MODELING OF UNDERWATER MANIPULATOR HYDRODYNAMICS WITH APPLICATION TO THE COORDINATED CONTROL OF AN ARM/VEHICLE SYSTEM

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FOR THE DEGREE OF
DOCTOR OF PHILOSOPHY

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August 1995
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Abstract

For users of unmanned underwater vehicles, manipulators have become a valuable tool in performing a wide variety of tasks, from scientific sampling in the ocean to maintenance and construction of underwater structures. In some situations, the addition of manipulators to an underwater vehicle poses significant control challenges due to the hydrodynamic interactions between the arm and the vehicle: When the arm is moved while the vehicle is hovering in open water, the large hydrodynamic forces acting on the arm can cause the vehicle to “swim” away from its desired station, degrading the operator’s ability to position accurately the manipulator end point.

To compensate for this dynamic coupling, the nature of the hydrodynamic forces acting on the manipulator must be well understood. This dissertation describes efforts to characterize the fundamental hydrodynamics of a single-link arm undergoing typical robotic slews. The product of this characterization is a new accurate real-time-implementable model of the hydrodynamic forces and torques acting on a circular cylinder (length/diameter = 9.1) swinging rapidly about one end through moderate angles (< 120 degrees) in a start-stop fashion. This model was developed through a balanced combination of theoretical development and experimentation. A two-dimensional potential-flow-theory analysis for a cylinder undergoing unsteady motions formed the starting basis for the hydrodynamic model. This analysis was extended semi-empirically to three dimensions using a strip-theory methodology. Valuable insight into the behavior of the hydrodynamic forces was gained through experimental flow visualization and direct measurement of forces at locations along the span of the cylinder. State-dependent drag and added-mass coefficients were identified from force and torque measurements using a strategy developed in this work. This research represents the first experimental investigation of the hydrodynamic forces acting on underwater manipulators.

As an example application of the new hydrodynamic model, the model was used to predict the arm/vehicle interaction forces for a system consisting of a free-swimming vehicle with a movable single-link arm. With this model of the arm/vehicle interaction forces, a coordinated arm/vehicle control strategy was developed. To demonstrate the effectiveness of this controller, experiments were conducted using the OTTER vehicle at the Monterey Bay Aquarium Research Institute (MBARI). Under this model-based approach, interaction forces acting on the vehicle due to arm motion were predicted and fed into the existing
vehicle position feedback controller. Using this method, vehicle station-keeping capability was greatly enhanced: Errors at the manipulator end-point were reduced by a factor of over six when compared to results when no control was applied to the vehicle, and by a factor of 2.5 when compared to results using only position feedback for controlling the vehicle subsystem. Using the coordinated-control strategy, arm end-point settling times were reduced by a factor of three when compared to those obtained with arm and vehicle position feedback control alone. These dramatic performance improvements were obtained with only a five percent increase in the total applied thrust.
To Amber
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The research presented in this dissertation has benefited greatly from the associations which I have enjoyed during my time as a graduate student in the Aerospace Robotics Laboratory (ARL) at Stanford University. I am fortunate to have as members of my reading committee individuals who have been instrumental in my development as a researcher and have played influential roles in the progress and completion of this dissertation. My principal advisor, Professor Stephen Rock, has provided vital direction and encouragement at the critical stages of my research. I am thankful for his example as a teacher and research advisor. I am grateful to Professor Robert Cannon for establishing and nurturing, in the ARL, a first-rate environment for performing experimental research. I appreciate the guidance of Professor Dan DeBra, who was influential in my decision to come Stanford and was most encouraging during my transition to being a graduate student again. I am thankful for the efforts of Mike Lee, my advisor at MBARI, who has provided many opportunities for hands-on engineering and practical experimental research, from which I have learned much.

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Chapter 1

Introduction

This dissertation presents theoretical and experimental research on the modeling of hydrodynamic forces acting on underwater manipulators, and its specific application to the high-performance, coordinated control of an underwater arm/vehicle system. For such systems, inertial coupling is significant, but hydrodynamic coupling dominates (by a factor of five to ten for the manipulators and motions considered here). This research was conducted as part of a joint program between the Aerospace Robotics Laboratory (ARL) at Stanford University and the Monterey Bay Aquarium Research Institute (MBARI) from 1991 to 1995.

1.1 Motivation

Undersea robotic vehicles have the potential to enhance our ability to explore the world's oceans. The potential impact of such vehicles has been demonstrated by the success of scientific vehicles such as JASON at Woods Hole Oceanographic Institution (WHOI) and Ventana at MBARI, shown in Figure 1.1. Currently, remotely operated vehicles (ROVs) are being used to explore and map biological, chemical, geological, and physical phenomena in the deep ocean.

While these ROVs have extended undersea scientific capabilities in a significant way, at this point underwater robots are not only expensive, they are also extremely difficult to control. Completion of scientific tasks often requires two or three skilled operators working in unison. Even simple tasks, such as picking up geological or biological specimens, are burdensome for the trained pilot. The potential of truly-capable underwater robots
remains largely untapped, in part due to their unwieldy characteristics, which currently make efficient completion of seemingly easy tasks arduous or impossible.

Figure 1.1: The Ventana Remotely Operated Vehicle

Based in Moss Landing, California, the Ventana ROV is operated on a daily basis in the Monterey Bay by MBARI. Originally designed for work in the offshore industry, Ventana has been outfitted with high-quality cameras, lights, sampling devices, sensors, and a manipulator for performing science missions. Ventana is a mid-size vehicle — 3 m long, 2 m high and 1.5 m wide, with a mass of 2000 kg.

Today's advanced commercially-available manipulator systems are typically tele-operated, using a passive “master” arm to control an underwater “slave” robot arm. These systems are limited in a fundamental way by the skill, coordination, and endurance of the human operator. High-speed, precise motion is precluded by the restrictions of current human/machine interfaces.

Because of the deficiencies with current operational underwater-vehicle systems and their as-yet-unrealized potential for far-reaching impact, the incentives for the development of vehicles that are more capable and more economic to use are very great. The goal of increasing underwater-vehicle capabilities is being pursued in numerous ways, with much of the current research effort being directed toward improving control-system performance, from the lowest to the highest levels of the control hierarchy.
CHAPTER 1. INTRODUCTION

Human/machine systems can be made easier to use and more productive through control strategies that incorporate increased autonomy, such as Object-Based Task-Level Control [1]. Task-Level Control exploits both the computational power of computers for low-level control and the cognitive capabilities of humans for real-time perception and decision making. Unlike artificial-intelligence-based methods, which disregard the powerful decision-making skills of the human mind, or tele-presence methods, which burden the user with the need to control detailed arm movement continuously, Task-Level Control makes efficient use of computers and humans in a natural complementary way.

With the advent and implementation of higher levels of autonomy in the control of underwater robotic systems, it is becoming possible to position manipulators with greater speed and accuracy. Control strategies, such as Task-Level Control, are capable of providing commands that exploit the full capabilities of manipulators for quick, precise motions. This ability to position rapidly the manipulator creates new challenges for the low-level vehicle control system because of the large hydrodynamic forces acting on the arm as it moves through the water. Hydrodynamic forces on the arm couple dynamically into the vehicle system, increasing the difficulty of regulating the position and attitude of the vehicle.

The relevance of hydrodynamic coupling is increased further for smaller underwater vehicles, which are becoming increasingly popular due to improvements in technology and their reduced cost. Operational expenses for underwater vehicles are most easily reduced by making the vehicle smaller, thereby reducing the demands for a large support vessel and crew. For small vehicles, the addition of manipulators poses an even greater control challenge, because hydrodynamic coupling forces have a greater effect on their vehicle dynamics than for larger, more stable vehicles.

As the human operator is relieved of low-level control responsibilities, the limitations on system performance shift to the control systems implemented to execute tasks for the human. To enable high-performance control of a manipulator end-point from a free-swimming vehicle base, low-level control systems that deal effectively with the complex hydrodynamics of fast motion must be developed. The development and implementation of such a controller is one of the main goals of this dissertation.

In the future, underwater robotic vehicles (URVs) will be more intuitive, less taxing, and easier to learn to use. Future vehicles will be operable by a single scientist-user, in a manner similar to a person driving a car, rather than by a team of highly skilled (and overburdened) pilots. Not only will future vehicles be easier to use, but their capabilities
will be greatly enhanced. Even now, URVs are expanding their roles from observation and collection to on-site underwater experimentation through the emplacement and servicing of sensors and experiments in the ocean. *Robotic manipulation* is a key enabling technology to bring about this advanced new capability.

1.2 Research Challenges in Vehicle Control

1.2.1 Small, Highly-Maneuverable Vehicles

Remotely operated vehicles, such as the Ventana vehicle (see Figure 1.1), are generally quite stable statically. A combination of foam packs and ballast are used to create large buoyancy and gravity restoring forces which passively restrict the pitch and roll attitude of the vehicle to the horizontal plane. In addition to restricting the attitude of the vehicle, the open frame design of ROVs with large foam packs typically results in a bluff shape for the vehicle. This bluff shape results in large drag forces, further limiting the operating regime of the vehicle to low speeds.

In contrast to conventional ROVs, some vehicles (which are usually relatively small compared to their ROV counterparts) are designed with efficiency and maneuverability as capabilities of high priority. The cost for efficient transit capability for many smaller underwater vehicles is reduced static stability due to a small separation distance between their centers of gravity and buoyancy. For vehicles designed to travel efficiently through the water, the requirement for a small frontal drag area limits the achievable static stability. Often static stability is reduced further by the desire for maneuverability out of the horizontal plane. To maintain stability, these vehicles rely on closed-loop feedback control.

**The OTTER Vehicle** To enable experimentally based research and testing of new ideas in the ARL/MBARI program, a testbed vehicle has been designed and developed. OTTER, an Ocean Technologies Testbed for Engineering Research, is pictured in Figure 1.2 below. The OTTER vehicle is about 2.1 m long, 0.95 m wide, and 0.45 m tall. It has a dry mass of 145 kg and is enclosed in a streamlined shell. Designed for maneuverability, OTTER has two main drive thrusters and six maneuvering thrusters that allow control over motion in three dimensions. In its current configuration, the vehicle uses a tether for communications and for trickle charging of batteries. For the experiments reported in this dissertation,
a single-link arm (1.0 m long, 7.1 cm diameter) was mounted on OTTER. A complete description of the OTTER vehicle hardware is given in Chapter 2.

![OTTER Underwater Robotic Vehicle](image)

**Figure 1.2: The OTTER Underwater Robotic Vehicle**

OTTER is a small, highly maneuverable underwater robotic vehicle designed by the ARL/MBARI team to serve as a testbed for conducting engineering research to improve the function and performance of underwater robot systems.

One of the primary design goals for OTTER was to allow rapid transit from the surface, where OTTER is deployed, to the location of interest, perhaps thousands of feet below the surface. With its maneuverability, small frontal area, and low-drag shape, OTTER is capable of traveling efficiently from one location to the next simply by pointing in the correct direction (even if is out of the horizontal plane), and of driving forward to its desired destination. As a result of its small size, maneuverability, and streamlined shape, OTTER
is more sensitive to disturbances than a typical ROV. This makes certain control tasks, such as positioning and station-keeping, very challenging.

1.2.2 Hydrodynamic Coupling Between Arm and Vehicle

In comparison to the inertial forces acting on manufacturing or space manipulators, the additional hydrodynamic forces acting on an underwater manipulator are extremely large and very complex: For typical rotational motions, the peak joint torques required to execute a motion underwater can be an order of magnitude larger than the peak joint torques required in air. For an arm/vehicle system, the hydrodynamic coupling forces represent significant disturbances to the vehicle-control system. These forces, generated as the arm moves, can even cause the vehicle to "swim" away from its desired station, limiting the vehicle's effectiveness in performing manipulation tasks.

In addition to being very large, the fluid forces acting on an underwater manipulator are quite complex, making modeling and simulation of these forces a significant challenge even for geometrically simple manipulator systems. The complexity of the hydrodynamic forces is a direct consequence of the complex nature of the three-dimensional, separated, high-Reynolds-number flow around an arm as it moves, and particularly as it rotates about one end. When expressed as functions of the parameters describing the motion of the arm (angular position, velocity, and acceleration), the hydrodynamic forces are nonlinear, time-varying, and uncertain (due to sensitivity to the "initial conditions" of the fluid).

Along with the hydrodynamic forces, the hydrostatic forces due to the gravity and buoyancy forces acting on the arm can be significant. Their significance is due primarily to their dependence on configuration: As the arm moves, the centers of buoyancy and gravity of the arm move, changing the hydrostatic moments applied by the arm to the vehicle. For accurate control of vehicle attitude and arm-tip location, these changing moments must be taken into account properly.

The significance of fluid coupling between a manipulator and a vehicle is demonstrated in the sequence of photographs shown in Figure 1.3. In this sequence, a single-link arm mounted on the front of the OTTER vehicle was moved back and forth from -45 degrees to +45 degrees. Each 90-degree slew was completed in two seconds with a slight pause before initiating the next slew. To demonstrate the significance of the coupling, all control commands to the thrusters on OTTER were disabled. As the sequence of photographs emphatically illustrates, there is a high degree of coupling between the arm and the vehicle,
Figure 1.3: **Arm/Vehicle Fluid-Dynamical Coupling**

The arm was moved back and forth from -45 degrees to +45 degrees. Each 90 degree slew lasted two seconds. For this sequence, all control commands to the thrusters were disabled. The vehicle rolled as much as 18 degrees in both directions from its nominal horizontal position. In yaw, the vehicle rotated about 15 degrees from its initial heading angle.
due primarily to the hydrodynamic forces acting. Most significantly affected were the roll and yaw degrees of freedom. In roll, the vehicle was disturbed up to 18 degrees in both directions from its nominal attitude. In yaw, the vehicle rotated about 15 degrees from its initial heading angle.

1.3 Research Goals

There were four primary objectives of the research presented in this dissertation:

1. Obtain a fundamental physical understanding of the hydrodynamic forces specific to underwater manipulators and manipulation. As a starting point, the manipulator was chosen to be a slender circular cylinder \((L/D = 9.1)\) actuated by a motor to swing about one end.

2. Based upon the understanding of the physics obtained through experimentation and theoretical analysis, develop a model of the hydrodynamic forces acting on the arm. As the targeted use for the model was control, it was required that the model produce information useful for control, namely hydrodynamic forces and torques as functions of the state and state derivatives of the system. An additional demanding requirement for using the model in control was that it be implementable in real time. This requirement made current state-of-the-art techniques from the field of computational fluid dynamics unsuitable for this application; and this is what led to pursuing instead a demonstrably sound semi-empirical approach to modeling.

3. Develop a control approach for the arm/vehicle system that exploits information made available by the hydrodynamic model to improve the performance of the system in a significant way.

4. Verify the success of the modeling and control approaches experimentally.

The specific task addressed in this research was to achieve high-performance station-keeping control of an underwater robotic system with a manipulator undergoing high-speed motions. By maintaining the vehicle's position precisely (despite the disturbances caused by the manipulator motion) precise positioning of the manipulator end point is made much easier. (And of course concurrent vehicle tasks like photography are made viable.) A schematic of this task is shown in Figure 1.4. The point-to-point positioning task was chosen
as the evaluation experiment for this research for two reasons. First, it is a generic task representative of other tasks of interest for an underwater system, such as sampling, science-experiment emplacement, or pick-and-place maneuvers. Second, successful point-to-point positioning of the manipulator end-point requires high-performance control of the entire arm/vehicle system. Such control is extremely difficult to achieve without compensating for the hydrodynamic interaction forces explicitly in the control of the arm/vehicle system.

![Vehicle-Station-Keeping Task](image)

**Figure 1.4: Vehicle-Station-Keeping Task**

*The targeted task for the arm/vehicle system was to keep the vehicle on station while moving the arm back and forth between two positions in a fashion characteristic of a pick-and-place maneuver. Because of the large hydrodynamic interaction forces, quick, accurate positioning of the arm end-point requires high-performance control of the total system.*

Though end-point position of the manipulator was not sensed and controlled *directly* in the experiments of this dissertation, end-point error (as determined from vehicle position and attitude and arm position measurements) was a valuable single-number metric for judging the quality of the vehicle-station-keeping control. Based upon the accuracy of the individual arm and vehicle sensors, the end-point data calculated are accurate to within 4 cm. As more accurate methods for direct sensing of the manipulator end point are developed for use in the underwater environment, the control strategies developed in this
dissertation can be extended in a straightforward way to take advantage of these sensing improvements.

1.4 Arm/Vehicle Control Approach

The hydrodynamic forces acting on an underwater manipulator as it moves are very large. For a free-swimming vehicle, these forces on the manipulator couple into the vehicle dynamics, resulting in disturbances to the vehicle's control system. However, unlike some vehicle disturbances (tether forces, changing currents, etc.) which are highly unpredictable, the hydrodynamic forces on an arm are largely a function (though complex and uncertain) of the arm state and its derivatives, and therefore can be predicted. The accuracy of the prediction is of course dependent on the correctness of the model used. The arm/vehicle control approach of this dissertation is based on the premise that the interaction forces between an arm and a vehicle can be predicted, and that this information can be used to overcome largely the degrading effects of the interaction forces.

The objective of this approach, called dynamically coordinated control, is to take advantage of physical understanding of system dynamics explicitly in the control of the arm/vehicle system. In the research presented here, this physical understanding is contained in an accurate model of the manipulator hydrodynamic forces. This predictive decoupling control can only compensate for disturbances that can be anticipated, such as those due to arm motion. To provide regulation capability and robustness to unanticipated disturbances (e.g., changing currents), the decoupling control is combined with vehicle position feedback control.

Figure 1.5 shows a simplified schematic diagram of the coordinated-control strategy. Under this approach, hydrodynamic and inertial forces generated from the motion of the arm are modeled in real time as the motion progresses. Based on the predicted interaction forces (and the vehicle position feedback), thrust commands are sent to the thrusters to counteract the forces that will be generated by the arm motion. In this way, the control of the arm and the vehicle are “coordinated.” Ideally, the interaction forces would be perfectly cancelled by the thruster forces, resulting in zero net force and torque on the vehicle. Practically, as will be shown, near-perfect cancelation was achieved.

These hydrodynamic interaction forces are dependent on the motion of the arm with respect to the water, which is a function of both the vehicle and arm motions. For the
station-keeping tasks of this research, where the vehicle is held nearly motionless by the control system, the dependence of the hydrodynamic forces on the vehicle motion becomes negligible, as will be shown in the results of Chapter 5 (hence the dashed line in Figure 1.5).

The control approach presented here was developed under the assumption that the hydrodynamic forces on an underwater manipulator can be modeled accurately and in real time. The primary advantage of doing so is quicker, more-precise end-point control. (An additional advantage of using a model to predict the hydrodynamic forces acting is that no additional sensors are required. This is important for reasons of cost, simplicity, and reliability.) As the coordinated-control strategy developed here required an accurate model, hydrodynamic modeling of an underwater manipulator was a major emphasis of this research.

1.5 Research Challenges in Hydrodynamic Modeling

The above section motivates using an accurate hydrodynamic model in the control of an arm/vehicle system. In this section, the research needed to develop such models is described.
A tremendous amount of work has been performed in the areas of robot arm dynamics and control. Similarly, a huge amount of work has been done by fluid dynamicists investigating flow over cylinders [2, 3]. However, no experimental work (prior to the research reported here) has been done to explore specifically the fundamental issues unique to underwater robotic manipulation. As a result of this lack of experimental work, previously existing models failed to incorporate significant (but non-obvious) hydrodynamic effects, and therefore were capable of predicting the forces acting on an underwater arm with only limited accuracy. In this research, the attributes characteristic of underwater manipulators and their motion have been identified and their significance further understood. These attributes are discussed in the following paragraphs.

**Swinging Motion** The motions of the links of a robotic manipulator typically have a large rotational component, characterized by the swinging of the link about one end. This follows from the fact that a vast majority of manipulators (if not all commercially available underwater manipulators) are articulated mechanisms with rotary joints. In fluid dynamics, research has not addressed specifically the rotation of a long slender bluff body about one end, but rather has focused on cylinders undergoing rectilinear motions relative to the fluid. Yet swinging motion significantly affects the forces acting, since it results in flows that are highly three-dimensional, rather than two-dimensional as in the case of translational motion.

**Rapid Acceleration and Deceleration** Typical robotic motions involve moving from one position to another, with a period of rapid acceleration from an initial position followed by a period of rapid deceleration to a final position. Most of the work in the fluids literature involves steady, uniform flows over cylinders.

**Motion Through Small Angles** Robotic motions are often quite short and transient — generally less than 120 degrees in a few seconds. For such limited motions, the flow over a manipulator link is very transient, often not developing fully to a quasi-steady state. Because of the transient nature of the flow, the forces are also very transient, making them more difficult to model.

**Significant End Effects** Fluid dynamics researchers often go to great lengths to eliminate end effects and other extraneous effects from their experiments. To them, these effects confuse and complicate the issues central to their research. In the case of an underwater
manipulator, end effects are central to the problem. All manipulators have ends. End effects are especially significant since they affect the forces that contribute the most to the hydrodynamic torques acting at the joints of the robot — those near the tip.

**Passing Through Disturbed Water** Unlike a robot arm operating in air, an underwater robot arm can be significantly affected by the wake it has left behind. For quick pick-and-place-type motions, the forces acting on an arm can be dependent on the state of the water, which in turn is a function of previous arm motions.

**Lift Forces** As a robot arm moves though water, vortices are shed into the wake. Under certain conditions, these vortices can introduce very large lift forces that act normal to the plane of the manipulator’s motion.

The above paragraphs serve to illustrate that, for even a single-link manipulator like the one studied in this research, the characteristics of the manipulator and its motion result in hydrodynamic forces that are quite complicated and difficult to characterize.

### 1.6 Contributions

In meeting the goals of this research, the following contributions have been made in underwater manipulator hydrodynamics and underwater robotic vehicle control:

- A fundamental new physical understanding of the hydrodynamics critical to the transient control of a swinging underwater manipulator has been developed. This understanding was developed through a balanced combination of new theoretical and semi-empirical analysis and experimental verification. From a hydrodynamic standpoint, robotic systems and their typical motions possess several unique attributes that require special attention. In this research, these attributes have been identified and their significance demonstrated.

- A highly-accurate, real-time implementable semi-empirical model of the hydrodynamic in-line forces acting on a swinging circular cylinder has been formulated. The form of the model was developed beginning with a two-dimensional analysis of a circular cylinder undergoing unsteady motion in an incompressible, inviscid, and stationary fluid with discrete vortices trailing the cylinder to model the feeding layers and wake.
This model was then extended to the three-dimensional regime that is central to moving-robot-arm hydrodynamics. State-dependent added-mass and drag coefficients were identified from position, velocity, acceleration, torque, and force measurements along the arm. Further understanding into the nature of the forces acting was obtained through experimental flow visualization. The development of this model represents a significant contribution, in that it is the first experimental characterization of forces typical to underwater manipulator motions.

- To support the model development, a novel sensor for measuring forces at locations along the span of a swinging cylinder has been devised. The sensor is composed of a thin aluminum ring (of the same diameter as the cylinder) suspended on cantilevers which are instrumented to measure the force acting on the ring. This sensor allowed state-dependent drag and added-mass coefficients at different locations along the length of the cylinder to be determined directly from force measurements taken during constant-acceleration slews of the arm.

- An approach for simultaneous identification of state-dependent drag and added-mass coefficients for a cylinder undergoing a constant acceleration motion has been developed. These coefficients were identified from measurements of force, torque, position, velocity, and acceleration based on two assumptions. First, it was assumed that the coefficients are functions of how far the cylinder has traveled only. This assumption followed directly from the theoretical analysis of the forces acting on an accelerating cylinder and was borne out in experiments over a wide range of motions. Second, it was assumed that the coefficients do not change instantaneously and can be modeled effectively by smooth spline functions of cylinder travel. The validity of this assumption is reinforced by the flow visualization, which shows the gradual development of the flow as the motion progresses, and by the excellent agreement between the results predicted by the model and the experimentally measured values.

- To enable the hydrodynamic experiments conducted in this research, a test-tank facility has been designed and constructed in the Stanford Aerospace Robotics Laboratory. Included in this facility are actuators, sensors, and mechanical components for the hydrodynamic testing of the single-link, single-degree-of-freedom arm discussed in this thesis and the components for the hydrodynamic testing of multiple-link, multiple-degree-of-freedom manipulators in future research.
• Using the OTTER vehicle, the significance of arm/vehicle hydrodynamic coupling for a small, agile underwater vehicle has been demonstrated experimentally. For arm motions at moderate speeds, it was shown that the end-point positioning accuracy of the manipulator was degraded due to the vehicle motions caused by the hydrodynamic forces generated from the arm motion. This was the case even when the vehicle per se was under closed-loop position/attitude control.

• The dramatic performance benefits of coordinated arm/vehicle control have been demonstrated experimentally for the first time ever. The coordinated-control approach developed in this research incorporates decoupling information from the arm hydrodynamic model into the vehicle controller to counteract the hydrodynamic interaction forces that act between the arm and the vehicle. Using the coordinated-control approach, the manipulator end-point errors were reduced significantly (by a factor of 2.5) when compared to those errors generated when using the conventional approach of independent arm and vehicle controllers.

