PORTABLE AUTOMATED DRIVER FOR UNIVERSAL ROAD VEHICLE DYNAMICS TESTING

DISSERTATION

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ABSTRACT

Automating road vehicle control can increase the range and reliability of dynamic testing, which is currently limited by the use of human test drivers. Many tests specify precise steering or brake inputs which human drivers are only able to approximate, adding uncertainty to the test results.

The goal of this project is to develop a portable control system, or “Automated Test Driver” (ATD), to automate many test driving functions. Though controlling steering, brake, and throttle functions, the ATD will still allow a human to sit in the driver’s seat if desired. The system is lightweight and compact enough to fit in the smallest passenger vehicle without substantially affecting its inertial properties, yet versatile enough to also control a large SUV. Test technicians can install the ATD in about an hour without permanent vehicle alteration, and later return the vehicle to stock condition for normal manual driving. Before testing begins, the ATD is paced through a brief tuning procedure to ensure stable and precise performance despite variation among test vehicles.

Though human test drivers are still the best choice for some types of tests, automation opens up a world of opportunity. As demonstrated by a variety of tests on three different vehicles, the ATD repeatably follows a predefined path with a given speed profile, or it can perform either path following or speed regulation independently. The ATD also precisely executes open-loop steering, throttle, or brake commands. It can automatically conduct tedious yet demanding tasks such as brake wear-in or understeer gradient tests, with greater accuracy than a human driver. Any event requiring precise timing or specific actuation displacements or velocities suits this system particularly well.
Key components for the Automated Test Driver have been designed by SEA Limited, Inc. This work explores the best means for integrating the SEA handwheel and brake and throttle actuators with a comprehensive control system.
To the glory of God.
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1.1 Introduction and Motivation

Each year millions of dollars and countless man-hours are spent around the world by automotive manufacturers, government agencies, consumer groups, and others doing road vehicle testing. Cars and trucks of all kinds are submitted to an endless battery of tests to ensure their safety, to gauge their performance, to assess their reliability, and to compare these metrics to other vehicles on the market. Yet, though the level of computer integration and automation built into the vehicles has dramatically increased over the past decade, the task of testing these vehicles is still largely performed by human drivers.

1.1.1 Human drivers limit vehicle testing

Unfortunately, the use of human test drivers often limits vehicle testing. For specific, precise, and repeatable inputs through the steering wheel, accelerator, or brake pedal, even the best human driver falls short of the capabilities of a well-designed machine. And as is the case when comparing output responses to test inputs in any experiment, the quality of the output is only as good as the quality of the input.

Every physical system has a characteristic input-output relationship. Given a series of identical inputs to similar systems, differences in the output correspond to differences between the physical systems. If robot-driven vehicles produce much more uniform test
inputs, differences in the test results will be more likely to reveal true differences between the vehicles.

As an example of the importance of uniform test input, consider the NHTSA rollover testing maneuver [1]. First the driver must give a 720°/sec steering input to a specified steering angle\(^1\). Then the driver waits for the vehicle roll rate to drop to less than ±1.5°/sec, and reverses the steering, at the same rate, to the same steering angle in the opposite direction. A professional driver can execute those commands fairly well, but even a slight variation in the timing or steering rate may be the difference between dangerously tipping up and safely plowing out. The resulting rollover rating can have a big impact on the sales success of the vehicle. With the stakes so high, the accuracy and repeatability of such a test is crucial.

Yet many standard vehicle tests are written with the limitations of human drivers in mind [2]. The SAE Service Brake Structural Integrity Test Procedure for passenger cars [3], for instance, stipulates that the brake pedal force should be rapidly ramped from zero to 200 lb at a rate of 2500 lb/s. To achieve this rate, however, “instantaneous rates can vary from 1000 to 4000 lb/s”, and only the ramp portion from 40 to 160 lb is required to fall even within this broad range. The requirements of the test are satisfied whether the maximum force is developed in 0.060 s or 0.104 s (73% longer).

Improving the uniformity of these tests – something possible with automated testing – would make their results more meaningful. Tightening the testing criteria would improve the quality of the results and allow more reliable differentiation between the performance of two different vehicles.

\(^1\) The angle which produced 0.3g lateral acceleration in a “Slowly Increasing Steer” maneuver.
1.1.2 Automation can reduce test driving job hazards

Automated testing could also eliminate some job hazards to human test drivers, and the current ATD incarnation is but a step away from enabling that. In rollover testing, drivers now depend on outriggers to protect them when the vehicle tips up. But outriggers, mounting hardware, or mounting brackets can fail, especially given that the vehicle frame was not designed for this type of device or loading. It would be safer simply to remove the driver from the vehicle. It would also be cheaper for the testing facility to eliminate both their inventory of expensive titanium outriggers and the need to retrofit vehicles with five-point safety harnesses.

This added safety equipment also has potential to affect the outcome of the tests. Automotive manufacturers have argued for years that the cumbersome outriggers used by some testing organizations adversely affect a vehicle’s roll moment of inertia, perhaps enough to cause the vehicle to tip up under circumstances where a vehicle without outriggers would not have. If the test is done with an unmodified vehicle driven by a robot, this argument disappears.

But not only could a robot driver improve the performance of standard, canned routines, it could also add significant flexibility to vehicle testing. The ability to follow a predetermined path, with or without a human driver, has great potential. SEA Ltd. (SEA), sponsor of this research, was recently asked to help test a type of security barrier. It was desired to accelerate a fully-loaded semi-trailer to highway speed and crash it into a barrier—obviously not a job for a human driver. The test facility did not have a straight track section long enough to accelerate the truck to the desired speed, so it would have to somehow guide itself around the oval track until it reached speed, then veer off into the barrier. Certainly a simple line-following controller could have been developed for the project, with some form of cruise control adapted to govern the speed; but if a proven, GPS-guided robot driver were on the shelf, this project could have been completed without all of the development and testing costs associated with the proposed control. Of
course the robot driver itself may be a casualty of such a test, but that would at least be a quantity known up front.

Human test drivers, to be sure, are not obsolete. They can still provide invaluable qualitative feedback concerning behaviors for which no electronic sensor exists. They are far superior to computers at synthesizing steering wheel feedback, noises, vibration, visual cues, and gut-level accelerometers to characterize the performance of a car. But for tests where repeated, consistent inputs are necessary in order to compare specific outputs, computers and machines can often do a better job.

### 1.1.3 Available devices for automating vehicle testing

Autonomous vehicle control is not a new area, but it has not often been applied to the field of vehicle testing. A handful of automotive manufacturers and at least one private company (ABD [4]) have developed test driving equipment with some degree of autonomy, but literature in the public domain remains scarce. Most testing today, even at world class test facilities such as the 4500-acre TRC (Transportation Research Center, Inc.) in East Liberty, Ohio, continues to be done by human drivers.

To this date, only one published work involves a robot driver integrating all three driver inputs into a device designed for dynamic testing of multiple vehicles. From 1998 to 2000, Volkswagen partnered with the German government to develop a system for automating durability testing on their punishing road course [5, 6]. This $10 million dollar system was effective, though at last report it still fell short of its goals. The authors state that the human test driver will not yet be replaced “due to some technical and cost-benefit limitations.” Specifically they cite the system’s dependence on an auxiliary battery-based power supply, which limits their test period below their 24-hour goal.

But with a look at the Volkswagen system, other limitations are readily apparent (Figure 1). Not only does the power supply add considerable weight and consume the trunk volume, but the massive electronics package fills the rear seats and roof rack. Clearly
this system lacks the flexibility to work with compact cars, and the sizable load on the roof would increase any vehicle’s rollover tendency. An affordable, adaptable robot driver has the potential to revolutionize vehicle testing.

![Figure 1: Volkswagen Autonomous Test Driver (after [5])](image)

Despite nearly two decades of serious research into the development of autonomous or semi-autonomous vehicles, most notably by the California PATH program (Partners for Advanced Transit and Highways), a computer-driven consumer automobile remains but a vision of the future. Though autonomous vehicles are not nearly ready for public use on open roads, they certainly can be made safe for limited operation in a controlled testing environment.

### 1.2 Purpose of the Study

It is the goal of this systems engineering study to combine steering, brake, and throttle actuators with global positioning feedback and a supervisory control system to create a portable, integrated Automated Test Driver (ATD) suitable for dynamically testing any road vehicle from the smallest subcompact to a Class 3 truck. A friendly interface will guide the user among various operating modes, including open-loop steering or brake/throttle maneuvers, path following with speed control, and recording a new path.
Further details on the system’s capabilities and limitations are found in Sections 1.3 and 1.4.

The actuators involved in the project have been developed by SEA Ltd. The handwheel actuator is part of their Automated Steering Controller (ASC), a system capable of reproducing any pre-programmed or manually-input steering sequence. This study will employ both the SEA handwheel as well as a servo-driven pedal actuator for both the brake and accelerator, also developed by SEA.

Many control input devices have been created and studied which are vehicle-specific. In such cases the vehicle’s handling response and powertrain can be accurately modeled for control purposes. The response of the brake system can also be more easily predicted when all of the plant parameters are known: pedal linkage arrangement, vacuum booster design, master cylinder volume, and rotor diameter, among others. Often the authors have even modified the vehicle to allow auxiliary throttle\(^2\) or brake actuation, or to allow direct measurement of key state variables, such as the master cylinder hydraulic pressure for a brake system.

Most devices in published literature also depend on knowledge of a variety of vehicle parameters, some of which may not be available without conducting additional tests. The best nonlinear vehicle handling models, for instance, account for weight shifting due to pitch and roll, steering deviations from compliance and roll, and other predictable phenomena stemming from vehicle-dependent parameters. The control task is made much easier when the designer can focus on the characteristics of a single vehicle.

The proposed robot driver for vehicle testing, on the other hand, has a unique and challenging set of design criteria. First and foremost, it must be able to accurately control

\(^2\) In this paper, “throttle” control shall be synonymous with accelerator control. Though it is understood that throttling is specific to gasoline engines, the robot driver is intended to perform equally well with diesel, and even hybrid, powertrains.
a vehicle with only limited knowledge of vehicle parameters and without a lengthy tuning procedure. The system must also be easily transportable between vehicles. Installation in a given vehicle should be quick and require no permanent or damaging vehicle modifications for equipment mounting. The equipment must nonetheless be mounted securely enough to prevent damage during high-g test maneuvers.

The ATD, for maximum utility, must be compatible with a wide range of motor vehicles. It must be adaptable for all passenger cars, light trucks, and sport utility vehicles. It would also be desirable to include military vehicles and commercial vans and trucks in its repertoire. A possible future extension would even include heavy trucks up to Class 8. But regardless the vehicle size, the robot driver system should have as small a size and weight as possible, in order to easily fit in the smallest of passenger cars and to minimize its impact on vehicle behavior.

Though the system will independently control the vehicle, it must be designed to allow a human driver to sit in the driver’s seat and override the controls when necessary. The entire system may be installed in a given vehicle, but part of a battery of tests may utilize only some of the robot’s actuators. For instance, it may be simpler for a human driver to steer and accelerate during a brake testing routine, with the robot controlling only the brake pedal. Should a deer jump onto the test track, the operator must still be able to reach and actuate the brake pedal.

Other safety features must be given careful consideration. The control computer should monitor feedback sensors to ensure the signal data is valid and the sensors are performing correctly. Newly programmed paths should also be verified with a driver in the vehicle before attempted without one. Though this list is far from complete, it gives an indication of the types of safety issues which arise.
1.3 Capabilities of the Automated Test Driver

This system may be programmed to carry out particular driving tests, such as the Consumers Union double lane change avoidance maneuver and the dynamic portion of the NHTSA rollover test. The constant radius test for determining steady-state directional control behavior [7] is also a great candidate for the robot driver. This demanding test requires the driver to gradually increase speed while repeatedly looping a circular path, recording the steering angle, speed, and other data. A robot driver has the potential to stay closer to the intended path using smaller corrective handwheel movements, all while recording more accurate steering measurements.

The ATD is able to follow an arbitrary user-given path, constrained only by the vehicle’s handling ability. Potential uses for the path-following function are many. For instance, tire wear could be compared for multiple tire types driving the same vehicle over the same aggressive road course. Or lateral cornering limits could be evaluated by gradually increasing speed on a circle track until the road grip is lost.

Any event with precise timing or requiring specific actuation displacements, velocities, or accelerations would also suit this system well. Such tests include brake fade testing, like the SAE Simulated Mountain Brake Performance Test [8], braking distance testing, or the standard burnishing procedures associated with these types of tests. The latter requires two hundred 40-0 mph stops at 12 ft/s², tedium that no human driver would enjoy. With some additional work to integrate the requisite temperature sensors, the entire SAE J1247 test could feasibly be automated.

The braking portion of the ATD alone has many applications. Modern braking systems have complex active safety devices which are not properly exercised or tested with simple straight-line brake actuation. Testing technologies such as Brake Assist [9], Antilock Braking Systems (ABS), brake force distribution control, vehicle stability control, and some forms of adaptive cruise control (ACC) are all potential applications for this robot driver.
Another design feature is separately operable lateral (steering) and longitudinal control modules. For example, it is possible to install only the throttle and brake actuators in a vehicle, along with the attendant longitudinal control elements (sensors and computers), and conduct braking tests with a human driver doing the steering.

1.4 Scope of the Study

Some of the robot driver applications have already been discussed, but let us specifically define what the robot will and will not do. First, this system is intended exclusively for a controlled testing environment, and will only employ the sensors required to calculate and / or update the vehicle’s position. No sensors (vision, radar, ultrasonic, optical, etc.) are planned to enable fully autonomous vehicle operation. The robot driver only follows predetermined paths or sequential timed or triggered operations; it has no mechanism for obstacle detection or avoidance. When operated without a driver, the vehicle will be able to be remotely stopped. But it will remain the test operator’s responsibility to maintain a safe environment.

Second, the path-following function of the robot driver is chiefly intended for quasiplanar, high friction road surfaces. Partial system testing (i.e., with a human controlling one or two of the driving inputs) on low friction surfaces could still be facilitated.

Third, the speed-control function is designed to maintain constant speeds and follow simple speed profiles based on time or distance. It is a low bandwidth controller, not meant for precise tracking of highly dynamic drive cycles such as the Federal Urban Driving Schedule (FUDS).

A final limitation is that longitudinal controls of the robot driver will function only in vehicles equipped with automatic transmission. Additional actuators for clutch and shift knob would add considerable complexity to this already challenging project, though they may be considered for future work.
1.5 Document Overview

The remaining sections of this work are organized as follows. Chapter 2 contains a review of relevant literature, surveying developments in dynamic vehicle control and related topics. Research methods are the focus of Chapter 3, detailing the design, implementation, and test procedure for the Automated Test Driver. Chapter 4 displays the results of the experimental tests as well as analytical comments on the ATD performance. Chapter 5 includes a brief summary as well as some suggestions for future work.

The Appendix contains graphical test results for every trial in the standard battery of validation tests, as well as results for a number of additional tests. The document concludes with the Bibliography.
CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

A sizable body of research exists in the field of automotive and, more generally, surface vehicle automated control. Some authors focus on a particular control subsystem, such as braking, whereas others investigate the bulk dynamics of an entire vehicle, or even group of vehicles. The last category (groups of vehicles), in particular, has drawn a great deal of attention over the past fifteen years.

2.1.1 Outline of Literature Review

The literature review will begin with a brief introduction to the study of Automated Highway Systems, a field to which much research in autonomous vehicles is indebted. Following this will be a detailed survey of existing automated vehicle testing equipment, complementing the highlights given in Section 1.1.3.

Because of the significant breadth of this topic, the literature review will be separated into two major areas. First, discussion of longitudinal dynamics will incorporate modeling and control issues in acceleration and braking. Second, a review of relevant work in lateral dynamics will include handling, steering response, and path-following topics. Though often treated separately, these two areas are not truly independent of one another; at the end are listed some studies which investigate the improvements possible by modeling the coupling between longitudinal and lateral effects.
After the discussion of vehicle dynamics and control come topics regarding various navigation sensor options as well as the system hardware architecture of similar studies.

2.1.2 Automated Highway Systems

Much of the published work in autonomous vehicle control has focused on Automated Highway Systems (AHS), or Intelligent Vehicle Highway Systems (IVHS) [10,11,12,13,14,15,16,17,18,19,20, and others], many of which were funded through the California PATH program. PATH, or Partners for Advanced Transit and Highways (though referred to by some authors as “Program on Advanced Technology for the Highway” [12,20]), was established in 1986 and is administered by the Institute of Transportation Studies (ITS) at the University of California, Berkeley. Motivated by the congested highways in that state and projections that the congestion would continue to worsen, researchers showed how highway throughput (vehicles per lane per hour) could be significantly increased by implementing autonomous vehicle control and coordination.

Instead of laying down more asphalt to accommodate the increased traffic, they propose to simply decrease the amount of pavement that each car requires. By reducing the spacing between vehicles and grouping these vehicles in “platoons” of, for example, eight vehicles, such an automated highway could support an effective throughput of about 4300 vehicles per lane per hour at 65 mph. By contrast, a normal highway with manually operated cars at the same speed has a throughput of approximately 2000 vehicles per lane per hour [21]. This reduced spacing (6.5 m) is safely possible only by virtue of the much faster reaction time of an autonomously-controlled vehicle compared to a human-controlled one.

Nearly all of these studies focus on the operation of a platoon of vehicles and offer algorithms tailored to maintain a specified intervehicle spacing. These studies are also generally more concerned with a smooth and comfortable ride than a robot driver would need to be in a testing environment. Some insights from the studies, however, can naturally be extracted for single vehicle modeling and control.
2.2 Existing Automated Vehicle Testing Equipment

In a discussion proposing a new tool for test driving automation, a survey of existing automated vehicle test driving equipment is essential. This Section draws from both academic and commercial literature.

A number of companies produce automated steering systems of some type. As mentioned in the introduction, SEA Ltd. markets an Automated Steering Controller (ASC, Figure 2, left) [22] which is capable of reproducing any pre-programmed or manually-input steering sequence. The ASC boasts 57 Nm torque at 720 deg/s, with a peak torque of 70 Nm and maximum speed of 1800 deg/s. Operational details of this unit are described in [23] and [24].

ATI/Heitz offers a similar steering unit called the Heitz Sprint 3 [25]. A peak torque of 60 Nm (optionally 80 Nm) is possible up to 1500 deg/s, and 25 Nm can be continuously sustained at 2000 deg/s. Unlike the other devices listed here, programs for the Sprint 3 are flashed to a plug-in EPROM, instead of direct download from a laptop or other user interface.

Stähle GmbH [26] also produces a variety of vehicle testing equipment. Like SEA and ATI/Heitz, they sell an open-loop type steering actuator. Stähle’s “Steeringpilot for cars” (SSP2000, Figure 2, right) is capable of 50 Nm maximum torque and 1800 deg/s steering velocity.
Figure 2: Steering Controllers from SEA Ltd (left [22]), ATI/Heitz (middle [25]), and Stähle (right [26])

Also in Stähle’s repertoire is a unit dubbed an “Autopilot for cars” (AP500S, Figure 3), is designed for vehicle speed control. This unit, which includes actuators for the accelerator, brake, clutch, and shift knob, is intended to be used only on a chassis dynamometer.

Figure 3: Stähle AP500S Autopilot for Cars [26]

Anthony Best Dynamics (or ABD, [4]) is a company concerned almost exclusively with automotive test equipment. Among their products are steering robots with three different nominal torque levels: 30 (Figure 4), 60, and 150 Nm. These clutchless direct drive units spin the steering wheel as well as the steering shaft, so an operator cannot keep his hands on the wheel during operation. They may, however, be mounted with a fixture enabling direct steering torque measurement.
Unlike the Stähle and ATI/Heitz units, ABD advertises a path-following capability for their steering controllers. When combined with a GPS-assisted motion pack, their steering controllers may be used to follow preprogrammed paths or retrace routes input by a user.

ABD also offers brake and throttle actuators with various force and speed ratings. The brake actuator shown at right in Figure 4 is capable of 1400 N pedal force; a high speed unit is also available. The ABD web site notes that some coordination of steering and brake maneuvers is possible.

As for fully autonomous vehicle testing, the lone example remains the system from Volkswagen discussed in the introduction [5, 6]. This cumbersome system is clearly not feasible for testing vehicles smaller than the vans displayed in Figure 1.

Naturally this has not been an exhaustive list of autonomous vehicle implementations; such a list would run many pages. Rather, this review was intended to capture the subset of devices designed to automate some facet of dynamic vehicle testing, the category into which the ATD fits. Certainly very sophisticated autonomous vehicles are plentiful in recent literature, but these systems are designed to fit and drive one vehicle, generally
with permanent modification required, with no thought to portability or adapting to various vehicles’ dynamics.

2.3 Longitudinal Dynamics

Beyond the survey of existing equipment for automated vehicle testing, a discussion of techniques and models in longitudinal and lateral control is worthwhile. We begin with the former.

Figure 5: General Longitudinal Dynamics Model

On the simplified model above, longitudinal dynamics are viewed as a two-input, one-output system [14]; other relevant factors are treated as disturbances. Longitudinal dynamics will be dealt with in the two broad subcategories of these inputs, acceleration and braking. Again the two subcategories are linked; consequently, some authors treat them together. One such study is especially relevant to this research and will be dealt with at the end of this Section.
2.3.1 Acceleration

Generally speaking, the more that is known about a physical system, the better the response of that system to a given input can be predicted. With this in mind, several authors turned to elaborate powertrain models to facilitate speed control [12,14,17,27]. While it is appealing to construct vehicle speed as an explicit function of throttle angle, this type of model is very difficult to derive and is highly dependent on specific powertrain parameters. Gathering this volume of detail for each different test vehicle and adding it to the controller is not only proscribed by the considerable effort involved, but all of the necessary parameters may simply not be available. Thus a more general model must suffice for the speed control of our robot driver, perhaps one that works with a limited set of discrete modes, or one that adapts or calibrates itself to a given vehicle.

