Development of Automatic Vehicle Headway Control

Law and a Simulation Tool

by

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Vehicle following and its effects on traffic flow has been an active area of research. Human Driving involves reaction, delays, and human errors that have adverse effects on traffic flow. We can eliminate human errors by introducing a computer control system.

The purpose of this research was to develop and evaluate a control law and a simulation tool for the study of automatic vehicle headway control. This research considers longitudinal control of a platoon of vehicles on automated highways. A new way of designing control law for vehicle following is presented by introducing safe boundary concept — the
trail vehicle should never exceed the maximum safe velocity and at the same time keeps the passengers comfort when accelerating or decelerating except under emergency circumstances. After finding the safe boundary, we design the automatic control law and then using our simulation tool to simulate its performance, adjust parameters until we reach a satisfactory result.

System dynamics concept and basic individual vehicle motion laws are used through the research. System dynamics provides a common foundation that can be applied wherever we want to understand and influence how things change through time. We look at the platoon system as a whole and study all the objects, such as vehicle dynamics, road condition, motor dynamics, in this system interact with one another.

A third-order nonlinear, Car-following, PID control law is designed using System Dynamics concept. System dynamics’ simulation language DYNAMO and Spreadsheet are have been used for our development of a simulation tool. A Satisfactory result is found after the extensive simulation which indicates that the platoon assumptions are achievable using the advanced technologies, like automatic vehicle control, radar, and sensors.
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Chapter 1

INTRODUCTION

"The road to a stronger national economy is not found in clogged streets and highways, slowing traffic to a crawl or in water shortages, open garbage dumps and polluted streams. Instead national strength will come through the provision of those everyday facilities whereby industries, commerce, and agriculture can thrive and citizens can enjoy the benefits of our national way of life". (Grigg 1993)

1.1 Background

In the economic development of a nation, transportation plays a very important role. For example, in the United States it represents more than 6% of GNP with more than 10% of the average family budget devoted to transport. In 1993, there were a total of 210,861,980 vehicles in the United States, 93.9% of them was highway vehicles. 2,307,906,000,000 vehicle-miles occurred (domestic transportation). Out of them, 2,296,669,000,000 miles were traveled on the highways which is 99.5% of the total. Car ownership has increased steadily over the last decade. In the United States, we have 800 vehicles per 1000 people. As the
rate of car ownership increases, there is a resultant growth in traffic volume without corresponding increases in the transport infrastructure. This has led to increases in various degrees of traffic congestion and accidents. In the year 1994, the United States sustained approximately 55,000 fatalities and 94.1% of them was motor vehicle fatalities. It is believed that the above trends in the United States is not too dissimilar from other developed countries. Presently, there is no immediate trend to indicate that this will slow down, unless various technical and political initiatives are taken.

As a response to these challenges, many national and international research and development programs have been adopted in various parts of the world. The program in the United States was formerly known as IVHS and now is named ITS. A major part of this program is called AHS(Automated Highway System) which is, in general, to remove as much human involvement as possible from the system through computer control and automation.

1.2 Research Objective

The objective of this research is:

- First, to develop a control law using the basic individual vehicle motion laws to achieve a safe and smooth vehicle following objective. This control law should have the ability to be implemented into the real world under normal conditions. The definition of normal generally means benign
environmental conditions and faultless operation of all the hardware, both on the vehicle and on the roadside.

- Second, to develop a simulation tool which taken into account all the influencing factors on a vehicle traveling on a AHS lane, and to test the control law developed in the first step. We may need to change some parameters until we reach a satisfactory result. This simulation tool should also be suitable for other control laws developed by other researchers in order to test and evaluate their results.

1.3 Motivation for the Research

An automatic vehicle control system is very sophisticated and has many factors affecting it. It is impossible to implement such a system into the real world without knowing how it will work. This requires extensive computer simulations that capture the coupling of the control system logic with the movements of vehicles on a prototype test facility. Therefore, a computer simulation model capable of testing and evaluating different headway control models under different operating conditions is a necessary tool for AHS development.

To study a sophisticated system like the automatic vehicle control system, we can either break it up into smaller and smaller pieces or look at this system as a whole. For the study of automatic vehicle control system, we want to understand and influence how, for example, acceleration and velocity, change through time. We want to unify our knowledge on vehicles dynamics, roadway conditions, central processor characteristics,
weather conditions, etc. System dynamics is capable of performing these functions. System dynamics is effective because it builds on the reliable part of our understanding of systems (vehic le dynamics, roadway conditions, central processor characteristics, weather conditions, etc) while compensating for the unreliable part. The modeling process separates consideration of underlying assumptions from the implied behavior, there is less inclination for people to differ on assumptions.

1.4 Scope of the Research

The formulation of a mathematical control law and development of a computer simulation tool are the emphasis of this research. This is seen as the first step in the design and implementation of a real world controller. Basic individual vehicle motion laws are thoroughly studied in this research. Various vehicle headway control theories and strategies are also discussed, and some of them are incorporated in simulation tool development. By repeated experimentation in this laboratory, assumptions are tested.

1.5 Contribution to the Body of Knowledge

This research represents a basic, step-by-step way of designing a control law for automatic vehicle headway control. This is accomplished by using the fundamental individual vehicle motion laws to find the safe
boundary. Based on comparisons of performance, an improved control law is found. The advantage of this control law is that it has the capability of making a near real-world simulation by incorporating the internal and external factors affecting vehicles operating on AHS lane.

This research also develops a simulation tool using the System Dynamics concept, the interaction between variables forming feedback loops. This gives a new way of implementing System Dynamics as well as a new way for AHS researchers to test and evaluate their results before applying them to the real world.
Chapter 2

LITERATURE REVIEW

2.1 AHS History

The concept of Advanced Highway System is not a new one. In the 1970's research was carried out to test the viability of automated vehicles by the automobile industry. Renewed interest in the AHS was kicked-off with the idea of Intelligent Vehicles (IVs) in the early part of this decade. Optimism over success of a full fledged AHS derives from the vast advances that were made in microchip technology. Needless to say the last decade has witnessed phenomenal improvement in the power of microchips and at the same time the market has witnessed a sharp drop in the cost of producing them. AHS promises to deliver a substantial improvement in capacities ranging from 100% to 1000% depending upon several factors. If implemented these will take care of capacity requirements for the better part of the next century. Apart from the tremendous improvement in capacities, a full fledged AHS promises to reduce accident rates, alleviate the congestion problem, aid elderly drivers (with the greying of North America, elderly drivers will form a substantial percentage of total drivers by the beginning of next century).
What exactly is AHS? The present vision of functional AHS is platoons of passenger cars electronically linked moving at high-speeds equipped with highly responsive sensors to steer and control the platoon. This high efficiency in controlling the platoon is achieved by an extremely high-level of coordination between a pilot car and other cars in the platoon and computing precise location coordinates of the platoons with the help of road-side beacons and feeding all these to a main computer. Substantial gains in capacities are expected through minimizing the spacing between the successive cars to as low as 1 m while at the same time cruising at high speeds, and fast reacting sensors for detecting and avoiding emergencies. Capacity gains can also be achieved by lateral guidance system which controls the lateral movement of the platoons thereby reducing the lateral distance between platoons, thus increasing the number of lanes available for platoons. AHS, itself, is not a single entity but consists of scores of sub-systems acting in coordination to make such a system to function smoothly. The chief components that constitute a AHS are Automated Longitudinal Guidance System (ALOGS), Automated Lateral Guidance System (ALAGS), and Automated Steering Control System (ASC). Basic research on perfecting these systems is still going on.

2.2 AHS Goal

The mandate for an automated highway comes from the December 1991 Intermodal Surface Transportation Efficiency Act (ISTEA), Part B, Section 6054(b), which states:
"The Secretary of Transportation shall develop an automated highway and vehicle prototype from which future fully automated intelligent vehicle-highway systems can be developed. Such development shall include research in human factors to ensure the success of the man-machine relationship. The goal of this program is to have the first fully automated roadway or an automated test track in operation by 1997. This system shall accommodate installation of equipment in new and existing motor vehicles."

The goal of an Automated Highway System (AHS) is to significantly increase safety and highway capacity without having to build new roads, by adding intelligence to both the vehicles and the roadside. Inter-vehicle spacing at which human drivers can be made safe by the use of automatic controls. Thus, congestion can be relieved by closely packing the vehicles on an automated highway. Each vehicle in an AHS is controlled by its on-board computer. The individual vehicle control can be broadly divided into two classes: longitudinal control (maintaining a safe distance with respect to the preceding vehicle) and lateral control (keeping the vehicle in the center of the lane and changing lanes as required).

The longitudinal controllers use sensors to obtain relative velocity and distance with respect to the car ahead as well as self-state sensors that provide measurements of its own velocity, acceleration, engine state, etc. Both the accelerator and the brake are automatically controlled to achieve
the desired result. In an AHS, the longitudinal controller of the vehicle must possess the following functionality:

- Autonomous/Adaptive Cruise Control

The task of an Autonomous Cruise Controller (ACC) is to follow the preceding vehicle at a safe distance. In a non-platoon architecture, this controller is used by all vehicles. In a Platoon architecture such a controller is used by the leader of a platoon, when it is not involved with a maneuver.

- Follower Control Law

The task of this controller is to maintain a small (in the range of 1-4 m) fixed distance from the preceding vehicle. This controller will be used by the followers of a platoon. In addition to the sensors mentioned above, the follower control law needs to know the state of the lead vehicle of a platoon.

- Transition Maneuvers

Involves control laws for lane change, entry and exit maneuvers as well as join and split maneuvers used to form and break up platoons in a platoon environment.
2.3 AHS System Study Background

Early research in the area of the automatic control of individual rubber-tired vehicles was conducted by the General Motors Corporation[49], Ohio State University[5], the Japan Governmental Mechanical Laboratory[52], the U.K. Road research Laboratory[9], Ford Motor Company[15], and the Japan Automobile Research Institute[42]. In 1986, the Federal Railroad Administration initiated the Northeast Corridor study, one part of which was focused on the automatic highway[70] as a means of relieving congestion in this corridor. Here, TRW Systems Group considered intercity travel as part of the Northeast Corridor studies and examined a representative automatic highway system that would accommodate rubber-tired passenger vehicles; the general system concept requires the use of synchronous network control. An extension of this effort reported estimates of cost and performance characteristics of the representative system. Partly in response to the needs defined in the Department of Housing and Urban Development (HUD) report, "Tomorrow's Transportation Today"[17], an increasing effort was devoted to automated ground transport for urban areas. Some of this effort, which was focused on automated guideway transit[1], is also applicable to AHS. The application of dual-mode vehicles for intra- and inter-urban or intercity travel has been studied, and several system concepts have been developed and reported in the literature. For example, General Motors[28] and Rohr Industries[55] have been studied the use of rubber-tired vehicles for intra-urban applications. In the former, the vehicles would be under synchronous network management and electronically steered while on the automated
portion of the system guideway, whereas the latter approach employed an asynchronous management policy and utilized a mechanical steering technique.

The use of automated pallets has also been frequently suggested as a means of providing intracity and intercity travel. Wilson et al. [74] have envisioned the use of the palletized automated transit (PAT) system for urban application in transporting ordinary vehicles and freight containers along a guideway network. Here, the pallet is supported by unflanged steel wheels and laterally constrained by mechanical guidance and switching arms, with the propulsive force being supplied through a con-railway drive technique. A tracked aircushion pallet system has been reported in the literature by Transportation Technology, Inc. [75] for urban application, wherein the pallets would be moved throughout the systems by means of a linear induction motor. A multi-model system consisting of pallets that would carry automobiles, completely enclosed, on elevated guideway has been suggested by TRW Systems as a means of providing intercity travel along densely related corridor areas. The pallets would be equipped with dc (direct circuit) motors for propulsion, and steel wheels on steel tracks would provide support and guidance for the pallet vehicles.

In addition to the synchronous and asynchronous network efforts previously mentioned, other investigators has addressed the macroscopic problem of overall network control. Researchers at the Applied Physics Laboratory of Johns Hopkins University examined the trade-offs involved in choosing a vehicle management system by making a comparison of synchronous and asynchronous vehicle management strategies and deriving
the effects of alternate routing algorithms on system performance. A computer simulation of vehicle platoon merging into a transit network under quasi-synchronous network control was conducted by Dais and York[16]. Lucus[44] have suggested a synchronous vehicle management policy that incorporates a dynamic scheduling algorithms to allocate guideway space to vehicles in such a manner that the capacity of critical network modes is not exceeded. The operation of this strategy sometimes requires a controlled vehicle to adjust its position in a moving traffic stream (move forward or backward) prior to entering a merge mode. Dynamic scheduling was considered in the study of automated dual-mode network control. The approach utilizes the synchronous concept, but does not employ preprogramming or origin-destination slot reservations. Rather, it uses a combination of path reservations through interchanges. Here, control tasks concerned with individual vehicles, such as scheduling and maneuvering, are handled at the local level, and those concerned with vehicles in aggregate are handled at the central level.

Most of the studies in this area have been conducted using computer simulations of theoretical network configurations; however, some of the network analyses have been site specific. For example, Howson[36] developed a simulation model of an automated roadway network for the metropolitan Detroit, MI area. The simulation effort was conducted to determine system performance as a function of the various design parameters, to develop operation software, and to evaluate the computer hardware needs of a real-world system. Stefanek and Wilkie[67] examined the impact of a hypothetical dual-mode vehicle system on transportation in the Detroit area using data supplied by the Detroit Region Transportation
and Land Use Study to form the user trip matrix. The study results indicate that a considerable improvement in traffic flow can be obtained through the introduction of a dual-mode network, but no figures of merit were established to assess the economic viability of the dual-mode concept. Various forms of dual mode transportation were analyzed in order to assess the economic viability of the dual-mode concept in an urban environment. The study was conducted in a Boston 1990 scenario and system performance was examined for the specially designed small systems. The results obtained from the study indicated that further technological development is warranted to arrive at a viable system. A similar study for automated intercity travel was suggested in a report of the Northeast Corridor Transportation Project that was prepared for the Department of Transportation[8]. Based on study of Northeast Corridor population growth pattern and the region’s transportation systems, the report suggests that effort be expanded to define and establish the desirability of moving toward automated highways for intercity travel along the corridor. To provide a sufficient level of high capacity for long term, the report recommends and expansion of the automated highway research and development program to define and evaluate possible concepts.