• As an initial experimental investigation into the hydrodynamics of underwater manipulators, a significant contribution of this research is represented by a broad physical understanding of the nature of manipulator hydrodynamic forces and thus of their significance to underwater robotic systems. As a product of this work, many new and exciting areas for future research in underwater manipulation have been identified (see Chapter 6).

1.7 Reader’s Guide

This chapter has introduced, in a general way, the topics and contributions of the research presented in this dissertation. The remainder of this dissertation is organized as follows:

In Chapter 2 the experimental apparatus used in this thesis are described. A complete description is given of the Stanford test tank facility and the sensors used to conduct single-link flow-visualization and modeling experiments. Additionally, the OTTER vehicle and the experimental setup used for the arm/vehicle coupling experiments are described in detail.

Chapter 3 is an overview of the hydrodynamic forces acting on a circular cylinder as it swings about one end through the water. In this context, a review of the relevant research
is given. In this chapter, hydrodynamic forces are classified, and their effect on manipulator dynamics and control is demonstrated and discussed.

Chapter 4 focuses on the development of a hydrodynamic model of the \textit{in-line} forces (along the direction of travel) acting on the single-link arm. Results of flow visualization and force measurement experiments are given. The theoretical development of the model is described, and a comparison is made between the experimentally measured in-line forces and torques and those predicted by the model.

In Chapter 5 the central topic of hydrodynamic coupling forces between an arm and a vehicle is addressed. The significant effect of these forces on the behavior of a total underwater robotic system is demonstrated with experimental results from the OTTER vehicle. A control approach that takes advantage of hydrodynamic model information from the arm in the control of the vehicle is proposed, discussed, and then demonstrated experimentally with strong results: The considerable performance benefits of coordinating the vehicle control with the motion of the arm are shown.

Chapter 6 concludes this dissertation with a summary of the results of this research and with recommendations for future research in the area of control of underwater manipulator systems.

Appendix A contains mechanical design drawings for the force sensor developed in this research. This sensor was used for measuring forces at different locations along the length of the arm.

In Appendix B, full details of the two-dimensional potential-flow theory analysis of the flow over a moving cylinder are given. The results of this analysis formed the basis for the hydrodynamic model developed in this research.

Appendix C presents experimental data showing the open-loop and closed-loop dynamic response of the OTTER vehicle.

Finally, Appendix D illustrates how the arm and vehicle degrees of freedom were determined and gives the kinematic relations for calculation of the arm end-point position.
Chapter 2

Experimental Apparatus

This chapter gives a description of the apparatus used to conduct the experiments reported upon in this thesis. Two experimental platforms are described: the Stanford test-tank facility where hydrodynamic modeling experiments were performed, and the OTTER vehicle that was used for the arm/vehicle control experiments. A third section of the chapter describes the computer system that was used in both setups.

2.1 Stanford Test Tank Facility

To support the experimental research in underwater manipulation, a small test-tank facility was constructed on the Stanford University campus. This section describes the tank facility including the experimental apparatus used to conduct the tests.

2.1.1 Test Tank

The tank is a 2.3 m diameter by 1.5 m deep and made of 7 mm thick polyethylene (see Figure 2.1). Facilities for water filtration and circulation, water disposal, and emergency containment and drainage have been implemented. Surrounding the tank is an aluminum pipe superstructure from which experimental devices, such as manipulators or thrusters, can be suspended. During initial experiments, it was found that the structure was not stiff enough. Low-frequency modes of the structure were excited by the shedding of vortices from the arm. To add stiffness and raise the resonant frequency, the superstructure was extended to attach to the ceiling, and stainless steel stringers attached to the ceiling were added.
2.1.2 Single-Link Arm

Throughout the course of this research a variety of “arms” have been experimented with. The great majority of the work presented here was conducted with a 7.1 cm diameter, 0.65 m long hollow PVC cylinder shown in Figure 2.2. The arm is mounted so that it rotates about one end in a swinging motion. A cylinder of circular cross-section was chosen because of its axisymmetric shape, which makes it well-suited for an underwater manipulator. Its shape is easily fabricated, it accommodates actuator components (motors, reducers) well, and hydrodynamic forces are independent of the orientation of its cross section with respect to the flow (due to axial symmetry).

2.1.3 Actuator

The actuator used to move the arm is a 1/2 horsepower (373 W) variable reluctance (VR) motor manufactured by Semifusion Corporation. The motor drives a 60:1 harmonic drive reducer. The motor and harmonic drive are contained in a waterproof housing that is filled with mineral oil for pressure compensation. Pressure compensating the housing allows a low-pressure shaft seal to be used, which results in much lower seal friction. The motor
commutation electronics, which incorporate a Motorola 68HC11 microcontroller, are also contained within the housing.

2.1.4 Sensors

To aid in the characterization of the hydrodynamic forces, several different types of sensors were used. A schematic drawing of the actuator and sensors is shown in Figure 2.3. The following paragraphs describe the different sensors used.

**Capacitive Encoder** All of the actuators (*e.g.*, thrusters, pan and tilt) in the OTTER project are pressure compensated, and thus filled with mineral oil. This is true of the single-link arm as well. Since optical encoders don’t function when immersed in oil, a special capacitive encoder was designed by Mike Lee of MBARI. The encoder is used for commutation of the motor as well as for closed-loop control purposes. The particular encoder used with the single-link arm has a resolution of 4096 counts per motor revolution. With the 60:1 reduction, this results in a resolution of 682 counts per degree of revolution of the output shaft. The encoder is incremental, so registration is required on start-up of the system.
An estimate of the arm angular velocity was obtained by taking the first difference of the position signal and then filtering it digitally with a fourth-order Butterworth low-pass filter with a 50 Hz cut-off frequency.

**Accelerometer** To give a measurement of the angular acceleration of the arm, a linear solid-state accelerometer was mounted in the arm at a known distance from the arm hub. The accelerometer was mounted with its sensitive axis normal to the longitudinal axis of the arm and in the horizontal plane to which the arm’s motion is confined. Mounted in this way, the sensor was relatively insensitive to gravitational force: Any gravitational force sensed by the accelerometer due to rotation of the sensitive axis out of its nominal horizontal plane (due to vibration of the arm) was several orders of magnitude smaller than the accelerations caused by rotations of the arm.

![Schematic Diagram of Single-Link Arm, Actuator, and Sensors](image)

**Figure 2.3: Schematic Diagram of Single-Link Arm, Actuator, and Sensors**

*This schematic shows the configuration of the arm and sensors that were used for the hydrodynamic experiments performed in the Stanford Test Tank. The underwater housing and support structure have been omitted from the drawing for clarity.*

**Torque Sensors** Measurements of output-shaft torque and lift moment at the arm hub were made available through a specially-designed beam element connecting the output shaft
to the base of the arm (see Figure 2.3). This beam element was outfitted with two full strain-gage bridges, one sensitive to bending moments in the horizontal plane about the drive shaft, and the other sensitive to bending moments in the vertical plane of the arm about an axis through the hub and normal to both the arm and drive-shaft axes. These sensors measure the output shaft torque \( T \) due to inertial and in-line hydrodynamic forces and the lift moment \( M_{\text{L}} \) due to transverse forces generated from the shedding of vortices.

**Force Sensor** To enable the measurement of hydrodynamic forces at different locations along the length of the arm a novel force sensing arm segment was designed. A photograph of the sensor installed in the arm is shown in Figure 2.2. The sensor, which replaces a one inch thick segment of the arm, is composed of a thin aluminum ring of the same diameter as the arm suspended on cantilevers. These cantilevers are instrumented with a full strain-gage bridge that is sensitive to the bending moments in the cantilevers caused by the hydrodynamic drag forces acting on the aluminum ring.

Arm segments were designed so that the sensor could be moved to any one of five different locations along the arm (see Figure 2.3). In this manner, forces could be measured at different locations to learn how the hydrodynamics varied along the span of the arm. Figure 2.4 shows the force sensor components disassembled. The inner and outer PVC segments of the arm are connected rigidly by the inner plug which passes through the load cell ring and attaches to the outer plug. The ring is supported by the instrumented cantilevers, which also attach to the inner plug.

When fully assembled and prepared for taking data, the sensor is covered in a thin latex sheath to prevent water from passing through the sensor as the arm is moved. Initial tests showed that without this precaution, errors result because the pressure distribution around the arm is affected. The sensor was designed to have a sensing range of \( \pm 4 \) N. Mechanical design drawings of the sensor are included in Appendix A.

The development of this sensor allowed the hydrodynamic forces at different locations along the span to be measured *directly*, rather than estimated from pressure measurements. This was important, because for the purposes of manipulator control, forces rather than pressures are the quantities of interest. With the direct measurement of the hydrodynamic force, it was possible to identify the characteristics of the drag and added-mass coefficients at the different locations. Furthermore, the sensor also allowed the force measurements to
be correlated with the torque measurements at the hub. The force sensor was a valuable tool for developing an understanding of the hydrodynamics of the single-link arm.

2.1.5 Signal Conditioning

The analog force, torque, and acceleration signals were each amplified and filtered using analog circuitry prior to being sampled by the analog input (A/D) board. Gains were chosen so that at maximum load or acceleration, the amplified output would be at the maximum acceptable input voltage of the A/D (±12 V). A two-pole analog low-pass filter with a 50 Hz roll-off frequency was used to prevent aliasing.

2.2 OTTER Vehicle with Single-Link Manipulator

For the arm/vehicle coordinated-control experiments presented in this thesis a single-link arm was mounted on the OTTER vehicle. The following sections discuss the hardware implementation for these experiments. Experiments were carried out in the MBARI test
tank located in Moss Landing, California. The tank is 12 m in diameter and 4 m deep. Additional information on the OTTER vehicle hardware can be found in [4].

2.2.1 Vehicle Overview

The OTTER vehicle is about 2.1 m long, 0.95 m wide, and 0.45 m tall and weighs about 145 kg in air. A photograph of the vehicle with the arm mounted is shown in Figure 2.5. The main structural element of the vehicle is a 0.36 m diameter by 1.25 m long aluminum pressure housing that houses the on-board computers and sensors. Two 0.12 m diameter housings of the same length contain NiCad batteries that provide approximately 750 W-h of power. The battery modules are mounted underneath the main housing.

![OTTER Vehicle With Single-Link Arm](image)

**Figure 2.5: OTTER Vehicle With Single-Link Arm**

*This photo shows the OTTER vehicle with the single-link manipulator mounted. The manipulator is 7.1 cm in diameter and 1.0 m long, while the vehicle is 2.1 m long, 0.95 m wide and 0.45 m tall.*

The pressure housings are surrounded by eight ducted thrusters that provide propulsion to the vehicle. All components are mounted to a welded stainless-steel frame that surrounds the main housing and runs the length of the vehicle. The vehicle is covered by a streamlined fiberglass shell. Additional buoyancy is provided by fiberglass-covered redwood blocks stored within the shell. Because the shell is free flooding, the effective mass and inertia of
the vehicle underwater are significantly higher than in air. A schematic diagram showing the OTTER vehicle and its various components is shown in Figure 2.6.

![Schematic Diagram of OTTER Hardware Architecture](image)

**Figure 2.6: Schematic Diagram of OTTER Hardware Architecture**

*This schematic shows the hardware configuration of the OTTER vehicle for the arm/vehicle control experiments performed in this research.*

### 2.2.2 Arm Configuration

The arm used for the arm/vehicle control experiments was 7.1 cm in diameter and 1.0 m long. This length was chosen because it has roughly the same effective length as a prototype manipulator that has been designed for OTTER when it is in a nominal operating configuration. The arm was mounted from the fore-port corner of the vehicle frame and tilted down at an angle of 60 degrees from the horizontal. This configuration, shown in Figure 2.7, was chosen because it places the arm in the region most likely to be the workspace of a future manipulator. With the arm mounted in this way, all of the vehicle degrees of freedom were affected by the hydrodynamic forces generated as the arm moved.
2.2.3 Thrusters

Propulsion is provided to the vehicle by two drive thrusters and six maneuvering thrusters. The drive thrusters consist of one-horsepower (746 W) variable reluctance (VR) motors with 5:1 planetary-gear reducers. The maneuvering thrusters use smaller 1/4 horsepower (187 W) VR motors with 3.5:1 planetary-gear reducers. The drive thrusters (drive-port and drive-starboard) are mounted at the rear of the vehicle, in the horizontal plane and pointed forward, and are intended to provide the high-power propulsion needed to move rapidly in the forward direction during transit operations. Also mounted in the horizontal plane are the fore and aft lateral thrusters, which together with the drive thrusters are used to maneuver the yaw ($\psi$) and the $x,y$ position of the vehicle.

The remaining four maneuvering thrusters are mounted in the vertical direction with one on each “corner” of the vehicle. Accordingly, they are known as the vertical-fore-port, vertical-fore-starboard, vertical-aft-port, and vertical-aft-starboard thrusters. These thrusters provide control of the pitch ($\theta$), roll ($\phi$), and vertical ($z$) degrees of freedom of the vehicle.
Each thruster housing contains electronics for commutation and control of the VR motors. Motorola 68HC11 microcontrollers incorporated in each thruster unit are used to implement high-bandwidth closed-loop velocity controllers. Experiments have shown very good correlation between thrust produced and propeller velocity squared [5] in steady conditions. Taking advantage of this relationship, a calibration curve for each thruster describing the relationship between thrust and the square of the output shaft velocity was generated. These thrust/velocity relationships were used in the vehicle controller to map commanded thrusts to commanded motor velocities, which were then sent to the thrusters. Communication between the main controller and the 68HC11 in each thruster was done at 31.25 kbaud using a custom serial communications protocol.

2.2.4 Sensors

In order to control the position and attitude of the vehicle, a variety of sensors were used. The following paragraphs briefly describe these sensors.

SHARPS The horizontal position of the vehicle was measured using the Sonic High Accuracy Ranging and Positioning System, also known as SHARPS, made by the Marquest Group, Inc. SHARPS is a long-baseline acoustic system that uses three fixed acoustic transducers to find the position \((x, y, z)\) of a fourth. In the tests reported here, three transducers were mounted around the perimeter of the MBARI test tank, and a fourth was mounted on the vehicle. The SHARPS system provided position updates at about 2.5 Hz with 2 cm of accuracy. Only the \(x\) and \(y\) positions from the SHARPS system were used. A pressure transducer was used to measure depth because of its improved robustness and higher update rates.

Dual-Axis Inclinometer The pitch and roll of the vehicle were measured using a Model SSY0018 dual-axis inclinometer from Spectron Systems Technology, Inc. This sensor has a usable range of ±45 degrees from horizontal and a resolution of 0.01 degree. This unit provided analog voltage outputs that were sampled at 100 Hz.

Flux-Gate Compass A KVH Industries ROV 1000 flux-gate compass provided measurements of vehicle heading. The compass has an accuracy of one degree and a resolution of 0.1 degree in a magnetically clean environment. In this application, the heading signal was
corrupted slightly by electromagnetic noise from the power electronics and cooling fan motors within the main pressure housing. Fortunately, the test-tank material was fiberglass so that electromagnetic interference due to the tank was not an issue. The ROV 1000 provides updates at 10 Hz.

**Depth Sensor** The depth of the vehicle was inferred from hydrostatic pressure measurements taken using a pressure transducer. The pressure measured at the vehicle's location is directly proportional to its depth below the surface. This sensor is used in place of the depth measurement from SHARPS for two reasons. First, because of its simplicity, it is more reliable. Second, since it is an analog sensor, measurements are available at the full 100 Hz sample rate of the vehicle controller rather than the 2.5 Hz update rate available from SHARPS.

**MotionPak** The MotionPak sensor package from Sytron-Donner uses solid-state gyros and servo accelerometers to provide measurements of the three linear accelerations \((x, y, z)\) and three angular rates \((\omega_x, \omega_y, \omega_z)\). In the application presented here, the linear accelerations were not used. The angular rates were used to provide damping in the control of the vehicle attitude.

### 2.3 Computer System

The computer systems and software used for the hydrodynamic modeling experiments at Stanford and the arm/vehicle control tests at MBARI were very similar. A summary of the computer hardware and software used is given in the following sections.

#### 2.3.1 Computer Hardware

The computer hardware used for this research consisted of UNIX-based Sun workstations and a VME-based real-time computer system, both networked to a central file server by Ethernet. For the hydrodynamic experiments at Stanford, the real-time computer hardware consisted of a Motorola MVME-167, 68040-based single-board computer and a Xycom XVME-500 16-channel analog-input (A/D) board with 12 bits of resolution. Communication to the arm motor was done at 31.25 kbaud over one of the MVME-167 serial ports.
CHAPTER 2. EXPERIMENTAL APPARATUS

The vehicle computer hardware architecture was very similar with the exception of two additional VME cages (one top-side, one on-board) to support vehicle control and communications between the vehicle control station and the vehicle (see Figure 2.6). For the experiments carried out for this dissertation, the three VME cages in the system were connected by Ethernet to enable high-speed communications between the components of the system.

2.3.2 Computer Software

The real-time software developed for the experiments of this thesis were written initially in the C programming language and later ported to C++. The code was developed on the Sun workstation and then cross-compiled for real-time implementation on the Motorola single-board computers. The real-time code was developed for use under ControlShell [6, 7], a software framework for real-time systems developed by Real-Time Innovations, which runs on top of the VxWorks operating system [8, 9].

For the arm/vehicle experiments, real-time communication between the top-side communications cage, the arm controller, and the vehicle controller was carried out using the Networked Data Delivery Service (NDDS) [10] over the Ethernet connection between the three VME cages. Commands to the arm/vehicle system were issued from a graphical user interface running on the Sun workstation under ControlShell.

The experimental data displayed in this dissertation was collected using Stethoscope [11], a real-time data monitoring and collection software tool. Not only was Stethoscope useful for data collection, but it was also extremely useful for software debugging and on-line analysis of data.

After collection, experimental data were analyzed, reduced, and plotted using MATLAB, a numeric computation and visualization software package from the MathWorks [12, 13]. MATLAB was also used for the development of hydrodynamic models and for the analysis and evaluation of different vehicle control strategies.
Chapter 3

Hydrodynamic Forces on Underwater Manipulators

The purpose of this chapter is to provide a general overview of the fluid-dynamics field with specific emphasis on topics that are germane to the underwater-manipulator research presented in this dissertation. The chapter begins with an introduction to the research problem considered, that of developing a physical understanding of the hydrodynamic forces acting on a swinging circular cylinder. A detailed review of the pertinent literature is given to provide context for the modeling research presented in Chapter 4. Some results of early experimental investigations of hydrodynamic forces acting on swinging cylinders are presented. This initial work was pivotal in determining subsequent research directions. To conclude the chapter, a summary is given.

3.1 Introduction

The dynamics of common dry manipulators can be fairly complicated, largely due to the number of links and degrees of freedom involved. Many approaches exist for accurately modeling the rigid-body dynamics of these systems. In contrast, the accurate modeling of the large hydrodynamic forces acting on a multiple-link, multiple-degree-of-freedom underwater arm is well beyond the current state of the art in the fields of fluid dynamics and robotics. Due to the complexity of the flow, the forces acting on an underwater manipulator undergoing typical motions are extraordinarily complicated. Prior to the work presented in
this dissertation, there had been no experimental research investigating manipulator hydro-
dynamics. One of the challenges in performing this research was defining a research problem
that was both tractable and generically useful to the underwater robotics community.

3.1.1 The Swinging Circular Cylinder: An Important First Step

For the research reported here, a circular cylinder swinging in a plane about one end was
identified as the starting point for investigations into the hydrodynamics of underwater
robot arms. A swinging circular cylinder was chosen for several reasons.

- It represents a challenging, yet manageable problem that could be studied at a funda-
damental level. From a fundamental understanding of the physics of the single-link
system, extensions can be made to accommodate more complex systems. Results
from the single-link investigations represent a solid foundation upon which further
understanding can be built.

- Though it is composed of only a single link with a single degree of freedom, it shares
many of the unique hydrodynamic attributes of a multiple-link, multiple-degree-of-
freedom arm. These attributes are discussed in the following section.

- A cylinder of circular cross section was chosen because of its axisymmetric shape,
which makes it well-suited for an underwater manipulator. Due to axial symmetry,
hydrodynamic forces are independent of the orientation of the arm cross section with
respect to the flow. The circular cross section is also a convenient geometry for water-
proof housings for motors, electronics, and sensors.

Experimental research into the hydrodynamics of the swinging circular cylinder rep-
sents an important first step to making further progress in comprehending the hydrody-
namics of a full-arm system. A physically sound understanding of such a system will be
achieved only through incremental, experimentally-based investigations, each building upon
the fundamentals discovered in previous successful efforts.
3.1.2 Unique Hydrodynamic Attributes of Manipulators

From a fluid dynamics standpoint, the characteristics of manipulators and their motions are truly unique. These characteristics, which were introduced in greater detail in Chapter 1, are briefly reiterated below.

1. **Highly Rotational**  The motion of robotic links typically has a large rotational component. For the single-link studies here, motion is purely rotational. Rotation induces complex three-dimensional flows.

2. **Rapid Acceleration**  Robotic links move in phases of rapid acceleration and deceleration. Motion is seldom of constant velocity.

3. **Short Motions**  Robotic motions are most often over short distances. Links generally move less than 120 degrees during a slew so the flow forces often never reach steady state.

4. **End Effects**  Robot links have relatively small length-to-diameter ratios ($5 < L/D < 15$) and exposed tips. The effects of flow around the end of arms is very significant.

5. **Previously Generated Wakes**  For an underwater robot undergoing rapid repetitive motions, the wake left from previous motions can affect the forces acting on the arm.

6. **Lift Forces**  Under certain conditions, lift forces normal to the plane of the arm’s motion from vortex shedding can be very large.

For the hydrodynamic model of the in-line forces acting on a single-link arm developed in Chapter 4, attributes 1 through 4 were found to be most significant and were dealt with effectively in the modeling approach. For the motions tested, the effects of previously generated wakes, although noticeable, were relatively small and were not considered. Lift forces due to vortex shedding, which are discussed below, were not addressed in the in-line model. The conditions under which lift forces are large are avoidable for manipulators, and were averted in the experiments conducted here.

For a single-link manipulator like the one studied in this research, the characteristics of the manipulator and its motion result in hydrodynamic forces that are quite complicated
and difficult to characterize. Future efforts with multiple-link arms will undoubtedly require more complexity and sophistication in the hydrodynamic model. Before full manipulators can be addressed, the fundamentals must be understood. This is the primary motivation and justification for pursuing the modeling of a single-link arm.

3.1.3 Physical Characterization of Hydrodynamic Forces

The flow and pressure distribution around a body can be described by the Navier-Stokes equations (3.1–3.3) and the continuity equation (3.4). For an incompressible Newtonian fluid, these equations can of course be written as [14]

\[
\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = f_x - \frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)
\]

(3.1)

\[
\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = f_y - \frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right)
\]

(3.2)

\[
\frac{\partial w}{\partial t} + u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = f_z - \frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)
\]

(3.3)

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0
\]

(3.4)

where \( u, v, w \) represent the velocity components in the \( x, y, z \) directions respectively; \( f_x, f_y, f_z \), the components of the body force per unit mass in the corresponding directions; \( p \), pressure; and \( \nu \), the kinematic viscosity of the fluid.

Since for a given scenario, the motion of the fluid is completely described by the above equations, it is possible in principle to compute the flow behavior by numerical methods. For bluff bodies like cylinders, where separation occurs, the flow over the body and into the wake is very complex. Obtaining numerical solutions even for relatively simple conditions, such as uniform steady flows at moderate Reynolds numbers, is extremely challenging [15]. Numerical analysis of the flow past a cylinder undergoing motions characteristic of a manipulator is beyond the current state of the art in computational fluid dynamics. Even if obtaining an accurate prediction of the hydrodynamic forces were easily possible from the numerical solution of Equations 3.1 through 3.4, the computational requirements would make such an approach infeasible for the real-time control application addressed in this dissertation.

To address the modeling of forces on submerged bluff bodies, such as underwater structures or vehicles, researchers have developed methods based on a combination of simplified
CHAPTER 3. HYDRODYNAMIC FORCES ON UNDERWATER MANIPULATORS

hydrodynamic analysis (e.g., potential-flow theory) and empirical measurement and observation. These methods allow the dominant phenomena present in a particular flow situation to be modeled effectively and in many cases more accurately than is possible by numerical solution of the Navier-Stokes equations.

For a submerged body, such as a manipulator or vehicle, the net hydrodynamic force acting is due to the pressure distribution surrounding the body. This total force can be expressed as the sum of component forces. The primary components of the hydrodynamic force on a submerged bluff body are commonly characterized as follows: drag, added mass, fluid acceleration, and lift.

Not only is the characterization of the total hydrodynamic force as a sum of these components useful from an analysis and modeling perspective, it is also beneficial because it provides a basis for the development of physical intuition into the nature of the forces involved. A complete discussion of these hydrodynamic force components can be found in Newman [16] and Sarphaya and Isaacson [2].

**Drag** Two types of drag exist, *skin drag* which is due to shearing of the fluid in the boundary layer as it flows over the body, and *pressure drag* which is due to the distribution of pressure around the body. For bluff bodies undergoing relatively fast motions, like the cylinders considered here, the flow over the arm is separated. Therefore, skin drag is negligible, leaving pressure drag as the primary source of drag. Drag is typically considered a function of the square of the relative velocity of the the body with respect the fluid.