2.3.1.1 Adaptive Cruise Control: Conventional

Recent studies on Adaptive Cruise Control (ACC) offer some suggestions. Adaptive cruise control actually has two meanings, both of which will be discussed here. One, such as described by Liubakka et. al., is the desire to enable one conventional speed control module to “provide acceptable performance over a wide range of vehicle lines and operating conditions” without recalibration for different vehicles [28]. This is similar to the goal of the proposed robot driver’s throttle control, although in our case some rudimentary calibration may be necessary.

Though conventional speed control is typically used at higher speeds simply to maintain a desired velocity by compensating for wind and grade disturbances, such a controller could be modified for use by the robot driver. Having surveyed the application of various control techniques to their problem, Liubakka et.al. found that a well tuned PI controller “is hard to beat” for speed control. “The problem is how to keep the PI controller well tuned, since both the system and operating conditions vary greatly” [28]. They settled on a slow adaptive algorithm which is initialized with speed-dependent gains.
2.3.1.2 Adaptive Cruise Control: Extended Functionality

The other meaning of adaptive cruise control is the one popularly associated with the ACC acronym, where a vehicle so equipped will automatically slow its speed to preserve a minimum safe distance, or headway, between it and a slower vehicle ahead of it. Some implementations also include emergency braking or collision warning / collision avoidance (CW / CA) schemes, and thus overlap the present accelerator control topic with the next one, braking. Versions of ACC, under various trade names, have been available on production vehicles since 1999, when Mercedes-Benz offered a Bosch product on their S class; in 2007 it remains a feature limited to premium-priced vehicles.

Hoess et. al. [29] studied three different variants of ACC: (1) using only the throttle, (2) using the throttle plus access to the transmission control unit (to provide additional deceleration by downshifting), and (3) using both the throttle and brake. Variant (2) would be difficult to implement for vehicles from a wide range of manufacturers. But this appears immaterial, since the authors reported that access to the transmission ECU yielded only marginal gains over use of the throttle alone.

Riley et. al. [30] extend the production cruise control feature with a brake algorithm. They describe a learning mode where the braking performance of the vehicle is gauged by a series of smooth stops on a surface with high coefficient of friction. This type of system identification could be also applied to calibrate the robot driver to a test vehicle.

Riley also examined the decision between using throttle control or brakes for deceleration. Throttle control, or slowing the vehicle by a combination of normal road loads (aerodynamic drag, rolling resistance) and powertrain friction, is typically used for deceleration requirements of less than 0.1 g. Also, the effectiveness of the engine braking is heavily dependent upon vehicle characteristics such as aerodynamic drag coefficient and transmission logic. Thus they decided to focus on brake-related deceleration and limit reliance on throttle control.
It may also be prudent to mimic these authors’ development strategy. Implementation was separated into two phases: Phase I, where basic ACC functionality was realized, and Phase II, which extended the utility to stop and go traffic conditions. Similarly for the robot driver, throttle based speed control and brake actuator development should be independently developed before any integration is attempted.

### 2.3.2 Braking

The other half of vehicle longitudinal control is braking. It may seem as though braking control should be easily transportable from one vehicle type to another, because automotive braking systems remained quite uniform for over half a century. The 1920 Model A Duesenberg was the first production car to be equipped with four wheel hydraulic brakes, and the 1928 Pierce-Arrow followed as the first with a vacuum booster [31, 32]. Even though it took Ford until 1939 to adopt hydraulic brakes – the last major manufacturer to do so – the basic system remained the same (see Figure 6) until the advent of practical ABS in the early 1990’s.

![Figure 6: Conventional Brake System (after [45])](image)

Figure 6: Conventional Brake System (after [45])
But despite the relative homogeneity of brake systems, neither modeling nor controlling one is a trivial task. Any force transmission system formed from a chain of mechanical, hydraulic, and pneumatic elements is naturally difficult to characterize. The vacuum booster alone, source of the “power assist” on most systems, is responsible for much of the system’s highly nonlinear behavior. The booster not only contains deadzones, spring preloads, and nonlinear fluid flow dynamics, but the booster dynamics are also dependent on the engine manifold dynamics (in the case of gasoline engines), which are also nonlinear [45]. And though many automobiles today retain the classic design, variants are rapidly gaining popularity as more and more vehicles include ABS or even brake-based electronic stability control.

2.3.2.1 Brake Actuator Design

Of primary concern when developing a mechanical braking apparatus is ensuring it will have the speed and force capability to match the criteria for any standard (or other reasonable) test. The SAE J229 test procedure [3], as mentioned before, requires an average application rate of 2500 lb/s up to 200 lb pedal force. Though the test permits an 100 lb overshoot (i.e., 300 lb maximum instantaneous force), it is not required that the actuator develop more than 200 lb. Both SAE J1247 [8] and the now-cancelled SAE J937 [33] braking tests stipulate a similar 200 lb maximum force.

Good design procedure, however, would certainly prescribe a healthy margin above the 200 lb requirement to compensate for actuator friction, wear, and misalignment between the actuator and pedal. Due to the pivoting (and thus nonlinear) travel of the typical brake pedal, some degree of misalignment is unavoidable when using any linear style of actuator.

Human factors studies are also of interest to quantify the brake force capability of human drivers. Eaton et. al. [34] found the 5th percentile pedal force capability for women was 100 lb, where brake effort was defined as the “highest force applied at the brake pedal for a 0.5 sec interval in a dynamic task.” Among all female test subjects driving a full size
sedan, the mean maximum pedal force was 159 lb. In another study, Segel et. al. [35] confirmed roughly the same result as Eaton for the 5th percentile female, and reported 200 lb and 250 lb force for the 50th and 80th percentile females respectively.

2.3.2.2 Brake System Modeling

Regardless of the final form of the ATD braking controller, it will be instructional to have a good mathematical model of the vehicle brake system. D.K. Fisher performed a classic mechanical and hydraulic system analysis of the standard, vacuum booster assisted brake system [36]. He proceeded through each component in the mechanical linkage, booster, master cylinder, brake lines, and drums, creating an 18-state dynamic model. Khan et. al. updated and streamlined this model in 1994 [37], but still required 10 states. Such detail may be required for braking system design tasks but is too complex a model for brake control. It would be too difficult to gather the necessary parameters, and the calculations would be too cumbersome for real time control.

In 1995, Gerdes et. al. [10] developed a model of brake system dynamics “tailored to the demands of control applications”. According to the authors, this 3-state model, “despite its simplicity, meets or exceeds the accuracy of previous models.” Such a model may be applicable for the robot driver, though it still requires considerable data (such as component masses) which will be difficult to determine without either access to manufacturer files or disassembling the system. The use of average values for these parameters, however, may still facilitate acceptable system performance. Alternately, a brief system identification routine may provide the missing parameters for a particular vehicle.

Looking at more current literature, brake systems have changed markedly over the past decade. Two authors [38,39] detailed modeling and simulation tools to support the design of modern brake systems. In many systems the conventional pedal-to-booster-to-master cylinder brake system is no longer routed directly to the brake calipers/shoes, but rather feeds a computer-controlled brake actuator (Figure 7). The brake actuator consists
of a hydraulic pump, reservoir, and valves which are directed by the ECU (Electronic Control Unit) to individually control the brake fluid pressure at each wheel.

![Diagram of Advanced Braking System]

**Figure 7: Advanced Braking System**

It is possible that, despite the nonlinear nature of the braking function and the variety of brake types, a generic model could be used for the robot driver. For the tasks it has been given, such a model may guide the brake controller to acceptable performance.

### 2.3.2.3 Brake System Control

Lennon and Passino [40] studied what they call the “base braking” problem, seeking to develop a controller which can ensure that the braking torque commanded by the driver will be achieved. They were not interested in the panic stop situation, but rather in the factors causing the normal brake pedal force (required to achieve a given deceleration on dry high-friction pavement) to vary. Though a variety of factors contribute to this problem, the primary factor studied here was pad/rotor temperature. The authors modeled brake action with an experimentally-determined linear relationship between brake fluid pressure and the stopping force on the car, multiplying the resultant force by a nondimensional scaling factor termed “specific torque”. Specific torque was defined to
reflect the variations in the brake torque as the brake pads increase in temperature. Several control methods were tested to reduce braking variations due to temperature effects: a fuzzy model reference learning controller, a genetic model reference adaptive controller, and a general genetic adaptive controller. Conclusions were limited, however, due to the lack of experimental verification of these algorithms.

Lennon and Passino were working to extend the results of a previous study [41] which evaluated Proportional-Integral-Derivative control (PID), lead-lag, autotuning, and model reference adaptive control (MRAC) techniques. While several of these techniques were highly successful, they included no compensation for environmental or other factors, such as the effect of brake temperature considered in [40].

Many studies implement brake control via alternate actuators which bypass the mechanical pedal linkage. Some such controllers still function through the vacuum booster, though many are connected directly to the brake fluid line. Yi and Chung [42], an example of the latter design, propose a brake control law for vehicle collision warning / collision avoidance (CW/CA) systems. Though this study is not of direct interest for the ATD, it does carry the braking model all the way to stopping distance estimation. Brake hydraulic dynamics are approximated as a linear second-order system, where brake torque is considered proportional to the brake fluid pressure at the wheel. But stopping distances are estimated by a function which includes an estimate of tire/road friction, which can be calculated either using wheel slip [43] or using the engine and wheel speeds [44].

More applicable to this work are studies such as those done by Maciuca, Gerdes, and Hedrick [45,46], where the actuator was mechanically linked to the brake pedal. These authors explored sliding mode control for this application, chosen due to its robustness in the presence of modeling errors and disturbance inputs.
2.3.3 Integrated Longitudinal Control

Combining both elements of longitudinal control, Ioannou and Xu [14] present several throttle and brake control systems, again with the focus toward platoon operation and automatic vehicle following. Their development is thorough and easily generalized to the problem of single vehicle control. They also extensively discuss the problem of integrating both throttle and brakes to maintain desired speed.

These authors tested and compared three throttle controllers, both through simulation and on vehicles: PID, PID with gain scheduling, and adaptive control. Their adaptive controller is essentially a PID controller where a parameter estimator or an adaptive law is used to update the gains based on measured performance data. All three controllers performed acceptably:

The fixed gain PID controller is the simplest one, but it causes some oscillation in acceleration and deteriorates the riding comfort at low speed. The adaptive controller has the fastest response and is somehow vehicle independent, but it requires more calculations. The performance of the PID controller with gain scheduling is between that of the other two controllers.

Ioannou and Xu separate the two input, one output system into two completely separate subsystems: (1) throttle angle to speed, and (2) braking command to speed (Figure 8). This was possible because the two controllers were never allowed to act simultaneously; a logic switch in the supervisory control activated one or the other.
The brake controller design is “based on the way that a good driver uses the brake and accelerator pedals.” For instance, both pedals are never used simultaneously. The brake is used only for rapid and large decelerations, and frequent switching between the two is avoided. A hysteresis zone was used to prevent frequent switching. Though their hysteresis was designed for vehicle following, a simplified version (such as shown in the figure below) would suffice for the robot driver.

![Figure 8: Longitudinal Vehicle Model (after [14])](image)

Ioannou and Xu report good results from treating the brake subsystem as a first order low pass filter with some time delay. In their design, however, “both the time delay and the dynamics of the brake subsystem are ignored” because the time delay is noticeable only
when the brakes are first applied and because the dynamics of the system are “much faster than those of the drivetrain.”

2.4 Lateral Dynamics

The lateral dynamics of an automobile are also difficult to model and precisely control. Guldner et al. [70] summarize the difficulties. “Due to the wide range of possible operating conditions (vehicle mass, road adhesion, etc.), automatic steering is a task for robust control. Steering actuator saturation and limited state information represent further challenges to control design. In addition, passenger comfort and safety considerations have to be taken into account.” It is a topic, however, which has drawn a good deal of attention over the past half century, so many tools are available for its study.

2.4.1 Vehicle Dynamics Overview

Numerous models are also available to relate vehicle lateral response to the steering input. Point mass models are the simplest example but are probably not sufficiently accurate for a vehicle control application such as this. The “bicycle model”, on the other hand, is still quite simple yet has proven accurate enough to be the basis for much research in vehicle dynamics. Though many variants of this model exist (e.g., [47, 93]), the common element is the reduction of four wheels to two, typically with the steering wheel in front and a fixed wheel in back (as on a bicycle).

Beyond the bicycle model, four wheel models are necessary for increased accuracy (e.g., [48, 49]). These incorporate, at a minimum, load transfer effects from vehicle acceleration, roll, and pitch. Some include many more subtle effects such as roll steer, bounce steer, camber thrust, and tire relaxation length.

The chief uncertainty in all these models lies in the tire-road interface, a phenomenon which is yet to be fully understood. Numerous models exist to approximate this facet of a vehicle’s behavior (e.g., [50, 51, 52, 53]), but most are rather complex.
As complexity is added to these models, two things occur which affect their usefulness for control applications. First, the number of vehicle parameters required to use the model increases, decreasing the ease with which the model is applied to a range of vehicles. Second, computation time increases, if the model is used in the control loop (in an observer, for instance), and this invariably decreases the effectiveness of the controller. Thus a suitable tradeoff between accuracy and complexity must be determined for the robot driver application.

2.4.2 Path-Following Algorithms

A number of path-following algorithms are discussed in the literature. Follow-the-carrot [54] and pure pursuit [55, 56] are geometrical algorithms using a desired position to calculate the intended vehicle trajectory. The second of these, pure pursuit, is a simple and efficient algorithm which is well understood, having been the basis for many different studies in robotics and autonomous vehicle control (e.g., [57, 58, 59]). In brief, the procedure is as follows: First the controller finds the point on the planned path (“goal point”) which is a given distance from the vehicle (“lookahead distance”). Then the steering angle is calculated to drive the vehicle along the arc of constant curvature connecting the current position and the goal point (see Figure 10). This steering command is recalculated every cycle of the control loop.

Unfortunately there is an inherent tradeoff in selecting the lookahead distance ($L$). Short lookahead is desirable for closely following the finer features of the path, but it also tends to destabilize the system. One can intuitively see that with a short lookahead, any deviation from the path will bring a sharp corrective steer, causing the vehicle to overshoot the path (possibly starting unstable oscillations). This principle is illustrated in Figure 11. Thus most applications settle for a larger lookahead to assure stability, though its damping effect may smooth over some sharp path features; the magnitude of $L$ determines the maximum path curvature that the algorithm can successfully track.
Any implementation of pure pursuit must take into consideration two of the method’s limitations. First, the pure pursuit method does not incorporate a model of the dynamic capability of a vehicle or its actuators\(^3\), with potentially dangerous results. For instance,

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\(^3\) A number of authors have simply stated that the pure pursuit method does not account for vehicle dynamics. This is not strictly true, however; calculation of the steering angle to match the desired arc trajectory includes not only the Ackerman Angle but also a term dependent on the vehicle’s understeer coefficient.
the algorithm could prescribe a sharp change in curvature at high vehicle speed. To mitigate this danger the programmer can impose speed-dependent steering angle or steering rate limits. In addition, the path can be analyzed beforehand to assess the odds of successful tracking at a given speed.

Hogg et. al. [57] and Kelly [60] investigate the improvements possible with various modifications to the pure pursuit algorithm. Hogg notes that pure pursuit, having only one parameter ($L$), is easy to tune, and it performs well when the vehicle is started on or near the path. But if the lateral path error becomes large, the method can be unstable. The stability improves if the method is adaptive, where the lookahead is made a function of lateral path error.

Looking closely, however, this is just another way of viewing the same problem previously discussed. Proper choice of lookahead distance depends not only on lateral path error but also on vehicle speed, vehicle handling characteristics, and controller delay. (The latter factors will be further explored in a subsequent Section on control stability.) For implementation in vehicle testing, with all sorts of possible paths at the full range of automotive speeds, the controller must naturally be designed to robustly accommodate this variety.

Wit et. al. [61] propose “vector pursuit”, a technique based on the theory of screws. This method takes into account not only the desired future position of the vehicle but also its desired orientation at that goal point. Though more computationally intensive, the method is demonstrated to be less dependent on the lookahead distance selected. Its chief advantage, however, is good performance with very short lookahead distances. Using reasonable lookahead, pure pursuit performs just as well\(^4\), and sometimes better; differences are minor at all larger values of lookahead.

\(^4\) As measured by standard deviation of position error and heading error.
One metric where vector pursuit consistently outperforms its competitors is overshoot of vehicle position after a jog in the path. In this case, increasing lookahead does not seem to improve the position-only based algorithms, where the vector pursuit response remains robust throughout the range of lookahead values tested.

PID control was used in a fourth path-following algorithm, developed independently by at least two authors [57, 62]. As with pure pursuit, a path goal point is selected based on a given lookahead distance. In this case, however, an error is computed on which the PID control acts. That error is the angle between the current vehicle heading and the vector pointing toward the goal point (Figure 12). Sidhu [62] points out, however, that this method is only a trigonometric variation of pure pursuit; in PID control, the steering command is a proportional gain applied to the angle $\theta$, whereas pure pursuit ultimately applies a gain to $\sin(\theta)$. If the vehicle is reasonably close to the path, $\theta$ will be small and $\sin(\theta) \approx \theta$.

![Figure 12: PID Path Following](image)

Though the methods are quite similar, the “gain” in the pure pursuit method has geometric significance derived from the vehicle steering geometry, whereas the gain in
the PID method is arbitrary. But the PID controller has the ability to act on the rate of change of error, something that basic pure pursuit lacks. Hogg et. al. report that a combination of these two methods, a simple average of the steering commands calculated separately by each, yielded consistently better results than either method alone.

Tseng, et. al. [63] developed another preview-type algorithm tied to vehicle-dependent parameters. This algorithm, in which the steer angle is calculated using both kinematic relationships and vehicle dynamics relationships, is explored in depth in Section 3.3.2.2 (page 80).

Figure 13: Sharp et. al. Driver Model Scheme (after [64])

R.S. Sharp et. al. extended the preview concept from a single goal point to a vector of such points, suggesting that “one cannot imagine a real driver using single-point preview” [64]. Their steering command is generated from a weighted average of instructions from a path point directly adjacent the vehicle to a path point far ahead.
(Figure 13). Though implemented only in simulation, this algorithm demonstrated excellent winding-track path following on a nonlinear model of a Formula One racer at speeds up to 90 m/s.

### 2.4.3 Path Planning

Path planning is a term typically used in connection with vehicles or robots which must decide their paths for themselves, proceeding efficiently to a destination but avoiding obstacles encountered on the way. Though such logic is beyond the scope of our proposed robot driver, some of the principles from this area may be applied.

For this research it is assumed that the path is entirely defined by a human operator before the vehicle even starts. Nonetheless, path-following algorithms often employ various methods to interpolate between given waypoints and thus smooth both control output and vehicle behavior. Some authors use simple linear interpolation [62], others use polynomial fits [56] or Bezier splines [65]. Munoz et. al. [66] argue for \( \beta \)-splines based on computational efficiency.

Beyond path smoothing there are other significant benefits to preprocessing a path [66]. Major considerations include keeping the maximum path curvature \( \gamma \) (or minimum path radius \( r \)) within the limits of a vehicle, and planning speeds along a path subject to a vehicle’s kinematic and dynamic constraints. The first could be achieved simply by restricting the minimum path radius to values greater than the vehicle minimum turn radius. The second, speed planning, could be implemented by defining an upper bound for lateral acceleration and restricting the maximum velocity accordingly (Eqn. (1)). This obviously applies only in scenarios where a vehicle’s limit handling performance is not being evaluated.
\[ v_{\text{max}} = \sqrt{a_{L,\text{max}}} \gamma \]  
\[ \gamma = \frac{1}{r} \]

2.4.4 Control Stability

Murphy [58] and Ollero and Heredia [59] published studies analyzing the inherent stability of the pure pursuit method as well as the impact of delay and other parameters on stability. Murphy devised a closed form expression representing the condition for stability:

\[ \sin^{-1}\left(\frac{L - v\tau}{L + v\tau}\right) > T \sqrt{\frac{v}{d\tau}} \]  

\( L = \) lookahead  
\( v = \) velocity  
\( \tau = \) time constant of position detection system (approx. as 1st order system)  
\( T = \) computational and other system delay

Ollero and Heredia linearized the vehicle kinematic equations for constant velocity, enabling the path tracking problem to be formulated for every value of vehicle speed as a linear control problem. This step allowed the dynamic of the steering actuation system to be appended to the model as additional linear equations. The resulting model enables linear control methods to be applied for the automated vehicle steering.

Their linearized stability analysis is easily justified: “Local stability of an equilibrium state of a nonlinear system may be examined by stability of the linearized system around the equilibrium state. This analysis applies only in the neighborhood of the equilibrium state. In the path tracking problem, the vehicle is trying to follow the path, so the vehicle’s state will be in the vicinity of the equilibrium state, and therefore the above assumption is acceptable” [59].

Figure 14 summarizes the findings of this study. Lookahead has been nondimensionalized by dividing by vehicle velocity (V) and steering actuator time.
constant (T), and the delay term (Tau) is likewise normalized by T. Gamma ($\gamma_p$) represents path curvature. Taking $\gamma_p = 0$, or a straight line, as the worst case, these results can be used directly in a steering controller to ensure the pure pursuit lookahead stays well within the stable region.

![Figure 14](image.png)

Figure 14: Pure Pursuit Stability with Delay (after [59])

2.4.5 Other Control Options and Related Issues

A number of other specific steering control options and issues exist which do not fit into the categories already presented. Four such options are discussed below.

