DESIGN AND TESTING OF A PNEUMATICALLY PROPELLED UNDERWATER GLIDER FOR SHALLOW WATER

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Summary

This report details the design and testing of a pneumatically propelled underwater glider. The vehicle was designed as a platform for motion control experimentation, and to explore the use of novel actuator designs to improve performance in shallow water and significant currents. The glider’s pneumatic buoyancy engine is capable of rapidly inflating an elastomeric bladder to 5 liters. (This displacement is an order of magnitude greater than that of legacy buoyancy engine designs.) The buoyancy engine was shown to operate reliably at 25 m depth. However, the compressibility of the bladder and associated change in tank weight (from exhausting air with each dive) presented significant challenges in trimming the vehicle. The attitude of the glider is controlled by translating and rotating a semi-annular mass. Because of the geometry of this mechanism, the glider is not restricted to a range of roll attitudes (i.e. the glider has unlimited roll authority and can “flip over”). By flipping over the glider may employ asymmetric hydrodynamic surfaces while preserving the same flow-relative geometry during both descents and ascents. Such asymmetric hydrodynamic surfaces (e.g. cambered hydrofoils, dihedral, wing twist) may be used to improve efficiency and performance. The ability to operate in both upright and inverted orientations requires reducing the contribution of the rigid body (minus the moving mass) to the bottom heaviness of the vehicle. A moving acoustic long-baseline ranging system was developed to position the glider while it was underway. The performance of this system was characterized experimentally in terms of ping success rate for various transducer geometries and depths in a shallow-water, rocky bottom lake.

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Chapter 1

Introduction

Sunlight penetrates the shallow waters of the coastal ocean creating a euphotic zone that hosts many plants, algae, reefs and other organisms of interest to marine biologists and of commercial value to industry. The littoral zone is also of strategic importance to the Navy. The task of sampling the coastal ocean is increasingly being carried out by autonomous underwater vehicles (AUVs). However, the presence of significant currents, that are often dynamic and spatially varying, can adversely affect the mobility and efficiency of AUVs operating in this environment. Moreover, the increased rate of biofouling in the euphotic zone can cripple long endurance robotic platforms.

Underwater gliders are capable of traversing hundreds of kilometers over deployment times of weeks, or months, because their motion largely consists of steady, stable, low power glides. Energy intensive buoyancy changes occur infrequently. The lack of external moving parts makes gliders quiet (and stealthy). This also makes them more robust to corrosion or biofouling than their propeller driven counterparts with external moving control surfaces. From robustness and endurance considerations, underwater gliders are ideal candidates for shallow water operations.

Traditional “legacy” gliders, such as Spray [1], Slocum [2] and Seaglider [3], were designed to operate in the deep waters of the open ocean. However, the unique challenges associate with shallow waters motivates developing new glider designs with improved capabilities. The performance of a glider is determined in part by its inherent design (i.e. actuator capabilities, hydrodynamics, and energy storage) and in part by the way it is operated (i.e. guidance, navigation, and control). The goal of this work was to develop a underwater glider platform for testing novel motion platting strategies (such as those discussed in [4, 5, 6, 7]) and to explore new actuator designs that can improve performance in significant currents.

A few gliders have been developed specifically for the coastal ocean. These include the shallow water Slocum Battery [8] and more recently the Coastal Glider [9]. The shallow water (200 m) Slocum Battery was designed with a single-stroke piston style buoyancy engine. It employs a rudder to yaw (rather than rolling with a laterally moving mass). This helps achieve a tight turning radius (of about 7 m) and improves maneuverability [10]. However the Slocum Battery only displaces 0.5 L of water with it’s buoyancy engine, resulting in a
top speed of 25 cm/s. Shallow water currents often exceed this speed and can potentially dominate the dynamics of a slow moving glider. The Coastal Glider has been in development for the past decade and is now being commercialized. It features a hydraulic pump based buoyancy engine that drives a piston to create large volume changes of 5 liters. It can reach speeds in excess of 1 m/s making it a good candidate for shallow water operations.

In addition to these gliders, the XRay \[11\] and ZRay\[12\] have been developed for long distance and long duration flight. These Liberdade class gliders differ significantly from earlier legacy designs. They are large blended wing body gliders (6m and 20m wingspan respectively). Their design was inspired by the comprehensive study \[13\] that recognized the improvements to be gained in net transport economy by increasing the scale, buoyant lung capacity, and improving the hydrodynamic design of legacy gliders.

This report details the design and testing of a pneumatically propelled underwater glider, the Virginia Tech Underwater Glider \[14, 15\]. Key features of the design are illustrated in Figure 1.1. The glider’s pneumatic buoyancy is capable of rapidly inflating an elastomeric bladder to 5 liters. The attitude of the glider is controlled by translating and rotating a semi-annular mass. A cylindrical moving mass mechanism allows the glider to “flip over” and employ asymmetric hydrodynamic surfaces. The glider was designed with open-source software. The glider during testing is shown in Figure 1.2 and key specifications are given in Table 1.1.
Figure 1.2: The Virginia Tech Underwater Glider prior to deployment in Claytor Lake, VA

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body Length</td>
<td>6.3 ft</td>
</tr>
<tr>
<td>Full Length (w/Antenna)</td>
<td>9.3 ft</td>
</tr>
<tr>
<td>Diameter</td>
<td>9.0 in</td>
</tr>
<tr>
<td>Mass</td>
<td>56 kg</td>
</tr>
<tr>
<td>Max Volume Change</td>
<td>5 Liters</td>
</tr>
<tr>
<td>Endurance</td>
<td>6 hours</td>
</tr>
<tr>
<td>Dives to 100m</td>
<td>11 (5L) or 21 (3L)</td>
</tr>
<tr>
<td>Dives to 50m</td>
<td>19 (5L) or 35 (3L)</td>
</tr>
<tr>
<td>Roll Control Authority</td>
<td>unlimited</td>
</tr>
</tbody>
</table>

Table 1.1: Specifications of the Virginia Tech Underwater Glider
Chapter 2

Buoyancy Control

Underwater gliders control their depth rate by controlling their net weight. This can be achieved by either adjusting the weight of the vehicle (e.g. releasing drop weights, or flooding ballast tanks) or by adjusting the buoyancy (e.g. inflating a bladder, or displacing water in a flooded chamber using a piston). The mechanical device that adjusts buoyancy is sometimes called the “buoyancy engine”, the “variable buoyancy device”, or the “buoyancy control system”.

The total buoyancy $B$ of a glider is the sum of the fixed buoyancy of the glider’s rigid body $B_{RB}$ (for shallow waters we may neglect hull compressibility, and assume this quantity is fixed) and the variable buoyancy generated by the buoyancy engine $B_{BE}$. For a glider with “dry weight” $W = mg$ (where $m$ is the glider’s mass and $g$ is gravitational acceleration), the net force on the glider is referred to as the net weight $\tilde{W} = W - B = W - B_{RB} - B_{BE}$. For the glider to descend $B_{RB} < W$ is required, and there is a a value of $B_{BE}$ that gives neutral buoyancy ($\tilde{W} = 0$). The buoyancy engine is typically designed to produce equal forces in both ascents and descents.

2.1 Shallow Water Buoyancy Engine Requirements

Legacy gliders operate at horizontal speed of about 0.6 kts. At such low speeds, typical currents in the coastal ocean can dominate the glider’s dynamics. This gap in speed motivates the design of faster gliders with larger buoyant lung capacities. Gliders are most efficient while in a steady equilibrium glide. Thus it is desirable to minimize the time it takes to transition from a dive to an ascent, and vice versa. This is particularly important in shallow waters where there may not be sufficient depth to establish a steady glide. We refer to the time that it takes the buoyancy engine to change between minimum and maximum net weight as the buoyancy transfer time. (Note that the speed of the moving mass actuator will also determine the time it takes to achieve a steady glide.)

In the design of the Virginia Tech Underwater Glider (VTUG) the endurance was required to be long enough for a full day of field testing (about a 6 hour deployment). The max-
Table 2.1: Buoyancy engine design requirements

- **Buoyant Lung Capacity**: $\pm 5\%$ (± 2,500 cc assuming a 50 kg glider)
- **Buoyancy Transfer Time**: $\leq 10$ sec. (167 cc/s)
- **Max. Operational Depth**: 100 m
- **Endurance**: 6 hours
- **Weight**: $\leq 30$ lbs ($\approx 25\%$ of vehicle dry weight)
- **Size**: $\leq 20$ in. length ($\approx 25\%$ of vehicle length)

The maximum operating depth was limited to 100 m. The buoyancy engine was required to have a large buoyant lung capacity (an order of magnitude larger than previous gliders) and a small buoyancy transfer time (less than 10 seconds). These requirements are summarized in Table 2.1.

### 2.2 Selecting the Buoyancy Control Mechanism

There have been many mechanisms proposed for the purpose of buoyancy control. Traditional buoyancy control mechanisms include piston-cylinder designs [8], hydraulic pump-powered bladders [1, 3] and the use of ballast tanks (as employed by manned submarines). Thermally driven phase-change buoyancy engines that harvest energy have been proven [2]. Chemical reactions with surrounding seawater to produce gases for buoyancy control have been proposed [16, 17]. Using shape memory alloys to change the hull displacement for buoyancy control has also been investigated [18]. A detailed discussion of the working principle behind each of these technologies is beyond the scope of this work. In the following, we briefly discuss the main reasons for choosing a pneumatic buoyancy engine.

A piston-cylinder buoyancy engine designed with a 8 inch diameter face (about the diameter of a typical glider hull) would require over 8,000 lbs of force to actuate at 100 m depth. Pretensioning the piston or reducing the piston face area (at the cost of a larger stroke) would reduce this force requirement. However, the power requirements to achieve the desired transfer time would remain prohibitive. Commercially available linear actuators (electric motor/lead-screw or hydraulic models) designed for industrial applications were determined to be too large and heavy to be viable options.

Oil-bladder and pump designs used by legacy gliders are efficient at high pressures and large depths. To meet the buoyancy engine requirements, a fast and large displacement pump would be required. A common challenge associated with pump based buoyancy engine designs is the presence of air bubbles in the working fluid. Such air bubbles may be difficult to remove and under some conditions may result in a reduction or loss of pumping power. A preliminary study of this concept concluded that commercially available pumps would not meet the required specifications with a reasonable weight and volume.
A pneumatic buoyancy engine was ultimately chosen for the VTUG. Energy stored in the form of compressed air, housed in a diving cylinder (scuba tank), can be used directly to inflate an elastomeric bladder for buoyancy. This approach does not have the efficiency losses associated with converting electrical to mechanical energy, as in battery powered designs. The pressure differential between the ambient water at 100 m and a scuba tank is over 3,400 psi. (In theory, the tank’s pressure differential allows inflating the bladder at depths up to 2 km.) A rapid inflation of a large volume can be achieved by simply opening a valve. In this design, the buoyancy engine is an open system and air must be exhausted to the atmosphere to reduce the buoyancy of the bladder once it is inflated. The endurance of the vehicle is limited by the capacity of the tank.

2.3 Design of a Pneumatic Buoyancy Engine

2.3.1 Conceptual Design

Scuba tanks with compressed air are an ideal choice for a pneumatic buoyancy engine because they are designed to operate in water, are safe, and can be readily re-filled at local dive shops. For the VTUG, a “pony” sized tank was found to have sufficient air capacity for the mission endurance required. (A pony tank is a smaller scuba tank typically used by divers as an extra air supply in emergency situations.) A steel tank was selected because of its higher air capacity to weight ratio relative to aluminum tanks. The specifications of the tank are given in Table 2.2.

To assess the performance of the pneumatic buoyancy engine concept, we assume that the air behaves according to the ideal gas law:

\[ pV = nRT \]  \hspace{1cm} (2.1)
where

\[ p = \text{pressure (Pa)} \]
\[ V = \text{volume (m}^3\text{)} \]
\[ n = \text{amount of gas (moles)} \]
\[ R = \text{universal gas constant } \left( 8.314 \; \text{J \cdot mole} / \text{K} \right) \]
\[ T = \text{temperature (K)} \]

The initial amount of gas \( n_{\text{tank}} \) inside the tank at room temperature can be computed from (2.1) using the parameters in Table 2.2. We find that \( n_{\text{tank}} \approx 31.1 \) moles in this case. For a given buoyant lung setting, the volume of water displaced by the buoyancy engine is

\[ V_\eta = \eta W / \rho_{\text{H2O}} g \]

where \( \rho_{\text{H2O}} \) is the density of water. At a given depth \( z \) (and corresponding ambient pressure \( p_{\text{amb}} \)) the ideal gas law (2.1) can be used to compute the amount of air \( n_{\text{cycle}} \) required to displace the volume \( V_\eta \) under ideal conditions. (This neglects the additional pressure required to stretch the bladder and other losses.) The ideal number of available dives for a given scuba tank is then \( N = n_{\text{tank}} / n_{\text{cycle}} \). Contours of fixed \( N \) are shown in Figure 2.1, for various bladder displacements and depths, given the scuba tank parameters in Table 2.2. At the most extreme design point (5 L displacement, 100 m depth) we expect an endurance of 13 dives, while at shallower depths many more dives can be expected. (Replacing the scuba tank in the field can further extend the endurance.)

![Figure 2.1: Number of dives](image)

In the previous discussion, it was assumed the bladder was inflated to the same volume at the same depth with each dive. However this will generally result in different ascent/descent profiles with each dive, due to a combination of two factors: a) the vehicle becomes lighter as the tank is depleted, and b) as the ambient pressure changes with depth, the bladder will
expand/compress unless the pressure is actively regulated (with feedback control). In the following we discuss these two effects.

