,1,2);
plot(time,boom_eqn1); % Boom position data
xlabel('time [sec]');
ylabel('boom angle [deg]');
grid on
subplot(4,1,3);
plot(time,hydrofoil_eqn1); % Hydrofoil position data
xlabel('time [sec]');
ylabel('hydro angle [deg]');
grid on
subplot(4,1,4); % Load cell data
plot(time,torque);
xlabel('time [sec]');
ylabel('torque [Nm]');
grid on

% Choose data window manually. Selection of two points to choose data.
% This method helps us eliminate the transients in the system.
[x,y] = ginput(2);
[il] = find(x(1)<time,1);
il = il-1;
[ir] = find(x(2)<time,1);
hold on
subplot(4,1,2);
plot(time(il),boom_eqn(il)*(180/pi),'rx');
plot(time(ir),boom_eqn(ir)*(180/pi),'rx');
%____________________________
% Modify data sets between the two selected data points for all 4 % signals
time_mod = time(il:ir);
cart_mod = cart_meters(il:ir);
boom_mod = boom_eqn(il:ir);
hydro_mod = hydrofoil_eqn(il:ir);
load_mod = torque(il:ir);

if (save_plots == 'yes'), % Saving plots in pdf format
    fn = sprintf('a_raw%d',i);
    print(gcf,'-dpdf', fn);
    fn = sprintf('%d_raw',i);
    print(gcf,'-dpdf', fn);
end

%Plots only for the new selected range after eliminating the %transients
figure(2)
subplot(2,1,1);
plot(time_mod,boom_mod); %Boom angular position plot after selecting ... 
    title(title_name);
    xlabel('time [sec]');
    ylabel('boom angle [rad]');
    grid on;
subplot(2,1,2);
plot(time_mod,hydro_mod); %Hydrofoil angular position plot after ... selecting the data
    xlabel('time [sec]');
    ylabel('hydrofoil angle [rad]');
    grid on;
if (save_plots == 'yes'), %save plot
    fn = sprintf('c_hydro_angle%d',i);
    print(gcf,'-dpdf', fn);
    fn = sprintf('%d_hydro_angle',i);
    print(gcf,'-dpdf', fn);
end

%____________________________
%calculate boom angular velocity to get power produced
figure(3);

win = 107; %number of data points to include in the data window
order = 2; %order of polynomial to fit data
[xnew,ang_vel] = deriv_poly(time_mod,boom_mod,win,order);
subplot(2,1,1) % time versus boom angular velocity
title(title_name);
xlabel('time (s)');
ylabel('boom angular velocity (rad/sec)');
grid on;
subplot(2,1,2) % time versus boom angular position
plot(time(mod(ceil(win/2):end-floor(win/2)),boom_mod(ceil(win/2):end-floor(win/2)));
xlabel('time (s)');
ylabel('boom angular position [rad]');
grid on;
if (save_plots == 'yes'), % save plot in pdf
    fn = sprintf('d_derivative%d',run_number);
    print(gcf,'-dpdf', fn);
    fn = sprintf('%d_derivative',run_number);
    print(gcf,'-dpdf', fn);
end

% Plot for power produced for 1 cycle
figure(4)
power = - (torque(il+floor(win/2):ir-floor(win/2))).*ang_vel; % power = torque*angular velocity
plot(xnew,power);
title(title_name);
xlabel('time (s)');
ylabel('Power (W)');
avg_power = trapz(xnew,power)/(xnew(end)-xnew(1)) % average power...
    produced for the cycle
avg_pow(j) = avg_power;
hold on;
plot([xnew(1),xnew(end)],[avg_power,avg_power],'r');
power_text = sprintf('average power = %2.1f Watts',avg_power); % text on...
    plot to see where the average power lies
text(xnew(1)+0.5,min(power)-0.1,power_text);
grid on
if (save_plots == 'yes'), % save plot
    fn = sprintf('g_power%d',run_number);
    print(gcf,'-dpdf', fn);
    fn = sprintf('%d_power',run_number);
    print(gcf,'-dpdf', fn);
end
j=j+1;