**Added Mass** When a body moves relative to the fluid in which it is immersed, a portion of the fluid surrounding the body moves as well. When a body is accelerated relative to the fluid, the acceleration of the surrounding fluid with the cylinder results in an increase in the force required to produce the acceleration. The "added mass" is the quotient of the additional force required divided by the acceleration of the body. Although motion of a body always results in motion of the added mass of surrounding fluid, an additional force due to the added mass is not produced unless there is relative acceleration between the fluid and the body. Note that an added-mass force is produced in the case of a cylinder accelerating in a still fluid and in the case of cylinder fixed in an accelerating fluid. Any time there is relative acceleration between the submerged body and the surrounding fluid, there is an added-mass force produced.
Fluid Acceleration  Along with the added-mass force, an additional “fluid acceleration” hydrodynamic force is produced whenever a fluid is accelerated past a body. This force is similar to the buoyancy force experienced by submerged bodies, which is equal to the weight of water displaced by the body and oriented in the direction opposite to the gravity vector. The fluid acceleration force is equal to the mass of the water displaced by the body times the acceleration of the fluid and acts in the direction of the fluid acceleration. In physical terms, for a volume of fluid to accelerate, a pressure gradient must exist in the fluid. This pressure gradient produces a fluid-acceleration force on the submerged body. Unlike the added-mass force, which is a function of relative acceleration between the body and the fluid, the fluid-acceleration force is a function of the fluid acceleration relative to a Newtonian reference frame. In the case of a fixed cylinder in an accelerating fluid, there is a fluid-acceleration force (in addition to the added-mass force). For a cylinder accelerating in a still fluid, there is no fluid-acceleration force that accompanies the added-mass force produced. For the research presented here, the fluid was not accelerating, so fluid-acceleration forces were not present.

Lift  Lift on bluff bodies, such as cylinders, is caused by the shedding of vortices into the wake. Lift forces act in the direction normal to the relative velocity of the body. As fluid particles flow into the leading edge of a cylinder, they are impelled in the boundary layer along top and bottom edges of the cylinder. Near the widest section of the cylinder, the flow separates forming upper and lower shear layers that bound the wake. Since fluid in the shear layers near the surface of the cylinder is moving slower than the free stream, the shear layers roll into the near wake of the cylinder forming discrete swirling vortices. These vortices are typically shed in a regular oscillating pattern (often referred to as the Kármán vortex street [17]) from each side of the cylinder. The formation and shedding of vortices alters the pressure distribution around the cylinder. The fluctuating pressure distribution results in a lift force that oscillates at the vortex shedding frequency. It should be noted that the drag force is affected, but only very slightly, by changes in the pressure distribution due to the shedding of vortices.
3.2 Background

In the literature, the hydrodynamic forces acting on circular cylinders are divided into two categories: *in-line* forces, which act in a direction collinear with the direction of motion, and *transverse* forces, which act in a direction normal to the direction of motion. Figure 3.1 shows a schematic representation of this simple decomposition. In-line forces are composed generally of the drag, added-mass, and fluid-acceleration forces.\(^1\) Transverse forces are due to lift from vortex shedding.

![Diagram showing in-line and transverse forces](image)

**Figure 3.1: In-line and Transverse Hydrodynamic Forces**

*The hydrodynamic forces acting on cylinders are often categorized as in-line, collinear with the direction of motion, and transverse, normal to the direction of motion.*

The following sections give a brief review of the literature dealing with the flow forces acting on submerged cylinders. Research based on computational-fluid-dynamics (CFD) methods is not considered here due to the high computational requirements that currently make these methods unsuitable for the real-time control application of this research.

Thus, the focus of this review is on semi-empirical methods, which are actively being pursued in parallel with computational techniques. In addition to the references provided

\(^1\)The problem of non-collinear cylinder and fluid accelerations, under which the in-line/transverse decomposition would break down, is not commonly considered in the fluids research literature. Typically, analyses consider a stationary cylinder in a moving fluid or a moving cylinder in a still fluid.
below, several texts [16, 2, 3] dealing with marine structures provide excellent overviews of
the field of semi-empirical analysis of hydrodynamic forces acting on cylinders, as they are
a common structural element in the construction of platforms, piers, and other underwater
structures.

3.2.1 In-line Forces

Morison’s Equation Since the early 1950s, Morison’s equation has been the basis for
most of the semi-empirical work on the modeling of hydrodynamic in-line forces on cylinders.
It was originally developed to model the force exerted by surface waves on piles, and has
subsequently been used to model forces due to steady, unsteady, and oscillatory flows.
Morison’s hypothesis was that the total force acting on a cylindrical object immersed in a
moving fluid is made up of two components: 1) a drag force proportional to the square of
the velocity and 2) a virtual-mass force proportional to the accelerative force exerted on the
mass of water displaced by the cylinder [18]. Morison’s equation is commonly expressed as

\[
F = C_m \left( \frac{\rho D^2 L}{4} \right) \dot{U} + C_d \left( \frac{\rho D L}{2} \right) U |U|.
\]  

(3.5)

Morison’s equation cannot be derived directly from first principles for the general flow
conditions to which it is commonly applied. Under different assumptions, the drag and
virtual-mass components can be derived independently using dimensional analysis and
potential-flow theory respectively. Under Morison’s hypothesis they are summed to give
the total hydrodynamic force.

In 1963, Sarpkaya showed that for the special case of a cylinder immersed in a constant-
acceleration flow, that the form of Morison’s equation can be derived directly using potential-
flow theory if vorticity in the feeding layers and wake is taken into account [19]. Rather
than being constant, the coefficients \(C_d\) and \(C_m\) were found to be state-dependent functions
of how far the fluid had accelerated past the cylinder. In the present research, the analysis
of Sarpkaya is extended to accommodate the case of a cylinder accelerating through still
water and to incorporate the three-dimensional effects characteristic of rotational motion.

Outside of the techniques of computational fluid dynamics, Morison’s equation is the
state of the art for modeling the hydrodynamic forces on submerged cylinders. Determina-
tion of the coefficients \(C_m\) and \(C_d\) for different flow regimes is an active area of re-
search [20, 21, 22, 23]. In addition to predicting the magnitude of wave forces on offshore
structures, the form of Morison's equation is also used for predicting hydrodynamic forces on underwater vehicles. Using experimentally obtained coefficients, Morison's equation accurately predicts forces within the specific flow regime to which the coefficients apply.

**Strip Theory**  As the motion of the swinging circular cylinder is rotational, it makes sense to draw on the body of knowledge developed in the study of helicopter rotor aerodynamics. Perhaps the most commonly used method for modeling the aerodynamic lift and drag on a helicopter rotor is blade-element theory [24], also known as strip theory. Under the strip-theory approach, the rotor is divided into multiple elements. The primary assumption of this analysis method is that each element of a propeller or rotor can be considered as an airfoil segment that follows a helical path. Lift and drag are calculated from the resultant velocity acting on the airfoil, with each element acting independently from adjoining elements. The thrust and torque of the rotor are obtained by summing the individual contributions of each of the elements along the length of the rotor.

While there are similarities between the motion of a helicopter rotor and that of a robot arm, there are major differences:

- Helicopter rotors are flexible. Arms are nearly rigid.
- Rotors operate in a very significant “downwash.” There is no downwash over a robot arm.
- Rotors operate in a compressible fluid. Underwater robots operate in an incompressible fluid.
- The flow over rotors is predominantly attached. The flow over cylindrical arms is separated.
- A helicopter rotor's function is to produce lift. A cylinder produces (hopefully) none.

From a fluid dynamics viewpoint, these differences are very significant. In spite of these differences, the strip-theory method is a viable approach for the modeling of the in-line hydrodynamic forces acting on an underwater manipulator. It has been used in the modeling approach of this research and in the approaches proposed by others for the modeling of manipulator hydrodynamics [25, 26].
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The application of strip theory is most convenient when standard textbook values for the hydrodynamic coefficients (e.g., $C_d = 1.2$ and $C_m = 1$) can be used. Sarpkaya [2] identified the conditions under which strip theory can be applied with standard coefficient values.

1. The flow over a narrow slice or strip of the body is essentially two-dimensional.
2. The interaction between adjacent strips is negligible.
3. The end effects are relatively small.
4. The body is sufficiently slender.

Experimental results presented in the following chapter will show that for an underwater manipulator, only the fourth condition is valid. Consequently, development of accurate strip-theory models for underwater manipulators requires experimental identification of valid coefficients to account for the three-dimensional characteristics of the flow.

3.2.2 Transverse Forces

For a cylinder moving through a fluid, transverse lift forces that act on the cylinder are generated. These forces are due to vortices that are shed into the wake from alternating sides of the cylinder. The alternating frequency and the strength of the lift force are functions of the cylinder’s velocity. The Strouhal number, $S$, is a ratio of shedding frequency (in Hz) times diameter to the velocity of the cylinder.

$$ S = \frac{fD}{U} \quad (3.6) $$

For a wide range of Reynolds numbers ($200 < Re < 2 \times 10^5$), the Strouhal number is approximately 0.2. An alternative interpretation is that the spacing of vortices in the wake is approximately equal to five cylinder diameters. For a given cylinder velocity, the shedding frequency can be easily calculated.

Empirical estimates of the lift coefficient due to vortex shedding have been made by many researchers [2] with very little agreement. Discrepancies are due primarily to differences in experimental setups. Cylinder end-effects, turbulence of the stream, and cylinder rigidity can each affect vortex-generated lift dramatically.

Characterization of transverse forces on cylinders using analytical methods is difficult for several reasons. First, separation and vortex shedding are time-dependent. Prediction
of the position of separation points on a circular cylinder in a time-dependent turbulent boundary layer flow is beyond the capability of current analytical methods. Second, there is a great deal of difficulty in describing the kinematics of the flow field in the vicinity of a cylinder. Exactly what happens to the fluid flow as it passes over a cylinder and into the wake is not completely understood. Third, it is difficult to extrapolate two-dimensional analytical findings to a flow situation that is highly three-dimensional. Even in steady, uniform flow, flow at different positions along the length of a cylinder is not coherent. Vortex shedding tends to occur in pockets. The flow field at one point on a cylinder may be very different from the flow field at other locations along its span. Much of the past and current research on vortex shedding is directed towards filling these holes in the general understanding of the phenomenon [27, 28].

**Vortex-Excited Oscillations** In flexible structures, vortex shedding can result in self-excited oscillations. These vortex-excited oscillations have caused catastrophic structural failures in bridges, smoke stacks, offshore drill-pipe columns, and offshore structures. Because of the severity of this problem, it is an area of ongoing research. Several books and a large number of papers have been devoted to this topic [3, 29, 30, 31, 32, 33].

One of the key characteristics of vortex-excited oscillations is lock-in, or synchronism. Lock-in occurs when the Strouhal frequency approaches the resonant frequency of the structure. At lock-in, the vortex shedding frequency corresponds to the resonant frequency of the structure, not the appropriate Strouhal frequency. Vortex shedding and resonant vibrations become synchronized, hence the term synchronism. Since the shedding of vortices occurs at a frequency other than the normal Strouhal frequency, lock-in is also called Strouhal frequency suppression (even though the amplitude of the resonant vibrations is much larger).

Under lock-in conditions, the strength of the lift forces on the cylinder increases. This is because the shedding becomes correlated along the span of the cylinder. Rather than shedding in cells along the span that are only lightly correlated from one to the next, a single vortex sheet is shed.

As with fixed cylinders, characterization of transverse forces on flexibly mounted cylinders is very difficult. Much of the current research deals with classifying the different vortex-shedding regimes and understanding the effects of different structural and flow parameters on the nature of the vortex shedding and structural vibrations [34, 35, 36, 37].
3.3 Exploratory Experiments

While much has been done with respect to flow over cylinders, no prior experimental work has been done to determine the significance of hydrodynamic forces for underwater robotic systems. In the early phases of the research reported here, experiments were performed to gain insight into the nature and significance of manipulator hydrodynamic forces. This section presents some early results that served to set the direction for subsequent research.

3.3.1 In-line Forces: Air Versus Water

Because of the effects of viscous drag and added mass, the in-line forces on a rotating single-link arm are much larger underwater than in air. Figure 3.2 shows measurements of motor torque caused by in-line forces on a single link moving in air and underwater. The peak torque required for the prescribed motion is only 3.1 N-m in air while it is 17 N-m in water — almost a factor of six difference.

![Graph showing in-line torque comparison between air and water](image)

**Figure 3.2: In-line Torque: Air Versus Water.**

*This plot shows that for a typical robot motion, the required joint torque underwater is much larger than in air.*

In addition to having a larger peak force in water than in air, the hydrodynamic forces are non-zero mean over the duration of the slew. For multi-body systems, this results in
much larger coupling impulses being applied between bodies in water than for those in air. For the motion shown in Figure 3.2, the net impulse torque applied by the motor to the arm in air is practically zero, while the net impulse torque applied in water is about 7 N-m·sec. This impulse torque represents the net amount of coupling between components for a given motion — in this case the coupling between the single link and the fixed base. When operating in conjunction with other degrees of freedom, this non-zero-mean coupling makes precise location of the manipulator end effector very difficult. Results presented in Chapter 5 demonstrate that hydrodynamic coupling forces generated by the manipulator motion of an underwater robot can have a significant degrading effect on the performance of the system.

3.3.2 In-line Forces Versus Transverse Forces

Except for very slow motions, in-line forces are always significant for underwater manipulators. The significance of transverse forces is highly dependent on the occurrence of lock-in. Lock-in happens when the vortex-shedding frequency coincides, for a sustained period of time, with a lightly-damped resonant mode of the system. When this happens, the shedding of vortices along the length of the arm becomes correlated, resulting in much larger lift forces.

Figure 3.3 compares the relative magnitudes of in-line and transverse forces for several typical robotic motions where lock-in did not occur. It can be seen that the in-line forces were much larger, especially for the shorter slews. For the multi-body coupling issues addressed in this research, in-line forces were much more influential, and therefore were the focus of the hydrodynamic modeling efforts.

While lock-in was avoided in the research conducted here, it can occur under certain conditions that are feasible for manipulators. Figure 3.4 shows in-line and transverse forces for a situation where lock-in did occur. To obtain the data shown, the single link was rotated at high speed through two complete revolutions. While this is not a common motion for a manipulator, the data show the relative significance of lift forces generated by lock-in. It can be seen that the peak lift torques generated by the transverse forces were as large as the motor shaft torque caused by the in-line forces. While the shaft torque was relatively constant during the constant velocity phase of the motion, the lift torque oscillated at 6 Hz, which coincided with a low-frequency resonant mode of the structure to which the arm was mounted.
Figure 3.3: **Hydrodynamic In-line and Transverse Torques.**

*For a wide range of typical single-link arm motions, the measured transverse torques are much smaller than the in-line torques.*

For underwater manipulators, lock-in is possible when the following three conditions occur simultaneously:

1. The manipulator, or the structure to which it is mounted, has a lightly-damped low-frequency resonant mode.

2. Arm velocities are fast enough so that for the given arm diameter, the Strouhal frequency approaches the frequency of the resonant mode. Blevins [3] suggests that lock-in is possible if the shedding frequency is within 30 percent of the resonant frequency.

3. Portions of the arm travel a sufficient distance so that numerous (at least 4–6) vortices are shed.

For the data of Figure 3.4, these conditions were satisfied. A lightly-damped resonant mode existed due to flexibility in the structure to which the arm was mounted. For the high-speed motion shown, Strouhal shedding frequency at the arm tip was 7 Hz. Because
Figure 3.4: In-line and Transverse Torques Under Lock-in Conditions.

When lock-in occurs, the torque from the transverse forces becomes very significant. The 6 Hz oscillation frequency corresponds to the lowest resonant mode of the arm system. Clipping of the transverse torque data is due to saturation of the analog input channel.

The arm traveled through two full rotations many vortices were shed, exciting the resonant mode, causing lock-in to occur.

The undesirable effects of lock-in can be reduced or eliminated in several ways.

- Raising the affected resonant frequency of the system by stiffening the arm or the structure supporting it, or by reducing the mass of the arm.

- Increasing the damping of the dominant lightly-damped modes of the system.

- Lowering the vortex shedding frequency, either by reducing the speed of the motion or by increasing the diameter of the arm.

- Altering the afterbody flow over the arm so that the flow remains attached or the regular formation of vortices is disrupted.

There are a number of ways to alter the afterbody flow over an arm using add-on devices so that regular vortex shedding is suppressed, including helical strakes, shrouds, fairings,
or splitters [38, 3]. Figure 3.5 shows cross-section schematics of two approaches that were evaluated on the single-link arm — a pivoting streamlined fairing and a trailing splitter plate. Figure 3.6 shows a measurement of lift forces for identical slew of the arm alone, the arm with the streamlined fairing, and the arm with the splitter plate. It can be seen that a dramatic reduction in lift force was realized for both of the add-on devices used. In both cases, the devices prolonged the attachment of the flow, causing the vortices to be shed at a distance from the arm where their effect was minimized. In addition to reducing lift forces, the streamlined shape of such devices often helps reduce pressure drag forces as well.

![Diagram of Streamlined Fairing and Splitter Plate](image)

**Figure 3.5: Schematic of Streamlined Fairing and Splitter Plate**

The fairing and splitter plate are free to pivot on the cylinder and trail behind the cylinder as it moves.

The data presented above shows that under certain conditions, lift forces generated from vortex shedding can be very large. Because of their potential to excite system resonances in a harmful way, the effects of vortex shedding cannot be ignored in the operation of an underwater manipulator system, especially when low-frequency resonances exist and motions are over significant distances at high speed. Should lock-in occur, however, its undesirable effects can be eliminated in a straightforward manner through the combination of methods described above.

### 3.4 Summary

This chapter has presented a brief introduction to the underwater manipulator modeling research conducted as part of this dissertation. As a starting point for experimental investigations, a single-link arm of circular cross section was selected. Although simple in function, the characteristics of a single link and its motion result in hydrodynamic forces
Figure 3.6: Reduction of Vortex-Induced Lift

Data show that shaft bending moments due to lift forces are reduced when a pivoting hydrofoil shroud or splitter plate is used. Large moments observed initially when the hydrofoil shroud was used were due to the flipping of the shroud. Clipping of data is due to saturation of the analog input channel.

that are quite complicated and difficult to characterize. With a solid understanding of the hydrodynamics of a single-link arm, efforts can be extended to multi-link systems.

To provide context for the hydrodynamic modeling presented in Chapter 4, a review of relevant literature addressing the characteristics of hydrodynamic forces acting on submerged cylinders was given. While a tremendous amount of work has been done to understand the forces generated from fluid flow over cylinders, no prior experimental work has been done to consider the specific combination of unique attributes that define the flow over an underwater manipulator and the resulting hydrodynamic forces. As an initial effort to understand the hydrodynamics of manipulator systems, Chapter 4 presents the development and validation of a hydrodynamic model for a single-link arm.

Early experiments in this research helped determine the relative significance of in-line and transverse forces. For the apparatus and motions considered here, lock-in conditions were avoided. Therefore, transverse forces were small compared to the in-line forces generated. The experimental results of this research indicate that unwanted vortex-induced
oscillations in underwater manipulator systems can be avoided initially through careful design or later eliminated by add-on devices. If the requirements of the application permit, a simpler approach to avoid the degrading effects of lock-in is to alter the motion of the manipulator — typically to slow it down. Because in-line forces are unavoidable and dominant, they are the focus of the hydrodynamic modeling efforts presented next.
Chapter 4

Modeling of Hydrodynamic In-Line Forces

Chapter 3 provided an overview of relevant research on the hydrodynamic forces acting on moving submerged cylinders. Background and experimental work were presented to provide motivation for the research work presented here. This chapter addresses the modeling of the hydrodynamic in-line forces acting on a swinging circular cylinder. The chapter begins with a brief introduction that outlines the benefits of and applications for manipulator hydrodynamic models. This is followed by a review of literature in the area of underwater manipulator modeling. The modeling approach taken in this research is then presented. A comparison between the hydrodynamic torques predicted by the model and experimentally measured values is given. The chapter concludes with a brief summary.

4.1 Introduction

The primary goal of the research presented in this chapter was to develop a solid understanding of the hydrodynamic forces acting on an underwater manipulator as it moves through the water, and based on this understanding, to develop an accurate predictive model of the hydrodynamic forces. For this research, the main motivation for developing a model was to enable high-performance automatic control of an underwater arm/vehicle system. To make this possible, it was necessary that the model be implementable in real time using computers on board the vehicle. For this reason, state-of-the-art methods from the field of computational fluid dynamics were not applicable.
CHAPTER 4. MODELING OF HYDRODYNAMIC IN-LINE FORCES

Aside from the benefit of providing insight into the physics of a complex and interesting system, a number of useful applications exist for underwater manipulator hydrodynamic models. In this work, a hydrodynamic model was used to provide disturbance-canceling control to an underwater-vehicle control system where hydrodynamic coupling forces between the arm and vehicle were very large. Other potential control applications include manipulator feedback control design and optimal control of manipulators. As accurate models for multiple-degree-of-freedom, multiple-link manipulators become available, solutions to optimal-trajectory planning problems for underwater arms will become useful. For example, using accurate hydrodynamic models, minimum-time or minimum-energy trajectories could be determined for a given manipulator task specification that take into account the hydrodynamics of the motions involved.

Another interesting application for manipulator hydrodynamic models is the real-time simulation of underwater robotic systems. Realistic vehicle simulations have potential for use in the training of underwater vehicle pilots in much the same way aircraft simulators are used for training airplane pilots.

Finally, in the design of an underwater manipulator system, an accurate model of the hydrodynamic forces involved can be beneficial in determining the dynamic torque requirements involved in executing a specified motion. With a knowledge of the torque requirements, motors, reducers, force and torque sensors, and link dimensions can be designed and selected correctly with greater certainty.

The recent papers of Lévesque and Richard [26], McMillan et al. [25], and Tarn et al. [39] have each addressed the modeling of underwater robotic systems. The focus of these papers is the efficient simulation of underwater vehicles and manipulators with many degrees of freedom. The models presented were each strip-theory-based and recommended using standard constant values for the drag and added-mass coefficients. The latter two papers give an approach for calculating an estimate of the added-mass coefficient, but do not suggest a specific value for the drag coefficient. Lévesque and Richard do not include added-mass or other acceleration terms in their formulation, but suggest a value of 1.1 for the drag coefficient. None of those models was validated experimentally.

Research presented in this chapter, which represents the first experimental investigation of hydrodynamic forces acting on underwater manipulators, shows that the conditions under which the use of standard coefficients is valid are violated for typical robotic motions due
to span-wise flow and end effects. In the model developed here, these three-dimensional-flow effects are accounted for explicitly in the state-dependent hydrodynamic coefficients identified from experiments.

4.2 Modeling Approach

The hydrodynamic modeling approach taken in this dissertation consisted of a complimentary balance of theoretical development and experimentation. Experimental flow visualization was used to determine the behavior of surrounding water as a rotating cylinder was moved through it. This provided clear physical insight into the nature and characteristics of the hydrodynamic forces acting on the cylinder. A two-dimensional theoretical analysis for a cylinder moving through an inviscid fluid was performed. This provided a theoretically valid model form which was then extended to 3-D using the strip-theory approach.

Measurements of force along the span of the rotating cylinder and torque at the cylinder hub were used for identification of drag and added-mass coefficients and to obtain further physical insight into the characteristics of hydrodynamic forces. Measurements of torque at the hub of the cylinder were also used in the experimental validation of the hydrodynamic modeling results. The following sections discuss the individual phases of the model development in detail.

4.2.1 Experimental Flow Visualization

Valuable physical insight into the flow phenomena of a particular system can be gained through flow visualization.\(^1\) Because hydrodynamic forces are completely dependent on the characteristics of the flow, flow visualization provides intuitive physical understanding of the behavior of hydrodynamic forces for a given situation. As underwater manipulators and their motions have several unique attributes, flow visualization furnishes a means for understanding the effects of these attributes and their impact on the characteristics of the hydrodynamic forces involved.

The flow visualization results presented here were obtained by applying small amounts of a highly concentrated paste of food-grade dye to the cylinder at different locations and letting it dry in place. The arm was then submerged, and videotape footage of the arm moving through specified slews was recorded. As the arm moved, the dye applied to the arm

\(^1\) "You can see a lot by just looking." Yogi Berra
surface dissolved into the surrounding water, leaving a visual image of the water’s motion. Video was taken from a top view with the camera looking down on the plane of arm motion and also from a down-the-arm view with the camera attached to the arm at the hub looking out toward the tip. Constant acceleration/constant deceleration motions were considered.

**Video Data — Top View** Figure 4.1 shows flow visualization photos for the arm from the top view. From the sequence of video frames, several unique features of the flow can be seen. In the first frame, it can be seen that the flow is attached initially over the full surface of the cylinder. As the motion progresses, evidence of separation along the trailing edge of the arm can be seen (frame 2). The line of separation moves from the trailing edge of the cylinder to the top as the cylinder rotates (frames 2–5). Along the leading edge, prior to separation the flow is normal to the span. Flow separation is most evident from the span-wise flow radially outward that occurs after the flow separates. The flow along the trailing side (base) of the arm from the hub toward the tip results from the pressure differential between the near-ambient base pressure at the hub and the lower base pressure near the tip. There is also flow around the tip of the cylinder from the ambient-pressure region beyond the tip to the lower-base-pressure area just inside the tip. In the final three frames, the formation of a horseshoe-shaped vortex provides evidence that vortices are shed from the arm in discrete cells rather than long coherent sheets.