The weight of air in the tank at full pressure (3,442 psi) is about 2 lbs. Assume that the vehicle is trimmed to be neutrally buoyant with a tank full and a bladder displacement of 2.5 L. Then if every ascent corresponds to a bladder displacement of 5 L (fully inflated) and every descent to a displacement of 0 L (fully deflated), the change in (maximum and minimum) net weight as the tank becomes depleted is given by Figure 2.2. We find that as the tank becomes lighter, the range of attainable net weights shifts towards higher buoyancy and the trim point will change significantly during the course of a single deployment.

The second effect results from the changes in ambient pressure around the elastomeric bladder. Let $p_{blad}$ accounts for the pressure resulting from the elastic forces in a stretched bladder. The bladder reaches an equilibrium volume when the ambient pressure $p_{amb}$ and elastic bladder pressure $p_{blad}$ is balanced by the internal bladder pressure $p_{int}$:

$$p_{amb} + p_{blad} - p_{int} = 0.$$  

(2.2)

The ambient pressure is given by the hydrostatic equation

$$p_{amb} = p_{atm} + \rho_{H2O}gz ,$$  

(2.3)

where $p_{atm}$ is the atmospheric pressure. Assume that the elastic pressure $p_{blad} = p_{blad}(V)$ depends only on the volume of the stretched bladder. The internal bladder pressure is given by the ideal gas law

$$p_{int} = \frac{nRT}{V}.$$  

(2.4)
Then the equilibrium condition (2.2) is rewritten as

\[ p_{\text{atm}} + \rho_{\text{H}_2\text{O}}gz + p_{\text{blad}}(V) - \frac{nRT}{V} = 0. \]  

(2.5)

In practice, we may measure \( p_{\text{blad}} \) directly using a differential pressure sensor that compares the ambient water pressure to the internal bladder pressure. Figure 2.3 shows an experimental setup and resulting pressure-buoyancy curve for the bladder used on the VTUG. The experiment was conducted by submerging the buoyancy engine in a container and measuring the weight of water displaced by the bladder at various differential pressures \( p_{\text{blad}} \). (As the bladder was inflated, the displaced water drained through a hole in the container wall directly onto a scale.) The \( p_{\text{blad}}(V) \) curve-fit to the data in Figure 2.3 is a fourth order polynomial constrained to intersect the origin. (Note that the curve in Figure 2.3 actually gives buoyancy with pressure, however it is trivial to obtain the corresponding volume.)

![Experimental setup and resulting pressure-buoyancy curve](image)

(a) Experimental setup  (b) Fourth order fit to data (constrained to intersect the origin)

Figure 2.3: Experimentally determined bladder pressure vs. buoyancy curve

Given the relationship \( p_{\text{blad}}(V) \) we may use (2.5) to obtain an expression \( n(z, V) \) that gives the amount of air required to inflate the bladder to a given volume, and a given depth. In Figure 2.4 contours of constant \( n \) are plotted in the \( z-V \) plane. These contours illustrate the open-loop response of the bladder, with a fixed amount of air, subject to changes in depth. If we follow the contours starting along the maximum depth \( z = 100 \) m we find that for most bladder displacements \( (V > 0.8 \) L) the bladder volume expands to the 5 L and saturates prior to reaching the surface.

The opposite effect occurs during descents. As the glider descends, the bladder is compressed and for displacements \( V < 2 \) L (at the surface) the volume quickly approaches zero. From this analysis it is clear that to maintain a fixed, non-zero, bladder displacement, the bladder \( p_{\text{blad}} \) must be actively regulated by exhausting air during ascents and inflating air during descents.
Suppose that the bladder displacement is regulated to a fixed volume using two valves which we refer to as the inflator (to inject air into the bladder) and deflator (to exhaust air to the atmosphere). Holding $V$ fixed, and differentiating (2.5)

$$\rho_{H2O}g \dot{z} = \dot{n}RT / V.$$  \hspace{1cm} (2.6)

From (2.6) we may determine the air exchange rate $\dot{n}$ required for a given depth rate $\dot{z}$. The range of expected depth rates will depend on the buoyant lung setting and drag of the glider. A conservative estimate for the maximum depth rate can be obtained by equating the net weight to the drag force. This gives the terminal velocity

$$\dot{z}_{\text{max}} = -\text{sgn}(\tilde{W}) \sqrt{2|\tilde{W}| / \rho_{H2O}SC_D},$$  \hspace{1cm} (2.7)

where $S$ is the surface area and $C_D$ is the drag coefficient. Depending on the orientation of the glider the drag coefficient may vary. (We assume the worst case orientation where the glider is pointed downwards and is most streamlined. In this case, the surface area can be taken as the hull cross-sectional area.) Equations (2.7) and (2.6) give a conservative estimate of the required amount of air to be exhausted/inflated by the valves to maintain a fixed bladder volume. This estimate is shown in Figure 2.5 for several drag coefficients, where each curve is generated by computing $\dot{z}_{\text{max}}$ and $\dot{n}$, from (2.7) and (2.6), for varying $\tilde{W}$. It is interesting to note that there is a maximum air exchange rate during descents. At $\tilde{W} = 0$ the bladder displacement is 2.5 L corresponding to neutral buoyancy and thus $\dot{z}_{\text{max}} = 0$ and $\dot{n} = 0$. As $\tilde{W}$ decreases, the bladder becomes completely deflated and there is no need to exchange air, and again $\dot{n} = 0$. In-between these two conditions the maximum required inflation rate occurs. On the other hand, we find that the air exchange rate magnitude is
monotonically increasing with $\dot{W}$ during ascents. The most demanding air exchange rate occurs when the drag coefficient is low during highest speed ascents.

Figure 2.5: Required air exchange rate to maintain a fixed bladder volume at the corresponding depth rate for various drag coefficients

Having established the required air exchange rates it remains to select actuators (valves and tubing) to meet these requirements. Generally, vendors of flow control valves provide flow coefficients ($C_v$) or volumetric flow rate curves for various differential pressures. The flow coefficient for gas flow [19] is given by

$$q = NC_v p_1 \left( 1 - \frac{2\Delta p}{3p_1} \right) \sqrt{\frac{\Delta p}{p_1 T_1}} ,$$  \hspace{1cm} (2.8)

where

- $q =$ volumetric flow rate (std. ft$^3$/min)
- $p_1 =$ inlet pressure (psia)
- $p_2 =$ outlet pressure (psia)
- $\Delta p = p_1 - p_2,$ differential pressure (psid)
- $T_1 =$ absolute temperature (R)
- $N = 22.67$ (constant depending on unit choice).

In Figure 2.6(a), the deflator flow coefficient $C_v$ is calculated from (2.8) for a range of bladder displacements and depths. For a given bladder displacement and depth, the corresponding maximum depth rate and air exchange rate are calculated using (2.7) and (2.6), assuming $C_D = 0.1$ (the worst case). Recall that $m_{air} = n M_{air}$ relates an amount of air $n$ to the mass.
of the air $m_{\text{air}}$ where $M_{\text{air}}$ is the molecular weight of air. Then the volumetric flow rate is

$$q = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{\dot{n}M_{\text{air}}}{\rho_{\text{air}}}.$$  

For sizing the deflator, $\Delta p = p_{\text{blad}}(V)$, and the hydrostatic equation (2.3) can be used to compute $p_2 = p_{\text{amb}}$. Then $p_1 = p_{\text{amb}} + \Delta p$. In Figure 2.6(a), we find that the deflator requirements (to maintain a fixed bladder volume) increase with depth and bladder displacement.

A similar analysis can be used to size the inflator. The flow into the bladder is driven by the large pressure differential between the scuba tank and the internal bladder pressure. Thus the effects of changes in depth can be ignored. In Figure 2.6(b) we find that the $C_v$ is significantly lower than that of the deflator. Further, there is a bladder displacement that maximizes the required $C_v$, and this is related to the maximum air exchange rate $\dot{n}$ seen in Figure 2.5.

![Figure 2.6](image)

(a) Deflator valve requirements  
(b) Inflator valve requirements

Figure 2.6: Sizing the pneumatic buoyancy engine control valves for a fixed bladder volume

Note that these previous analyses assumed the bladder is already inflated to a desired volume. To determine the flow coefficient require for a given buoyancy transfer time one must study the dynamic process of inflating or deflating the bladder. We do not characterize this process here but we note that it is largely driven by the pressure differential. In the case of inflating the bladder, the flow rate can be assumed fixed (since the change in tank pressure is small during a single inflation). Whereas during deflation, the change in pressure (and thus flow rate) is given by $p_{\text{blad}}(V)$ and cannot be assume fixed. The above analysis was used to select the deflator (model: iQValves Standard PFCV-203321) and inflator (model: Enfield Technologies LS-V15) valves.
2.3.2 Detailed Design

A schematic of the major components in the pneumatic buoyancy engine is given in Figure 2.7. A standard 1st stage dive regulator (model: Zeagle DS-V) is fitted to the scuba tank. This regulator contains a diaphragm that outputs both a high pressure and low pressure. The high pressure (HP) air port is used for sensing the pressure remaining in the tank, and a low pressure (LP) air port is used to inflate the bladder. The regulator is designed to deliver the LP air at a constant 160 psid above ambient. Early work studying the feasibility of the pneumatic buoyancy engine concept involved a test rig with on/off solenoid flow control valves (see Section 10.1). In the design of the glider’s buoyancy engine proportional solenoid valves were used instead. (Note: some valves do not seal completely when they are “closed.” Such valves have a vendor specified “leakage rate”. The inflator valve’s leakage was suppressed by installing an additional solenoid in-line to open the flow to the inflator, prior to an inflation.)

The pressure sensor inlet and bladder exhaust were fitted with 25 μm filters to prevent debris from entering the system. To prevent water from entering the bladder where the air is exhausted a check valve was used. The “cracking pressure” of a check valve is the minimum pressure differential required to open the valve. Since the bladder pressure is relatively low, a very low cracking pressure (0.02 psi) medical grade check valve was chosen. However, this valve was relatively small and restricted the flow. Thus a series of parallel check valves were installed across a manifold. All of the pneumatic tubing consisted of perfluoroalkoxy (PFA) teflon tubing and Swagelok fittings. The Swagelok fittings provide an easy method for installing, maintaining, and altering the pneumatic system through the use of two-ferrule tube fittings and a swaging technique. The latex bladder was originally fabricated for use in a blood pressure measurement device (sized to measure blood pressure around an adult thigh, model: W. A. Baum Co. Inc. #1844). This bladder was modified by installing a large diameter fitting (Figure 4.2(e)) to allow larger tubing to the deflator (Figure 4.3(d)). To increase the pressure in the bladder, and improve deflation rates, a covering for the bladder was constructed from olefin fabric. This restricted the volume of the bladder while allowing
higher pressures.

An external view of the pneumatic buoyancy control system fabricated for the glider is shown in Figure 2.8. The bladder pressure was regulated using a proportional-integral-derivative (PID) controller.

![Images of the pneumatic buoyancy control system](image1.jpg)

(a) Modified bladder with deflator fitting  
(b) Large diameter deflator tubing  
(c) Internal pneumatic control and sensing  
(d) Flooded chamber removed to expose bladder with olefin fabric jacket, exhaust manifold, and tank

Figure 2.8: Interior and exterior of the buoyancy control system

### 2.3.3 Longitudinal Placement of the Bladder

The buoyancy engine of the glider can generate large pitching moments and strongly influence the attitude of the glider. Since the glider is generally in a nose up attitude during ascents, and nose down attitude during descents, it may seem appropriate to place the bladder towards the nose in front of the vehicle’s center of gravity. In such a configuration, increases and decreases in the buoyancy of the bladder relative to the neutral buoyancy state result in a pitching moment in a favorable direction. However, this makes achieving a communications stance on the surface more difficult. (See Section 10.5.1 for a detailed discussion.) To mitigate
this issue some gliders have a secondary bladder used solely to assist in elevating the tail on
the surface. The placement of the bladder depends heavily on the design of the pitch control
mechanism and requires a detailed analysis of the available pitch attitudes under a range of
operating conditions.

For the VTUG, we found that placing the bladder in the nose was not feasible as the pitching
moments in this configuration were too large for the moving mass to have sufficient pitch
authority. A compromise was achieved by placing the bladder forward of the CG but “inside”
the hull to reduce the bladder moment arm. This required a section of the vehicle to be
flooded.

2.3.4 Bladder Pressure, Pitch Attitude, and Depth Rate Coupling

One challenge associated with placing the bladder forward of the CG is that this causes a
coupling between bladder pressure, pitch, and depth rate. The result is that any open-loop
equilibrium gliding state is unstable.

Consider a vehicle in a steady equilibrium descent and suppose that at a given instant the
bladder feedback control loop is turned off. The bladder pressure will decrease as the glider
descends and this will cause a nose-down pitching moment. A steeper pitch attitude will
further increase the bladder’s depth and reduce its volume. The vehicle will become less
buoyant and, together with a steeper flight path angle, the depth rate will increase.

This coupling can also occur on the surface when the glider is in a communications stance. A
perturbation in pitch attitude (e.g. from waves or wind) may cause the bladder (located in
the nose) to plunge deeper below the surface. Even with a small depth change the change in
ambient pressure (and therefore bladder pressure) may be significant (e.g. a change in depth
of 0.5 m gives $\approx 0.7$ psi change in ambient pressure). As discussed in Section 10.5.1 the
VTUG executes its communications stance with just a marginally positive buoyancy. Thus,
even a small bladder pressure change may be sufficient for the vehicle to become negatively
buoyant. In both of these cases, a well designed bladder feedback controller is needed to
mitigate these effects.
Chapter 3

Attitude Control

The attitude of the glider in pitch and roll is controlled using an internal translating and rotating (cylindrical) mass that adjusts the vehicle’s center of gravity. A single moving mass is used for both pitch and roll control.