Since the mid-1960’s. FHWA has supported efforts that have evolved to encompass system-level aspects, critical subsystem development, and the building and instrument of 4-mi automated highway test facility[27] for the evaluation and demonstration of various aspects of AHS operations. In 1976, FHWA initiated a study to determine the desired degrees of both guideway passivity and vehicle-based intelligence[8].
Vehicle-based intelligence can be greatly enhanced through the application of on-board microprocessor technology.

There is current widespread interest in employing microprocessors in vehicles; however, relatively little effort has been expended on using microprocessors for higher-lever functions such as lateral and longitudinal control, and the associated on-board on-line decision-making that is essential to the functioning of an intelligent vehicle. The few reported efforts include studies at the General Motors Proving Grounds[66], at OSU[5], and at Boeing Aerospace Corporation[34]. The former involved instrumentation of a 2.3-mi (3.7-km) test facility to control two automated test vehicles for normal guideway operation including the merging and diverging functions. The OSU effort involved the implementation and field tests of lateral and longitudinal microprocessor-based controllers. The latter study was laboratory evaluation of a microprocessor-based longitudinal controller for an automatic vehicle. This is an area in which much research and basic development remains to be accomplished.

A considerable effort has been focused on lateral control and longitudinal control. Two approaches to the former: “radar” steering and “wire-follower”(Automotive Engineering, April, 1996) are most applicable to AHS. The only work on radar steering has been done at OSU, where the lateral position of a vehicle, relative to lane center, is obtained by reflecting X-band energy off a guidewall and processing the returned signal. In a wire-follower configuration, the lateral position of a vehicle is obtained via a signal from a current-excited cable embedded in the roadbed. Here, advanced work has been done at both General Motors[34]
and Ohio State University[18] where several lateral controllers have been designed, implemented and evaluated under field conditions. In essence, excellent lateral control was achieved (i.e., close tracking, good insensitivity to disturbance forces, and a comfortable ride). It should be noted that the radar approach would be suitable for a passive guideway whereas the wire-follower approach would be suited for an active one.

The corresponding longitudinal control problem has three aspects: the specification of a vehicle's desired state, the sensing of its actual state, and the realization of a satisfactory controller. The first of these is largely a function of the selected system structure. In the passive guideway case, one approach toward sensing of a vehicle's longitudinal state is by a car-following radar. Current research efforts are focused on an anti-collision radar, which is a forerunner of one that could be employed for automated car-following. Steady progress has been reported, and a suitable radar for long-range sensing may be available within a decade. If it were desired to employ a platoon approach to vehicle management, then a very short range radar would be required. The earliest work on car-following controllers was done by General Motors Corporation in conjunction with Radio Corporation of America in the late 1950's. Subsequently, additional efforts were undertaken in Japan and at OSU. Car-following controllers have also been designed and evaluated for Automatic vehicles; however, as these designs were intended for low-speed large-time headway situations, they are not relevant to the most stringent AHS operating environment.
2.4 AHS Controller Design Background

2.4.1. Drew's Formula

Drew proposed a nonlinear Car-Following model in his book--
Traffic Theory and Flow[19]. Although his theory was derived from human
driver behavior, it is still very useful for our design of automatic vehicle
controller. He assumes that the traffic stream is a superposition of vehicle
pairs where each vehicle follows the vehicle ahead according to a specific
stimulus-response equation that approximates the behavioral and
mechanical aspects of the driver-vehicle-road system. Although this control
law was not developed for AHS, we can still use it as a lead vehicles
control law under certain circumstances.

The general equation for this control is given by:

\[ x_{i+1}(t) = a_m \frac{x_i(t - T) - x_{i-1}(t - T)}{[x_i(t - T) - x_{i+1}(t - T)]^n} \]

Where \( T \) is the time lag of response to the stimulus, \( a_m \) is a constant of
proportionality.

The equation above states that the acceleration of a vehicle at a
delayed time \( T \) is directly proportional to the relative speed of the lead and
the trail vehicles and it is inversely proportional to the relative distance of
the lead and the trail vehicles. This can be considered as one of the earliest
PID controller designs.
2.4.2 Dynamic Aspect Control: Berkeley's Design

Researchers at the University of California, Berkeley developed nonlinear, third order, ordinary differential equations for the design of controller which are given below:

\[ \ddot{x_i} = b_i(x_i, \dot{x_i}) + a_i(x_i)u_i \]

\[ a_i(x_i) = \frac{1}{m_i \tau_i(x_i)} \]

\[ b_i(x_i, \dot{x_i}) = -\frac{2K_{di}}{m_i} \dot{x_i}x_i - \frac{1}{\tau_i(x_i)} (x_i + \frac{K_{di}}{m_i} x_i^2 + \frac{d_{mi}}{m_i}) \]

Where the subscript \( i \) indicates the \( i^{th} \) vehicle; \( X_i \) is the position of this vehicle with respect to a fixed roadside reference; \( x_i \) and \( \dot{x_i} \) are its velocity and acceleration, respectively; \( m_i \) is its mass; \( \tau_i \) is the time constant of its engine; \( u_i \) is the engine input; \( K_{di} \) is the aerodynamic drag coefficient and \( d_{mi} \) is the mechanical drag.

2.4.3 Mechanical Aspect Control

V.K. Narendran and J.K. Hedrick[32-34] took the engine dynamics into consideration and presented a combined vehicle model.
The engine dynamics are captured through two engine states—mass of air in the intake manifold, $m_a$, engine speed, $\omega_e$. The dynamics are represented as:

$$m_a = m_{ai} + m_{ao}$$

$$\omega_e = (t_{net} - t_{pump}) / j_e$$

$$m_{ai} = \beta_1 PRI(p_m / p_a) TC(\alpha)$$

Where $m_{ai}, m_{ao}$ are the mass rates of air flow into and out of the intake manifold respectively; $p_a, p_m$ are atmospheric and manifold pressures respectively and $PRI$ and $TC$ are nonlinear functions; $t_{net}, t_{pump}$ are the net engine and pump torque respectively; $j_e$ is the effective engine inertia. 

*Figure 2-1* is a schematic of the engine.

![Engine Schematic](Image)
The longitudinal dynamics equation is as below:

\[ mV = (f_{\text{drag}} + f_{\text{fractive}}) + f(X) \]

Where \( m \) and \( V \) are vehicle mass and velocity, respectively; \( f_{\text{drag}} \) and \( f_{\text{fractive}} \) are drag forces and net tire forces, respectively; \( f(X) \) arises from the components of forces along the corresponding directions due to the vehicle roll, pitch and yaw movements, it is given as \( f(X) = f_r(f_p + sr) \); where \( sr \) is the slip ratio of the tire.

In addition, the brake torque \( (t_{br}) \) has also been included and is modeled as a first lag.

\[ t_{br} = (t_{br,c} + t_{br}) / \tau_b \]

Where \( t_{br,c} \) is the commanded brake torque; \( \tau_b \) is the time constant of the brake.

The inputs to the model are the throttle angle, \( \alpha \), steering angle, \( \delta \), and \( t_{br,c} \).

Several assumptions were made to solve the problem.

1. Pitch dynamics are neglected but the rolling dynamics are kept.
2. No slip between the wheels and the road, so we can relate the vehicle speed to the engine speed.
3. The road super-elevation and gradient angles have been neglected.
A PID Controller can be designed based on the above discussion. The sliding surface \( S_a \) is defined as a function of the error between actual position( \( R \) ) and desired position( \( R_{des} \)).

\[
S_a = e_a + \lambda_{a,1} e_a + \lambda_{a,2} \int e_a dt
\]

\[
\omega_e = f(X) + c_{11}(X)t_{net} + c_{13}(X)\delta_f + c_{14}(X)\delta_f^2
\]

Where \( e_a = R - R_{des} \).

Differentiating \( S_a \) and make it equal to \(-k_a S_a\). We have the control equations:

\[
U_a = k_a S_a + \lambda_{a,1} e_a + \lambda_{a,2} e_a - R_{des}
\]

\[
V_a = (x_s - y_s)^2 - (x_t - y_t)^2 - (R - \hat{R})^2 / R
\]

\[
b_a = V_a + t_{11}(X)(\psi - \psi_d) - t_{12}(X)\dot{\psi} - t_{13}(X)\ddot{\psi} +
\]

\[
t_{14}(X)f_{\alpha,1}(\psi - \psi_d, \psi_d) + t_{15}(X)f_{\text{drag}, \psi} / m + t_{16}(X)f_{\text{drag}, \psi}
\]

\[
t_{\text{net,des}} = (-U_a - b_a + a_{12}(X)t_b, + a_{13}(X) \delta_f + a_{14}(X)\delta_f^2) / c_{net}
\]

Where \( c_{net} \) is a constant. The desired throttle angle \( \alpha_{des} \) is calculated from the table look-up procedure based on \( t_{\text{net,des}} \) and \( \omega_e \).
Chapter 3

MANUAL CONTROL

3.1 Introduction

Human driving involves reaction times, delays, and human errors that affects traffic flow adversely. The slinky effects, oscillations, and long settling times are observed in the human driver models. We will review the three different human driving models later. The structure of Human Driving model is shown in Figure 3-1:

![Diagram](image.png)

Figure 3-1: Structure of the Human Driving Model
3.1 Linear Follow-the-Leader Model

This model is based on the assumption that each individual driver reacts in some specific fashion to a stimulus from the vehicle(s) ahead. The Figure 3-2 shows the block diagram for this model:

![Block Diagram](image)

Figure 3-2: Linear follow-the-leader Model

It is determined that the average delay time (reaction time) for a driver is 1.5s, and the gain is approximately 0.37 sec\(^{-1}\). The stimulus can be defined as the difference in distance (drivers are not able to perceive the difference in velocity as easily as the difference in distance) and the driver’s response is an acceleration command to the vehicle.
3.2 Linear Optimal Control Model

If we assume that a human driver has the ability to perform a linear optimal control. Their participation are based on a quadratic cost function that penalizes the weighted sum of the square of the inter-vehicle spacing and the square of the relative velocity. Since these weights differ from driver to driver and are, therefore, unknown, this optimal control approach can only be used to come up with the structure of the controller the human driver mimics. And further more, if we take these factors, such as driver’s reaction time, the neuromuscular dynamics, and non-linearity of the vehicle dynamics into consideration, the modified model will be more realistic (from the control point of view) and more nonrealistic (from the driver point of view). The structure is shown in Figure 3-3. The stimulus here are the weighted sum of the square of the inter-vehicle spacing and the square of the relative velocity and also noise from the outside are assumed to be zero.

![Diagram](image)

Figure 3-3: Linear Optimal Control Model
3.3 Look-Ahead Model

The hypothesis for this model is that the human driver has the ability to observe the behavior of three vehicles directly ahead of him. The block diagram of this model is shown in Figure 3-4.

Figure 3-4: Look-Ahead Model

The logic block tests the majority of acceleration. If the majority direction is the same as the 1st lead vehicle, the following vehicle keeps the tracking. If the majority direction is the same as the 2nd lead vehicle, the following vehicle switches to track the 2nd lead vehicle.

3.4 Concluding Remarks

The human driver models considered previously try to mimic the control action of the driver with a certain controller.
This controller is rather simple and consists of a delay that models the human driver reaction time. All models considered assume a simple first-order model for the vehicle dynamics. To test if these models can be used in the controlling of a platoon of vehicles, we use a simple spreadsheet program. Suppose we have three vehicles one following another. \( V_L, V_T1, V_T2 \) are their velocities. \( X_L, X_T1, X_T2 \) are their positions. \( X_R1 \) and \( X_R2 \) are the relative spacing between the lead vehicle--1st trail vehicle and the lead vehicle—2nd trail vehicle, respectively. Their initial speeds are 35\( m/s \), relative initial spacing is 1\( m \). The driver's reaction time is set to be 1.5\( s \). The lead vehicle starts to decelerate from time zero at 5\( m/s \) (maximum deceleration rate accepted by most of the literature).

The allowable relative speed when the trail vehicle hits the lead vehicle is set to be 3\( m/s \) (Current federal regulations require that a bumper sustain a 4 \( Km/h \) impact without damage). The simulation results are given in Figure 3-5.

It is found that the 1st trail vehicle hits the lead vehicle at time 0.65\( s \) with a relative speed of 4.55\( m/s \) which is greater than allowable(3\( m/s \)). The conclusion is that the human driver is not capable of performing satisfactorily for vehicle platoon control. We must replace the human control with a more sophisticated one that is based on a more realistic model of vehicle dynamics and driven by a computer and physical sensors.
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Figure 3-5: Simulation Results

Computer control will eliminate human reaction time, be more accurate, and be capable of achieving much better performance. Better performance will translate into smoother traffic flows, improved flow rate, less pollution, and safer driving.
Chapter 4

AUTOMATIC CONTROL

4.1 Introduction

The word control can be defined as a scheme (or an algorithm) that makes a machine behave according to our wishes. The methods used to accomplish the control objectives can be classified, very generally, as open-loop controls and closed-loop or feedback controls.

Open-loop control is much simpler and less expensive. No sensors are needed to measure the variables and provide the feedback. However, it is not very accurate (compared with feedback control), and its accuracy can vary without this variation being detected by the system.

Closed-loop control is inherently more accurate than the open-loop system because of its principle of operation. It can be designed to provide extreme accuracy in the steady state (for almost all steady-state operating conditions). When we design controllers for vehicles traveling on AHS (Automated Highway Systems) whose intra-platoon spaces are about 1 m and inter-platoon spaces are about 10 m, the controllers' accuracy are the most important issue that we are dealing with. One of the major reasons for choosing to use the feedback control systems in our vehicle following
simulation is the need for extreme accuracy beyond the capacity of the human driver, or of an open-loop control. It is also for the accuracy over long periods of driving, which may be impractical for a human driver because of the effects of fatigue. As a matter of fact, the feedback control systems are being used by all the literature in developing the vehicle following simulation models.

Although the feedback control system presents many advantages, we should not neglect some disadvantages of this kind of system. The system will be complicated by the increased number of components, such as sensors in our case. The system may oscillate from the desired output. However, a feedback control is generally considered superior to an open-loop system.