The most significant observation from the top-view flow visualization data is that the flow over a rotating circular cylinder is very three-dimensional. Flow along the span and around the tip of the arm act to relieve low-pressure areas along the trailing side of the arm, which are the main source of drag. Because of the 3-D flow, lower-than-normal values of the drag coefficient are expected than for the more commonly investigated 2-D case.

**Video Data — Down-the-Arm View** Figure 4.2 shows flow visualization data for the arm from the down-the-arm view. A patch of dye was applied to the trailing edge of the cylinder at a distance 0.8L out from the hub. For each image, the ratio $s/D$ gives the number of diameters that the cylinder has traveled at the point where the dye was applied. From this perspective, the development of the wake as the cylinder swings can be observed. This is useful since the drag forces on the cylinder are largely a function of the characteristics of the flow separation. In frame 1, it can be seen that the flow is completely attached initially: Here the flow is potential in nature, and the hydrodynamic forces are
Figure 4.1: Flow Visualization — Top View

This video sequence shows the flow over the arm looking down on the arm from above. The key features of the flow from this perspective are flow along the trailing side of the arm from the hub toward the tip and flow around the tip. In the final images, the formation of a large horseshoe vortex can be seen. These features illustrate the complex three-dimensional structure of the flow over a rotating cylinder.
Figure 4.2: Flow Visualization — Down-the-Arm View

This video sequence shows the flow over the arm when looking down the arm from the hub toward the tip. Dye was applied to a point 8/10 of the distance from the hub to the tip. The key feature of the flow from this point of view is the drastic change in the structure of the flow as the cylinder is accelerated from rest. Initially the flow is completely attached. As the motion progresses, the flow becomes fully separated.
due completely to fluid inertia and to skin friction (with skin friction being very small). As the cylinder travels, separation occurs and a pair of symmetric vortices begins to form in the near wake (frame 2). As the motion progresses, these vortices grow, and the separation points move to the top and bottom edges of the cylinder (frames 3 and 4). As the vortices grow stronger, pressure drag on the cylinder increases. The wake first widens (frame 5) and then narrows (frame 6) as the flow becomes fully established.

The key observation that can be made from the down-the-arm-view flow-visualization pictures is that there is a drastic change in the structure of the wake as the cylinder accelerates from rest. Initially the flow is completely attached. As the cylinder moves the flow develops to a state where it is fully separated. Because of this transition in the behavior of the flow, it is expected that the drag and added-mass coefficients will not remain constant, but will change to reflect the instantaneous characteristics of the flow. Understanding the behavior of the forces during this transition period is of major importance for manipulators, because their motions are often short enough to occur *predominantly* within this transition.

### 4.2.2 Theoretical Analysis

To arrive at a model form, upon which a full three-dimensional model of the hydrodynamic forces on a rotating cylinder could be built, a two-dimensional potential-flow analysis of a circular cylinder undergoing unsteady motions was developed from first principles.

Potential-flow theory is a standard analysis approach that is introduced in many fluid mechanics texts, \( e.g., [14] \). The main assumptions of potential-flow theory are that the fluid is inviscid and the flow is irrotational. In addition, in this analysis the fluid is assumed to be incompressible. In regions outside the boundary layer and the wake, the assumptions of an inviscid fluid and irrotational flow are completely valid. The drag effects due to the shearing of the boundary layer (skin drag) are negligible for the Reynolds numbers considered, while drag effects due to separation and formation of the wake (pressure drag) are very large. The challenge in using a potential-flow analysis for a separated flow lies in accounting correctly for the effects of separation and the wake. In this analysis, the feeding layers and the wake are modeled using a number of discrete vortices. Based on the experimental flow-visualization photos showing the development of the wake, placement of these vortices in the feeding layers and wake is justified.
Potential-Flow Theory Background  To provide some background for the analysis presented below, a brief introduction into the foundations of potential-flow theory is given. Irrotational fluid flows, by definition, have zero vorticity. Vorticity, $\omega$, is defined as the curl of the velocity vector $\mathbf{v}$ where $\mathbf{v} = u\mathbf{i} + v\mathbf{j}$:

$$
\omega = \text{curl } \mathbf{v} = \nabla \times \mathbf{v} = 0.
$$

(4.1)

Because the curl of the velocity vector of the flow is zero, the velocity can be written as the gradient of a scalar function $\phi(x, y, t)$ called the velocity potential. In two dimensions,

$$
\mathbf{v} = \nabla \phi = \frac{\partial \phi}{\partial x} \mathbf{i} + \frac{\partial \phi}{\partial y} \mathbf{j}.
$$

(4.2)

Thus, the components of the velocity vector, $u$ and $v$ are related the velocity potential function by the equations

$$
u = \frac{\partial \phi}{\partial x},
$$

(4.3)

$$v = \frac{\partial \phi}{\partial y}.
$$

(4.4)

The underlying principle upon which potential-flow theory is based is the conservation of mass. When applied to an infinitesimal volume within a flow, this principle leads to the continuity equation. For an incompressible fluid, the continuity equation yields

$$
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0.
$$

(4.5)

More concisely,

$$
\nabla \cdot \mathbf{v} = 0.
$$

(4.6)

In view of Equations 4.3 and 4.4 and the continuity equation, the velocity potential, $\phi$ must satisfy the relation

$$
\nabla^2 \phi = \frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} = 0
$$

(4.7)

which is called Laplace’s equation.

Another useful scalar function is the equation of a streamline. A streamline is a curve everywhere parallel to the flow. The equation of a streamline or stream function, $\psi(x, y, t)$, is defined by the equations

$$
u = \frac{\partial \psi}{\partial y}
$$

(4.8)

$$v = -\frac{\partial \psi}{\partial x}
$$

(4.9)
From the stream function definition and the irrotationality condition (Equation 4.1, it can be seen that the stream function also satisfies Laplace’s equation, that is,

\[ \nabla^2 \psi = \frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = 0. \quad (4.10) \]

In two dimensions as considered here, the methods of complex variable theory can be employed to aid in providing solutions to Laplace’s equation. Let the complex variable \( z \) be defined as \( z = x + iy \) and define a new function \( w(z) \) called the complex velocity potential as

\[ w(z) = \phi + i\psi. \quad (4.11) \]

This function, which satisfies Laplace’s equation \( (\nabla^2 w = 0) \), can be used to describe many flow fields of engineering interest. For the flow over a body, a description of the flow using the complex velocity potential enables expressions for the forces on the body to be determined. These forces can be calculated from the pressure distribution around the body as determined by the Bernoulli equation; or, using the complex velocity potential, they can be calculated directly from Blasius’ theorem [40].

**Two-Dimensional Analysis**  At this point an abridged version of the two-dimensional analysis for the problem of interest is given. The full details of this analysis are presented in Appendix B. The analysis considers the flow of a nominally incompressible, inviscid fluid over a cylinder that is moving through it with unsteady motions. The wake and feeding layers are modeled using discrete vortices with independent positions, velocities, and strengths. Figure 4.3 shows a schematic representation of the 2-D cylinder and its wake. The 2-D portion of this analysis is similar in approach to that done by Sarpkaya [41, 19] for a stationary cylinder immersed in a moving fluid.

Using Blasius’ theorem, the forces acting on a moving circular cylinder in a still fluid in the presence of a specified number of singularities can be calculated by

\[ F_X - iF_Y = \frac{i\rho}{2} \int_C \left( \frac{dw}{dz} \right)^2 dz - i\rho \frac{\partial}{\partial t} \int_C \omega d\zeta + \rho A \frac{dW}{dt} \quad (4.12) \]

where \( F_X \) and \( F_Y \) represent drag and lift forces respectively with each integral being evaluated around the contour of the cylinder circumference \( C \).

With the aid of the circle theorem [40], the complex velocity potential \( w \), which describes the flow situation pictured in Figure 4.3, can be written as

\[ w = \frac{U(t) c^2}{z} + \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln(z - z_k) - \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln \left( z - \frac{c^2}{z_k} \right) \quad (4.13) \]
where \( U(t) \) represents the unsteady velocity of the cylinder, \( c \) represents the radius of the cylinder, and \( \Gamma_k \) and \( z_k \) represent the strength and position of the \( k \)th vortex. This expression is composed of a moving doublet term to model the cylinder, a term to model the \( m \) real vortices in the feeding layers and wake, and a term to model the \( m \) image vortices in the cylinder. The image vortices are included to ensure that the boundary conditions on the surface of the cylinder are satisfied. Motion of the reference frame fixed at the center of the cylinder is accounted for by

\[
\overline{W} = U(t)
\]  

where \( \overline{W} \) represents the velocity of the reference frame.

By substituting Equations 4.13 and 4.14 into Equation 4.12, the following expression for \( F_X \) results:

\[
F_X = -\rho \sum_{k=1}^{m} \Gamma_k (v_k - v_{ki}) + \rho \sum_{k=1}^{m} q_k \frac{\partial \Gamma_k}{\partial t} - \frac{\pi}{4} \rho D^2 \frac{dU}{dt}
\]  

(4.15)

Based on an understanding of how vortices in the wake affect the in-line forces, Equation 4.15 makes sense physically. The first two terms account for the effect of vortices in the feeding layer and the wake. When the vortices are small, their contributions to the drag are small. As the vortices grow, their contribution to the total in-line force increases. The third term is an “added-mass” term, which is a function of the acceleration of the cylinder.

For constant acceleration motion of the cylinder, dimensional analysis shows that

\[
\frac{\Gamma_k}{UD}, \frac{u_k}{U}, \frac{v_k}{U}, \frac{v_{ki}}{U}, \frac{p_k}{D}, \frac{q_k}{D}, \frac{p_{ki}}{D}, \frac{q_{ki}}{D} \text{ are functions of } \frac{s}{D} \text{ and } Re,
\]
where $s$ is a measure of how far the cylinder has traveled, $D$ is the cylinder diameter, $Re$ is Reynolds number, $u_k$ and $v_k$ are velocity components (of the center) of the $k$th real vortex, $u_{ki}$ and $v_{ki}$ are velocity components of the $k$th image vortex, $p_k$ and $q_k$ are position coordinates of the $k$th real vortex, and $p_{ki}$ and $q_{ki}$ are position coordinates of the $k$th imaginary vortex.

Sarphaya and Garrison [19] showed empirically, with experiments over a broad range of Reynolds numbers ($100 < Re < 5 \times 10^5$),\(^2\) that

$$\Gamma_k \frac{u_k}{UD}, \frac{v_k}{U}, \frac{u_{ki}}{UD}, \frac{v_{ki}}{U}, \frac{p_k}{D}, \frac{q_k}{D}, \frac{p_{ki}}{D}, \frac{q_{ki}}{D}$$

are functions of $\frac{s}{D}$ only.

Taking this into consideration, Equation 4.15 can be reduced to the following:

$$F_x = -C_m(s/D) \cdot \frac{\pi}{4} \rho D^2 \frac{dU}{dt} - C_d(s/D) \cdot \frac{1}{2} \rho D U^2$$  \hspace{1cm} (4.16)

where

$$C_m(s/D) = 1 - \sum_{k=1}^{m} \frac{4 q_{ki} \Gamma_k}{\pi D UD}$$ \hspace{1cm} (4.17)

$$C_d(s/D) = 2 \sum_{k=1}^{m} \frac{\Gamma_k}{UD} \left( \frac{v_k}{U} - \frac{v_{ki}}{U} \right) - 2 \sum_{k=1}^{m} \frac{q_k}{D} \frac{\partial (\Gamma_k/UD)}{\partial (s/D)}.$$ \hspace{1cm} (4.18)

The key outcome of this 2-D analysis is that for a cylinder undergoing constant acceleration motions (relative to nominally still water), the hydrodynamic drag and added-mass coefficients, $C_d$ and $C_m$, are functions of how far the cylinder has traveled only. In other words, the instantaneous values of $C_d$ and $C_m$ are independent of the cylinder’s acceleration, its velocity, or the associated Reynolds number and are dependent only on the cylinder’s displacement from its initial position at rest.

**Strip-Theory Extension to 3-D** Using a standard strip-theory approach, the theoretically rigorous results of the 2-D analysis can be extended semi-empirically to three dimensions. This approach is diagrammed in Figure 4.4. The forces acting on a thin segment of the arm are calculated using a form of Equation 4.16:

$$dF_i = -C_{mi}(s_i/D) \cdot \rho \frac{\pi D^2}{4} i d_i \dot{\theta} - C_{di}(s_i/D) \cdot \frac{1}{2} \rho D l^2_i d_i \dot{\theta} \ddot{\theta}.$$ \hspace{1cm} (4.19)

\(^2\)For the high-speed arm motions presented here, the value of the peak Reynolds number at the tip of the arm was typically less than $1.5 \times 10^5$
The hydrodynamic in-line torque and force acting at the hub can be calculated using the following simple relations.

\[
dT_i = l_i dF_i
\]

\[
T_{hyd} = \sum_{i=1}^{n} dT_i
\]

\[
F_{hyd} = \sum_{i=1}^{n} dF_i
\]

where \( n \) is the number of segments used in the model.

Figure 4.4: Diagram of Strip-Theory Implementation

This figure shows how strip-theory was applied to extend the 2-D hydrodynamic analysis to 3-D.

The conditions under which strip theory can be applied using standard values for coefficients are outlined in Section 3.2. Figures 4.1 and 4.2 show that the flow around a swinging circular cylinder is very much three-dimensional. The assumptions under which standard values for the hydrodynamic coefficients (determined under two-dimensional flow conditions) can be used are violated for typical robotic motions due to span-wise flow and end effects. Strip theory cannot be applied to situations where these 3-D flow effects are significant unless these effects are accounted for appropriately. Careless application of coefficients obtained from 2-D flow situations to the 3-D flow around a manipulator will result in inaccurate predictions of the hydrodynamic forces acting. If 3-D flow is significant, the model
implementations must correct for this through experimental identification of appropriate coefficient values.

4.2.3 Measurement of Force and Torque

Measurements of local in-line forces along the span of the cylinder and torque at the hub were used for several purposes. First, they allowed experimental identification of the hydrodynamic coefficients of Equation 4.19 above. Second, they allowed the hydrodynamic model for the swinging cylinder to be validated experimentally. Finally, they provided quantitative physical understanding of the forces acting on the system. Some of the physical insights gained through force measurement are discussed here, while coefficient identification and experimental validation are explained in later sections.

Figure 4.5 shows in-line force data taken at several locations along the length of the cylinder for a variety of constant-velocity motions. To obtain these data, the force sensor (described in Chapter 2) was placed at one of five locations along the span. With the sensor in a given location, the cylinder was rotated at a constant angular velocity for just under one revolution. Each data point represents the mean force over three runs at the same velocity. There are six curves shown (one curve for each angular velocity) with each curve having five data points (one data point for each sensor location). As an indicator of how the results varied from one run to the next, the mean force values for individual runs are plotted for the fifth angular velocity.

Several interesting trends can be seen in the data of Figure 4.5. First, for a given speed the measured force increases as the sensor is moved from the hub towards the tip. This is expected, since the local linear velocity increases linearly as the sensor is moved from the inner locations toward those near the tip. At location 5, however, for each angular velocity the measured force is lower than at location 4. This drop in force is due to flow around the tip of the arm, which relieves the low-pressure area along the trailing edge of the arm, causing the drag forces near the tip to be reduced. The forces measured at location 3 are also lower than expected when compared to the data measured at locations 1, 2, and 4. This seems likely to be due to the nature of the radial flow induced along the length of the cylinder, as seen in the flow visualization.

The data of Figure 4.5 correlate well with the flow visualization photos of Figure 4.1. For a cylinder undergoing a constant-velocity motion, the in-line force on the cylinder is made up predominantly of drag due to separation of the flow over the cylinder. Separation of the
flow results in a low-pressure region along the base (trailing side) of the cylinder, which is the source of drag. As the speed increases, the base pressure drops, increasing the drag. Along the base of the arm, it can be reasoned that fluid will flow from regions where the base pressure is highest (lowest drag) to regions where the base pressure is lowest (highest drag). Based upon this reasoning, the local drag-force measurements alone indicate that water is flowing along the trailing edge of the arm from regions near the hub out toward location 4, and around the tip in toward location 4. This is in complete agreement with the results of the flow visualization experiments.

4.2.4 Model Synthesis

Given the relationships of Equations 4.19 through 4.21 and measurements of local forces, hub torque, and angular position, velocity, and acceleration of the single-link arm, the challenge remains to determine how the hydrodynamic coefficients, $C_d$ and $C_m$, vary with respect to the travel or displacement ($s/D$) of the cylinder. Two independent approaches for identifying the behavior of $C_d$ and $C_m$ have been taken here. In the first approach, measurements of force at five locations along the span, along with measurements of position,
velocity, and acceleration, were used in conjunction with Equation 4.19 to characterize the 
drag and added-mass coefficients at each of the locations.

In the second approach, measurements of torque at the hub of the cylinder are used to 
determine the effective drag and added-mass coefficients for the entire length of the cylinder. 
By combining Equations 4.19 through 4.21, and assuming that the effects of the local drag 
and added-mass coefficients can be modeled accurately by global average coefficients \( \bar{C}_d \) 
and \( \bar{C}_m \), the following relationship can be written:

\[
T_{hyd} = - \sum_{i=1}^{n} \left( \bar{C}_m(s_i / D) \cdot \rho \frac{\pi D^2}{4} l_i^2 \frac{d\bar{\theta}}{d\theta} + \bar{C}_d(s_i / D) \cdot \frac{1}{2} \rho D l_i^3 \frac{d^2\bar{\theta}}{d\theta^2} \right). 
\] (4.23)

Measurements of torque and angular position, velocity, and acceleration were used with 
Equation 4.23 to determine \( \bar{C}_d \) and \( \bar{C}_m \).

Both coefficient identification approaches have advantages and disadvantages. The advantage of identifying local force coefficients is that it provides quantitative information 
about the hydrodynamics at locations along the span of the arm; and such information 
is useful because it provides added understanding of the complex three-dimensional flows 
involved in manipulator motions. The operational disadvantages of this method are that 
it requires a large amount of data to characterize the coefficients at many locations and 
that it is not easily extendible to other manipulators. Identification of local coefficients for 
a manipulator of different geometry would require fabrication of sensor components and 
specialized apparatus to perform the required tests.

The advantage of identifying global torque coefficients is that less data gathering and 
reduction is required and that this method is much more easily extended to a manipulator 
of different geometry. The disadvantage of this approach is that information about local behavior of the drag and added-mass coefficients is not captured. While local-force coefficients 
provide enhanced understanding of the radial dependence of the forces acting on an arm, 
the information they provide is not essential for control purposes. For control, information 
about interaction forces and torques at the hub is of prime importance. Though simpler in 
implementation, models using torque-based coefficients provide the necessary information 
for control without sacrificing model accuracy. Results from both coefficient identification 
approaches, showing good agreement between the two, are presented below.
Hydrodynamic-Coefficient Identification Method  Equations 4.19 and 4.23 express relationships between the hydrodynamic force and torque acting and the angular displacement, velocity, and acceleration of the arm — all measured quantities in this research. The unknown parameters of interest are the hydrodynamic drag and added-mass coefficients. A correct characterization of the state-dependent behavior of these coefficients allows accurate prediction of the hydrodynamic forces acting on the arm. The primary challenge in characterizing these coefficients is that for a particular instant in time, Equations 4.19 and 4.23 are both underdetermined. There are two unknown quantities \((C_d \text{ and } C_m)\) with only one expression to relate them (either 4.19 or 4.23).

Sarpkaya [2] proposed a procedure for overcoming this problem. This procedure involved writing expressions for the hydrodynamic force at two instants in time separated by a short time interval and assuming that the coefficients remained constant over the interval. This results in a system of two equations and two unknowns that can be easily solved. Attempts to apply this approach in the present research were unsuccessful: Coefficient values were extremely sensitive to variations in the measurements, particularly to the rapid, but small variations in the force that were due to real hydrodynamic effects. Under this approach, the identified coefficient values varied wildly from one data set to the next, and on many occasions the results were counter-intuitive (e.g., negative coefficient values).

To overcome the difficulties in simultaneously identifying values for \(C_d \text{ and } C_m\), a new approach was developed as part of this research. This approach models \(C_d \text{ and } C_m\) as a series of cubic-spline polynomials that are functions of \(s/D\). The locations of the spline points, and hence the shape of the polynomial functions modeling the hydrodynamic coefficients, are determined using nonlinear optimization. A unique feature of this approach is that it allows the user to incorporate previously gained knowledge of the hydrodynamics into the coefficient profiles.

Based on flow visualization and analysis results, several conclusions regarding the behavior of the hydrodynamic coefficients can be inferred that are helpful in their identification. First, from Figure 4.2, it can be seen that the flow is fully attached initially. Under these conditions, potential flow theory accurately predicts that — at that point in time — \(C_d = 0\) and \(C_m = 1\).

Second, from flow visualization it was found that up to a displacement of roughly seven or eight diameters, the flow changed gradually, but after that the configuration of the wake changed only slightly. The flow was fully separated. From this it can be inferred that after
a given initial displacement the drag coefficient settles to a positive value that varies only slightly. Due to span-wise flow and flow around the tip, the drag coefficient is probably less than the normal value of 1.2 associated with the translational motion of a cylinder with no end-effects.

Finally, from the video sequences of Figure 4.2, it can be seen that the structure of the flow changes gradually over the period of time from when the motion starts until when the wake is fully developed. From this it is reasonable to assume that the coefficients change gradually as well during the startup transient. This gradually varying behavior of the coefficients is well-modeled by the smooth cubic-spline polynomials used to approximate the functional relationship between \( C_d \) and \( C_m \) and \( s/D \).

Figure 4.6 illustrates the polynomial spline models used for the drag and added-mass coefficients prior to fitting to the experimental data. Each curve is composed of three types of spline points: one fixed point, five shaping points, and one scaling point. For both coefficients, the fixed points correspond to the start-of-motion values of the coefficients predicted by potential-flow theory: In the initial instants of motion, the value of \( C_d \) is fixed at 0 and the value of \( C_m \) is fixed at 1. These two points remain unchanged during the optimization process.

For each coefficient, five equally spaced shaping points are used to vary the shape of the coefficient profile. Movement of these shaping points allows the coefficient curves to take on a wide variety of shapes that model properly the state-dependent behavior of the coefficients.

Each coefficient curve is scaled both in the travel dimension and the coefficient magnitude dimension by a single scaling point. Movement of this point adjusts the scale of the entire coefficient profile between the fixed point and the scaling point. Location of this scaling point effectively determines the steady-state value of the coefficient and the travel value after which the coefficient is assumed to be constant.

Given measurements of arm position, velocity, acceleration, and hub torque \((T_{meas})\), the nonlinear optimization problem of fitting the hydrodynamic coefficients to the experimental torque data can be posed as follows:

Find \((d_1, d_2, d_3, d_4, d_5, d_s, m_1, m_2, m_3, m_4, m_5, m_s, \tau_s)\)

\[
\text{to minimize } \sum_{i=1}^{m} (T_{hyd_i} - T_{meas_i})^2 \quad (4.24)
\]
where $T_{hyd}$ comes from Equation 4.23 and $m$ is the number of data points being considered. This same approach was used to identify local force coefficients. For the coefficient data presented below, a total of 12 different arm slews at four different accelerations were used to evaluate the coefficients. For each of the 12 data sets, measurements of hub torque and arm position, velocity, and acceleration at 200 to 500 time points (depending on the duration of the slew) were used to determine the best-fit location of the coefficient spline points. The identification of one coefficient profile for multiple runs of different accelerations is in accordance with the theoretically derived dependence of the coefficients on travel only.

This approach of modeling the coefficients as cubic splines and then using nonlinear optimization to vary the shape of the splines to determine the state-dependent behavior of the coefficients allows the problem of determining two parameters from one equation to be solved by considering not just each instant in time individually, but by optimizing the coefficient profile shapes (using a relatively small number of parameters) over their
entire transients. This approach not only enables solution of the problem, but it allows previously determined physical information about the coefficients’ behavior to be included in the parameterization of the coefficient profiles.

**Experimentally Identified Coefficients** Figure 4.7 shows values for the *torque-based* drag and added-mass coefficients that were obtained using the optimization procedure described above. The drag coefficient starts off initially at zero as constrained by the fixed spline point. The drag coefficient rises rapidly and peaks at a value just larger than 1.0 after about 3.7 diameters of travel. The drag coefficient then drops off and settles in at a steady-state value of 0.78 after 8 diameters of motion. The added-mass coefficient starts off initially at a value of 1.0 and drops off to a value of 0.2 after 4.2 diameters of motion. The added-mass coefficient then rises to its steady-state value of 0.36.

Variation of the coefficients for different slews is indicated by the data points marked with ‘*+*’ symbols. These points were determined by performing the optimization procedure on three groups of data from four different slews (rather than one group of data from 12 slews).

