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An Impedance Model Approach for Adaptive Cruise Control

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THESIS

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ABSTRACT

The objective of this thesis is to design an intelligent cruise controller for the motion control of a vehicle platoon and discuss the relationship between following distance and signal sampling frequency. For safe vehicle following, an Adaptive cruise control (ACC) law is considered in an impedance model based approach. The impedance model is developed to guide the dynamic behavior of the following vehicles. In order to assure the safety of the vehicle platoon, it is necessary to regulate the distance between vehicles at an appropriate value. The proposed control law uses relative speed and spacing as well as preceding acceleration information to choose a proper control action for maintaining a desired following distance. Simulations are carried out in Matlab®/Simulink workspace to verify the stability and performance for both multiple vehicles control and adaptive cruise control

CHAPTER 1: INTRODUCTION

While people are concerning about the transportation safety and capacity on the highway, the automated vehicle following has been a major way to reduce the potential safety issues and save labors. The modern automated vehicle control technology allows vehicles to travel in groups with a close and safe following distance. These applications lead to a reduction in the amount of space used by a number of vehicles on a highway.

1.1 Conventional Cruise Control

The Cruise Control (CC) in most nowadays automotive models is a system that automatically controls the speed of a vehicle. The system will maintain a target speed set by the driver by controlling the throttle of the car. In the automatic vehicle control, the purpose of cruise control (CC) is to maintain a desired speed when there is no vehicle ahead in a certain distance, thus a vehicle may stay in cruise control state if it has no vehicle immediately in front of it and has enough distance to a preceding vehicle. The controller for cruise control uses a feedback and feed-forward control law of the form:

$$a_d = v'_d - k(v - v_d) \quad (1-1)$$

Where a_d is the desired acceleration of the vehicle, v is the speed of the vehicle, v_d is the desired speed of the vehicle, and k is a gain.

However, if we need to create groups of vehicles travelling automatically in a close-distributed platoon, a Cruise Control Law is obviously not enough for this application.

1.2 Intelligent Cruise Control

An automatic vehicle control system designed to control the longitudinal velocity at a driver's set value as well as the speed of and the distance to a leading vehicle has several advantages including the safety, performance and efficiency on a highway. While the traditional cruise control system control the velocity at preset speed value, the Autonomous Intelligent Cruise Control (AICC), system usually uses information from the preceding and following vehicles to adjust the vehicle's velocity to that of the lead vehicle and keep it at a safe distance to the lead one .

1.3 Adaptive Cruise Control/Cooperative ACC

The adaptive cruise control (ACC) system, which is a general term meaning improved or advanced cruise control, is more and more used in modern vehicles. Some advanced control technologies including automatic braking or dynamic set-speed type controls are often utilized in these systems. Usually a radar or laser-based sensor is used to measure the following distance in order to make the vehicle to slow down when moving close to a preceding vehicle and accelerate again to the preset speed when traffic allows.

Usually during automatic vehicle following, the control objective is to maintain a desired spacing from its preceding vehicle as well as limit the speed under a maximum value, therefore it is a vehicle following problem that the vehicle control system will involve both speed control and distance control.

The purpose to introduce the ACC law is to regulate the range between vehicles to a preset value and to adjust the speed of the directed vehicle(s) to the speed of preceding vehicles. Also, if communications between vehicles is added to ACC, it may be called Cooperative Adaptive Cruise Control (CACC). Apparently, if a control system is designed to

work for ACC, it should also work for CACC because relative distance, speed and acceleration can be calculated based on the information passed by the other vehicle through communication. Actually, there are no big differences between the control laws for ACC and CACC, except the operating logic and communication link [1]. The design issues for the ACC or CACC control Law would be discussed in Chapter 2 and Chapter 3

1.4 Velocity Headway

As mentioned above, each follower vehicle is expected to maintain a safe distance to the vehicle in front of it for safe vehicle following purpose. The vehicle's performance and braking capabilities, road conditions, aerodynamics and sensor update frequency is needed to be considered to estimate the required safety distance [3]. To make this easy to calculate, a safe distance policy [3] is usually adopted for the follower vehicle which is given by:

$$R_d = h \cdot (v_d^2 - v^2) + \lambda \cdot v + \mu \quad (1-2)$$

Where v is the speed of the vehicle, v_d is the desired speed of the vehicle, and h ,

λ and μ are positive constants that rely on the braking capabilities and for the directed vehicles.

For a tight vehicle following maneuver, the speed of the follower vehicle is approximately equal to the desired speed. Therefore, the safety distance policy can be approximated by:

$$R_d = \lambda v + \mu \quad (1-3)$$

Which is called velocity dependent or headway and known as “constant time headway policy”[3].

If road slip is not severe, vehicle speed measurement can be based on wheel speed which is quite reliable for longitudinal control. The constant λ is a time constant which is used to ensure the directed vehicle has enough cushion time when the downstream traffic changes

1.5 Impedance Control

The spring-damper relation or so called ‘impedance’ relation is widely used to represent the interaction with uncertain environment. The impedance control approach for vehicle following couples all following vehicles and leads to a dynamical system that resemble a series of mass/spring/damper systems [4]. The impedance control for automatic vehicle applications often includes a system consisting of a series of mass/spring/dampers. This kind of system is dissipative, that is, the energy of the system decays to zero eventually (i.e., there is no velocity difference between masses, and distance between neighboring masses is equal to the original length of spring). Therefore, it is expected that if the vehicle controller is designed under this framework, the whole system tends to be stable and free of slinky-type effects.

1.6 Problems Statement

To summarize the first Chapter, it may be easy to just point out the major problems that need to be solved:

- 1). the control design tasks for ACC are to design a controller which determines desired net force or torque.
- 2). Design an impedance model that is able to simulate the functionality of the controller.

3). Verify the controller stability in the both continuous and digital control system

This article concentrates on a controller design. And at last we need to discuss the stability issues in continuous and digital system domain and verify the control law with simulation. Organization of this thesis is as follows; In Chapter.2, different papers on automatic vehicle control will be compared and discussed. The major methodology for the controller design will be presented in Chapter.3. In Chapter.4 and Chapter.5, the control design and simulation results will be presented. The last two Chapters will interpolate the results and make the concluding remarks.

CHAPTER 2: LITERITURE REVIEW

In this chapter, several papers and literatures will be discussed about typical design strategies about intelligent cruise control for single vehicle as well as multiple vehicles. The first literature will talk about the major intelligent cruise control applications, such as vehicle model, different levels for cruise control and switch strategies for those controllers. The next two papers will focus on different control strategies on this application. The controllers for intelligent cruise control can be divided into two parts: higher level and lower level controller. These two papers discussed those two different types of controller respectively. The last paper would cover a typical model for multiple vehicles control.

2.1 Intelligent Cruise-Control Applications

Different from the conventional cruise-control (CC) systems which control the speed at a desired vehicle speed, ACC systems accommodate their speed when there is another vehicle ahead on the roadway and follow the leader vehicle at a preset following distance using range detectors such as radar and/or lidar[1]. Otherwise, ACC will transit to conventional CC which means it returns to the preset speed when there is no vehicle just in front of it. In a typical intelligent Cruise Control system, the action of follower vehicle in demand is to apply cruise control (CC) if the road ahead is clear, if not it will follow the lead vehicle in a certain distance.

, Some modern vehicle models, such as the Nissan Q45 and FX45, the Mercedes Sclass, the Lexus 330 and 430, the Audi A8, and select Jaguar and Cadillac, already has available ACC systems installed [1]. The vehicles can use these ACC systems to obtain the information that needed for speed and distance control. In the industry, these controllers are usually composed by two different parts: lower level controller and higher level controller.

Figure 2.1 shows the typical composition of the system and connections of these two parts [1].

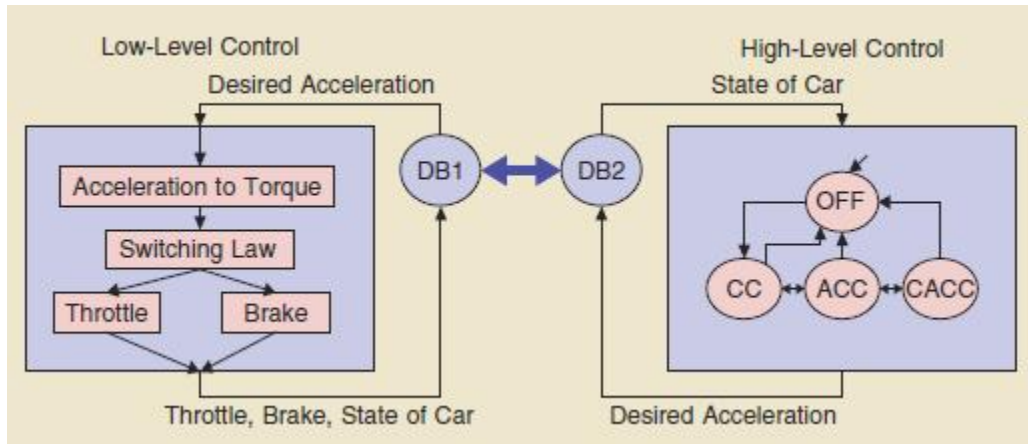


Figure 2.1 The Composition of Vehicle Control System in Modes [1]

Cruise Control (CC), Adaptive Cruise Control (ACC) and Cooperated ACC (CACC) are different states in the upper level controller. This controller will use only the information from the subject vehicle's line-of-sight sensors (i.e. radar or lidar) in ACC mode, and use information from the wireless communication link together with that from those sensors in CACC mode [1].