4.2 Controller Design Requirements

The purpose of designing a high-performance intra-platoon vehicle controller is not to completely avoid rear-end collisions from those close longitudinal spaces (as few as 1m) but to provide a modest relative speed in the event a failure happens. The failure can be a sudden deceleration of the lead vehicle for some reason, etc. For a inter-platoon vehicle controller, it should be designed so that a total stop be reached in the event of a failure and no collisions occur for all vehicles involved. The very close longitudinal spaces between vehicles within a platoon have important implications for the design of a longitudinal control system.
1. It is essential that the spaces between consecutive vehicles be controlled extremely accurate so that the dynamic variations are no more than a fraction of the nominal space. The control system must be sufficiently fast to avoid excessive errors even in highly dynamic maneuvering, but it also have sufficient damping to avoid overshoots that could compromise the small spaces between consecutive vehicle bumpers.

2. The control has to ensure that the platoon remains asymptotically stable, so that disturbances are damped out the further one proceeds along the platoon.

4.3 Three Types of Controller

Three types of controllers are being used in the design. Proportional Feedback Control(P controller), Proportional-Integral Feedback Control(PI Controller) and Proportional-Integral-Differential feedback Control(PID Controller).

4.3.1 Proportional Feedback Control

When the feedback control signal is made to be linearly proportional to the error in the measured output, we call the result Proportional Feedback. The general form of proportional control is:

\[ u = K \cdot e \]

where \( u \) is input, \( e \) is error and \( K \) is a constant
Proportional controller can be viewed as an amplifier with a “knob” to adjust the gain up or down. The system with proportional control may have a steady-state offset (or droop) in response to a constant reference input and may not be entirely capable of rejecting a constant disturbance.

4.3.2 Proportional-Integral Feedback Control

When the integral control is added to the Proportional control, they are named Proportional-Integral feedback control. The reason to add the integral control is to reduce or eliminate constant steady-state errors, but this benefit typically comes at the cost of worse transient response. Integral feedback has the form:

\[ u(t) = \frac{K}{T_I} \int_0^t e \, d\eta \]

where \( T_I \) is the integral, or called reset time.

4.3.3 Proportional-Integral-Derivative Feedback Control

If we introduce the derivative term into the PI controller, we have the Proportional-Integral-Derivative feedback control. The derivative feedback is used in conjunction with proportional and/or integral feedback.
to increase the damping and generally improve the stability of a system. It takes the term:

\[ u(t) = K \cdot T_e \cdot e \]

For the PID controller, the control signal is a linear combination of the error, the time integral of the error, and the time rate of change of the error. All three gain constants are adjustable. It has the form:

\[ u(t) = K_1 \cdot e + K_2 \cdot \int_0^t e(\tau)d\tau + K_3 \cdot e \]

The PID controller is sometimes able to provide an acceptable degree of error reduction simultaneously with acceptable stability and damping. PID controllers are so effective that PID control is standard in most of the processing industries. For the above reasons, the PID control is being used in our vehicle following simulation.

### 4.4 Different Control Methods

There are two fundamentally different ways of designing longitudinal controllers for road vehicles, which are known as Asynchronous( Car-following ) and Synchronous( Point-following ) controllers.
4.4.1 Car-following Control

The Car-following control systems use the differences between the state of each vehicle and the state of its predecessor as the error signals for regulation.

We consider two vehicles on a AHS highway, one following another. The position of the vehicle to a reference spot is $x$. The relative distance of the two vehicles is $d$. The velocity of the vehicle is $x$. Figure 4-1 shows the two vehicles:

![Diagram showing car-following control](image)

Figure 4-1: Car-Following Control

4.4.2 Point-Following Control

In contrast to the Car-following control, the Point-following control systems uses the differences between the position and/or velocity of each vehicle and the corresponding position and/or velocity of a virtual
reference point that moves along the roadway at constant speed as its error signals.

We consider a platoon of $N$ vehicles on a AHS highway. Each vehicle in the platoon is assigned a slot of length $L$, the distance to a reference spot is $x_i$ and the deviation of the actual position to the assigned position is $\Delta_i$. Figure 4-2 shows the point-following control:

![Figure 4-2: Point-Following Control](image)

The design requirements decide which one is better. The comparison is given in the next section.

4.4.3 Comparison of the two types of control

The advantages and disadvantages of Car-Following control and Point-Following control are given in Table 4-1. The comparison assumes that all required technologies are available for the two control modes.

Table 4-1 is given below for the detailed comparison:
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<thead>
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<th>Advantages</th>
<th>Disadvantages</th>
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</thead>
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<td><strong>Car-Following Control</strong></td>
<td></td>
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<tr>
<td>Extremely flexible in a accommodating trains or platoons or vehicles of diverse length</td>
<td>Control system on vehicle is complicated:</td>
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<tr>
<td></td>
<td>- sensing spacing, speed, and acceleration relative to lead vehicle</td>
</tr>
<tr>
<td>Spaces between vehicles can be adjusted as speed changes</td>
<td>- communication between vehicles</td>
</tr>
<tr>
<td>Speed can be adjusted easily to adapt to demand shifts or incidents</td>
<td>- has to be able to minimize interactions among vehicles (asymptotic stability)</td>
</tr>
<tr>
<td>Communication burden of roadway is minimized</td>
<td>- has to operate different modes for lead vehicle</td>
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<tr>
<td>Can still operate without communication between vehicle and roadside</td>
<td>Vehicles need multiple communication paths to/from other vehicles and the roadway</td>
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<td>Normal and emergency control modes are very similar</td>
<td>Flow instabilities are possible if something goes wrong</td>
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<tr>
<td><strong>Point-Following Control</strong></td>
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<tr>
<td>Simple vehicle control implementation—each vehicle follows a well-defined target</td>
<td>Slot length must be long enough for worst-case condition, which limits the capacity</td>
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<tr>
<td>Only one communication path is needed (to roadway)</td>
<td>Not easily adaptable to sudden demand shifts</td>
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<td>Merging, routing, scheduling simplified by fixed, discrete spacing increments</td>
<td>Failure of a vehicle or a wayside system produces shutdown</td>
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<tr>
<td>De-couples vehicle control from system management is needed</td>
<td>A separate back-up communication system in case of roadway communication</td>
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<td>Not suitable for platoons of variable length</td>
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The Ohio State University researchers were among the first in the investigation and research of the AHS. Their efforts placed emphasis on car-following control system modeling at their earlier stage. But they switched their efforts to the point-following approach later on because of the lack of an essential component of a car-following control system—a forward looking, vehicle-based radar system that has the accuracy of $\pm 0.3\,m$ over the range from 10-100$m$. This is a special case in the AHS research. In general, the point-following control system can be considerably simpler to design, but do not offer as much flexibility or as high a potential lane capacity as the car-following system. For our purpose of flexibility and accuracy, we use the car-following control method in our simulation.
Chapter 5

SYSTEM DYNAMICS

"Promoting the rigorous study of systems and their behavior using the principles of feedback, dynamics and simulation."

5.1. Introduction

In 1956 Jay W. Forrester, holding the basic patent for magnetic core memory, decided to put aside his electro-mechanical research and apply the principles of feedback control to socioeconomic systems. Accepting a position at MIT's School of Industrial Management, Professor Forrester began applying these principles to the problems of managing a corporation, resulting in the publication in 1961 of Industrial Dynamics, a comprehensive treatment of the use of feedback principals and simulation to aid in the management of a company.

Over the past three decades System Dynamics has been applied broadly in such areas as environmental change, economic development, social unrest, urban decay, psychology and physiology. There has been a corresponding growth in the base of tools being developed and applied
including things such as causal loop diagramming, chaos theory, statistical analysis and interactive learning environments.

Though eclectic in content and methods, System Dynamics retains certain underlying principles that form an important bridge between reality and our ability to understand:

- Concentration on dynamics and feedback relationships.
- Representation of decision making behavior based on actual information availability.
- Explicit recognition of disequilibrium and the process of adjustment.
- Incorporation of nonlinear relationships when appropriate.
- Quantification of unmeasured but important concepts and relationships.

By maintaining these principles work done in System Dynamics is kept both understandable and relevant.

5.2 Structure

System Dynamics is a method for studying the world around us. Unlike other scientists, who study the world by breaking it up into smaller and smaller pieces, system dynamicists look at things as a whole. The central concept of system dynamics is understanding how all the objects in a system interact with one another. A system can be anything from a steam engine, to a bank account, to a basketball team. The objects and people in a system interact through "feedback" loops, where a change in one variable
affects other variables over time, which in turn affects the original variable, and so on.

An example of this is money in a bank account. Money in the bank earns interest, which increases the size of the account. Now that the account is larger, it earns even more interest, which adds more money to the account. This goes on and on. Another example of a simple feedback loop which we have all experienced is adjusting the water tap to reach a desired temperature. You turn the faucet, feel the temperature, and compare it to the desired temperature. You continue to adjust the water, with smaller and smaller adjustments, until you reach the desired temperature.

What System Dynamics attempts to do is understand the basic structure of a system, and thus understand the behavior it can produce. Many of these systems and problems which are analyzed can be built as models on a computer. System dynamics takes advantage of the fact that a computer model can be of much greater complexity and carry out more simultaneous calculations than can the mental model of the human mind.

5.3 Applications

The professional field known as system dynamics has been developing for the last 40 years and now has a world-wide and growing membership. System dynamics combines the theory, methods, and philosophy needed to analyze the behavior of systems in not only
management, but also in environmental change, politics, economic behavior, medicine, engineering, and other fields. System dynamics provides a common foundation that can be applied wherever we want to understand and influence how things change through time.

The System Dynamics process starts from a problem to be solved to a situation that needs to be better understood, or an undesirable behavior that is to be corrected or avoided. The first step is to tap the wealth of information that people possess in their heads. The mental data base is a rich source of information about the parts of a system, about the information available at different points in a system, and about the policies being followed in decision making. The management and social sciences have in the past unduly restricted themselves to measured data and have neglected the far richer and more informative body of information that exists in the knowledge and experience of those in the active, working world.

System Dynamics uses concepts drawn from the field of feedback control to organize available information into computer simulation models. A digital computer as a simulator, acting out the roles of the operating people in the real system, reveals the behavioral implications of the system that has been described in the model. The first articles based on this work appeared in the Harvard Business Review (Forrester, 1958). From over three decades in system dynamics modeling have come useful guides for working toward a better understanding of the world around us.
5.4 Dynamo—A Simulation Language

DYNAMO was the first system dynamics simulation language, and for a long time the language and the field were considered synonymous. An expert computer programmer, Richard Bennett, worked for Professor Forrester when he was writing the 1958 article, "Industrial Dynamics—A Major Breakthrough for Decision Makers," for the Harvard Business Review. That article is chapter two of Industrial Dynamics. For that article Professor Forrester needed computer simulations and asked Bennett just to code up the equations so they could run them on the computer. However, Dick Bennett was a very independent type. He said he would not code the program for that set of equations but would make a compiler that would automatically create the computer code. He called the compiler "SIMPLE," meaning "Simulation of Industrial Management Problems with Lots of Equations." Bennett's insistence on creating a compiler is another of the important turning points; it accelerated later modeling that rapidly expanded system dynamics. Jack Pugh has since extended the early system dynamics compilers into the very influential DYNAMO series. The language was made commercially available from Pugh-Roberts in the early 1960s. DYNAMO today runs on PC compatibles under DOS/Windows. It provides an equation based development environment for system dynamics models.

5.5 Application to Automatic Vehicle Control

What makes using system dynamics different from other approaches
of studying complex systems, like automatic vehicle control, is the use of feedback loops—how a system is connected by feedback loops which create the non-linearity found.

For the study of automatic vehicle control system, we want to understand and influence how, for example, acceleration and velocity, change through time. We want to unify our knowledge on vehicles dynamics, roadway conditions, central processor characteristics, weather conditions, etc. Systems dynamics is capable of performing all we need. System dynamics is effective because it builds on the reliable part of our understanding of studying systems(vehicles dynamics, roadway conditions, central processor characteristics, weather conditions, etc) while compensating for the unreliable part. The modeling process separate consideration of underlying assumptions from the implied behavior, there is less inclination for people to differ on assumptions. The system dynamics emphasis on endogenous behavior. This is just the same way that we design a vehicle controller. We look at the individual characteristics of the vehicles; consider how they are interconnected and controlled in a platoon; and evaluate the dynamic behavior implied by their feedback loops. We do not attempt to improve our results by using only information from outside but by the vehicle dynamics as well. We do not assume that our control exists in a state of equilibrium that is affected only by exogenous events that impact the vehicle from outside.

From the above discussion, we found out that systems dynamics has advantages compared to other approaches for our purpose and task. It is a
very good choice in simulation of the automatic vehicle control, although the software related to it is still not very user friendly.
Chapter 6

DEVELOPMENT OF CONTROL LAW

6.1 Introduction

The control law we designed should be able to follow the lead vehicle at a safe distance and to track an optimal velocity as closely as possible. Extensive simulation and experimentation by other researchers suggested that it is very unlikely that a single law will be able to produce acceptable performance when faced with various large displacements from the equilibrium point. It is a better way to utilize different laws in different cases. The four cases considered here are:

- Case 1: The velocity of the lead vehicle is faster than the trail vehicle, and the relative distance between the two vehicles is larger than the safe distance.

- Case 2: The velocity of the lead vehicle is faster than the trail vehicle, but the relative distance between the two vehicles is smaller than the safe distance.
• Case 3: The velocity of the lead vehicle is slower than the trail vehicle, but the relative distance between the two vehicles is larger than the safety distance.

• Case 4: The velocity of the lead vehicle is slower than the trail vehicle, and the relative distance between the two vehicles is smaller than the safety distance.

6.2 Mathematical Analysis

In a safe vehicle following, the impact velocity in a collision should never exceed an acceptable limit. High-speed collisions are most likely to occur if the lead vehicle applies and holds maximum braking while the trail vehicle in the progress of catching up with it. The maximum safe approach velocity for the trail vehicle is then defined as the top speed that would allow it to hit the lead vehicle at the acceptable impact velocity if the vehicle ahead were to apply and hold full brake.

