Passive and Active Attitude Stabilization for a Tow-fish

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Abstract

High precision oceanographic sensing applications often require precisely regulated sensor platforms. This paper investigates the problem of stabilizing the longitudinal motion of a streamlined sensor platform, towed in a two stage arrangement, using servo-actuated tail fins and an internal moving mass actuator. The results suggest that, while tail fins can adequately regulate the attitude of such a platform at moderate speeds, a moving mass actuator can significantly improve disturbance rejection at lower speeds.

1 Introduction

Advances in sensor technology have opened exciting possibilities for ocean science. For example, acoustic Doppler current profilers (ADCPs) allow scientists to instantaneously measure all three components of fluid velocity throughout a column of water. By translating an ADCP along a horizontal path, one may obtain a complete velocity profile in a vertical plane. Because ADCPs drastically improve measurement efficiency, they make it feasible for scientists to scan the ocean for sparsely distributed oceanographic features.

One such application for an ADCP involves searching for and characterizing small-scale ocean turbulence [3]. Regions of strong turbulent mixing are sparsely distributed and time-varying, making localization difficult using conventional sampling methods. Using an ADCP, however, pockets of turbulence can be quickly identified. To accurately characterize ocean turbulence, the sensor's attitude be precisely regulated. A common way to isolate a sensor from wave disturbances and surface currents is to mount the sensor on an immersed, towed platform called a "tow-fish." This arrangement is adequate for many ocean science applications, however the oscillatory motion of the towing vessel is unavoidably transmitted to the tow-fish through the tether. It has been shown that a two stage towing arrangement can significantly attenuate sensor attitude disturbances due to ship motion [5, 10]. Such an assembly consists of a primary tether, connecting a depressor weight to the surface vessel, and a secondary tether linking the tow-fish to the depressor weight. A two tether configuration, as depicted in Figure 1, allows for independent adjustment of the tow-fish depth and the distance between the tow-fish and depressor weight using two winches aboard the towing vessel.

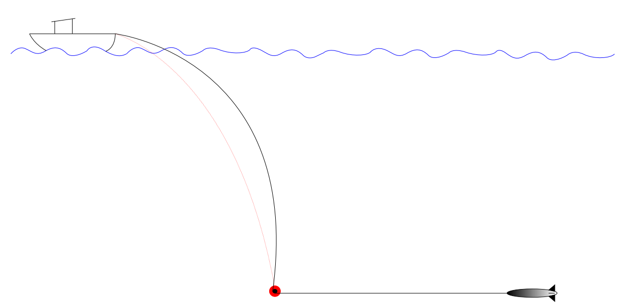

Figure 1: Two stage towed sensor platform.

Careful design can produce a tow-fish with adequate passive stability characteristics for a particular sea state and towing speed. If one wishes to use

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the vehicle over a range of sea states and towing speeds, then the tow-fish may also require active stabilization using, for example, servo-actuated tail fins. An adaptable sensor platform is particularly important for turbulence mapping, where one "trawls" for turbulent hot spots at a moderate speed and then samples these turbulent regions with greater resolution at a lower speed.

One problem that often arises for towed platforms is a steady-state attitude bias due to inertial or hydrodynamic asymmetries. This bias can be eliminated by deflecting the fin actuators, however this steady offset in the control increases the chance of actuator saturation when disturbances occur, particularly at low speed where large fin deflections are necessary to generate control moments. One solution to this problem is to include an internal, servo-actuated mass which can trim the vehicle's center of gravity as necessary. As a bonus, the moving mass could be used to augment the vehicle's disturbance rejection capability at low speed.

In Section 2, a dynamic model is presented for the longitudinal motion of a tow-fish with stern plane actuators and an internal moving mass. Section 3 describes linear control design results which verify that an internal moving mass can effectively augment the attitude control authority of a tow-fish at low speed. We conclude in Section 4.

2 Modeling

We present a standard model for the dynamics of a tow-fish, augmented with an internal moving mass actuator. Consider an immersed prolate, spheroidal body with total mass m equal to the mass of displaced fluid. Fix a right-handed coordinate frame in the spheroid principal axes with the 1-axis aligned with the spheroid axis of symmetry and the 2-axis pointing starboard. Let $\boldsymbol{\Omega} = [\Omega_1,\ \Omega_2,\ \Omega_3]^T$ and $\boldsymbol{v} = [v_1,\ v_2,\ v_3]^T$ represent the angular and translational velocity of the body with respect to inertial space, expressed in terms of this body-fixed coordinate frame. Suppose that all of the mass is uniformly distributed, except for a servo-actuated point mass $\bar{m}$ located at the point $\boldsymbol{r}_{\bar{m}}$. As described in [9], the system dynamics can be cast as a non-canonical Hamiltonian control system, in the special case where no non-conservative forces or torques act. More generally, one can include viscous and propulsive effects as exogenous forces.