2.4.5.1 Robust Steering by Yaw Rate Control

Jürgen Ackermann has published a number of papers (incl. [67, 68, 69]) detailing the advantages of controlling yaw rate rather than steering angle for robust vehicle control. Employing principles from linear system theory, Ackermann finds that “integrating unit feedback of the yaw rate error makes the yaw mode unobservable from the front axle lateral acceleration and thereby takes uncertainty out of the steering transfer function” [67].
With the path-following methods previously described, the steering angle is treated as having a direct geometrical relationship with the path of the vehicle. In reality, of course, there is a series of highly nonlinear links in the control chain between the steering wheel and the vehicle motion which make this geometrical approximation valid only at slower speeds and mild actuator inputs. Ackermann’s method recognizes this limitation and treats steering angle instead as a yaw rate reference input.

2.4.5.2 Sliding Mode Path Tracking

Dr. Ackermann teamed with Prof. V. Utkin [68, 70, 71] and others to explore application of sliding mode control to vehicle control. The properties of disturbance rejection and order reduction are appealing for this application, as well as its relative robustness to modeling imperfections. The authors report good results from application of this method, such as smaller deviations from path than a baseline linear controller. Yet the sliding mode controller also had its disadvantages; a more oscillatory behavior was evident in steering angle rate and lateral acceleration [68].

Guldner et.al. [70] develop lateral control without need for path preview. To make it insensitive to speed variation, motion equations and sampling rate is based on distance derivatives instead of time derivatives. “Obviously, fast motion requires a higher sampling rate than slow motion. Sampling data is collected with regard to covered distance rather than with regard to elapsed time, i.e. \( \Delta d = v \Delta T \).”

2.4.5.3 Sideslip Angle Estimation

Estimation of slip angle\(^5\), or vehicle sideslip, is critical to problems in vehicle dynamics. While measurement of the forward speed (in the direction the body is pointed) can

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\(^5\) Body slip (vehicle sideslip) is the angular difference between vehicle heading (the direction the vehicle body is pointing) and the direction the vehicle is actually moving. Tire slip is the angular difference between vehicle trajectory and the direction of the tire.
generally be related to wheel rotational speed, it is more difficult to precisely determine the direction of the actual vehicle velocity.

Nishio et. al. [72], in developing a vehicle stability control system, propose a method to estimate vehicle sideslip angle using only inputs from standard automotive grade sensors. They note that a vehicle model estimator (observer) gives typically accurate results except that it is not robust against changes in road conditions, exactly the circumstance for which stability control systems are needed. So from the sensed quantities of yaw rate, lateral acceleration, steering angle, and vehicle speed, their estimator combines a vehicle model observer with a pseudo-integral method to add this necessary robustness.

Other interesting quantities are available from their estimator, including road-tire friction and road bank angle. A flag for vehicle spinout is also available, and must be available since the vehicle model observer cannot estimate the slip angle accurately when the vehicle spins out. Spinout is indicated (after a given steering command) when the lateral acceleration plateaus at a value lower than the expected threshold. The judgment threshold is adjusted based on the aforementioned estimate of road-tire friction.

2.4.5.4 Electronic Copilot for Safety and Vehicle Diagnosis

As discussed in the introduction, practical implementations of autonomous drivers must include some supervisory system to ensure safe operation. At least one study took this concept far beyond a basic safety monitor; also incorporating comprehensive real-time vehicle diagnostics, they described their system as an “electronic copilot” [6]. In an attempt to replace the surveillance possible by a human test driver, these authors devised a system to optically analyze the instrument cluster, compare acoustical signals against an expected norm, and to monitor electrical signals such as those available through the CAN bus.
2.5 Integrated Lateral / Longitudinal Control

Though most autonomous vehicle controllers separately treat the lateral and longitudinal control aspects, some authors specifically addressed the coupling between them [13, 15]. Lim and Hedrick [15] identified and characterized various coupling effects and designed a combined controller which compensated for these. Because such coupling effects become increasingly significant as maneuvers involve higher accelerations, larger tire traction forces, or reduced road friction, designers of a controller for dynamic vehicle testing must keep these effects in mind.

Lateral / longitudinal coupling effects fall into three classifications: kinetic coupling, tire force coupling, and weight shift coupling. One example of kinetic coupling is the steer angle effect, where the lateral cornering force of the front wheels actually has a component in the longitudinal direction. Another example is the observation that the (lateral) centripetal force is a function of longitudinal velocity (and curvature). Tire force coupling, the second effect, is illustrated by noting that given a certain coefficient of tire-road friction, the resultant of the lateral and longitudinal forces on each tire is limited by a function of the direction of the resultant.

Weight shift in the vehicle can also have a significant effect. When the weight shift is induced by longitudinal acceleration, it can have a considerable effect on lateral dynamics. Jack rabbit starts, for example, shift a significant fraction of the vehicle weight off the front wheels, thereby reducing the lateral force available for steering correction. Lim and Hedrick note that weight shift induced by lateral acceleration, on the other hand, has a negligible effect due to two mechanisms which naturally adjust the slip ratio to evenly distribute longitudinal force: the drivetrain differential, for the case of acceleration, and the independence of brake torque from the angular wheel speed.

Implementing a sliding mode lateral controller incorporating compensation for lateral / longitudinal coupling, these authors report 20% less deviation from path than with a gain-
scheduled PID controller. Because a longitudinal slip ratio estimate is currently unavailable, their longitudinal controller modeling these effects could not be used.

2.6 Navigation Sensor Options and Sensor Fusion

The capabilities of navigation sensors, especially those in the affordable realm, have grown dramatically over the past decade. GPS has gone from the exclusive tool of the military to sub-$100 toys, now embedded in cell phones and PDAs. As receivers have become more commonplace, they have also become available in much faster and more accurate forms. In the early 1990’s, researchers at Carnegie Mellon concluded that GPS feedback, available at only ~1 Hz, was alone insufficient to control a vehicle even at low speeds [83]. At this writing, however, RTK-GPS (Real Time Kinematic GPS) is commonly available with 1cm (1-σ) horizontal accuracy and position output up to 10 Hz.

Inertial measuring devices have also matured, now available in integrated six-axis packages with built in fusion and filtration\(^6\). Though MEMS\(^7\)-based IMUs are very affordable compared to their ancestors, they are prone to irregular drift. GPS, on the other hand, is relatively slow but has bounded error. The fusion of an IMU with accurate GPS to correct drift provides a powerful tool for vehicle control feedback. Section 3.5 (pg. 88) explores both of these sensor types in more depth.

Studies have investigated different combinations of sensors for different applications. Lenain et. al. [73] employ a single GPS antenna to direct the automatic steering of a farm vehicle. Tractors operate almost constantly in high slip conditions (much more so than automobiles), making the steering control problem more difficult. Trying to keep costs

\(^6\) These packages are often referred to as Inertial Measuring Units (IMU).

\(^7\) Micro-Electromechanical Systems (MEMS) are the “integration of mechanical elements, sensors, actuators, and electronics on a common silicon substrate through microfabrication technology” (http://www.memsnet.org/mems/what-is.html).
down with only a single sensor, these authors compensate for the sliding conditions by incorporating them in their model. Instead of using a conventional car-like dynamic model, the tractor is treated as a vehicle with two steering axles.

Other authors attack the sliding problem with a plethora of sensors. O’Connor et. al. [74] used an array of four GPS antennas to determine vehicle attitude. Hailing the advantages of GPS over optical forms of guidance, they note that GPS requires neither good weather, daylight, nor pavement markings or other visual cues for operation. Two suggestions from this study should be remembered. First is the simple observation that their system performed better when an optimally estimated state was used in the controller rather than the unfiltered measured states direct from the sensors. Second, they neglected to compensate for the lateral displacement of the GPS antennas due to roll motion of the body, which could easily account for 2.6cm of apparent lateral movement per degree of roll. Without such compensation, especially on uneven ground, it is difficult to determine how far the vehicle reference point truly deviates from its path.

Another sensor gaining ground in navigational applications is the GDS, or Geomagnetic Direction Sensor (also known as a magnetometer). This device uses the Earth’s magnetic field to give an output corresponding to the direction it is pointing. Though this type of sensor gives drift-free output in world coordinates, it is affected by disruptions that a ferrous vehicle will cause in the magnetic field; to compensate this effect, the sensor must be calibrated in each installation.

Noguchi et. al. [75] compared automated farm tractor steering by GDS alone and a combination of GPS and GDS. Control by GDS alone is really a type of dead reckoning, since it outputs only heading. These authors concluded, however, that GDS alone would be suitable for use as a short-term controller when the GPS signal is lost. Applying Kalman filtration (discussed below) to estimate the states from each sensor and using a two-dimensional Probability Density Function to fuse GPS and GDS sensor results, they achieved a lateral error less than half the stated accuracy of the GPS sensor.
Hogg et al. [57] combined inclinometers, MEMS rate gyros, accelerometers, and GPS to create their own navigation sensor package for use on a 20 kg tracked mobile robot. Ultimately they recommend employing one of the commercially available, fully integrated MEMS Inertial Measurement Units with digital output for increased noise immunity. When drift in angular rates is checked with magnetometers, a complete navigation solution is possible, typically referred to as an Attitude Heading Reference System (AHRS).

In his 2004 PhD dissertation, Jihan Ryu [76] demonstrated the potential of such a full sensor suite. Employing only automotive grade (i.e., inexpensive) inertial sensors and two laterally spaced GPS antennas, he accurately estimated important vehicle parameters such as tire cornering stiffness, understeer gradient, and roll stiffness from data collected in test maneuvers.

The primary arithmetic tool for combining the output of multiple sensors is the Kalman Filter, dubbed by one author “Navigation’s Integration Workhorse” [77]. In a modern paper on navigational sensing, the Kalman Filter really needs no introduction. The recursive method outlined by R. E. Kalman in 1960 [78] is almost universally present in the solution of discrete data linear filtering problems found in autonomous navigation. In essence, the Kalman Filter is “a multiple-input, multiple-output digital filter that can optimally estimate, in real time, the states of a system based on its noisy outputs” [77] by minimizing the mean of the squared error. “The filter is very powerful in several aspects: it supports estimations of past, present, and even future states, and it can do so even when the precise nature of the modeled system is unknown” [79].

Practical applications of the algorithm, however, “require careful attention to adequate statistical modeling and numerical precision” [77]. Though the exact pattern of sensor noise is never available, one must be able to characterize it with some accuracy for the Kalman Filter to work properly.
2.7 System Hardware Architecture

A number of authors reported on practical issues encountered when constructing or testing their autonomous vehicles. Hogg et al. [57] note that “careful consideration has to be given to the entire signal path from the sensor to the A/D converter.” They found that excessive noise in the signal caused a dramatic increase in the perceived sensor drift rate. To reduce the effects of the noise, they recommend separating and isolating power supplies solely for the gyros and accelerometers, isolating power and signal traces on the sensor and control board, and ensuring that signal traces are properly shielded and routed to avoid cross-talk between axes.
CHAPTER 3

RESEARCH METHODS

3.1 Overview

The ATD design is the culmination of development efforts in hardware, control, safety, and evaluation. These four areas provide the topical outline for this chapter.

3.2 Hardware

The first topic, hardware, itself breaks down into subtopics. The actuators will be discussed first, followed by processor, sensor, and electronics choices. Finally, the size, mass, and packaging of the system will be assessed in order to evaluate how close it comes to its goals of portability, impact on test vehicle mass distribution, and ease of installation. A schematic overview of the hardware is shown in Figure 15.

3.2.1 Steering Actuator

The Automated Test Driver (ATD) includes two actuators: a steering handwheel unit, and a brake and throttle actuator. The handwheel unit, shown mounted on a display stand in Figure 16, was developed as part of SEA Ltd.’s Automated Steering Controller (ASC). This actuator consists of a DC servomotor with gear reduction, clutch, and housing. Actuator torque is reacted through two suction cups affixed to the windshield.
The handwheel housing includes a normal steering wheel with two thumb buttons. The right button disengages a clutch to allow the servomotor to take steering control, and the left button enables the servoamplifier and cues the controller to begin. Both buttons must remain depressed else control reverts to the driver.

The handwheel’s LED display (top center) gives the operator a digital speed output, replacing the dashboard speedometer which is mostly obscured by the housing. Flanking the LED panel are amber and green lights which communicate program status to the operator. For instance, a flashing amber light indicates that the operator should straighten the front wheels and press the left button to zero the steering encoder. Once completed, the amber light ceases flashing and remains on. When the program is ready to run, the green light will flash to indicate that the programmed sequence will start as soon as both buttons are depressed; the operator need not look back to the laptop display for this information.
Installation of the handwheel, while not difficult, is the most time-consuming part of ATD setup. More precisely, initial installation is time-consuming, because it requires the removal of the driver’s airbag, for vehicles so equipped. Once the standard handwheel is removed, an adaptor (universal or vehicle-specific) is fixed to the steering column spline, and the handwheel bolts easily to the adaptor. (The option does exist to attach the actuator directly to the stock handwheel, but this option is seldom used because the extra height brings the handwheel rather close to the operator.)

3.2.2 Brake and Throttle Actuator

SEA Ltd. has also built a brake and throttle actuator (BTA, Figure 17) for testing purposes. One servomotor drives a ballscrew which, acting through a mechanical linkage, advances the accelerator or brake pedal, depending on direction of rotation. On each pedal is clamped a small plate; the actuator rods bolt to these plates through a spherical bearing. The current design is capable of applying 270 N (60 lbf) to either pedal.
The linkage is adjustable to fit a wide range of pedal positions and spacing, and yet allows a human foot comfortably between the actuator rods. Whenever desired, the driver can release the footswitch; this cuts power to the motor and allows him to instantly regain pedal control. Because neither pedal linkage is rigidly connected to the ballscrew, the driver may also override the actuator and apply either brake or throttle at any time without having to fight motor torque or inertia.

Several features combine to hold the BTA in place. First, the substantial mass of the base plate (16 kg) is sufficient to prevent the reaction torque from lifting the actuator off the floor. Second, the rear edge of the plate is positioned against some feature at the rear of the footwell, either a ridge in the floorboard or the rails to which the driver’s seat is mounted. This edge counteracts the rearward pedal reaction force. Third, a ratchet strap is threaded from posts on the plate’s rear corners to the driver seat base, further securing the actuator. Finally, Velcro pads on the plate bottom adhere to the carpet and resist shifting of the plate. Figure 18 shows the BTA installed in two of the test vehicles.
Pedal action will naturally vary from one vehicle to another. In some vehicles, the brake or throttle pedal travels a significant distance before effecting any braking torque or acceleration, respectively. This deadband, left uncorrected, would complicate the control action. Thus the deadband is, for the most part, eliminated during installation by adjustment of the threaded actuator rods. A small deadband is allowed in the brake pedal to prevent excessive heat buildup in the brakes during testing.

For simplicity, pedal displacement feedback is accomplished through the motor’s optical encoder. Though direct measurement would be more accurate, no tests are yet envisioned which would require high accuracy for this metric. The kinematic relationship of the actuator linkage was examined for a wide range of pedal positions. From plots of actuator versus pedal displacement it was determined that, unless the pedals were significantly offset (either horizontally or vertically) from the connection point on the actuator, the relationship was very nearly linear with unity slope.

3.2.3 Electronics Enclosure and Battery Box

The electronics enclosure houses the ATD’s main computer, servo amplifiers, Ethernet switch, and a battery charger (Figure 19). National Instruments’ CompactRIO (cRIO-9012), powered by a 400-MHz microprocessor running a real-time operating system, is
the heart of the electronics box. This real-time controller is mated to an 8-slot backplane for modular I/O, itself driven by a 3-million gate Field Programmable Gate Array (FPGA).

![CompactRIO with 4-slot backplane and Electronics Enclosure](image)

**Figure 19:** CompactRIO with 4-slot backplane (left) and Electronics Enclosure (right)

Various real-time platforms were considered for use, including Simulink-coded Motorola and DSpace boards. CompactRIO was selected for its combination of affordability and ease of implementing an attractive GUI (graphical user interface) through LabVIEW.

Also residing in the electronics enclosure are two analog AMC servo amplifiers: a brushed-DC version for the handwheel, and a brushless-DC version for the BTA. An Ethernet switch interconnects the GPS/INS unit, CompactRIO, and a laptop computer used for control interface.

Finally, a dual-voltage battery charger is included to trickle-charge the battery box used to power the amplifiers. The trickle charger is sufficient to keep the system operating indefinitely for most testing scenarios. It is possible that for continuous path-following operation for very long periods with only brief breaks between, the operator may need to alternate between two battery boxes. This was never experienced by the author, however; indeed, one battery box could operate the system for sixty minutes of actual driving time without the charger even on.
3.2.4 Vehicle State Sensor

For following predefined paths, the feedback mechanism of choice for modern autonomous vehicles is a combination of two sensor types: GPS (Global Positioning System) and INS (Inertial Navigation System). GPS yields accurate position, heading, and velocity information with bounded error but is limited in bandwidth. Inertial sensors, by contrast, can have high bandwidth but their integrated outputs are susceptible to drift. The nature and capabilities of GPS and INS are expanded in Section 3.5.

By combining (or “fusing”) these two sensors with an appropriate mechanism, vehicle state output could be both accurate and of high frequency. At present, the best mathematical mechanism for sensor fusion is the Kalman Filter, described in Section 2.6.

3.2.4.1 Development

Initial work on the path-following system was done with the combination of a two-axis speed/distance sensor and the yaw axis of an inertial sensor (Figure 20). The Corrsys / Datron Correvit S-400 is a non-contact optical sensor “for slip-free measurement of longitudinal and transversal dynamics”. In digital mode it outputs distance and body slip angle; in analog mode, it outputs longitudinal and lateral speeds.
The inertial sensor was the Crossbow IMU400CA-100. This inertial measurement unit (IMU) combines three rotational gyros and three linear accelerometers in one housing, all of MEMS construction. As with the optical sensor, both analog and digital outputs are available at 100 Hz.

![Sensor](image1.png)

**Figure 20: Corrsys/Datron S-400 Optical Speed Sensor (left) and Crossbow IMU400 (right)**

The combination of these two sensors proved reasonably effective for path following over short distances. The weakness lies mainly in the S-400, which must be mounted on the perimeter of the vehicle (offset from the tracking reference point). This sensor is difficult to properly align to the vehicle body and loses accuracy at low speeds. Given these challenges and the inability to completely remove the sensor drift, a lateral variation of 1-2 meters was typical at the end of a 150m test path. To prevent the accumulation of error, a high-precision global position feedback is necessary, such as is available with GPS.

A RTK-GPS (Real-Time Kinematic GPS) system was then purchased to complement the other sensors. Two NavCom NCT-2030M GPS receivers are used (Figure 21), one as a base station and one as a rover unit. The base station is positioned at a fixed location and broadcasts the real-time GPS error correction to the rover via an RF link. In this mode, the NCT-2030M boasts a horizontal positioning accuracy of 1 cm (RMS) and broadcasts output at up to 10 Hz.
Though this three-sensor system provided suitable vehicle state feedback, it was bulky, somewhat redundant, and time-consuming to set up. Thus the optical sensor was omitted, and work commenced to integrate the GPS and inertial data with a Kalman filter. The first step was to convert the output from the three accelerometers and three rate gyros into position, velocity, and acceleration data in a local coordinate system. Then the Kalman filter was applied to integrate the two sensors’ outputs.

3.2.4.2 Commercial GPS/INS Unit

During this development, a commercial navigation unit was made available for the project. The RT3002 (Figure 22) combines a GPS receiver and INS system in one compact package (234 x 120 x 80 mm). When receiving correction signals from the NavCom base station, this unit broadcasts 100Hz vehicle states at advertised accuracies of 2cm (1σ) in position, 0.05 km/h (RMS) in velocity, and 0.1° (1σ) in heading. Full specifications are shown in Table 1.

Figure 22: RT3002 by Oxford Technical Solutions
The RT3002 took the place of the previous inertial sensor and GPS receiver, and relieved the central processor of the sensor fusion and filtration duties. This unit provides state output in three ways: a 72-byte packet is available via either serial or UDP, or a slightly more complete set of states is available via CAN.

GPS/INS solutions are also available commercially from other companies. The Applanix POS LV210, for instance, incorporates ring laser gyros instead of MEMS gyros, advertising much lower noise and drift. GeneSys and Honeywell manufacture similar units, employing either MEMS or fiber-optic gyros (FOG). RaceLogic produces a similar series of “VBOX” systems which will measure and store vehicle dynamic data but not broadcast it real-time.

<table>
<thead>
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<th>Positioning</th>
<th>L1/L2 Kinematic</th>
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<td><strong>Velocity Accuracy</strong></td>
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<td>- Linearity</td>
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</tr>
<tr>
<td><strong>Lateral Velocity</strong></td>
<td>0.2%</td>
</tr>
<tr>
<td><strong>Update Rate</strong></td>
<td>100 Hz</td>
</tr>
<tr>
<td><strong>Calculation Latency</strong></td>
<td>3.9 ms</td>
</tr>
<tr>
<td><strong>Power</strong></td>
<td>9-18 V d.c. 15W</td>
</tr>
<tr>
<td><strong>Dimensions (mm)</strong></td>
<td>234 x 120 x 80</td>
</tr>
<tr>
<td><strong>Weight</strong></td>
<td>2.2 kg</td>
</tr>
<tr>
<td><strong>Operating Temperature</strong></td>
<td>-10 to 50°C</td>
</tr>
<tr>
<td><strong>Vibration</strong></td>
<td>0.1 g²/Hz 5-500 Hz</td>
</tr>
<tr>
<td><strong>Shock Survival</strong></td>
<td>100G, 11 ms</td>
</tr>
<tr>
<td><strong>Internal Storage</strong></td>
<td>500 MB</td>
</tr>
<tr>
<td><strong>Twin Antenna</strong></td>
<td>No</td>
</tr>
<tr>
<td><strong>Upgradeable GPS</strong></td>
<td>Yes (RT3003)</td>
</tr>
</tbody>
</table>

Table 1: RT3002 Specifications [80]
3.2.5 Size, Mass, and Installation

Compactness, lightweight, and ease of installation were three of the goals for the Autonomous Test Driver. Table 2 gives one measure of the degree to which those goals were attained. Photos in Sections 3.2.1 and 3.2.2 demonstrate that any vehicle which has room for a human driver could easily accommodate the actuators as well. The RT3002 may be mounted in any orientation, anywhere in the vehicle, as long as an Ethernet cable can connect it to the electronics box. The laptop computer would typically remain on the passenger seat or floor beside the driver, though devices are available to mount this more securely.