The maximum safe velocity is a function of the spacing between vehicles and the lead vehicle. The curve defining the safe velocity consists of two sections. In the first acceleration section, the trail vehicle is far enough from the lead vehicle that maximum deceleration will stop the lead vehicle before the trail vehicle hits it. The safe velocity curve in this section can be found by assuming that the emergency jerk of both vehicles are large and the lead vehicle could reach maximum deceleration essentially immediately, and after the pure time delay, so could the trail
vehicle. $a_{\text{min}}$ is the absolute value of the maximum deceleration. The equations of motion are:

$$x_{\text{lead}}(t) = \Delta x + v_{0\text{lead}}t - a_{\text{min}}t^2 / 2$$

$$v_{\text{lead}}(t) = v_{0\text{lead}} - a_{\text{min}}t$$

$$x_{\text{trail}}(t) = v_{0\text{trail}}t - a_{\text{min}}(t - d)^2 / 2$$

$$v_{\text{trail}}(t) = v_{0\text{trail}} - a_{\text{min}}(t - d)$$

where:

- $v_{0\text{lead}}$ and $v_{0\text{trail}}$ are the velocities of the lead and trail vehicles at the moment the lead vehicle decelerates.
- $x_{\text{lead}}(t)$ and $x_{\text{trail}}(t)$ are the positions of the lead and trail vehicles at the moment that both vehicles stopped.
- $a_{\text{min}}$ is the absolute value of the maximum emergency deceleration.
- $\Delta x$ is the spacing between the rear of the lead vehicle and the front of the trail vehicle.
- $d$ is any delay between the rear of the lead vehicle applies maximum braking and the time when the trail vehicle does.

We now determine the position of the lead vehicle when it stops.

$$t_{\text{stop}} = \frac{v_{0\text{lead}}}{a_{\text{min}}}$$

$$x_{\text{lead}}(t_{\text{stop}}) = \Delta x + v_{0\text{lead}}^2 / 2a_{\text{min}}$$

where:
• $t_{\text{stop}}$ is the time that the lead vehicle and the trail vehicle from braking to stop.

• $a_{\text{min}}$ is the absolute value of the maximum emergency deceleration.

• $\Delta x$ is the spacing between the rear of the lead vehicle and the front of the trail vehicle.

Next we determine when the trail vehicle reaches $v_{\text{allow}}$. At this time the vehicles should collide.

$$t_{\text{hit}} = (v_{\text{trail}} + a_{\text{min}} d - v_{\text{allow}}) / a_{\text{min}}$$

$$x_{\text{trail}}(t_{\text{hit}}) = v_{\text{trail}} (v_{\text{trail}} + a_{\text{min}} d - v_{\text{allow}}) / a_{\text{min}} - (v_{\text{trail}} - v_{\text{allow}})^2 / 2a_{\text{min}}$$

where:

• $t_{\text{hit}}$ is the time from the lead vehicle brake to trail vehicle hit the lead vehicle.

• $v_{\text{allow}}$ is the allowable impact velocity when trail hits the lead vehicle.

Equating $x_{\text{lead}}(t_{\text{stop}})$ and $x_{\text{trail}}(t_{\text{hit}})$ and simplifying gives:

$$(v_{\text{trail}} + a_{\text{min}} d)^2 = v_{\text{lead}}^2 + v_{\text{allow}}^2 + 2a_{\text{min}} \Delta x + (a_{\text{min}} d)^2$$

Assuming that the delay is small and the final term can ignored, this equation can be written:

$$v_{\text{safe}} = -a_{\text{min}} d + \sqrt{v_{\text{lead}}^2 + v_{\text{allow}}^2 + 2a_{\text{min}} \Delta x}$$
If the initial assumption of a large jerk is not made, and \( v_{allow} = 0 \), the equation becomes:

\[
v_{safe} = -a_{\min} d + \sqrt{v_{lead}^2 + (v_{lead} - v_{trail}) \cdot a_{\min}^2 / j_{max}} + 2a_{\min} \Delta x + (a_{\min} d)^2
\]

where:

- \( j_{max} \) is the absolute value of the maximum negative jerk possible in braking.

For \( j_{max} = 50m/s^3 \), the extra term is not significant. This equation can be put in the same form as those in (Sklar et al. 1979).

We conclude the above derivation by defining this equation for the first part of the safe velocity curve (explained later in Safe Boundary):

\[
v_{safe}(v_{lead}(t), \Delta x(t)) = -a_{\min} d + \sqrt{v_{lead}(t)^2 + v_{allow}^2} + 2a_{\min} \Delta x(t)
\]

(1)

where:

- \( v_{safe} \) is the first portion of the maximum safe velocity curve.
- \( a_{\min} \) is the absolute value of the maximum emergency deceleration.
- \( v_{lead} \) is the velocity of the lead vehicle. It can be determined by adding the signals from the relative velocity sensor and the on board velocity sensor in the lead car of the trail vehicle.
- \( v_{allow} \) is the allowable relative impact velocity.
• $\Delta x$ is the spacing between the rear of the lead vehicle and the front of the trail vehicle.

• $d$ is any delay between the rear of the lead vehicle applies maximum braking and the time when the trail vehicle does.

If full braking does not stop the lead vehicle before the trail vehicle hits it, the safe velocity is given by:

$$v_{safe2} (v_{lead} (t)) = -a_{min} d + v_{lead} (t) + v_{allow},$$

where:

• $v_{safe2}$ is the second part of the safe velocity curve.

For a safe, time-optimal merge situation (case 1), the trail vehicle tracks the maximum safe velocity curve defined by these equations. The vehicle’s initial and final velocities are usually well under the safe curve. To reach the safe curve, the trail vehicle can accelerate at the maximum comfortable level, or maximum possible level. To slow to $v_{lead}$ at the end of the merge, the trail vehicle decelerates at the maximum comfortable level. The deceleration curve can be written as a function of $v_{lead}$ and $\Delta x$.

The equations of motion are:

$$x_{lead} (t) = \Delta x + v_{0lead} t + a_{lead} t^2 / 2$$

$$v_{lead} (t) = v_{0lead} + a_{lead} t$$
\[ x_{\text{trail}}(t) = v_{0\text{trail}} t - (a_{\text{com}} - a_{\text{lead}}) t^2 / 2 \]

\[ v_{\text{trail}}(t) = v_{0\text{trail}} - (a_{\text{com}} - a_{\text{lead}}) t \]

\[ \Delta v_{\text{final}} = -a_{\text{com}}^2 / 2 j_{\text{com}} = \Delta v_{\text{initial}} + a_{\text{com}} t_{\text{final}} \]

\[ t_{\text{final}} = -(a_{\text{com}}^2 / 2 j_{\text{com}} + \Delta v_{\text{initial}}) / a_{\text{com}} \]

\[ \Delta x_{\text{final}} = \Delta x_{\text{desired}} + a_{\text{com}}^2 / 6 j_{\text{com}} = x_{\text{lead}}(t_{\text{final}}) - x_{\text{trail}}(t_{\text{final}}) \]

So, we have the equation for deceleration curve:

\[ v_{\text{decel}}(v_{\text{lead}}(t), \Delta x(t)) = v_{\text{lead}}(t) + \sqrt{2a_{\text{com}} (\Delta x(t) - \Delta x_{\text{desired}})} \]  

(3)

where:

- \( v_{\text{decel}} \) is the desired velocity profile for the deceleration part.
- \( a_{\text{com}} \) is the absolute value of the maximum comfortable acceleration or deceleration.
- \( x_{\text{desired}} \) is the desired intravehicle spacing.

The acceleration part of the trail vehicle trajectory can not be easily expressed in terms of \( \Delta x \) and \( v_{\text{lead}} \). The initial relative velocity and relative acceleration of the two vehicles are variables, as is the acceleration capability of the trail cars. Safety is least critical in this region, however, and a desired velocity need not be defined. The desired acceleration is given by:

\[ v_{\text{accel}} = a_{\text{com}} \]  

(4)
At high speeds, most vehicles can not achieve \( a_{com} \), which is taken to be \( 2 \text{ m/s}^2 \). Let \( a_{\text{max}}(v) \) the maximum acceleration a vehicle can reach. In order to find a equation for this variable. We have several basic equations. Please refer to Appendix A for details. The power developed by a standard IO engine driven vehicle is given by:

\[
P = \frac{1}{375} \cdot F_p \cdot V
\]

where:
- \( P \) is power, \( hp \).
- \( F_p \) is propulsive force, \( lb \).
- \( V \) is the speed of the vehicle, \( \text{mi/h} \).

\[
F_a = F_p \cdot f
\]

where:
- \( F_a \) is the force for acceleration, \( lb \).
- \( f \) is the total resistance, \( lb \).

\[
f = f_i + f_g + f_c
\]

where:
- \( f_i \) is the inherent resistance, \( f_i = 0.01T + 0.0001TV + 0.0026CAV^2 \), \textit{cars and buses}.

\[
f_i = 7.6T + 0.09TV + 0.002AV^2, \textit{trucks}. (T \text{ is in 1000lbs} \text{ (Taborek, 1957)})
\]
- \( f_g \) is the gradient resistance, \( f_g = T \frac{P}{100}, \text{lb} \).
\( f_c \) is the curvature resistance, \( f_c = \frac{T V^2}{g r} \left( \frac{lb}{ft^2/s^2} \right) \left( \frac{ft}{s^2} \right) \left( \frac{ft}{lb} \right) \).

- \( T \) is vehicle weight, \( lb \).
- \( p \) is percent grade.
- \( r \) is radius of curvature, \( ft \).
- \( g = 32 \, ft/s^2 \).

From the equations above, we have:

\[
a_{max} = \frac{F_s}{T} = \frac{375P}{V} - f_i - f_s - f_c
\]

Since the above equation is very nonlinear, we can also use a rough linear approximation:

\[
a_{max} = 2.26 \, m/s^2 - (0.05 \, s^{-1}) v_{tail}
\]

The road is assumed to be flat. The vehicles are assumed to have automatic transmission in third gear.

The phase-plane trajectory in includes abrupt changes in acceleration at each point where curves intersect (shown later in Safe Boundary). These sections must be smoothed so as not to violate the jerk comfort constraints. The following changes are made in the basic profile to smooth the changes in acceleration.
1. Maximum comfortable jerk must be applied. The deceleration curve should bring the trail vehicle to a spacing and velocity at which applying the jerk comfort limit will bring it to the final desired spacing and velocity. The difference in acceleration between the lead and trail vehicles before applying that jerk is $-a_{com}$. So the time required to reach zero acceleration is $a_{com} / j_{com}$. Assuming the final velocity is $v_{lead}$, the initial velocity before applying the comfortable jerk is given by:

$$v_{trail} = v_{lead} + \frac{a_{com}^2}{j_{com}} - a_{com}^2 / a_{com} = v_{lead} + aj_{com}$$

The final spacing should be $\Delta x_{desired}$, so the initial spacing before applying the comfortable jerk is given by:

$$\Delta x = \Delta x_{desired} + a_{com}^3 / aj_{com}^2 - a_{com}^3 / a_{com}j_{com} + a_{com}^3 / 6j_{com}^2$$

$$\Delta x = \Delta x_{desired} + a_{com}^3 / 6j_{com}^2$$

The initial blend values are the final values for the deceleration curve. On the deceleration curve, comfort level deceleration is assumed when the lead vehicle is maintaining a constant velocity. When the lead vehicle is accelerating or decelerating, the acceleration difference between vehicles is the comfort level.

Solving these equations for $v_{trail}$ gives:
\[ v_{trail} = v_{lead} + \sqrt{a_{com}^4 / 4j_{com}^2 + 2a_{com} (\Delta x - \Delta x_{desired})} = a_{com}^3 / 6j_{com} \]

The resulting equation can be rewritten:

\[ v_{decel} = v_{lead} + \sqrt{a_{com}^4 / 4j_{com}^2 + 2a_{com} (\Delta x - \Delta x_{desired} - a_{com}^3 / 6j_{com})} \tag{6} \]

where:

- \( a_{com} \) is the absolute value of the maximum comfortable acceleration and deceleration.
- \( \Delta x_{desired} \) is the desired intra-vehicle spacing.

2. To make the transition from the acceleration curve to the first part of the safe velocity curve, \( j_{com} \) must be applied at the correct point on the acceleration curve. Acceleration along the first part of the safe velocity curve can be estimated as \( a_{lead} - a_{com} / 2 \). The total change in acceleration from the acceleration curve to the safe curve is then \( a_{lead} - a_{trail} - a_{com} / 2 \). Assuming that the maximum comfortable jerk is used, the time required to make this change in acceleration is: \( (0.5a_{com} + a_{trail} + a_{lead}) / j_{com} \). The change of the velocity during this time is:

\[ (a_{trail}^2 - (0.5a_{com} - a_{lead})^2) / 2j_{com} \]

Define \( v_{trail}, v_{lead}, \) and \( Dx_e \) as the expected trail and lead vehicle velocities and relative spacing, respectively, at time \( t + (0.5a_{com} + a_{trail} - a_{lead}) / j_{com} \) if the trail vehicle were to apply and hold \( j_{com} \) until that time. To end the transition on the safe curve, negative maximum
jerk is needed when the velocity given by the safe curve at \((v_{\text{lead}}, \Delta x_e)\)
is \(v_{\text{trail}}\). The transition begins when the following inequality is satisfied:

\[
v_{\text{safe}} \leq v_{\text{trail}} + (a_{\text{trail}}^2 - (0.5a_{\text{com}} - a_{\text{lead}})^2) / 2j_{\text{com}}
\]  

These calculations are based on the assumption that the jerk of the lead vehicle during the transition is zero. To generalize the transition to handle all cases, the jerk of the trail vehicle must be \(j_{\text{lead}} - j_{\text{com}}\) during the transition.

3. A similar calculation can be made for the transition between the second part of the safe curve and the deceleration curve. In this case, the time in the transition is \(a_{\text{com}} / j_{\text{com}}\). Negative maximum comfortable jerk should be applied when:

\[
v_{\text{decel}} \leq v_{\text{trail}} + (2a_{\text{trail}}a_{\text{com}} - a_{\text{com}}^2) / 2j_{\text{com}}
\]

When we design a controller for the above states described. A general feedback control law which takes the same form in all of these states can be expressed as:

\[
u = x = x_{\text{desired}} + k_a(x_{\text{desired}} - x) + k_v(x_{\text{desired}} - x)
\]

where:
\[ k_a, k_v \text{ are controller gains.} \]

\[ x_{\text{desired}} \text{ is given by equations (1), (2), and (6), respectively.} \]

And now derive the terms \( x_{\text{desired}} \) and \( \dot{x}_{\text{desired}} \) which are the first and second time derivatives of the desired velocity---\( x_{\text{desired}} \).