In order to calculate the torque required, the desired subject vehicle acceleration, which is generated by the upper controller, will be transmitted to the lower level controller. The desired torque will be the input for the switching law. The switching law will decide whether to apply brakes or throttle and in what amount depending on their capability.

The full vehicle model usually used for lower level controller development in practice is an 11- state model as shown in Figure 2.2 [12]. This model consists of the vehicle dynamics model, throttle and brake system dynamics, a two-state model for the spark ignition engine, and models of the torque converter, transmission, and wheel slip.

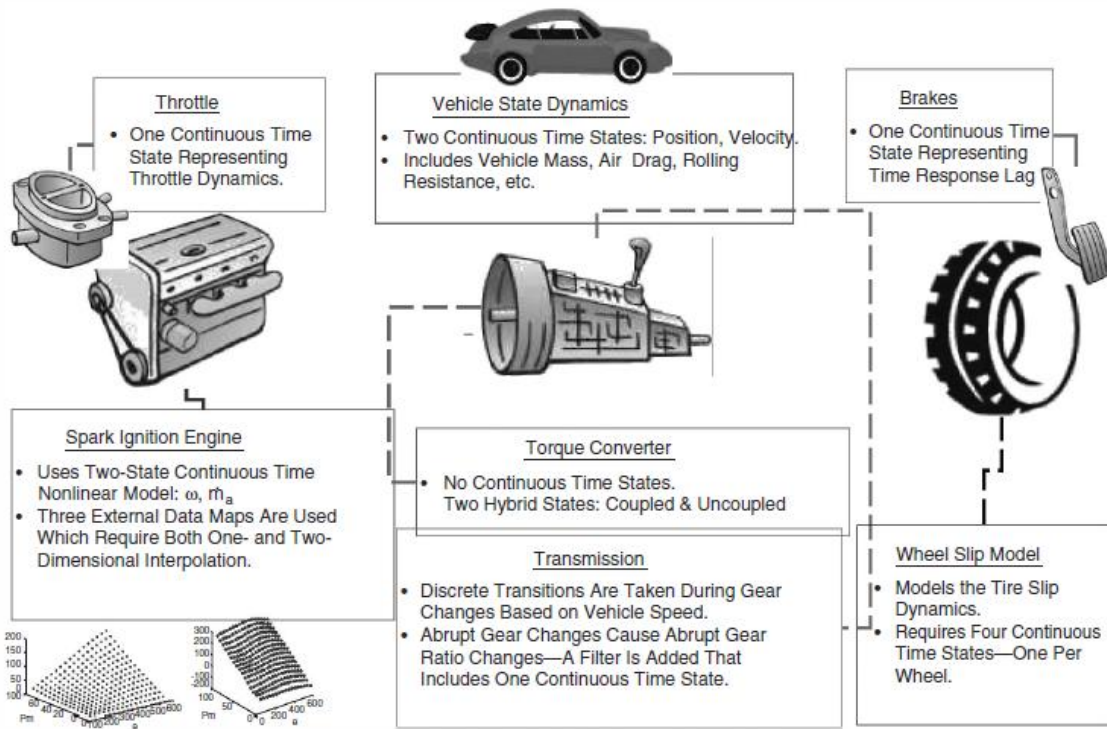


Figure 2.2 The Full Vehicle Model [12].

The vehicle speed can be sensed by a gyroscope or encoder installed on the wheel. From the speed and parameters for transmission and torque converter, the speed of engine can be obtained. Then, the desired throttle angle is computed using an engine map with regard to engine speed. The brake control is fulfilled by adjusting the master cylinder pressure.

The state of the car is usually governed by the following distance (x_d) and the subject vehicle speed (v). In a typical ACC system, some constants will be set in advance in order to apply transitions between different states. Figure 2.3 shows the state transition diagram of the MEMS based ACC system. This system behavior consists of four states, which are, 'Halt', 'Accelerate', 'Cruise' and 'Retard' [5]. Following distance is a system level input to the controller sensed by the radar, while the vehicle velocity is an inter-system input whose output is an input to the ACC system.

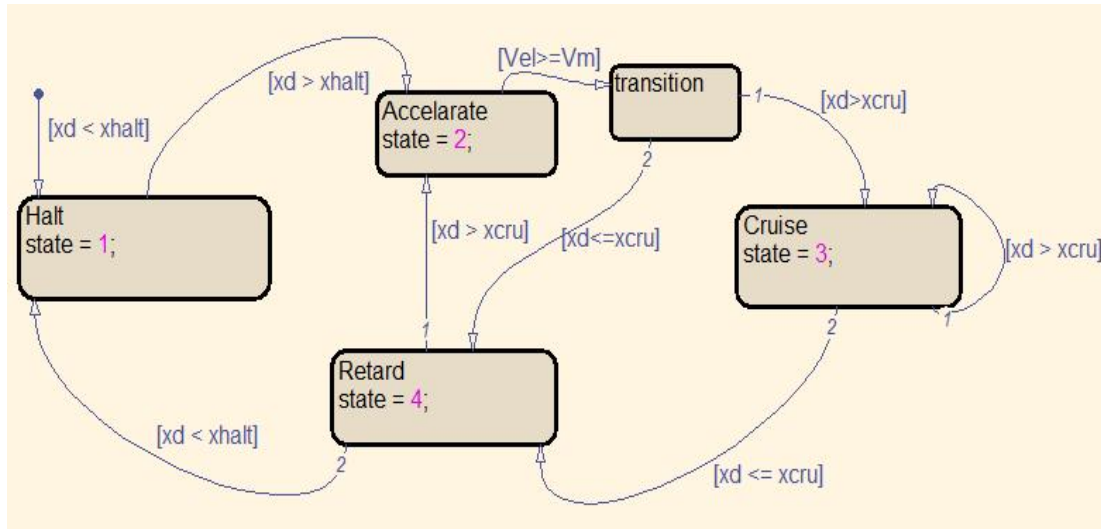


Figure 2.3 State Transition Model for ACC system in Simulink/Stateflow

In this Graph, x_{halt} indicates the following distance when the vehicle stops; x_{cru} means the following distance with regard to the vehicle speed; v_m stands for the maximum cruise speed (desired speed) for the subject vehicle. The switching laws would make sure that the controller outputs the corresponding state of the car according to the sensed variables and parameters.

The use of these sensors used for sensing the following distance and closing rate requires heavy filtering, because they are normally subject to noise, update frequency and drop-outs, and this, in turn, introduces delays into the system and reduce the ability of the ACC system. However, in cooperative ACC (CACC) systems, because the forward-looking sensor is assisted by a wireless communication link, it will offer real time leader-to-follower updates of critical information.

2.2 Control Strategies on Lower and Upper Level Controller

Now that, the widely used control strategy for upper level controller is so-called ACC. However, for the lower level controller, the design is mainly based on the vehicle model and the assumptions, therefore, the lower control strategies varies with the vehicle models used for the control system.

2.2.1 Typical ACC Control Design Strategy

ACC has been a hot research topic in recent years in both academics and car manufacturers. Among these researches, Persson, M. et al. consider higher performance ACC which claimed suitable for both low and high speeds [9]. Liang C.-Y et al. consider the string stability when several ACC vehicles drive closely [10]. And Xiao-Yun Lu, et al. Propose some sliding-mode based designs of ACC which claimed have higher acceleration/deceleration capability, safety and suitability for both ACC and CACC [2]. These papers focus on different aspects. [10] mainly concerned about the vehicle behaviors on different speeds, while [2] discussed about the ACC strategy for multiple vehicles, however, [2] had a more comprehensive view for the ACC design. Particularly, [2] covered two different choices of sliding modes for ACC designs.

To design ACC using the sliding mode control scheme, the first thing is to select a surface or a manifold (i.e., the sliding mode) such that the system trajectory exhibits desirable behavior when confined to this manifold. Then proper feedback gains needs to be decided so that the system input trajectory will just intersect and stay on the surface.