The only remaining component is the bulkiest one, the electronics box. A spacious box was selected for developing this prototype; its contents could easily fit into a smaller enclosure. The chosen box, however, does not even fill a normal-sized footwell.

In a 2-seat vehicle such as the Mazda Miata, the electronics box would reside on the passenger floorboard with the battery box beside it. The RT3002 would be mounted between the driver’s and passenger’s seats, or perhaps the deck behind the seats, leaving the passenger’s seat free for the laptop. In larger vehicles, placement of components is naturally more flexible.

In its current form, the system mass is 57.7 kg (127 lb), two-thirds of the weight of the average American male. The actuators represent the heaviest components. The massive base plate of the brake and throttle actuator was intended to help stabilize it against pedal forces. Securing the actuator by other means could allow significant weight reduction, if that were desired. Total system volume is 0.077 m³, or 2.7 ft³.
<table>
<thead>
<tr>
<th>Component</th>
<th>Mass (kg)</th>
<th>Dimensions (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Handwheel unit</td>
<td>14.8</td>
<td>Ø380 x 173</td>
</tr>
<tr>
<td>Brake &amp; throttle actuator</td>
<td>17.6</td>
<td>380 x 375 x 120 max</td>
</tr>
<tr>
<td>Electronics box</td>
<td>7.5</td>
<td>356 x 221 x 340</td>
</tr>
<tr>
<td>Battery box</td>
<td>11.4</td>
<td>356 x 132 x 162</td>
</tr>
<tr>
<td>GPS/IMU</td>
<td>2.2</td>
<td>234 x 120 x 80</td>
</tr>
<tr>
<td>Laptop computer</td>
<td>2.2</td>
<td>325 x 260 x 40</td>
</tr>
<tr>
<td>Cables &amp; antennae</td>
<td>2.0</td>
<td>–</td>
</tr>
<tr>
<td><strong>System total:</strong></td>
<td><strong>57.7</strong></td>
<td><strong>0.077 m³</strong></td>
</tr>
</tbody>
</table>

Table 2: Size and Mass of ATD Components

3.3 Control

As vital as the hardware just discussed is the control system to govern it. Control exists in this system at several different levels and in as many different structures. The organization of the control system will be discussed first, followed by details of the development and implementation at each level.

3.3.1 Organizational Structure

The ATD control hardware is a system of four computers, arranged as shown in Figure 23. This figure is patterned after one created for Ohio State University’s ION (Intelligent Off-road Navigator), a 2005 entry in the DARPA Grand Challenge [81].

The sensing computer filters and combines the inertial and GPS data into a set of simple position coordinates for use in the central computer. This will take some processing
burden from the central computer and speed up the main control loops. In this case, the RT3002 serves as the sensing computer.

A “Windows” laptop computer will serve as the user interface and handle high-level system control. It accepts the mode of operation (e.g., autonomous path following, brake test, etc.) from the user and initiates the appropriate programs to configure or execute that operation. These programs may be run on the laptop or on CompactRIO, as appropriate.
A third computer, CompactRIO, is responsible for low-level control actions. Based on instructions received from the high-level controller and on feedback from navigation sensors and the local controller, the central computer will update reference values for steering, throttle, and brake.

The local controller, here the FPGA embedded in the CompactRIO backplane, drives the servo actuators to the reference values passed from the low level controller. It also handles analog and digital signal I/O as well as relay actions.

3.3.1.1 Hybrid System Description

The ATD controller can be described as a “hybrid system”, combining continuous dynamical systems with a discrete system. The system operates in a variety of discrete modes with distinct goals, but the execution of each mode takes place in the continuous time domain.
\[ \Psi = \text{Translates state information to switching signals.} \]

\[ \Phi = \text{Generates events for Finite State Machine based on system state or sensor data thresholds.} \]

\[ \Gamma = \text{External Control Events (e.g., user mode selection, test parameter input, control overrides)} \]

Figure 24: Hybrid System Diagram
The hybrid system diagram in Figure 24 is one method of organizing the structure of a complex control problem (modeled after [82]). The problem breaks into three components: the finite state machine, the continuous system, and the interface between these two.

3.3.1.1 Finite State Machine

The finite state machine comprises a system of discrete states or modes in which the system is expected to operate. Such state diagrams are supported in the State Diagram Toolkit of LabVIEW and the Stateflow toolbox of Matlab/Simulink.

To illustrate the concept, a simple state machine example is shown in Figure 25 for the logic in a standard cruise control. The system has three states: Cruise, Speed up, and Manual. A cruise control system is always in one of these three states, but in only one state in a given instant. Certain cues (or “events”) from outside the state machine cause the system to change from one state to another. For instance, when the system is in “Cruise” and the driver taps the brake pedal, the system reverts to “Manual” control.

![Figure 25: Standard Cruise Control Logic (State Machine Model) (after [82])](image)

Figure 26 below shows the state machine employed in the ATD user interface. Descriptions for the various states and selected interface events follow in Table 4 and 4.
Figure 26: Finite State Machine for User Interface

<table>
<thead>
<tr>
<th>STATE</th>
<th>STATE DEFINITION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Init</td>
<td>Initialize user interface.</td>
</tr>
<tr>
<td>Select Mode</td>
<td>Display the appropriate mode choices available to the user and wait for input.</td>
</tr>
<tr>
<td>Configure PF</td>
<td>Configure the path follower: solicit user choice of path, speed profile, path</td>
</tr>
<tr>
<td></td>
<td>reference position, and other instructions to the controller.</td>
</tr>
<tr>
<td>Path Follow</td>
<td>Execute current path following configuration.</td>
</tr>
<tr>
<td>Configure OL</td>
<td>Configure the desired open-loop steering, brake, or throttle exercise.</td>
</tr>
<tr>
<td>OL</td>
<td>Execute currently-configured open-loop exercise.</td>
</tr>
<tr>
<td>Record Path</td>
<td>Record path and speed driven by operator, process and display results, and save</td>
</tr>
<tr>
<td></td>
<td>path and speed profile files for future use.</td>
</tr>
<tr>
<td>Exit</td>
<td>Stop all running programs except RT3002.</td>
</tr>
</tbody>
</table>

Table 3: State Descriptions for Figure 26
### Table 4: Selected Event Descriptions for Figure 26

<table>
<thead>
<tr>
<th>EVENT</th>
<th>INTERFACE CONDITIONS TO GENERATE THE EVENT ($\phi$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>newPF</td>
<td>Button depressed: user desires to configure new path follower.</td>
</tr>
<tr>
<td>not config’d</td>
<td>Any number of conditions caused configuration failure (i.e., RT3002 not initialized).</td>
</tr>
<tr>
<td>configured</td>
<td>Path follower configuration program successfully completed.</td>
</tr>
<tr>
<td>done</td>
<td>Path-following routine complete or aborted by user via handwheel buttons or footswitch.</td>
</tr>
<tr>
<td>repeatPF</td>
<td>Button depressed: user desires to repeat the last path-following routine.</td>
</tr>
</tbody>
</table>

#### 3.3.1.1.2 Continuous System

The continuous, or dynamic, system is the real-time portion of the hybrid system. This portion is common to every control system. In the cruise control example above, the continuous system is the longitudinal dynamics of the vehicle and the inputs which affect it. It could be represented by the illustration in Figure 27 below.

![Continuous Time System (Cruise Control)](image)

**Figure 27:** Continuous Time System (Cruise Control)
The ATD consists of two distinct axes of control, lateral and longitudinal. Linkage between the low-level and local controllers is shown in Figure 28. The left column is the steering control and the right column is the brake and throttle control.

Note the nested control architecture employed [83]. For example, the steering planner (path-following algorithm) sends a desired steering angle ($\delta_{des}$) to the ASC servo controller, which then drives the handwheel to the desired angle and leaves the steering planner free to update $\delta_{des}$.

**Figure 28: Control System Overview**

### 3.3.1.1.3 Interface

The final block of the hybrid system is the interface, which links the finite state machine, the continuous system, and data from external sensors. No control actions are generated.
here; it is simply a communication bridge between these elements. The interface consists of two functions, $\Psi$ and $\Phi$. $\Psi$ translates state information to switching signals used in the continuous system, and $\Phi$ monitors vehicle states and sensor data to generate event triggers for the FSM.

An example of $\Psi$, going back to the cruise control illustration, would occur when the vehicle is in “Manual” state and the driver pushes the “set” button. The state switches from “Manual” to “Cruise”, so the interface tells the continuous system to switch from manual to automatic throttle control and begin to perform as a regulator, maintaining speed at the given set point. An example of $\Phi$ would occur when the vehicle is in “Speed up” state and reaches the preset desired speed. The interface determines this signal threshold is reached and passes this event to the state machine, which then switches from “Speed up” to “Cruise”.

### 3.3.1.2 User Interface

The operator’s link to the hybrid system just described is through the graphical user interface (not to be confused with the “interface” of Section 3.3.1.1.3, which is within the hybrid system). This ATD user interface is resident on the laptop. To illustrate its function, the sequence of events will be shown for an operator who desires to record a certain manually-driven path then have the ATD retrace the path automatically.

The first panel seen by the operator is shown in Figure 29. Pressing the lower left button will bring up Figure 30, the program for recording path and speed. The operator first gives the new path a filename, specifies the separation between path points, and indicates whether the RT3002 should be expecting correction signals from a base station. Also selected is whether the reference position, or (0,0) coordinate on the new path, will be manually-entered or if it will be the point where the vehicle starts. The latitude and longitude of this reference position are stored in the path file.
Figure 29: Main ATD Operation Panel

Figure 30: "Record Path and Speed" Front Panel
After recording the new path, the program will return to the main operation panel (Figure 29). Since this path has never been run by the ATD, a new configuration file must be written. “Configure New Path Follower” is selected on the main panel, calling the program shown in Figure 31.

![Figure 31: Configure Path Follower](image)

Here the operator browses for and selects the new path and speed files by name, and enters a name for this new configuration file (or accepts the default). The operator also picks other options, such as whether to run the path as a continuous loop, automatically transitioning from the last path point back to the first.

The choice is made whether to use the reference position (lat / long) stored in the path file, to use the vehicle’s current position, or to manually enter a value. This allows the
flexibility to run a given path anywhere the user desires, regardless of where it was collected.

Once successfully configured, the ATD automatically proceeds to the program which executes the maneuver. On the right in Figure 32 is the program running on CompactRIO, where the vehicle parameters have already been entered. The steering algorithm requires an estimate of vehicle wheelbase, CG-to-front axle distance, steering ratio, and understeer gradient. These values are entered once, when the program is first run, and remain in place until ATD is shut down.

The left side of Figure 32 shows the simple operation panel for the path-follower. Here the configured run may be started and stopped, and any status or error messages from the low-level controller may be viewed. Though the stop button is functional, typically the program is stopped by the operator releasing a handwheel button or the footswitch. When the run is complete, the host program on the left closes, and the operator is again returned to the main operation panel. To repeat the previous run, “Repeat Last Path Follower” opens the host program directly, and the system awaits the “Start ATD” button. With a driver in the vehicle, however, no action will start unless the handwheel buttons and footswitch are depressed.
3.3.2 Lateral Control Implementation

Among the various options available for lateral path-following control, two methods were experimentally tested. The first is Sidhu’s proportional-derivative type controller introduced in Section 2.4.2 [62]. The second is a kinematic algorithm analogous to, but less simplistic than, Pure Pursuit. Ultimately a combination of these two methods was used.

3.3.2.1 PD Heading Error Control

Initial ATD development, along with extensive simulations, was conducted with the PD Heading Error Control method. This path-follower consists of two functions, the path planner and the control law.
3.3.2.1.1 Path Planner

The path planner smoothes the control action, smoothes the desired trajectory into one the vehicle can actually follow, and makes the tracking success robust to variation in the path point spacing. This subprogram emulates the forward-looking function of a human driver, comparing the current vehicle orientation with the intended path to calculate an achievable “goal point” ahead (see Figure 33).

Figure 33: Goal Point Calculation

As with all preview-based path-following algorithms, Sidhu’s method employs a goal point which is some “lookahead distance” down the path from the vehicle. Typically no discrete path point lies exactly one lookahead distance from the vehicle, thus the path planner finds the two bounding path points and interpolates between them. One could avoid interpolation and simply find the path point nearest “L” meters from the vehicle; the resulting steering command, however, becomes jerky unless the path points are quite close together. Interpolation smoothes the actuator motion and makes the system tolerant of widely-spaced path points. This effect is illustrated by simulation in Figure 34.
The interpolation mechanism works as follows: The planner finds the nearest path point more than one lookahead distance from the vehicle (point A in Figure 33). Then the planner looks at the path point just previous to A, which will be called point B. If point B is in the vehicle’s forward field of view, then the interpolated goal point will lie between A and B. If, on the other hand, point B lies somewhere behind the vehicle, the goal point will be interpolated between A and the vehicle’s current position, P. Though this method means that the goal point can lie off the intended path, it makes for a much more robust interpolation scheme. When interpolation is performed with one point ahead and one point behind the vehicle, it is easy for the algorithm to confuse the forward and reverse path directions.

The PD Heading Error algorithm can easily handle path vectors both widely- and closely-spaced. Yet when interpolating as described, close spacing between points in the path definition array yields the best adherence to desired path (see Figure 35). But though increasing path point density improves following ability, it comes at the price of computation time, which is critical for a real-time process. And higher density is only beneficial to a point; in practice, any path point spacing less than the order of the lookahead distance does not improve path-following accuracy. For the battery of tests
published in this work, path point spacing was 0.2 m for paths with fairly tight corners and 1.0 m for higher speed paths with more gradual turns.

![Effect of path point density, Speed = 10 mph, Lookahead = 2.6 m](image)

**Figure 35: Effect of Path Point Spacing**

After computing the goal point, the path planner verifies that this point is in the vehicle’s forward field to avoid tracing a reverse path. It then stores the coordinates of the nearest path point ahead of the vehicle for reference in the next iteration. This saves computation time by allowing the path planner to only consider the relevant portion of the path (the limited segment just ahead of the vehicle), savings which can be considerable if the path definition array is large.

There is also a use for the path planner module outside the normal control loop. When the operator chooses to have the system store a path which he or she has manually driven, the GPS path points are collected and stored. The resulting path definition array is very large and very closely-spaced. Before this new path is written to file, the collected data is processed through the path planner to generate a smaller, regularly spaced path definition array.
3.3.2.1.2 Control Law

Once the goal point is determined by the path planner, the control law calculates the steering command necessary to reach it. This calculation is, in essence, quite simple. The controller first determines the angle between the vehicle heading and the goal point bearing (θ in Figure 33), termed the heading error. Then PD control is applied to this heading error, and the result becomes the steering command. See Equation (4), where δ is steering angle.

\[ \delta = K_p \theta + K_d \frac{d\theta}{dt} \]  

In other words, the PD operator acts to drive the difference between vehicle heading and goal point bearing to zero. Though the basic calculation is simple, other subtleties must be included to adapt the controller for different speeds, path curvatures, and different vehicles.

Though this is a reference-tracking control problem, no feedforward is explicitly included. However, the preview-based nature of Sidhu’s algorithm makes it inherently feedforward. The algorithm does not act on current lateral deviation from the path, but rather on anticipated future deviation. Further improvement may be possible by analyzing the path curvature at or beyond the goal point and modifying the control law accordingly, but this method has not yet been explored.

3.3.2.1.3 Simulation

In order to investigate speed-dependent or vehicle-dependent parameter adaptation, the system was modeled in LabVIEW to conduct path-following simulations [84]. Simulations were performed on four different vehicles in an attempt to classify behavior by vehicle type or characteristic. The vehicles used are listed in Table 5.
Vehicle lateral dynamics were modeled with a 3-DOF state-space system including roll motion and compensation for multiple steering and suspension compliances. The model is given in Equation (5), with coefficients defined in Equations Error! Reference source not found. through (17). Nomenclature for this model follows the equations.

$$\begin{bmatrix}
m & 0 & m_h & 0 \\
0 & I_z & I_x & 0 \\
m_h & I_x & I_x & 0 \\
0 & 0 & 0 & 1
\end{bmatrix} \begin{bmatrix}
\dot{V} \\
\dot{r} \\
\dot{\phi}
\end{bmatrix} = \begin{bmatrix}
F_{Y_t} & F_{Y_s} & 0 & F_{Y_p} \\
M_{z_t} & M_{z_s} & 0 & M_{z_p} \\
0 & M_{X_t} & -B_f - B_r & M_{X_p} \\
0 & 0 & 1 & 0
\end{bmatrix} \begin{bmatrix}
V \\
r \\
\phi
\end{bmatrix} + \begin{bmatrix}
F_{Y_{t_k}} \\
M_{z_{t_k}}
\end{bmatrix} \delta_k \tag{5}
$$

$$F_{Y_t} = -2 \left( \frac{C_f' + C_r'}{U} \right) \tag{6}$$

$$F_{Y_s} = -mU - 2 \left( \frac{aC_f' + bC_r'}{U} \right) \tag{7}$$

$$F_{Y_p} = 2C_f' \tag{8}$$

$$M_{z_t} = -2 \left( \frac{aC_f' - bC_r' - C_Mf - C_Mr}{U} \right) \tag{9}$$

$$M_{z_s} = -2 \left( \frac{a^2C_f' - aC_{Mf} + b^2C_r' + bC_{Mr}}{U} \right) \tag{10}$$
\[ M_{z_y} = 2 \left( e_f \left( aC_f^r - C_{Mf}^r \right) - e_r \left( b^2 C_r^r + bC_{Mr} \right) + C_r^r \left( aC_{\phi_r} - bC_{\phi_r} \right) \right) \]  \hspace{1cm} (11) \]

\[ M_{z_y} = 2 (aC_f^r - C_{Mf}^r) \]  \hspace{1cm} (12) \]

\[ M_{x_y} = -m_s h U \]  \hspace{1cm} (13) \]

\[ M_{x_y} = -K_f - K_r + m_s gh \]  \hspace{1cm} (14) \]

\[ C_f^r = \frac{C_f}{1 + K_{mf} C_{mf}^r - K_{rf} C_f} \]  \hspace{1cm} (15) \]

\[ C_r^r = \frac{C_r}{1 + K_{mr} C_{mr} - K_{rr} C_r} \]  \hspace{1cm} (16) \]

\[ I_{xz} = m_s ch \]  \hspace{1cm} (17) \]

<table>
<thead>
<tr>
<th>UNITS</th>
<th>VEHICLE PARAMETERS</th>
</tr>
</thead>
<tbody>
<tr>
<td>( m )</td>
<td>kg</td>
</tr>
<tr>
<td>( m_s )</td>
<td>kg</td>
</tr>
<tr>
<td>( I_{zz} )</td>
<td>kg-m(^2)</td>
</tr>
<tr>
<td>( I_{xx} )</td>
<td>kg-m(^2)</td>
</tr>
<tr>
<td>( H_{CG} )</td>
<td>m</td>
</tr>
<tr>
<td>( h )</td>
<td>m</td>
</tr>
<tr>
<td>( a )</td>
<td>m</td>
</tr>
<tr>
<td>( b )</td>
<td>m</td>
</tr>
<tr>
<td>( c )</td>
<td>m</td>
</tr>
<tr>
<td>UNITS</td>
<td>VEHICLE PARAMETERS</td>
</tr>
<tr>
<td>-----------</td>
<td>-------------------------------------------------</td>
</tr>
<tr>
<td>$K_{ff}$ rad/N</td>
<td>front lateral force suspension compliance steer</td>
</tr>
<tr>
<td>$K_{fr}$ rad/N</td>
<td>rear lateral force suspension compliance steer</td>
</tr>
<tr>
<td>$K_{Mf}$ rad/(Nm)</td>
<td>front aligning torque suspension compliance steer</td>
</tr>
<tr>
<td>$K_{Mr}$ rad/(Nm)</td>
<td>rear aligning torque suspension compliance steer</td>
</tr>
<tr>
<td>$\varepsilon_f$ deg/deg</td>
<td>front roll steer</td>
</tr>
<tr>
<td>$\varepsilon_r$ deg/deg</td>
<td>rear roll steer</td>
</tr>
<tr>
<td>$C_{gf}$ deg/deg</td>
<td>front camber to roll coefficient</td>
</tr>
<tr>
<td>$C_{gr}$ deg/deg</td>
<td>rear camber to roll coefficient</td>
</tr>
<tr>
<td>$K_f$ Nm/rad</td>
<td>front overall roll stiffness</td>
</tr>
<tr>
<td>$K_r$ Nm/rad</td>
<td>rear overall roll stiffness</td>
</tr>
<tr>
<td>$B_f$ Nm/(rad/s)</td>
<td>front overall roll damping</td>
</tr>
<tr>
<td>$B_r$ Nm/(rad/s)</td>
<td>rear overall roll damping</td>
</tr>
<tr>
<td>$K_{sr}$ rad/rad</td>
<td>handwheel steering ratio</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>UNITS</th>
<th>TIRE PARAMETERS</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{cf}$ N/rad</td>
<td>front tire cornering stiffness</td>
</tr>
<tr>
<td>$C_{cr}$ N/rad</td>
<td>rear tire cornering stiffness</td>
</tr>
<tr>
<td>$C_{Mf}$ Nm/rad</td>
<td>front tire aligning moment stiffness</td>
</tr>
<tr>
<td>$C_{Mr}$ Nm/rad</td>
<td>rear tire aligning moment stiffness</td>
</tr>
<tr>
<td>$C_\gamma$ N/rad</td>
<td>tire camber stiffness</td>
</tr>
<tr>
<td>UNITS</td>
<td>MODEL VARIABLES, STATES</td>
</tr>
<tr>
<td>-------</td>
<td>-------------------------</td>
</tr>
<tr>
<td>$V$ m/s</td>
<td>lateral velocity</td>
</tr>
<tr>
<td>$r$ rad/s</td>
<td>yaw rate</td>
</tr>
<tr>
<td>$\phi$ rad</td>
<td>roll angle</td>
</tr>
<tr>
<td>$\delta_K$ rad</td>
<td>front tire steer angle</td>
</tr>
<tr>
<td>$\delta_{bw}$ rad</td>
<td>handwheel steer angle</td>
</tr>
</tbody>
</table>

Because tire cornering force was modeled as a linear function of slip angle, simulations were constrained to maneuvers designed for lateral acceleration at or below 0.3 g. Steering actuator and linkage dynamics were modeled using a 2nd-order transfer function developed by Brown [23].