We have:

\[
v_{\text{desired}} = \frac{\partial v_{\text{desired}}}{\partial \Delta x} \Delta x + \frac{\partial v_{\text{desired}}}{\partial v_{\text{lead}}} v_{\text{lead}}
\]

\[
v_{\text{desired}} = \frac{\partial v_{\text{desired}}}{\partial \Delta x} \Delta x + \frac{\partial v_{\text{desired}}}{\partial v_{\text{lead}}} v_{\text{lead}} + \frac{\partial v_{\text{desired}}}{\partial v_{\text{trail}}} v_{\text{trail}} + \frac{\partial v_{\text{desired}}}{\partial v_{\text{lead}}} v_{\text{lead}}
\]

In a conclusion, a control law has been found on the above discussion. This control law uses a multiple modes controller for automatic vehicle headway control. Namely, deceleration mode controller, first part acceleration mode controller, second part acceleration mode controller. For acceleration control mode, a simple first order control law can be used. The equations are given as follow.

1. Deceleration mode controller:

\[
x_{\text{desired}} = a_{\text{lead}} + \frac{a_{\text{com}}(v_{\text{lead}} - v_{\text{trail}})}{\sqrt{a_{\text{com}}^4/4j_{\text{com}}^2 + 2a_{\text{com}}(\Delta x - \Delta x_{\text{desired}} - a_{\text{com}}^3/6j_{\text{com}}^2)}}
\]
\[ x_{\text{desired}} = j_{\text{lead}} - \frac{a^2_{\text{com}} (v_{\text{lead}} - v_{\text{trail}})^2}{(\sqrt{a^4_{\text{com}} / 4j^2_{\text{com}}} + 2a_{\text{com}} (\Delta x - \Delta x_{\text{desired}} - a^3_{\text{com}} / 6j^2_{\text{com}}))^3} + \frac{a_{\text{com}} (a_{\text{lead}} - a_{\text{trail}})}{\sqrt{a^4_{\text{com}} / 4j^2_{\text{com}}} + 2a_{\text{com}} (\Delta x - \Delta x_{\text{desired}} - a^3_{\text{com}} / 6j^2_{\text{com}})} \]

2. First part acceleration mode controller:

\[ x_{\text{desired}} = \frac{v_{\text{lead}} a_{\text{lead}} + a_{\text{min}} (v_{\text{lead}} - v_{\text{trail}})}{\sqrt{v^2_{\text{lead}} + v^2_{\text{allow}} + 2a_{\text{min}} \Delta x}} \]

\[ x_{\text{desired}} = \frac{v_{\text{lead}} j_{\text{lead}} + a_{\text{lead}} (a_{\text{lead}} + a_{\text{min}}) - a_{\text{min}} a_{\text{trail}}}{\sqrt{v^2_{\text{lead}} + v^2_{\text{allow}} + 2a_{\text{min}} \Delta x}} - \frac{(a_{\text{lead}} v_{\text{lead}} + a_{\text{min}} (v_{\text{lead}} - v_{\text{trail}}))^2}{(\sqrt{v^2_{\text{lead}} + v^2_{\text{allow}} + 2a_{\text{min}} \Delta x})^3} \]

3. Second part acceleration mode controller:

\[ x_{\text{desired}} = a_{\text{lead}} \]

\[ x_{\text{desired}} = j_{\text{lead}} \]

4. For \( v_{\text{acc}} \) a simple first order control law can be used which is also suggested in most of the literature:
\[ u = k_a (a_{com} - x) \]

### 6.3 Safe Boundary

Safe boundaries are found using the equations given above. By using the Spreadsheet as a simulation tool, we found out the safe boundary for all the four cases. The parameter values used in this simulation are given in Appendix A. The four cases are:

- **Case 1**: The velocity of the lead vehicle is faster than the trail vehicle, and the relative distance between the two vehicles is larger than the safety distance.

- **Case 2**: The velocity of the lead vehicle is faster than the trail vehicle, but the relative distance between the two vehicles is smaller than the safety distance.

- **Case 3**: The velocity of the lead vehicle is slower than the trail vehicle, but the relative distance between the two vehicles is larger than the safety distance.

- **Case 4**: The velocity of the lead vehicle is slower than the trail vehicle, and the relative distance between the two vehicles is smaller than the safety distance.
6.3.1. Case 1 Safe Boundary

Figure 6-1: Relative Distance Versus Velocity

Figure 6-2: Relative Distance Versus Relative Velocity
6.3.2. Case 2 Safe Boundary

Figure 6-3: Relative Distance Versus Velocity

Figure 6-4: Relative Distance Versus Relative Velocity
6.3.3. Case 3 Safe Boundary

Figure 6-5: Relative Distance Versus Velocity

Figure 6-6: Relative Distance Versus Relative Velocity
6.3.4. **Case 4 Safe Boundary**

![Case 4 Safe Boundary](image)

**Figure 6-7: Relative Distance Versus Velocity**

![Relative Distance Versus Relative Velocity](image)

**Figure 6-8: Relative Distance Versus Relative Velocity**
6.3.5 Figure Analysis

From Figures 6-1 through 6-8, the safe boundaries were found for all the cases. The boundaries are defined as the intersected areas of $V-a$, $V-s_1$, $V-s_2$, $V-d$ which are safe acceleration velocity, safe 1 velocity, safe 2 velocity, and safe deceleration velocity, respectively. For case 1, the safe boundary starts from $V-a$ at relative distance 100$m$ and meets $V-s_1$ at relative distance 65$m$ the boundary will go along $V-s_1$ until at the relative distance 30$m$ when it meets $V-s_2$, and goes along $V-s_2$ until at the relative distance 2$m$, $V-d$ is applied to the 1$m$ of desired distance. The safe area will be all the space under the boundary. For case 2, the safe boundary starts from $V-d$ to $V-s_2$ until desired space. For case 3, the safe boundary is same as defined in case 1. For case 4, the safe boundary is same as defined in case 2. After finishing this step, a simulation tool should be developed to simulate the designed control law.
Chapter 7

DEVELOPMENT OF SIMULATION TOOL

7.1 Introduction

What makes using system dynamics different from other approaches of studying complex systems, like automatic vehicle control, is the use of feedback loops—how a system is connected by feedback loops which create the non-linearity of the system.

A third-order nonlinear control law is found to be accurate enough for our design purpose. This third-order nonlinear control law takes three level variables: acceleration, velocity, distance.

7.2 Causal Diagram

We assume an identical design speed for all the vehicles operating on each section of the automated highway based on the roadway conditions. The model is established based on the following description.
A platoon consists of a certain amount of vehicles (maybe based on roadway condition or vehicle types in the platoon) traveling on an AHS lane at a certain design speed. A safety distance between the lead and trail vehicle is found from the safe boundary, which is a function of the speed and deceleration capacity of the vehicle.

For safety considerations, a vehicle must communicate with its platoon leader. The lead vehicle will travel at the design speed at equilibrium. If relative distance between the lead and trail vehicles exceeds the safe boundary, a deceleration, or brake will apply. Maximum acceleration or deceleration rate and maximum jerk rate apply when accelerating or decelerating. Comfortable jerk and acceleration rate apply under normal conditions. Maximum acceleration and jerk apply under emergency circumstances. There are two time lags in the model, sample time and power train response time. Other delays are assumed to be zero. Weather factors are taking into account. Vehicle dynamics are considered.

Based on the above verbal descriptions, we can draw the causal diagram to help us understand how all objects in this system affecting each other. Three pair of level variables are underlined. AL and AT, VL and VT, XL and XT. Figure 7-1 shows the Causal Diagram.

Figure 7-1 Causal Diagram for Automatic Control
Where:

- ACOM - ACCELERATION AT COMFORT.
- AL - ACCELERATION OF LEAD VEHICLE.
- ALMAX - ACCELERATION OF LEAD VEHICLE AT MAXIMUM.
- ALR - ACCELERATION OF LEAD VEHICLE RELATIVE.
- AT - ACCELERATION OF TRAIL VEHICLE.
- ATMAX - ACCELERATION OF TRAIL VEHICLE AT MAXIMUM.
- ATR - ACCELERATION OF TRAIL VEHICLE RELATIVE.
- CL - AIR RESISTANCE PARAMETER, LEAD VEHICLE.
- CT - AIR RESISTANCE PARAMETER, TRAIL VEHICLE.
- DRA - DELAYED RELATIVE ACCELERATION.
- DRV - DELAYED RELATIVE VELOCITY.
- DRX - DELAYED RELATIVE POSITION.
- FAL - FORCE OF ACCELERATION, LEAD VEHICLE.
- FAT - FORCE OF ACCELERATION, TRAIL VEHICLE.
- FPL - FORCE OF PROPULSION, LEAD VEHICLE.
- FPT - FORCE OF PROPULSION, TRAIL VEHICLE.
- FRL - FORCE OF RESISTANCE, LEAD VEHICLE.
- FRT - FORCE OF RESISTANCE, TRAIL VEHICLE.
- FRCL - FORCE OF RESISTANCE FROM CURVE, LEAD VEHICLE.
- FRCT - FORCE OF RESISTANCE FROM CURVE, TRAIL VEHICLE.
- FRGL - FORCE OF RESISTANCE FORM GRADIENT, LEAD VEHICLE.
- FRGT - FORCE OF RESISTANCE FROM GRADIENT, TRAIL VEHICLE.
- FRIL - FORCE OF RESISTANCE FROM INHERENT, LEAD VEHICLE.
- FRIT - FORCE OF RESISTANCE FROM INHERENT, TRAIL VEHICLE.
- GCL - GRAVITATIONAL CONSTANT, LEAD VEHICLE.
- GCT - GRAVITATIONAL CONSTANT, TRAIL VEHICLE.
- GL - GRADIENT, LEAD VEHICLE.
- GT - GRADIENT, TRAIL VEHICLE.
- JCOM - JERK COMFORT.
- JMAX - JERK AT MAXIMUM.
- JRLD - JERK OF LEAD VEHICLE DELAYED.
- JRLI - JERK OF LEAD VEHICLE INITIAL.
- JRTD - JERK OF TRAIL VEHICLE DELAYED.
- JRTI - JERK OF TRAIL VEHICLE INITIAL.
- JRTID - JERK OF TRAIL VEHICLE INITIAL DESIRED.
- Ka AND Kv - GAIN.
- ML - MASS, LEAD VEHICLE.
- MT - MASS, TRAIL VEHICLE.
- PL - POWER, LEAD VEHICLE.
- PT - POWER, TRAIL VEHICLE.
- RA - RELATIVE ACCELERATION.
- RL - RADIUS OF CURVATURE, LEAD VEHICLE.
- RT - RADIUS OF CURVATURE, TRAIL VEHICLE.
- RV - RELATIVE VELOCITY.
- RX - RELATIVE POSITION.
- RXD - RELATIVE POSITION DESIRED.
- SSL - CROSS SECTION AREA, LEAD VEHICLE.
- SST - CROSS SECTION AREA, TRAIL VEHICLE.
- VL - VELOCITY OF LEAD VEHICLE.
- VLR - VELOCITY OF LEAD VEHICLE RELATIVE.
- VT - VELOCITY OF TRAIL VEHICLE.
- VTR - VELOCITY OF TRAIL VEHICLE RELATIVE.
- XL - POSITION OF LEAD VEHICLE.
- XT - POSITION OF TRAIL VEHICLE.
Chapter 8

SIMULATION RESULTS

8.1 Introduction

The main simulation task—automatic vehicle controller design is performed using DYNAMO III on a Mainframe computer. The supporting simulation is done using the Spreadsheet. Four cases considered are:

- Case 1: The velocity of the lead vehicle is faster than the trail vehicle, and the relative distance between the two vehicles is larger than the safety distance.
- Case 2: The velocity of the lead vehicle is faster than the trail vehicle, but the relative distance between the two vehicles is smaller than the safety distance.
- Case 3: The velocity of the lead vehicle is slower than the trail vehicle, but the relative distance between the two vehicles is larger than the safety distance.
- Case 4: The velocity of the lead vehicle is slower than the trail vehicle, and the relative distance between the two vehicles is smaller than the safety distance.
8.2 Simulation Results

The purpose of designing a simulation tool is to test and evaluate the control law developed in Chapter 6. This simulation tool is developed in Chapter 7. After the extensive simulation, a satisfactory result is found by setting vehicle gain $K_a$ and $K_v$ as 8 and 16, respectively. The simulation results are given in the following figures.

Figure 8-1: Case 1 Velocity-Time Diagram
Figure 8-2: Case 1 Acceleration-Time Diagram

Figure 8-3: Case 1 Relative Distance-Time Diagram
Figure 8-4: Case 2 Velocity-Time Diagram

Figure 8-5: Case 2 Acceleration-Time Diagram
Figure 8-6: Case 2 Relative Distance-Time Diagram

Figure 8-7: Case 3 Velocity-Time Diagram
Figure 8-8: Case 3 Acceleration-Time Diagram

Figure 8-9: Case 3 Relative Distance-Time Diagram
Figure 8-10: Case 4 Velocity-Time Diagram

Figure 8-11: Case 4 Acceleration-Time Diagram
Case 4: Relative Distance

Figure 8-12: Case 4 Relative Distance-Time Diagram
8.3 Figure Analysis

Figure 8-1 through Figure 8-3 are Case 1 Velocity-Time, Acceleration-Time and Relative Distance-Time Diagrams, respectively. This Case is considered to be merge following situation. The trail vehicle will accelerate at $A_{com}$ (Comfortable acceleration) until it reaches the Safe 1 following situation according to the Case 1 safe boundary defined in Chapter 6, then Safe 2 following situation and finally deceleration following situation until the vehicle reaches the desired relative distance, designed as 1 m in this research. The results are found to be satisfactory – smooth and safe following.

Figure 8-4 through Figure 8-6 are Case 2 Velocity-Time, Acceleration-Time and Relative Distance-Time Diagrams, respectively. This Case is considered to be slow deceleration following situation. The trail vehicle will decelerate at $A_{com}$ (Comfortable deceleration) until it reaches the Safe 2 following situation according to the Case 2 safe boundary defined in Chapter 6, and finally deceleration following situation until the vehicle reaches the desired relative distance, designed as 1 m in this research. The results are found to be satisfactory – smooth and safe following.