As a first step toward a complete control design, we consider only the longitudinal tow-fish dynamics; equivalently, we assume that Ω_1, Ω_3, and v_2 are identically zero. We also assume that

$$\boldsymbol{r}_{\bar{m}} = \begin{pmatrix} r_{\bar{m}_1} \\ 0 \\ \gamma \end{pmatrix} \tag{1}$$

where γ is constant; the point mass is offset a distance γ below the vehicle's symmetry axis and it moves parallel to that axis, under the influence of a control force u_{int}.

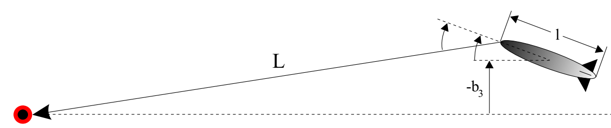

Figure 2: Planar model of the towed sensor platform.

We will assume that an inextensible, neutrally buoyant secondary cable of length L attaches the tow-fish to a depressor weight. The cable is attached to the nose of the tow-fish, a distance $l/2$ from the body coordinate origin. Rather than simulate the full two stage, tethered vehicle dynamics, we assume that the secondary cable exerts a constant force $\bar{F}$ directed toward the nominal depressor weight location, as shown in Figure 2. This assumption can be relaxed to account for depressor motion due to disturbances and to the dynamic interaction between the depressor and tow-fish. In this preliminary study, however, we simplify the problem in order to isolate the tow-fish dynamics from the rest of the system.

The tow-fish dynamics do not depend on the vehicle's surge position however, to determine the line of action of the towing force, one needs to know the heave position, which we denote b_3. The kinematic equations of interest are therefore

$$\begin{aligned} \dot{\theta} &= \Omega_2 \\ \dot{b}_3 &= -v_1 \sin\theta + v_3 \cos\theta, \end{aligned} \tag{2}$$

where θ represents the pitch angle. Define the generalized inertia matrix $\mathbb{I}(r_{\bar{m}_1}) =$

$$\begin{pmatrix} J+\bar{m}(\gamma^2+r_{\bar{m}_1}^2) & 0 & -\bar{m}r_{\bar{m}_1} & \bar{m}\gamma \\ 0 & m_1 & 0 & \bar{m} \\ -\bar{m}r_{\bar{m}_1} & 0 & m_3 & 0 \\ \bar{m}\gamma & \bar{m} & 0 & \bar{m} \end{pmatrix},$$

where J, m_1, and m_3 include added inertia and added mass, as appropriate. (See [2], for example.) The dynamic equations are

$$\begin{pmatrix} \dot{\Omega}_2 \\ \dot{v}_1 \\ \dot{v}_3 \\ \ddot{r}_{m_1} \end{pmatrix} = \mathbb{I}(r_{\bar{m}_1})^{-1}$$

$$\left\{ \begin{pmatrix} -\bar{m}r_{\bar{m}_1}(2\dot{r}_{m_1}+v_1)\Omega_2+(m_3-m_1)v_1v_3 \\ -(m_3v_3-\bar{m}r_{\bar{m}_1}\Omega_2)\Omega_2 \\ (m_1v_1+2\bar{m}\dot{r}_{\bar{m}_1})\Omega_2 \\ \bar{m}(r_{\bar{m}_1}\Omega_2-v_3)\Omega_2 \end{pmatrix} + \right.$$

$$\left. \begin{pmatrix} -\bar{m}g(r_{\bar{m}_1}\cos\theta+\gamma\sin\theta)+\mathcal{T}_{2_v}+\frac{1}{2}\bar{F}l\sin\beta \\ \mathcal{F}_{1_v}+\bar{F}\cos\beta \\ \mathcal{F}_{3_v}+\bar{F}\sin\beta \\ u_{\mathrm{int}} \end{pmatrix} \right\}$$

where $\mathcal{T}_{2_v}$, $\mathcal{F}_{1_v}$, and $\mathcal{F}_{3_v}$ are components of the torque and force due to viscous effects:

$$\begin{aligned} \mathcal{T}_{2_v} &= \mathcal{T}_{2_{\mathrm{body}}}+\mathcal{T}_{2_{\mathrm{tail}}} \\ \mathcal{F}_{1_v} &= \mathcal{F}_{1_{\mathrm{body}}}+\mathcal{F}_{1_{\mathrm{tail}}} \\ \mathcal{F}_{3_v} &= \mathcal{F}_{3_{\mathrm{body}}}+\mathcal{F}_{3_{\mathrm{tail}}}. \end{aligned}$$