Three parameters are available to determine the behavior of the proposed steering algorithm: lookahead ($LA$ in Figure 33), and $K_p$ and $K_d$ from Equation (4). Based on prior experience [85], the dominant parameters are lookahead and proportional gain ($K_p$), with derivative gain ($K_d$) contributing a secondary effect. Therefore, though all trials were run using multiple values of $K_d$, the trials focused on variation of the first two parameters.

The optimal combination of proportional gain and lookahead was computed for each trial based on two criteria. The first criterion was path-following error, which was calculated by finding the total area between the intended path and the actual vehicle trajectory, then dividing by the length of the path traveled.

The second criterion was controller stability. An FFT (Fast Fourier Transform) was calculated for the steering command on each trial. From observation, there was a critical frequency which corresponded to excessive steering oscillation on these maneuvers.
Trials revealing a spectral magnitude higher than a given threshold at the critical frequency were discarded, even if they followed the path more closely than the others.

The controller stability criterion was only used on trials above 15 mph (24 kph). Below this speed, the critical frequency was too close to the frequencies otherwise required for completing the steering maneuver.

![Figure 36: Sample Simulation Lateral Shift Maneuver](image)

Trials were conducted on a lateral-shift path, such as is shown in Figure 36, at multiples of 5 mph (8 kph) for combinations of integer values of $K_p$ and $LA$. Sample results for one such test, for the Jeep Cherokee tested at 15 mph on a lane-change path requiring 0.5 m/s$^2$ centripetal acceleration, are shown in Figure 37. The arrow points to the asterisk indicating the trial with lowest error, denoting the optimal combination of $K_p$ and $LA$ for this path and speed.
Figure 38 shows the effect of $K_d$ on optimal path tracking error for the 1994 Ford Taurus. (The error value plotted at each speed represents the lowest error for any tested combination of control parameters.) Some derivative gain is clearly helpful, especially at higher speeds. Larger $K_d$ yields slightly better results above 55 mph, but this margin is negligible.

For all vehicles simulated, a derivative gain of 1 gives very good results across the entire range of speeds, including results for lower speeds not shown in Figure 38. Therefore $K_d = 1$ was used for all subsequent analysis, and further control adaptation focused on the effects of proportional gain ($K_p$) and lookahead ($LA$).
As can be seen from the Taurus results in Figure 39 (left), optimal lookahead is well-represented by a line when plotted versus speed. This was also the case for the other vehicles simulated. When the linear fits for the four simulated vehicles are overlaid (Figure 39 right), it becomes clear that optimal lookahead is only weakly dependent on vehicle type, if at all. One curve fit, shown in Equation (18), would suffice for the Taurus, Jeep, and Excel. The Expedition, 55% heavier than the next closest vehicle, may benefit from slightly longer lookahead due to its mass-related slower response (Equation (19)).

\[
LA(m) = 0.40 \text{mph} + 0.2 \quad (18)
\]

\[
LA(m) = 0.44 \text{mph} + 0.2 \quad (19)
\]
As for determining a rule for the third control parameter, proportional gain, simulation results were difficult to interpret. Too many vehicle-dependent factors are involved to attempt a scheduling rule based on any one or two of them. In contrast to the empirical simulation-based approach used for $K_d$ and $LA$, a more theoretical approach is prudent when selecting proportional gain.

A proportional gain rule based on “yaw gain”, such as that developed in [84], may be practical (though it has not yet been experimentally tested). Yaw gain is defined by Gillespie [47] as the ratio of yaw rate to steer angle, $r/\delta_k$. The essence of steady state steering control lies in understanding this ratio for a given vehicle. So far, the algorithm has chosen the goal point based on the optimal lookahead distance. Then based on vehicle speed, one particular yaw rate will cause the vehicle to intercept this goal point.

Finally, the controller must order the proper steering angle to achieve this desired yaw rate. That is where, for the proposed control law, proportional gain becomes involved. Begin with Equation (4), neglecting for now the derivative term:
\[ \delta_{hw} = K_p \theta \]  

The desired yaw rate \( r_{des} \) is implicit in bearing error \( \theta \).

\[ r_{des} = \frac{\theta}{t} \]  

where \( t \) is the time until the vehicle reaches the goal point. This \( t \) can be calculated as the ratio of the lookahead distance to vehicle speed. Rearranging this, one can calculate a theoretical \( Kp \) value as shown in Equations (22) and (23). Note that yaw gain, the denominator term in parentheses in Equation (23), is stated in terms of handwheel angle rather than tire angle.

\[ \theta = r_{des} t = r_{des} \frac{LA}{vel} \]  

\[ K_p = \frac{\delta_{hw}}{\theta} = \left( \frac{\delta_{hw}}{r_{des}} \right) \frac{vel}{LA} = \left( \frac{1}{r_{des} / \delta_{hw}} \right) \frac{vel}{LA} \]  

Yaw gain is dependent on many factors, including vehicle speed, mass, wheelbase, and tire properties. These properties are typically not available for every test vehicle, making theoretical determination difficult. But any common steady-state test maneuver, such as those outlined in [7], can be used to quickly identify the yaw gain for a vehicle.

A sample calculation can be made using data from the experimental test shown in Figure 36 (2 m/s² lateral shift at 20 mph, LA = 5 m). The measured yaw rate, goal point bearing, and handwheel angle for this test are shown in the same figure at right. Though not a steady state test, the ~1.5-second period around time = 15 sec approaches this.

First estimate the yaw gain from Figure 36 at time = 15 sec.

\[ \frac{r}{\delta_{hw}} = \frac{13 \text{deg/s}}{83 \text{deg}} = 0.157 \text{s}^{-1} \]
This yaw gain was similarly calculated from experimental results on the same vehicle and same speed at 1 m/s² and 3 m/s². Results (0.153 and 0.155 s⁻¹, respectively) were within 3% of the value shown in Equation (24).

Now calculate a theoretical $K_p$ using Equation (24) with the known lookahead and velocity:

$$
K_p = \left( \frac{1}{0.157 \text{s}^{-1}} \right) \frac{8.9 \text{m/s}}{5 \text{m}} = 11.4
$$

(25)

This compares favorably with the optimal $K_p$ of 12 calculated in simulation for this scenario [84]. As mentioned, however, the control law based on the relationship shown in Equation (23) has yet to be validated in simulation and experiment.

### 3.3.2.1.4 Results for PD Heading Error Control

The right plot in Figure 40 shows path following error for a 1997 Honda CR-V employing the PD Heading Error control strategy. The vehicle gradually accelerated from a standstill to 11 m/s, reaching 0.4 g on this sloped asphalt surface with 2.8% grade. Here the rules developed above were applied for $K_d$ and $LA$, but proportional gain was manually-tuned to a fixed value. Allowing a maximum deviation of 5 cm (2 in.) from the intended path on this run, this control strategy obviously has potential.
Yet the difficulty of selecting an appropriate proportional gain remains. Proportional gain is the key link between steering angle and yaw rate, varying with a vehicle’s speed, mass, wheelbase, steering ratio, tire properties, and more. In lieu of pursuing an experimental parameter identification routine to construct a model-based proportional gain, it was decided to switch to a different control strategy where the key vehicle parameters were built-in.

### 3.3.2.2 Kinematic Algorithm Based on Vehicle Parameters

The lateral control strategy ultimately used was a mix of the PD Heading Error method and a kinematic algorithm. Preserved from the former method was the path planner module outlined in Section 3.3.2.1.1 as well as the guideline for lookahead determination. What the new algorithm offered was a way to determine proportional gain based on kinematic relationships and vehicle dynamics laws, knowing only four key vehicle parameters.

The kinematic algorithm presented here is a modification of the algorithm presented by Tseng, et. al. in [63]. Figure 41 shows the schematic from which the kinematics are
derived. The shaded vehicle represents the actual current position, and the dotted-outline vehicle represents the vehicle position if it were then precisely on path. This kinematic algorithm, too, is a proportional controller; here the proportional gain acts on the projected lateral deviation from the path \(d\), rather than the angular heading error. See Equation (26).

\[
\delta = K_p d = K_p (y + y_{lead})
\]  

(26)

Refer now to Figure 41 to outline the kinematic algorithm derivation. Note first the following definitions, where \(p\) is preview distance (lookahead) and \(L_b\) is the distance between the rear axle and the vehicle CG:

\[
y_{lead} = y_{projected} + y_{road}
\]  

(27)

\[
p' = p + L_b.
\]  

(28)

Table 6 shows the notation used for this derivation.
<table>
<thead>
<tr>
<th>Variable Name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>δ_{hw}</td>
<td>handwheel steer angle</td>
</tr>
<tr>
<td>δ_{rw}</td>
<td>roadwheel steer angle</td>
</tr>
<tr>
<td>k</td>
<td>vehicle understeer gradient</td>
</tr>
<tr>
<td>K_p</td>
<td>proportional gain</td>
</tr>
<tr>
<td>K_{sr}</td>
<td>steering ratio</td>
</tr>
<tr>
<td>LA, p</td>
<td>lookahead or preview distance</td>
</tr>
<tr>
<td>L</td>
<td>vehicle wheelbase</td>
</tr>
<tr>
<td>L_b</td>
<td>distance between rear axle and vehicle</td>
</tr>
<tr>
<td>u</td>
<td>vehicle forward speed</td>
</tr>
</tbody>
</table>

Table 6: Notation for Kinematic Algorithm Derivation

Then from trigonometry, assuming $\theta$ is a small angle,

$$y_{lead} = p' \sin \theta = p' \theta$$  \hspace{1cm} (29)

$$p' = R(2\theta).$$  \hspace{1cm} (30)

Solving Equation (30) for $\theta$ and substituting into (29), we rearrange to find a relationship for $R$.

$$y_{lead} = p' \theta = \frac{p'^2}{2R}$$  \hspace{1cm} (31)

$$R = \frac{p'^2}{2y_{lead}}$$  \hspace{1cm} (32)

Returning to the theory of vehicle dynamics, recall the transfer function relationship between lateral acceleration and roadwheel steer angle derived from the bicycle model [86], shown in Equation (33). In steady-state, and defining understeer gradient $k$ as shown in Equation (34), this reduces to Equation (35).
\[
\left(\frac{a_y}{\delta_{rw}}\right)(s) = \frac{I_y s^2 + L b (c_f c_r / u)}{m l_x s^2 + \left[ m (a^2 c_f + b^2 c_r) / u + L c_f c_r / u^2 \right] + L^2 c_f c_r / u^2 + m (b c_r - a c_f)}
\]  

(33)

\[
k = \frac{m g (b c_r - a c_f)}{L c_f c_r}
\]  

(34)

\[
\left(\frac{a_y}{\delta_{rw}}\right)_{ss} = \frac{u^2}{L + k u^2}
\]  

(35)

Rearrange Equation (16) in terms of steer angle \( \delta_{rw} \), then replace lateral acceleration by the square of forward velocity divided by curve radius (Equation (36)). Substitute for \( R \) the terms from Equation (32), and finally convert the roadwheel steer angle to handwheel angle (Equation (38)).

\[
\delta_{rw} = \frac{L + k u^2}{u^2} a_y = \frac{L + k u^2}{u^2} \left(\frac{u^2}{R}\right)
\]  

(36)

\[
\delta_{rw} = \frac{2 (L + k u^2)}{p^2} y_{lead}
\]  

(37)

\[
\delta_{hw} = \frac{2 (L + k u^2)}{(p + L_b)^2} K_{sr} y_{lead} = K_p y_{lead}
\]  

(38)

Then the proportional gain defined by this algorithm is expressed in Equation (39), replacing the variable \( p \) with \( LA \).

\[
K_p = \frac{2 (L + k u^2)}{(LA + L_b)^2} K_{sr}
\]  

(39)

Implementation of this kinematic algorithm was simplified by minor modifications. Rather than separately calculate the three components of \( d \) shown in Figure 41, translate it into familiar terms from the previously-discussed algorithm. Figure 42 illustrates that
the projected lateral path deviation is simply the product of lookahead and sine of the heading error.

![Diagram of lateral offset calculation](image)

Figure 42: Modified Lateral Offset Calculation

With this modification, the proportional gain acts not only on \( y_{\text{lead}} \) but on \( y \) (the current lateral deviation) as well. Tseng, et.al. begin the derivation as if the proportional gain acts on \( d \), the total projected path deviation (Equation (26)). Their derivation concludes, however, with Equation (37), which is in terms of \( y_{\text{lead}} \) only. No explanation is offered as to whether \( y \) is separately treated. Making the adjustment described in Figure 42, however, has yielded good results.

### 3.3.3 Longitudinal Control Implementation

Throttle and braking control are combined in one algorithm on the ATD. Though this was necessitated by the mechanical arrangement of the actuator, it did not prove a limitation.
This controller will be discussed from the perspective of a desired speed-profile tracking problem. The ATD is capable of following either speed vs. time or speed vs. distance profiles. An example of the former is shown in Figure 43.

![Figure 43: Sample Speed vs. Time Profile](image)

A feedforward mechanism based on the static process characteristic (SPC), as described by Astrom and Hagglund [87], speeds reaction time in this setpoint-tracking controller.

![Figure 44: Sample Static Process Characteristic (SPC) for Brake and Throttle Actuator](image)
The SPC is “a curve that gives the steady state relation between process input $u$ and process output $y$”. For the speed tracking problem, accelerator pedal position is the process input and vehicle speed is the process output. At least that is how the SPC is treated; obviously the brake pedal comes into play in the final controller.

A sample static process characteristic, computed for the F-150, is shown in Figure 44. Actual data points collected are denoted by the blue circles, and the red line is the 3rd order polynomial curve fit for the data. The dependent axis is the servomotor position in degrees, where positive angles are acceleration and negative angles are braking.

Strictly speaking, the plot axes should be reversed for a static process characteristic. However, this plot was designed to reflect the SPC as it is used by the ATD program, where the controller receives a desired speed command and must output an appropriate feedforward pedal position.

Note how the third order fit, while not perfect, captures an essential characteristic of the throttle vs. speed relationship. With the actuator centered (at zero position), the vehicle is idling, moving forward typically at $2 – 2.6$ m/s. Zero speed does not correspond to zero pedal displacement, but rather some amount of brake pedal displacement. The curve in Figure 44 reveals that the feedforward action will actuate the brake for all speeds below $1.5$ m/s.

Currently this feedforward tuning is only semi-automatic. One program drives the actuator to discrete pedal positions and outputs speed values, and another program converts the data to corresponding polynomial coefficients. The operator must relay the data points from the first program to the second. Complete automation of this process, a simple step, is now underway.

Complementing the feedforward mechanism is a proportional-integral controller acting on the difference between desired and actual speed. Different gains are implemented for the braking and accelerating directions. One set of gains has been found satisfactory for
the vehicles tested to date. The proportional gain in braking is much higher than that used for throttle, because the brake action is not assisted by a feedforward component.

3.4 Safety

As with any autonomous vehicle, even those in a controlled environment, solid safety measures are needed to protect lives and equipment. Complex systems like the ATD have countless failure modes; the designer must carefully plan to eliminate or minimize damage from all conceivable failures. Containment of failures falls into two categories: internal and external.

3.4.1 Internal Containment

Many problems can be detected and resolved by the control system itself. Sensor failure is one example. If GPS signal is lost for a period, the controller will still be able to function, but the navigational position data will begin to degrade. Currently internal containment is implemented only at the start of a test. Before a run can commence, communication between all computers is verified and the quality of the RT3002 output must exceed a specified level (thus the reason the operator indicates whether a GPS base station is being used on the configuration screens shown in Section 3.3.1.2). If any test fails, the operator receives an error-specific message on the laptop interface.

Because test runs are often short, checking the system at test initiation catches most problems. For those which do occur in the middle of a test, external containment measures are present.

3.4.2 External Containment

If the problem is of a nature which the controller can either not detect or not contain, a test operator must intervene. To facilitate this, safety switches on the actuators have been hard-wired to cut power to the motors when they are released (Figure 16 and Figure 17).
On the handwheel, releasing the button also reengages the clutch (by spring energy) and returns steering control to the operator. Also, as mentioned while discussing the brake and throttle actuator, neither pedal linkage is rigidly connected to the ballscrew. This allows the driver to override the actuator and apply either brake or throttle at any time without having to fight motor torque or inertia.

3.5 GPS and Inertial Sensors

The GPS/INS feedback sensor, or “motion package”, not only closes the ATD control loop but also provides all system performance validation data. Other means, such as laser networks, do exist to provide external validation, but these are generally cumbersome and costly. Since GPS and inertial sensors are so critical for the ATD, they warrant some discussion concerning their theory, operation, and application.

3.5.1 Global Positioning System

The Global Positioning System (GPS) consists of at least 24 satellites GPS traveling at an altitude of 20,200 km in very precise orbits about the Earth [89]. The satellite orbits are designed such that at least six satellites should be in view at all times from any place on Earth.

The system was initially established in the 1970’s for the U.S. military, but was extended for civilian use in 1983. Even then, the signal remained scrambled to limit accuracy for nonmilitary uses until 2000. Similar systems have been launched by the former Soviet Union (GLONASS) and, more recently, the European Union (Galileo).

GPS satellites transmit information to Earth on microwave-frequency signals. GPS receivers interpret this information, calculating the travel time of satellite signals by comparing the time a signal was transmitted with the time it was received. By multiplying this time by the speed of the wave, the receiver can determine how far each signal traveled. Using this distance information from at least four satellites, and knowing
where these satellites are with respect to the Earth and one another, the GPS receiver finds the intersection of these spheres and determines its own position. Based on this process, known as trilateration, a standard GPS receiver can achieve a position accurate to within approximately 10 meters.

GPS accuracy, however, can be enhanced by many means. “Differential” GPS, or DGPS, requires an additional receiver fixed at a known location. The stationary receiver compares its known location to what it calculates using GPS, and sends this error correction to roving units. Errors are induced by fluctuating travel times of satellite signals, typically caused by reflection in the ionosphere or orbital path errors. The stationary receiver simply quantifies this error for the satellites it is receiving and allows roving receivers nearby to correct their calculations accordingly.

Because these errors are location dependent, the usefulness of differential corrections is proportional to the proximity of the stationary receiver, or base station. One network of base stations operated by the U.S. government is known as WAAS, or Wide-Area Augmentation System. This signal is available anywhere in the United States, and nominally improves position accuracy to 3 meters. Corrections are also publicly available through radio beacons operated by the U.S. Coast Guard. Though radio beacon coverage is still more dense near large bodies of water, thirty-seven broadcast stations are currently operational in the Nationwide DGPS program, and this number continues to grow [88].

Commercial differential signals are available for an annual subscription fee. OmniSTAR, Landstar, and Starfire are three examples, providing a range of offerings with sub-meter accuracy. OmniSTAR, for instance, operates a series of base stations along the periphery of the U.S. [89]. These base stations process the GPS signals they receive, calculate position-dependent error signals, then rebroadcast this error packet to subscribers over their own communications satellite.

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For optimal accuracy at a particular site a local base station can be implemented, achieving position accuracy as good as 1 cm (RMS). Many higher-end commercial GPS receivers can be configured to operate either as a “rover” unit or as a base station, broadcasting corrections through an integral or separate RF transceiver.

Position accuracy may also be improved by receiving multiple satellite signal frequencies. The most common civilian GPS frequency, L1, is broadcast at 1575.42 MHz [89]. If a receiver compares data on this frequency with the same data on L2 (1227.60 MHz), it can better estimate the signal path error caused in the ionosphere.

Position accuracy is not the only feature which can be improved. Velocity can be measured using the Doppler shift on GPS signals with an accuracy of 30 mm/s (1σ, horizontal velocity) even without differential corrections [76]. Vehicle attitude can also be determined quite accurately with yet another technique: use the phase difference of carrier waves received at two antennas mounted on the same vehicle.

As powerful a tool as GPS is, it does have limitations. The signal will pass easily through clouds, but may be distorted by heavy rain and is certainly blocked by trees and buildings. A receiver may also become confused when some satellite signals are received both directly and reflected off a nearby wall (known as “multi-path” error). Because the satellites are spread across the sky, GPS systems may be unable to obtain a fix even when the sky directly overhead is not obscured, such as in a canyon. Tall buildings in cities create “urban canyons”, areas where perhaps only one or two satellites are available.