Due to the discontinuity of the sliding mode control law, the control system based on this law has the ability to direct trajectories or inputs to the sliding mode in limited time (i.e., stability of the sliding surface is better than asymptotic). However, when the trajectories

arrive at the sliding surface, the system will take over the role of the sliding mode. Therefore, selection of the sliding modes and reach-ability conditions is crucial for designs based on this scheme.

The design procedure with different sliding modes would be discussed in next Chapter. This part mainly discussed the general process adopted by [2] to design such a controller.

The first thing in this paper is to choose a sliding surface. A sliding surface was usually chosen on a plane with control parameter and derivative of the parameter as two coordinates. An example is shown on the Figure.2.4.

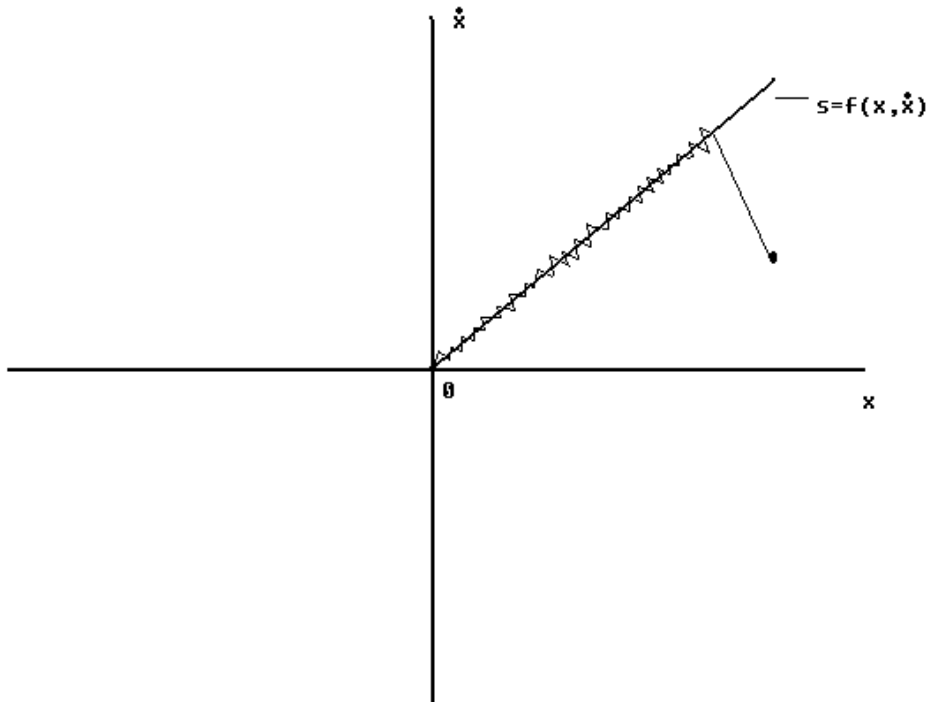


Figure 2.4 The Selection for Sliding Surface

In the Figure 2.4, a sliding surface s is chosen as a linear relation between x and \dot{x} (e.g. $s = k \cdot x + \dot{x}$). The ideal sliding mode is $s = 0$ which means the reduced order relationship

on the sliding manifold is $\dot{x} = -k \cdot x$. Any input trajectory on this plane can reach the sliding surface through the reach-ability condition which is usually defined as:

$$\dot{s} + \gamma \cdot s = 0 \quad (2-1)$$

However, since the term \dot{s} contains the second derivative of x , before solving the control, an error model which must include term \ddot{x} should be developed first. To illustrate this process, if term x indicates distance, then v means speed and a stands for acceleration. Thus, for example, the model can be written as:

$$\dot{x} = v \quad (2-2)$$

$$\dot{v} = a = \frac{1}{M}(F) \quad (2-3)$$

where M and F are the mass and force respectively. At last, after substituting the expression for \dot{s} and s , the force can be solved through the reach-ability condition.

For the sliding surface s , sliding reach-ability conditions, even if available for chosen surface, can not guarantee the closed-loop system stability [2]. The reason is that, if we are controlling the reference headway (see section 1.4),

$$R_d = \lambda v + \mu \quad (2-4)$$

in which v is a variable. In the stability considerations, only the reach-ability condition $s = 0$ in the ideal sliding mode does not guarantee the stability of the closed-loop system. Instead, one has to consider the reminder dynamics on the sliding manifold with unknown v taken into consideration. To achieve this, [2] suggested using a new pair of coordinates for the closed-loop system. Since the sliding mode control is not adopted in this thesis, the method

for stability analysis will not be discussed here. The indirect Lyapunov method for stability analysis can be found in section 3.2 of [2].

This paper [2] also discussed Implementation related issues which could affect the robust performance of the control system as well as the transition from ACC to CACC in a control design viewpoint. However, these discussions are not related to the main contents of this thesis.

2.2.2 The Lower Level Controller Configuration

On the other hand, the lower level controller often consists of a throttle controller, brake controller, and switching logic. The brake controller is used for deceleration that cannot be achieved by engine torque alone. If no brake is required for the vehicle, the throttle controller will take over to accelerate or decelerate the target vehicle. The role of the switching logic is to properly activate and deactivate the throttle and brake controllers based on the required net torque at the current state. When the computer continuously computes the required throttle angle, if the required throttle angle is greater than the minimum throttle angle, the logic determines that the throttle controller alone is adequate to generate the desired control action, and no brake torque will be applied [3]. Otherwise, the throttle controller will be deactivated which means it will keep the throttle angle at the minimum value and brake control will be applied to generate the enough brake torque.

2.3 Multiple Vehicles Control Design Strategy

For multiple vehicles control in a platoon, there is a potential disadvantage of the approach described above that the disturbances or errors may propagate both forward and

backward within a platoon. Therefore, the information from the follower vehicle should be considered in the control system

The multiple vehicles control scheme, i.e. The Vehicle Platoons control system, consists of a vehicle guidance model and individual vehicle controllers. Shladover and No. et al. proposed guidance models which using a global communication and a local communication respectively to transmit the velocity and acceleration signals to each follower vehicle [7, 8]. Zhang et al. developed a control algorithm for the vehicle platoon using the information of the leading and following vehicles [3].

[3] claimed that their approach guarantees individual vehicle stability as well as platoon stability under the constant spacing safety policy. This result leads to the following important conclusion: the design of the platoon stable vehicle follower controller under constant spacing policy is possible through the use of the relative speed and spacing information from both the controlled vehicle's immediate predecessor and follower.

Fig 2.4 shows the control strategy in [3]. They designed a controller for the subject vehicle using information from both the controlled vehicle's immediate predecessor and the controlled vehicle's immediate follower to achieve automatic vehicle following, where V_n , V_{n+1} and V_{n-1} are the velocities of vehicles n , $n+1$ and $n-1$, respectively,

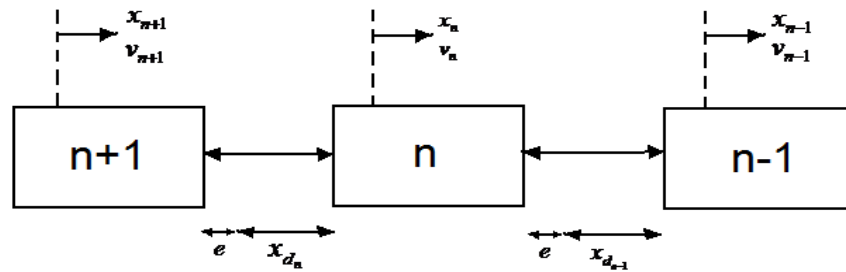


Figure 2.5 Configuration of Vehicle Following Maneuver

The individual vehicle controller should be coupled with the guidance model to execute the control signal. The vehicle control is accomplished by longitudinal and lateral control

using a driving and steering actuator respectively. For longitudinal control, various methods has been proposed such as sliding mode control with the Lyapunov function based approach [2] besides the conventional PID controller [6].

Other than these researches, a guidance model using the spring-damper relation (or say impedance relation [see Chapter 1.5]) is adopted in [4]. Impedance control does not attempt to track motion and force trajectories but rather to regulate the mechanical impedance specified in a certain guidance model. The local interaction between the lead and follower vehicles can be specified by spring-damper relations [4]. Therefore, the trajectory still has to be defined, because in impedance control, only the impedance parameters will be defined. So, indeed, the impedance control with conceptual serial spring-damper model is kind of force control scheme to generate the appropriate trajectory according to the force exerted from the environment and the impedance. Figure 2.5 shows an example for this impedance model in [4].