3.1 Pitch Attitude Requirements

Typically the pitch actuator must be design to generate sufficient pitching moments to meet a desired:

1. range of pitch attitudes during descents,
2. range of pitch attitudes during ascents,
3. pitch attitude during surface communication.

A free body diagram of the forces acting on a glider in a steady, equilibrium glide is shown in Figure 3.1. (Note that we omit the hydrodynamic lift and drag forces here for simplicity.) It is assumed that all of the forces act along the vehicle centerline, with the exception of the moving mass weight $W_{MM}$ (which we assume acts a distance $z_{MM}$ below the centerline). The attainable pitch angles will change with the buoyancy generated by the buoyancy engine $B_{BE}$, and the weight of the tank $W_T$. The remaining weight of the rigid body (excluding $W_{MM}$ and $W_T$) is denoted $W_{RB}$. Similarly, the buoyancy of the rigid body (excluding $B_{BE}$) is denoted $B_{RB}$. The longitudinal location through which each force acts is measured from the back of the vehicle (indicated by $l$ in Figure 3.1). Then the conditions for equilibrium are:

\[ \sum F_z : W_{MM} + W_T + W_{RB} - B_{RB} - B_{BE} = 0 \]  \hspace{1cm} (3.1)

\[ \sum M : (l_{MM} + z_{MM} \tan \theta)W_{MM} + l_TW_T + l_RB W_{RB} - l_RB B_{RB} - l_BE B_{BE} = 0 \]  \hspace{1cm} (3.2)

where $\theta$ is the pitch angle.
If the moving mass can travel from \( l_{MM} \in [l_{MM,\text{min}}, l_{MM,\text{max}}] \), then using (3.1) and (3.2) the range of possible pitch attitudes can be computed for a given bladder buoyancy and tank weight. The center of gravity (CG) and center of buoyancy (CB) positions in the vertical plane are given by:

\[
CG_x = \frac{l_{MM}W_{MM} + l_TW_T + l_{RB}W_{RB}}{W_{MM} + W_T + W_{RB}} \quad (3.3)
\]
\[
CG_z = \frac{z_{MM}W_{MM}}{W_{MM} + W_T + W_{RB}} \quad (3.4)
\]
\[
CB_x = \frac{l_{RB}B_{RB} + l_{BE}B_{BE}}{B_{BE} + B_{RB}} \quad (3.5)
\]
\[
CB_z = 0 . \quad (3.6)
\]

By adjusting the moving mass position, tank weight, and bladder buoyancy parameters, one may determine if the vehicle is capable of meeting the desired range pitch attitudes. Inevitably the mass properties of the manufactured vehicle will deviate from those assumed during the design. Uncertainty analysis must be incorporated into the design process, and trim adjustments must be made by installing ballast (e.g. lead) or buoyancy (e.g. syntactic foam) onto the vehicle as needed. Note that when the vehicle is on the surface, the problem of determining the CB is much more complex because one must account for the discontinuity in the fluid density. (The buoyancy of the portions of the glider exposed to the air is negligible.) On the VTUG, the tank was placed close to CB to reduce the effect on the pitch trim of the vehicle.

### 3.2 Actuator Sizing

To size the actuators, we compute the force required to move the mass by summing the frictional forces and weight along the axis of the glider as shown in Figure 3.6. A lead-screw type pitch actuator was chosen to move the mass. A “triple-start” lead-screw (with three independent threads) was used to maximize the travel per turn of the screw. (Each turn resulted in \( d_{LS} = 1/4 \) in. of travel). The required lead-screw torque \( \tau_{LS} \) was computed...
by assuming the rotational work from one turn of the lead-screw ($\tau_{LS}2\pi$) is equal to the translational work of moving the mass: $(F_t + W_{RM}\sin\theta)d_{LS}$. Thus

$$\tau_{LS} = \frac{(F_t + W_{RM}\sin\theta)d_{LS}}{2\pi}. \quad (3.7)$$

Figure 3.2: Determining the translational motor torque required $\tau_{LS}$ to rotate the lead-screw mechanism

A motor and gear-head were selected to meet the required lead-screw torque with a sufficient factor of safety. Since some of the moving mass is symmetric about the body axis, only a fraction of the moving mass weight is used to roll the vehicle. Figure 3.3 shows the forces and moments acting on the rotating mass where $F_t$ is the bearing friction, $W_{RM}$ is the weight of the rotating portion of the moving mass, and $\tau_G$ is the force required to rotate the mass. The moving rotational moving mass weight $W_{RM}$ should be sized to overcome the buoyancy of the wings at the surface (see Section 10.5.2 for a detailed discussion).

Figure 3.3: Determining the rotational motor torque required $\tau_G$ to rotate the gear mechanism

### 3.3 Detailed Design

The moving mass weighs 11.2 lbs, of which the rotational portion weighs about 8.1 lbs. Using polymer plain bearings, the linear portion of the mass slides along the large diameter shaft (that also houses the scuba tank), and rotates as shown in Figure 3.4 and 3.5. The shaft along which the mass moved allows a linear travel range of 15 inches with a maximum speed of 0.42 in/s. The angular range is unrestricted and has a maximum angular rate of 70 deg/s.
The linear portion is driven by a triple-start lead-screw that is supported by a ball bearing on the forward endcap, a lead screw nut fixed to the moving mass, and an Oldham coupler attached to a gearmotor. Limit switches are installed on each end of the moving mass’s linear travel length. These are used during a startup homing routine to calibrate the encoder’s position. When the rotational portion of the mass is actuated, a reaction torque is applied to the lead-screw. To reduce this torque, a stiff support shaft was installed parallel to the lead-screw.

(a) Foremost position  
(b) Aftmost position

Figure 3.4: Longitudinally moving actuator

(a) 0 degree position  
(b) 180 degree position

Figure 3.5: Rotating moving mass actuator

The rotating portion consists of polymer plain bearings in a housing that rotates around the linear portion (acting as a shaft, see Figure 3.6). The rotating portion is made up of a large diameter spur gear and an annular semi-circular mass. A pinion gear motor drives a spur gear to rotate the entire assembly. Power and signals are passed from the motor controller near the tail of the vehicle to the gearmotor using a coiled cable.

Each degree of freedom of the moving mass is driven by a Maxon ECmax30 motor with a 500 count encoder. While the motors are the same, different planetary gearheads were chosen in each case to meet different torque requirements. Measurements for attitude control are provided by an attitude heading and reference system (model: Microstrain 3DM-GX-25 AHRS).
3.4 Bottom Heaviness for a Glider with a Cylindrical Actuator

In the following we compare the properties of a glider designed with unlimited roll control (and a cylindrical moving mass actuator) to that of a traditional glider with limited roll control (and a rectilinear moving mass actuator). This comparison will help underscore some key differences in these respective designs. Consider the orientation of a two-dimensional disk, representing the cross-section of the glider, subject to the forces $W_{RB}$, $W_{MM}$ and $B_{RB}$ as shown in Fig. 3.7. For the purposes of this example, we neglect the tank weight and assume the disk is neutrally buoyant with $B_{RB} - W_{RB} - W_{MM} = 0$. Assume that buoyancy acts at the center of the disk. Let a clockwise rotation be denoted by the angle $\phi$ (the roll angle). Then there are two torques about the center: the torque from the weight of the rigid body $\tau_{RB} = -\sin \phi W_{RB} z_{RB}$, and the torque from the weight of the moving mass $\tau_{MM}$. The motion of the disk is given by $\tau_{RB} + \tau_{MM} = I \ddot{\phi}$, where $I$ is the inertia.

First, consider a linear moving mass actuator design. The moving mass position, expressed
in the body frame \((y_B, z_B)\), is given by \((y_{MM}, z_{MM})\). We assume that \(z_{MM} > 0\) is fixed, whereas \(y_{MM} \in [-\bar{y}_{MM}, \bar{y}_{MM}]\) can vary. The torque from the moving mass is then \(\tau_{MM} = (y_{MM} \cos \phi - z_{MM} \sin \phi)W_{MM}\). At equilibrium \(\ddot{\phi} = 0\), and for a given \(y_{MM}\) the equilibrium roll angle is:

\[
\phi = \left( \frac{y_{MM}W_{MM}}{z_{RB}W_{RB} + z_{MM}W_{MM}} \right).
\]

(3.8)

Generally, \(z_{RB} > 0\) and the rigid body torque \(\tau_{RB}\) provides a restoring moment that significantly contributes to the roll stability of the vehicle (at the cost of a decreased range of attainable roll angles). It is clear from (3.8) that a rectilinear moving mass will be limited in roll to some range \(|\phi| < \pi/2\).

Now consider a cylindrical moving mass actuator design. Let the position of the mass be given in the body frame as \((r \cos \alpha, r \sin \alpha)\) where \(r\) is the distance of the moving mass from the center of the disk, and where \(\alpha\) is the angle of the mass position measured clockwise from the \(y_B\) axis. Define the angle \(\alpha' = \alpha - \phi\). The torque from the moving mass is then \(\tau_{MM} = r \cos \alpha' W_{MM}\). By varying \(\alpha' \in [0, 2\pi]\) the following range of equilibrium roll angles may be obtained:

\[
\phi = \arcsin \left( \frac{rW_{MM}}{z_{RB}W_{RB} \cos \alpha'} \right).
\]

(3.9)

From (3.9) we find that the moving mass must be designed such that \(rW_{MM} \geq z_{RB}W_{RB}\) for the glider to have unlimited roll control. However, if \(z_{RB} \neq 0\) then it is clear that the roll dynamics will be different in both upright (\(\phi = 0\)) and inverted orientations (\(\phi = \pi\)). (In one orientation \(\tau_{RB}\) may provide a restoring force, while in the other it will be destabilizing.) To mitigate this effect the glider should be trimmed such that \(z_{RB} = 0\). In our hypothetical example, \(z_{RB} = 0\) suggests that any nonzero \(W_{MM}\) could rotate the disk to a stable equilibrium point. However, in practice, \(W_{MM}\) must be large enough to rotate the glider in the presence of viscous damping moments. Further, \(W_{MM}\) must be large enough such that, in the absence of a restoring rigid body torque, it provides adequate roll stability in the presence of disturbances.
Chapter 4

Wing and Tail Design

4.1 Wing Sizing

The wings were initially sized to be approximately equal to those of legacy gliders. Holding the wing size fixed makes it possible to evaluate the performance improvements resulting from a larger buoyant lung capacity alone. The scaled planform drawings of the legacy gliders Spray [1], Slocum [2], and the Seaglider [3] were imported into computer-aided drafting (CAD) software. These drawings were used to estimate the details of the legacy glider wing and tail geometry that are not explicitly stated in the literature. (Note that these estimates are based on older publications and do not reflect their modern variants. Further, in calculating the wing and tail geometry the fuselage area was ignored.) The baseline wing for the VTUG was chosen to be a straight rectangular wing with a symmetric airfoil (NACA-0015) and a span and chord comparable to that of legacy gliders, see Table 4.1 for a comparison. Since the net weight of the VTUG is an order of magnitude larger than that of legacy gliders, the wing loading is proportionally larger as well.

The relatively small wings of underwater gliders are often placed far aft of the center of gravity. In such a position the wings act like a combined wing and horizontal tail. A positive (nose-up) perturbation in angle of attack increases the lift force on this aft wing/tail and results in a restoring moment. However, when fuselage effects are considered, the hydrodynamics of the glider do not necessarily contribute to longitudinally (static) stability [20]. Instead, longitudinal static stability is primarily driven by the restoring moment resulting from the offset between the center of gravity and center of buoyancy.

4.2 Wing Harness

The chief advantage of unlimited roll control (i.e. the ability to “flip over”) is that the vehicle may employ various asymmetric wing and tail geometries to improve performance.
Most existing gliders cannot flip over and thus they are typically designed to be symmetric about the vehicle’s horizontal plane. This ensures that the hydrodynamic forces are equivalent in both ascents and descents.

A wing harness was designed to hold the wing in a number of asymmetric geometries, including various dihedral angles (affecting roll stability), wing incidence angles (affecting drag) and at various longitudinal positions (affecting pitch stability). However, to change other wing parameters such as wing camber, twist, or taper ratio, new wings must be fabricated.

A schematic of the wing harness design is shown in Figure 4.1. A large aluminum ring was designed to fit around the hull. To adjust the longitudinal position of the wing harness

<table>
<thead>
<tr>
<th></th>
<th>Net Weight, $W$ (lbs)</th>
<th>Span, $b$ (in)</th>
<th>Mean Geometric Chord, $c$ (in)</th>
<th>Wing Loading, $W/S$ (psf)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seaglider</td>
<td>0.33</td>
<td>29.4</td>
<td>6.68</td>
<td>0.24</td>
</tr>
<tr>
<td>Spray</td>
<td>0.49</td>
<td>38.7</td>
<td>4.13</td>
<td>0.44</td>
</tr>
<tr>
<td>Slocum</td>
<td>0.57</td>
<td>27.1</td>
<td>3.53</td>
<td>0.86</td>
</tr>
<tr>
<td>VTUG</td>
<td>5.51</td>
<td>40.0</td>
<td>4.00</td>
<td>4.96</td>
</tr>
</tbody>
</table>

Table 4.1: Comparison of wing geometry and wing loading
(within a 22 in. range), the ring is moved along a groove machined in the top and bottom of the hull and fastened using set screws. A series of radial holes were drilled in the ring to allow a sliding fixture to be fastened to the ring at various dihedral angles (between 0 - 20 deg. in 4 deg. increments). The sliding fixture held a captive bolt that was allowed to revolve in a groove and change the wing incidence angle by \( \pm 30 \) deg. Figure 4.2 illustrates the range of possible wing geometries provided by the wing harness.

4.3 Vertical Tail Sizing and Placement

The vertical tail was sized with respect to the baseline wing, described in Table 4.1. The wing was assumed fixed at the longitudinal position in the center of the available range provided by the wing harness. Several approaches were used to size the tail. Various tail sizes were iteratively considered and compared to legacy gliders. The center of gravity was estimated to be at the geometric center of each glider analyzed.