With such limits, systems relying heavily on GPS may be restricted to open spaces. But manufacturers have recently been working to minimize these limitations with sophisticated tracking techniques and complementing GPS with other sensors. Another option is to simply increase the number of satellites used in the solution by allowing the receiver to interpret signals from GLONASS, the Russian parallel to GPS. Soon another option will be available as well; the European Union has been launching navigation
satellites as part of a new system known as Galileo. Galileo will “definitely become operational in 2008” [90], according to a 2004 EU press release.

Even operating under a wide-open sky, GPS can exhibit irregular behavior. When a new satellite comes over the horizon, or another passes under it, the position solution can jump around. This problem can also be evident when a vehicle passes near an obstruction which temporarily blocks one or more satellites from view.

In addition to this, GPS offers a relatively low bandwidth for the processed position, velocity, and orientation output. The combination of these factors dictates that, for real-time high speed vehicle navigation purposes, GPS feedback alone is insufficient. Many sensors are available to complement GPS for this purpose, such as fifth-wheel encoders or their optical counterparts. The most popular solution, however, is to combine GPS with inertial sensors, as are described in the following section.

3.5.2 Inertial Sensors

Though ancient efforts in navigation were aided by tools such as the magnetic compass and sextant, modern inertial navigation began with the development of the gyroscope. Leon Foucault first publicized the gyroscopic effect in 1852 [91], but it was not until the early twentieth century that the gyroscope was developed as a directional reference. In aircraft, gyros were also used for artificial horizons and rate-of-turn indicators; on ships, they provided a stable platform for directing gun fire. One of the largest technological leaps was the German V2 rocket from World War II, demonstrating the use of inertial feedback for airborne guidance over long distances.

Inertial navigation systems (INS) fall into one of two categories: stable platform and strapdown. All of these early examples required a stable platform, or a gimbal mounting which allows the rotating mass to maintain a constant orientation in the inertial frame of reference (isolated from the rotational motion of the vehicle). Even today, the most accurate inertial systems require a stable platform. Instruments used to navigate
submarines underwater for months, or placed on spacecraft headed for distant planets, require the 0.0001 deg/hr gyroscopic error these units can provide [91].

The cost of stable platform devices, however, is an obstacle to widespread commercial use; in their place has come strapdown inertial navigation technology. Strapdown devices, as their name suggests, do not require gimbal mountings. Rotating-mass gyroscopes are replaced with other devices which measure the gyroscopic effect. These devices include ring-laser gyro, fiber-optic gyro, and MEMS gyro. These are ordered generally in descending order of accuracy and cost, though there is naturally a range of quality in the manufacture of each type.

Regardless the type of gyroscopic system employed, however, the gyros are only half of the INS. The gyroscopes maintain a constant inertial frame of reference for a vehicle, but accelerometers must provide information on the vehicle’s linear motion. Three orthogonally-oriented accelerometers can sense any component of acceleration, which is then integrated into vehicle velocity and, in turn, position. Accelerometers track the linear motion of the vehicle, but the gyroscopes indispensably dictate in which direction the linear changes should be accounted.

Inertial navigation systems are capable of very high bandwidth, but all are prone to drift. Any error in the gyroscopic reference frame causes error in accounting of acceleration, which is then continuously integrated. Especially in the Earth’s gravitational field, any error in attitude makes some component of the gravitational vector appear to be acceleration in another plane, quickly causing large lateral velocity and position errors. Though error rates for the sensors themselves may be limited, there is no bound to the errors in the integrated values in inertial navigation.

For this reason, the output of lower cost inertial navigation units cannot be used alone for vehicle feedback in a test driver such as the one proposed. On the other hand, GPS receivers have limited bandwidth but bounded error. Thus the two sensor types are quite complementary.
3.5.3 Sensor Fusion

Yet though GPS and INS data are complementary, accurately combining data from the two sources is not trivial. One factor is the physical separation of the inertial sensors and GPS; this separation implies that they do not measure the same dynamics. These sensors are affected differently by a vehicle’s roll and body slip, for instance. Various mathematical means exist for handling this separation, known as the “lever arm” effect (e.g., [92]).

Another complication arises from the propensity of GPS to jump, as described before, or to give erroneous data near partial obstructions. The entrance and exit from a tunnel (or even short underpass) is one such notorious case of the latter. As the vehicle enters a tunnel, its satellites disappear from view one at a time, from one horizon to the other. The last few fixes (say at 10 Hz) are computed from satellites on one side of the vehicle, generally nearer the horizon. The position and velocity computed from this skewed aspect make the vehicle appear to be drifting at an angle to its actual heading. Then in the tunnel, no GPS is available, so the inertial solution proceeds with the most recent (erroneous) heading, quickly integrating into a large position error. Cases like this must be handled as exceptions in the fusion process.

3.5.4 Converting Position Data to Local Coordinates

The position output of a GPS system is typically given in latitude, longitude, and altitude (LLA). These coordinates, however, are not directly useful for computing position relative to a local datum or distance from a known path. Position data is more intuitive and easier to manipulate when converted to the local tangential plane.

This conversion must begin with a discussion of reference frames. Even our starting point, LLA, is subject to various standards. Contrary to popular perception, a given latitude and longitude does not uniquely specify one given point on Earth. Rather, one must also know the reference datum in which those coordinates are specified. Because
the Earth is not spherical, not even a regular ellipsoid, different datums are more accurate in different parts of the Earth.

WGS-84 is the most commonly used reference datum in North America, and is often the default for commercial GPS receivers, and will thus be specified here. This datum is actually a set of coordinates defining an ellipsoid, shown in Table 7, as agreed upon by the World Geodetic System Committee in 1984. In this system, then, one obtains a position specified by angular measurements north or south of the equator, east or west of the Greenwich (or prime) meridian, and height above or below the ellipsoid surface.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>length of the semi-major axis (equatorial)</td>
<td>$R = 6378137.0$ m</td>
</tr>
<tr>
<td>length of the semi-minor axis (polar)</td>
<td>$r = R(1 - f)$</td>
</tr>
<tr>
<td>flattening of the ellipsoid</td>
<td>$f = (R - r) / R$</td>
</tr>
<tr>
<td>major eccentricity of the ellipsoid</td>
<td>$e = \sqrt{2 - f}$</td>
</tr>
<tr>
<td>Earth’s rotational rate</td>
<td>$\Omega = 7.292115 \times 10^{-5}$ rad/s</td>
</tr>
</tbody>
</table>

Table 7: WGS-84 Parameters, after [91]

The second reference frame to mention is the inertial frame, which has its origin at the Earth’s center and axes which do not rotate relative to the “fixed” stars. Next is the Earth frame, otherwise known as ECEF (Earth-centered, Earth-fixed). It also has its origin at the center of the Earth, but has X- and Y-axes which rotate with the Earth. Here the X- and Y-axes both lie in the equatorial plane, and the X-axis passes through the prime meridian (Figure 45, left).
The navigation frame is the one most useful for terrestrial navigation covering a small area. It is also known as the local tangent plane (LTP) or NED, for its three components North, East, and Down. Finally, there is the body frame, with axes fixed to the body of the vehicle, regardless of its orientation. Strapdown inertial data is most easily output in body-frame axes, for it is to these which the device is “strapped down”. The SAE coordinate system definitions, shown at right in Figure 45, are used for the body axes.

To employ LLA information given by a GPS/INS motion pack, two steps are necessary. First, convert LLA to ECEF coordinates with the following method [91]. Defining \( N \), the radius of the Earth’s curvature at a given latitude (\( \phi \)) as

\[
N = \frac{R}{\sqrt{1 - e^2 \sin^2 \phi}}. \tag{40}
\]

Then given longitude (\( \lambda \)) and defining height (\( h \)) as

\[
h = \text{altitude above mean sea level} + \text{geoidal separation (m)} \tag{41}
\]

the position coordinates in ECEF are
\[
X_e = (N + h) \cos \phi \cos \lambda \quad (42)
\]
\[
Y_e = (N + h) \cos \phi \sin \lambda \quad (43)
\]
\[
Z_e = \left( \frac{r^2}{R^2} \right) N + h \sin \phi . \quad (44)
\]

To convert these ECEF coordinates to the navigation frame, the coordinate transformation in Equation (45) is prescribed.

\[
\begin{bmatrix}
N \\
E \\
D
\end{bmatrix}
= \begin{bmatrix}
- \sin \phi \cos \lambda & - \sin \phi \sin \lambda & \cos \phi \\
- \sin \lambda & \cos \lambda & 0 \\
- \cos \phi \cos \lambda & - \cos \phi \sin \lambda & - \sin \phi
\end{bmatrix}
\begin{bmatrix}
X_e \\
Y_e \\
Z_e
\end{bmatrix}. \quad (45)
\]

Note that Equation (45) works only to transform changes in position, not to convert a specific XYZ location to NED. In other words, \( X_e \) is really \( \Delta X_e \) (and so forth).

### 3.5.5 Converting Inertial Sensor Data to Inertial Navigation in Local Coordinates

With the axes systems now defined, we can return to the inertial sensor to discuss its navigational function. Recognize first that unless the inertial sensor is gimbal-mounted, output from the six axes of a typical inertial measurement unit cannot be directly used for navigation. For the more-common strapdown devices, the inertial data must be continually converted from the body-frame axes to an external reference frame.

To output dynamic data useful for navigation, strapdown inertial systems must be given their initial orientation. Alternately, this initial orientation may be calculated by comparing IMU output to GPS output during an initialization routine. Once the unit orientation is known (and updated every clock cycle per output from its own gyroscopes), the accelerometer output can be properly integrated to determine vehicle velocity and position.

Maintaining the IMU orientation may be done with several different mathematical tools, including quaternion methods and the “direction cosine matrix”. Quaternion methods are
a bit more computationally-intensive, but eliminate the discontinuity present in the other methods when pitch or roll angles exceed 90°. The direction cosine matrix, on the other hand, suffices when vehicle orientation is not expected to see these limits. The latter method will be outlined here [91].

IMU orientation is contained within the direction cosine matrix $C^b_n$. The sub- and superscripts denote that this matrix converts acceleration data from the body frame to the navigation frame. Each time step, the sensor computer executes the following procedure. Employing $p$, $q$, and $r$, the three rotational rates of roll, pitch, and yaw respectively (in rad/s), first form the skew symmetric form of the rate vector $\omega^b_{nb}$ as shown in Equation (46). $\omega^b_{nb}$ is the rotation rate of the body frame with respect to the navigation frame (LTP), measured in body coordinates. This skew symmetric matrix is used in the next step to update the direction cosine matrix.

$$\Omega^b_{nb} = \begin{bmatrix} 0 & -r & q \\ r & 0 & -p \\ -q & p & 0 \end{bmatrix}$$  (46)

Now compute the change in the direction cosine matrix per Equations (47) and (48).

$$\dot{C}^n_b = C^n_b \Omega^b_{nb}$$  (47)

$$\Delta C^a_n = \dot{C}^n_b \Delta t$$  (48)

Read the body specific forces ($f_b = [f_x, f_y, f_z]^T$) output by the accelerometers and convert to m/s$^2$. Now apply the appropriate version of what is known as the navigation equation. Described in Equation (49) is the navigation equation for operation in the local tangent plane. Vehicles (e.g., aircraft) which travel for longer times or over larger distances may be better tracked using a different reference frame. These various formulations are described in [91].
\[ \dot{V}_e^n = C_b^n f_b^n - [2 \omega_{ie}^n + \omega_{en}^n] \cdot V_e^n + g_i^n \] (49)

\( \dot{V}_e^n \) is the time rate of change of velocity in the navigation frame (LTP). The first term on the right, \( C_b^n f_b^n \), represents the body frame accelerations transformed to LTP. The next term, \( [2 \omega_{ie}^n + \omega_{en}^n] \cdot V_e^n \), are the Coriolis corrections. One may choose to ignore this term in some circumstances. Here, for instance, GPS will typically be available at 10 Hz to update the inertial solution. The Coriolis effect may also be smaller than the sensor noise level for lower quality MEMS gyros and accelerometers. And again, the distances traveled will be relatively small and speeds rather low (compared, for instance, to a Mach 2 missile). The final term of the navigation equation, \( g_i^n \), is the local gravitational vector in the navigation frame.

From the LTP accelerations calculated in Equation (49), integrate to update the vehicle velocity and position (Equations (50) and (51) respectively).

\[ V_k^n = V_{k-1}^n + \dot{V}_e^n \Delta t \] (50)

\[ P_k^n = P_{k-1}^n + V_{k-1}^n \Delta t \] (51)

### 3.6 ATD Functionality and Evaluation

The Automated Test Driver’s proposed functionality can be divided into three categories: full vehicle control, lateral control, and longitudinal control.

To properly evaluate the ATD, its functions must be tested on a variety of vehicles. Performance was assessed on three vehicles which fairly represent the intended spectrum of usage, described in the next section. Specific evaluation methods are listed in Table 8.
### Table 8: Proposed ATD Functionality and Evaluation

<table>
<thead>
<tr>
<th>Control Type</th>
<th>Controller Function</th>
<th>Evaluation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Vehicle Control</td>
<td>Follow a predefined path and speed profile (autonomously)</td>
<td>Measure deviation from path and speed profile, verify controller stability.</td>
</tr>
<tr>
<td>Lateral Control</td>
<td>Follow a predefined path (steering only)</td>
<td>Measure deviation from path and verify controller stability.</td>
</tr>
<tr>
<td></td>
<td>Follow prescribed handwheel input, e.g. sine sweep</td>
<td>Compare actual actuator position to desired, evaluate controller bandwidth.</td>
</tr>
<tr>
<td>Longitudinal Control</td>
<td>Brake: Match desired pedal displacement</td>
<td>Measure deviation from desired pedal force / pedal displacement. Need smooth actuation.</td>
</tr>
<tr>
<td></td>
<td>Throttle: Accelerate at desired rate to desired final speed</td>
<td>Measure deviation from desired speed and monitor acceleration. Need smooth actuation.</td>
</tr>
<tr>
<td></td>
<td>Throttle / Brake: Follow desired speed profile</td>
<td>Measure deviation from desired speed. Check smooth actuation and sensible transition between throttle and brake.</td>
</tr>
</tbody>
</table>

**3.7 Test Procedure**

The ATD was evaluated with path-following maneuvers in full-autonomous mode as well as open-loop maneuvers for characterizing vehicle response. The test vehicles and series of tests employed are described below.
3.7.1 Vehicles

Three vehicles were selected for the battery of tests: a 2000 Toyota Camry, 1997 Honda CR-V, and a 1992 Ford F-150 XLT pickup truck. To demonstrate the adaptability of the ATD to different vehicles, the test vehicles chosen represent a sampling from three different vehicle genres: the passenger sedan (Camry), the small sport utility vehicle (CR-V), and the full-size truck (F-150). Specifications for all vehicles are listed in Table 9.

![Test Vehicles: Camry (left), CR-V (middle), F-150 (right)](image)

The 2000 Camry was in the best condition with the newest tires. Its steering was precise and responsive compared to the other test vehicles, though naturally less so than a sports car. The ride was smooth and very quiet.

The 1997 CR-V begins to show evidence of its high mileage. Its tires are in good condition, but the original shock absorbers permit substantial body lean. Suspension bushings are worn and cracked, transmitting more noise and vibration to the driver.

Despite low mileage, the 1992 F-150 showed its years with a large steering backlash and uneven steering effort in right and left directions. It is an unsophisticated vehicle designed for hauling large payloads; without that load, the suspension is very stiff and jarring.
<table>
<thead>
<tr>
<th></th>
<th>Toyota Camry CE</th>
<th>Honda CR-V</th>
<th>Ford F-150 XLT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model year</td>
<td>2000</td>
<td>1997</td>
<td>1992</td>
</tr>
<tr>
<td>Approximate miles</td>
<td>106,000</td>
<td>145,000</td>
<td>63,000</td>
</tr>
<tr>
<td>Engine</td>
<td>2.2L 4-cyl (150 ft-lb / 136 hp)</td>
<td>2.0L 4-cyl (133 ft-lb / 126 hp)</td>
<td>4.9L 6-cyl (265 ft-lb / 145 hp)</td>
</tr>
<tr>
<td>Transmission</td>
<td>4-spd automatic</td>
<td>4-spd automatic</td>
<td>4-spd automatic</td>
</tr>
<tr>
<td>Test weight (kg)</td>
<td>1551</td>
<td>1586</td>
<td>2183</td>
</tr>
<tr>
<td>Tires</td>
<td>P195/70SR-14 Yokohama Avid Touring (5000 mi)</td>
<td>P205/70TR-15 Yokohama Avid TRZ (25000 mi)</td>
<td>235/75-15 Firestone Wilderness AT (good tread)</td>
</tr>
<tr>
<td>Length (m)</td>
<td>4.800</td>
<td>4.480</td>
<td>5.410</td>
</tr>
<tr>
<td>Height (m)</td>
<td>1.397</td>
<td>1.674</td>
<td>1.803</td>
</tr>
<tr>
<td>Wheelbase (m)</td>
<td>2.667</td>
<td>2.620</td>
<td>3.378</td>
</tr>
<tr>
<td>Weight as tested (F/R)</td>
<td>60.0% / 40.0%</td>
<td>52.6% / 47.4%</td>
<td>54.4% / 45.6%</td>
</tr>
<tr>
<td>CG height (mm)</td>
<td>548</td>
<td>669</td>
<td>693</td>
</tr>
</tbody>
</table>

Table 9: Test Vehicle Specifications

3.7.2 Test Sites

Unless otherwise noted, all tests reported here were conducted at one of two sites. The first was a large parking lot for a structure abandoned five years ago. The asphalt is still in decent shape, though the surface is littered with patches of gravel and debris. It also has an irregular pattern of drainage slopes and shallow channels for runoff.
The second test site was the Vehicle Dynamics Area (VDA) at Transportation Research Center (TRC) in East Liberty, Ohio, shown in Figure 47. The VDA comprises a 50-acre (200,000 m²) asphalt rectangle enclosed by a 3.4-km banked oval track. A 1% runoff slope runs from north to south. The quality and regularity of this surface was, of course, excellent.

![Vehicle Dynamics Area (VDA) at TRC](image)

Figure 47: Vehicle Dynamics Area (VDA) at TRC

### 3.7.3 Initialization Procedure

First the RT3000 configuration must be completed, establishing the orientation of the inertial unit (red box) in the vehicle and the position of the GPS antenna relative to the red box. Then the LabVIEW program is run for finding the throttle static process characteristic (SPC). A third order curve fit of actuator position vs. desired speed is obtained from the data thus collected, and these coefficients are put into the main ATD control program. Also set in the ATD program are the four vehicle parameters used to calculate the lateral control algorithm gain.
Following this process, the ATD is ready to run. If the understeer gradient for this vehicle (one of the controller inputs) is not known, then the first maneuver conducted will determine this value. The operator enters an estimate of understeer gradient into the program and uses the ATD to run a Constant Radius Test (CRT), as outlined in SAE J266 [7]. Several other common tests could be manually conducted to determine understeer gradient, but once the ATD has been installed and initialized, it is much faster to use the ATD to conduct the CRT than any manual method.

Even with imperfect tuning, the ATD can stay close enough to the desired circle to accurately assess understeer gradient. Sometimes the operator must make a small adjustment to the lookahead parameter to improve behavior on the first few runs. For instance, the initial settings on the CR-V yielded an oscillatory steering controller. With one adjustment, the results of the second CRT attempt were good enough to be published in Chapter 4. On the F-150, stability was achieved with one adjustment, but a second adjustment and trial was made to improve path following. For the Camry, no adjustment was necessary; it employed the same exact set of parameters from the first test to the last.

3.7.4 Tests

<table>
<thead>
<tr>
<th>Test</th>
<th>Trials</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant Radius Test (CRT)</td>
<td>3x clockwise, 3x counterclockwise</td>
</tr>
<tr>
<td>Figure Eight</td>
<td>Six continuous loops at 5, 8, and 10 m/s</td>
</tr>
<tr>
<td>Random Path</td>
<td>Three identical trials, each with varying speed</td>
</tr>
<tr>
<td>Oval Path</td>
<td>One or more loops at 14, 20, and 25 m/s</td>
</tr>
</tbody>
</table>

Table 10: Battery of Automated Tests

The ATD was subjected to a battery of four tests to evaluate its ability to simultaneously control all three driver inputs on three different vehicles. The tests include a circle test
(or constant radius test), figure eight, random path, and an oval. Through these tests, the path-following and speed control accuracies of the controller were judged. The test regimen for each vehicle is summarized in Table 10.

### 3.7.4.1 Constant Radius Test (CRT), SAE J266

The SAE standard test procedure J266 was written in order to establish a recommended practice for determining surface vehicle steady-state directional control behavior. As written in the 1996 test bulletin,

In the absence of elastic and kinematic steer effects – for example, at very low speeds – the Ackerman turn radius [see Figure 48] is defined geometrically by the wheelbase and by front wheel and rear wheel steer angles. At increasing speed, steady turning results in centrifugal force, which produces deflections in steering, suspension, and tire systems. These include lateral force deflection steer, aligning torque deflection steer, roll steer, tire slip angles from lateral force and camber force, etc. When expressed in degrees per g of lateral acceleration and lumped together, these “cornering compliances” produce steer angles and tire slip angles in front and rear which modify the Ackerman turn radius. Cornering compliances subtract from the front and rear Ackerman steer angles as shown in [Figure 48]. Cornering compliances greater in the front than in the rear increase path radius from the Ackerman condition and produce understeer; while cornering compliances greater in the rear than in the front reduce path radius, causing oversteer. The difference between the total front and rear cornering compliance is called understeer/oversteer gradient, expressed in degrees per g. Likewise, the change in steering wheel angle required to maintain a given radius with increasing lateral acceleration is called steering wheel angle gradient, the change in roll angle with lateral acceleration is called roll angle gradient, etc. [7].
From this diagram, the Ackerman Steer Angle (or simply, Ackerman angle) may be geometrically defined in Equation (52). For vehicles with front-wheel steering only, and assuming small steer angles, this relation reduces to Equation (53).

\[
\sin \delta_F - \cos \delta_F \tan \delta_R = \frac{L}{R_A} \quad (52)
\]

\[
\delta_f = \frac{L}{R_A} \quad (53)
\]

The test procedures described in SAE J266 are designed to measure these factors, as well as others such as yaw rate gain (degrees per second per degree of steer angle) and sideslip gain (degrees per degree of steer angle). The standard outlines several common steady-state tests suitable for determining steady-state directional control behavior, including

- Constant radius test with slowly increasing speed and steer (CRT)
- Slowly increasing steer with constant speed
- Slowly increasing speed with constant steer
From these, the CRT was chosen for ATD automation because it does not require a large test area. The CRT can also be conducted by increasing the speed at discrete intervals, but this takes much longer and tends to heat up the tires (affecting test results).