The important thing to note from these results is that they correlate very well with the results from both the flow visualization and the 2-D theoretical analysis. For \( s/D = 0 \), the drag and added-mass coefficient values are fixed at 0 and 1 respectively, as potential-flow theory predicts for the case of fully attached flow. As the flow separates and a symmetric vortex pair begins to form in the near wake, the drag coefficient rises. The drag coefficient reaches its peak when the vortices are at their largest size just prior to being shed. When the symmetric vortices can no longer coexist in a stable manner in the near wake, vortex shedding begins. When this happens the drag coefficient drops off, after which it rises to its steady-state value corresponding to the fully developed wake.

The dependence of the drag coefficient on the development of vortices in the wake is shown by Equation 4.18: When no vortices are present; the drag coefficient is zero; as vortices grow and move in the wake, the drag coefficient rises accordingly.

Because of the three-dimensional flows along the span and around the tip, the steady-state drag coefficient (0.78) is considerably lower than the value of 1.2 expected for a translating cylinder with no end effects. Flow around the tip and along the span relieve the low-pressure area at the base (trailing side) of the arm near the tip, resulting in lower drag forces and hence a lower mean drag coefficient.
Figure 4.7: **Drag and Added-Mass Torque Coefficients**

*This plot shows the state-dependent behavior of the drag and added-mass coefficients identified from measured torque data. The state-dependent characteristics of the coefficients correlate very well with flow-visualization results. The status of the wake corresponding to values of the coefficients at four different travel states is shown in the sketches across the top of the plot.*

From the behavior of the added-mass coefficient, it can be concluded that the presence of vortices in the near wake reduces the amount of fluid that is accelerated locally as the cylinder is accelerated. This result is in complete agreement with the results of the two-dimensional potential-flow-theory analysis shown in Equation 4.17: With no vortices in the wake, $C_m = 1$; as separation occurs and vortices grow in the wake, the added-mass coefficient drops off from this initial value. As long as separation occurs and vorticity is shed into the wake, it is to be expected that the added-mass coefficient will be smaller than the value of 1 predicted by potential-flow theory (without separation effects).
Figure 4.8 shows plots representing the drag and added-mass coefficients identified experimentally from force measurements made at five different locations \( l_1 = 0.42L, l_2 = 0.54L, l_3 = 0.66L, l_4 = 0.79L, l_5 = 0.91L \) along the span of the cylinder. Of fundamental interest is the dependence of the coefficient values on radial location. It can be seen that there are trends common to all of the coefficients of each type, and that there are significant variations from location to location. The common features can be attributed to each location experiencing similar flow phenomena, mainly due to the development of the wake as the cylinder accelerates. For each of the drag-coefficient curves, the coefficients rise from their initial starting value of 0, peak at their maximum value, undershoot slightly, and then settle to a steady state value. With the exception of the curve from location 5, each of the curves peaks between 2.5 and 3.5 diameters of travel. The drag-coefficient curve for location 5 peaks at about 4.5 diameters.

As an indicator of the variability in the coefficients, data points from three separate optimizations are plotted for the drag and added-mass coefficients identified for location 4.

The added-mass-coefficient curves all follow a similar pattern of starting off at 1, then gradually decreasing to a minimum value, then rising slightly to settle at a steady-state value. The minimums of the added-mass-coefficient curves occurred between 3 and 4 diameters of travel, with the exception of the curve representing location 5, which reached a minimum at about 5 diameters of travel. Delays in the development of the flow at location 5, as evidenced by the lag in its coefficient curves relative to the curves of the other locations, can be attributed to the effects of flow around the tip interacting with the flow over the arm.

The differences between the drag and added-mass coefficients between the locations are undoubtedly due to the three-dimensional flow effects induced by the swinging motion. The low values of the steady-state drag coefficients at locations 1 and 5 are due to the proximity of these locations to the ambient-pressure regions near the tip and the hub. Flow from these ambient-pressure regions into the base area reduces the drag acting near the tip and the hub.

While the wide variation among the local force coefficients is somewhat surprising, the coefficients identified from the local force measurements are in very good agreement with those coefficients obtained from torque measurements. By comparing the combination of Equations 4.19 through 4.21 with Equation 4.23, it can be seen that in steady state the individual force coefficients \( (C_d_i, C_m_i) \) and the global torque coefficients \( (\bar{C}_d, \bar{C}_m) \)
Figure 4.8: Local Drag and Added-Mass Force Coefficients

These plots show the state-dependent behavior of the drag and added-mass coefficients identified from force measurements taken at locations along the span \( l_1 = 0.42L, l_2 = 0.54L, l_3 = 0.66L, l_4 = 0.79L, l_5 = 0.91L \). Each curve is labeled with numbers corresponding to the location from which it was obtained. While the curves are all of similar shape, there is a significant variation in the magnitudes of the coefficients at the different locations. Hub torque is of course influenced most by forces labeled 4 and 5, because of their larger moment arm.

are related by the following expressions:

\[
C_d = \frac{\sum_{i=1}^{5} l_i^3 d_i C_{d_i}}{\sum_{i=1}^{5} l_i^3 d_i}, \quad (4.25)
\]

\[
C_m = \frac{\sum_{i=1}^{5} l_i^2 d_i C_{m_i}}{\sum_{i=1}^{5} l_i^2 d_i}. \quad (4.26)
\]

If Equations 4.25 and 4.26 are used in conjunction with experimentally evaluated force coefficients \( (C_{d_1}, C_{d_2}, \ldots, C_{d_5} \text{ and } C_{m_1}, C_{m_2}, \ldots, C_{m_5}) \), steady-state values for \( \bar{C}_d \) and \( \bar{C}_m \)
can be calculated indirectly. Using this approach, $\bar{C}_d$ was calculated to be 0.75 and $\bar{C}_m$ was calculated to be 0.39. These values are in very good agreement with the steady-state values of the coefficients obtained directly from torque measurements: $\bar{C}_d = 0.78$ and $\bar{C}_m = 0.36$.

Because of the dependence of the local force coefficients on $s_i/D$, comparison of the local force coefficients with the global torque coefficients during their transient phases cannot be done by simply applying Equations 4.25 and 4.26, as was done for their steady-state values. The most straightforward method for evaluating the agreement between the local-force-based coefficients and the total-torque-based coefficients in their entirety is a comparison of results from the implementation of Equations 4.19-4.21 using local-force-based coefficients and Equation 4.23 using total-torque-based coefficients. Figure 4.9 shows results from this approach using an ideal constant-acceleration/constant-velocity/constant-acceleration trajectory as a source of position, velocity, and acceleration information.

As with the steady-state values of the coefficients, the agreement between the torques calculated with the local-force-based coefficient and those calculated with the total-torque-based coefficients is very good. Differences in the drag and inertial torques in the first second of motion are due primarily to variations in the coefficients during their transient phases. Thereafter, variations are due primarily to differences in the steady-state values of the coefficients.

### 4.2.5 Modeling Approach Summary

The above sections have presented the approach taken in this dissertation for the modeling of the forces acting on a single-link manipulator. Results from flow-visualization, local-force-measurement, and total-torque-measurement experiments were shown. A theoretical analysis, which served as the foundation for the model, was presented. Finally, a new approach was given for identifying, from experimental data, the state-dependent coefficients for drag and added mass.

### 4.3 Experimental Results

In this section, experimental results validating the accuracy of the model developed in the previous section are given. For the results presented here, the arm was divided into 10 equal segments. For simplicity, global torque-based coefficients were used, although force-based coefficients would have given similar results (as Figure 4.9 ensures). This hydrodynamic
Figure 4.9: Correlation of Torque-based and Force-based Coefficients

These plots compare the drag and inertial components of the hydrodynamic torque as calculated from Equations 4.19–4.21 using local-force-based coefficients and Equation 4.23 using total-torque-based coefficients. The position, velocity, and acceleration profiles used for the comparison are shown in the top plot. Drag and inertial components of the hydrodynamic torque are shown in the middle and bottom plots respectively.

The model is relatively undemanding from a computational point of view and has been implemented at sample rates up to 500 Hz on a 68030-based real-time processor using four segments.

Figures 4.10 through 4.12 compare modeled hydrodynamic torque predictions from the model developed here and two other models from the literature with the actual hydrodynamic torques that were measured experimentally. Figure 4.10 shows results from a short 30-degree, 0.4-second slew, Figure 4.11 shows results from a 60-degree, 0.6-second slew, while Figure 4.12 shows results from a longer 120-degree, 1.2-second slew. Results from three different slews are given to demonstrate that the model developed here is valid for a wide range of motions. In each figure, plot (a) shows the time history of the arm
motion. Plot (b) shows results from the model developed as part of this research using state-dependent coefficients. Plot (c) shows results from a model implemented according to Lévesque and Richard [26] wherein the effects of added-mass were not considered and the drag coefficient was assumed to have a constant value of 1.1. Plot (d) shows results from the model inferred from McMillan, et al. [25] in which the drag coefficient had a constant value of 1.2 and the added-mass coefficient had a constant value of 1.

Figure 4.10: Hydrodynamic Modeling Results — Short Motion

These plots show hydrodynamic modeling results for a single-link manipulator undergoing a short, quick swinging motion. Plot (a) shows the trajectory of the arm motion. Plot (b) compares measured torques with those produced by the model from this research. Plot (c) shows results from a model implemented as suggested in [26]. Plot (d) shows results from a model implemented as suggested in [25].

The results of Figures 4.10 through 4.12 demonstrate the superior modeling accuracy of the approach presented in this dissertation over the most applicable of the other methods presented in the literature. On average, modeling errors\(^3\) were reduced by 4.5 times when comparing the model of this research with the other two models considered. This higher

---

\(^3\)Model errors were calculated by normalizing the sum of the errors for the data sets presented in Figures 4.10–4.12 according to \( \sum_{i=1}^{m} \frac{|T_{\text{meas}} - T_{\text{hyd}}|}{\sum_{i=1}^{m} |T_{\text{meas}}|} \), where \( m \) is the number of data points.
Figure 4.11: Hydrodynamic Modeling Results — Medium-length Motion

These plots show hydrodynamic modeling results for a single-link manipulator undergoing a swinging motion of moderate length. Plot (a) shows the time history of the arm motion. Plot (b) compares measured torques with those produced by the hydrodynamic model of this research. Plot (c) shows results from a model implemented according to [26]. Plot (d) shows results from a model inferred from [25].

level of accuracy is enabled by the use of state-dependent drag and added-mass coefficients. During initial portions of the slews, these coefficients take on values corresponding to values predicted by potential flow theory ($C_d = 0$ and $C_m = 1$). As the motion progresses, the coefficients transition to take on steady-state values corresponding to the three-dimensional separated flow situation ($C_d = 0.78$ and $C_m = 0.36$).

In plot (c) of Figures 4.10, 4.11, and 4.12 it can be seen that the constant coefficient model fails to capture the effects of the inertial torques, which dominate at the beginning and end of the slew, and that it overestimates the drag during the middle portion of the slew. In the results of plot (d), this constant-coefficient model initially models the inertial torques very well, but later overestimates both the inertial and drag torque components. The major sources of error in the models presented in plots (c) and (d) are due to assuming the coefficients to be constant and assuming that the drag coefficients have the same values
as those corresponding to the translational motion of a cylinder \((C_d = 1.1 - 1.2)\), which they do not because of the radial flow induced by the swinging motion. In other words, the errors can be attributed to using coefficients from a two-dimensional, steady-flow situation to describe a three-dimensional, unsteady flow.

While the data presented in Figures 4.10 and 4.12 are for constant-acceleration/constant-deceleration motions, it should be noted that the model developed here has been applied with equal success to fifth-order-spline trajectories, which are common for robotic manipulators. Fifth-order trajectories are desirable because they result in smooth acceleration commands to the system, which minimizes wear on the manipulator joints and avoids unwanted excitation of system resonances.
4.4 Implementation Issues

Local Force Coefficients Versus Global Torque Coefficients  The measurement of forces at different locations along the arm has been beneficial in developing an understanding of the hydrodynamics of the flow around the single link. In addition, local force measurements have aided in the development of the hydrodynamic model discussed above.

From an implementation standpoint, however, the use of coefficients identified from torque measurements is more convenient for two reasons. First, for an accurate force-coefficient-based model to be developed, the identification of state-dependent force coefficients at many locations is required. The identification process for each location requires that a significant amount of data be gathered and that a nonlinear optimization be performed. To model a single link accurately, 10 to 20 force-coefficient profiles would be required. In contrast, an accurate model using the global torque coefficients requires that only two coefficient profiles be determined.

The second reason for using torque-based coefficients is that they are more easily identified for manipulators of different geometries. Because of their dependence on the three-dimensional aspects of the flow that are unique to the particular arm considered, the force-coefficient results of Figure 4.8 are not directly extendible to arms of different geometries. The measurement of local forces in this research required the development of a custom sensor and related hardware to allow the movement of the sensor from one location to the next. Using such an approach for the development and validation of hydrodynamic models for a variety of manipulators is too burdensome to be feasible. Torque measurements at the hub of an arm, on the other hand, can be made with commercially-available sensors and would require only a small amount of fabrication. The coefficient-identification approach outlined above could be easily applied to other geometries if coefficients were based on torque measurements.

Model Accuracy Versus Number of Segments  A reasonable expectation is that the accuracy of the hydrodynamic model will increase with the number of segments used to model the arm. Figure 4.13, which shows modeling error plotted against the number of model segments used, confirms this expectation. An interesting result is that the modeling error does not decrease significantly when more than four model segments are used. These
results demonstrate that using a large number of segments, which increases the computational requirements, is not necessary to achieve very accurate results using the model developed above.

![Graph showing RMS error versus number of segments](image)

**Figure 4.13: Modeling Error Versus Number of Segments**

*This plot shows the relationship between modeling error and the number of segments used to model the arm. It can be seen that relatively accurate results are obtained using only a few segments — a positive result from a computational perspective.*

For the model and coefficients implemented, RMS modeling error does not fall below three percent. While three percent is a quite acceptable error for the purposes investigated here, it is obvious that model accuracy is limited by error sources other than the number of segments used. Hydrodynamic forces generated as the arm moves are very sensitive to the conditions of the fluid and very small fluctuations in the arm motion. Because the hydrodynamic model of this thesis considers the flow forces on a macro scale by emulating the net effect of the individual fluid-particle motions, it is unable to model some of the second-order effects caused by the inherent small variations in the flow. While the dominant hydrodynamic effects of arm motion are captured very well by the combination of coefficients and measurements used in the model, resolution of some of the smaller force variations (which vary from one run to the next) is not possible without considering motion of the individual fluid particles. Since accurate modeling at this micro level is not presently possible for the
complex flows considered, the approach taken here represents the best solution for modeling of hydrodynamic forces.

**Drag Versus L/D Ratio** The forces acting on a cylinder are dependent wholly on the behavior of the flow. For the rotating single link considered here, the flow is three-dimensional. The three-dimensional characteristics of the flow around a single link are dependent on its geometry, primarily its length-to-diameter ratio \( (L/D) \). The parameter most affected by changes in \( L/D \) is the drag coefficient.

Figure 4.14 illustrates the effect of the \( L/D \) ratio on the average torque-based drag coefficient determined from constant-velocity motions of the arm. These drag coefficients were determined directly from measurements of joint torque and joint-angle velocity using the strip-model approach of Equation 4.23 with \( C_d \) constant and the added-mass portion of the equation set to zero.

![Drag Coefficient Versus L/D Ratio](image)

**Figure 4.14: Drag Coefficient Versus L/D Ratio**

This plot shows the relationship between the drag coefficient and the length-to-diameter ratio for a swinging cylinder. The drag coefficient was determined from torque measurements taken from a variety of constant velocity motions of the arm. As a reference, the drag coefficient for a long cylinder undergoing purely translational motion is 1.2.
CHAPTER 4. MODELING OF HYDRODYNAMIC IN-LINE FORCES

It can be seen that the effective drag coefficient increases with $L/D$. This is because for arms of high $L/D$, end-effects act over a smaller percentage of the span and because the motion becomes more translational near the tip of a high $L/D$ arm than that of a low $L/D$ arm. In essence, 3-D flow effects become less significant with increasing $L/D$. It is expected that for extremely high $L/D$ ratios (beyond what is feasible for manipulators) the drag coefficient will approach values typical for a translating cylinder ($C_d = 1.2$).

To obtain the most accurate modeling results, coefficients for arms of different configurations should be identified based on experimental torque measurements. If this is not possible, a valid engineering approach is to scale the drag-coefficient curve of Figure 4.7 ($L/D = 9.1$) by an appropriate amount as indicated in Figure 4.14. For example, for an arm of $L/D = 6$, the steady-state value of the drag coefficient of Figure 4.7 would need to be reduced by a factor of 0.94 (0.80/0.85) to account for the increase 3-D flow due to the smaller ratio of $L/D$. When measurement of torque for calibration purposes is not possible, this technique represents a valid engineering approach for extending the results of Figure 4.7 to arms of different geometries. During the course of this research, this approach has been shown to give good results. Careful comparison of the results of Figure 4.14 with those of Figure 4.7 will show that the steady-state drag coefficient for the constant-acceleration case is about eight percent lower than that of the constant velocity case. This is because acceleration promotes stability and attachment of the flow over the surface of the arm resulting in slightly lower drag forces [2]. This acceleration effect is partially responsible for the significantly lower drag coefficients identified in the experiments discussed above. This represents another characteristic of robot motion that is not well-modeled by the standard drag coefficient.

**Acceleration Measurement** For the model identification and experiments presented above, angular acceleration of the arm was sensed using a linear accelerometer placed in the arm at a known distance from the hub. These measurements were very important in developing an understanding of the manipulator hydrodynamics. In implementing the hydrodynamic model in the arm/vehicle control application of Chapter 5, desired joint acceleration was used in place measured joint acceleration. This approach has the benefit of reducing the sensor requirements for the system. In this section, the impact on model accuracy of using desired acceleration as an estimate of actual acceleration is briefly addressed.
Figure 4.15 shows modeling results using both measured and desired acceleration in the model. While the model is in better agreement with the torque measurement when measured acceleration is used, the use of desired acceleration in the model gave very good results. Errors in the model using desired acceleration are most significant at the very beginning and end of the slew when the actual acceleration did not track the desired acceleration as accurately due to the dynamics of the actuator. When using the desired-acceleration signal in the model, it was found that the best results were obtained when fifth-order trajectories were used to command the arm. Using these smooth trajectories, the desired-acceleration command was more accurately tracked by the arm resulting in improved modeling of the hydrodynamic forces.

![Graph showing arm position and torque over time](image)

Figure 4.15: Model Results Using Desired-Acceleration Signal

*These plots compare hydrodynamic modeling results when measured-acceleration and desired-acceleration signals are used in the model.*
4.5 Summary

In this chapter, the development of an accurate model of the hydrodynamic forces acting on a single-link manipulator has been presented. This model is the product of a balanced combination of theoretical analysis and experimental research. Flow visualization experiments were conducted to learn about the behavior of the flow over the arm and to gain insight into the hydrodynamic forces generated by the arm’s motion. A rigorous two-dimensional potential-flow-theory analysis, which took into account the effects of vortices in the wake, was performed for a cylinder undergoing unsteady motions. The results of this analysis were then extended semi-empirically to three dimensions using strip theory. Measurements of force and torque were used to gain further physical insight into the hydrodynamic forces acting on the arm and as a basis for the identification of hydrodynamic drag and added-mass coefficients. To enable the identification of these coefficients, a new approach based on modeling the state-dependent behavior of the coefficients as cubic-spline functions of arm travel was developed. The net result of these efforts was the implementation and experimental validation of a very accurate hydrodynamic model of the forces acting on swinging single-link arm.
Chapter 5

Coordinated Control of an Arm/Vehicle System

The research presented in Chapter 4 showed that the hydrodynamic forces acting on a single-link arm can be modeled accurately and in real time for a wide range of arm motions. As shown in Chapter 1 (and further demonstrated in this chapter), hydrodynamic coupling forces between an arm and a vehicle can degrade the end-point positioning performance of an underwater robotic system in a major way.

The goal of the control research presented in this chapter is to develop an approach to take advantage of manipulator model information in the control of a system, composed of an arm mounted on an underwater vehicle, to improve the station-keeping control of the vehicle, thereby enabling faster, more accurate positioning of the arm end point. By improving the control of the arm/vehicle system at this low level, future underwater robotic systems will be able to take full advantage of performance benefits offered by high-performance task-level human/machine interfaces.

This chapter begins with a discussion of pertinent research in the control of underwater vehicles and manipulators. The station-keeping control problem addressed in this research is then described. The concept of dynamically coordinated control of an arm/vehicle system is introduced. Next, the implementation of this control strategy on the OTTER vehicle is described in detail. Finally, experimental results using the coordinated-control approach are presented and compared with standard control strategies.
5.1 Background

Controlling underwater vehicles and robots to enable them to perform useful functions in the deep ocean represents a difficult problem that has challenged researchers for many years. Lead by the initial work of Yoeger and Slotine [42], the application of sliding-mode control techniques to underwater vehicles became an active area of interest [43, 44, 45]. The motivation for using the sliding-mode approach is to enable robust control of the uncertain nonlinear vehicle system. Other research has focussed on using adaptive or neural-network control methods to deal with uncertainty in the plant model [46, 47, 48, 49].

Several recent papers have addressed the modeling of underwater robotic systems [26, 25, 39]. The focus of these papers is the efficient simulation of underwater vehicles and manipulators with many degrees of freedom. These efforts are outlined in greater detail in Section 4.1 of the previous chapter. As with underwater vehicle control, much of the research in underwater manipulator control has focussed on the application of sliding-mode control methods and adaptive or neural-net control techniques. Westerman explored the application of neural networks to the problem of trajectory generation for an underwater manipulator. Unfortunately, only control strategies were presented with no experimental results [50]. In another effort, Liceaga [51] presented simulation results for a manipulator using sliding-mode control.

Only Yoeger and his colleagues have presented experimental results on the specific topic of underwater manipulator control [52]. Their research investigated specifically the compliance control of a manipulator actuator and in general the concept of how manipulators can be used to apply forces to objects in the environment. Yoeger's valuable research has one important emphasis, force control. The research of this dissertation has another, motion control, which is addressed in the present chapter.

In the underwater-vehicle community, two theoretical efforts are of direct relevance to the coordinated arm/vehicle control approach taken in this research. In the work of Mahesh, Yuh, and Lakshmi [53], an adaptive controller for coordinating vehicle and arm motion was proposed. The arm and vehicle were considered as a single unit and an adaptive controller was developed for the whole system. This required a discrete-time approximation to the full nonlinear arm/vehicle dynamics to be implemented in the control. The success of the approach is dependent on the controller's ability to adapt accurately to rapidly changing hydrodynamic coefficients. The approach has been demonstrated using only a computer
simulation of the planar motion of a vehicle. Experimental validation of the effectiveness of
their approach has not been demonstrated.

In a second, related effort, Koval [54] proposed a model-based feedforward control ap-
proach for the stabilization of an underwater manipulation robot. In this work, the com-
putational feasibility of a real-time hydrodynamic model implementation was addressed. Few
implementation details were provided. No simulation or experimental results were given.

Outside the underwater realm, relevant control research has been performed in the
Aerospace Robotics Laboratory at Stanford University. Koningstein investigated the dy-
namics and control of a two-armed free-flying space robot [55], and Vasquez considered the
problem of cooperative manipulation from a platform with unknown motion [56]. The work
of this dissertation differs in that while the complexity of multiple cooperating manipulators
is not yet considered, the hydrodynamic forces considered here are much larger and more
difficult to model than the inertial forces dealt with by Koningstein and Vasquez.

Unlike the underwater vehicle and manipulator references presented here, the focus of
this dissertation is not on adapting to existing uncertainty, but rather on improving sys-
tem performance by exploiting detailed knowledge of the system dynamics. While adaptive
approaches are often useful, premature reliance on learning control methods can result in
inferior performance due to a lack of physical understanding: Like conventional control
methods, adaptive or learning control schemes can only be improved by a strong under-
standing of the fundamental physical phenomena at hand.

The approach taken here involves augmenting the existing vehicle feedback control with
information based on a fundamental physical understanding of the manipulator hydrody-
namics, in a way that benefits the control of the entire system. To achieve good results,
this approach requires an accurate model of the manipulator hydrodynamics, which was the
goal of the research of Chapter 4. In this chapter, the coordinated-control approach using
the model of Chapter 4 is described and experimental validation is given.

5.2 Vehicle Station-Keeping Control Problem

A common work scenario for an underwater robot on a science mission is to perform a
manipulative task, such as picking up a rock, sampling a biological specimen from the
water column, or moving a science instrument, while holding the position and attitude of
the vehicle fixed or "on station." Such tasks are examples of jobs where the station-keeping
control of the vehicle is very important. By using control to keep the vehicle motionless, not only is the manipulation made easier, but the challenge of visually tracking the desired work scene is made less burdensome as well.

Station keeping is made difficult due to the presence of disturbances in the ocean environment, of which currents and tether forces are two. Another is the large hydrodynamic coupling force generated as the arm moves through the water. If precise positioning of the manipulator end point is required, even relatively small motions at moderate speeds can have significant degrading effects. It is that important problem that is addressed here.

For the coordinated arm/vehicle control experiments discussed in this chapter, a single-link arm was mounted on the OTTER vehicle, which is shown in Figure 2.5. Under the coordinated-control approach, the vehicle feedback controller is augmented with information about the hydrodynamic interaction forces between the arm and the vehicle. This information is produced in real time using a very accurate model (developed during this work) of the hydrodynamic forces acting on a single-link arm.