First, an empirical method (Raymer [21]) was used. This method uses the tail volume coefficient:

\[
C_{vt} = \frac{L_{vt} S_{vt}}{b S},
\]

where \( L_{vt} \) is the distance between the aerodynamic center of the wing and the aerodynamic center of the vertical tail, \( S_{vt} \) is the vertical tail area, \( b \) is the wingspan, and \( S \) is the wing area.

Next, the coefficient \( C_{n_\beta} \) (the variation of yawing moment coefficient with sideslip angle) was estimated. An analytical approach (Roskam [22]) that included the effects of sweep, taper ratio and aspect ratio was used to estimate \( C_{n_\beta} \). Also, a potential flow solver (Athena Vortex Lattice (AVL) [23]) was used to model the legacy glider and VTUG geometry and estimate \( C_{n_\beta} \). The AVL models of the gliders are illustrated in Figure 4.3. (Note that the effects of the fuselage were ignored in these models.) The results of these analyses are given in Table 4.2.

<table>
<thead>
<tr>
<th></th>
<th>( C_{vt} ) (non-dim.)</th>
<th>( C_{n_\beta} ) (1/rad)</th>
<th>( C_{n_\beta} ) (1/rad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seaglider</td>
<td>0.19</td>
<td>0.824</td>
<td>0.831</td>
</tr>
<tr>
<td>Spray</td>
<td>0.46</td>
<td>0.904</td>
<td>0.811</td>
</tr>
<tr>
<td>Slocum</td>
<td>0.34</td>
<td>2.317</td>
<td>1.829</td>
</tr>
<tr>
<td>VTUG</td>
<td>0.30</td>
<td>2.013</td>
<td>1.410</td>
</tr>
</tbody>
</table>

Table 4.2: Comparison of tail volume and \( C_{n_\beta} \) coefficients
We find from Table 4.2 that legacy gliders have tail volume coefficients $C_{vt} \approx 0.2 - 0.45$. This is much larger than that of sailplanes ($C_{vt} \approx 0.02$) or even jet transports ($C_{vt} \approx 0.09$) [21]. This difference may be attributed, in part, to the fact that gliders typically bank to turn and do not have rudders. In the absence of a rudder, the relatively large vertical tail helps reduce the sideslip angle in a turn. Comparing the estimated $C_{n\beta}$ coefficient for legacy
gliders, we find a reasonable agreement in trend and magnitude between the analytical and numerical estimates. The VTUG’s vertical tail was iteratively sized to obtain a $C_{vt}$ and $C_{n_\beta}$ within the range of legacy glider designs. The resulting tail has a chord length of 3 in, a span of 25 in, and is placed about 26 in aft of the nominal wing position.

### 4.4 Wing Modifications

The Reynolds number for legacy gliders is approximately $Re \approx 3 \times 10^4$. John McMasters (a renowned aerodynamicist who worked on the Liberdade class underwater gliders Stingray and XRay [24]) found that operating at Reynolds numbers below $10^5$ results in a dramatic decrease in the maximum lift-to-drag ratio of smooth airfoils [25] (see Figure 4.4). The baseline VTUG wing operating at 1 m/s has $Re \approx 1 \times 10^5$. However, it is clear from Figure 4.4 that, for smooth airfoils, an even higher Reynolds number is desired.

To explore this idea, a larger wing (60 in. span, and 12 in. chord) was constructed (see
A. Wolek, T. Gode, C. A. Woolsey, J. Quenzer and K. A. Morgansen

Figure 4.4: Maximum airfoil lift-to-drag ratio with Reynolds number (McMasters, Technical Soaring, 1980) [25]

Figure 4.5: Modified glider wings

For this larger wing, Re ≈ 3 × 10^5 (assuming a 1 m/s speed) and the wing loading is closer to that of legacy designs with W/S = 1.1 psf.

4.5 Gliding Performance

Consider the free body diagram of a glider in a steady glide, as shown in Figures 4.6. Summing the net weight \( \bar{W} \), lift \( L \), drag \( D \) along the \((x_s, z_s)\) axes, gives the equilibrium conditions:

\[
\Sigma F_{x_s} : \quad -D - \bar{W} \sin \gamma = 0 \, ,
\]

\[
\Sigma F_{z_s} : \quad L - \bar{W} \cos \gamma = 0 \, ,
\]

where \( \gamma \) is the flight path angle (measured positive clockwise from the horizontal). The velocity triangle \((u, w, V)\) shown in Figure 4.6 with (4.2) gives the depth rate \( w \) (positive
Assuming the parabolic drag law (valid for low speed flight) \( D = \frac{1}{2} \rho_{\text{H}_2\text{O}} V^2 S (C_{D0} + K C_{L}^2) \) and the definition of lift coefficient \( L = \frac{1}{2} \rho_{\text{H}_2\text{O}} V^2 S C_L \) then (4.2) and (4.3) give

\[
\tan \gamma = \left( \frac{L}{D} \right)^{-1} = \left( \frac{C_{D0} + K C_{L}^2}{C_L} \right). \tag{4.6}
\]

We may rearrange (4.6) into a quadratic function of \( C_L \). Solving this equation gives an expression \( C_L(\gamma, K, C_{D0}) \). We find the (for a real-valued solution) the shallowest glide slope possible is \( \gamma_{\text{min}} = \arctan(2\sqrt{K C_{D0}}) \). The speed can then be expressed as

\[
V = \frac{2|\tilde{W}| \cos \gamma}{\rho_{\text{H}_2\text{O}} S C_L(\gamma, K, C_{D0})}. \tag{4.7}
\]

Using (4.4) and (4.5) with (4.7) we may construct the so called “spider plot” of forward speed vs. depth rate by varying \( \gamma \in [\gamma_{\text{min}}, \pi/2] \) for a number of \( |\tilde{W}| \). A sample spider plot for the Seaglider is shown in Figure 4.7. The hydrodynamic parameters assumed for the Seaglider are taken from [20], however we have assumed a hypothetical bladder that gives up to 2.5 L of buoyancy to demonstrate the expected performance gains from a larger bladder. We find that at glide slopes of about 45 deg. a buoyancy equivalent to a displacement of 2.5 L gives a speed of about 1.75 m/s.
Figure 4.7: Spider plot: predicted performance of a hypothetical Seaglider equipped with a high displacement bladder. Lines of constant flight path angle (green) and angle of attack (red) are superimposed.
Chapter 5

Structures

A driving factor in the structural design of the VTUG was to reduce machining complexity, cost, and to maximize the use of commercially available materials.

5.1 Hull Sizing

The hull was designed as a series of connected, constant thickness, tubes. The majority of the glider’s components were machined from aluminum. (Specifically, Al-6061-T6 alloy was used and the material properties in Table 5.1 were assumed for sizing purposes). The use of ribs and stringers, or a tailored compressible hull, was not considered. The American Society of Mechanical Engineers (ASME) standard for construction of pressure vessels [26] was used to determine the required thickness of the hull. (Design guidelines are given in Subsection A, Part UG-28: Thickness of Shells and Tubes Under External Pressure of Section VIII, Division 1: Rules for Construction of Pressure Vessels [26].) The hull was sized assuming it was one long tube approximately equal to the vehicle’s expected length \( L = 62 \) in. The diameter was assumed to be \( D = 9 \) in. The ASME sizing method was iteratively used for a number of thicknesses \( t \). It was found that \( t = 1/4 \) in. met the working pressure requirement (160 psi) with a factor of safety of \( \approx 3.5 \). This is a large factor of safety and the thickness of the tube could be reduced to save weight. However, the ends of the tube must be thick enough to allow an O-ring bore to be machined, and to hold threads for a number of set

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of Elasticity, ( E )</td>
<td>10.3 ( \times 10^6 ) psi</td>
</tr>
<tr>
<td>Poisson’s Ratio, ( \nu )</td>
<td>0.33</td>
</tr>
<tr>
<td>Tensile Yield Strength, ( \sigma_Y )</td>
<td>38 ksi</td>
</tr>
<tr>
<td>Density, ( \rho )</td>
<td>0.0975 lbs/in(^3)</td>
</tr>
</tbody>
</table>

Table 5.1: Material properties of AL-6061-T6
screws. Machining such a bore is challenging for long thin tubes. After consulting with machinists, it was determined that the 1/4 in. thickness was a reasonable size that could be machined to tolerance for the tube lengths desired (note that a lathe “steady rest” tool was required). This thickness was also a stock tube size that was readily available.

The ASME code result were also verified analytically. The hoop stress, axial stress, and collapse pressure of a cylindrical thin-walled pressure vessel, with ends capped, under a uniform external pressure are given in Roark’s formulas for stress and strain [27] (Chapter 13, Table 13.1, Case 1c). The hoop stress for a given external pressure \( q \), at points away from the hull ends, is:

\[
\sigma_{\text{hoop}} = \frac{qR}{t} \tag{5.1}
\]

and the axial stress is:

\[
\sigma_{\text{axial}} = \frac{qR}{2t} \tag{5.2}
\]

The external collapsing pressure of a thin tube under uniform external pressure is:

\[
q_{\text{collapse}} = \frac{t}{R} \left( \frac{\sigma_y}{1 + (4\sigma_y/E)(R/t)^2} \right) . \tag{5.3}
\]

Evaluating the conditions (5.1)-(5.3) for the design of the hull we obtained the factors of safety shown in Table 5.2. It is clear from Table 5.2 that the most concerning condition is the external collapse pressure.

We also considered the buckling of the hull. The critical pressure for buckling of a thin tube with closed ends held circular under uniform external pressure is given in Roark’s formulas for stress and strain [27] (Chapter 15, Table 15.2, Case 20a):

\[
q_{\text{buckle}} = \frac{0.8E t}{R} \left( \frac{1}{n^2 \left[ 1 + \left( \frac{nL}{\pi R} \right)^2 \right]^2 + \frac{n^2 t^2}{12R^2(1 - \nu^2)} \left[ 1 + \left( \frac{\pi R}{nL} \right)^2 \right]^2} \right) , \tag{5.4}
\]

where \( n \) is the number of lobes formed by the tube in buckling, and \( R \) is the inner radius [27]. For \( n = 2, 3, 4 \) we obtained the critical pressures and factors of safety given in Table 5.3. From Table 5.3 we find that the first buckling mode with \( n = 2 \) has the lowest safety factor of 3.5.
### 5.2 Marinization

Since the glider was designed to operate only for short durations in fresh water, a rigorous marinization of all components was not required. Nonetheless, several methods were used to minimize the effects of corrosion. Regular polishing of the hull is one method of removing oxidation and preventing corrosion (see Figure 5.4(b)).

![Figure 5.1: Hull fabrication and maintenance](image)

(a) The hull was machined from a series of stock aluminum tubes
(b) Polishing the hull removes oxidation and prevents corrosion

Anodizing and painting is a more permanent corrosion prevention method. Alternately, a chromate conversion coating can be applied to small parts without the need for specialized tools. This process consists of cleaning the aluminum parts using Alumiprep 33 and brushing or immersing them in Alodine 1201. The result is a coating with a bronze appearance (see Figure 5.2). While we found this method produced good results for small parts by immersion, applying Alodine 1201 evenly to the entire hull with a brush was difficult and not pursued.

Lastly, a sacrificial anode was also employed by the VTUG for corrosion resistance. This consisted of a small piece of zinc, designed for use with boats, threaded into the aft endcap.

### 5.3 Endcap Sizing

The most common pressure vessel caps are either hemispherical, ellipsoidal or otherwise curved sections. However, to simplify manufacturing it is desirable to construct the endcap

<table>
<thead>
<tr>
<th>Number of lobes, $n$</th>
<th>$q_{\text{buckle}}$ (psi)</th>
<th>Factor of Safety</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>558</td>
<td>3.5</td>
</tr>
<tr>
<td>3</td>
<td>1,201</td>
<td>7.5</td>
</tr>
<tr>
<td>4</td>
<td>2,125</td>
<td>13.3</td>
</tr>
</tbody>
</table>

Table 5.3: Analysis of hull buckling criteria
from a flat plate. The endcap was sized using the ASME standard for construction of pressure vessels [26]. (Design guidelines are given in Subsection A, Part UG-34: Unstayed Flat Heads and Covers of Section VIII, Division 1: Rules for Construction of Pressure Vessels [26].) A flange type design was chosen (see Figure 5.3).

\[ t_{\text{endcap}} = d \sqrt{\frac{C q}{\sigma_y E}}, \]  

(5.5)

where \( d \) is the inner diameter, \( C \) is a factor depending on the method of attachment (\( C = 0.33 \) using the method in Figure 5.3) and \( E \) is the joint efficiency for the weld. In the VTUG design the endcap is attached using a series of set screws that are sized with a very high safety margin, therefore it is assumed to be equivalent to a perfect weld, \( E = 1 \). From (5.5) the required thickness is \( t_{\text{endcap}} = 0.32 \) in.

To verify this result analytically, we considered the stress and deflection of a flat circular plate of constant thickness with fixed edges, subject to a uniform pressure load, as given in Roark’s formulas for stress and strain [27] (Chapter 11, Table 11.2, Case 10b). Performing the analysis one finds the above determined thickness satisfactory. The final endcaps were oversized to a 0.5 in. thickness to accommodate a number of threaded fittings that were installed in each endcap face.
5.4 O-ring Seals

An easily machinable and reliable design for sealing the endcaps was desired. Each endcap was machined with a pair of O-ring grooves (for redundancy) and one set screw groove to secure the endcap (see Figure 5.5). The hull was bored to Parker O-ring handbook specifications [28], whereas the dimensions of the endcap plug and O-ring grooves were slightly modified to ease installation (within acceptable O-ring stretch and squeeze tolerances). We found that endcap removal and cleaning was easiest with a silicone compound (such as Chemplex 710) rather than a petroleum-based grease (such as Parker O-Lube).