When performing the CRT with a human test driver, it is advantageous to use a relatively large path radius, at least 100 ft and often 200 ft. The larger radius lessens the errors in calculated results due to path wandering and fluctuation in vehicle speed. If the ATD can improve path and speed performance, accurate and repeatable results can be achieved using smaller circles than required for a human driver. Tests for this project were done with a 30-m (98.4 ft) radius circle as shown in Figure 49. Each vehicle performed three tests in both clockwise and counterclockwise directions.

SAE J266 specifies a maximum acceleration of 1.5 kph/s when conducting the CRT with a slowly increasing speed. The speed profile used for this test brought the vehicle to an initial speed of 2 m/s to determine the Ackerman steer angle (at near-zero lateral acceleration), then accelerated the vehicle at less than one-third the maximum rate, or 0.43 kph/s. The top speed (12.25 m/s) was designed to achieve 5 m/s² lateral acceleration on the 30 m radius circle before bringing the vehicle to a stop.

Steering wheel angle gradient, as described above, is a measure of a vehicle’s understeer gradient. (“Oversteer” will be dropped from the term “understeer/oversteer gradient”
since the steering characteristic of virtually every production vehicle falls within the understeer range.) Understeer gradient is determined on the CRT by plotting roadwheel steer angle, in degrees, versus lateral acceleration in g’s. Roll gradient (deg/g), another metric which is useful for characterizing the vehicle’s propensity to roll, is determined by measuring the slope of the roll angle when plotted against lateral acceleration.

3.7.4.2 Figure Eight

![Figure Eight Path and Speed Profile](image)

Figure 50: Figure Eight Path and Speed Profile

Two 25m radius ends with centers 80m apart are joined by straight sections to form the figure eight (Figure 50). This path alternates left- and right-hand turns and shows the controller’s ability to transition between curved and straight sections. It is also particularly suitable for continuous looping. Each vehicle was tested for six (continuous) loops on the figure eight at three speeds: 5, 8, and 10 m/s (11.2, 17.9, and 22.4 mph).

3.7.4.3 Random Path

The random path shown in Figure 51 exercises the controller’s ability to follow an irregular trajectory and repeatedly cross over its own track. It also features a varied speed profile reaching up to 15 m/s (33.6 mph), matching a suitable speed to each path portion. To accommodate the speed profile, this path was not tested with repeated loops; though
with a different speed profile, it could easily be followed in loop mode. Each vehicle conducted three trials on the random path.

![Random Path and Speed Profile](image1.png)

**Figure 51: Random Path and Speed Profile**

### 3.7.4.4 High-speed Oval

The 3.37-km oval path in Figure 52 was used to test the higher speed performance of the controller. The path was traced by each vehicle for 1-3 loops at 14, 20, and 25 m/s (31.3, 44.7, 55 mph respectively). The Camry also followed the oval at 30 m/s (67.1 mph).

![Oval Path and Speed Profile](image2.png)

**Figure 52: Oval Path and Speed Profile**
3.7.4.5 Open-loop Tests

Once a fully-autonomous path following controller has been developed, it is a simple matter to program portions of the controller to be used in an open-loop manner. Standard steering routines such as the J-turn, NHTSA Fishhook, and sine sweep can be implemented using user- or speed-based triggers, or triggered by any other measured vehicle parameter. Open-loop throttle and brake actions can similarly be programmed.

Thus far this capability has been exercised to a lesser extent on the ATD. J-turn and sine sweeps have been used for steering parameter identification, and throttle step response has been measured to examine time constants for various throttle inputs.

The 2005 Chevrolet Equinox donated to Ohio State University as part of General Motors’ ChallengeX competition was paced through an entire day of testing with the ATD. In addition to a Constant Radius Test and step-steer commands to evaluate the steering control, brake tests were conducted to evaluate the student-built stability control system. The ATD accelerated the Equinox to a given speed, then applied the brake as a percentage of full displacement (e.g., 50% of total travel) on a split-μ surface.

To date, one limitation is that the ATD servo control loop tuning is less than optimal. It performs transparently for low-frequency demands such as are normally seen for path-following maneuvers, but loses accuracy on high-slope ramps or sinusoids above 4 Hz.

3.7.5 Automated Data Analysis

One of the principal benefits of the ATD is the capability of rapid, automated data analysis. Analysis is performed in Matlab on the system laptop. At the conclusion of each run, the processing script starts by downloading the data file from the low-level controller (CompactRIO). The user is prompted for a name with which to save this run, after which plots for path-following performance (path vs. actual), path-following error vs. time and vs. distance, and actual and desired speeds appear. Also available are plots
of numerous other performance metrics such as lateral acceleration and angular rates and angles (e.g., roll angle).

For the Constant Radius Test, the analysis benefits are especially clear. The primary purpose for this SAE standard test is the determination of vehicle understeer gradient. As previously noted, the test may be conducted either with slowly increasing speed (≤ 1.5 kph/s acceleration) or at discrete speeds, picked to create increments of ≤ 0.05 g.

With the latter method, a test driver commonly maintains each speed for 30-45 seconds to ensure that the speed and path variations will be averaged into reasonably accurate data points. Ignoring the time to change speeds and assuming a test from 0.05-0.5 g, this means each test will last at least 5 – 7.5 minutes. With data collection at 100 or 200 Hz, this translates to unwieldy file sizes. The bulk test data is tediously broken into segments matching the desired discrete speeds, and an average steer angle is found for each segment. Plotted together, it looks like Figure 53, data for a manually-driven sample CRT done in the CR-V.

![Figure 53: CR-V Understeer Gradient Results for Manually-driven, Discrete-speed CRT (CW)](image-url)
The continuous-acceleration test data is easier to process; one simply fits a line to the plot of roadwheel steer angle versus lateral acceleration, and the slope of the line is the understeer gradient. (Strictly speaking, SAE J266 specifies that “any portion of the record in which speed change was greater than 1.5 kph/s, if used, should be so labeled” [7].) But since the shorter continuous test provides less data to average, path deviation error becomes magnified in the understeer calculation.

Compare the plots in Figure 54 for manually-driven (left) and ATD-driven (right) circle tests. Though in this case the ultimate least-squares understeer gradient results differ only by 0.06 deg/g, the ATD fit is clearly better than the manually-driven one. And in this case, the ATD test was done with a different set of tires than the manual trials, so one would not expect the two results to compare well.

Another way in which the ATD computation is more accurate has to do with the plot’s independent axis. Steer angle is typically plotted against computed rather than measured lateral acceleration for these graphs (and ultimately for the gradient calculations), since the computed value is much cleaner. But computed lateral acceleration ($v^2/R$) is only valid as long as the vehicle accurately tracks the circle radius. Compare the difference in

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**Figure 54:** CR-V Understeer Gradient from Continuous Acceleration CRT; Manually-driven (left), ATD-driven (right)
Figure 55 between computed and measured lateral acceleration for manually-driven (left) and ATD-driven (right) circle tests.
CHAPTER 4

RESULTS

4.1 Introduction

The ATD has been tested thus far on six vehicles: 2000 Toyota Camry, 1997 Honda CR-V, and 1992 Ford F-150 XLT (all mentioned in Chapter 3), plus a 2000 Jeep Grand Cherokee, 1991 Toyota Camry, and 2005 Chevrolet Equinox. The controller worked acceptably in all vehicles, though not equally well in each. Among the vehicles given the full battery of tests, the 2000 Camry demonstrated the best path following and the 1992 F-150 generally the worst. Speed control on the Camry, CR-V, and F-150 was roughly equivalent. Nonetheless, it can easily be argued in all test cases that the ATD performed more accurately and much more repeatably than might a human test driver. And the automated data processing, as evidenced here with the Constant Radius Test, can be dramatically faster than that performed for a manually-driven maneuver.

This chapter is organized as follows: The balance of the first section gives some guidelines to the interpretation of the test results. The two subsequent sections provide figures for the installation, initialization, and tuning of the ATD. Section 4.4 then displays the comprehensive results for two sample tests. Section 4.5 contains a comparison of the path-following performance of the three vehicles on each of the four main tests, while Section 4.6 compares their speed control performance. All results in sections 4.4, 4.5, and 4.6 were generated with the ATD controlling all three driver inputs, though a human driver was in the vehicle to initiate each test.
Section 4.8 contains sample results from several open-loop maneuvers such as a handwheel sine-sweep and throttle step response. For some of these tests the ATD controlled only the handwheel or brake and throttle, and for others it controlled all driver inputs. The chapter concludes with an analysis of the results.

4.1.1 Notes to Interpretation of Graphical Results

The standard SAE vehicle coordinate system is indicated with all the results. For instance, when looking at a plot of lateral acceleration, a positive result will indicate acceleration along the vehicle’s positive y-axis, or to the right. Table 11 lists the interpretation of values which are not otherwise obvious.

<table>
<thead>
<tr>
<th>Plotted Value</th>
<th>+ (Positive) Interpretation</th>
<th>- (Negative) Interpretation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Path-following error</td>
<td>vehicle is to left of desired path</td>
<td>vehicle is to right of desired path</td>
</tr>
<tr>
<td>Handwheel angle</td>
<td>degrees clockwise from zero</td>
<td>degrees counterclockwise from zero</td>
</tr>
</tbody>
</table>

Table 11: Interpretation of Various Plotted Values
Recall also that the data was sampled at 100 Hz. Therefore where dots are visible on the plot curve, such as the path-following snapshots, these points are spaced 0.01 sec in time.

### 4.1.2 Concerning Path-Following Error

The reader will notice the regular occurrence of a large path-following error during the first few seconds of many trials. This is due to a combination of two factors. The first is the inability of the human driver to put the vehicle in the precise position and orientation intended for the start of that path, whether that start point is an arbitrary point or a painted marker on the asphalt. The second factor is that the controller has been programmed to reduce its steering commands while the vehicle velocity is below 2 m/s, so the vehicle is slightly delayed in correcting its course. The latter feature results from the observation that no position or heading correction is possible when the vehicle is not moving; thus the controller is prevented from demanding sharp steering changes at low speed which may harm the power steering pump.

### 4.2 Portability and Installation

The hardware installed easily in all six vehicles, though simple wooden spacers were necessary to level the brake and throttle actuator in the CR-V and Cherokee due to their uneven floorboards. In the two Camrys, the electronics box was located in the back seat, and three occupants could comfortably observe the test. In the truck and SUV-type vehicles, cabling allowed the box to be secured in the cargo area, keeping at least four passenger seats available.

The RT3000 GPS/INS unit was located along the longitudinal axis (centerline) of all vehicles, either in the front or the back seat area. The sensor data output was displaced longitudinally to the vehicle center of gravity and vertically to the approximate roll axis. The latter displacement was intended to minimize the appearance of lateral path following error due only to vehicle roll.
Installation time was only measured for the last vehicle tested. The 2000 Camry (Figure 57) had never been outfitted with the ATD before the TRC test date, and thus made a good candidate for evaluating installation time. The researchers had by this time developed proficiency installing and uninstalling the system dozens of times in the CR-V and F-150, plus two times in the Cherokee. For the Camry, then, the total installation time was about one hour: twenty minutes for one person to disassemble the airbag and steering column, followed by forty minutes for two people to install the ATD hardware.

Figure 57: 2000 Camry Installation

In the left figure, the electronics box and battery box are contained within the blue plastic crate on the rear seat. This crate is not entirely necessary, but does help secure the equipment. It is often easier to strap down one crate or wedge it between seats rather than individually securing each component.

4.3 Initialization and Tuning

Assuming the laptop is powered with LabVIEW and Matlab running, the initialization and tuning process takes about fifteen minutes. This includes configuration of the RT3000, establishing the throttle static process characteristic, and input of vehicle
parameters in the ATD program. If the vehicle understeer gradient is not known, an additional fifteen minutes should be allowed for this test before the system is ready for full vehicle control on an arbitrary path. Thus for a vehicle which has not yet been tested, for which minimal parameters are known, a total of 1.5 hours should be allotted for installation and tuning before the vehicle is ready for path-following maneuvers with the ATD. By way of comparison, it often takes at least this long to instrument a vehicle with the sensors and data acquisition system which are used for manually-driven tests.

In addition to the tuning process already discussed, minor modifications to two steering variables were made. Table 12 shows the parameter adjustments for each test run in the format $L_{A \text{supp} 1} / L_{A \text{supp} 2}$ (e.g., “2 / 0.2”). $L_{A \text{supp} 1}$ is a constant supplemental lookahead (in meters) added to the speed-based lookahead value calculated by second-order curve fit. $L_{A \text{supp} 2}$ is another addition to lookahead, but this smaller factor augments only the lookahead value used to compute lateral offset $d$ in the steering control law, $\delta = K_p d$, derived on page 80. $L_{A \text{supp} 2}$ is not added into the lookahead value used to calculate $K_p$.

<table>
<thead>
<tr>
<th></th>
<th>CRT</th>
<th>Figure Eight</th>
<th>Random</th>
<th>Oval</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>CW</td>
<td>CCW</td>
<td>5 m/s</td>
<td>8 m/s</td>
</tr>
<tr>
<td>Camry</td>
<td>2 / 0.2</td>
<td>2 / 0.2</td>
<td>2 / 0.2</td>
<td>2 / 0.2</td>
</tr>
<tr>
<td>CR-V</td>
<td>0 / 0.05</td>
<td>0 / 0.05</td>
<td>2 / 0.2</td>
<td>2 / 0.2</td>
</tr>
<tr>
<td>F-150</td>
<td>1 / 0.4</td>
<td>1 / 0.4</td>
<td>2 / 0.4</td>
<td>2 / 0.4</td>
</tr>
</tbody>
</table>

Table 12: Steering Controller Tuning Parameter Adjustment

Additional lookahead ($L_{A \text{supp} 1}$) was necessary to ensure controller stability during the tests. The second factor ($L_{A \text{supp} 2}$) was added in an attempt to reduce tracking error.

A few observations regarding this table may be instructive. First, $L_{A \text{supp} 1}$ was generally smaller for the Constant Radius Test (CRT), which is a quasi-steady-state situation, than for the other, more dynamic maneuvers. (The exception is the Camry, which employed the same parameter adjustments for all tests; smaller values would likely have worked for
the Camry in the CRT, but no others were tried.) In steady-state tests, the perturbations exciting instability are smaller and less frequent than those in the more dynamic tests, thus not requiring lookahead distances as large.

A second observation is that larger lookahead increases were necessary at higher speeds. This follows the trend for lookahead itself, which must naturally be larger at higher speeds. This pattern suggests that the lookahead rule itself is not doing its job.

The speed-based lookahead rule was determined using simulation (page 77), and never explicitly evaluated with experiment. It is likely that the system model did not capture all computational delays and nonlinearities of the real system, thus leading to a lookahead rule too aggressive for reality. In order to eliminate the need for this secondary tuning step, one should either improve the model and determine a new lookahead rule or develop a new rule experimentally. Another option is changing to a multiple preview point method such as that suggested by Sharp et. al. [64], reducing the dependence on the location of a single preview point (goal point).

4.4 Sample Test Results

Though the full test results for the complete battery of tests is included in the appendix, full results for two sample tests will be given in this section as an introduction to the capabilities of the ATD.

4.4.1 SAE J266 Constant Radius Test (Camry)

Because the understeer gradient of the 2000 Camry was not known, this test, as described in Chapter 3, was conducted first. The controller was given the estimated value of 3.5 deg/g to begin, in addition to the other known vehicle parameters (wheelbase and CG-front axle distance). Because steering ratio was not present on the Expert Autostats® printout, a value of 15.0 was visually estimated. Based on experience with other
vehicles, the curve-fit lookahead rule was augmented by 2.0 m to assure stable performance. Results for one clockwise CRT are shown in Figure 58 through Figure 62.

Figure 58: Camry: CRT Path-Following (left) and Speed Control (right)

Figure 59: Camry: Path-Following Error (left) and Handwheel Angle (right)
Figure 60: Camry: Lateral Acceleration (left) and Path-Following Snapshot (right)

In Figure 60 (left), measured lateral acceleration is compared to that calculated by $\frac{v^2}{R}$. The measured data has been phaseless-filtered with an 8th order Butterworth filter with an 8 Hz cutoff frequency.

Figure 61: Camry: Roll Gradient (left) and Understeer Gradient (right) for 30m Radius CRT
The plots in Figure 61 show two important results from the Constant Radius Test. Here, computations from both clockwise and counterclockwise tests are shown on each graph. At left is the calculated roll gradient, and at right is the calculated understeer gradient, both in degrees/g.

For the three trials, understeer gradients of 2.53, 2.51, and 2.54 deg/g were obtained. Understeer gradients for the counterclockwise test (shown in the appendix) were not quite as closely-grouped: 2.75, 2.81, 2.65 deg/g. The lowest of these was nonetheless less than 6% different than the highest.
The Ackerman angles calculated in the clockwise test were 5.33°, 5.33°, and 5.32° respectively, 4.7% larger than the theoretical value of 5.09° computed from wheelbase and path radius. This small difference between experimental and theoretical Ackerman angles, however, is typical for all types of tests. The counterclockwise test produced Ackerman angles of -5.34°, -5.31°, and -5.36°, showing both process repeatability and that the handwheel encoder had been accurately zeroed. Full results for all tests are displayed in Table 13, with further analysis in sections 4.5.1.3 and 4.5.1.4.

As reflected in Figure 62, the maximum deviation from path for all three trials was 11 cm (4.33 in), with a maximum of 8 cm (3.15 in) for trials 2 and 3. (As mentioned in this chapter’s introduction, the excursion at the very beginning of any run is ignored.) Path-following was held this tight despite lateral acceleration up to 0.52 g.

Compare now results from the ATD-driven CRT with those of a manually-driven test. One should not put too much value in this comparison, for the manual test was driven by a graduate student and not a professional test driver. On the other hand, this highlights another advantage of the ATD: no special physical skills are required for accurate test driving.

Two plots below show path-following error for manually-driven Constant Radius Tests. The test in the left plot was conducted using continuous acceleration, whereas the test in the right plot was done with the discrete speed option. The left plot shows the manually-driven result superimposed on the ATD result, both for the same vehicle in the same direction. Clearly the ATD-driven result is much better in terms of both accuracy and consistency.
4.4.2 Winding Path (Camry, 14 m/s)

A complete sample result is also provided for the Camry as it traversed the winding path laid out on TRC’s Vehicle Dynamics Area (Figure 64). Results are shown for the first loop, run at 14 m/s (31 mph). Path-following error on this very dynamic test averaged approximately 28 cm with one excursion to 50 cm when path curvature abruptly changed from positive to negative. Lateral acceleration ranged from +0.70 g to -0.39 g, as shown in Figure 65.

Figure 63: CR-V: Manually-driven Constant Radius Tests

Figure 64: Camry: Winding Path (14 m/s)
4.5 Path-Following Performance

Having shown two sample comprehensive test results for the ATD, path-following results will be compared for all three test vehicles on the constant radius test, figure eight, random path, and high-speed oval.
4.5.1 Constant Radius Test (CRT)

Results below are all for a 30 m (98.4 ft) radius circle test conducted with slowly-increasing speed. The Camry tests were conducted at TRC, whereas the CR-V and F-150 were tested on an uneven, gravel-strewn parking lot. Again, full test results showing many more performance parameters on all CRT trials are in the appendix.

Figure 66: Constant Radius Test (CW) for Camry (top left), CR-V (top right), and F-150 (bottom)
4.5.1.1 Clockwise CRT

While the Camry and CR-V remained quite close to the desired clockwise curve, the F-150 strayed increasingly from the circle as lateral acceleration went up (Figure 66). At the end (near 0.5 g), the truck reached a peak of 50 cm from the desired path. A similar trend is evident for all vehicles in the counterclockwise test (Figure 67), though at only half the magnitude. The poorer performance of the F-150 may be in part attributed to its larger steering deadband, which is uncompensated by the controller. See Section 4.9 for further discussion on this deadband.

The relatively poor parking lot test surface also affected the higher-g performance of both the CR-V and F-150. Both of these slid substantially on gravel patches toward the end of their runs.

4.5.1.2 Counterclockwise CRT

It is interesting that the Camry and CR-V held much closer to the path on the clockwise CRT than on the counterclockwise CRT shown in Figure 67. Not that the CCW-run 22 cm path deviation is unacceptable when tracking a 0.5 g curve, but it is twice as high as the error in the opposite direction. The F-150, on the other hand, shows the opposite trend; it’s CCW error is about half that of the CW run.