%something something
%Calculate the average tow cart velocity

figure(5)
cart_win = 301; %select data window for cart position
[time_c_vel,cart_vel] = deriv_poly(time_mod,cart_mod,cart_win,2);
figure
plot(time_c_vel,cart_vel); %plot time versus cart speed
grid on
xlabel('time [sec]')
ylabel ('velocity of tow cart [m/s]')

avg_cartspeed= mean(cart_vel)%Find average cart speed
hold on;
plot([xnew(1),xnew(end)],[avg_cartspeed,avg_cartspeed],'r');
cartvel_text = sprintf('average cart velocity = %3.3f m/s','avg_cartspeed);
text(xnew(1),min(cart_vel),cartvel_text);

if (save_plots == 'yes'), % save plot
  fn = sprintf('h_cartvelocity%d',run_number);
  print(gcf,'-dpdf', fn);
  fn = sprintf('%d_cartvelocity',run_number);
  print(gcf,'-dpdf', fn);
  end
%
% Plot between angular velocity of boom and torque

figure(6);
plot(ang_vel, load_mod(ceil(win/2):end-floor(win/2)));
title(title_name);
xlabel('boom angular velocity (rad/sec)');
ylabel('torque (N.m)');
grid on;
end
%
%Plot the complete plot for power for change in hydrofoil angles

figure(7)
beta= [10,20,30,40,50,60,70,80,82,83,84,85];
plot (beta, avg_pow, 'o-');
ylabel('average power [W]');
xlabel('Hydrofoil angle [Nm]');
grid on
if (save plots == 'yes'), % save plot
fn = sprintf('powerplot%d',1);
print(gcf,'-dpdf', fn);
fn = sprintf('%d powerplot',1);
print(gcf,'-dpdf', fn);
end

A.3 Code for entire range of data with repeatability tests

clear all
clc

save plots = 'yes';
% loading four files to compare their repeatability for ceratin angles
load('fullplot');
load('rep50');
load('rep80');
load('rep84');

% Plot full range of the tested parameter
a = fullplot(1,:); % hydrofoil angles
b = fullplot(2,:); % average cycle power produced for cycle

% check repeat test for torque w.r.t. beta angles

figure(1)
plot(a,b,'-o','linewidth',2,'color','b')
hold on
plot(c,d,'*','color','r')
hold on
A.4 Microcontroller Code for Hydrofoil Control (Arduino)

```c
/*
   This program sets up the dip switches for changing the hydrofoil angle of attack
*/
#include <Servo.h>

Servo myservo; // create servo object to control a servo
#define encoder0PinA 2  //Digital pin used to connect the first signal of... quadrature encoder
#define encoder0PinB 3  //Digital pin used to connect the second signal... of quadrature encoder
volatile double encoder0Pos = 0; //Variable to read the encoder position
double degCount = 0;  //Variable to read the encoder position in degrees

double hydro_ang[] = {0.0, 40.0, 50.0, 60.0, 70.0}; // array of desired... +/- hydrofoil angles in [deg] values to be chosen from dip switch... inputs
double hydro_R;  //in deg
double hydro_D;  //in deg
double hydro_R_servo; // in servo command value
double hydro_D_servo; // in servo command value
int hydro_middle = 93; //servo command which makes the hydrofoil have a ... beta angle of zero
double slope = 0.383;  //slope conversion for beta angles to servo command... values
```
int state = 0; // state of the system:
    // 0 when in deploy stroke
    // 1 when in return stroke
int one; // dip switch for when beta is zero
int two; // dip switch from two to five take in values for different ...
    // hydrofoil angles
int three;
int four;
int five;
int six;
int value=0;

// Used to find the servo position when beta is zero
int button1; // Increment of 1
int button2; // Decrement of 1