![Figure 5.4: Installation of endcap](image)

(b) Endcap and O-Rings being lubricated

(a) Bore of the hull being lubricated

5.5 Pressure Relief Valve and Vent Plug

Heating of electrical components within the hull can cause the air inside the pressure vessel to expand. In addition, the buoyancy control system houses some high pressure components that can potentially pressurize the pressure vessel to a dangerous level. As a safety precaution an anodized aluminum shallow depth pressure relief valve (Model: Prevco LP-PRV) was permanently installed in both the main hull and in the buoyancy control module. This valve opens at a differential pressure of 5 psid to exhaust air from inside the pressure vessel.

Further, a vent plug (Model: Prevco No. 4 Vent Plug) was used to exhaust any excess pressure (below 5 psid) remaining in the hull after a deployment. This vent plug also helps in installing and removing the endcaps. By removing the vent plug the hole in the endcap allows air to move into and out of the hull preventing pressure from building up as the endcap is installed.

To prevent water condensation on electronics that heat up during a deployment, humid air contained in the hull may be removed by pulling a vacuum and “backfilling” it with dry nitrogen gas. While the vent plug port allows this option, this procedure was not necessary for the typically short deployments of the VTUG.
Figure 5.5: Sample endcap design
Chapter 6

Recovery Aids

A number of recovery aids are used to help locate the glider in adverse weather conditions, aid in bringing the vehicle aboard, and to prevent the loss of the vehicle in emergency situations. The glider’s tail houses a series of light emitting diodes (LEDs), an LED driver, the GPS, radio modem and antenna, as well as a solenoid mechanism to actuate the dropweights. The tail module is physically isolated from the remaining vehicle so that damage to the tail boom would not flood the main hull. Electrical signals are passed between the tail and the glider’s main hull using a short wet-mateable cable (as shown in Figure 6.1).

Figure 6.1: Tail module connected to aft end of the hull

6.1 Flashers

A series of four LEDs (model: Luxeon Rebel 6500K Cool White) were embedded in a clear polyvinyl chloride (PVC) housing in the tip of the tailboom. These LEDs were used to improve visibility of the glider during retrieval. They could also be used to indicate some basic status information during deployment (e.g. using a particular flash pattern during power on, prior to beginning a dive, or to indicate an emergency). The LEDs were intended
to flash for short durations. (Continuously powering them for long periods of time may overheat components in the assembly.)

![Image of tailboom containing GPS and LED flashers](image)

**Figure 6.2: Tailboom containing GPS and LED flashers**

### 6.2 Drop-weights

A drop-weight mechanism was designed to release weights and force an ascent in the event of a buoyancy system failure. The primary requirement was for the weights to be heavy enough to cause the vehicle to ascend if all of the buoyancy from the bladder was lost. Assuming the vehicle trims to neutral buoyancy with $B_{BE,0}$ (and a full tank) the drop-weights must have a net weight $\tilde{W}_{DW} > B_{BE,0}$ to be effective.

Several designs for the drop-weight release system were considered, including a galvanic release, shearing a thin attachment wire with a solenoid, or using a pyrotechnic fastener. Ultimately, a mechanism consisting of two interlocking cylindrical masses released by retracting a pin was designed (as shown in Figure 6.3(a)). The drop-weights interlock such that two concentric holes align, allowing a retractable pin to protrude from the tail pressure vessel and hold the weights fixed. Upon retracting the pin, the drop-weights are free to rotate and fall under their own weight.

A schematic of the drop-weight design is shown in Figure 6.4(a). The resistance forces of the dynamic O-ring $F_{\text{seal}}$ can be estimated using the Parker O-Ring Handbook [28]. The pin is spring loaded, providing a Hookian force $F_{\text{spring}} = kx$ where $k$ is the spring coefficient and $x$ is the spring deflection. The net weight of the drop-weights acting on the pin results in a frictional force $F_f = \mu\tilde{W}_{DW}$ where $\mu$ is the friction coefficient between the steel pin and aluminum drop-weights in water. The ambient pressure force $F_{\text{amb}} = \rho_{\text{H2O}}gzA$, where $A$ is the pin face area, acts to depress the pin at depth. If the available solenoid force $F_{\text{solenoid}}$ satisfies

$$F_{\text{solenoid}} > F_{\text{spring}} + F_{\text{seal}} + F_f,$$

then the drop-weight can be actuated at any depth. (With increasing depth $F_{\text{amb}} > 0$ and this reduces the required $F_{\text{solenoid}}$). Note that available solenoid force typically increases as...
A second requirement is for the spring to be sufficiently stiff, such that the spring deflection $x$ is significantly less than the thickness of the drop-weight-plate $x_{DW}$. (This is required so that the pin is not inadvertently retracted at depth causing the dropweights to fall). Thus we have the conservative condition (neglecting friction forces), that at maximum depth

$$x = \frac{F_{\text{amb}}}{k} \ll x_{DW}.$$ 

Based on these design criteria a solenoid (model: Ledex 5EC) and spring were selected. The drop weight assembly also included the vertical stabilizers and fairings as shown in Figure 6.3(b).

### 6.3 Emergency Locater

A standalone acoustic pinger (model: Vemco V9-1) can be attached to the glider’s hull during untethered deployment. This small device, originally designed for tagging fish, provides a continuous ping for 20 days. In the event that the glider does not surface a diver with a specialized receiver may attempt to locate the source of the ping and recover the vehicle.

### 6.4 Boat-side recovery

To aid in recovering and deploying the glider, a retractable glider “chute” was designed as shown in Figure 6.5(a). To assist in handling the glider while it is near the boat (e.g. to prevent collisions and to steer the glider towards the chute) a “snare” consisting of a hollow pipe threaded with a rope and a carabiner was used to hook the glider.
Figure 6.4: Dropweights

(a) Forces acting in dropweight design

(b) Dropweights with fairings

Figure 6.4: Dropweights
Figure 6.5: Aids for deploying and recovering the glider

(a) Glider chute

(b) Glider snare
Chapter 7

Electronics

Most of the electronics were housed in the “main electronics bay” located in the tail section of the vehicle as shown in Figure 7.1. In the following section we detail the computer, power, sensor, actuator and communication electronics used on the VTUG.

Figure 7.1: Main electronics bay

7.1 Computer

The glider computer consists of a series of connected boards that conform to the PC/104-Plus embedded computer standard. A collection of such boards is often called a “stack”. At minimum, a power board and a single board computer are required in a stack. Additional functionality is provided by connecting peripheral boards. The PC/104-Plus standard ensures that boards from various manufactures can be integrated into one computer. This standard is commonly used in applications requiring a rugged and small computer. The components of the VTUG PC/104 stack are listed in Table 7.1, along with some selected specifications.
<table>
<thead>
<tr>
<th>Description</th>
<th>PC/104-Plus Stack</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Supply</td>
<td>Jupiter (JMM-SIO-XT)</td>
<td>50 Watts Power</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7-30 V Input</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5 V Output (10A)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>12 V Output (2A)</td>
</tr>
<tr>
<td>Main Computer</td>
<td>Pegasus (PGS800-256)</td>
<td>500 MHz, AMD LX800 Processor</td>
</tr>
<tr>
<td></td>
<td></td>
<td>256 MB RAM</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4 USB 2.0 Ports</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2 RS-232 Serial Ports</td>
</tr>
<tr>
<td>Input/Output</td>
<td>Diamond (DMM-16-AT)</td>
<td>6-channel, 16-bit A/D</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4-channel, 12-bit D/A</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8 digital inputs</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8 digital outputs</td>
</tr>
</tbody>
</table>

Table 7.1: Glider PC/104 computer components

### 7.2 Power

The VTUG is powered by 3-cell lithium polymer (LiPo) batteries. The nominal voltage of each battery is about 11.1 V (and over 12 V at full charge). On the VTUG, two such batteries are connected in series to provide approximately 24 V. Advantages of LiPo batteries are that they have a high energy density, high discharge rates, and relatively flat discharge curves. (A flat discharge curve of voltage (V) vs. capacity (mAh) implies that relatively small voltage drops are expected for most of its operation. However, this is typically followed by a sudden and rapid voltage drop as the battery nears depletion.) Knowledge of proper handing techniques is necessary to prevent common safety hazards associated with LiPo batteries. They must be charged with a specialized charger that balances the voltage in each cell, thus a total of 12 conductors are needed to charge two batteries. (The VTUG was not designed with this many waterproof conductors dedicated for battery charging, thus one limitation of using these batteries is that the endcap had to be removed after each deployment to recharge the vehicle, as shown in Figure 7.2.)

It is desirable to be able to switch between external power and battery power to reserve battery charge prior to deployment. To achieve this a latching relay was used to place both external and battery power sources in parallel prior to cutting off external power, as shown in Figure 7.3. (Note: This method requires the main power switch to be the only path to ground. If an alternate path to ground exists when the switch is flipped, then any components along this path (e.g. VGA, ethernet connectors) may be damaged by excessive current.) A disadvantage of this approach is the need to return to external power to unlatch the relay and remove power from the glider. Other vehicles sometimes use a magnetic switch instead to cut the main power. (In such designs the the switch is activated by manually removing a magnet on the exterior of the vehicle.)
The external power provided to the glider was typically either a DC power supply (in the lab), or a 12 V boat battery (in the field). The boat battery voltage supply was increased to 24 V using a DC-DC converter (model: Vicor Corporation V28B24C150BL). A diode was used to protect against inadvertent reverse polarity while connecting power. To power the glider, a wet-mateable cable was plugged into the tail of the vehicle and into part of the ground station node shown in Figure 7.4. The power draw by the glider was monitored using a Watt meter (model: AstroFlight Inc. 101Q). Typical idle power consumption was about 48 Watts (approx. 2 A at 24 V).

To scale the voltage of the battery such that it can be read by the I/O board a simple voltage divider and op-amp follower circuit was used, see Figure 7.5(a). The A/D converter was set for an input range of 0 - 5 V, and the battery voltage range (min. 15V, max 25V) was scaled to approximately 2.5 - 4.5V so that it can be measured. A calibration curve was developed that maps the op-amp output to true battery voltage, see Figure 7.5(b).
A custom sensor was constructed to detect leaks. The sensor consists of two parallel wires bound by heat shrink tubing with bare conductors exposed in regular intervals as shown in Figure 7.6(a). This sensor can be shaped to conform to the hull interior and glued into place (Figure 7.6(b)). On the VTUG, three such sensors were used (two in the main hull, one in the buoyancy control module).

When the exposed conductors come in contact with water, current flows between the parallel wires (with a large resistance). A voltage divider circuit was designed (Figure 7.7) to take advantage of this effect. When the leak sensor encounters water, the circuit is closed and a voltage drop is detected. The circuit was tested by applying a water droplet and observing the response.
7.3.2 Depth Pressure Sensor

To test and calibrate the pressure sensor (model: GE UNIK 5000, range: 0 - 200 psia) in the lab, a “dead weight tester” was used. This device uses a gas cylinder to lift a calibrated set of weights and outputs a precise pressure. The calibration of the depth sensor within the expected range is shown in Figure 7.8.

![Figure 7.8: Calibration of the depth sensor using a dead weight tester]
7.3.3 Bladder Pressure Sensor

A differential bladder pressure sensor (model: GE UNIK 5000, range: 0 - 5 psid) was used to measure the pressure difference between one port exposed to the ambient water and the other port exposed to the air pressure inside the bladder. Benchtop tests showed an unexpected change in the pressure sensor reading with orientation of the transducer (when both ports were exposed to ambient pressure). This effect was most likely caused by the weight of the fluid within the differential pressure sensor itself. To characterize this error, an experiment was conducted in which the transducer was moved through a 180 deg. range while recording the measured pressure as shown in Figure 7.9.

![Test setup for rotating pressure sensor through 180 degrees](image1)

![Pressure sensor output](image2)

Figure 7.9: An experiment was designed to measure the change in reported bladder pressure resulting from the orientation of the transducer

7.3.4 Travel Limit Sensors

Upon powering the motor controllers, a homing routine is initiated that seeks to find a reference point by turning the motors in a predefined direction until a limit switch signal is activated. Initially, hall effect sensors were used for this purpose however these require a magnet to be installed on the moving mass. This magnet interfered with the magnetometer and was replaced by a contact switch instead (as shown in Figure 7.10).
7.3.5 Inertial Measurement Unit

It is desirable to mount the inertial measurement unit (IMU) parallel to the body axis and near the CG to minimize the error in the computed Euler angles. If the IMU is mounted in an orientation different than that of the body axis, then the IMU reports local Euler angles and a computation involving rotation matrices is required to compute the true body axis Euler angles. This is particularly problematic if the rotation matrix involves the heading angle. (The heading angle is measured by a magnetometer and is often a noisy measurement that is susceptible to interference.) The IMU on the VTUG was iteratively mounted in several locations to minimize magnetic interference. While the mounting location away from the motors, shown in Figure 7.11, was best, the interference was still a significant source of error.

After mounting the IMU inside the glider, the vehicle orientation was adjusted using a bubble level such that the glider was horizontal with wings level. The pitch and roll bias of the IMU was then recorded by collecting IMU data for several minutes as shown in Figure 7.12.

To characterize the interference resulting from moving the mass on the magnetometer, the glider was left stationary while the moving mass moved to various linear and rotational
Figure 7.12: Determining the bias in the inertial measurement unit mounting location positions (see Figure 7.13). We found that rotational movement of the mass had the greatest effect on the heading error and resulted in deviations up to $\approx 13$ deg.