Although the single-link arm is quite simple mechanically, hydrodynamically it is quite complex due to the unsteady, three-dimensional flows developed as the arm moves. Thus, even though it does not possess all of the functionality of a full manipulator, the single link allows testing and validation of the coordinated-control concept.

For this work, it was necessary to develop a more advanced model, because many of the cogent hydrodynamic attributes of robotic motion (e.g. short unsteady swinging motions, radial-flow and tip-flow effects, etc.) had never been addressed by any previous model. As subsequent hydrodynamic models for manipulators increase in sophistication to handle accurately multiple links and degrees of freedom, the control approach presented here can be extended in a straightforward way to accommodate these systems.

The end-point positioning and station-keeping performance of different controllers was tested by commanding the vehicle to maintain station while moving the arm from one position to another and then back to the original position. Figure 5.1 shows a schematic representation of this manipulator positioning task. Precise point-to-point positioning of the manipulator end point requires high-performance control of the entire arm/vehicle system. Such control is difficult to achieve without compensating for the hydrodynamic interaction forces explicitly and precisely in the control of the total arm/vehicle system.
5.3 Approach

To address the problem of arm/vehicle dynamic coupling, a coordinated-control strategy is put forth and its underlying philosophy is discussed. A complete description of the implementation of this strategy on the OTTER vehicle is described. This section concludes with an explanation of the experimental test strategy used to verify the usefulness of the coordinated-control approach.

5.3.1 Dynamically Coordinated Control

The central idea of dynamically coordinated control is to take advantage of physical understanding of system dynamics explicitly in the control of the arm/vehicle system. In the context of the control problem addressed here, this physical understanding is embodied in an accurate model of the manipulator hydrodynamic forces. Under the coordinated-control approach, hydrodynamic and inertial forces generated from the motion of the arm are modeled in real time as the motion progresses. Based on the predicted interaction forces, thrust
commands are sent to the thrusters to counteract the forces generated by the arm motion. In this way, the control of the arm and the vehicle are “coordinated.” The experimental results presented in this chapter demonstrate the validity of this coordinated-control approach for the arm and the vehicle.

Figure 5.2 shows a simplified schematic diagram of the coordinated-control strategy. The main control components are the hydrodynamic model, the arm controller, vehicle controller, the arm-trajectory generator, and the vehicle-trajectory generator. The manipulator hydrodynamic model was developed and discussed in detail in Chapter 4. The arm and vehicle controllers are discussed in the following sections.

![Diagram of coordinated control](image)

**Figure 5.2: Block Diagram of Primary Coordinated-Control Components**

This schematic shows a simplified block diagram for the coordinated-control approach. The primary components are the hydrodynamic model, the manipulator and vehicle controllers, and the manipulator- and vehicle-trajectory generators. The coordinated control is composed of the position feedback controllers (on the manipulator and vehicle) together with the decoupling control produced from the hydrodynamic model of the interaction between the manipulator and the vehicle.

For the station-keeping experiments of this dissertation, the vehicle-trajectory generator supplied zero-reference commands for each of the vehicle degrees of freedom. When the vehicle is held relatively motionless by the control, the contribution of the vehicle's motion to the hydrodynamic forces acting on the arm becomes negligible. For coordinated arm/vehicle tasks requiring vehicle motion, the effect of vehicle motion on the hydrodynamic forces on the arm would have to be considered explicitly.
The control approach presented here was developed assuming the availability of an accurate hydrodynamic model. The primary benefit of this model-based control approach was the performance increase achieved by effectively eliminating one of the main disturbances on the system. Using the modeling approach presented here to predict the hydrodynamic forces has the additional benefit of maintaining high reliability and low cost — no additional sensors are required.

5.3.2 Vehicle Feedback Control System

For the fairly low speeds characteristic of station-keeping operation, the motion of the independent degrees of freedom of the vehicle are very lightly damped. This is due to the signed-quadratic relationship between velocity and fluid drag. At very low speeds, the drag is almost non-existent. At high speeds the drag forces are extremely large. For the low speeds involved in station keeping, the $x$, $y$, and $z$ translational motions and the yaw motion can be modeled approximately as $1/s^2$ plants, while the pitch and roll motions can be modeled as lightly damped second-order systems. Further details of the open-loop dynamic characteristics of the OTTER vehicle are given in Appendix C.

To provide control over the individual vehicle degrees of freedom, classical proportional-integral-derivative (PID) feedback controllers were used for each of the quantities $x$, $y$, $z$, $\phi$, $\theta$, and $\psi$. State-of-the-art position and attitude sensors provided the measurements required to achieve high-quality feedback control for each of the vehicle degrees of freedom. The integral portion of the control was implemented so that it was active only when the desired vehicle velocity was zero. In this way, good steady-state error performance was achieved while preserving good transient response.

Figure 5.3 shows a schematic representation of the vehicle feedback control implemented on the OTTER vehicle. The control loop was implemented digitally with a 100 Hz update rate. However, $x$ and $y$ position information from SHARPS was available at only 2.5 Hz. Yaw information from the flux-gate compass was produced at 10 Hz.

For station-keeping operations, the vehicle-trajectory generator produces constant desired-position commands. These position commands are transformed into the vehicle-body frame for use by the controller. The vehicle controller takes in body-frame-referenced position and velocity measurements for the vehicle degrees of freedom and, based on the errors, produces a vector of three forces ($F^x_{cmd}$, $F^y_{cmd}$, $F^z_{cmd}$) and three torques ($\tau^x_{cmd}$, $\tau^y_{cmd}$, $\tau^z_{cmd}$) to be applied to the vehicle about its center of mass in the vehicle frame. Based on the thruster
configuration, a vector of eight thrust commands \( T_{cmd} \) for the vehicle can be calculated. This is done by recognizing that the \( T_{cmd} \) required to produce the desired \( F_{cmd} \) and \( \tau_{cmd} \) can be calculated from the relation

\[
\begin{bmatrix}
F_{cmd} \\
\tau_{cmd}
\end{bmatrix} = R_v T_{cmd}
\]

(5.1)

where \( R_v \) is the 6 \( \times \) 8 thruster matrix map that takes into account the position and orientation of the thrusters relative to the center of mass of the vehicle. Rearranging Equation 5.1 results in

\[
T_{cmd} = R_v^\dagger \begin{bmatrix} F_{cmd} \\ \tau_{cmd} \end{bmatrix}
\]

(5.2)
where $R^d_1$ is the pseudo-inverse\(^1\) of $R_v$. Since the configuration of the thrusters on the vehicle is known a priori, $R_v$ and $R^d_1$ can be computed beforehand without impacting the speed of the real-time implementation.

Past research has shown that good agreement between the commanded thrust, $T_{cmd}$ and the actual thrust produced by the thrusters can be achieved by using velocity feedback to control directly the thruster-motor velocity, and taking advantage of the relationship between thrust produced and the angular velocity of the output shaft squared [5]. It has been found that for the steady-thrust case that

$$T = k_\omega |\omega|.$$  \hfill (5.3)

In other words, the motor-velocity command required to produce a desired thrust command can be calculated from

$$\omega_{cmd} = k_T \text{sgn}(T_{cmd}) \sqrt{|T_{cmd}|}. \hfill (5.4)$$

Values of $k_T$ for each of the thrusters were determined from thruster-velocity measurements together with spring-scale measurements of thrust on the vehicle. From Equation 5.4, thruster-motor velocity commands can be determined for each of the thrusters. These commands are sent to the thrusters, causing the vehicle to move in the desired way.

Achievable feedback gains were limited in two ways. First, low update rates on $x$, $y$ and yaw limited the amount of proportional feedback that could be applied without causing an instability. In $x$ and $y$, derivative information was available only from differencing the 2.5 Hz SHARPS measurement, so it was difficult to get the good vehicle-velocity information needed for high derivative gains. Second, noise on the sensors (particularly yaw) resulted in noisy control signals that caused excessive damage and wear to thrusters when very high gains were used. These signals could have been filtered more, but this would have resulted in more phase delay, which would have had a destabilizing effect. Even with these sensor limitations, in the absence of arm motion, good positioning performance was achieved using PID control on the individual degrees of freedom of the vehicle. Appendix C provides further details regarding the closed-loop vehicle control and shows plots of the closed-loop station-keeping performance of OTTER’s individual degrees of freedom.

\(^1\)using the Moore-Penrose pseudo-inverse, $A^d = A^T (AA^T)^{-1}$
5.3.3 Arm Feedback Control System

For the single link, the in-line hydrodynamic forces, though nonlinear and somewhat uncertain, provided damping and stability to the arm dynamics. Because of the well-damped dynamic characteristics of the arm, high angular-position-feedback gains were achievable. Using straightforward implementation of classical control methods, very good position control was attained.

Figure 5.4 shows a schematic block diagram of the arm controller implemented for the experiments described in this dissertation. This implementation takes advantage of the 1 kHz, high-gain velocity-feedback controller internal to the motor electronics. By controlling the arm motor in “velocity control” mode, the arm actuator behaved as a velocity source.

230 Hz sample rate

Figure 5.4: Arm Feedback Control Block Diagram

This schematic shows the arm feedback control block diagram. The controller is implemented digitally at 230 Hz. The arm motor was velocity controlled by a 1 kHz, high-gain feedback controller implemented on the 68HC11 microcontroller, which is part of the motor commutation electronics.

A fifth-order trajectory generator was used to provide smooth desired commands to the arm-joint-position controller. Desired velocity commands direct from the trajectory generator were sent to the motor as feedforward signals. A proportional arm-joint-position feedback loop was closed around the internal arm-joint-velocity feedback loop to provide control of the arm joint angle $\alpha$. The sample rate was limited to 230 Hz by the achievable serial communication bandwidth between the VME cage and the 68HC11 microcontroller in the motor housing.
5.3.4 Coordinated-Control Implementation

As mentioned previously, the coordinated-control approach implemented here involved augmenting existing independent arm and vehicle controllers with information about their dynamic interaction to produce improved control of the system. Figure 5.5 shows a schematic representation of the coordinated-control approach implemented on the OTTER system. The coordinating information between the two systems came from the manipulator hydrodynamic model and the decoupling thrust commands it generated.

![Coordinated-Control Implementation Block Diagram](chart)

Figure 5.5: Coordinated-Control Implementation Block Diagram

This schematic shows the block diagram of the coordinated-control scheme as implemented on the OTTER vehicle. The arm controller runs in one VME cage at 230 Hz, while the vehicle controller runs in a second VME cage at 100 Hz. The hydrodynamic model information is sent from the arm controller to the vehicle controller over an Ethernet connection at 60 Hz.
Implementation Assumptions

The essential pieces of information necessary for implementation of the single-link manipulator hydrodynamic model developed and described above are the position, velocity, and acceleration of the link relative to the water. In implementing this control approach for the station-keeping application, four key assumptions were made:

1. Vehicle motions are small and do not contribute significantly to the net motion of the manipulator relative to the water.

2. Desired arm-joint acceleration is a good approximation of the actual joint acceleration.

3. The water through which the arm is moving is still.

4. Transverse forces on the arm are insignificant in comparison to the in-line forces.

Under the condition that the vehicle and arm controllers are functioning as intended, the first two assumptions are reasonable and valid. The benefit of the first assumption is an increase in the control bandwidth due both to a reduction in computational complexity and to a decrease in the amount of information passed over the limited-bandwidth communication link between the arm and the vehicle. The benefit of the second assumption (which is standard for many model-based robot controllers) is that measurements of individual joint accelerations are not required. The third assumption is valid for the tests reported here: In the large MBARI tank, fluid motion is due solely to thruster discharge, which does not impinge on the arm. The validity of the fourth assumption has been demonstrated experimentally (see Section 3.3.2). For the apparatus used and the types of motions considered here, strong periodic vortex shedding does not become well-established, hence transverse forces are comparatively very small.

Figure 5.6 shows the hydrodynamic model performance under the assumptions outlined above. The agreement between the measured arm-joint torque and the modeled arm-joint torque was quite good. Some errors existed at the beginning and end of arm trajectories due to accelerations caused by joint flexibility which were not incorporated into the model when the desired acceleration signal was used. Errors during the middle of the trajectory (when the torque peaked) were due primarily to vehicle motions which were not included in the model. In spite of the simplifying assumptions made, it can be seen that good modeling accuracy was maintained.
Figure 5.6: Measured and Modeled Joint Torque on OTTER Vehicle.

This plot compares measured joint torque with the modeled joint torque when the arm was moved back and forth through ±45 degrees on the OTTER vehicle.

Implementation Description

Feedback controllers were applied to the arm and vehicle as described in the above sections. In addition, the hydrodynamic model and decoupling-control connection were added to complete the coordinated-control implementation as shown in Figure 5.5. To provide the rate information required for the model, the arm-position signal was pseudo-differentiated using a digital filter. Position information came directly from the motor encoder, while the desired acceleration signal was used to provide the required acceleration information.

The hydrodynamic model was implemented as described above. The output of the model was a vector of three forces \( F_{hyd}^x, F_{hyd}^y, F_{hyd}^z \) and three torques \( \tau_{hyd}^x, \tau_{hyd}^y, \tau_{hyd}^z \) acting about the base of the arm joint. These were the forces and torques required to counteract the forces generated by the motion of the arm. As with the feedback control, a thruster
configuration map was used to determine the required decoupling-control commands to the thrusters, \( T_{dcpl} \) to counteract the hydrodynamic coupling forces.

\[
T_{dcpl} = R_a^t \begin{bmatrix} F_{hyd} \\ T_{hyd} \end{bmatrix}
\]

(5.5)

\( R_a \) was the \( 6 \times 8 \) thruster matrix map that took into account the position and orientation of the thrusters relative to the base of the arm.

The vector \( T_{dcpl} \) of decoupling thruster commands was sent to the vehicle controller 60 times per second over an Ethernet connection between the arm and vehicle card cages. These decoupling commands were summed directly with the feedback commands, \( T_{feedback} \), to produce the total thrust command to be sent to the thrusters, \( T_{cmd} \).

As shown in Figure 5.5, the model-based decoupling control implemented here is calculated based upon the desired arm acceleration and upon measurements of arm position and velocity. As such, the output of the decoupling controller is a combination of nonlinear, model-based feedforward and feedback control (though not error based). This nonlinear feedforward and feedback combination is used to “cancel” the undesirable nonlinear coupling between two components of the system. In this regard, it is similar computed-torque control [57], which uses model-based feedback and feedforward loops to linearize and decouple the controlled system.

Combining vehicle position and attitude feedback control with decoupling control, as depicted in Figure 5.5, results in a vehicle controller that possesses the positive attributes of both types of control. Position feedback control provides regulation capability, robustness to disturbances, and robustness to plant model uncertainties. However, feedback control is inherently error-based. This implies that a position error must exist before the controller does anything in response. In this situation, a predictive model providing decoupling control commands to cancel undesirable dynamic interactions is very useful. Rather than waiting for a position error to build up, the decoupling control predicts what the control command should be to cancel the error-generating interactions in the system before the errors occur.

The limitation of the decoupling control implemented here is that since it was designed to cancel a particular interaction, it (alone) does not reject errors due to unknown disturbances or uncertainty in the plant model: Its nonlinear feedforward and feedback loops are only active when the arm moves, and even when active they do not provide the position and attitude regulation capability required for precise control of an underwater vehicle system.
For the problem considered here, a balanced combination of decoupling control and position feedback control provides the best solution.

### 5.3.5 Experimental Test Strategy

To determine the value of the proposed coordinated-control strategy, four different vehicle controllers were implemented and tested. In each of the four evaluated vehicle controllers, the arm control used was identical (see Figure 5.4). The four different controllers evaluated are shown in Figure 5.7.

**No Vehicle Control** In this case, both the decoupling-control path from the arm hydrodynamic model and the vehicle feedback control loop were open. With no control active, the effects of arm motion on the uncontrolled vehicle dynamics were observed.

**Feedback Control Only** In this control configuration, the vehicle-feedback-control loop was closed while the decoupling-control loop from the arm model to the vehicle remained open. In this case, the effects of arm motion on the closed-loop vehicle dynamics were seen, and the disturbance rejection capabilities of closed-loop control were demonstrated. The feedback control design implemented for these experiments was the same as is used on a daily basis on the OTTER vehicle.

**Decoupling Control Only** In this implementation, the vehicle-feedback-control loop remained open while the decoupling path from the hydrodynamic model to the vehicle was closed. Using this configuration, the effectiveness and accuracy of the decoupling control application were determined.

**Feedback with Decoupling Control** In this case, both the vehicle-feedback-control loop and the decoupling-control loop from the arm model were closed. In this control configuration, the performance benefits of combining the decoupling control, which provides predictive coordination between the motion of the arm and control of the vehicle, with the vehicle feedback control, which provides robustness to disturbances and uncertainty, were tested.
Figure 5.7: Block Diagrams of Four Controllers Tested
This schematic shows the block diagrams of the four different controllers implemented on the OTTER vehicle. In all four cases, the arm feedback control was operational. In the No Vehicle Control case, both the decoupling path and the feedback loop on the vehicle were open. In the Feedback Control Only case, the decoupling path was open, while the feedback path was closed. In the Decoupling Control Only case, the feedback loop was open, while the decoupling path was closed. In the Feedback with Decoupling Control case, both the decoupling loop and the feedback loop were closed.
The Feedback Control Only test case, along with the No Vehicle Control case, provided performance baselines against which the Feedback with Decoupling Control approach were compared.

5.4 Experimental Results

This section presents experimental results from the coordinated arm/vehicle control portion of this research. Data from two different types of tests are presented — multiple-swing motions and single-swing motions. In Figures 5.8 through 5.12, 5.15 through 5.21, and 5.25, data are shown from experiments where the arm was swung back and forth multiple times between ±45 degrees. In Figures 5.13, 5.14, and 5.22 through 5.24, data are presented where the arm was slewed a single time from one point to the other.

Several different types of data are presented including image sequences from video footage of the experiments, arm-angle tracking data, arm end-point error data, vehicle error regulation data, and thruster-usage data. Using these results comparisons are drawn between the different controller types. The results demonstrate the benefits of both feedback control and decoupling control and their complementary attributes that result in the best control behavior when both feedback and decoupling control are combined.

5.4.1 Video Image Sequences

Figures 5.9 through 5.12 show image sequences taken from video footage shot during arm/vehicle control experiments with the OTTER vehicle. In each sequence, the images were taken at approximately one-second intervals during the first three swings of the arm in a multiple-swing sequence. Figure 5.8 shows a typical time history of the arm joint angle for the multiple-swing motions of the arm. The image sequences give a qualitative feel for the performance of the different controllers.

No Vehicle Control  Figure 5.9 shows images taken for the No Vehicle Control case. It can be seen that the open-loop roll mode of the vehicle was excited by the motion of the arm. Errors in roll were as large as 18 degrees in both directions from the horizontal. A significant error in yaw can also be observed. During this sequence, the vehicle drifted about 15 degrees in yaw from its initial heading. These images demonstrate that the hydrodynamic coupling forces involved in moving an arm at moderately fast speeds were very large and that (if not
**Figure 5.8: Arm Joint-Angle Time History**

This figure shows the arm joint angle versus time for the multiple-swing arm motion results shown in Figures 5.9 through 5.15.

coped with) they have a significant degrading effect on the station-keeping capability of a small, agile vehicle such as OTTER.

**Feedback Control Only** When compared with the No Vehicle Control case, the benefits of position and attitude feedback control, as shown in the sequences of Figure 5.10, are readily apparent. Errors in yaw and roll were reduced, but still very significant. The closed-loop roll mode, although much more damped than the open-loop mode, was still excited by the arm motion. Roll motions were as large as nine degrees in both directions. The yaw angle of the vehicle varied as much as eight degrees from its nominal position. While the benefits of closed-loop control are obvious from this sequence of images, the disturbances introduced from the arm/vehicle coupling still resulted in substantial deviations in the vehicle's position and attitude.

**Decoupling Control Only** The sequence of Figure 5.11 illustrates the performance of the controller in the Decoupling Control case. It can be seen that the influence of the arm motion on the vehicle was greatly reduced. Errors in roll and yaw were noticeably smaller.
Figure 5.9: Arm/Vehicle Response | No Vehicle Control

For this sequence, all control commands to the thrusters were disabled. The vehicle rolled as much as 18 degrees in both directions from its nominal horizontal position. In yaw, the vehicle rotated about 15 degrees from its initial heading angle.
Figure 5.10: Arm/Vehicle Response — Feedback Control Only

In this sequence, the vehicle was under position and attitude feedback control. Roll errors were as large as nine degrees in both directions, while yaw errors were as large as eight degrees.
Figure 5.11: Arm/Vehicle Response — Decoupling Control Only

For this sequence, vehicle feedback control was disabled, decoupling control was activated. Roll errors were less than three degrees. Yaw errors were kept below four degrees. Vehicle position drifted slightly due to lack of feedback.
Figure 5.12: **Arm/Vehicle Response — Feedback with Decoupling Control**

In this sequence, both decoupling and feedback control were activated. Roll errors were held below two degrees. Yaw errors were less than three degrees.
Since the application of the decoupling control did not perfectly cancel the interaction forces, the open-loop roll mode was excited slightly by the combination of arm and thruster forces acting.

**Position Feedback and Decoupling Control Combined**  Figure 5.12 shows the performance results obtained using the Feedback with Decoupling Control approach. As with the Decoupling Control Only case, the hydrodynamic interaction forces were largely cancelled by the decoupling component of the control. The advantage of adding the position and attitude feedback control was that the remaining errors from the inexact decoupling control were further reduced by the error regulation of the feedback control. With the addition of position feedback control, robustness to system uncertainty was provided, and the tendency of the vehicle to drift off station was eliminated. These benefits are demonstrated clearly in a more quantitative way in the following sections.

### 5.4.2 Arm End-Point Positioning Performance — Single Swing

While not sensed and controlled directly in these experiments, arm end-point position error is the most useful indicator of the quality of the performance of the arm/vehicle controller. For the results presented here, end-point position data were generated using measurements of the vehicle position and attitude combined with measurements of the arm joint angle. These measurements were applied to the known position kinematics of the system to generate a measure of the end-point position. Based upon accuracies of the different sensors, the end-point position data are accurate to within 4 cm. As suitable methods for direct sensing of the manipulator end point are developed for the underwater environment, the coordinated-control strategy developed here can be extended in a straightforward way to enable further improvements in end-point control that direct end-point sensing will allow. A full description of the end-point kinematics is given in Appendix D.

In evaluating end-positioning performance from the single-swing tests, two different types of data are examined. First, in Figure 5.13 settling-time performance data are presented for each of the four different controllers. Second, in Figure 5.14 time histories of end-point errors for a single sweep of the arm are shown.

**Arm End-Point Settling Time**  Figure 5.13 shows plots of the distance of the arm end point from the desired target for the four different controllers. For the 90-degree slews
considered, the distance traveled by the end point was about 1.5 m. Here settling time is
defined as the time required to stay within five percent (of the total distance traveled) of
the target — in this case ±7.5 cm.

![Graphs showing Arm End-Point Settling Time](image)

**Figure 5.13: Arm End-Point Settling Time**

This figure gives an indication of the settling-time performance of the different controllers. Here, settling time is defined as the time required to settle within five percent (of the total distance traveled) of the desired target. Note that without feedback control, the end point doesn’t come within ±5% of the target. Settling times are approximately 6.5 seconds for the Feedback Only case and 2 seconds for the Feedback with Decoupling case.

These settling-time plots demonstrate quite strikingly the importance of the feedback component of the vehicle control: Without feedback, the arm end point either fails to come within five percent of the target (as in the No Vehicle Control case) or it fails to remain within the five-percent error bound around the target point (as in the Decoupling Control Only case).

In the Feedback Control Only case, the time required to settle to within five percent of the target was about 6.5 seconds. As the arm moved, significant errors in roll, yaw, $x$, and
y resulted. Coming and staying within the error bound required these errors to be reduced, which took a substantial amount of time.

In the Feedback with Decoupling Control case, the observed settling time was about 2 seconds. This represents an improvement of over three times compared to the settling time of the Feedback Control case. Because the vehicle stayed on station, the settling time corresponded directly to the duration of the slew.

**Arm End-Point Error** In Figure 5.14 end-point errors for the Feedback Only and Feedback with Decoupling controllers are compared for a single-swing motion of the arm. Single-swing motion results are instructive since they allow the arm/vehicle system response to be observed in the period of time both during and following the occurrence of the interaction forces. In this way, the benefits of predicting the interaction disturbance and applying decoupling control to prevent the build up of errors can be seen. The upper plot in Figure 5.14 shows, for reference, the 90-degree, two-second trajectory followed by the arm for the single-sweep motion case. The lower plot shows the end-point error histories for the two controllers.

In both the Feedback Only and Feedback with Decoupling cases, the initial end-point error spike that occurred during the arm transient was due to the arm joint-angle tracking error. After the completion of the arm motion, the end-point error quickly settled to the one to three centimeter range in the Feedback with Decoupling case. This error resulted from both arm joint-angle errors and vehicle position and attitude errors.