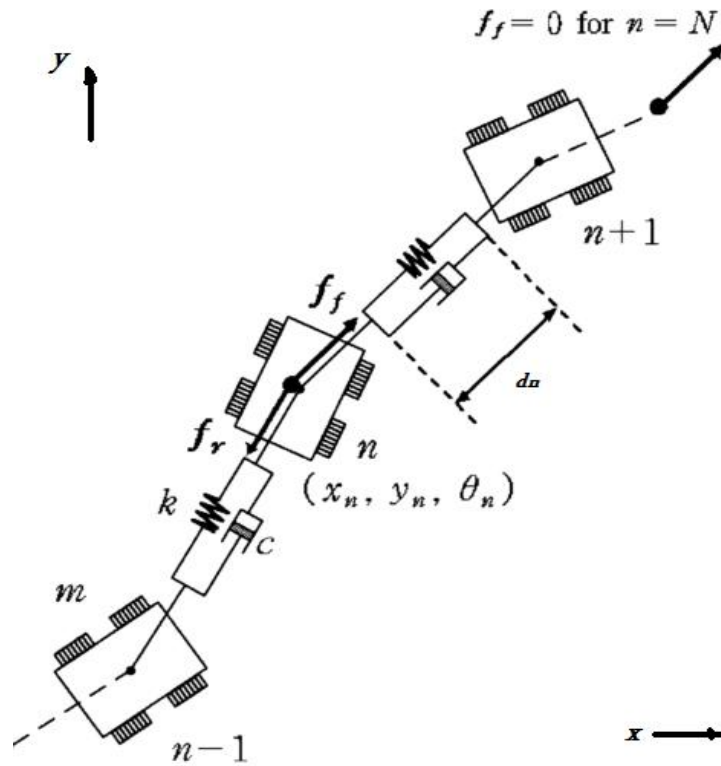


Figure 2.6 The Serial Spring-damper Model [4]

CHAPTER 3: VEHICLE MODEL AND CONTROL DESIGN

In this chapter, a simple vehicle model for control design will be first developed and the control law design will be followed

3.1 Vehicle Model

In order to implement the controller for the vehicle following purpose, a longitudinal vehicle model is necessary. The automotive power-train is usually composed by three segments: an engine, a transmission (including a torque converter), a drive train (including rubber tires), and all other components that can influence the longitudinal performance of an automobile. A simple functional description of such a system is shown in Figure 3.1[3]. This is typically a system consists of several subsystems with different inputs and outputs.

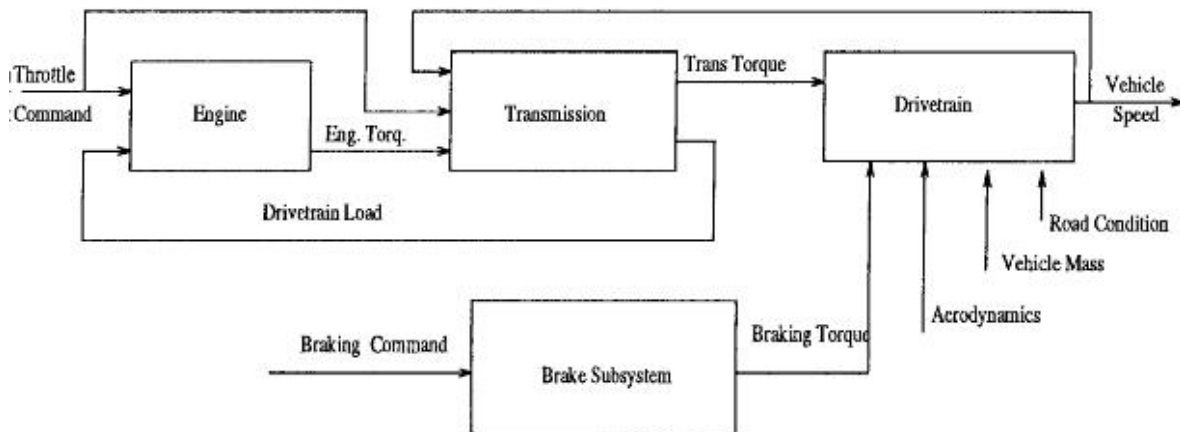


Figure 3.1 Longitudinal Vehicle Model [3]

For longitudinal control, the system in Fig 3.1 can be considered as a two-input (throttle angle and brake torque) and one outputs (vehicle speed) system[2]. The other inputs such as aerodynamic drag, road load, and vehicle mass are treated as disturbances. This two-inputs-one-output system can be subdivided as two major parts. The first part consists of the engine and transmission systems, and the second part is the drive-train system.

Since the nonlinear system described above is complicated and highly nonlinear, it is too difficult to design a controller based on this system in a short period. A simplified vehicle longitudinal model that can represent the basic dynamics of the vehicle needs to be designed to simplify the task of the controller.

J. K. Hedrick, et al [11]. proposed a simple three state vehicle model for control. In order to describe the vehicle model they made, I need to make following assumptions at first:1) The torque converter is locked; 2) No torsion of the drive axle; 3) No slip at wheels; 4) The gear ratio is locked.

The states that need to be controlled in this process are:

1. Engine speed (ω_e)
2. Net Torque from the engine (T_{net})
3. Vehicle Speed (v)

According to [11], the flow rate of the air in the intake manifold is governed by the continuity equation, which is:

$$\dot{m}_a = \dot{m}_{ai} - \dot{m}_{ao} \quad (3-1)$$

Where \dot{m}_{ai} and \dot{m}_{ao} are the mass flow rates into and out of the intake manifold. The empirical relationships for these rates are:

$$\dot{m}_{ai} = \dot{m}_{MAX} \cdot TC(\alpha) \cdot PRZ \quad (3-2)$$

$$\dot{m}_{ao} = c_1 \cdot \eta_{vol} \cdot m_a \cdot \omega_e \quad (3-3)$$

Where the \dot{m}_{MAX} is the maximum flow rate corresponding to a fully open throttle valve. The function TC is a nonlinear function of throttle angle α . The function PRZ is the normalized pressure influence function which is a nonlinear function of the pressure ratio $P_r = P_m / P_{atm}$, where P_{atm} is the atmosphere pressure and P_m is calculated using ideal gas law, which is:

$$P_m = \frac{RT_m}{M_{air} V_M} m_a \quad (3-4)$$

Then, the rotational dynamics of engine is given by:

$$\sum J \cdot \dot{\omega}_e = T_{net} - T_{load} \quad (3-5)$$

where T_{net} is the net torque output from the engine normally defined as the difference between the combustion torque and other losses. It is empirically known to be a nonlinear function of engine speed and manifold pressure. T_{load} is the effective load on the engine.

Also, the vehicle speed and engine speed are related by the relation:

$$v = R_t R_g \omega_e \quad (3-6)$$

Where R_t is the effective tire radius, and R_g should be a variable that depends on the vehicle gear ratio, but in this application it will be assumed as a constant. Consequently,

$\sum J$ is the effective engine inertia which includes engine, torque converter, driveshaft, and vehicle inertias. Its functional form is thereby:

$$\sum J = J_e + J_{t,g} + R_g^2 (J_{wf} + J_{wr} + \dots) \quad (3-7)$$

where J_e is the engine torque, $J_{t,g}$ is the transmission torque at a particular gear ratio, J_{wf} and J_{wr} are the inertias of front and rear wheels respectively, and M is vehicle mass. And term T_{load} is composed by all longitudinal dynamics terms:

$$T_{load} = R_g (T_{br} + C_a v_2 + R_t F_r) \quad (3-8)$$

where T_{br} is the total brake torque, C_a is the aerodynamic drag coefficient and F_r is the total rolling and friction force.

3.2 Basic Control Design

This section presents the control laws for vehicle following based a simple vehicle model shown in section 3.1. Using spacing and headways as the inter-vehicle distance control strategy, several control techniques are investigated for consideration. And at last a control law using impedance relations will be presented.

Consider two vehicles travelling on a straight lane of highway. The front one is called lead vehicle and the other one is called follower vehicle. This control task is to design an upper level controller which determines the net torque output from the engine (T_{net}). And this output in turn leads to a throttle and brake control command in a lower level. But this design will only focus on the upper level controller in order to analyze the relationship between the T_{net} and other design parameters. The configuration of the vehicle following maneuver is shown in Figure 3.2.