// the setup routine runs once when you press reset:
void setup() {
    // for dipswitch inputs
    pinMode(encoder0PinA, INPUT); // Input 1 from encoder
    pinMode(encoder0PinB, INPUT); // Input 2 from encoder
    attachInterrupt(0, doEncoderA, CHANGE); // to trigger the interrupt ...
    attachInterrupt(1, doEncoderB, CHANGE);
    pinMode(4, INPUT); // dipswitch1
    pinMode(5, INPUT); // dipswitch2
    pinMode(6, INPUT); // dipswitch3
    pinMode(7, INPUT); // dipswitch4
    pinMode(8, INPUT); // dipswitch5
    pinMode(9, INPUT); // dipswitch6
    pinMode(10, INPUT); // for pushbutton to zero servo +1
    pinMode(12, INPUT); // for pushbutton to zero servo -1
    pinMode(11, OUTPUT); // servo output
    pinMode(13, OUTPUT); // status led output
    myservo.attach(11); // attaches the servo on pin 11 to the servo object
    Serial.begin(9600); // initialize serial communication at 9600 bits per ...
    second
}
// the loop routine runs over and over again forever
void loop() {
  six = digitalRead(9); // switch 6 has to be ON to take in any new value ...
  for the beta angle
  degCount = encoder0Pos*360.0/4000.0; // convert encoder count to degrees
  if (six == 1) { // do setup of the hydrofoil angles
    one = digitalRead(4);
    two = digitalRead(5);
    three = digitalRead(6);
    four = digitalRead(7);
    five = digitalRead(8);
    value = one*1 + two*2 + three*3 + four*4 + five*5; // to check which ...
    switch is ON
    hydro_D = hydro_ang[value-1]; // to assign the deploy angle from the ...
    hydrofoil array
    hydro_R = (-1)*hydro_D; // Assign the return hydrofoil angle
    hydro_D_servo = hydro_D*slope + hydro_middle; // Final deploy angle is...
    found by converting this angle in terms of servo count
    hydro_R_servo = hydro_R*slope + hydro_middle; // Final return angle is...
    found by converting this angle in terms of servo count
    if (value == 1) { // manually set the zero hydrofoil position
      button1 = digitalRead(10);
      button2 = digitalRead(12);
      delay(500);
      if (button1 == 1) {
        hydro_middle++;
      }
      else if (button2 == 1) {
        hydro_middle--;
      }
      myservo.write(hydro_D_servo); // sets the servo ...
      position according to the scaled value
    }
  }
  // print the results to the serial monitor. Only when switch 1 is ON
  Serial.print(one);
  Serial.print(two);
  Serial.print(three);
  Serial.print(four);
Serial.print(five);
Serial.print(six);
Serial.print('\t');
Serial.print(value);
Serial.print('\t');
Serial.print(hydro_angle[value-1]);
Serial.print('\t');
Serial.print("buttons=");
Serial.print(button1);
Serial.print(button2);
Serial.print(hydro_middle);
Serial.print("Encoder = ");
Serial.println(encoder0Pos);

delay(10);
}

else { // no setup just quickly read and control the servo for flips

if (encoder0Pos == 111 && state==0) {
  //encoder count of 111 is 10 ...
  myservo.write(hydro_D_servo); //set servo at hydro_D_servo position
  state=1;
}

if (encoder0Pos == -111 && state==1){ //encoder count of -111 is -10 ...
  myservo.write(hydro_R_servo); //set servo at hydro_R_servo position
  state=0;
}

}

void doEncoderA(){
  // look for a low-to-high on channel A
  if (digitalRead(encoder0PinA) == HIGH) {
    // check channel B to see which way encoder is turning
    if (digitalRead(encoder0PinB) == LOW) {
      encoder0Pos = encoder0Pos + 1; // CW

    }
  }
}
else {
    encoder0Pos = encoder0Pos - 1;  // CCW
}

}  // must be a high-to-low edge on channel A

else { // check channel B to see which way encoder is turning
    if (digitalRead(encoder0PinB) == HIGH) {
        encoder0Pos = encoder0Pos + 1;  // CW
    }  // check channel B to see which way encoder is turning
    else {
        encoder0Pos = encoder0Pos - 1;  // CCW
    }  // check channel B to see which way encoder is turning
}

void doEncoderB() {
    // look for a low-to-high on channel B
    if (digitalRead(encoder0PinB) == HIGH) {
        // check channel A to see which way encoder is turning
        if (digitalRead(encoder0PinA) == HIGH) {
            encoder0Pos = encoder0Pos + 1;  // CW
        }  // check channel A to see which way encoder is turning
        else {
            encoder0Pos = encoder0Pos - 1;  // CCW
        }  // check channel A to see which way encoder is turning
    }  // look for a low-to-high on channel B
    else { // check channel B to see which way encoder is turning
        if (digitalRead(encoder0PinA) == LOW) {
            encoder0Pos = encoder0Pos + 1;  // CW
        }  // check channel B to see which way encoder is turning
        else {
            encoder0Pos = encoder0Pos - 1;  // CCW
        }  // check channel B to see which way encoder is turning
    }  // check channel B to see which way encoder is turning
}
A.5 LabVIEW Data Collection Code

Fig. A.1: LabVIEW code used to collect data from all four analog signals
Appendix B

Calibration

All of the sensors that were used in this work were calibrated for their specific application. Four analog sensors, one load cell, and three potentiometers were calibrated.