The angle between the line of action of the towing force and the body 1-axis is denoted β. (See Figure 2.)

We assume that the only viscous torque is due to the moment provided by the lift force on the stern planes, that is, $\mathcal{T}_{2_{\mathrm{body}}}=0$. This assumption is consistent with the expectation that the dominant source of hydrodynamic moment over the body arises from the potential flow model of the hydrodynamics, and is therefore captured in the added mass and inertia terms [8].

The viscous forces acting on the body are

$$\begin{pmatrix} \mathcal{F}_{1_{\mathrm{body}}} \\ \mathcal{F}_{3_{\mathrm{body}}} \end{pmatrix} = \begin{pmatrix} \cos\alpha & -\sin\alpha \\ \sin\alpha & \cos\alpha \end{pmatrix} \begin{pmatrix} \mathcal{D} \\ \mathcal{L} \end{pmatrix}$$

where $\mathcal{D}$ and $\mathcal{L}$ represent the drag and lift force and $\alpha=\tan^{-1}\left(\frac{v_3}{v_1}\right)$ is the "angle of attack." Standard modeling assumptions suggest defining

$$\begin{pmatrix} \mathcal{D} \\ \mathcal{L} \end{pmatrix} = -\begin{pmatrix} C_{D_0}+\epsilon(C_{L_\alpha}\alpha)^2 \\ C_{L_\alpha}\alpha \end{pmatrix} P_{\mathrm{dyn}}S_{\mathrm{body}},$$

where $C_{D_0}>0$ is the drag coefficient at zero angle of attack, $C_{L_\alpha}>0$ is the slope of the lift coefficient, and ϵ is a "correction factor" to make empirical data consistent with theory. The term $P_{\mathrm{dyn}}=\frac{1}{2}\rho V^2$ is the dynamic pressure defined in terms of the fluid density ρ and the tow-fish speed $V=\sqrt{v_1^2+v_3^2}$. The term S_{body} represents a reference area, typically the body's frontal or planform area.

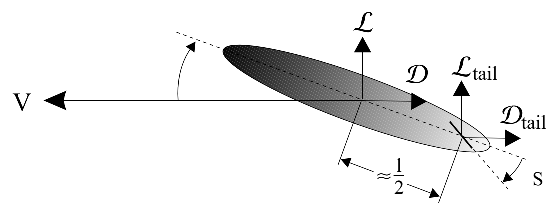

Figure 3: Sketch of the viscous forces.

The torques and forces acting on the body due to the stern planes depend on their effective angle of attack, the sum of α and the fin deflection angle δS, which we treat as a control. Specifically,

$$\begin{pmatrix} \mathcal{F}_{1_{\mathrm{tail}}} \\ \mathcal{F}_{3_{\mathrm{tail}}} \end{pmatrix} = \begin{pmatrix} \cos\alpha & -\sin\alpha \\ \sin\alpha & \cos\alpha \end{pmatrix} \begin{pmatrix} \mathcal{D}_{\mathrm{tail}} \\ \mathcal{L}_{\mathrm{tail}} \end{pmatrix}$$

where

$$\begin{pmatrix} \mathcal{D}_{\mathrm{tail}} \\ \mathcal{L}_{\mathrm{tail}} \end{pmatrix} = -\begin{pmatrix} \tilde{C}_{D_0}+\tilde{\epsilon}(\tilde{C}_{L_\alpha}(\alpha+\delta S))^2 \\ \tilde{C}_{L_\alpha}(\alpha+\delta S) \end{pmatrix} P_{\mathrm{dyn}}S_{\mathrm{tail}}.$$