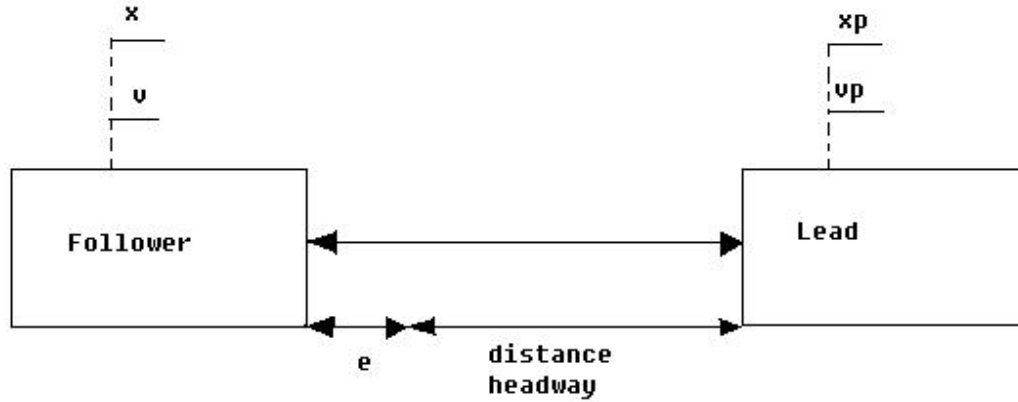


Figure 3.2 Schematic of Longitudinal Control

The Spacing error e is defined as the difference between the absolute distances of lead and follower vehicle. The distance headway is designed using “constant time headway policy” (see Chapter 1)

The basic control design purpose is as following:

- 1). The follower vehicle closed loop system should be stable
- 2). The spacing errors (e), resulting from the lead vehicle maneuver, should go to zero
- 3). The effect of velocity change on the spacing error should be as small as possible

And it is assumed following measurements are available:

- 1). Lead vehicle’s velocity (v_p)
- 2). Range and closing rate to the lead vehicle (x_r, v_r)
- 3). Engine speed

3.2.1 Preliminaries

The following notations will be used throughout the article:

x_r ----- Relative distance; (can be measured by Radar)

v_r ----- Relative velocity; (can be Measured by Radar)

a_r ----- Relative Acceleration;

x ----- Follower vehicle distance

v ----- Follower vehicle velocity

a -----Follower vehicle acceleration

x_p ----- Preceding distance

v_p ----- Preceding speed (measured by radar)

a_p ----- Preceding acceleration (may not be available)

M ----- Vehicle mass

T_{load} ----- Total Force or Torque of the rolling resistance and friction

T_{net} ----- Net output torque expected from engine or brake force

To avoid complexity in calculations, it is assumed that the starting point of the follower vehicle is the initial point which is set to be 0 and the travelling direction is set to be the positive direction(x-coordinate). Then, some basic relations are listed below:

$$x_r = x_p - x ; \quad v_r = v_p - v ; \quad a_r = a_p - a \quad (3-9)$$

The initial conditions are:

$$x(0) = 0, x_p(0) = L > 0$$

$$v(0) = v_p(0) = v_r(0) = 0 \quad (3-10)$$

From the notations and assumptions, we will have

$$\dot{v}_r = a_r, \quad \dot{x}_r = v_r \quad (3-11)$$

And the choice of distance headway is crucial for following safety, thereby, the headway is defined by the “constant time headway policy” which is:

$$x_d = \alpha v + \mu \quad (3-12)$$

where α and μ are positive constant design parameters

3.2.2 Prior work

Xiao-Yun Lu, et al. [3] used sliding control strategy to design the control law. In sliding control, usually a surface is defined as a function of error and derivatives of the error and/or integrals of the error. The surface is defined so that the state will exponentially decay along the surface to the desired point [1].

For example, they defined a sliding surface:

$$s = (v_r - v_d) + k(x_r - x_d) \quad (3-13)$$

The desired point is $s = 0$, so the reduced order dynamics can be written as:

$$\frac{d}{dt}(x_r - x_d) = -k(x_r - x_d) \quad (3-14)$$

Anouck R. Girard, et al.[1] clearly defined the error similar to Paper 2 to get the control law.

They define the error e as:

$$e = x_r - x_d \quad (3-15)$$

According to literature 1, if we define a sliding surface: $s = \dot{e} + k \cdot e$. From the sliding reachability condition: $\dot{s} = -\gamma \cdot s$. Then, $\ddot{e} + k_d \cdot \dot{e} + k_p \cdot e = 0$, where $k_d = k + \gamma$, $k_p = \gamma \cdot k$.

The control law can be obtained by feedback linearization:

$$a = a_p - \ddot{x}_d + \dot{k}_d e + k_p e \quad (3-16)$$

However, it is not necessary to define error as distance error, if we define error as velocity error, which is:

$$e = v_r = v_p - v \quad (3-17)$$

Then, $\int e = \int v_r = \int v_p - \int v = (x_p - x_d) - x = x_r - x_d$

For sliding control, the surface s can be written as: $s = v_r + k \cdot (x_r - x_d)$. This situation was also discussed in [2].

So it will be relatively easier if we took speed error as feedback error since its integral has to be the distance error in ideal conditions (not include the integral errors) but the opposite way will be false

Soo-Yeong, Yi and Kil-To, Chong proposed a guidance model shown in Figure 3.3[4]. In this system, the vehicles are guided with 'serial chains of spring-dampers'. The impedance control with conceptual serial spring-damper model is kind of force control scheme to generate the appropriate trajectory according to the force exerted from the uncertain environment.

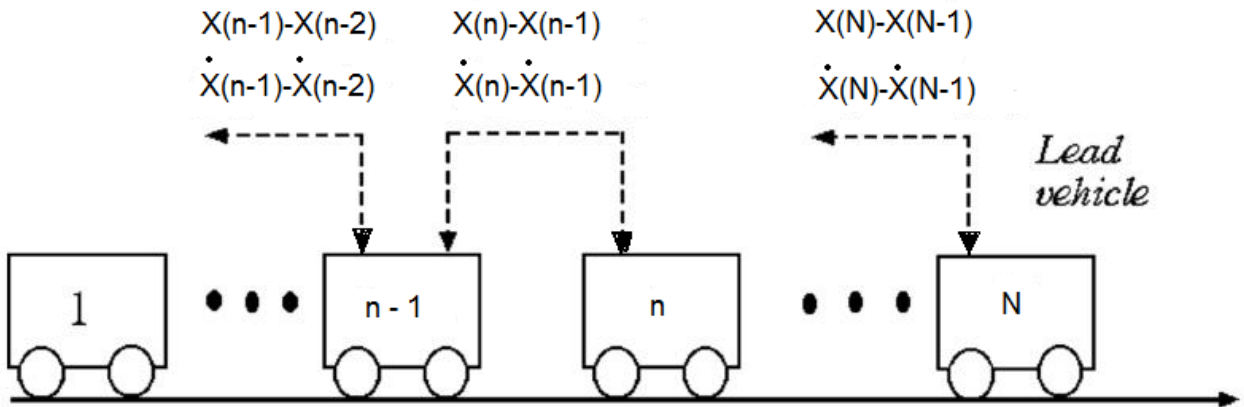


Figure 3.3 Serial Spring-Damper Guidance Model [4]

$x_{(1,2,3,4)}$ stands for the position of each vehicle. Since each vehicle in this platoon is connected with conceptual springs and dampers, so the force control strategy can be given as:

$$F_f = k(x_{n-1} - x_n - x_d) + c(\dot{x}_{n-1} - \dot{x}_n)$$

$$F_r = k(x_n - x_{n+1} - x_d) + c(\dot{x}_n - \dot{x}_{n+1}) \quad (3-18)$$

Here x_d denotes distance headway that should be maintained for safety purpose. It should be noted that $F_f = 0$ for the first lead vehicle and $F_r = 0$ for the last vehicle. From the Newton's law of motion the equilibrium of force can be expressed as:

$$M \cdot \ddot{x}_n = F_f - F_r \quad (3-19)$$

However, this model based approach has some drawbacks. The fact that the impedance before and behind an object vehicle are coupled with the position and velocity of this vehicle becomes a heavy burden to obtain the engine control force. To simplify this impedance relation, the suggested guidance model which shown in Figure 3.4, will be single impedances impacting only a single vehicle.

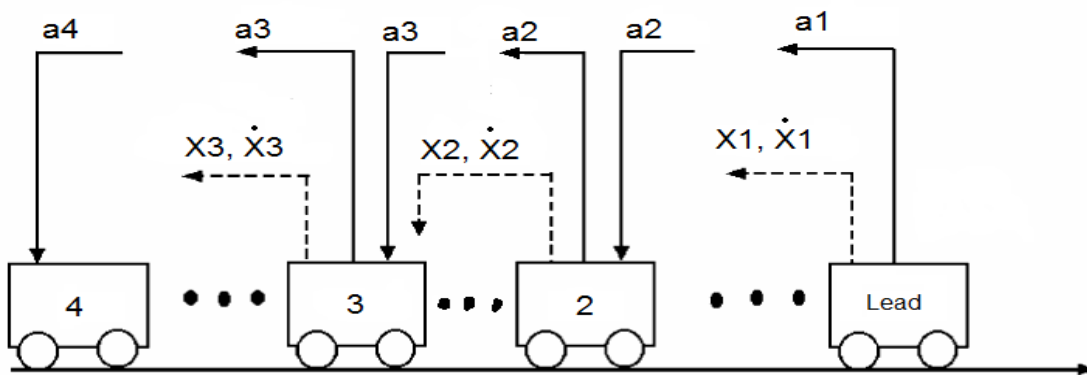


Figure 3.4 Spring-Damper Model with only One Side Information

Since this control design only involves two vehicles, so this controller based on the mass/spring/damper dynamics will be relatively stable. However, ignoring the information from the follower vehicle in turns leads to slinky-type effects, which will decrease the safety when travelling on the highway.