B.1 Load Cell

Standard known weights were used to calibrate the load cell. The weights were measured using a standard digital scale.

B.1.1 Tension Only: Vertical Mount

The load cell was initially calibrated by simply hanging weights on it when it was oriented vertically with the top of the load cell being held rigid and the bottom end being free to hang mass on. The load cell is rated to 7.45 N so the range of mass used for the calibration was varied from 0 N to 5.49 N. A linear fit is seen from Fig. B.1 and table B.1. The best fit line is given by:

\[ F_m[N] = -2.2494V_o - 0.5417 \]  (B.1)
Table B.1: Load cell calibration when only tension loading is taken into consideration (vertical orientation)

<table>
<thead>
<tr>
<th>Output Voltage [V]</th>
<th>Force [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.24</td>
<td>0</td>
</tr>
<tr>
<td>0.96</td>
<td>-0.67</td>
</tr>
<tr>
<td>1.84</td>
<td>-1.06</td>
</tr>
<tr>
<td>3.66</td>
<td>-1.87</td>
</tr>
<tr>
<td>5.49</td>
<td>-2.68</td>
</tr>
</tbody>
</table>

Fig. B.1: Load cell calibration
B.1.2 Tension and Compression: Horizontal mount

A test run was done with this calibration and data was collected in LabVIEW. As the system is symmetric during the deploy and return stroke, the forces during these two strokes were also expected to be equal in magnitude but opposite in direction. However, from the reading in LabVIEW, the force was greater during one stroke than the other. This was believed to be due to the load cell used, is primarily a unidirectional load cell used only to measure only tension loading. A description of the load cell used from the manufacturer’s website (Phidgets.com): “Load cells are designed to measure force in one direction. They will often measure force in other directions, but the sensor sensitivity will be different, since parts of the load cell operating under compression are now in tension, and vice versa.”

Since we were observing differences in load when the load cell was in tension versus when it was in compression, a testbed was created to calibrate the load cell in both compression and tension. A new test setup was designed and manufactured to test this load cell under tension and compression. Table B.2 shows the calibration data.
Fig. B.3: Calibration of load cell in tension and compression

Fig. B.2 shows the setup for the load cell calibration. The calibration equation for the load cell is

\[ F_m[N] = -2.2168V_o - 0.2802 \]  \hspace{1cm} (B.2)

After calibrating, it was observed that there was not much difference in slope for loading in tension \((slope = -2.27)\) or compression \((slope = -2.2)\) as shown in Fig. B.3. Since the load cell sensitivity was the same in compression and in tension, we hypothesized that the difference in load values in tension and compression were due to the low geometric tolerances of the wood brake. Symmetric loading in compression and tension was observed in tests with the newly designed aluminum brake, which
provides a better surface contact with the boom shaft than the earlier wood brake.

B.2 Tow cart position potentiometer

A 10-turn 10kΩ potentiometer is used for measuring the distance between the tow-cart and the start of the tank. Several positions of the tow cart were selected and their sensor values were measured and the distance from the start of the tank and the two cart was measured using a carpenter’s tape measure (which had a resolution of 1/16 of an inch). At the beginning of the run, $V_o$ is maximum and as the distance increases, the value of $V_o$ drop linearly. Table B.3 and Fig. B.4 shows the calibration data collected. A linear fit was done and the calibration curve for the towcart position sensor is given by

$$\text{Distance}[\text{in}] = -0.8402V_o + 4.521 \quad (B.3)$$

Table B.3: Tow cart calibration

<table>
<thead>
<tr>
<th>Output Voltage [V]</th>
<th>Tow cart Travel [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.18</td>
<td>1.01</td>
</tr>
<tr>
<td>3.13</td>
<td>1.89</td>
</tr>
<tr>
<td>1.95</td>
<td>2.88</td>
</tr>
<tr>
<td>0.89</td>
<td>3.78</td>
</tr>
</tbody>
</table>
Fig. B.4: Tow cart Calibration
The only problem associated with this potentiometer was that the nut holding the potentiometer shaft was extremely loose. So the sensor assembly was removed and the nut was securely tightened. Daily checks were performed and showed that the nut did not loosen over time.