This could be accounted for by a few factors. First, any vehicle’s steering action is not necessarily identical in both directions. Anyone who has collected data to compute steering ratio can attest to this; though steering ratio ultimately is represented by one number, experimental curves of handwheel vs. roadwheel angles are neither linear nor symmetric about the y-axis (zero steer angle). The linear fits are reasonably valid, of course, but not perfect.
Secondly, the zero steer position recorded by the operator at the beginning of the test could be offset to one side or the other. Typically this is set while the driver attempts to track a straight line at low speed. It is difficult to determine the exact zero position when there is any left-right slope on the test surface, any wind buffeting the vehicle, or any backlash in the steering mechanism. Because the ATD controls only the handwheel angle, without any direct feedback of the roadwheel angle, steering error will result from any zero-angle discrepancy between the controller and reality.

Finally, even if the steering were perfect, vehicles may exhibit different lateral dynamics in left- and right-hand turns. This can be due to uneven static weight distribution or
unequal load shifting caused by spring and suspension component variance. Other suspension compliance effects can also come into play.

4.5.1.3 Understeer Gradient Determination

Table 13 displays the understeering gradients and Ackerman steering angles calculated for all CRT trials. Examining the difference in CW / CCW Ackerman angles, a small handwheel zeroing error is apparent for the CR-V and F-150. This will affect path-following accuracy, but since the understeer gradient (“k”) computation is the slope of the steer angle versus lateral acceleration line, it does not depend on the exact steer angle, and is thus insensitive to zeroing errors.

<table>
<thead>
<tr>
<th></th>
<th>CW</th>
<th>CCW</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>k (deg/g)</td>
<td>Ackerman angle (deg)</td>
</tr>
<tr>
<td>Camry</td>
<td>Trial 1</td>
<td>2.53</td>
</tr>
<tr>
<td></td>
<td>Trial 2</td>
<td>2.51</td>
</tr>
<tr>
<td></td>
<td>Trial 3</td>
<td>2.54</td>
</tr>
<tr>
<td>CR-V</td>
<td>Trial 1</td>
<td>3.56</td>
</tr>
<tr>
<td></td>
<td>Trial 2</td>
<td>3.49</td>
</tr>
<tr>
<td></td>
<td>Trial 3</td>
<td>3.58</td>
</tr>
<tr>
<td>F-150</td>
<td>Trial 1</td>
<td>4.87</td>
</tr>
<tr>
<td></td>
<td>Trial 2</td>
<td>4.84</td>
</tr>
<tr>
<td></td>
<td>Trial 3</td>
<td>4.75</td>
</tr>
</tbody>
</table>

Table 13: Understeer Gradient Test Results
Table 14: Calculated Understeer Gradient and Ackerman Angle Variation

<table>
<thead>
<tr>
<th></th>
<th>CW k (deg/g) variation</th>
<th>CCW k (deg/g) variation</th>
<th>CW Ack. angle (deg) variation</th>
<th>CCW Ack. angle (deg) variation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Camry</td>
<td>0.03</td>
<td>0.16</td>
<td>0.01</td>
<td>0.05</td>
</tr>
<tr>
<td>CR-V</td>
<td>0.09</td>
<td>0.11</td>
<td>0.03</td>
<td>0.02</td>
</tr>
<tr>
<td>F-150</td>
<td>0.13</td>
<td>0.04</td>
<td>0.01</td>
<td>0.06</td>
</tr>
</tbody>
</table>

4.5.1.4 Roll Gradient Determination

Another calculation automated by the ATD, in conjunction with performance of a constant radius test, is the determination of roll gradient. Here the CRT must be conducted on a regular surface so that the roll angle may be corrected for any surface...
slope. For this set of tests, only the Camry performed its CRT on a regular surface (at TRC). To understand the need for a regular surface, compare the unfiltered, unadjusted roll angle measurements for Constant Radius Tests in the Camry at TRC (Figure 68, top) and in the other vehicles on a parking lot (middle and bottom). It would be difficult to correct the latter measurements without recording one complete circle at very low speed for use as a “bias” adjustment.

The left and right Camry roll gradients were determined independently with clockwise and counterclockwise tests. Roll gradient plots for the first CW/CCW trials appear at left in Figure 69, for the second trials in the middle, and for the third trials at right; tabulated results are shown in Table 15. Variation of calculated roll gradient was 0.06 deg/g (1.1%) for the three clockwise trials and 0.03 deg/g (0.6%) for the counterclockwise trials.

<table>
<thead>
<tr>
<th>Camry</th>
<th>CW Roll Gradient (deg/g)</th>
<th>CCW Roll Gradient (deg/g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Trial 1</td>
<td>5.39</td>
<td>5.36</td>
</tr>
<tr>
<td>Trial 2</td>
<td>5.33</td>
<td>5.37</td>
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<tr>
<td>Trial 3</td>
<td>5.36</td>
<td>5.34</td>
</tr>
<tr>
<td>Variation</td>
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<td>0.03</td>
</tr>
</tbody>
</table>

Table 15: Tabulated Roll Gradient and Variation
Figure 69: Camry Roll Gradient, Trials 1 (top left), 2 (top right), and 3 (bottom)

4.5.2 Figure Eight

Results below are for a figure eight path as shown in Figure 70. As with the CRT, the Camry tests were conducted at TRC while the CR-V and F-150 were tested on a parking lot. While all path-following plots are shown in this section, the appendix contains plots of lateral acceleration, handwheel angle, and other data for these tests.
4.5.2.1 Figure Eight at 5 m/s

Figure 70: Figure Eight Path

Figure 71: Figure Eight Path-Following at 5 m/s for Camry (top left), CR-V (top right), and F-150 (bottom)
4.5.2.2 Figure Eight at 8 m/s

Figure 72: Figure Eight Path-Following at 8 m/s for Camry (top left), CR-V (top right), and F-150 (bottom)
4.5.2.3 Figure Eight at 10 m/s

At all speeds the F-150 showed a tendency to track outside the curve (figures above); its error was negative while moving counterclockwise and positive while moving clockwise. The CR-V had a similar tendency, though smaller in magnitude. The CR-V’s error also barely dipped below zero, being centered on a small positive value.

Lateral acceleration plots, found in the appendix, show that the vehicles required nominally 1, 2.6, and 4 m/s\(^2\) respectively to navigate the figure eight at 5, 8, and 10 m/s.
This test gives evidence to the excellent repeatability of the ATD. Look for instance at the results for the Camry at 10 m/s in Figure 73 and Figure 74, demanding an uncomfortable 0.4 g. Though path error reaches 20 cm during this run, the tracks are very close together. In the plot of handwheel angle, the traces from six laps are indistinguishable except at a couple of points.

Figure 74: Camry Figure Eight at 10 m/s; Handwheel Angle (left) and Path-Following Snapshot (right)
4.5.2.4 Figure Eight Comparison with Human Driver

Figure 75: Camry Figure Eight with Human Driver (left) vs. ATD (right)
Figure 75 and Figure 76 show comparisons between figure eight paths driven by a human driver and the ATD. The author, not a professional test driver, performed driving duties.
for these trials. Manually-driven results would likely have been better with a professional behind the wheel. Even I got better as time went on; witness the improving trend in repeatability from the slowest trial in the Camry (first) to the fastest trial in the F-150 (last).

Manually following a painted line on the asphalt is challenging. I tried to keep the line visually in the center of the hood, which may not have been the driver’s strategy when the path was recorded. In any case, it may be more prudent to compare these graphs on repeatability rather than strictly on accuracy.

The Camry speedometer did not move until velocity exceeded 10 mph (4.5 m/s), and its smaller gage size made it more difficult to evaluate speed with a quick glance. The Camry’s quieter engine, too, makes the driver more dependent on visual feedback than in the F-150. Spending more time with eyes on gages, even at 5 m/s, translates to greater deviations from the path.

Note that the spikes in path-following error at the 175 m and 350 m points are due to a shortcoming in the data processing script. At these points, the script may determine that the vehicle is closer to a point on the crossing path than the path segment it is following. This behavior is more pronounced when the vehicle is further from its intended path.

These caveats aside, the ATD shows a more repeatable performance in all trials. Even in the best manually-driven case, the F-150 at 5 m/s, vehicle position varies by as much as 0.5 m from lap to lap. In the worst ATD-driven case, the F-150 at 10 m/s, vehicle position fluctuates less than half that much.

4.5.3 Random Path

Vehicles on the random path shown below started with an eastward heading. A time-varying speed profile was followed, beginning with 10 m/s, then accelerating to 15 m/s, and slowing to 6 m/s at the end (see pg. 143). Since each test consisted of one loop, all three vehicles may be compared on one plot. Again the Camry tests were conducted at
TRC while the CR-V and F-150 were tested on a parking lot. Additional detail is available in the appendix.

Figure 77: Random Path Vehicle Comparison

4.5.4 Oval

All tests on the 3.37 km oval plotted below were conducted at TRC. Vehicles started in a southeasterly orientation. Tests on CR-V were cut short by nightfall, and comparison for all vehicles is only available at 20 m/s (45 mph). The F-150 and Camry were additionally tested at 14 and 25 m/s, and the Camry was tested at 30 m/s (67 mph).

The abrupt change around t = 65 sec was created as the vehicles climbed out of the steeply-banked corner in turn two (a corner designed to guide vehicles into a figure eight trajectory on the VDA).
The Camry was tested up to 30 m/s (67 mph) on the oval, with lateral acceleration averaging 0.47 g on the northern turn.

Figure 79: Camry on Oval Path (30 m/s); Path Error (left), Lateral Acceleration (right)
4.6 Speed Control Performance

All tests in the previous section were conducted using the ATD for speed control as well as steering control. Comparative results for speed control performance are presented in this section. In addition, speed control plots for every vehicle/test combination can be found in the appendix.

4.6.1 Constant Radius Test (CRT) Speed Control

SAE J266 specifies a maximum acceleration of 1.5 kph/s when conducting the CRT with a slowly increasing speed. The speed profile used for this test brought the vehicle to an initial speed of 2 m/s to determine the Ackerman steer angle (at near zero lateral acceleration), then accelerated the vehicle at less than one-third the maximum rate, or 0.43 kph/s. The top speed (12.25 m/s) was designed to achieve 5 m/s² lateral acceleration on the 30 m radius circle before bringing the vehicle to a stop.
Smooth speed control at low speeds is rather difficult due to the combination of several factors. One factor is the low reflected vehicle inertia in first gear. (In higher gears, higher reflected vehicle inertia smooths out the torque variations in the powertrain.) Another is the fact that idle speed for these vehicles ranged from 2.1 – 2.6 m/s, requiring a transition from brake to throttle to maintain lower speeds. Still another factor is the deadband in the accelerator pedal. While much of this deadband can be removed by proper adjustment of the actuator linkage, it is still a very small pedal stroke separating idle speed from twice that. Combine that jump with the torque multiplication occurring in the torque converter, and one can see why automotive manufacturers typically set a lower limit around 13 m/s (30 mph) for factory cruise control. These factors explain the overshoot of 2 m/s and the sawtooth profile observed below 8 m/s in Figure 81.

![Figure 81: Speed Control for CRT – Vehicle Comparison](image)

**4.6.2 Figure Eight Speed Control**

A sixty-second window was compared for all vehicles on the figure eight path at 5 m/s and 10 m/s (Figure 82). All three vehicles maintained 5 m/s within roughly 0.2 m/s, or
4%. The error band was significantly smaller for the Camry and CR-V at 10 m/s, but at this speed the F-150 was markedly slowed at the beginning of each curve.

Figure 82: Speed Control for Figure Eight (5 m/s left, 10 m/s right) – Vehicle Comparison

4.6.3 Random Path Speed Control

All three test vehicles followed the desired random path speed profile fairly well (Figure 83). The Camry took rather long to resume the target speed after both deceleration commands, perhaps indicative that the feedforward mechanism was not well-tuned in this speed range.
4.6.4 Oval Speed Control

All three vehicles maintained a steady-state speed within 0.8% of the 20 m/s target for the oval path (Figure 84).

Figure 83: Speed Control for Random Path – Vehicle Comparison

Figure 84: Speed Control for Oval (20 m/s) – Vehicle Comparison
4.7 Cumulative Test Results

Root-mean-square path-following error was calculated for all trials so that path-following and speed control performance trends may be compared for each vehicle. The RMS results are summarized in Figure 85 through Figure 88.

For the Constant Radius Test, all ATD-driven tests beat the RMS error of the manually-driven tests, excepting the F-150 in the clockwise direction. Some explanation has already been offered for the F-150’s uneven showing; further analysis is provided in Section 4.9.

![Figure 85: Constant Radius Test: RMS Path-Following Error (left), RMS Speed Error (right)](image)

As previously shown, manually-driven trials were also conducted for the figure eight path. The RMS results for all trials are in Figure 86. Here the ATD bested the human driver at every speed in both the Camry and F-150.
For the random path and oval tests, no painted path was available to enable manually-driven comparison. RMS results for these tests are shown in Figure 87 and Figure 88.
4.8 Sample Open-Loop Maneuvers

A wide variety of open-loop driving inputs is possible on the ATD with a modicum of programming effort, including the NHTSA Fishhook maneuver, sine with dwell, and more. Several routines have been implemented for assisting with system identification, including the three examples to follow. The first is a sinusoidal sweep (chirp) steering input, the second a steering step response, and the third is a throttle step response.

4.8.1 Open-Loop Steering Maneuver

Figure 89 shows results from a sine sweep exercise performed on the 1997 CR-V. This test was performed on a very poor asphalt surface, resulting in “noisy” roll and lateral acceleration traces. Compare these with results from the next sine sweep test conducted on the regular surface at TRC (Figure 90).
The sine sweep maneuver in Figure 90 was performed on the 2005 Chevrolet Equinox, Ohio State’s entry in General Motors’ ChallengeX Competition. Test speed was 17 m/s. The ATD provided a regular, accurate $\pm 45^\circ$ sinusoidal steering input up to 4Hz, except that it tended to overshoot by approximately $1.5^\circ$ toward the end.
Below is a staggered step-steer command also conducted in the Equinox. Steer angle is stepped in two 90° increments at 270 deg/s, then back to zero. The handwheel followed the command rather precisely, with mild underdamped ripples visible at the end of each step (2.2° overshoot).
4.8.2 Open-Loop Throttle Control

Figure 92 shows a sample of another open-loop test possible with the ATD, a throttle step input. Tests of this type may be used to characterize a vehicle’s powertrain response to various throttle commands.
4.9 Analysis of Results

Beyond the comments already included in the subtopics above, a few general topics merit discussion. The first topic is an attempt to explain the disparity between the ATD’s path-following accuracy on the Camry and CR-V compared to the F-150. At least two factors come into play: the truck’s steering backlash and its longer wheelbase.

Steering backlash has already been discussed as contributing to difficulty in precisely zeroing the steering encoder. This translates into an uneven performance in left- and right-hand cornering. Larger (uncompensated) backlash also means that small steering corrections avail nothing; only when error becomes larger would steering correction be sufficient to affect roadwheel angle. Even then, the steering correction would not be sufficient to fully correct the vehicle trajectory, because the backlash-induced deadband had not been accounted for. A remedy for this would be some sort of software deadband compensation.
The second factor, longer wheelbase, means that a higher steering angle is required for same radius curve (i.e., for same yaw rate). This principle is easily illustrated by the definition of the Ackerman angle, where steer angle \( \delta = \frac{L}{R} \) \( (L = \text{wheelbase}, \ R = \text{curve radius}) \). Though the steering control law factors wheelbase into its calculation (Equation (39)), the longer wheelbase only amplifies the problems caused by steering backlash.

Another worthwhile topic is the propensity for the controller to allow a small, constant offset from a straight path. This same characteristic was seen on all vehicles. It is not the same as constant offsets during a curved path, which may be due to suboptimal lookahead distance. From a control theory perspective, the addition of integral gain seems an obvious choice. And this may be prudent, but it cannot be added for all cases. For instance, in the Constant Radius Test, a vehicle always has a constant heading error. The path direction at the goal point is always different from its current heading, because the path is a continuous circle. If an additional term is added to the control law based on the integral of the heading error, the vehicle’s steering angle will continuously increase, causing it to spiral toward the center of the loop.

What may be done, however, is to modify the control law to separately treat the current lateral offset from path. Instead of treating current lateral offset as part of the total heading error, as shown in Figure 42, as a separate entity it could be integrated.

Also worth addressing is another change to the steering control law. Described in Equation (26), steer angle \( \delta = K_p d \) where \( d = LA\sin \theta \) and \( \theta \) is projected heading error. The latter equation was implemented as \( d = LA\theta \), though there was no good reason to use this small angle approximation. On the oval path, when lookahead was large due to higher speeds, the small angle approximation for \( \sin \theta \) would have yielded a value larger than needed; this at least partly explains why vehicles tracked consistently inside the circle.
CHAPTER 5

CONCLUSIONS

5.1 Summary

The goal of this project was to develop a lightweight, compact, easily-mounted, and low-cost control system to automate many test driving functions. The Automated Test Driver achieved its weight goal, besting its 100 kg target for combined system mass by over 42 kg, including all actuators, sensors, and electronics. The portability and installation goals were also met; trained personnel have indeed installed the ATD in one hour without altering or damaging the vehicle, except that the stock steering wheel is be temporarily replaced by a motorized handwheel unit. In another 15-30 minutes, the ATD has been tuned to a new vehicle with manual data input of only four vehicle parameters, three of which are commonly available, and the fourth can be quickly assessed by the ATD.

Operability goals were also met or exceeded. A test operator is able to select from standard test routines or preprogrammed paths, or may program an arbitrary path and / or velocity profile. In addition, the ATD has automated the execution of the Constant Radius Test for determining surface vehicle steady-state directional control behavior, as defined by SAE J266. The ATD speeds both execution and data analysis for this very common test, demonstrating excellent repeatability as judged by three consecutive trials. Path-following capability with automated speed control was verified on four dynamic
paths with three vehicle types, with paths ranging from low to high cornering force and speeds up to 30 m/s (67 mph).

No other device found in technical or commercial literature has claimed the capabilities listed above. Existing devices are either too cumbersome to fit in small cars, or they are vehicle-specific designs which modify or bypass stock throttle, brake, or steering linkages. Many also have the control system designed around the performance of one particular vehicle.

The ATD offers the vehicle test industry experimental flexibility as well as substantial improvements in accuracy and repeatability over a human driver.

The stated goals were achieved, but room is left for improvement. Though quite repeatable, both path tracking and speed control could be more accurate. Based on the analysis comments included throughout Chapter 4, various modifications could enhance ATD performance. A list of suggestions is also provided under the future work heading below.

5.2 Suggestions for Future Work

Though no formal stability analysis is presented for the steering controller used in the ATD, a stability mechanism is suggested from others’ analysis of a similar steering controller. Unyelioglu et. al. analytically prove their closed-loop system stable using Routh-Hurwitz analysis, having determined that “given any range of longitudinal speeds, there exists a sufficiently large lookahead distance ensuring the closed-loop stability for all speeds in that speed range” [93]. An analogous proof for the ATD steering law would be worthwhile.

Other potential improvements or extensions of this work:

- Evaluate robustness to vehicle parameter variation.
• The ATD speed-based lookahead rule was determined using simulation. This was one factor not verified experimentally. During the ATD tuning, operators consistently had to modify controller to add lookahead; a better rule could prevent need for this modification. It is likely that the mathematical model used in simulation does not capture all delays and nonlinearities of the real system. The choice exists to either improve the model and determine a new rule or develop a new rule experimentally, or to change the control law altogether to a multiple preview point method such as suggested by Sharp et. al. [64].

• Complete the RF link enabling ATD testing without a human driver in the vehicle. This would reduce driver exposure to tests which were either dangerous or hard on the body, such as vehicle endurance courses.

• Software steering deadband detection / compensation, perhaps based on tuning scheme which measures handwheel motor amplifier current during small, slow steer angle changes.

• Modify the control algorithm to allow the vehicle to track more accurately at very low speeds: instead of limiting steer angle at low speeds (the current protective measure), limit steer rate.
APPENDIX

TEST RESULTS

6.1 Constant Radius Test Results

6.1.1 Camry

Figure 93: CRT (CW) – Camry
Figure 94: CRT (CW) – Camry
Figure 95: CRT (CW) – Camry, Path-Following Repeatability for Three Trials
Figure 96: CRT (CCW) – Camry
Figure 97: CRT (CCW) – Camry
Figure 98: CRT (CCW) – Camry, Path-Following Repeatability for Three Trials
6.1.2 CR-V

Figure 99: CRT (CW) – CR-V
Figure 100: CRT (CW) – CR-V
Figure 101: CRT (CW) – CR-V, Path-Following Repeatability for Three Trials
Figure 102: CRT (CCW) – CR-V
Figure 103: CRT (CCW) – CR-V
Figure 104: CRT (CCW) – CR-V, Path-Following Repeatability for Three Trials
Figure 105: CRT (CW) – F-150
Figure 106: CRT (CW) – F-150
Figure 107: CRT (CW) – F-150, Path-Following Repeatability for Three Trials
Figure 108: CRT (CCW) – F-150
Figure 109: CRT (CCW) – F-150
Figure 110: CRT (CCW) – F-150, Path-Following Repeatability for Three Trials
6.1.4 Vehicle Comparison: Path-Following

Figure 111: CRT (CW left, CCW right) – Camry, CR-V, F-150 (top to bottom)
6.1.5 Vehicle Comparison: Speed Control

Figure 113: CRT (CW) – Speed Control Comparison
6.2 Figure Eight Path Results

6.2.1 Camry

Figure 114: Figure Eight (5 m/s) – Camry
Figure 115: Figure Eight (8 m/s) – Camry
6.2.2 CR-V

Figure 117: Figure Eight (5 m/s) – CR-V
Figure 118: Figure Eight (8 m/s) – CR-V
Figure 119: Figure Eight (10 m/s) – CR-V
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Figure 156: Winding Path (8 m/s) – Path-Following Comparison

Figure 157: Winding Path (10 m/s) – Path-Following Comparison
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