Based on these analyses above, this thesis will present a new control strategy for ACC which will not use sliding control scheme as usual but the impedance model approach. The sliding control in the nature is a special type bang-bang controller (on-off controller) that will switch abruptly between two states. The main strength of sliding mode control is its robustness. Because the control can be as simple as a *switching* between two states, it need not be *precise* and will not be sensitive to parameter variations that enter into the control channel. However, robustness will not be a main issue in this control design, since purpose of this design is to find the relationship of sampling period and other control design parameters. This new approach will couple the follower vehicle and the lead with spring and damper. This system will guarantee good stability and this new strategy may also reduce the slinky effects due to no information from behind.

3.3 Control Design and Stability

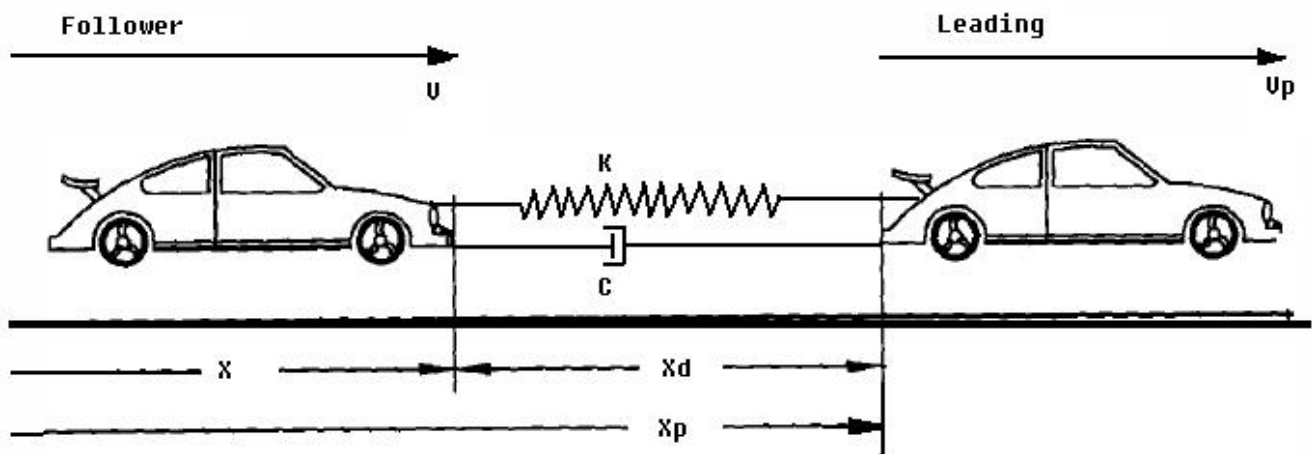


Figure 3.5 Vehicle Longitudinal Following Maneuver

Using the spring-damper model, the configuration of vehicle following strategy is shown in Figure 3.5. The leading and follower vehicles are connected with imaginary spring K with original length L and damper C . Then, the length change of the spring and the speed difference between the two vehicles are:

$$\Delta L = x_p - x - x_d = x_r - x_d \quad (3-20)$$

$$\Delta v = v_p - v = v_r$$

where $v_r = v_p - v$ and $x_r = x_p - x$ (see 3.2.1).

Then, the conceptual force applied on the follower should be:

$$F = K(x_r - x_d) + C \cdot v_r = K(x_r - x_d) + C \cdot \dot{x}_r \quad (3-21)$$

According to Newton's Law of motion, the acceleration caused by this conceptual force is:

$$a_f = F / m = \frac{K}{m}(x_r - x_d) + \frac{C}{m} \cdot \dot{x}_r \quad (3-22)$$

Now, to achieve the control action for the follower vehicle, suppose that two vehicles are at the initial state where $v(0) = v_p(0) = v_r(0) = 0$. If it is assumed that the preceding acceleration a_p is available, in a certain time Δt , there is a distance gain Δx and velocity $a_p \cdot \Delta t$ for the lead vehicle, then the acceleration for the follower caused by this distance gain is a_f . In order to catch up the lead and keep the following distance at x_d , the acceleration of the subject vehicle at this time should be given by $a = a_p + a_f$ which is greater than lead vehicle acceleration. The Figure 3.6 shows the control law for the follower

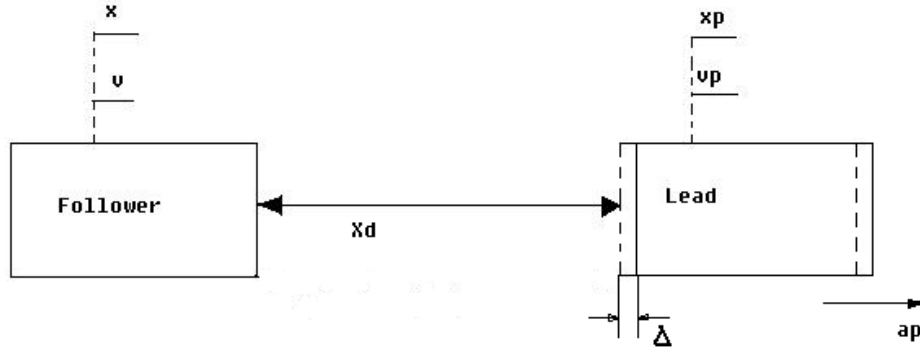


Figure 3.6 Vehicle Longitudinal Control Maneuver

Therefore, from (3-21), the control law achieved after feedback linearization is:

$$a = a_p + a_f = a_p + \frac{K}{m}(x_r - x_d) + \frac{C}{m} \cdot \dot{x}_r \quad (3-23)$$

Compare the equation above to the control Law from sliding mode control:

$$a = a_p - \ddot{x}_d + k_d \cdot \dot{e} + k_p \cdot e = a_p - \ddot{x}_d + k_d(\dot{v}_r - \dot{v}_d) + k_p(x_r - x_d) \quad (3-24)$$

(3-23) just simply dropped out the term \ddot{x}_d and \dot{v}_d . Because this two terms both came from the term x_d (derivative and second derivative of x_d), ideally they would not affect the stability of the spacing control too much

Also, x_d is the distance headway for spacing control, and ideally the sum of this headway and spacing error should be the actual following distance. This headway is defined as:

$$x_d = \alpha \cdot v + L \quad (3-25)$$

where α and L are both constants and greater than zero.

From (3-23) and (3-25), and using the relations (see 3.2.1):

$$x_r = x_p - x; v_r = v_p - v; a_r = a_p - a \quad (3-26)$$

$$\dot{x}_r = v_r; \dot{v}_r = a_r \quad (3-27)$$

Finally, the overall differential equation for control law (3-23) is obtained that:

$$\ddot{x}_r + \left(\frac{C}{m} + \frac{K \cdot \alpha}{m}\right) \cdot \dot{x}_r + \frac{K}{m} x_r = \frac{K}{m} \cdot L + \frac{K \cdot \alpha}{m} \cdot v_p \quad (3-28)$$

Since v is unknown in (3-25), but v_p , v_r and x_r are measurable (see 3.2.1), the equation is organized with x_r as output, $x_{id} = \alpha \cdot v_p + L$ as input.