**B.3 Boom angle potentiometer**

A 3-turn 1 kΩ linear potentiometer is used to measure the boom angular position. The excitation voltage for the potentiometer was 5 VDC and was provided by a regulated power supply. Five angles were selected for the calibration test. For these angles the voltage reading ($V_o$) was recorded. To determine the angle of the boom with respect to the sides of the tank, a linear measurement and geometrical relations were used. The position of the boom forms a triangle as shown in Fig. [B.5](#).

An inexpensive 24-inch caliper (resolution 1/16th inch), was used to measure two sides of a triangle ($d_1$ and $d_2$) see Fig. [B.5](#). We assumed that the cross-bar of the tow-cart was perpendicular to the tank rails and used this to orient our measurements. We used the following equation to determine the boom angle:

\[
\theta = \arctan\left(\frac{d_1}{d_2}\right)
\]
Table B.4: Boom position calibration

<table>
<thead>
<tr>
<th>Output Voltage [V]</th>
<th>Boom Angle [degrees]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.21</td>
<td>-27</td>
</tr>
<tr>
<td>3.16</td>
<td>-15</td>
</tr>
<tr>
<td>3.09</td>
<td>0</td>
</tr>
<tr>
<td>3.0</td>
<td>15</td>
</tr>
<tr>
<td>2.95</td>
<td>27</td>
</tr>
</tbody>
</table>

\[
\theta[\text{degrees}] = (\tan^{-1}(d_1/d_2)) \times 180/\pi \tag{B.4}
\]

The calibration equation is:

\[
\theta[\text{degrees}] = -207.65V_o + 640.07 \tag{B.5}
\]

Figure B.6 shows the calibration for the angular position of boom.

Fig. B.7 shows the assembly for the boom potentiometer. To ensure that the
Fig. B.7: Figure shows the boom potentiometer assembly on the tow cart. A wooden bracket and a set screw has been used to ensure that the potentiometer provides true positioning of the boom.
potentiometer doesn’t slip, a wooden bracket is used between the potentiometer and the brake. Also, a set screw is used to avoid any relative motion between the potentiometer and the shaft.

B.4 Hydrofoil angle potentiometer

A 3-turn 1 kΩ potentiometer was used to measure hydrofoil angle. The excitation voltage for the sensor was 5 VDC and was provided by a regulated power supply. The servomotor is set at a position where the hydrofoil angle ($\beta$) is zero. To determine that the hydrofoil was at a zero angle relative to the boom a method was needed to determine the hydrofoil angle. The rectangular blank from which the foam was cut was attached to the hydrofoil to give a straight surface which gave a consistent line relative to the chord line of the hydrofoil. A straight edge of approximately 13 inches long was attached to the flat surface of the foam block to extend the flat line. Using a standard 6 inch caliper, two measurements (10.5 inches apart) were made between the straight edge attached to the foam block surrounding the hydrofoil and the boom arm of the system. A zero angle was defined to be when the straight edge was parallel to the boom. When both readings of the caliper were the same, the hydrofoil would be at a zero relative angle. The servomotor is now rotated to different angles. These angles can again be measured with the help of a caliper and trigonometry as was done for the boom angles. The potentiometer readings ($V_o$) at these positions are also recorded and shown in table [B.5](#) and Fig. [B.8](#)

$$\beta[degrees] = 188.05V_o - 522.46$$  \hspace{1cm} (B.6)

After a test was carried out and analyzed, it showed that the hydrofoil didn’t return to its initial position. On further investigation, it was found that the metal shaft that runs through the hydrofoil was slipping with respect to the hydrofoil. This
Table B.5: Hydrofoil calibration

<table>
<thead>
<tr>
<th>Output Voltage [V]</th>
<th>Hydrofoil angle [degrees]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.01</td>
<td>43</td>
</tr>
<tr>
<td>2.9</td>
<td>24</td>
</tr>
<tr>
<td>2.79</td>
<td>0</td>
</tr>
<tr>
<td>2.66</td>
<td>-22</td>
</tr>
<tr>
<td>2.55</td>
<td>-41</td>
</tr>
</tbody>
</table>

Fig. B.8: Hydrofoil calibration
was fixed by applying epoxy between the metal shaft and the outside fiberglass and wrapping and embedding carbon fiber tow into the epoxy. During the next test, the data recorded that there was still a shift in the position of the hydrofoil. This was because the potentiometer connection with the hydrofoil was slipping. This was corrected by using epoxy to glue the shaft adapter to the gear. No further slipping was observed.