The various parameters are defined as above, but are referred to the stern planes rather than the body. We neglect the pure hydrodynamic moment generated by the stern planes. Thus, the only viscous moment acting on the body is due to lift and drag on the stern planes,

$$\mathcal{T}_{2_{\mathrm{tail}}} = -\left(\left(\frac{1}{2}l\cos\alpha\right)\tilde{C}_{L_\alpha}(\alpha+\delta S)+\right.$$

$$\left.\left(\frac{1}{2}l\sin\alpha\right)\left(\tilde{C}_{D_0}+\tilde{\epsilon}(\tilde{C}_{L_\alpha}(\alpha+\delta S))^2\right)\right)P_{\mathrm{dyn}}S_{\mathrm{tail}}.$$

3 Stabilization at Low Speed

This section describes the results of preliminary regulator design for the tow-fish described in Section 2. The parameters used in the simulations correspond to a 4 : 1 spheroid which is 2 meters long. It is assumed that the tow-fish is neutrally buoyant, that 80% of its mass is uniformly distributed throughout the spheroid, and that the remaining 20% is concentrated at a point whose position is given by (1) with $\gamma = \frac{1}{8}$ meter. A critically damped spring-damper (k = 5 kN/m and b = 1 kN/(m/s)) is included in the model to restrict excursions of the point mass; the spring-damper force is included in the total control force applied to the point mass:

$$u_{\text{int}} = -b\dot{r}_{\bar{m}_1} - kr_{\bar{m}_1} + u_{\bar{m}},$$

where $u_{\bar{m}}$ remains to be determined.

Lift and drag coefficients for the body were computed using data presented in [4]. Lift and drag coefficients for the two stern planes are based on standard, symmetric airfoil data, assuming an aspect ratio of 3 and a length of $\frac{1}{4}$ meter. We let L = 50 meters.

Because the nominal motion of the tow-fish is a dynamic equilibrium, and disturbances are expected to be small, it is reasonable to consider the dynamics linearized about the equilibrium. Let $\boldsymbol{x} = [b_3,\ \theta,\ \Omega_2,\ v_1 - \bar{V},\ v_3,\ r_{\bar{m}_1},\ \dot{r}_{\bar{m}_1}]^T$, where $\bar{V}$ is the constant, nominal tow-fish speed, and let $\boldsymbol{u} = [\delta S,\ u_{\bar{m}}]^T$. The linearized dynamics are

$$\dot{\boldsymbol{x}} = \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}\boldsymbol{u}$$

where $\boldsymbol{A}$ and $\boldsymbol{B}$ are the appropriate Jacobian matrices evaluated at $(\boldsymbol{x}, \boldsymbol{u}) = (\boldsymbol{0}, \boldsymbol{0})$. A linear-quadratic regulator was designed to minimize the cost

$$J = \int_0^\infty \left(\frac{1}{2}\tilde{\boldsymbol{x}} \cdot \boldsymbol{Q}\tilde{\boldsymbol{x}} + \frac{1}{2}\tilde{\boldsymbol{u}} \cdot \boldsymbol{R}\tilde{\boldsymbol{u}} \right) dt, \qquad (3)$$

where $\tilde{\boldsymbol{x}}$ and $\tilde{\boldsymbol{u}}$ are the non-dimensional state and control obtained by normalizing mass, length, and time by m, l, and $l/\bar{V}$, respectively. The penalty matrices were chosen as

$$\begin{aligned} \boldsymbol{Q} &= \text{diag}(1000, 1000, 0, 0, 0, 0, 0), \quad \text{and} \\ \boldsymbol{R} &= \text{diag}(100, 1) \end{aligned}$$

in order to heavily penalize excursions in depth and pitch, as well as large stern plane deflections.

We consider two cases, a relatively a high nominal towing speed $\bar{V}$ and a relatively low towing speed. Note that if one were to solve the LQR problem for arbitrary $\bar{V}$, one would obtain a state feedback gain which is parameterized (or "scheduled") by $\bar{V}$. Alternatively, one might solve the problem for a variety of speeds and interpolate the resulting set of gains according to the actual towing speed. While good closed-loop performance would not be guaranteed in cases where the towing speed varies rapidly, the gain scheduling approach could be quite effective for the proposed application, since the nominal towing speed remains constant.