Figure 7.13: Characterizing the interference of the moving mass on the magnetometer reading
7.4  Actuators

7.4.1  Flow Controllers

Each solenoid (the inflator and deflator) is driven by a separate valve controller as shown in Figure 7.14. These drivers take an analog signal and amplify it appropriately to open or close the solenoid by the desired amount. The driver circuit boards have potentiometers that allow adjusting several operating parameters (e.g. dead band, dither, max current draw). The inflator solenoid was driven by the Enfield Technologies, D1 PWM Valve Driver. The deflator solenoid was driven by the iQValves Standard PFCV Signal Amplifier.

![Figure 7.14: Proportional solenoid valve drivers in the buoyancy engine](image)

7.4.2  Motor Controllers

Each Maxon motor (for mass rotation and mass translation) requires its own dedicated controller. These controllers are housed in the main electronics bay next to the batteries (see Figure 7.15). The EPOS2 motor controller takes the encoder input, travel limit signals, and supplies power to the motor coils. The controller is capable of operating in a number of modes including position, velocity, or acceleration modes. The controller communicates with the PC/104 through a universal serial bus (USB).

7.4.3  Drop-weight Trigger

A digital output was used to control the drop-weight because the analog ports of the PC/104 were all occupied. The drop-weight is suitable for being triggered by a digital signal because it only has two states. A circuit was designed to supply power to the drop-weight solenoid whenever a logical high signal was detected (see Figure 7.16). For this purpose a metal-oxide-semiconductor field-effect transistor (MOSFET) was used. A pull down resistor is used to keep the gate input at GND, in case a wire comes loose or if nothing is being commanded on the digital port. When the input reaches a 3-5 V, then full power (24 V) is supplied to the...
solenoid. A fly-back diode is used to prevent reverse voltage from damaging the MOSFET when the solenoid is turning off.

![Figure 7.15: Motor controllers during assembly of the electronics main bay](image)

7.4.4 Air Supply Solenoid Relay

As discussed in Section 2.3.2, the inflator valve has some associated leakage rate even when it is “closed”. If the scuba tank valve is left opened while the bladder pressure is not being regulated, then this will cause the bladder to slowly fill up and rupture. To prevent this, a flow control valve was placed in series with the inflator to shut off the air supply when the inflator was not in use. This solenoid was triggered by sending an analog voltage to a
op-amp and relay circuit (see Figure 7.17) that then provided full power (24 V) to the flow control solenoid.

![Relay Circuit Diagram]

**Figure 7.17: Relay circuit to power air supply valve**

### 7.5 Communication

There are four modes of communication with the glider: directly connecting to the PC/104, establish a secure shell over ethernet, creating a serial socket over radio, or using acoustic channels. For details regarding the acoustic communication, see Section 8. During benchtop tests it is possible to connect a VGA monitor and keyboard directly to the PC/104. When the glider is sealed, a 16-pin wet mateable cable plugged into the tail provides an ethernet connection. This high-bandwidth mode of communication is used to update the glider’s software and download log files after field trials. When the glider is deployed the radio (model: Freewave FGR2 Series, 900 MHz) is the primary method of communication.

### 7.6 Pinouts and Connectors

Deans connectors were used for high-amp connections (e.g. to connect the batteries). For most other connections either Powerpole or D-sub connectors were used depending on the number of connectors required and the space available. Molex connectors were used to connect to the WHOI micro-modem and EPOS motor controllers. For external connections, Teledyne wet-mateable cables were used to connect the transducer. Subconn micro wet-mateable connectors were used to join the buoyancy engine to the main hull, the tail to the main hull, and the ground station to the main hull.

A detailed pinout diagram and schematic is maintained for the interfaces between major electrical assemblies. Figure 7.18 shows the schematic of connections made on a breadboard that distributed power from the batteries to other systems of the vehicle, as well as routed signal from the D-sub endcap connector (shown in Figure 7.19). An early revision of the full schematic of electrical connections on the VTUG is shown in Figure 7.20.
Figure 7.18: Power distribution board and aft endcap connector

Figure 7.19: Aft endcap connector
Figure 7.20: Glider electronics system schematic
Chapter 8

Acoustic Positioning and Communication

8.1 Underwater Navigation and Communication

A number of positioning and communication systems exist for land and air vehicles operating autonomously within a relatively short distance (several miles) of a ground control station. In such applications, a radio and global positioning system (GPS) receiver often suffice to perform the communication and positioning tasks. However, because electromagnetic waves are strongly attenuated in water (especially seawater) these technologies cannot be effectively used with underwater vehicles while they are underway. Instead, acoustic methods are typically employed. Acoustics communication is relatively bandwidth limited and can be challenging in certain environments, especially where the speed of sound profile is not properly characterized or accounted for. Further challenges arise from multi-path, propagation delays and Doppler-shifts (when applicable). Nonetheless, this method of underwater communication remains the most effective and widely used.

In the absence of an acoustic positioning system, underwater vehicles typically employ the dead reckoning technique. Dead reckoning uses an initial position (e.g. obtained using GPS) and measurements from an inertial navigation system to estimate the position of the vehicle. (Often the inertial navigation system consists of accelerometers, gyroscopes and a magnetometer.) Sources of error associated with the technique include the inherent sensor error and exogenous disturbances [29, 30]. More advanced sensors, such as Doppler Velocity Logs (DVLs) or gyro-compasses may improve a dead reckoned estimate, however they ultimately suffer from the same sources of error.

Acoustic ranging systems provide independent position estimates that do not degrade with time (i.e. they do not suffer from error drift). They require deploying a number of transducers, and accompany acoustic modems, at known locations in the vicinity of an underwater vehicle. The distance between these transducers is called the baseline. Depending on the geometry of the transducer network they may be referred to as short baseline (SBL), ultra
short baseline (USBL), and long baseline (LBL) positioning systems (see Figure 8.1).

(a) A USBL is typically < 10cm
(b) A SBL is typically 20 − 50m
(c) A LBL is typically 100m − 10km
(d) A MLBL is similar to a LBL however the acoustic nodes are mobile

Figure 8.1: Types of acoustic baseline geometries

An acoustic ranging system works by successively sending an acoustic pulse (a “ping”) to each transducer in the network. Upon receiving a response, the travel time of sound between two transducers can be computed. Assuming some acoustic properties of the water, namely the speed of sound profile, the distance between transducers may be estimated. Further, these measurements are used to estimate the unknown position of the underwater vehicle. The estimation algorithms may either be static or dynamic. Static estimates are based purely on the geometry of the system (e.g. the intersection point of a series of spheres is estimated using a least-squares solution). Dynamic estimates (e.g. a Kalman filter) incorporate a dynamic model of the vehicle and additional sensor measurements.

While much of the previous discussion was limited to discussing acoustic positioning. The same acoustic network can be used to communicate with the vehicle while it is underway. However, because this communication is bandwidth-limited, generally only the most critical information is communicated in this way (e.g. occasional updates on vehicle health and status, or high-level commands from human operators). For a thorough discussion of underwater navigation techniques, relevant estimation algorithms, and details pertaining to the VTUG acoustic positioning and communication system see [31].
The LBL positioning system used by the VTUG consists of three mobile nodes/beacons built to serve as part of the acoustic network. One of the three beacons is designed to be on-board the boat and accessible to operators. This beacon is referred to as the ground station (GS) node. It houses some additional hardware for communicating with and powering the glider. The remaining two nodes are deployed on anchored, inflatable rafts (see Figure 8.2). We refer to these two beacons as B1 and B2, respectively.

![Figure 8.2: Beacon deployed on a raft](image)

![Figure 8.3: All three beacons during a tank test](image)
Figure 8.3 shows the three nodes during a communication test of the acoustic transducers in a water tank at Virginia Tech. The electrical components are housed in an aluminum 80 × 20 frame, placed inside a rugged Pelican case. The frame has two levels, as illustrated in Figure 8.4.
Figure 8.4: Hardware within the acoustic beacon

(a) First level of the beacon’s frame

(b) Second level of the beacon’s frame

(c) Antenna and acoustic transducer
Power is provided by a 12 V marine deep-cycle battery. A power inverter converts this into an alternating current (AC) source to power the laptop (other devices in the system use this direct current (DC) source directly). The beacon is equipped with a GPS, radio, and an acoustic modem. The acoustic modem, developed by Woods Hole Oceanographic Institute (WHOI), consists of several components: the micro-modem, power amplifier and co-processor (see [32] for a detailed description of these components). Serial communication from these devices is converted to a laptop-compatible USB connection. An array of switches provides independent power to each device to assist with debugging (and in some cases to avoid powering unused devices.) A 25 m Teledyne wet-mateable cable is used to submerge the transducer to a predetermined depth. To prevent the cable from swaying due to currents, it was weighed down by running it through a 4 foot plastic pipe. All the components within the Pelican case are friction fit using foam to prevent them from moving around. Table 8.1 gives a summary of the technical specifications of the components in each of the three nodes.

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Battery</td>
<td>12 V, 575 CCA (cold cranking amps)</td>
<td>DieHard RV 27M</td>
</tr>
<tr>
<td>Inverter</td>
<td>500 W, Two 110 V AC outlets</td>
<td>Black&amp;Decker 500</td>
</tr>
<tr>
<td>Laptop</td>
<td>1.5 GHz, 4 GB RAM, Intel i3-3120M</td>
<td>Lenovo G580</td>
</tr>
<tr>
<td></td>
<td>- Ubuntu 12.04 (Precise Pangolin)</td>
<td></td>
</tr>
<tr>
<td>Radio</td>
<td>900 MHz</td>
<td>Freewave FGR2</td>
</tr>
<tr>
<td></td>
<td>115.2/153.6 kbps speed</td>
<td></td>
</tr>
<tr>
<td>Antenna</td>
<td>12°, 150W</td>
<td>DATA-LINC (3db)</td>
</tr>
<tr>
<td></td>
<td>Frequency: 896-940 MHz</td>
<td>Omni-directional antenna:</td>
</tr>
<tr>
<td></td>
<td>Nominal impedance: 50Ω</td>
<td>EAN0900WC</td>
</tr>
<tr>
<td>GPS</td>
<td>FTDI interface</td>
<td>GS407 U-Blox5H</td>
</tr>
<tr>
<td></td>
<td>2 Hz refresh rate (NMEA)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Sarantel omni-directional geo-helix S-type active antenna</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Accuracy: ~ 10m</td>
<td></td>
</tr>
<tr>
<td>Acoustic Modem</td>
<td>12 V, 4 KHz bandwidth (25-29 KHz)</td>
<td>WHOI Micro-Modem</td>
</tr>
<tr>
<td></td>
<td>QPSK modulation</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Texas Instruments 1 GHz FPP DSP</td>
<td></td>
</tr>
<tr>
<td>Acoustic Transducer</td>
<td>Resonance frequency: 28 KHz</td>
<td>BTech BT-1RCL</td>
</tr>
<tr>
<td>Wet Mateable Cable</td>
<td>Max depth: 700 m</td>
<td>Teledyne Belden 8412</td>
</tr>
<tr>
<td></td>
<td>25 m shielded 20AWG 3 pin cable</td>
<td></td>
</tr>
<tr>
<td>Serial to USB hub</td>
<td>Four port RS-232 to USB serial adapter</td>
<td>StarTech ICUSB2324I</td>
</tr>
</tbody>
</table>

Table 8.1: Node hardware specifications
The acoustic modem wiring diagram is shown in Figure 8.5.

8.3 Software Design

The acoustic communication software was designed to be compatible with the VTUG’s software. It was developed using the Robot Operating Systems (ROS) architecture. The computer runs on the Ubuntu 12.04 operating system. The open-source “Goby Underwater Autonomy Project” [33] library was used for interfacing with the WHOI acoustic modems via National Marine Electronics Association (NMEA) sentences. Some common NMEA commands used to configure the modems and perform the modem-to-modem pinging are listed in Table 8.2.
Table 8.2: Sample NMEA commands used by the WHOI acoustic modems (adapted from [34])

<table>
<thead>
<tr>
<th>Command</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$CCCFQ,NNN</td>
<td>Query NVRAM configuration parameter</td>
</tr>
<tr>
<td></td>
<td>e.g. $CCCFQ,AGN: Prints the current value of the “analog gain”</td>
</tr>
<tr>
<td></td>
<td>e.g. $CCCFQ,ALL: Prints a list of all current configuration parameters</td>
</tr>
<tr>
<td>$CCCFG,NNN,vv</td>
<td>Used to set NVRAM configuration parameter, host to modem</td>
</tr>
<tr>
<td></td>
<td>NNN: Name of the NVRAM parameter to set</td>
</tr>
<tr>
<td></td>
<td>vv: New value to set</td>
</tr>
<tr>
<td></td>
<td>e.g. $CCCFG,AGN,250: Sets the analog gain parameter to 250</td>
</tr>
<tr>
<td>$CCMPC,SRC,DEST</td>
<td>Mini-packet ping command</td>
</tr>
<tr>
<td></td>
<td>To ping a beacon to compute one-way-travel-time</td>
</tr>
<tr>
<td></td>
<td>SRC: Source (unit designated as ping originator)</td>
</tr>
<tr>
<td></td>
<td>DEST: Destination (unit designated as receiver of ping)</td>
</tr>
<tr>
<td>$CAMPA,SRC,DEST</td>
<td>A ping has been received</td>
</tr>
<tr>
<td></td>
<td>When a ping is received at the destination beacon, the modem sends this</td>
</tr>
<tr>
<td></td>
<td>command to the user</td>
</tr>
<tr>
<td></td>
<td>SRC: Source (unit designated as transmitter)</td>
</tr>
<tr>
<td></td>
<td>DEST: Destination (unit designated as receiver of ping)</td>
</tr>
</tbody>
</table>

The modem is configurable with a number of various settings that are detailed in [34]. A typical transmission consists of “data packets” which are further broken up into “frames”. Two modes of communication are available: frequency shift keying (FSK) and phase shift keying (PSK). Each of these modes can accommodate a different configuration of packets/frames and their respective memory sizes. The later mode appeared to be more robust in the field and was used with the VTUG.