Fig. 3.9 shows the hydrofoil potentiometer assembly. The slipping of the potentiometer was observed between the wood and the plastic pinion connection. To remove the slipping, epoxy was applied at this junction.
Fig. B.9: The hydrofoil potentiometer assembly is shown in this figure. Epoxy has been used between the wood and the pinion to ensure that there was no more slipping of the potentiometer.
Appendix C

Error Analysis

The uncertainties in all of our measured quantities deserves a careful analysis. Several of the quantities that are reported in this thesis are calculated from the experimental measurements. The uncertainties in these calculated quantities can be determined by propagating the error in the measurements through the calculation using standard tools. However, to begin the uncertainties in the measured quantities will be determined.

C.1 Tow Cart Position Error

The tow cart position was determined using a 10-turn wire-wound potentiometer. It was calibrated against a standard construction tape measure which was read to the nearest 1/32 inch. All measurement were taken with respect to the inside edge of the tank liner (used as the origin of the system or the zero position). The voltage reading from the potentiometer was converted using the DAQ’s 16 bit A/D converter. The cart was placed in a fixed position and position data from the DAQ was taken. The noise level in signal for a constant reading was determined. As can be seen in Fig. C.1 the data is normally distributed and has a standard deviation of 0.021 inches. Assuming that our uncertainty in this measurement due to signal noise is twice the standard deviation, this would result in an uncertainty of 0.043 inches. However, there are other sources of uncertainty which may be larger. In particular the measurements
were calibrated against a standard measuring tape whose precision was 1/32 inch. However, given tape sag, deformation, and uneven lines of sight, the uncertainty of our calibration standard is likely 1/16 inch (±0.0625 inch). To get a better idea of the uncertainty of the calibration procedure, a single distance measurement was done five times. Table C.1 shows the readings on the measuring tape. If we assume that the measurements are normally distributed, then the standard deviation ($S_d$) for these readings is 0.04 inches. Using the same assumption stated earlier, that the uncertainty of the calibration procedure is approximately ±2S. Since the calibration procedure has a larger uncertainty than the noise level in the reading, the error for the tow cart measurements is ±0.08 inches.
C.2 Load Cell Error

To check the accuracy of the scale that was used to calibrate the load cell, a single mass was measured using our scale (Cen-Tech Model 9534), a triple beam balance, a laboratory grade-Mettler Toledo Classic Plus AB204-S/FACT (Fully Automatic Calibration Technology). The readings of the mass on the three scales were: Cen-Tech scale: 145 g, Triple Beam Balance: 144.6 g, Mettler Toledo Lab Scale: 144.4999 g. Thus the uncertainty of the Cen-Tech scale that was used to measure the weights for the load cell calibration is approximately ±1 g. If the local gravitational constant is assumed to be $9.81 m/s^2$, then this results in an uncertainty of the calibration standard of ±0.00981 N.

Another source of error is the noise level of the analog force signal. Fig. C.2 shows the distribution of the load cell measurements for a constant applied load (using a weight). The standard deviation of the signal is 0.0051 N and if we assume the uncertainty is approximately twice the standard deviation, the uncertainty of the
signal is $\pm 0.0102 N$. Since the uncertainty due to signal fluctuations and noise is the larger of the two errors, the uncertainty in the load cell measurements is $\pm 0.01 N$.

### C.3 Boom Angular Position

The boom angular position was not measured directly but instead calculated using linear distance measurements and trigonometric relations. A suitable position for the boom was chosen and with the help of two distances $d_1$ and $d_2$, the angle $\theta$ was calculated. The orientation of $d_1$ and $d_2$ are shown in Fig. C.3.