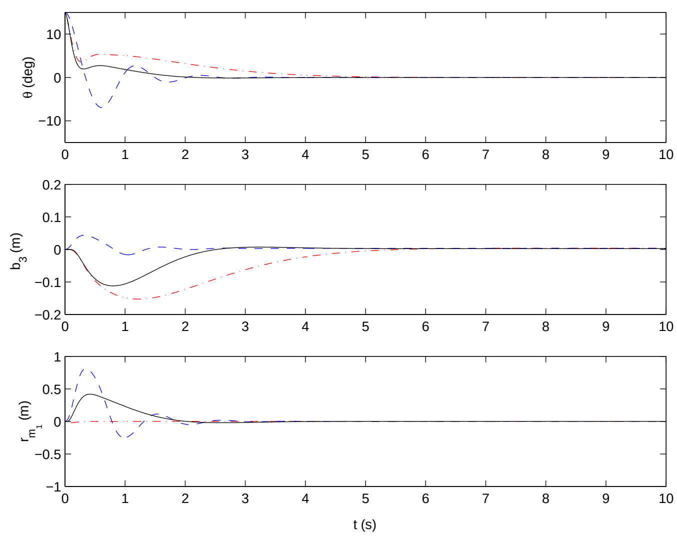

Figure 4: Attitude response to a pitch disturbance ($\bar{V}$ = 3 m/s).

Consider first the case where the tow-fish moves at a nominal speed $\bar{V}$ = 3 m/s, or roughly 6 knots. This is a moderately high speed which might correspond to "search mode," in which an ocean scientist seeks an oceanographic feature of interest. At this speed, the stern planes are quite effective at regulating the tow-fish attitude. Figure 4 presents the results of three simulations involving a 15 degree initial pitch angle. This initial condition might result, for example, from an impulsive pitch disturbance. In Figure 4 and all figures which follow,

- dashed-dotted lines correspond to $u_{\bar{m}} = 0$,

- dashed lines represent the case where the stern planes remain fixed ($\delta S = 0$), and
- solid lines represent the case where both actuators are used.

Rather than simply use the gain matrix computed using both actuators, the LQR problem was solved separately for each case to ensure that the three different system responses truly minimize (3), given the available control inputs.

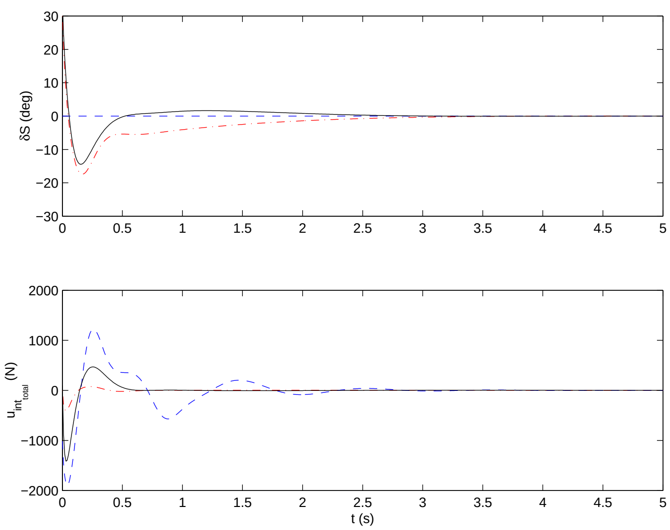

Figure 5: Input response to a pitch disturbance ($\bar{V} = 3$ m/s).

In the simulation shown in Figure 4, the response due to combined actuation is superior to the response using either the stern planes or the moving mass alone. The control effort required to actuate the moving mass, however, far exceeds the capability of any physically reasonable actuator; see Figure 5. Fortunately, at this relatively high speed, the stern planes alone are quite effective at rejecting the disturbance. Further tuning of the stern plane gains could presumably yield even better performance. At lower speeds, this is not the case.

Suppose the tow-fish moves at a nominal speed $\bar{V} = 0.5$ m/s (or roughly 1 knot). This speed is relatively low and might correspond to a sampling mode in which the tow-fish is used to make high resolution measurements of a particular oceanographic feature. In Figure 6, it is clear that the stern plane acting alone does not adequately reject the disturbance, either in pitch or in depth. The moving mass actuator is much more effective at this low speed. Moreover, the control effort required to actuate the moving mass is comparatively small, as can be seen in Figure 7. It should also be noted that the stern plane deflections at this lower speed are much larger, making it more likely that these actuators would saturate.