The ROS software architecture is based on the idea that the robot’s functions can be broken down into task-specific “nodes” that communicate via predefined “topics”. A high level chronological sequence of events that results in a LBL ranging cycle and a “heartbeat” (a brief vehicle health status) message being communicated by the glider is shown in Figure 8.6. (In Figure 8.6, boxes with a black border denote specific ROS nodes involved.)
Figure 8.6: Sequence of events for LBL ranging and heartbeat communication
Chapter 9

Software

Human operators can command and control the glider via the “ground station”. In more complicated operations the system may also include a series of mobile acoustic beacons, as discussed in Chapter 8. Software was developed for the both the glider, acoustic beacons, and ground station within a single project.

9.1 Motivation

The Virginia Tech Underwater Glider software was created with easy portability and extendability in mind. The code is open source and is licensed under the GNU General Public License v3 <http://www.gnu.org/licenses/gpl.html>. The code should be:

- Open source and freely distributable so that anybody can contribute to the project.

- Easily accessible so that university students, who typically have limited time to familiarize themselves with the project, can be involved in the development.

- Portable between different glider platforms with minimal effort. The principal functions that all underwater gliders share should hence be created at an abstract software level that is independent of the detail of the physical devices that implement those functions. This separation of general functionalities from the physical devices allows swapping the underlying software module with ease, and without any change to the interface that the software provides towards the user.

9.2 Operating System

The operating system used on the glider and acoustic beacons is Ubuntu 12.04 (Precise Pangolin). Note that the glider runs a server version of Ubuntu 12.04 that requires less
computational power. This operating system was used because of its reliability, minimal CPU requirements, and the availability of open source software.

### 9.3 Robot Operating System

The Robot Operating System (ROS) was created by the Willow Garage project. It runs on Ubuntu Linux (among other platforms) and was developed with a view toward portability, extendability, and a relatively easy learning curve for new developers. ROS is a community driven, open-source collection of software packages designed to interface with robot hardware bringing the user both low and high level control. It was chosen for use on the glider for its repository of existing code and for its nodal architecture, that manages and simplifies the communication of multiple processes (nodes). A chief advantage of this architecture is that similar tasks can be grouped into independent nodes that may be executed independently. For example all of the sensing, control, or estimation tasks can be grouped into three separate nodes. Nodes communicate through the use of messages with a pre-defined structure called topics. Inter-node communication is managed by a ROS Master. This approach makes the system more robust to failure (e.g. a single node can fail and be restarted without crashing the entire system) and also modular (e.g. a single node can be updated, replaced or run at various frequencies). Various parameters (e.g. control gains, speed of sound estimate, etc.) are contained in human readable text files (ending with extension .yaml) that can be easily modified in the field.

### 9.4 Software Architecture

A schematic of the node-topic structure employed by the VTUG is shown in Figure 9.1. Similarly, the node-topic structure employed by an acoustic beacon is shown in Figure 9.2. In the following, the role of each node is briefly described:

- **/comms**: manages the communication of the glider with the ground station and other acoustic nodes. It serves three primary roles:
  1. Relay radio communications:
     - Command queue editing (add, insert, remove, refresh, clear)
     - Broadcast of state estimates
  2. Broadcast heartbeat data (acoustically). (A heartbeat is a short message that contains vital glider information.)
  3. Ping LBL beacons and process ping response + beacon GPS data (acoustically).

This node must be running in any experiment where the glider is going to be remotely operated. The comms node behaves reactively to incoming data from the comms hardware or to data transmission requests from associated nodes.
• /acomms: interacts with the WHOI micro-modem hardware. The comms and beacon nodes send transmit requests and receive data messages from the acomms node. This node utilizes the Goby Underwater Autonomy Project libraries.

• /radio: provide read/write access to a port which connects the glider and ground station over a (relatively low bandwidth) serial link. The comms and beacon nodes relay command queue information and state data through the radio node.

• /monitor: detects anomalies in incoming sensor measurements and issues alerts, error codes and abort commands in emergency situations.
• /estimation: contains the filters, smoothers, and estimators that are used to process sensor measurements and generate state estimates (e.g. first-order low-pass filter, or extended Kalman filter).

• /scheduler: manages the command queue that contains high-level commands. When execution begins, this node advances through the queue passing the commands to the guidance node as they are completed.

• /guidance: implements the control strategies that convert high-level commands to low-level reference signals (implementation of outer-loop controllers). Some example controllers include go-to-depth, go-to-roll, comms-stance, and sawtooth.

• /control: implements the low-level control. Reference signals that can be passed to the control node include depth rate, turn rate, pitch angle, etc. Some of the reference values that can be regulated are actually looped around other controllers. At the lowest level, the control node produces actuator signals for:
  – internal moving mass linear position
  – internal moving mass rotational position
  – inflator solenoid valve position
  – deflator solenoid valve position
  – drop weight solenoid on/off
  – LED array on/off

Currently all feedback control is accomplished with PID controllers.

• /sensorData: aggregates sensor measurements (e.g. from the glider’s input/output board, GPS unit, AHRS unit, etc.) and produces a publishes a sensor data topic.

• /datacollect: subscribes to sensor data, state estimates, and control status which are written to respective log files.

• /GPS: Since the GPS is a sensor this node only publishes incoming data. This node handles the processing of GPS information output by the gpsd daemon.

• /EposManager: controllers accept commands for the motor positions as well as produce measurements of those positions. Hence the node both listens for commands (sent from control node) as well as publishes measurements (received by sensor node). Third party software is used for this node.

• /IMU: Since the IMU is a sensor this node only publishes incoming data. Third party software is used for this node.

• /beacon: serves to relay communication between the beaconGUI and the glider via the radio node, and to keep track of the beacon location (via GPS node) and respond appropriately when the glider pings the beacon during LBL ranging.
9.5 Graphical User Interfaces

During benchtop testing, when the glider is connected to the ground station via an ethernet cable, the benchtestGUI node is used to display glider sensor data, status information and accept commands from the operator. Figure 9.3 shows the commands and display tabs that are available to the operator using the benchtestGUI. Similarly, during field testing, radio and acoustic communication is managed using the beaconGUI interface. This GUI is used for assembling/editing the command queue, displaying state data transmitted by the glider, visualizing heartbeat information sent from the glider, visualizing beacon/glider health status and also issuing emergency abort commands if necessary. The interface available with radio communication is shown in Figure 9.4, while the interface available with acoustic communication is shown in Figure 9.5.

For a detailed discussion of the software see the Developer’s Manual [35] and the User’s Manual [36].

9.6 Development Tools

The glider’s software was developed using the QT Creator and QT Designer. Cmake was used as the fundamental make tool. Doxygen and LaTeX were used to produce documentation. RabbitVCS was used for version control.
Figure 9.4: Screenshot of the beaconGUI interface with radio communication

Figure 9.5: Screenshot of the beaconGUI interface with acoustic communication
Chapter 10

Testing

10.1 Buoyancy Engine Feasibility

To determine the feasibility of the pneumatic buoyancy engine concept, and to test a number of components selected, a prototype buoyancy control system was fabricated. The prototype consisted of a large scuba tank, a pressure vessel housing on/off type solenoid valves and bladder pressure and depth sensors, as well as flooded chamber that housed the elastomeric bladder, see Figure 10.1. The rig was trimmed to remain vertical.

![Figure 10.1: Feasibility test of the pneumatic buoyancy engine](image)

Tests of this system were conducted in January 2012 at War Memorial Swimming Pool on Virginia Tech’s campus. In these experiments a bang-bang controller was designed to maintain a desired depth within some deadzone. The pneumatic system was able to reliably make the rig ascend and descend. However, finely regulating the bladder pressure was made difficult due to the coarse pressure changes provided by the on/off type solenoids. The influence of the inflator on the bladder pressure was disproportionately greater than that of the deflator. This test motivated revisiting the valve sizing and selection approach. As a
result of further analysis, new proportionally controlled valves were selected. The diameter of the tubing and valve orifice in the deflation assembly was increased to improve the deflation rate.

### 10.2 Depth Rating Validation

In August 2012, the glider was tested in Lake Washington by deploying it with a tether to a depth of 25 m (see Figure 10.2). The purpose of the test was to verify the bladder is capable of inflating at this depth, and also to confirm the vehicle is properly waterproofed.

![Glider in tank for trimming at the University of Washington](image1)

![Glider being launched from a sailboat](image2)

![Tethered glider in Lake Washington](image3)

Figure 10.2: Lake Washington Depth Rating Test

### 10.3 Sawtooth Glides in Pool

A series of pool tests culminated with the glider performing a sawtooth glides across War Memorial Swimming Pool in March 2013 (as shown in Figure 10.3(a)). The glider employed a modified bladder that used a fabric covering to restrict expansion beyond some volume and allow higher pressures (up to 5 psi). This resulted in faster deflation rates and made the bladder less susceptible to small changes in ambient pressure (as might occur from the vehicle pitching or changing depth). The data from this test is given in Figure 10.3(b). The
glider was able to traverse the width of the diving well (≈ 20 m) in about 2.5 minutes. It is important to note that the pool is much too shallow for the glider to establish a steady glide and the sawtooth here consists of inefficient transient motions. The bladder pressure oscillated near the surface and further gain tuning was required. The target depth of the glider during the sawtooth (black dashed line, top right subplot in Figure 10.3(b)) was overshot by about 1.5 m. Because of the limited depth of the swimming pool, the moving mass was only able to travel about 6 in. before the glider changed its direction. The associated descent and ascent pitch attitudes were about ±60 deg.

(a) Motion of glider across swimming pool

(b) Corresponding data

Figure 10.3: Pool sawtooth test
10.4 Buoyancy Engine Performance

Extensive testing of the buoyancy engine was conducted at Claytor Lake, VA. Figure 10.4 shows a series of dives to target depths of 15 m then to 20 m conducted in September 2013. In these experiments the inflator command was saturated to 30% to prevent excessively fast inflations. Dives to 20 m had about 3-4 m of overshoot and resulted in a change in tank pressure of about 70 psi. These experiments demonstrated the feasibility and reliability of the pneumatic buoyancy engine concept.

![Graphs showing bladder pressure, depth, command, and tank pressure over time.](image)

Figure 10.4: Buoyancy engine tests at Claytor Lake, VA

10.5 Trim and Gain Tuning

The goal of “trimming” the glider is to achieve a desired weight, buoyancy, and balance that meets performance and stability requirements. This requires carefully designing the vehicle for a range of center of buoyancy (CB) and center of gravity (CG) positions that meet these requirements. Estimating the mass properties and locations of all components in the vehicle during the detailed design and fabrication stage is essential to minimizing the amount of trim adjustments required once the vehicle is assembled. (Ideally, this also includes a sensitivity analysis that ensures the design is robust to small deviations in the expected values.)

The glider was trimmed for a number of operating conditions: the communication ("comms") stance (where the tail sticks above the water to improve reception), a desired depth rate and pitch attitude during ascents or descents, and the ability to roll completely around with one revolution of the moving mass. (Trimming is also required whenever a major component is changed or when moving from fresh water to sea water and vice versa.)
However, uncertainty in the design and fabrication process may require adjustments to be made. To determine the required adjustments in trim one may either attempt to measure the mass properties directly and compute the required adjustments, or make changes in the field (or in a trimming tank) based on observations of the vehicle’s motion.

The mass properties of the vehicle can be measured directly if appropriate facilities are available. The buoyancy of the entire vehicle can be measured by submersion (observing the change in water level). Weight can be measured by placing the glider on a scale. To determine the CG and CB locations (along the longitudinal axis) the glider can be hung from two or more spring scales both inside and outside a trimming tank. The position of the CG and CB locations in the vertical plane is more difficult to measure using this approach. The inertia of the glider can be estimated using the bifilar or compound pendulum technique.

Trim adjustments may be made by: re-arranging existing components (when permissible), designing new replacement parts with different weight/buoyancy properties, using the moving mass and buoyancy engine for trim, or adding/removing buoyancy and weight from the vehicle. In this latter approach, often foam is used to add buoyancy and a dense metal is used to add weight. Syntactic foam is a type of foam frequently used for this purpose, however it is expensive and rather brittle. (It must be encapsulated with epoxy and cured once it has been shaped appropriately. This is a lengthy process that does not lend itself well to iterative adjustments.) Instead, a more machinable, inexpensive extruded polysyrene insulation foam may be used (model: Owens Corning Foamular 1000, with a compressive strength of 100 psi). Lead is often used as a trim weight, however tungsten powder is an alternative that is safer to handle and can be mixed with epoxy to form a liquid which can be poured into a mold that cures to a similar density as lead. This makes it easier to form the trim weights into irregular shapes. It is important to design accessible but unobtrusive locations along the vehicle’s body that can accommodate additional trim foam and weight. Ideally, any trim weights or foam should reside underneath the fairings such that it does not disturb the flow around the vehicle. Trimming is an iterative process that often requires a compromise between competing objectives. In the following we discuss a few such objectives.

10.5.1 Comms Stance Trimming

The communication “comms” stance (see Figure 10.13) is used to extend the tail and improve GPS and radio reception. This desired attitude suggests making the glider nose-heavy (i.e. moving the internal mass forward) and and very buoyant (to raise the tail above the water.) However, because the bladder is located near the front of the vehicle, increases in buoyancy tend to pitch the nose up and reduce the angle of the tail. Thus, there is a trade-off between these two objectives.

If the moving mass alone does not pitch the vehicle down adequately enough then the buoyancy and weight must be re-distributed such that the nose is heavier and the tail is more buoyant. While this may improve the comms stance, it will bias the range of attainable pitch attitudes towards steeper nose-down angles.