In the Feedback Only case, the end-point error remained quite large (approximately 20 cm) in the seconds following the completion of the arm slew. This error was largely due to yaw and roll errors in the vehicle's attitude. As the feedback controller responded to the errors caused by the interaction forces, the error was reduced. Because the bandwidth of the feedback control was relatively slow (due to the large mass of the vehicle), the end-point errors were still significant 15 seconds after the slew had ended. For the 20-second period shown, the mean end-point error for the Feedback Only case was 11 cm, while the mean error for the Feedback with Decoupling case was only 3.5 cm — over a factor of three reduction.
Figure 5.14: Arm End-Point Error — Single Swing

This plot shows time histories of the end-point errors for a single sweep of the arm. The mean error for the Feedback Only case was 11 cm, while the mean error for the Feedback with Decoupling case was 3.5 cm.

5.4.3 Arm End-Point Positioning Performance — Multiple Swings

For the multiple-swing motions of the arm considered, Figure 5.15 shows time histories of the arm end-point errors for the different controllers tested. It can be seen in the No Vehicle Control case that the end-point errors were largest, having a mean value of 28 cm. Errors were due both to the excited vehicle dynamics and to the drift of the vehicle from its desired station. In the case of Feedback Control Only, the drift errors were reduced, but the closed loop dynamics of the vehicle (particularly in roll and yaw) were excited. The net result was noticeable errors in end-point position (11 cm mean error). In the Decoupling Only case, oscillatory errors due to excited dynamics were reduced, but the vehicle tended to drift unopposed, causing sustained errors in the end-point position (9.1 cm mean error).

As expected, the best end-point error regulation was achieved in the Feedback with Decoupling Control case. In this case, the errors due to vehicle drift and excited vehicle
dynamics were both reduced to near zero, resulting in very good end-point error control with a mean error of only 4.6 cm. It can be seen that with combined decoupling and feedback control, that the end-point errors were reduced by a factor of six when compared with the No Vehicle Control case and a factor of 2.4 when compared to the Feedback Control Only case. In the Feedback with Decoupling Control case, where end-point errors were smallest, a more significant portion of the error can be attributed to the arm joint-angle error. During the arm slews, joint tracking errors were typically around 2 to 3 cm. In this particular application, errors in roll and yaw were the largest contributors to the arm-end-point error, especially in the cases of No Vehicle Control and Feedback Control Only.

![Graphs showing end-point error over time for different control cases.](image)

Figure 5.15: Arm End-Point Error — Multiple Swings

This plot shows a time history of the arm end-point error corresponding to the arm motion shown in Figure 5.8. Mean errors were as follows: No Vehicle Control—28 cm, Feedback Only—11 cm, Decoupling Only—9.1 cm, Feedback with Decoupling—4.6 cm.

5.4.4 Vehicle Error Regulation

The video image sequences of Section 5.4.1 showed dramatically the qualitative differences between the several control strategies. In this section, data that quantitatively demonstrates
the differences between the strategies are shown. This section presents plots comparing the error regulation performance of the controllers in each of the six vehicle degrees of freedom. Figures 5.16 through 5.18 present roll, yaw, and pitch data, while Figures 5.19 through 5.21 show data for the $x$, $y$, and $z$ degrees of freedom. The data presented here give an indication of the relative performance of each of the controllers in maintaining the station of the vehicle.

**Vehicle Attitude Errors**  Figure 5.16 shows time histories of vehicle roll error data for each of the four controller types considered. With no control effort, roll errors were very large (between ±18 degrees) as the open-loop roll mode of the vehicle was excited. The addition of attitude feedback control improved the roll error regulating performance, but the errors were still significant (between ±9 degrees). Decoupling control alone effectively countered much of the roll moment generated from the arm motion. In this case, decoupling control allowed the vehicle to respond to arm interaction forces before significant attitude errors were induced. Further improvement was realized when both decoupling and feedback control were combined together. Peak roll errors were always less than 1.5 degrees in this case.

Similarly, time histories of vehicle yaw error for the different controllers are shown in Figure 5.17. When no feedback or decoupling control was applied, the vehicle heading angle drifted significantly from its nominal position. Unlike the roll and pitch attitude degrees of freedom, the yaw degree of freedom has no passive restoring force inherent to its open-loop dynamics. Because of this, the yaw degree of freedom was fully dependent on feedback control to prevent drifting due to disturbances or uncertainty in the plant model. When feedback control alone was applied, the tendency to drift was reduced, but the closed-loop dynamics of the yaw controller became apparent. As the controller attempted to reject the yaw disturbance, it caused the vehicle to oscillate significantly in response (up to 9 degrees error). When decoupling control only was applied, the yaw disturbance due to arm motion effectively was cancelled resulting in much smaller yaw errors. With both feedback and decoupling control combined, the yaw errors were again small. Yaw errors were roughly three times smaller for the Decoupling Only and Feedback with Decoupling cases than for the Feedback Only case.

For the OTTER vehicle with the arm mounted in its present configuration, the pitch attitude was relatively unaffected by disturbances from the arm motion. Figure 5.18 shows
Figure 5.16: Vehicle Roll Error Versus Time

Vehicle roll error is plotted for each controller type. The corresponding arm motion is shown in Figure 5.8. In each case, the roll mode of the vehicle was excited by the arm motion. The effects of this excitation were reduced significantly by the decoupling control.

time histories of the vehicle pitch errors incurred for the four controllers. Even in the No Vehicle Control case, the pitch errors were quite small when compared with the roll and yaw errors. Under the other combinations of decoupling and feedback control, the errors were even smaller and did not contribute in a large way to the arm end-point errors.

Vehicle Position Errors When compared to the roll and yaw degrees of freedom, the contributions of the vehicle x-y-z degrees-of-freedom to the total error at the tip of the arm were relatively small. That is not to say, however, that they were completely insignificant. Because similar trends were observed between each of the translational degrees of freedom, they will be discussed together. Figures 5.19, 5.20, and 5.21 show the x, y, and z responses respectively.

When no control was applied, the vehicle translates significantly in response to the arm motion — 12 cm in x, 8 cm in y, and 3 cm in z. For the x and y degrees of freedom,
the application of feedback control alone helped correct the oscillations caused by the arm motion. In both $x$ and $y$, the response of the feedback controller to the coupling disturbances can be seen. In $z$, the application of feedback actually caused the error to increase slightly (although it is still quite small). This was probably due to large thrusts being applied by the vertical and lateral thrusters while the vehicle was rolled significantly. Errors in the thruster map or the application of the commanded thrust probably resulted in errors in the net thrust applied in the $z$-direction.

When decoupling control only was applied, the dynamic interaction effects of the arm motion were reduced significantly, however some drift of the vehicle position occurred due to the small errors in the cancelation of the interaction forces by the decoupling control. This problem was remedied once again by combining the decoupling and feedback control components: The resulting translational errors were smallest when both decoupling and feedback control were applied.
Figure 5.18: Vehicle Pitch Error Versus Time

This figure shows pitch error for each controller type tested. The corresponding arm motion is shown in Figure 5.8. The pitch mode of the vehicle was not significantly affected by the arm motions considered. Errors were relatively small in all cases.

Summary For each of the degrees of freedom of the vehicle, the best error regulation results were obtained when decoupling and feedback control were combined. The decoupling control effectively cancelled most of the dynamic coupling between the arm and the vehicle. Much of the remaining coupling effect was eliminated by the feedback control. The feedback control also provides for the rejection of other disturbances not present in these tests (e.g., tether forces, currents) and robustness to uncertainties in the plant.

5.4.5 Thruster Usage

In this section, a comparison of the control effort used by the Feedback and Feedback with Decoupling controllers is given (see Figures 5.22 and 5.23). These data demonstrate that the performance improvements illustrated above came about largely from an intelligent application of control with only a moderate increase in the total control applied.
Figure 5.19: Vehicle x-Position Error Versus Time

Vehicle x-position error is plotted for each controller type. The corresponding arm motion is shown in Figure 5.8. While errors were relatively small for all cases where control was applied, they were smallest for the Feedback with Decoupling Control case.

Also in this section, a comparison of the control distribution between the feedback loop and the decoupling loop for the Feedback with Decoupling Control case is given. Figure 5.24 gives an indication of the total amount of decoupling and feedback thrust applied by all thrusters. Single-swing motions of the arm were used to generate the thruster data for this section (see Figure 5.14).

**Horizontal Thrusters**  Figure 5.22 shows thruster responses for each of the horizontal thrusters on the vehicle for the Feedback Only and Feedback with Decoupling cases. For each thruster, it can be seen that the Feedback with Decoupling thrusts spiked up to comparatively large values during the slew, but then settled to very small values almost immediately after the slew ended. On the other hand, for the Feedback Only case it can be seen that the thruster commands were much smaller initially, but that the thrusters fired over much longer durations. Long after the thrusters had essentially shut off in the
Feedback with Decoupling case, they continued to fire in the Feedback case in an effort to regulate the errors in the system. These plots illustrate the predictive capability of the decoupling control, which in each case leads the feedback control in responding to the arm motion.

Note that the largest thrust command by far came from the lateral-fore thruster. This makes sense physically since the lateral-fore thruster was positioned very close to the base of the arm joint and the lateral forces generated by the arm were very large. The responsibility for countering these forces fell largely upon the lateral-fore thruster.

**Vertical Thrusters**  Figure 5.23 shows thruster responses for each of the vertical thrusters on the vehicle for the Feedback Only and Feedback with Decoupling cases. As with the horizontal thrusters, the vertical Feedback with Decoupling thrust commands spiked up to large values during the arm motion to compensate for the interaction forces generated.
Figure 5.21: Vehicle z-Position Error Versus Time

Vehicle z-position error is plotted for each controller type. The corresponding arm motion is shown in Figure 5.8. Errors were smallest for the Feedback with Decoupling Control case.

Unlike the horizontal thrusters, the vertical thrusters responded fairly quickly to the roll errors introduced in the Feedback Only case. Notice that in both control cases the vertical-aft thrusters remained on well after the completion of the arm motion. This was due to the integral control acting to zero the pitch error of the vehicle.

Comparison of Thruster Usage  Two useful indicators of thruster usage are peak thrust applied and the total thrust\(^{2}\) applied over the 20-second duration shown. Peak thrusts give a good indication of worst-case instantaneous demand on the thrusters, while total thrust usage gives a better indication the total thrust (and hence battery power) required over the duration of the test. For the thruster data shown in Figures 5.22 and 5.23, the peak thrusts were 1.8 to 2.7 times larger in the Feedback with Decoupling Control case, with the exception of

\(^{2}\)Total thrust was calculated by summing the absolute values of the thrusts applied over the duration of the test, while subtracting out any DC components. This is analogous to evaluating the expression \(\int |T_{\text{cw}}| \, dt\).
Figure 5.22: **Horizontal Thruster Usage**

This figure depicts the thrust commands sent to the four individual horizontal thrusters for the motion shown in Figure 5.13. Thrusts applied in the Feedback with Decoupling case are much larger, but are applied earlier and with much shorter duration.

The lateral-aft thruster where it was five times larger. While the peaks are much higher due to the feedforward component of the command coming from the hydrodynamic model, the thrust durations were shorter for the Feedback with Decoupling case. This is reflected in the relative values total thrust used in each control case: Only five percent more total thrust was used in the Feedback with Decoupling case than for the Feedback Only case.

The key result of this comparison is that for a slight increase in total thrust used, a very substantial performance improvement was realized (see Figure 5.13). This was accomplished by taking advantage of knowledge of the system dynamics and coordinating the control in a sensible way.

**Thrust Distribution** To understand how control effort was distributed between the decoupling path and the feedback loop in the case of Feedback with Decoupling Control, the total decoupling and feedback thrusts were computed and plotted. Figure 5.24 shows a time history of the total decoupling and feedback thrusts for a single-swing motion of the arm.
Figure 5.23: Vertical Thruster Usage

This figure depicts the thrust commands sent to the four individual vertical thrusters for the single-swing motion shown in Figure 5.13. Thrusts applied in the Feedback with Decoupling case are significantly larger during the transient phase. After the transient, both vertical-aft thrusters remain on due to integral control on the pitch error.

Total decoupling thrust was computed by summing the absolute value of the components for the decoupling command vector ($T_{\text{dcp}}$ in Figure 5.5) at each instant of time. Similarly, the total feedback thrust was computed by summing the components of the $|T_{\text{feedback}}|$ vector.

During the slew of the arm, the decoupling component of the signal spiked up as the hydrodynamic model predicted the interaction forces acting. Prior to and after the slew, the decoupling command was not quite zero. This was due to the incorporation of hydrostatic forces, caused by the slight buoyancy of the arm, into the decoupling control. It can be seen that the feedback control component was always non-zero and varied only slightly. The feedback was always non-zero due to the integral control acting on the independent degrees of freedom of the vehicle. The feedback varied only a little because the interaction forces were cancelled effectively by the decoupling control, and the errors remained small.
Figure 5.24: Thrust Distribution — Feedback with Decoupling Control

This figure illustrates how thrust is distributed between the decoupling loop and the feedback loop for the Feedback with Decoupling Control case. Shown are the sums of the absolute values of the thrusts applied to the individual thrusters for the decoupling loop and the feedback loop. The corresponding arm motion is shown in Figure 5.14. During the motion of the arm, the control effort is dominated by the decoupling portion. After the transient is over, the control is due almost completely to the feedback loop. A small amount of decoupling remains to counter the hydrostatic forces.

5.4.6 Possible Control Error Sources

While the above sections illustrate that the coordinated-control approach worked very well, it is also obvious that it did not perfectly cancel the effects of the hydrodynamic interaction between the arm and the vehicle. The purpose of this section is to discuss briefly the small errors that did exist in the implementation of the decoupling portion of the control, just as a matter of understanding the process.
In the implementation of the coordinated-control approach, there were three primary sources of error:

- **Hydrodynamic Modeling Error** — specifying the wrong desired thrust to counteract the hydrodynamic forces generated by the arm.

- **Thruster Calibration Error** — given a correct desired thrust, calculating the wrong desired thruster-motor velocity.

- **Thrust Implementation Error** — given a correct desired thruster-motor velocity, not being able to achieve it immediately and exactly due to thruster-velocity-control errors or given that the desired thruster-motor velocity is achieved, unmodeled dynamics cause the wrong thrust to be applied.

Each of these error sources is discussed briefly in the following paragraphs.

**Hydrodynamic Modeling Error**  Figures 4.10 through 4.12 of Chapter 4 demonstrates that the hydrodynamic model developed in this research does a very good job of predicting the hydrodynamic torques acting. Since actual position, velocity, and acceleration measurements of the arm’s motion were used in the hydrodynamic model, the errors in Figures 4.10 through 4.12 were due either to uncertainty in the coefficients or to errors in the model form itself. While it is expected that some modeling errors of this type will always exist (due to the extreme sensitivity of the forces to the initial conditions of the flow), it is important to note that they were relatively small in the results presented here.

Figure 5.6 shows modeling results with the arm mounted on the vehicle. The desired acceleration signal was used and the effects vehicle motions were not incorporated into the model. Even with these assumptions, the model agreement was quite good.

**Thruster Calibration Error**  The thrust data used for calibrating the thrusters were obtained under steady thrust conditions. The thrusters were given a constant velocity command and the thrust acting on the vehicle was measured using a spring scale. Using this method, measurements were accurate to approximately ±2 N. The thrusters were calibrated on the vehicle and were not calibrated independently. For example, the two drive thrusters were calibrated at the same time by measuring the thrust produced by both thrusters for a given velocity command and then dividing by two — essentially assuming that the
two thrusters have the same thrust characteristics. While generally accurate, this is not completely true, as each thruster has its own unique behavior.

**Thrust Implementation Error** The thruster motors on the OTTER vehicle were velocity controlled. From the inverse thruster model (obtained from the calibration data), motor-velocity commands to the thrusters were produced. Some errors were introduced into the system by the thrusters' inability to perfectly track the desired thruster-motor-velocity command. As an example of typical tracking performance, Figure 5.25 shows commanded and actual thruster velocities for the vertical thrusters. It can be seen that for all but the vertical-aft-port thruster, the velocity tracking performance was acceptable. This thruster was running into its saturation limits. Notice that for all of the thrusters, there was a slight lag (approximately 0.15 seconds) between the commanded and actual velocities.

![Velocity Tracking Performance](image)

**Figure 5.25: Thruster Velocity Tracking Performance**

*This figure shows velocity tracking performance for each of the vertical thrusters. Each thruster had a high-gain velocity feedback loop running on a microcontroller contained in the thruster housing. These plots illustrate that tracking performance was very good for the vertical thrusters with the exception the vertical-aft-port thruster, which entered saturation at higher velocity commands.*

In addition to velocity tracking errors, errors due to unmodeled or uncertain thruster dynamics can result in further errors in the thrust implementation. As mentioned above,
dynamics are not included in the inverse thruster model used to calculate the commanded thruster velocities.

In summary, while the overall performance of the system was very good from vehicle station-keeping and end-point-error and settling-time standpoints, it could be further improved by the elimination of modeling and thrust implementation errors. The greatest improvement could be realized if the thrusters were actually thrust controlled rather than velocity controlled. This would effectively eliminate thruster calibration and thrust implementation errors. A more accurate thrust implementation would reduce the vehicle motion, improving the accuracy of the hydrodynamic model as well. While direct control of thrust is most desirable, the issues of complexity, reliability, and accuracy in sensing the thrust for control would have to be addressed.

5.5 Summary

In this chapter, the coordinated-control approach taken for the control of an underwater arm/vehicle system has been described in detail. Relevant background information has been presented along with detailed discussion of the control system that was implemented and tested in a full system. A key underlying contribution to this control approach has been the development of accurate hydrodynamic modeling for an important arena of dynamic motion that had never before been addressed, theoretically or experimentally.

Experiments demonstrated that the hydrodynamic coupling forces between an arm and a vehicle can be very significant, resulting in large disturbances to the station-keeping control of the vehicle. Experimental results showed that substantial performance improvements can be realized in the control of an underwater arm/vehicle system by using the control techniques developed here. Tracking errors at the manipulator end point were typically reduced by a factor of 2.5 using the coordinated-control approach, when compared with the standard approach of independent arm and vehicle feedback controllers. Using the approach presented here, settling times at the manipulator end point were also reduced by over three times when compared to the values obtained using vehicle-position feedback control alone.

It is important to note that these dramatic performance improvements were achieved with only a five-percent increase in total control effort.
Chapter 6

Conclusions

This final chapter consists of two sections. The first section summarizes the results of the research presented in this thesis. The second section gives recommendations for future research in the control of underwater manipulation systems.

6.1 Observations and Summary of Results

This research has addressed the modeling of hydrodynamic forces acting on an underwater manipulator with specific application to the high-performance coordinated control of an underwater arm/vehicle system where hydrodynamic coupling between the arm and the vehicle were significant. The contributions of this work are detailed in Section 1.6. This research has led to the conclusions that follow.

6.1.1 Hydrodynamic Forces on Underwater Manipulators

Unique Characteristics  From a hydrodynamic point of view, underwater manipulators and their motions have a number of unique attributes that affect the flow over the manipulator as it moves through the water, thereby affecting significantly the forces acting during the motion. Transient rotational (swinging) acceleration through small angles of a relatively short cylinder is not well understood or characterized in the literature. The effects of these attributes were addressed directly in this work.

Transverse Forces  The focus of this work was primarily on the modeling of in-line forces (in the direction of motion). It was found that transverse forces, generated from the
shedding of vortices, are minimal as long as the conditions of lock-in are avoided. Lock-in occurs when the vortex shedding frequency approaches the frequency of any lightly-damped structural resonances in the system. Under this condition, the vortex shedding frequency synchronizes (locks-in) with the resonant frequency, resulting in self-exciting oscillations in a direction transverse to the motion of the arm.

For the experiments of this thesis, the lowest structural resonant frequency was typically several times higher than the dominant shedding frequency, so lock-in did not occur, and lift forces therefore were very small compared to the in-line forces. However, for long, sweeping motions of long, slender arms with low-frequency, lightly-damped vibratory modes, transverse forces can be extremely significant — often larger in magnitude than the in-line forces.

**In-Line Forces** In-line hydrodynamic forces are those acting in the plane of the arm's motion. The two-dimensional theoretical analysis of the forces on a circular cylinder undergoing a constant acceleration motion showed that the in-line force is composed of two terms. The first term is the product of the acceleration, the mass of the water displaced, and an added-mass coefficient. The second term is the product of the frontal area, the density, the velocity squared, and a drag coefficient. The theoretical analysis also showed that the added-mass and drag coefficients are state-dependent functions of how far the cylinder has traveled since the motion began.

The state-dependent behavior of the coefficients was determined experimentally from measurements of force, torque, position, velocity, and acceleration using an approach developed in this research that allows simultaneous identification of the drag and added-mass coefficients. The correctness of the state-dependent characteristics of the coefficients identified was reinforced by flow-visualization experiments where the development of the wake was examined.

From measurements of the in-line force taken at different locations along the span of the arm, it was found that the behavior of the local drag and added-mass coefficients varied from one location to another. This was due to the three-dimensional nature of the flow over the arm, and was validated by the flow visualization experiments that showed flow along the span of the arm and flow around the tip which was significant. This 3-D flow alters the pressure distribution around the arm, thereby affecting the values of the coefficients at different locations along the arm.
CHAPTER 6. CONCLUSIONS

An unanticipated benefit of the span-wise flow and flow around the tip was that the drag forces acting were reduced. The average value of the drag coefficient ranged from just 0.75 to 0.8 compared to values from 1.1 to 1.2 that would be expected for a long cylinder undergoing steady, rectilinear motion. This reduced drag is a direct result of the flow along the span and around the tip, which acts to relieve the low-pressure area along the trailing edge of the arm (which is the primary contributor to drag).

The hydrodynamic modeling effort culminated in the synthesis of a sound semi-empirical model that predicts the hydrodynamic in-line torque acting on a swinging circular cylinder with excellent agreement with experimental torque measurements over a wide range of motions of the single link. The accuracy of the model presented in this thesis represents a major improvement over models presented in the literature.

The keys to successful modeling were careful consideration of the attributes of manipulators and their motions, sound theoretical development, and thorough experimentation. The modeling work presented in this dissertation is especially significant because it represents the first experimental investigation of hydrodynamic forces acting on an underwater cylinder swinging about one end (such as a manipulator). Careful examination of the single-link manipulator case signifies an important first step in understanding the hydrodynamics of a multi-link, multi-degree-of-freedom manipulator.

6.1.2 High-Performance Control of an Underwater Arm/Vehicle System

Significance to Control of Hydrodynamic Forces In contrast to inertia and friction forces, hydrodynamic forces are extremely large, and therefore are most significant in the control of underwater manipulators and vehicles. Experimental results presented in this dissertation have shown that for small, maneuverable vehicles, motions of the arm can result in large disturbances to the vehicle, causing it to move from its desired station.

With the OTTER vehicle under closed-loop position control, 90-degree, two-second sweeps of the arm caused the vehicle to roll up to nine degrees and yaw up to eight degrees from the desired attitude. These perturbations resulted in errors as large as 30 centimeters at the arm end point. Even with shorter, slower motions, the disturbances are big enough to degrade the end-point positioning performance of the manipulator to an undesirable degree.

Coordinated Arm/Vehicle Control The underlying philosophy of the coordinated arm/vehicle control approach taken in this research was to incorporate information about
the motion of the arm in the control of the thrusters, so that the effects of hydrodynamic interaction force disturbances would be minimized. Information about the motion (position, velocity, and desired acceleration) was used in the hydrodynamic model of the arm to predict the interaction forces and torques acting between the arm and the vehicle. These interaction forces and torques were mapped and sent to the thrusters (combined with the feedback command) so that equal and opposite forces and torques were applied to counteract those generated by the motion of the arm. Ideally, the modeling, mapping, and thrust application would be perfect, and the vehicle would remain motionless — undisturbed by the motion of the arm. And indeed it was nearly so.

Results of experiments conducted in this research show that by incorporating information generated in real time by the arm hydrodynamic model into the vehicle control, the degrading effects of hydrodynamic interaction forces can be dramatically reduced. When compared to the case where the vehicle control was disabled, an improvement of over a factor of six in end-point positioning accuracy was realized when a combination of feedback and model-based decoupling control was used. A factor of 2.5 improvement in end-point positioning accuracy was seen when comparing conventional independent arm and vehicle feedback controllers with the coordinated-control approach presented in this work.

In addition to the improvement in positioning accuracy, a great improvement in the settling time of the manipulator tip was realized. Using the coordinated-control approach, the settling times were less than one third of those associated with the independent arm and vehicle controllers.

The dynamically coordinated arm/vehicle control research presented here represents the first experimental demonstration of the performance benefits of such an approach.

6.2 Recommendations for Future Work

As an initial experimental investigation into the hydrodynamics and control of an underwater manipulator system, this research has produced a number of new and exciting research directions for future researchers in this field. The following paragraphs briefly describe some of these possibilities.

**Hydrodynamic Modeling of Multi-Link, Multi-DOF Arms** The hydrodynamic research presented in this dissertation was limited to a single-link manipulator. To be most
useful to the underwater vehicle community, this work should be extended to address the hydrodynamics of multiple-link, multiple-degree-of-freedom arms. To be effective, this new research will need to progress incrementally, of course.