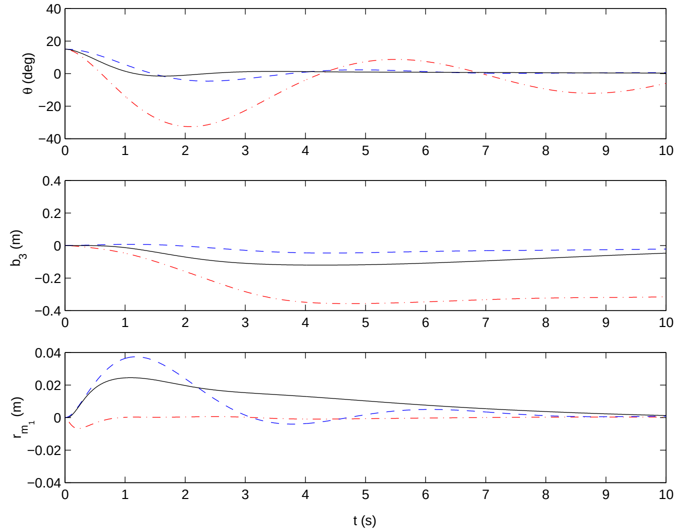

Figure 6: Attitude response to a pitch disturbance ($\bar{V} = 0.5$ m/s).

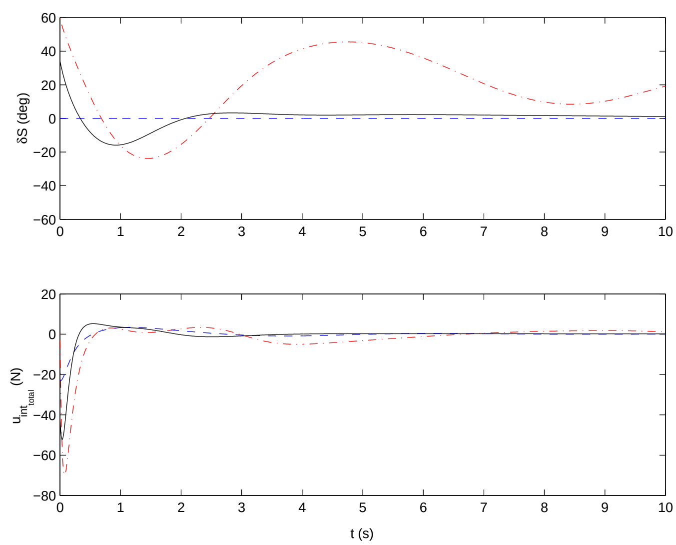

Figure 7: Input response to a pitch disturbance ($\bar{V} = 0.5$ m/s).

4 Conclusions and Continuing Work

This paper describes preliminary control design for a passively and actively stabilized towed sensor platform for measuring ocean turbulence. The tow-fish will be the final component in a two stage towing system. The tow-fish features stern plane actuators, for active disturbance rejection, and a servo-actuated internal mass to eliminate attitude biases due to inertial or hydrodynamic asymmetries. It is anticipated that the sensor platform will be useful in other applications, including sonar and video imaging.

For the turbulence mapping application, the tow-fish must operate over a range of towing speeds. Linear control design and analysis results suggest that tail fin actuators are effective at moderate towing speeds but are subject to saturation at low speed. By using the internal moving mass to augment the disturbance rejection capability at low speed, one may better regulate the vehicle attitude while reducing the likelihood of fin saturation.

These results are preliminary, but they motivate a continuing investigation of the problem of passive and active attitude stabilization for a tow-fish. Several issues remain to be addressed. The issue of fin actuator saturation can be studied in more detail by including this effect in the model throughout the control design process. The problem of integrator windup due to fin saturation can be addressed using a technique such as the method of [6], which was demonstrated for an unmanned underwater vehicle in [1]. It is perhaps even more important to consider the possibility of saturation for the moving mass actuator. Such an actuator might consist of a DC motor which drives a large mass along a lead screw. Certainly the actuator range of motion would be restricted. Also, the magnitude of the reaction force generated by the moving mass would be limited by friction in the assembly.

The two tether arrangement depicted in Figure 1 raises questions about hydrodynamic interactions between the tethers. Such interactions must be minimized to ensure that the tow-fish dynamics will not be adversely affected. An overview of modeling for immersed cables is presented in [7]. In addition to these issues, future work will address the effect of depressor weight oscillation due to surface waves, as modeled in [2], and the dynamic interaction between the tow-fish and depressor. Also, the control design and analysis will be extended to three dimensions by constructing an independent lateral-directional controller for the linearized tow-fish dynamics. Additional actuation for the lateral-directional system will include a rudder and lateral freedom for the moving mass.

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