Through testing with the VTUG, we found that the best comms stance was achieved when
the vehicle was just slightly positively buoyant. However, this configuration was susceptible to strong pitch-depth coupling. See Sections 2.3.3 and 2.3.4 for a detailed discussion.

These issues may be mitigated by carefully tuning the bladder pressure controller. We found that the gains required to maintain this sensitive comms stance were much different than those required during sawtooth glides. Thus the functionality of scheduling new gains during a comms stance was implemented in the software.

10.5.2 Roll Trimming

The ideal case for roll trim is when the rigid body CG and CB are aligned along the vehicle longitudinal axis (see Section 3.4 for a detailed discussion). It is desirable to adjust the bottom heaviness without perturbing the pitch trim (i.e. to only adjust the CG/CB locations in the vertical plane). To achieve this, neutrally buoyant pairs of weight and foam were placed at approximately the same longitudinal position (see Figure 10.6).

It is preferable for a shallow water glider with unlimited roll control to “flip over” on the surface, when transitioning from an ascent to a descent, rather than while it is underway. This maximizes the time available for the glider to maintain a steady glide. However, there are several challenges associated with flipping over on the surface. As the glider begins to roll its wings on the surface there is a discontinuous change in buoyancy as one wing leaves the water. The result is that the buoyancy of the other (submerged) wing is not balanced and therefore a roll moment acts to restore the vehicle to the level orientation (see Fig. 10.7). This restoring moment can be significant since the moment arm of the wings’ center of buoyancy is typically several times that of the moving mass. (Note that vertical stabilizers placed perpendicular to the wings reduce this effect.) The VTUG initially employed thick hydrofoil wings, however we found that these were too buoyant and prevented the vehicle from rolling on the surface. Flooded wings that were partially hollow were fabricated, however only a small improvement in roll performance was observed. Last, flat plate steel wings with minimal buoyancy were constructed and these wings enabled the glider to complete full 360 deg. rolls.
Figure 10.6: Pairs of neutrally buoyant trim foam (larger pink blocks) and trim weight (smaller black pieces) were used to adjust the position of the CG and CB in the vertical plane (only).

Another factor that was found to affect the ability of the glider to roll was the drag due to wind acting on the wings. A wing extending straight above the water has a relatively large surface area and the resulting drag forces may be significant. On calm days it was observed that by rolling clockwise, or counter-clockwise, the wings would act as paddles to change the heading of the vehicle. This effect may be exploited to re-orient the glider prior to initiating a dive.

(a) At depth the wings are submerged and do not induce a rolling moment

(b) On the surface there is a discontinuous change in wing buoyancy causing a restoring moment that works to keep the glider level

Figure 10.7: Effect of wing buoyancy on rolling moments
10.5.3 Pitch Trimming

As mentioned previously, the main objectives of pitch trimming is to ensure the moving mass is capable of achieving a desired range of pitch attitudes during ascents, descents, and during the comms stance. Three locations on the vehicle were used to adjust the pitch trim: the nose, the tail, and the flooded coupler section containing the buoyancy engine.

One way to trim for descent pitch attitude is to perform tethered “drop tests” in which the tethered glider is held negatively buoyant near the boat and iteratively released to about 10 m. Since the glider is not transitioning from a comms stance, the mass position and bladder pressure can be set before the glider is released to reduce the time it takes to achieve a steady glide. The descent attitude of the glider can be determined both qualitatively (in extreme cases) and quantitatively by reviewing attitude measurements. Based on these observations, trim adjustments can be made after each iteration. To reduce the number of test points, the pitch attitude can be observed with the mass at the limits of its range. For example, if the pitch attitude during a descent with the mass in its aftmost position is shallow (e.g. $\approx 0$ deg.), and then steep (e.g. $\approx 60$ deg.) with the mass in its foremost position, then it follows that intermediate positions will give intermediate pitch angles. This indicates that the trim of the vehicle is adequate for this condition and the problem then becomes to find the correct mass position for the desired intermediate pitch angle. A similar approach can be used to trim for the ascent pitch attitude however this is more involved because it requires performing a sawtooth dive and analyzing the data from the ascent.

In practice, we found trimming the vehicle in pitch was very difficult. This can be attributed to at least three factors: the tank becomes lighter during a trimming campaign, the bladder pressure has a large influence on the pitch attitude, and the glider’s attitude is sensitive to small changes in trim because of the small bottom heaviness. Although these factors were considered during the detailed process, their effect was underestimated and post-production changes to the design altered the desired balance of the vehicle. While for some specific configurations we could trim the pitch of the vehicle as desired, this configuration became obsolete after a few dives due to the change in the weight of the tank.

10.5.4 Bladder Pressure Gain Tuning

It is clear from the discussion in Section 2 that the correct flow valve position is a function of the depth rate, and may also depend on the bladder pressure command. (Recall the deflator valve is more effective when the bladder pressure is higher.) Extensive testing tuning the gains of the proportional-integral-derivative (PID) bladder pressure control loops showed that a single set of gains does not perform well for varying depth rates and bladder pressures. This lead to the development of a separate set of gains for the comms stance (see Section 10.5.1). One way of mitigating the challenges associated with bladder pressure gain tuning is to limit the bladder’s operation to being fully deflated during descents and a fully inflated bladder pressure during ascents. This reduces the number of operating conditions that must be tuned for at the cost of flexibility with the available buoyancy commands. Another approach, recommended for future work, is to design a model-based control law
that captures coupling between bladder pressure, pitch, depth rate, and the required actuator commands.

10.6 LBL Ranging System: Planar Motion Test

The acoustic LBL ranging system was tested in Claytor Lake, VA in November 2014. The objective of the experiment was to assess the performance of the system under ideal conditions in the operating environment (see Figure 10.8(a)).

The experiment consisted of deploying three LBL nodes (each on their own rafts) with transducers submerged to 15 ft. The glider’s motion was emulated by keeping the glider aboard a moving boat and submerging the transducer to 5 ft. Keeping the glider aboard the boat allowed us to record the true GPS position of the glider relative to all of the deployed beacons (each logging their own GPS position as well). Thus the accuracy of the LBL ranging system and our position estimates could be determined. For the purposes of this experiment only static estimation algorithms were implemented. All of the data collected was processed offline. The host vessel (and glider) were moved in a lawn mower pattern to cover a rectangular region in the vicinity of the deployed beacons as showed in Figure 10.8(b).

The full track of the boat was divided into two separate tracks for analysis. The first track (Track 1) comprised of the vertical (North-South) lawn mower paths (also called “swaths”). The second track (Track 2) comprised of horizontal (East-West) swaths. Figure 10.9 shows the estimation results for Track 1.

Figure 10.9(a) shows a comparison of the modem ranges and the true ranges obtained by using the GPS data from the nodes and glider. It can be seen that they both match quite well.

Figure 10.9(b) shows a histogram of pings with, and without, a successful response from the nodes. For the duration of the run, 134 successful pings were expected from each node. However, only 101, 102, and 98 successful pings were obtained from Nodes 1, 2, and 3 respectively. This shows some consistency in the performance of all the three nodes. However, the causes of the lost pings cannot be easily identified. Since extensive testing was done to ensure the robustness of the hardware and software in the lab setting it is likely that this is due to the properties of the underwater acoustic environment rather than a hardware or software error.

Figure 10.9(c) shows the ping locations on the glider path. It was observed that three consecutive successful pings in the 15 s LBL range cycle were not always obtained. Most of the time two successful pings in the 15 s cycle were obtained. Whenever three consecutive range measurements were available localization could be performed with higher accuracy. However, two successive range measurements were sufficient to localize the glider position because:

1. Only planar estimation is being performed, since the depth of the transducers is known \textit{a priori}. 
2. Due to the availability of GPS data of the boat, one of the solutions obtained via the algebraic solution could be eliminated as knowledge of the baseline was available.

Figure 10.9(d) shows the results of implementing the least squares [37] and Bancroft algorithms [38]. The estimates with three range measurements and those with two are combined together in the plot shown. As expected, their performances are similar. The results of the run are detailed in Table 10.1.
Figure 10.9: Testing the LBL system with a GPS-equipped boat emulating the glider using North-South oriented swaths (Track 1)

Table 10.1: Summary of results from Track 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Duration of run</td>
<td>34 min</td>
</tr>
<tr>
<td>Number of expected pings per beacon</td>
<td>134</td>
</tr>
<tr>
<td>Avg. error* for LS with 3 range measurements</td>
<td>10 m</td>
</tr>
<tr>
<td>Avg. error for LS with 2 range measurements</td>
<td>13 m</td>
</tr>
<tr>
<td>Avg. error for Algebraic Solution with 3 range measurements</td>
<td>9 m</td>
</tr>
<tr>
<td>Avg. error for Algebraic Solution with 2 range measurements</td>
<td>12 m</td>
</tr>
</tbody>
</table>

*Error refers to the difference between range calculated using GPS, and range from WHOI Micro-Modems with an assumed speed of sound

Figure 10.10 shows the results for Track 2, which are further detailed in Table 10.2. The
results are comparable with Track 1.

Figure 10.10: Testing the LBL system with a GPS-equipped boat emulating the glider using East-West oriented swaths (Track 2)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Duration of run</td>
<td>22 min</td>
</tr>
<tr>
<td>Number of expected pings per beacon</td>
<td>86</td>
</tr>
<tr>
<td>Avg. error* for LS with 3 range measurements</td>
<td>12 m</td>
</tr>
<tr>
<td>Avg. error for LS with 2 range measurements</td>
<td>16 m</td>
</tr>
<tr>
<td>Avg. error for Algebraic Solution with 3 range measurements</td>
<td>12 m</td>
</tr>
<tr>
<td>Avg. error for Algebraic Solution with 2 range measurements</td>
<td>15 m</td>
</tr>
</tbody>
</table>

Table 10.2: Results of Track 2
10.7 LBL Ranging System: Vertical Motion Test

An experiment was conducted to determine the optimal depth, $d$, to which the acoustic buoys transducers should be deployed. While in some cases this optimal placement may be determined by analysis of the speed of sound profile and acoustic properties of the operating environment, we did not have the expertise or capabilities to conduct such a study. Instead we sought to determine this parameter experimentally. A test was conducted with the glider transducer removed from the vehicle and connected to a fourth 25m wet-mateable cable. The depth $d$ of the beacons transducers was incrementally varied from: 5m, 10m, 15m to 20m. For each configuration $d$ the glider transducer was also lower to a particular depth (1m, 5m, 10m, 15m, or 20m) for 5 minutes. The test setup is sketched in Figure 10.11.

The resulting percentage of successful pings (from each beacon) was recorded. The results are shown in Figure 10.12. Note that the glider transducer was deployed from the Ground Station vessel, thus it was nearly coincident with Ground Stations transducer.

Figure 10.11: Testing of ping success for various LBL network depths. For a given fixed depth of all the acoustic beacons’ transducers (shown in yellow) a transducer emulating the glider’s motion was held for 5 minutes at a range of depths (shown in red).

10.8 Sawtooth Glides with Acoustic Positioning

The first tests that utilized the LBL system with the glider underway were conducted in February 2015 (see Figure 10.13). The sawtooth glide consisted of a switching between pre-defined mass positions and bladder pressure set-points at a target depth (to change from ascent to descent). Because there was no active pitch and roll control the glider was unable to attain a steady level equilibrium glide. It was also observed during this test that the acoustic ranging immediately ceased as the glider pitched down and began its dive. This
Figure 10.12: Ping success percentage for various depths of the acoustic beacons’ transducers

may be due to the transducer being mounted in the nose of the vehicle, and the body of the vehicle interfering with the transducer’s operation.

In an attempt to improve the positioning of the glider while underway, the transducer was moved from its position in the front of the nose (aligned with the longitudinal axis) to the top of the hull as shown in Figure 10.14.

Subsequent tests were conducted with closed-loop attitude feedback control to move the internal mass, as shown in (see Figure 10.15). Unfortunately, the new transducer location did not improve the acoustic ranging performance (again, no pings were successful during the sawtooth glide). From Figure 10.15 it is clear that a steady shallow glide could not be established with the given trimming configuration, bladder pressure controller and moving mass actuators. We believe that the lack of successful pings during the sawtooth dive may be a result of the following factors: a) as the glider descends and ascends the body of the glider may occlude the transducer located in the nose, b) the moving mass motors and solenoid valves introduce significant noise into the LBL system, especially during sawtooth dives when they are most active, and c) the bladder is regulated by exhausting air that forms numerous bubbles in the vicinity of the transducer that may cause scattering of sound.
(a) Visualization of glide orientation with depth and time: a circle indicates a successful range measurement, circle’s color indicates range measurement from a unique beacon

(b) Corresponding data

Figure 10.13: Sawtooth glide test with acoustic positioning
Figure 10.14: Various mounting locations of the acoustic transducer
Figure 10.15: Sawtooth glide test with acoustic positioning and attitude control
Chapter 11

Conclusion

In testing a novel underwater glider design and a long baseline ranging system in a shallow water lake, we identified several challenges associated with key glider systems that were designed to make the glider faster and more maneuverable. For example, a pneumatic buoyancy engine was found to generate large attitude disturbances that challenged the vehicle’s attitude control system, a cylindrical moving mass actuator. Despite the challenges we encountered, the pneumatic buoyancy engine was able to reliably control the depth of the glider and provide large and rapid buoyancy changes. This type of buoyancy engine may be useful in applications requiring the vehicle to extract itself from a layer of debris after a prolonged period of being inactive on the sea-floor (e.g. if the vehicle were to passively collect measurements and periodically surface to transmit data). Field tests of the Virginia Tech Underwater Glider underscore the challenges involved in developing new motion control concepts. However, the experimental results also suggest that pneumatically powered glider may be an effective mobile sensing technology in shallow water and strong currents.
Bibliography