For the same boom angle, five sets of measurements were done to characterize the uncertainty in the calibration standard. Another boom angle was selected and the same process is carried out again to determine if the uncertainty in the calibration procedure changed based on boom angle. Table C.2 and C.3 show the readings for the two measured angles. All lengths were measured using a large inexpensive vernier caliper with a stated precision of 1/128 inch. From the Fig. B.5, $\theta$ can be calculated by:
Table C.2: Boom angle measurement for position ‘1’ of the boom

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>5.664</td>
<td>19.609</td>
<td>16.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.623</td>
<td>19.632</td>
<td>16.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.711</td>
<td>19.64</td>
<td>16.2</td>
<td>0.077</td>
<td>0.16</td>
</tr>
<tr>
<td>5.632</td>
<td>19.64</td>
<td>16.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.664</td>
<td>19.68</td>
<td>16.05</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table C.3: Boom angle measurement for position ‘2’ of the boom

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>9.32</td>
<td>17.83</td>
<td>27.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.33</td>
<td>17.75</td>
<td>27.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.28</td>
<td>17.80</td>
<td>27.5</td>
<td>0.076</td>
<td>0.15</td>
</tr>
<tr>
<td>9.39</td>
<td>17.84</td>
<td>27.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.32</td>
<td>17.82</td>
<td>27.6</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[
\theta = \arctan\left(\frac{d1}{d2}\right) \tag{C.1}
\]

Thus the uncertainty in our calibration procedure for boom angles is ±0.16°.

Note that the errors for the calibration are similar for both angles measured. Thus it appears that the uncertainty due to the calibration procedure are roughly independent of boom angle.

However, as the boom angle is also measured via a potentiometer, the fluctuations and noise from that analog signal was also estimated to see if it’s magnitude was larger than the error in our calibration procedure. A histogram of 10,000 data points for a fixed boom angle is shown in Fig. [C.4]. The error due to noise fluctuations was found to be ±0.2°. From the results of these tests, the maximum of the two errors is used to represent \( \theta \). Thus, for this system boom angle is best described as \( \theta \pm 0.2° \).
C.4 Hydrofoil Angular Position

A similar method like the one used to calibrate the boom angular position, was adapted to calibrate the hydrofoil angular position. Fig. C.5 shows the orientation of the two sides $l_1$ and $l_2$ to find the hydrofoil angle $\beta$. Both lengths were measured using the same large caliper that was used in the boom angle calibration procedure. The equation for $\beta$ can be given by:

$$\beta = \arcsin\left(\frac{l_1}{l_2}\right)$$  \hfill (C.2)

Two hydrofoil angles were selected and $l_1$ and $l_2$ were measured for these angles. Table C.4 shows five repeated measurements of the hydrofoil angle using the calibration procedure. Table C.4 and C.5 show the measurements at two angles.

As was stated in the calibration procedure for the hydrofoil angle, a straightedge was attached to the hydrofoil to facilitate measurements. Due to deviations in this
Fig. C.5: Method used to measure the error while determining the hydrofoil angle

Table C.4: Hydrofoil angle measurement for position ’1’ of the hydrofoil

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>8.22</td>
<td>13</td>
<td>39.21</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8.25</td>
<td>13</td>
<td>39.39</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8.20</td>
<td>13</td>
<td>39.12</td>
<td></td>
<td>0.13</td>
</tr>
<tr>
<td>8.26</td>
<td>13</td>
<td>39.43</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8.20</td>
<td>13</td>
<td>39.12</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table C.5: Hydrofoil angle measurement for position ’2’ of the hydrofoil

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>5.70</td>
<td>13</td>
<td>11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.73</td>
<td>13</td>
<td>11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.68</td>
<td>13</td>
<td>11</td>
<td></td>
<td>0.07</td>
</tr>
<tr>
<td>5.71</td>
<td>13</td>
<td>11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.69</td>
<td>13</td>
<td>11</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
attachment point, it is assumed that the error associated with $l_2$ is larger than the precision of the caliper and is approximately $\pm \frac{1}{16}$ inch. The results showed that the uncertainty due to the calibration is $\pm 0.3^\circ$.

The noise level in the hydrofoil sensor can be determined from the histogram of a constant position reading (shown in Fig. C.6). The standard deviation, assuming a normal distribution, $0.1^\circ$, which would imply that the error due to signal noise is $0.2^\circ$.

Thus, errors in our calibration procedure are slightly larger than the signal noise and the hydrofoil uncertainty is $\pm 0.3^\circ$.

### C.5 Propagation of Error for Torque Calculation

There are a number of errors associated with the measurement of torque which is used to calculate the instantaneous power and also the average cycle power. These errors are associated with the uncertainty in the load cell measurements, boom angle
measurements, and the geometry of the friction brake design.