Figure 8-7 through Figure 8-9 are Case 3 Velocity-Time, Acceleration-Time and Relative Distance-Time Diagrams, respectively. This Case is considered to be slow acceleration following situation. The trail vehicle will follow the Case 3 safe boundary similar to the case 1. The results are found to be satisfactory – smooth and safe following.
Figure 8-10 through Figure 8-12 are Case 4 Velocity-Time, Acceleration-Time and Relative Distance-Time Diagrams, respectively. This Case is considered to be emergency braking following situation. This is the most difficult situation to be controlled. The trail vehicle will decelerate at $A_{\text{max}}$ (Maximum deceleration). The comfortable deceleration will no longer hold. The fluctuation here is great, but a safe following is reached.
CONCLUSIONS AND FURTHER RESEARCH

9.1 Introduction

This chapter summarizes the results of this research and suggests several directions that could be taken in the future for further study. This research uses basic vehicle motion laws and System Dynamics to form a control law and to develop a simulation tool.

9.2 Summary of Research

Both theoretical analysis and practical computer simulations are used in this research for a better controller and a simulation tool design. First, A third order nonlinear, Car-following, PID control law has been developed. This law uses a multiple modes controller for vehicle headway control in handling different operating situations, instead of a uniform control algorithm, to achieve a better control quality.
Second, a System Dynamics simulation tool has been developed. This tool uses feedback loops to study non-linearities. So we can understand and influence how, for example, acceleration and velocity, change through time. We can unify our knowledge on vehicles dynamics, roadway conditions, central processor characteristics, weather conditions, etc. This simulation tool is effective because it builds on the reliable part of our understanding of studying systems (vehicles dynamics, roadway conditions, central processor characteristics, weather conditions, etc) while compensating for the unreliable part. The modeling process separate consideration of underlying assumptions from the implied behavior, there is less inclination for people to differ on assumptions. By using this simulation tool we are able to look at the individual characteristics of the vehicles; consider how they are interconnected and controlled in a platoon; and evaluate the dynamic behavior implied by their feedback loops. We do not need to assume that our control exists in a state of equilibrium that is affected only by exogenous events that impact the vehicle from outside.

The control law has been simulated using the System Dynamics simulation tool for vehicle traveling on AHS. The weather conditions and road conditions are considered, such as, friction factor, resistance forces, gradient, curvature. The vehicle dynamics are also considered, such as, vehicle mass, power train delay time, cross section area. The performance of the control law and the simulation tool is found to be satisfactory which means that the control law developed in this research is capable of operating under any conditions.
9.3 Conclusions

It is found that in Case 1, the trail vehicle will accelerate at $A_{\text{com}}$ (Comfortable acceleration) until it reaches the Safe 1 following situation according to the Case 1 safe boundary defined in Chapter 6, then Safe 2 following situation and finally deceleration following situation until the vehicle reaches the desired relative distance, designed as 1 m in this research. The results are satisfactory – smooth and safe following.

For Case 2, the trail vehicle will decelerate at $A_{\text{com}}$ (Comfortable deceleration) until it reaches the Safe 2 following situation according to the Case 2 safe boundary defined in Chapter 6, and finally deceleration following situation until the vehicle reaches the desired relative distance, designed as 1 m in this research. The results are found to be satisfactory – smooth and safe following.

For Case 3, the trail vehicle will follow the Case 3 safe boundary similar to the case 1. The results are found to be satisfactory – smooth and safe following.

For Case 4, the trail vehicle will decelerate at $A_{\text{max}}$ (Maximum deceleration). The comfortable deceleration will no longer hold. The fluctuation here is great, but a safe following is reached.

The control strategy proposed in this research improves the safety and comfort of the longitudinal platoon traveling. The strategy also increases the robustness of the maneuvers to variable acceleration performance. By changing certain parameters in the control equations in
case of bad weather or road conditions, the maneuvers can also be made robust to variable deceleration performance.

It is found that vehicle automatic headway control is a complicated process involving a variety of factors -- human, mechanical, environmental, and roadway. These factors affect the properties of the control system greatly. It is very difficult to determine all these factors in an analytical model. System Dynamics simulation can afford a means of modeling this control process with various certain and uncertain factors, and therefore, it is the key to solve the automatic headway control problem.

System delay greatly influence the vehicle headway control process. It brings an additional feedback loop into the control system, and thus increases the order of the system. The delay in our automatic headway control system is the major cause of oscillations in the control system which must be fully considered.

9.3 Further Research

This research did not take into account the variation of the drag coefficient in a platoon of vehicles which suggests the importance of platoons having non-uniform spaces when operating with vehicles of different shapes. Since all platooning vehicles are to be equipped with on-board computers and engine performance sensors, it would be a relatively simple additional task to optimize spacing to achieve a local minimum power expenditure (minimum drag). This is a subject for further research.
In this research, only vehicle longitudinal control is considered, further research may conduct in the study of the effects on the lateral displacement of vehicles when collisions happen.

The simulation Language used in this research -- DYNAMO III, is still not very user friendly. A little program can be developed for us to export the output data and import them to Microsoft Excel for a better looking figure. This improvement will save us a lot of time.
REFERENCES:


18. Drew, Donald R. "Engineering 5104 Class Notes". Virginia Polytechnic Inst. & State University, 1995


APPENDIX A: PARAMETER VALUES FOR SIMULATION

- $|a_{com}| = 2 \text{ m/s}^2$. This value is currently accepted in the literature.

- $a_{\text{min}} = 5 \text{ m/s}^2$. This is the accepted value for maximum deceleration.

- $a_{\text{max}} = 2.26 \text{ m/s}^2 - (0.05 \text{ s}^{-1}) v_{\text{trail}}$. This is a rough approximation based on other's research.

- $|j_{\text{com}}| = 2.5 \text{ m/s}^3$. Most literature used this value.

- $|j_{\text{max}}| = 50 \text{ m/s}^3$. This is a physical limit on jerk.

- $v_{\text{allow}} = 3 \text{ m/s}$. The severity of injuries in automobile accidents is measured on the Abbreviated Injury Scale (AIS). Injuries rated from 3 to 6 on this scale are considered serious. Injuries of AIS = 2 are moderate. They are life threatening but may be temporarily incapacitating. Examples are simple bone fractures or major abrasions. It was found that for crashes at or below 3.3 m/s, no probability of fatalities or injuries rated AIS greater than 3.

- $\Delta x_{\text{desired}} = 1 \text{ m}$. This is the current intra-platoon spacing.

- $d = 200 \text{ msec}$. This can be achieved by redesigning the brake
system, sensors etc.

- \( k_a = 8, k_r = 16 \).

- \( ML = 3000 \text{ LB}, MT = 50000 \text{ LB} \).

- \( PL = 110 \text{ HP}, PT = 350 \text{ HP} \).

- \( GL = 1, GT = 1 \).

- \( SSL = 19 \text{ FT}^2, SST = \text{WIDTH} \cdot (HEIGHT - 0.0625) \text{ FT}^2 \).

  \[
  \text{WIDTH} = 8.5 \text{ FT, HEIGHT} = 13.5 \text{ FT}.
  \]

- \( GCL = 32 \text{ FT/S}^2, GCT = 32 \text{ FT/S}^2 \).

- \( RL = 10000 \text{ FT, RT} = 10000 \text{ FT} \).

- \( ALN = 0 \text{ M/S}^2, ATN = 0 \text{ M/S}^2 \).

- **CASE 1**: \( VLN = 10 \text{ M/S}, VTN = 5 \text{ M/S} \).

  **CASE 2**: \( VLN = 10 \text{ M/S}, VTN = 0 \text{ M/S} \).

  **CASE 3**: \( VLN = 10 \text{ M/S}, VTN = 35 \text{ M/S} \).

  **CASE 4**: \( VLN = 15 \text{ M/S}, VTN = 16 \text{ M/S} \).

- **CASE 1**: \( XRN = 35 \text{ M} \).

  **CASE 2**: \( XRN = 8 \text{ M} \).
CASE 3: $XRN = 25 \text{ M.}$

CASE 4: $XRN = 5 \text{ M.}$
APPENDIX B: INDIVIDUAL VEHICLE MOTIONS

The point of departure for describing the movement of a vehicle along its path is to consider the motion of that vehicle as if it were completely unaffected by the movement of other vehicles. In effect, this is a means of treatment of the best possible performance of that vehicle, for the interference of other vehicles can only tend to slow the progress of the vehicle. Once this best performance can be treated, then reductions in speed caused by other vehicles and other types of interference can be taken into account.

1. Equations of Motion:

The basic principles which underlie most of the relationships governing free vehicle movement are Newton’s laws of motion, as described in Principia published in 1687. These laws are:

1. Every body continuous in its state of rest or of uniform motion in a straight path unless the application of a force compels a change in that state.

2. An(unbalanced) force causes a proportional rate of change of momentum that takes place in the direction in which the force is impressed.
3. For every action between two contiguous bodies, there is a reaction which is equal, opposite, and simultaneous.

The first law may regarded as a special case of the second; and the second is the one most crucial to transportation engineering. The second states:

\[ F = \frac{d}{dt}(mv) \]

Where the force \( F \) acts in the same direction and sense as \( v \).

If the mass remains constant, then:

\[ F = m \frac{dv}{dt} = ma \]

Where \( a \) is the rate of change of velocity, or the acceleration. The mass of a body is, of course, equal to its weight in a gravitational field \( T \) divided by the acceleration due to gravity, usually termed \( g \).

In fact, mass can be thought of as the ratio of the force acting on a body to its acceleration, since we have:

\[ m = \frac{F_1}{a_1} = \frac{F_2}{a_2} = \ldots = \frac{T}{g} \]
2. Velocity and Acceleration Definition

\[ \text{Figure B-1: Motion of a vehicle in a plane. (a) Location. (b) Location vector difference. (c) Velocity. (d) Velocity vector difference.} \]

Velocity and acceleration are terms in constant, almost everyday, usage in our highly mechanized society, so they often are used without
worrying about precise definitions. However, because the correct application of vehicle following depends upon an accurate understanding of these terms, they must be defined precisely. Figure B-1 represents the motion of a vehicle in a plane.

The location of the vehicle at any instant can be represented by the direction and magnitude of a vector, the vector \( s_i \) specifying the location at time \( t_1 \), \( s_2 \) the location at \( t_2 \), etc.

Average velocity between two points is defined as the vector difference in locations divided by the difference in times as below:

\[
v_{12} = \frac{s_2 - s_1}{t_2 - t_1}
\]

Often we are not interested in the average velocity between two locations or points in time, but rather in the instantaneous velocity. This is defined as:

\[
v_i = \lim_{t_i \to t} \frac{s_2 - s_1}{t_2 - t_1}
\]

The direction of this vector will be tangent to the path the vehicle is following. Thus the instantaneous velocity of a vehicle moving on a path is a vector, tangent to the path at the point considered, of magnitude equal to the rate of progression on the auto's path, but it tells us nothing about the direction of the motion.
In the derivative form, the above equation is:

\[ v = \frac{ds}{dt} \]

Acceleration is the rate of change of velocity with respect to time. But since velocity is a vector, so must acceleration be a vector which is defined as:

\[ a = \frac{dv}{dt} \]

3. **Forces Producing Motion in Transportation**

The movement of any vehicle along its path is determined by the forces acting on that vehicle, according to Newton's laws. By consciously applying forces, the direction and magnitude of which are controlled, the velocity and path of the vehicle is determined.

The forces acting on a vehicle can be understood by consideration of the forces shown in *Figure B-2*.
Figure B-2: Forces acting on a vehicle in motion

Acting parallel to the vertical axis (Z) is the weight of the vehicle, the force $T$, which, if not reacted to by an equal and opposite force, would cause the vehicle to fall. If the vehicle is moving, there is an inherent resistance of the vehicle to motion $R$, acting along the axis of the vehicle's path, which also must be overcome. In addition, if the vehicle is
accelerating, then a net force, \( \frac{T \cdot a}{g} \), \( a \) being the acceleration and \( g \) the gravitational constant, must be the resultant force along its axis. Finally, the vehicle may be changing direction, in which case the centrifugal force \( C \) must be overcome.

The forces to overcome these resistance to the desired motion come from a combination of the propulsive unit, gravity, and the reaction of the surface (or medium) on which the vehicle is moving. The propulsive force \( M \) acts along the vehicle's path as shown. The reaction force of the supporting medium can include the propulsive forces, as in the case of wheeled vehicles (where the propulsive force is created by wheel to path friction), but after elimination of this component, the resulting support force \( S \) acts in a plane perpendicular to the vehicle axis as shown, and will include a component to overcome centrifugal force if the vehicle is turning. Notice that when the vehicle is changing elevation, the weight of the vehicle will have a component along the vehicle's axis, and therefore it will either aid or hinder motion. Thus there are many forces which must act on a transport vehicle in order that it move in the manner desired.

3.1. Inherent Resistance

3.1.1. Single Vehicle

All vehicles traveling on or in the earth or its atmosphere possesses an inherent resistance to motion, by which is meant the resistance to motion on a straight, level path at a constant speed. This resistance must be overcome if the vehicle is to maintain that speed, for otherwise this
inherent resistance would cause the vehicle to decelerate, ultimately to a stop.

*Table B-1:* Presents the inherent resistance formulas for wheeled road vehicles – automobiles, trucks, and buses:

### Automobiles and buses (Taborek, 1957)

\[ R = 0.01T + 0.0001TV + 0.0026CAV^2 \]

*where R = resistance, lb*

\[ T = \text{weight, lb} \]

\[ V = \text{velocity, mi/h} \]

\[ C = \text{air resistance parameter} \]

- Auto, \( C = 0.40 \) to 0.50
- Convertible auto, \( C = 0.60 \) to 0.65
- Bus, \( C = 0.60 \) to 0.70

\[ A = \text{maximum cross section area, } ft^2 \]

### Trucks (Society of Automotive Engineers, 1974)

\[ R = 7.6T + 0.097V + 0.002AV^2 \]

*where \( R = \text{resistance, lb} \)*

\[ T = \text{weight, 1000 lb or kips} \]

\[ V = \text{velocity, mi/h} \]

\[ A = \text{adjusted maximum cross section area, } ft^2 \]

\[ = \text{width times height less } \frac{3}{4} \text{ in or 0.0626 ft} \]

In general, this resistance for any given vehicle type and transport technology is a function of its weight and its velocity. It is important to consider weight variations primarily because the load being carried on a vehicle will vary from one assignment to another, and a rapid means of estimating resistance for each load is desired. Also, there is a reduction in
the weight of a vehicle which carries its own fuel as a trip progresses, and this often must be taken into account. The inherent resistance will be termed \( R(T, V) \) where the parentheses containing the total vehicle weight \( T \) (including fuel, load, etc.) and the velocity \( V \) indicates that the resistance depends on these two characteristics. The function \( R \), of course, will depend upon the particular vehicle type (size, type, etc.) and path.