For stability issues, (3-28) have poles:

$$\lambda_1 = \frac{-C - \alpha \cdot K - \sqrt{(C + \alpha \cdot K)^2 - 4K \cdot m}}{2m} \quad (3-29)$$

$$\lambda_2 = \frac{-C - \alpha \cdot K + \sqrt{(C + \alpha \cdot K)^2 - 4K \cdot m}}{2m}$$

Thus, the closed loop system is globally stable for following condition

$$(C + \alpha \cdot K)^2 \geq 4K \cdot m \quad (3-30)$$

Because if (11) is achieved, the poles of the closed loop system are always negative. But with K decreasing as $4K \cdot m \rightarrow 0$, the stability margin decreases as $\lim_{K \rightarrow 0} \lambda_2 = 0$

3.4 Implementation

In last section, a control law for maintaining the spacing between two vehicles in a safe distance is developed using the impedance relation. To implement this control law, the three-state vehicle model is needed (see section 3.1). Those states are:

1. Engine speed (ω_e)
2. Net Torque (T_{net})
3. Mass of air in manifold (m_a)

The equations below are the conditions on which the controllers based on. And the control law developed above needs to be coupled with these equations to get the engine speed or net torque.

$$\dot{m}_a = \dot{m}_{MAX} \cdot TC(\theta) \cdot PRI(m_a) - \dot{m}_{ao} \quad (3-31)a$$

$$\dot{\omega}_e = (T_{net}(\omega_e, m_a) - T_{load}) / \sum J \quad (3-31)b$$

$$\sum J = J_e + J_{t,g} + R_g^2 (m \cdot R_t^2 + 2 \cdot J_w) \quad (3-31)c$$

$$T_{load} = R_g \cdot (T_{br} + R_t \cdot F_r + C_a \cdot R_g^2 R_t^3 \omega_e^2) \quad (3-31)d$$

If assuming that R_g is a constant and using the relation $v = R_t \cdot R_g \cdot \omega_e$ (see section 3.1), it holds that:

$$a = R_t R_g (T_{net} - T_{load}) / \sum J \quad (3-32)$$

From (3-32) and (3-31)b, the net torque of the engine is given by:

$$T_{net} = \frac{a \cdot \sum J}{R_t \cdot R_g} + T_{load} \quad (3-33)$$

Where

$$a = a_p + a_f = a_p + \frac{K}{m} (x_r - x_d) + \frac{C}{m} \cdot \dot{x}_r \quad (3-34)$$

On the other hand, \dot{m}_a can be obtained through the numerical differentiation of m_a and the throttle angle θ can be found from the engine map by knowing function TC. If $\theta < \theta_0$ (θ_0 is the minimum throttle angle), according to the switching law, the brake should occur. The desired brake torque is given by:

$$T_{br} = (T_{net} - \Phi - \frac{a \cdot \sum J}{R_t R_g}) / R_g \quad (3-35)$$

Where, $\Phi = R_t R_g (F_r + C_a \cdot v^2)$

For example, use the data table below to calculate each parameter. This data is used by J. K Herdcik et al. in [11] .

Table 1 Powertrain Parameters (all units are in MKS system)

V_e = engine displacement	0.0049 m ³
V_m = intake manifold volume	0.00446 m ³
C_t = engine torque constant	1018686 Nm/k
J_e = engine & torque converter inertia	0.2630 kg m ²
MAX = max flow rate into intake manifold	0.684 kg/s
Δt_{it} = intake to torque production delay	5.48/we
Δt_{st} = spark to torque production delay	1.30/we
τ_f = fuel delivery time constant	0.05 sec
R_1 = first gear speed reduction ratio	0.4167
R_2 = second gear speed reduction ratio	0.6817
R_3 = third gear speed reduction ratio	1.0
R_4 = fourth gear speed reduction ratio	1.4993
R_5 = final gear speed reduction ratio	0.3058
$J_{t,1}$ = effective turbine inertia, 1 st gear	0.08202 kg m ²
$J_{t,2}$ = effective turbine inertia, 2 nd gear	0.07592 kg m ²
$J_{t,3}$ = effective turbine inertia, 3 rd gear	0.11388 kg m ²
$J_{t,4}$ = effective turbine inertia, 4 th gear	0.13150 kg m ²
J_{wf} = inertia of front wheel	2.565 kg m ²
J_{wr} = inertia of rear wheel	2.565 kg m ²

h_f = static axle to ground height of front wheel	0.33 m
h_r = static axle to ground height of rear wheel	0.33 m
M =vehicle mass	2148 kg
K_s =shaft siffness	6742 Nm/rad
F_{rf} = front tire rolling resistance	86.16
F_{rr} = rear tire rolling resistance	81.11
C_a = aerodynamic drag coefficient	0.53384 kg/m
$K_f(i)$ = tire slip proportionality, front (both side combined)	83710 N
$K_r(i)$ = tire slip proportionality, rear (both side combined)	79070 N
$\tau_{b,v}$ = vehicle brake torque time constant	0.1 sec
$\tau_{b,t}$ = total system brake torque time constant	0.25 sec
K_{bf} = front brake torque proportionality constant	
K_{br} = rear brake torque proportionality constant	

Let $\alpha = 0.4$ and $L = 2.0$, for maintaining the speed of 20m/s, the headway can be found as 10 meters. Assuming the gear ratio is locked at 3rd gear, then, the typical values of these parameters are:

$$\frac{\sum J}{R_t \cdot R_g} = 725.50 \quad (3-36)$$

$$\Phi = R_g R_T (F_r + C_a \cdot v^2) = 0.33 \cdot (167 + 0.5 \cdot 400) = 121 \quad (3-37)$$

Thus, from (14), the net engine torque is:

$$T_{net} = a \cdot (\sum J / R_t R_g) + \Phi = a \cdot (725) + 121 \quad (3-38)$$

With the headway control law, the net torque is given by:

$$T_{net} = 725 \cdot [a_p + \frac{K}{m} (x_r - x_d) + \frac{C}{m} \dot{x}_r] + 121 \quad (3-39)$$

However, in the implementation, if a_p is dropped due to it being difficult to measure, and let $K = 1800$ N/m, $C = 2000$ N/s, for an initial velocity error 0.1 m/s for a 2-vehicles platoon,

the net torque is around 140 Nm. But if a_p exist and is measurable, the torque would be much greater than this amount. Also, as this error increases, the torque required increases. From the control law, the torque required is inversely proportional to the headway.

If No terms in control (14) and (15) are dropped. This does not exclude the situation that some quantities may be estimated. For a measured signal c , due to measurement noise, some filters have to be used before it is fed into the controller. Filtering will cause some time delay and discrepancy $E_f(c)$ compared to the nominal signal. Thus there is measurement error $E_e(c)$ in practice [2]:

$$E_e(c) = E_m(c) + E_f(c) \quad (3-40)$$

where E_m is the error caused by measurement itself.

Suppose $E_e = T_{net} - T_{np}$ is the discrepancy. Because the controller is solved from (15), replacing T_{net} with $T_{np} + E_e$ in (14) leads to:

$$T_{np} = \left(\sum J / R_t R_g \right) \cdot \left[a_p + \frac{K}{m} (x_r - x_d) + \frac{C}{m} \dot{x}_r + \frac{E_e}{\sum J / R_t R_g} \right] + T_{load} \quad (3-41)$$

Therefore, due to a disturbance term with E_e . Some boundary layer naturally results in practice.

CHAPTER 4: ANALYSIS AND SIMULATION

4.1 Discrete-time control

4.1.1 Z-Transform

The Z-transform converts a discrete time-domain signal, which is normally a sequence of real or complex number, into a frequency-domain representation. It can be considered as a discrete equivalent of Laplace transform and is a good method to find the how the sampling frequency would affect the spacing control.

Still assuming the lead acceleration a_p is available, from the differential equation, the transfer function can be found from (3-28):

$$H(s) = \frac{X_r}{X_{id}} = \frac{K}{ms^2 + (C + K \cdot \alpha)s + K} \quad (4-1)$$

In order to make it clear, K, C, m are given the values as K = 2000, C=1800, m=2148. Hence, the transfer function can be written as:

$$H(s) = \frac{2000}{2148 s^2 + 2600 s + 2000} \quad (4-2)$$

To discretize this system using the triangle approximation with sample period Ts = 0.5 second and 0.1 second. Then the Z transform function is given as:

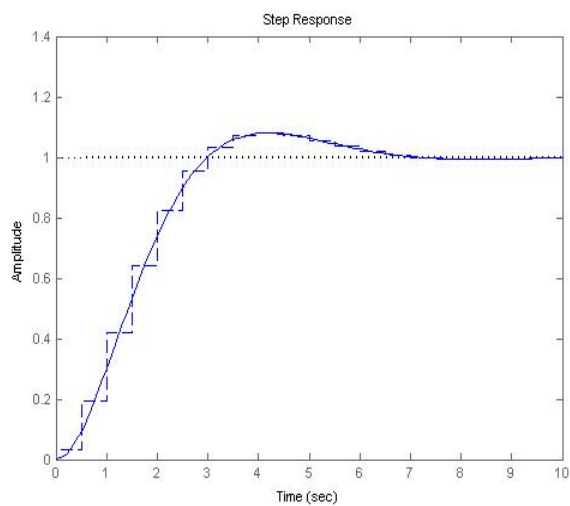
$$G(z) = \frac{0.0332 z^2 + 0.1136 z + 0.02451}{z^2 - 1.375 z + 0.546} \quad (4.3)$$

Sampling time: 0.5

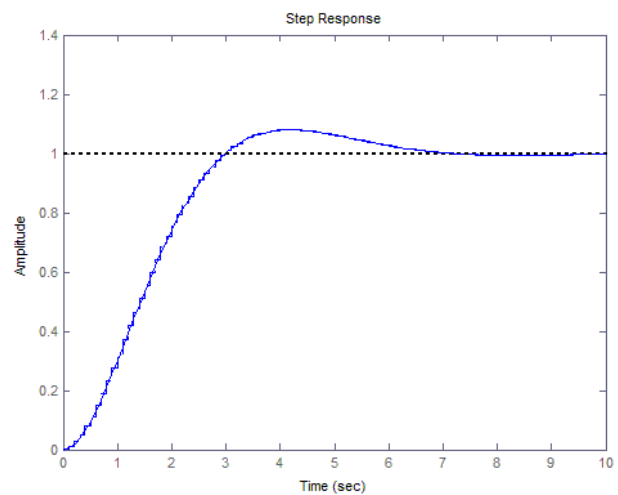
$$g(z) = \frac{0.001505 z^2 + 0.005841 z + 0.001417}{z^2 - 1.877 z + 0.886} \quad (4-4)$$

Sampling time: 0.1

Then, comparing the continuous and discretized step responses in Matlab workspace will lead to the results in Fig 4.1



(a) $T_s=0.5$ sec.