The torque applied on the brake from the slipping boom shaft is given by:

\[ \tau = lF \sin \phi \quad (C.3) \]

Fig. C.7 shows an exaggerated picture of what the load cell would look like if it was not perpendicular to the arm of length \( l \).

To determine the uncertainty in the calculated torque value we assume that the error in the three measurements are independent of each other. In this case, that seems like a fairly reasonable assumption since the load measurement and the length measurements are calibrated using different standards. On differentiating the above equation with respect to \( l \), \( F \) and \( \phi \) we get the following equations.

\[ \frac{\partial \tau}{\partial l} = F \sin \phi \quad (C.4) \]
\[ \frac{\partial \tau}{\partial F} = l \sin \phi \] (C.5)

\[ \frac{\partial \tau}{\partial \phi} = l F \cos \phi \] (C.6)

Assuming the errors are independent, the error can be estimated to be:

\[ \Delta \tau = \sqrt{\left( \frac{\partial \tau}{\partial l} \Delta l \right)^2 + \left( \frac{\partial \tau}{\partial F} \Delta F \right)^2 + \left( \frac{\partial \tau}{\partial \phi} \Delta \phi \right)^2} \] (C.7)

Dividing throughout by \( \tau \) to get the fractional error gives:

\[ \frac{\Delta \tau}{\tau} = \sqrt{\frac{(F \sin \phi \Delta l)^2}{F^2 l^2 \sin^2 \phi} + \frac{(l \sin \phi \Delta F)^2}{F^2 l^2 \sin^2 \phi} + \frac{(l F \cos \phi \Delta \phi)^2}{F^2 l^2 \sin^2 \phi}} \] (C.8)

\[ \frac{\Delta \tau}{\tau} = \sqrt{\left( \frac{\Delta l}{l} \right)^2 + \left( \frac{\Delta F}{F} \right)^2 + \frac{\cos^2 \phi}{\sin^2 \phi} \Delta \phi^2} \] (C.9)

The uncertainty in the force measurement was discussed above and was found to be \( \pm 0.01 \text{N} \). The length measurement was done using a set of standard machinist calipers and was found to be 2.775 inches. The precision of the calipers was 0.001 inches. However, since the measurement was not easy to obtain (center of two cylinders) 10 independent measurements were made. If a normal distribution is assumed for the measurements, then the standard deviation was found to 0.049 inches. Thus the uncertainty in the length measurement was 0.1 inches. Converting these distance measurements to metric gives a length of 7.048 \( \pm 0.254 \text{cm} \). Table C.6 shows the length recorded for these 10 measurements.

The angle between the centerline of the brake and the load cell is not easy to determine. The machined parts were manufactured to minimize the error in this alignment but our best guess is that the uncertainty in this angle is approximately \( \pm 1^\circ \).
Table C.6: Measurement of torque arm from pivot point. Table shows 10 readings to find the standard deviation of the distribution.

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Most of our load readings were on the order of 1 to 10 N. If we assume a load of approximately 5 N we can estimate the relative error in the torque calculation.

$$\frac{\Delta \tau}{\tau} = \sqrt{ \left(0.254 \frac{-7}{0.048}\right)^2 + \left(0.01 \frac{-5}{2}\right)^2 + \left(\frac{\cos^2(\pi/2)}{\sin^2(\pi/2)}\right)\left(\frac{\pi}{180}\right)^2}$$  \hspace{1cm} (C.10)

Note that due to the geometry of the system, the error due to misalignment of the load cell has, to first order, no effect on the relative error of the torque calculation.

Therefore the relative $\frac{\Delta \tau}{\tau}$ is approximately 0.04 or 4% of the calculated torque value. For the give example with the assumed load of 5 N on the load cell the torque would be $0.352 \pm 0.014 \text{ Nm}$.

C.6 Propagation of error for Average Cycle Power calculation

The ultimate goal is to characterize the error in the average cycle power calculations. Note that there is no simple way to do this using the uncertainties in the four base measurements. The average cycle power calculation requires that the system reach steady state which depends on the feedback control system and it’s own dynamics. It
also depends on the repeatability of the hydrofoil actuator and tow-cart drive system. Probably the best way to characterize the random error in the average cycle power calculations which will include all of these issues, is to repeat the same test several times to experimentally determine the overall variation in average cycle power. This was done for several tests and appears in the thesis. This does not take into account unseen systematic errors in the control system (for example) but the total uncertainty in the four base experimental measurements was determined and appears above.