The resistance of vehicles are estimated by the use of formulas which have been estimated from numerous tests of actual equipment in conjunction with theories as to the sources and nature of the frictional resistance and the flow of the air around the vehicle.

3.1.2 Platoon Vehicle

Several general conclusions are made when studying the drag coefficient of platoon vehicles.

1. The drag coefficient ratio for the lead vehicle and for each succeeding vehicle is independent of the number of vehicles in the platoon.
2. Each vehicle added to the platoon experiences a lower drag over most of the strong and weak interaction range, but may be subject to rather sharp, local drag changes (flowfield resonance). The final vehicle in the platoon experiences the least drag variation as vehicle spacing varies.
An overall measure of platoon drag performance may be obtained by defining an average drag coefficient ratio.

\[
(C_{D_p})_{\text{avg}} / C_{D_{avg}} = \left( \frac{1}{n} \right) \sum_{i=1}^{n} \left( C_{D_i} / C_{D_{avg}} \right)
\]

Where \(i\) is the \(i^{th}\) vehicle and \(n\) is the total vehicle in a platoon.

![Average Drag Coefficient for n-Vehicle Platoon](image)

Figure B-3: Average Coefficient

*Figure B-3* expressed platoon-averaged drag coefficient ratio as a function of separation distance and number of vehicles. Three separation distance are studied: 0.75 m, 1m and 1.5 m. The results suggest the importance of platoons having non-uniform spaces when operating with vehicles of different shapes. Since all platooning vehicles are to be equipped with on-board computers and engine performance sensors, it
would be a relatively simple additional task to optimize spacing to achieve a local minimum power expenditure (minimum drag).

3.2. Gradients

Vehicles of all modes undergo changes in elevation above sea level. In almost all applications the gradient is small enough for us to make an approximation:

\[ F_g = T \frac{P}{100} \]

where \( F_g \) = gradient force

\( T \) = weight of vehicle

\( P \) = percent grade

This approximation is very good for small gradients, such as main road grades of less than 5 percent.

3.3. Curvature Resistance

All vehicles must traverse curves or change direction. There are two important types of curves in the paths of vehicles. Horizontal curves are curves in the horizontal plane and do not result in any change in vehicle elevation. Vertical curves are in the vertical plane and result in a change in
elevation. Often the two are combined in actual curves in the path followed by a vehicle, although they can exist independently.

In the simplest case, the introduction of the curvature or change of direction results in a change, usually an increase, in the total resistance which must be overcome. This change in resistance is due to the centrifugal force which, provided the vehicle is traveling at constant speed on a path of constant radius, is equal to:

\[ F_c = \frac{T V^2}{g r} \]

where \( F_c \) = centrifugal force
\( r \) = radius of curvature
\( V \) = vehicle speed
\( g \) = gravitational constant

3.4. Propulsion Forces

There are almost as many means of giving locomotion to vehicles as there are different means of giving mobility to them. Forces are required in all transport technologies to overcome the resistance forces described previously, which include inherent resistance, forces required to cause the vehicle to follow the desired path, and forces required to cause the vehicle to accelerate and decelerate as desired.
In order to study the propulsion forces acting on a vehicle it is necessary to understand certain physical terms which are closely related to the forces referred to in Newton's laws that lie at the foundation of the analysis of vehicle motion. Work is defined as the product of a force acting over a distance, specifically:

\[ dW = F \cdot ds \cdot \cos \theta \]

where \( dW \) = differential unit of work

\( F \) = force

\( ds \) = differential displacement

\( \theta \) = angle between force and direction of displacement

Power is the rate of performing work, namely:

\[ P = \frac{dW}{dt} \]

At any point along the path followed by a force, then, the power is simply:

\[ P = F \cdot V \cdot \cos \theta \]

where \( V = \frac{ds}{dt} \)
Often in transportation the force and velocity are constant over long distances, and the force is directed along the path, resulting in constant power $FV$.

In order to easily remember the above relationship, some treatment of units is necessary:

$$P = \frac{1}{375} FV$$

where $P =$ power, hp  
$F =$ force, lb  
$V =$ speed, mi/hr

The basic relationship which is central to the propulsive aspects of vehicle performance is the relationship between the maximum available propulsive force and the speed of the vehicle. The reason for the connection with the vehicle’s speed is partly that the speed is important in the resistance which must be overcome by the propulsive forces and partly that for many types of vehicles the maximum propulsive force available is significantly influenced by the speed. The maximum force available for propulsion will also in general vary with various environmental conditions, such as the pressure and temperature of the air, possibly wind direction and velocity, and whether or not it is raining or snowing.
Most motor vehicles are propelled by a combination of a diesel or gasoline engine and a mechanical or hydraulic transmission, with the propulsion force being applied to the vehicle through the wheel-road interface. The results in a propulsive force versus speed of vehicle relationship are shown in Figure B-4:

Propulsion force at wheels

![Diagram showing propulsion force at wheels with vehicle speed on the x-axis and propulsive force on the y-axis, with transmission gear ratios labeled 1, 2, 3.]

*Figure B-4: Gasoline or diesel powered road vehicle propulsive force diagram*

The typical values for the available adhesion between the wheel and the road are shown in Table B-2:
Table B-2: Maximum Coefficients of Adhesion (Friction) between Rubber Tires and Roads:

<table>
<thead>
<tr>
<th>Road surface condition</th>
<th>Max. Rolling</th>
<th>Coefficient Sliding</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth concrete</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dry</td>
<td>0.75</td>
<td>0.64</td>
</tr>
<tr>
<td>Wet</td>
<td>0.60</td>
<td>0.50</td>
</tr>
<tr>
<td>Wet and oily</td>
<td>0.50</td>
<td>0.25</td>
</tr>
<tr>
<td>Asphalt</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dry</td>
<td>0.70</td>
<td>0.60</td>
</tr>
<tr>
<td>Wet</td>
<td>0.56</td>
<td>0.31</td>
</tr>
<tr>
<td>Snow</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hard-packed</td>
<td></td>
<td>0.20</td>
</tr>
<tr>
<td>With chains</td>
<td></td>
<td>0.30</td>
</tr>
<tr>
<td>Ice</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dry</td>
<td>0.14</td>
<td>0.10</td>
</tr>
<tr>
<td>With chains</td>
<td>0.25</td>
<td></td>
</tr>
<tr>
<td>Wet</td>
<td>0.12</td>
<td>0.06</td>
</tr>
</tbody>
</table>

Source: Fitch (1976, p.65).

These high values of coefficients of adhesion under normal circumstances, in conjunction with the actual retarding power of the brakes typically found in motor vehicles, result in extremely high possible deceleration rates. At this juncture, it is appropriate to point out the limits upon the maximum deceleration rate that passengers can withstand before becoming uncomfortable or actually being injured. In a series of tests conducted in the early 1930s, in conjunction with the development of a more advanced streetcar (Electric Railway Presidents’ Conference
Committee, 1975 reprint), it was found that seated passengers could remain comfortable at deceleration and acceleration rates up to 9 \text{ ft/s}^2 and a rate of change of acceleration up to 10 \text{ ft/s}^3. These values are still widely accepted as limits in the design of land transport vehicles.

4. Generalized Vehicle Performance Relationships

Road vehicle performance relationships has been developed which relate important vehicle performance characteristics to vehicle design, loading, and path characteristics.

![Acceleration and deceleration performance curves of road vehicles](image)

Figure B-5: Acceleration and deceleration performance curves of road vehicles:

- **Passenger Cars**, curve 1
5. WORK, ENERGY, AND FUEL CONSUMPTION

An increasingly important aspect of transport technologies is their consumption of fuel. As a matter of fact, in many developed nations transportation of all sorts consumes about one-fifth to one-quarter of the nation’s entire fuel consumption. With the realization that the supply of fuel is not inexhaustable, fuel conservation is becoming increasingly important.

Fuels are used in transportation primarily to overcome the various resistance to motion which are encountered: inherent, curvature, and gradient.

5.1. The Work Method

In many transportation technologies, as well as in many other fields in which fuel is used by machines to create forces, it has been observed that the fuel consumed by such machines is approximately proportional to
the total work performed by those machines. The basic relationship is the following:

\[ z = W \cdot r \]

where \( z = \text{fuel consumed} \)

\( W = \text{work performed} \)

\( r = \text{fuel rate} \)

If we specify the variation of the propulsive force at point \( x \) along a path as \( M_p(x) \), then the total propulsive work in traveling from \( d_m \) to \( d_n \) would be:

\[ W = \int_{d_m}^{d_n} M_p(x) \, dx \]

where \( M_p(x) = \text{fuel-consumption-related propulsive force at point } x \).

In practice, it would be very difficult to use the formula above for the estimation of propulsive work. A simplified formula for estimating propulsive work from a point \( d_m \) to a point \( d_n \) will be:

\[ W = \sum_{i=m}^{n} M_{p,i} (d_{i+1} - d_i) \]

where \( m \) and \( n \) are specified as the locations of interest.
It is extremely important that the force used in the above equation is the force created by the propulsion unit solely as a result of consumption of fuel. Thus, for example, the gradient force on a downgrade would not be considered part of this force. And the same case for the braking force.

The fuel rates for the diesel trucks are 0.40-0.50 lb/hp-h and for the gasoline autos are 0.003-0.005 gal/mi-lb (gasoline: 1 gal = 6.0 lb; diesel fuel: 1 gal = 7.1 lb)

5.2. Work and Energy

The concept of kinetic energy can be explained by considering the acceleration of a vehicle on a level path from rest. If the total resistance to motion of the vehicle is $R(v)$, then the total work performed on the vehicle in acceleration from speed $v_0$ at location $d_0$ and time $t_0$ to speed $v_1$ at time $t_1$ and location $d_1$ will be:

$$W = \int_{d_0}^{d_1} [R(v(x)) + T \frac{a(x)}{g}] dx$$

where $v(x) =$ velocity at each point $x$ on path

$$a(x) =$ acceleration at each point $x$ on path $= \frac{dv(x)}{dt}$
This work is composed of two parts, overcoming resistance and imparting acceleration. Concentrating on the latter only, and terming this work $K$, we have:

$$K = \int_{d_0}^{d_1} T \frac{dv(x)}{dt} \, dx = \int_{d_0}^{d_1} T \frac{v(x) \, dv}{g}$$

$$= \frac{1}{2} \frac{T}{g} (v_f^2 - v_o^2)$$

Table B-3: Some Characteristics of Typical Vehicles:

<table>
<thead>
<tr>
<th>Type</th>
<th>Weight, lb</th>
<th>Power, hp</th>
<th>Length, ft</th>
<th>Cross section, ft$^2$</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Small auto</td>
<td>3,000</td>
<td>100</td>
<td>14</td>
<td>19</td>
<td>4 seats</td>
</tr>
<tr>
<td>Large auto</td>
<td>4,000</td>
<td>200</td>
<td>18</td>
<td>26</td>
<td>6-9 seats</td>
</tr>
<tr>
<td>Bus</td>
<td>38,000</td>
<td>300</td>
<td>40</td>
<td>104</td>
<td>38-53 seats</td>
</tr>
<tr>
<td>Small truck</td>
<td>4,000</td>
<td>150</td>
<td>17</td>
<td>33</td>
<td>1 2 ton capacity</td>
</tr>
<tr>
<td>Medium truck</td>
<td>7,000</td>
<td>170</td>
<td>22</td>
<td>45</td>
<td>4 5 ton capacity</td>
</tr>
</tbody>
</table>
Table B-4: Design Vehicle Dimensions:

<table>
<thead>
<tr>
<th>Design Vehicle type</th>
<th>Overall</th>
<th>Overhang</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Height</td>
<td>Width</td>
</tr>
<tr>
<td>Passenger Car</td>
<td>4.5</td>
<td>7</td>
</tr>
<tr>
<td>Single Unit Truck</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Single Unit Bus</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Articulated Bus</td>
<td>10.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Combination Trucks</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Intermediate Semitrailer</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Large Semitrailer</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Interstate Semitrailer</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Triple Semitrailer</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Turnpike Double</td>
<td>13.5</td>
<td>8.5</td>
</tr>
<tr>
<td>Recreation Vehicle</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Motor Home</td>
<td>8</td>
<td>30</td>
</tr>
<tr>
<td>Car and Camper Trailer</td>
<td>8</td>
<td>49</td>
</tr>
<tr>
<td>Car and Boat Trailer</td>
<td>8</td>
<td>42</td>
</tr>
</tbody>
</table>

(Source: A Policy on Geometric Design of Highways and Streets, p21)
APPENDIX C: DYNAMO SOURCE CODE