As a first step, it is recommended that a two-link arm with a fixed elbow (both links in the plane of motion) be considered. By locking the elbow at fixed angles, the effects of configuration on the behavior of the flow over the arm can be studied. The inclination angle of the second link will affect the flow along the arm and around the tip, as well as the shedding of vortices into the wake (an asymmetric vortex pair may develop for certain angles of inclination). Both in-line and transverse forces will be influenced by the configuration of the arm. An understanding of how flow over one link affects the flow over the other link will have to be addressed as well. Different flow "regimes" may result when the elbow is positioned at different angles.

With the fixed-elbow arm well understood, a second step could consider the planar motion of a two-link arm with a moving elbow joint. The additional complexity of a second degree of freedom would have to be considered, as well as the possibility of transitioning between different flow regimes.

In subsequent steps, these results could be generalized to multiple-link, multiple-degree-of-freedom manipulators cooperating in close proximity to each other and to objects and organisms they may choose to capture and move carefully through the water.

Motion through Moving Fluid Although the models developed in this research could be extended in a straightforward way to do so, the motion of an arm through a moving fluid was not explicitly considered. Two possible causes of fluid motion that may be significant are ocean currents and wakes from previous arm motions. In both situations, the difficulty lies both in determining how the fluid is moving, and in accounting for the fluid motion in the hydrodynamic model.

Hydrodynamics of Links of Non-circular Cross-sections Cylinders of circular cross-section have been considered in this work. Another common shape for a manipulator link would be a square or rectangular cylinder. For cylinders that are not axis-symmetric (as is a circular cylinder), the forces acting would be dependent on the orientation of the cross-section relative to the flow. An understanding, based upon experimental results, of how
CHAPTER 6. CONCLUSIONS

lift and drag forces would be affected for different angles of attack of the arm would be essential.

**Reduced-Drag Manipulator Design** As mentioned many times before in this dissertation, the hydrodynamic forces resisting the motion of a manipulator through the water are very large. Because of this, manipulator motions (especially high-speed motions) can consume a significant amount of power. This is undesirable for battery-powered underwater vehicles. It is therefore important, from a power consumption standpoint, to reduce manipulator drag.

As shown in Chapter 3, devices such as pivoting fairings and splitter plates can result in reduced drag and lift forces. However, in an operational situation, such devices may interfere with the task or be infeasible. Other methods, such as dimpling the arm surface to promote attachment, or altering the arm’s geometry to induce more flow along the span or around the tip, may well prove beneficial to reducing the magnitude of the drag forces acting.

**Computational Fluid Modeling of an Underwater Manipulator** In the present work, the theoretical model development was based on the case of two-dimensional inviscid fluid flow over a cylinder undergoing unsteady motion. This 2-D model was extended using strip theory to model accurately the forces generated from a flow with significant three-dimensional components. Although not yet feasible for use in real-time control, the development of a full 3-D computational model of the hydrodynamic forces and moments on a single-link arm would represent a challenging and informative contribution. Such a model would need to consider nonsteady, rotational motion and end effects. Computational approaches could range from a discrete-vortex method based on potential-flow theory all the way to the solving of the full Navier-Stokes equations.

**Optimal Control of an Underwater Manipulator** Once an accurate model of the hydrodynamics of a multi-link, multi-degree-of-freedom manipulator is available, optimal control approaches for the motion control of the manipulator become very attractive. For example, minimum-time slews for a limited amount of control effort, or minimum-energy slews for a fixed final time could be considered. Also of interest would be slews where the
minimum interaction impulse is imparted by the arm to the vehicle. Maximum-interaction-impulse slews may also be of interest for “swimming” or dynamic repositioning of the vehicle.

Effects of Vortex Shedding on the Control of a Manipulator DOF In the general case of an underwater manipulator, lift forces generated from vortex shedding will act as disturbances to actuated degrees of freedom. While the effects of vortex shedding on structures has been studied extensively, their effects on an actuator under closed-loop control have not been addressed. In this situation, the control could be altered automatically to alleviate lock-in. (For example, when lock-in occurs, the controller bandwidth could be changed to break the lock.) Another problem of interest would be the use of an actuator under closed-loop control to reduce the effects of lock-in on a link with joint flexibility. In this experimental work, the joint could be made deliberately more flexible to exaggerate the effects of lift forces generated under lock-in conditions.

Adaptive Control of an Underwater Manipulator with an Unknown Payload A substantial amount of work has been done in the Aerospace Robotics Laboratory on the adaptive control of manipulators with unknown payloads [58, 59]. This work, for industrial and space robots, has focussed on the rapid identification of mass and inertia properties of an unknown payload. For an underwater manipulator the problem is significantly more complicated, since drag, lift, and added-mass characteristics would have to be identified as well.

Strategies for Fully-Coordinated Arm/Vehicle Control For repositioning of the manipulator end-point, the approach taken in this research was to maintain the station of the vehicle and to move the manipulator appropriately. For many repositioning tasks, the vehicle may be free to move or required to move to complete the task. For such cases, an understanding of how best to take advantage of the redundancy and the dynamic capabilities of the total system would be very useful. This understanding could result in formalized strategies for the coordinated control of an arm/vehicle system. In the ARL, research on precisely this problem has been done for space robots [60]. For underwater systems, further research to develop strategies that account for the large hydrodynamic forces will be required.
In this work, information from the hydrodynamic model was used to augment the vehicle control in a beneficial way. Efforts to investigate the use of information from direct measurements of force in the control the arm/vehicle system may be profitable as well.

**Sampling Task in an Unstructured Environment**  In current operational systems, sampling of rocks or biological specimens is extremely difficult. The successful demonstration of a sampling task carried out automatically from a task-level command (rather than joystick or tele-operator commands) would represent a huge advance in capability. By extending some of the advanced vision techniques developed in the ARL [61], it is possible that an error vector between the arm end-effector and the desired object could be generated. By closing the appropriate control loops around this error vector, coupled with astute impedance control when contact is made, the object could be picked up with care. Successful completion of such a sampling task would require new research advances in vision and in automatic control.
Appendix A

Force Sensor Design Drawings

The following pages show mechanical design drawings for the in-line force sensor that was developed as part of this research. The conceptual design of the sensor was done by Tim McLain, while the detailed mechanical design was carried out by Gad Shelef. Drawings for the each of the three main sensor components (cantilevered sensing ring, inner plug, and outer plug) are shown in Figures A.1, A.2, and A.3. Figure A.4 shows the components in their assembled form.
Figure A.3: Outer Plug
Figure A.4: Force Sensor Assembly
Appendix B

Hydrodynamic Model Analysis

This appendix presents the full details of the two-dimensional hydrodynamic analysis performed for this research. This analysis considers the flow of an incompressible, inviscid fluid over a circular cylinder undergoing unsteady motions. The wake and feeding layers are modeled using discrete vortices with independent positions, velocities, and strengths. Figure B.1 shows a schematic representation of the 2-D cylinder and its wake. The 2-D portion of this analysis is similar in approach to that done by Sarpkaya [41, 19] for a stationary cylinder immersed in a moving fluid.

Figure B.1: Two-Dimensional Hydrodynamic Model Schematic
This diagram illustrates the flow conditions modeled in the 2-D hydrodynamic analysis.

From Blasius’ theorem (cf. equation 9.52.5 of Milne-Thomson [40]), the forces acting on a moving circular cylinder in a still fluid in the presence of a specified number of singularities
can be calculated by

\[ F_X - iF_Y = \frac{i\rho}{2} \int_C \left( \frac{dw}{dz} \right)^2 dz - \frac{i\rho}{\partial t} \int_C \bar{w} d\bar{z} + \rho A \frac{d\bar{W}}{dt} \]  

(B.1)

where \( F_X \) and \( F_Y \) represent drag and lift forces respectively with each integral being evaluated around the contour of the cylinder circumference \( C \).

With the aid of the circle theorem (cf. equations 6.21.1 and 13.50.1 of [40]), the complex velocity potential \( w \), which describes the flow situation pictured in Figure B.1, can be written as

\[ w = \frac{U(t)c^2}{z} + \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln(z - z_k) - \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln \left( z - \frac{c^2}{z_k} \right) + \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln z \]  

(B.2)

where \( z \) is a complex variable, \( U(t) \) represents the unsteady cylinder velocity, \( c \) represents the radius of the cylinder, and \( \Gamma_k \) and \( z_k \) represent the strength and position of the \( k \)th vortex. While the last term of Equation B.2 comes directly from the circle theorem, it should be excluded from the expression for \( w \), since the vortices are generated by the relative motion of the cylinder with respect to the water. Because of this, the presence of the vortices does not result in a change in the net circulation around the cylinder, and the image vortices at the center of the cylinder are not appropriate to include in the complex potential. Therefore, Equation B.2 can be re-written as

\[ w = \frac{U(t)c^2}{z} + \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln(z - z_k) - \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln \left( z - \frac{c^2}{z_k} \right) \]  

(B.3)

This expression is composed of a moving doublet term to model the cylinder, a term to model the \( m \) real vortices in the feeding layers and wake, and a term to model the \( m \) image vortices in the cylinder. The image vortices are included to ensure that the boundary conditions on the surface of the cylinder are satisfied. Motion of the reference frame fixed at the center of the cylinder is accounted for by

\[ \bar{W} = U(t) \]  

(B.4)

where \( \bar{W} \) represents the velocity of the reference frame.

Equation B.1 can be evaluated most easily by considering each term independently:

\[ F_{X1} - iF_{Y1} = \frac{i\rho}{2} \int_C \left( \frac{dw}{dz} \right)^2 dz \]  

(B.5)

\[ F_{X2} - iF_{Y2} = -\frac{i\rho}{\partial t} \int_C \bar{w} d\bar{z} \]  

(B.6)

\[ F_{X3} = \rho A \frac{d\bar{W}}{dt} = \pi \rho c^2 \frac{dU}{dt} \]  

(B.7)
Equation B.5 can be solved by Lagally’s theorem. The steady part of the drag and lift forces acting on a cylinder with \( m \) vortices alone with no circulation or stream present can be written as (cf. equation 8.63.9 of [40] and equation 5 of [41]):

\[
F_{X_1} - iF_{Y_1} = \rho \sum_{k=1}^{m} \Gamma_k (-v_k - iu_k) \tag{B.8}
\]

where \( u_k \) and \( v_k \) are real and imaginary components of the velocity of the center of the \( k \)th real vortex. Equation B.6 can be re-expressed by changing the sign on \( i \) throughout as

\[
F_{X_2} + iF_{Y_2} = i\rho \frac{\partial}{\partial t} \int_C \frac{w}{z} \, dz. \tag{B.9}
\]

The integral on the right hand side of this expression can be integrated by parts [62] to give

\[
F_{X_2} + iF_{Y_2} = i\rho \frac{\partial}{\partial t} \left[(wz)\big|_C - \int_C z \frac{dw}{dz} \, dz\right]. \tag{B.10}
\]

Since there are no singularities on the contour of the cylinder, \((wz)\big|_C = 0\), so that Equation B.10 can be re-written as

\[
F_{X_2} + iF_{Y_2} = -i\rho \frac{\partial}{\partial t} \int_C z \frac{dw}{dz} \, dz. \tag{B.11}
\]

Letting \( w = w_1 + w_2 + w_3 \), where

\[
w_1 = \frac{U(t)c^2}{z} \tag{B.12}
\]

\[
w_2 = \frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln(z - z_k) \tag{B.13}
\]

\[
w_3 = -\frac{i}{2\pi} \sum_{k=1}^{m} \Gamma_k \ln \left(z - \frac{c^2}{z_k}\right), \tag{B.14}
\]

we have by the residue theorem

\[
-i\rho \frac{\partial}{\partial t} \int_C z \frac{dw_1}{dz} \, dz = -2\pi \rho c^2 \frac{dU}{dt} \tag{B.15}
\]

\[
-i\rho \frac{\partial}{\partial t} \int_C z \frac{dw_2}{dz} \, dz = 0 \tag{B.16}
\]

\[
-i\rho \frac{\partial}{\partial t} \int_C z \frac{dw_3}{dz} \, dz = -i\rho \sum_{k=1}^{m} \frac{c^2}{z_k} \frac{\partial \Gamma_k}{\partial t} - i\rho \sum_{k=1}^{m} \Gamma_k \frac{\partial}{\partial t} \left(\frac{c^2}{z_k}\right). \tag{B.17}
\]

Applying the results of Equations B.15 through B.17 to Equation B.11 gives the following expression for \( F_{X_2} + iF_{Y_2} \):

\[
F_{X_2} + iF_{Y_2} = -2\pi \rho c^2 \frac{dU}{dt} - i\rho \sum_{k=1}^{m} \frac{c^2}{z_k} \frac{\partial \Gamma_k}{\partial t} - i\rho \sum_{k=1}^{m} \Gamma_k \frac{\partial}{\partial t} \left(\frac{c^2}{z_k}\right). \tag{B.18}
\]
In this expression, the position of the center of the \( k \)th image vortex, \( c^2/z_k \), can be written as

\[
\frac{c^2}{z_k} = p_{ki} + iq_{ki}
\]  

(B.19)

where \( p_{ki} \) and \( q_{ki} \) are the real and imaginary coordinates of the \( k \)th image vortex position. The coordinates of the image vortices are related to the coordinates of the real vortices, \( p_{ki} \) and \( q_{ki} \), by the following expressions:

\[
p_{ki} = \frac{p_k c^2}{p_k^2 + q_k^2}
\]

(B.20)

\[
q_{ki} = \frac{q_k c^2}{p_k^2 + q_k^2}.
\]

(B.21)

The velocity components of the image vortices, \( u_{ki} \) and \( v_{ki} \), are simply the partial derivatives with respect to time of each of the vortex coordinates:

\[
u_{ki} = \frac{\partial p_{ki}}{\partial t}
\]

(B.22)

\[
u_{ki} = \frac{\partial q_{ki}}{\partial t}.
\]

(B.23)

Finally, \( F_{X_2} + iF_{Y_2} \) can be formulated as

\[
F_{X_2} + iF_{Y_2} = -2\pi \rho c^2 \frac{dU}{dt} + \rho \sum_{k=1}^{m} \frac{\partial \Gamma_k}{\partial t} (q_k - ip_k) + \rho \sum_{k=1}^{m} \Gamma_k (v_k - iu_k)
\]

(B.24)

Combining Equations B.7, B.8, and B.24 and re-arranging into real and imaginary parts gives

\[
F_X = -\rho \sum_{k=1}^{m} \Gamma_k (v_k - u_k) + \rho \sum_{k=1}^{m} q_k \frac{\partial \Gamma_k}{\partial t} - \frac{\pi}{4} \rho D^2 \frac{dU}{dt}
\]

(B.25)

\[
F_Y = \rho \sum_{k=1}^{m} \Gamma_k (u_k - u_k) - \rho \sum_{k=1}^{m} p_k \frac{\partial \Gamma_k}{\partial t}
\]

(B.26)

The remainder of this analysis focuses on the in-line force term \( F_X \). For constant-acceleration motion, dimensional analysis shows that

\[
\frac{\Gamma_k}{UD}, \frac{u_k}{U}, \frac{v_k}{U}, \frac{u_{ki}}{UD}, \frac{v_{ki}}{U}, \frac{p_k}{D}, \frac{q_k}{D}, \frac{p_{ki}}{D}, \frac{q_{ki}}{D} \text{ are functions of } \frac{s}{D} \text{ and } Re
\]

where \( s \) is the displacement of the cylinder, \( D \) is the diameter of the cylinder, and \( Re \) is the Reynolds number. Sarpkaya and Garrison [19] showed empirically by experimental measurements over a broad range of Reynolds numbers (\( 100 < Re < 5 \times 10^5 \)) that

\[
\frac{\Gamma_k}{UD}, \frac{u_k}{U}, \frac{v_k}{U}, \frac{u_{ki}}{UD}, \frac{v_{ki}}{U}, \frac{p_k}{D}, \frac{q_k}{D}, \frac{p_{ki}}{D}, \frac{q_{ki}}{D} \text{ are functions of } \frac{s}{D} \text{ only.}
\]
Letting
\[ \frac{\Gamma_k}{UD} = f_k(s/D) \] (B.27)
so that
\[ \Gamma_k = UD f_k(s/D) \] (B.28)
and differentiating \( \Gamma_k \) with respect to time gives
\[ \frac{\partial \Gamma_k}{\partial t} = \frac{dU}{dt} D f_k(s/D) + UD \frac{\partial f_k(s/D)}{\partial t}. \] (B.29)
By the chain rule
\[ \frac{\partial f_k(s/D)}{\partial t} = \frac{\partial f_k(s/D)}{\partial (s/D)} \frac{\partial (s/D)}{\partial t} = \frac{\partial f_k(s/D)}{\partial (s/D)} \frac{U}{D} \] (B.30)
so that
\[ \frac{\partial \Gamma_k}{\partial t} = D f_k(s/D) \frac{dU}{dt} + \frac{\partial f_k(s/D)}{\partial (s/D)} \frac{U}{D} \left( \frac{\partial f_k(s/D)}{\partial (s/D)} \right) U^2 \] (B.31)
or
\[ \frac{\partial \Gamma_k}{\partial t} = D \frac{\Gamma_k}{UD} \frac{dU}{dt} + \frac{\partial (\Gamma_k/UD)}{\partial (s/D)} U^2. \] (B.32)
By substituting the results from Equation B.32 into Equation B.25, the following result is obtained:
\[
F_X = -\frac{\pi}{4} \rho D^2 \frac{dU}{dt} \left[ 1 - \sum_{k=1}^{m} \frac{q_{k_i} \Gamma_k}{\pi D UD} \right]
- \frac{1}{2} \rho D U^2 \left[ 2 \sum_{k=1}^{m} \frac{\Gamma_k}{UD} \left( \frac{v_k}{U} - \frac{v_{k_i}}{U} \right) - 2 \sum_{k=1}^{m} \frac{q_{k_i} \partial (\Gamma_k/UD)}{\partial (s/D)} \right]. \] (B.33)
For constant-acceleration motions, the bracketed expressions of Equation B.33 are functions of \( s/D \) only. Therefore, \( F_X \) can be expressed as
\[
F_X = -C_m(s/D) \cdot \frac{\pi}{4} \rho D^2 \frac{dU}{dt} - C_d(s/D) \cdot \frac{1}{2} \rho D U^2 \] (B.34)
where
\[
C_m(s/D) = 1 - \sum_{k=1}^{m} \frac{q_{k_i} \Gamma_k}{\pi D UD} \] (B.35)
and
\[
C_d(s/D) = 2 \sum_{k=1}^{m} \frac{\Gamma_k}{UD} \left( \frac{v_k}{U} - \frac{v_{k_i}}{U} \right) - 2 \sum_{k=1}^{m} \frac{q_{k_i} \partial (\Gamma_k/UD)}{\partial (s/D)}. \] (B.36)
Equation B.34 represents the in-line hydrodynamic force undergoing constant-acceleration motions. The key outcome of this analysis is that for a cylinder undergoing constant acceleration motions, the state-dependent hydrodynamic drag and added-mass coefficients, \( C_d \) and \( C_m \), are functions only of how far the cylinder has traveled.
Appendix C

OTTER Vehicle Dynamic Response

This appendix presents data showing the closed-loop response of the OTTER vehicle to a disturbance force impulse. The results shown here were obtained by disturbing the vehicle from its nominal position using a long, slender pole. The disturbance impulses were applied over a period of approximately 2 seconds. In Figures C.1 and C.2 plots of the open-loop and closed-loop response of OTTER in the roll and pitch degrees of freedom are shown. Figure C.3 shows the closed-loop response of OTTER’s yaw degree of freedom. In Figures C.4 through C.6, the closed-loop response of the vehicle in the $x$, $y$, and $z$ degrees of freedom is shown.

The data from these figures provide insight into the closed-loop performance of the OTTER vehicle. Since roll and pitch degrees of freedom are passively stabilized by moments due to buoyancy and gravity forces, open-loop response data in these degrees of freedom are provided for comparison.
Figure C.1: Vehicle Roll Attitude Regulation

This plot shows the open-loop and closed-loop roll response of the OTTER vehicle. With no control applied, the resonant frequency of the roll mode was 0.14 Hz. In comparison, the closed-loop resonant frequency of the roll mode was 0.23 Hz. The damping ratio of open-loop mode was 0.11, while the closed-loop mode damping ratio was much higher. Though the nonlinear closed-loop behavior prevented calculation of the damping ratio directly, the motion was fully damped within one cycle of motion. For a linear system, this would correspond to a damping ratio of 0.4–0.5.
Figure C.2: Vehicle Pitch Attitude Regulation

This plot shows the open-loop and closed-loop pitch response of the OTTER vehicle to a force disturbance. In the open-loop case the resonant frequency of the pitch mode was 0.053 Hz, while the damping ratio was 0.16. When closed-loop control was applied, the resonant frequency of the pitch mode was 0.084 Hz and the damping ratio was 0.19.
Figure C.3: Vehicle Yaw Attitude Regulation

In this plot, the yaw attitude disturbance rejection performance of the closed-loop controller is shown. From the data shown, the frequency of the closed-loop resonant mode was calculated to be 0.06 Hz, while the damping ratio was calculated to be 0.12.
Figure C.4: Vehicle x-Position Regulation

The disturbance rejection performance of the closed-loop vehicle controller for the x-position degree of freedom is shown in this plot. The frequency of the closed-loop x-position resonant mode was 0.050 Hz. The damping ratio for this mode was 0.17.
Figure C.5: Vehicle y-Position Regulation

This plot shows the closed-loop y-position response to an applied force disturbance. From this data, the resonant frequency of the y-position closed-loop mode was calculated to be 0.048 Hz, while the damping ratio was calculated to be 0.17.
Figure C.6: Vehicle z-Position Regulation

In this plot, the closed-loop z-position response to an external force disturbance is shown. The resonant frequency of the z-position closed-loop mode was calculated to be 0.038 Hz. The corresponding damping ratio was determined to be 0.16.
Appendix D

Arm End-Point Kinematics

As a single-number measure of the quality of the station-keeping control of the vehicle, the manipulator end-point position error was calculated. End-point position was calculated from measurements of the vehicle position \((x, y, z)\) and attitude \((\psi, \theta, \phi)\), and arm joint angle \((\alpha)\) according to Equations D.1 through D.3, which represent the position kinematics of the arm/vehicle system. Desired end-point position was calculated similarly using the desired position and attitude commands in place of the measured values. Measurements of \(x, y,\) and \(z\) represent the displacement of the vehicle from the center of the tank where the positive \(x\) direction is North, the positive \(y\) direction is East, and the positive \(z\) direction is down toward the Earth's center. For the reference frame fixed at the center of gravity of the vehicle, the positive \(x_{bf}\) direction is out the nose of the vehicle, the positive \(y_{bf}\) direction is toward the starboard side, and the positive \(z_{bf}\) direction is down through the belly of the vehicle.

To represent the attitude of the vehicle a yaw-pitch-roll \((\psi-\theta-\phi)\) Euler angle representation was used. Yaw (heading) represents the rotation of the vehicle about the earth-fixed \(z\) axis. Pitch represents the rotation of the vehicle about an intermediate body-fixed \(y\) axis, while roll represents the rotation about the body-fixed \(x_{bf}\) axis.

The arm hub location is located at the point \((L_x, L_y, L_z)\) in the body-fixed reference frame. The tilt angle formed between the plane of the arm's motion and the \(x_{bf}-y_{bf}\) plane of the vehicle is \(\gamma\), which for the experiments of this dissertation was fixed at 60 degrees. The arm joint angle, \(\alpha\), is the angle formed between the longitudinal axis of the arm and the body-fixed \(x_{bf}-z_{bf}\) plane. Figure D.1 shows schematically how the position and attitude of the vehicle and the position of the arm are represented.
Figure D.1: Arm/Vehicle Position and Attitude

This schematic illustrates how the various degrees of freedom of the arm and the vehicle were determined.

The end-point position of the manipulator is given by the following relations:

\[ x_{\text{end}} = x + L_x c_\alpha c_\phi - L_y (s_\psi c_\phi - s_\theta s_\phi c_\psi) + L_z (s_\psi s_\phi + s_\theta c_\psi c_\phi) + L_a [s_\psi (s_\alpha c_\phi + s_\gamma s_\phi c_\alpha) + c_\phi [c_\gamma c_\theta c_\alpha - s_\theta (s_\phi s_\alpha - s_\gamma c_\phi c_\alpha)]] \]  \hspace{1cm} (D.1)

\[ y_{\text{end}} = y + L_x s_\psi c_\theta + L_y (c_\psi c_\phi + s_\psi s_\theta s_\phi) - L_z (s_\phi c_\psi - s_\psi s_\theta c_\phi) - L_a [c_\phi (s_\alpha c_\phi + s_\gamma s_\phi c_\alpha) - s_\phi [c_\gamma c_\theta c_\alpha - s_\theta (s_\phi s_\alpha - s_\gamma c_\phi c_\alpha)]] \]  \hspace{1cm} (D.2)

\[ z_{\text{end}} = z + L_y s_\phi c_\theta + L_z c_\phi c_\theta - L_x s_\theta - L_a [c_\phi (s_\gamma s_\phi c_\alpha + c_\theta (s_\phi s_\alpha - s_\gamma c_\phi c_\alpha))] \]  \hspace{1cm} (D.3)

where \( s_\psi = \sin(\psi) \) and \( c_\psi = \cos(\psi) \), and so forth for the remaining angles \( \theta, \phi, \alpha, \) and \( \gamma \).
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