************************************************************************************
NOTE
NOTE AUTOMATIC VEHICLE HEADWAY CONTROL MODEL
NOTE BY: MINGDONG YAO, MARCH, 1996
NOTE
************************************************************************************
NOTE
NOTE THIS IS A SIMULATION OF ONE CAR FOLLOWING
NOTE ANOTHER
NOTE LEVEL VARIABLES ARE: ACCELERATION, VELOCITY AND
NOTE DISTANCE
NOTE RATE VARIABLES ARE:
NOTE ALR, ACOM, JRLD, VLR, JRTD, ATR, VTR.
NOTE
************************************************************************************
NOTE
NOTE INSURE NOT NEGATIVE
MACRO POS(X)
  A POS.K=MAX(X.K,0)
MEND
NOTE
NOTE INSURE JERK WITHIN COMFORTABLE LIMIT
MACRO JLMT(X)
  A JLMT.K=MIN(X.K,2.5)
MEND
NOTE
NOTE INSURE ACCELERATION WITHIN COMFORTABLE LIMIT
MACRO ALMT(X)
  A ALMT.K=MIN(X.K,2)
MEND
NOTE
NOTE MAXIMUM ACCELERATION (LEAD VEHICLE)
  A ALMAX.K=3*FAL.K/ML
NOTE ALMAX = ACCEL. OF LEAD VEHICLE AT MAX. (M/S/S)
NOTE  FAL -- FORCE OF ACCELERATION, LEAD.  (LB)
NOTE  ML -- MASS, LEAD VEHICLE  (LB)
C  ML=3000
A  FAL.K=FPL.K-FRL.K
NOTE  FPL -- FORCE OF PROPULSION, LEAD.  (LB)
NOTE  FRL -- FORCE OF RESISTANCE, LEAD.  (LB)
A  FPL.K=375*PL/VEL.K
NOTE  PL -- POWER, LEAD VEHICLE  (HP)
C  PL=110
A  FRL.K=FRGL.K+FRIL.K+FRCL.K
NOTE  FRGL -- FORCE OF RESIS. FORM GRADIENT, LEAD  (LB)
NOTE  FRIL -- FORCE OF RESIS. FROM INHERENT, LEAD  (LB)
NOTE  FRCL -- FORCE OF RESIS. FROM CURVE, LEAD  (LB)
A  FRGL.K=ML*GL/100
NOTE  GL -- GRADIENT, LEAD VEHICLE
C  GL=1
A  FRIL.K=0.01*ML+0.0001*ML*VEL.K+0.0026*CL*SSL*VEL.K^2
NOTE  CL -- AIR RESISTANCE PARAMETER, LEAD VEHICLE
NOTE  SSL -- CROSS SECTION AREA, LEAD VEHICLE  (FT^2)
C  CL=0.4
C  SSL=19
A  FRCL.K=ML*VEL.K^2/(GCL*RL)
NOTE  GCL -- GRAVITATIONAL CONSTANT, LEAD VEHICLE
NOTE  RL -- RADIUS OF CURVATURE, LEAD VEHICLE  (FT)
C  GCL=32
C  RL=10000
NOTE  MAXIMUM ACCELERATION(TRAiL VEHICLE)
A  ATMAX.K=3*FAT.K/MT
NOTE  ATMAX -- ACCEL. OF TRAIL VEHICLE AT MAX.  (M/S/S)
NOTE  MT -- MASS, TRAIL VEHICLE  (LB)
C  MT=50000
A  FAT.K=FPT.K-FRT.K
NOTE  FPT -- FORCE OF PROPULSION, TRAIL.  (LB)
NOTE  FRT -- FORCE OF RESISTANCE, TRAIL  (LB)
A  FPT.K=375*PT/VT.K
NOTE  PT -- POWER, TRAIL VEHICLE  (HP)
C  PT=350
A  FRT.K=FRGT.K+FRIT.K+FRCT.K
NOTE  FRGT -- FORCE OF RESIS. FROM GRADIENT, TRAIL  (LB)
NOTE  FRIT -- FORCE OF RESIS. FROM INHERENT, TRAIL  (LB)
NOTE  FRCT -- FORCE OF RESIS. FROM CURVE, TRAIL  (LB)
A  FRGT.K=MT*GT/100
NOTE  GT -- GRADIENT, TRAIL VEHICLE
C  GT=1
A  FRTT.K=0.01*MT+0.0001*MT*VT.K+0.0026*CT*SST*VT.K^2
NOTE  CT -- AIR RESISTANCE PARAMETER, TRAIL VEHICLE
NOTE  SST -- CROSS SECTION AREA, TRAIL.  (FT^2)
C  CT=0.4
C  SST=19
A  FRCT.K=MT*VT.K^2/(GCT*RT)
NOTE  GCT -- GRAVITATIONAL CONSTANT, TRAIL VEHICLE
NOTE  RT -- RADIUS OF CURVATURE, TRAIL VEHICLE  (FT)
C  GCT=32
C  RT=10000
NOTE
NOTE
******************************************************************************************************************************************
NOTE  LEAD VEHICLE
NOTE
******************************************************************************************************************************************
NOTE  L  AL.K=AL.J+(DT)(JRLD.JK)
NOTE  AL--ACCELERATION OF THE LEAD VEHICLE  (M/S/S)
NOTE  JRLD -- JERK OF LEAD VEHICLE DELAYED  (M/S/S/S)
NOTE  ALN--INITIAL ACCELERATION OF THE LEAD VEHICLE
C  ALN=0
R  JRLD.KL=JRLI.K
NOTE  JRLI -- JERK OF LEAD VEHICLE INITIAL  (M/S/S/S)
A  JRLI.K=CONS
C  CONS=0
NOTE  C  JRDL=0
NOTE  L  VL.K=VL.J+(DT)(ALR.JK)
NOTE  VL -- VELOCITY OF LEAD VEHICLE  (M/S)
NOTE  ALR -- ACCEL. OF LEAD VEHICLE RELATIVE  (M/S/S)
R  ALR.KL=MIN(ALMAX.K,AL.K)
N  VL=VLN
C  VLN=30
NOTE  VLN -- INITIAL VELOCITY OF LEAD VEHICLE  (M/S)
NOTE
L XL.K=XL.J+(DT)(VLR.JK)
NOTE XL -- POSITION OF LEAD VEHICLE (S)
NOTE VLR -- VELOCITY OF LEAD VEHICLE RELATIVE (M/S)
R VLR.KL=VLR.K
N XL=XLN
NOTE XLN -- INITIAL POSITION OF LEAD VEHICLE (S)
C XLN=100
NOTE
NOTE
******************************************************************************

NOTE TRAIL VEHICLE
NOTE
******************************************************************************

NOTE
L AT.K=AT.J+(DT)(JRTD.JK)
NOTE AT -- ACCELERATION OF THE TRAIL VEHICLE
NOTE JRTD -- JERK OF TRAIL VEHICLE DELAYED (M/S/S)
N ATN=ATN
NOTE ATN--INITIAL ACCEL. OF THE TRAIL VEHICLE (M/S)
C ATN=0
NOTE
R JRTD.KL=DELAY3(JRTI,K,JRTD)
NOTE JRTI -- TRAIL VEHICLE JERK INITIAL (M/S/S)
NOTE JRD--JERK DELAY TIME (S)
C JRTD'=0.2
A JRTI.K=JLMT(JRTID.K+Ka*(ATD.K-AT.K)+Kv*(VTD.K-VT.K))
NOTE JRTID -- JERK OF TRAIL INITIAL DESIRED (M/S/S)
NOTE ATD -- ACCEL. OF TRAIL VEHICLE DESIRED (M/S)
NOTE VTD -- VELOCITY OF TRAIL VEHICLE DESIRED (M/S)
NOTE Ka, Kv -- VEHICLE GAIN
C Ka=8
C Kv=16
NOTE
NOTE GR1--GROUP 1 JERK
NOTE GR2--GROUP 2 JERK
NOTE XR--RELATIVE SPACE
A XR.K=XL.K-XT.K
NOTE

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NOTE  JRTI1--TRAIL VEHICLE JERK INITIAL: ACCELERATION PART
NOTE  JRTI2--TRAIL VEHICLE JERK INITIAL: SAFE1 PART
NOTE  JRTI3--TRAIL VEHICLE INITIAL: SAFE2 PART
NOTE  JRTI4--TRAIL VEHICLE INITIAL: DECELERATION PART
NOTE
A  JRTI1.K=MIN(Ka*(ACOM-AT.K),JCOM)
C  ACOM=2
NOTE
NOTE
A  EQU1.K=VL.K*JRLD.JK
A  EQU2.K=AL.K*(AL.K+AMIN)
A  EQU3.K=AMIN*AT.K
A  EQU4.K=SQR(VL.K*VL.K+VAOL*VAOL+2*AMIN*XRPP.K)
A  EQU5.K=(AL.K*VL.K+AMIN*(VL.K-VT.K))
A  EQU6.K=SQR(VL.K*VL.K+VAOL*VAOL+2*AMIN*XRPP.K)
NOTE  AMIN = MAXIMUM DECELERATION (M/S/S)
NOTE  VAOL = ALLOWABLE COLLISION VELOCITY (M/S)
C  AMIN=5
C  VAOL=3
NOTE
A  JRTI3.K=JLMT(JRLD.JK)
NOTE
NOTE
A  EQU10.K=SQR(EQU1.K)
A  EQU8.K=ACOM*ACOM*ACOM*ACOM/(4*JRCOM*JRCOM)
A  EQU9.K=2*ACOM*(XR.K-
XR+ACOM*ACOM*ACOM/(6*JRCOM*JRCOM))
A  EQU11.K=ACOM*(AL.K-AT.K)
C  JCOM=2.5
C  XRD=1

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NOTE
A ATD.K=CLIP(GRA.K,GRB.K,XR.K,30)
NOTE GRA--GROUP A ACCELERATION
NOTE GRB--GROUP B ACCELERATION
NOTE XR--RELATIVE SPACE
NOTE
NOTE
A ATD1.K=ACOM
NOTE
A ATD2.K=(VL.K*AL.K+AMIN*(VL.K-VT.K))/SQR(1.0+(VL.K*VL.K+VAOL*VAOL+2*AMIN*XRPP.K))
A XRPP.K=MAX(XR.K,-XR.K)
NOTE XRPP -- POSITIVE RELATIVE DISTANCE
A ATD3.K=ALMT(AL.K)
NOTE
NOTE
NOTE
A VTD.K=CLIP(GRI.K,GRII.K,XR.K,30)
NOTE GRI--GROUP I VELOCITY
NOTE GRII--GROUP II VELOCITY
NOTE
A VTD1.K=MIN(ACOM*SQR(2*XRPP.K/ACOM),80)
NOTE
A XRPP.K=MAX(XRN-XR.K,XR.K-XRN)
NOTE XRPP -- POSITIVE DISTANCE
C XRN=100
NOTE
A VTD2.K=MIN(AMIN*
X JRDT+SQR(VL.K*VL.K+VAOL*VAOL+2*AMIN*XRPP.K),80)
NOTE
A VTD3.K=MIN(-AMIN*JRDT+VL.K+VAOL,80)
NOTE
NOTE
NOTE
NOTE
L  VT.K=VT.J+(DT)(ATR.JK)
NOTE  VT -- VELOCITY OF TRAIL VEHICLE  (M/S)
NOTE  ATR -- ACCEL. OF TRAIL VEHICLE RELATIVE  (M/S/S)
R  ATR.KL=MIN(ATMAX.K,AT.K)
N  VT=VTN
NOTE  VTN -- INITIAL VELOCITY OF TRAIL VEHICLE  (M/S)
C  VTN=25
NOTE
L  XT.K=XT.J+(DT)(VTR.JK)
NOTE  XT -- POSITION OF TRAIL VEHICLE  (M)
NOTE  VTR -- VELOCITY OF TRAIL VEHICLE RELATIVE  (M/S)
R  VTR.KL=VT.K
N  XT=XTN
NOTE  XTN -- INITIAL POSITION OF TRAIL VEHICLE
C  XTN=0
NOTE
NOTE
NOTE
SPEC  DT=0.05/LENGTH=70/PLTPER=1/PRTPER=1
PLOT  JRLD=A, JRTD=1
PLOT  AL=B, AT=2
PLOT  VL=C, VT=3
PLOT  XL=D, XT=4
PRINT  AL, AT
PRINT  VL,VT
PRINT  XL, XT
PRINT  XR
PLOT  XR=X
RUN
## APPENDIX D: VEHICLE EXAMPLE

<table>
<thead>
<tr>
<th>BMW UK Model Range 1995</th>
<th>Comacts</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>316i</strong></td>
<td><strong>318ti</strong></td>
</tr>
<tr>
<td>Price</td>
<td>£13,350</td>
</tr>
<tr>
<td>Engine</td>
<td>4 Cylinder 8 valve</td>
</tr>
<tr>
<td>Capacity</td>
<td>1596 cc</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>9.7:1</td>
</tr>
<tr>
<td>Power Output</td>
<td>102 bhp</td>
</tr>
<tr>
<td>Trans. Manual</td>
<td>5 Speed</td>
</tr>
<tr>
<td>Switchable Auto</td>
<td>Yes</td>
</tr>
<tr>
<td>Dimensions Length</td>
<td>4210mm / 165.8in</td>
</tr>
<tr>
<td>Width</td>
<td>1698mm / 66.8in</td>
</tr>
<tr>
<td>Fuel Tank Capacity</td>
<td>52 litres / 11.4 gals</td>
</tr>
<tr>
<td>Boot Capacity</td>
<td>300-1030 litres / 10.6-36.4 cu ft</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>------------------</td>
<td>-------</td>
</tr>
<tr>
<td>Wheels</td>
<td>6J x 15 Steel</td>
</tr>
<tr>
<td>Steel</td>
<td></td>
</tr>
<tr>
<td>Tyres</td>
<td>185/65 R15</td>
</tr>
<tr>
<td>Trailer Load Unbrak</td>
<td>600kg / 1320lb</td>
</tr>
<tr>
<td>Trailer Load Brak</td>
<td>1250kg / 2750lb</td>
</tr>
<tr>
<td>Performance</td>
<td>Man</td>
</tr>
<tr>
<td>0-62mph (secs)</td>
<td>12.3</td>
</tr>
<tr>
<td>Max Speed (mph)</td>
<td>117</td>
</tr>
<tr>
<td>Fuel Consumption</td>
<td></td>
</tr>
<tr>
<td>Urban mpg</td>
<td>31.0</td>
</tr>
<tr>
<td>56mph mpg</td>
<td>49.6</td>
</tr>
<tr>
<td>75mph mpg</td>
<td>37.7</td>
</tr>
<tr>
<td>Average mpg</td>
<td>38.2</td>
</tr>
</tbody>
</table>
VITA

Mr. Mingdong Yao was born on January 30, 1970 in Harbin City, China. He moved with his parents to Tianjin City in 1982. After his graduation from No. 40 High School of Tianjin City, he studied in the Electrical Engineering Department at Hunan University, Changsha, China, for 5 years and graduated in July, 1993 with a Bachelor's degree in Communication Engineering (with honor).

He was awarded the prizes for excellent student every year from Hunan University. He won the first prize for the designing of an automatic radio control car model. His graduation paper for the Bachelor's Degree was related to the application of image processing to automatic vehicle location and control. He became very interested in ITS (Intelligent Transportation System).

Before he came to Virginia Tech., he worked in NEC Corporation and Motorola Corporation for two years. He had the chance to learn the most advanced technologies and deal with the most advanced equipment in Communication industry. He chose Virginia Tech. to continue his Master's degree study because its Center for Transportation Research was named one of the excellence in 1993. His Master's research focused on Automated Vehicle Headway Control. He is capable of handling Communication and Computer related ITS projects.

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