(b) $T_s=0.1$ sec.

Fig 4.1 Comparison of Continuous and Discretized Step Responses

This open loop system has an overshoot about 10%. Also, the z transform exactly transfer the response to discrete values at the preset sampling period. In the modeling, the zero-order hold is usually used for creating discrete-time values, and often followed by a continuous system.

For stability issues, if the system gain is equal to 1, the closed loop system characteristic function for (4-3) is given by $P(z) = 1+G(z) = 0$, which becomes:

$$1.0332 z^2 - 1.2614 z + 0.5705 = 0$$

The roots of the characteristic function are found to be:

$$Z1 = 0.6104 + 0.4237i, \quad Z2 = 0.6104 - 0.4237i$$

Since $|Z1|=|Z2|<1$, the system is stable.

Using the same method, transfer function (4-4) will have two poles that:

$$z1 = 0.9342 + 0.1155i, \quad z2 = 0.9342 - 0.1155i$$

Since $|z1|=|z2|<1$, this system is stable too.

4.1.2 Study on Digital Control System

Because, by using radar or lidar, the relative distance (i.e. following distance) and leading vehicle speed is measurable, there should be enough information to study on that how the update frequency of radar or lidar can affect the following distance.

A simplified closed-loop control system for transfer function (4.1) is shown in Figure 4.2. The system gain is preset to be 1 and the sampling periods for test are $T1 = 0.1$ sec and $T2 = 0.5$ sec.

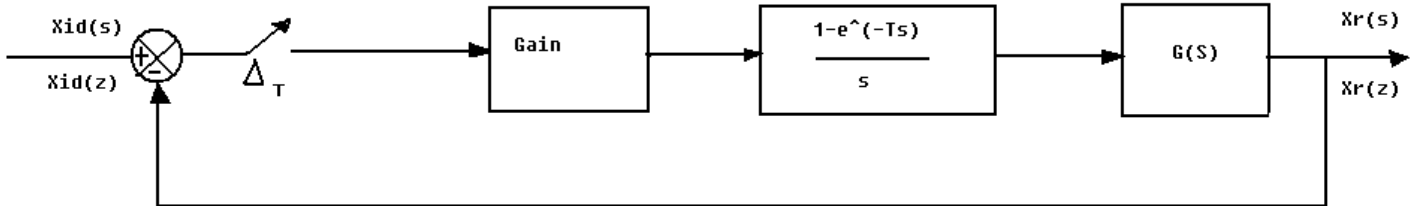


Figure 4.2 Closed-loop Control System for (4-1)

To model this system in Matlab/Simulink, $G_h(s)$ and $G(s)$ should be transferred to z -domain. For sampling period at 0.1 sec. and 0.5 sec., the z transfer functions have already been given at (4.2) and (4.3). Therefore, the Simulink solution for this system can be found at Fig 4.3

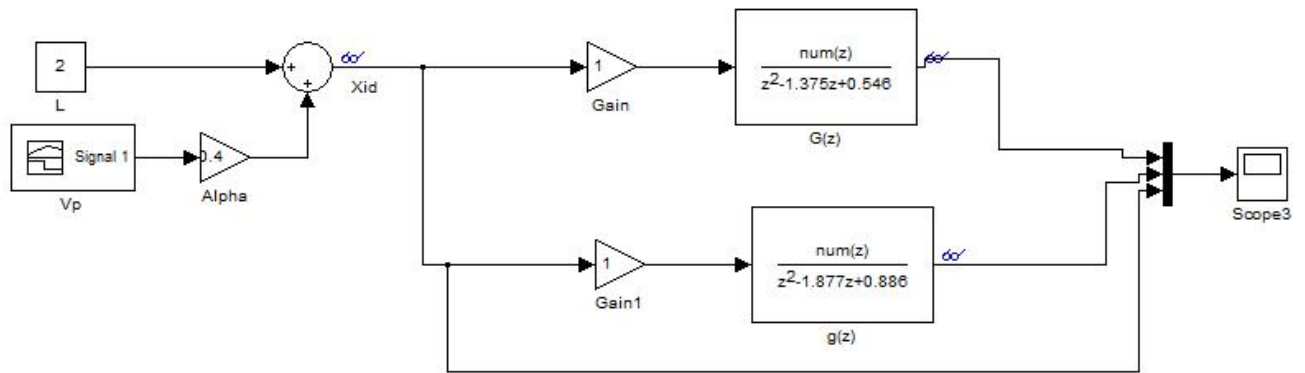


Figure 4.3 Simulink Solution for z-Transfer Functions

There are two test groups of signal for simulation using different driving maneuvers. In both groups, the ideal following distance is calculated by the lead vehicle speed.

Group 1

The leading vehicle speed increases from 0 to 20 m/s in 20 sec. then keeps at this value for some time, and at last decelerate to stop.

Group 2

The leading vehicle keep at speed of 20 m/s for 50 sec. then suddenly stop in 3 mins. The simulation results are shown in Figure 4.4

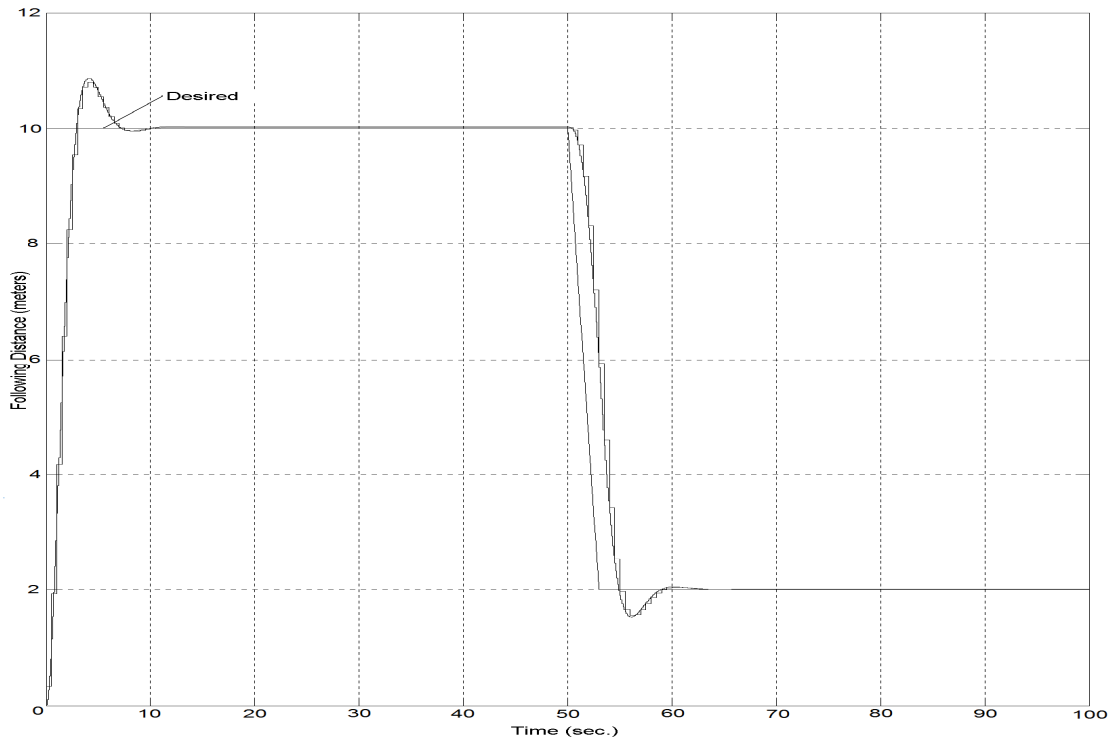
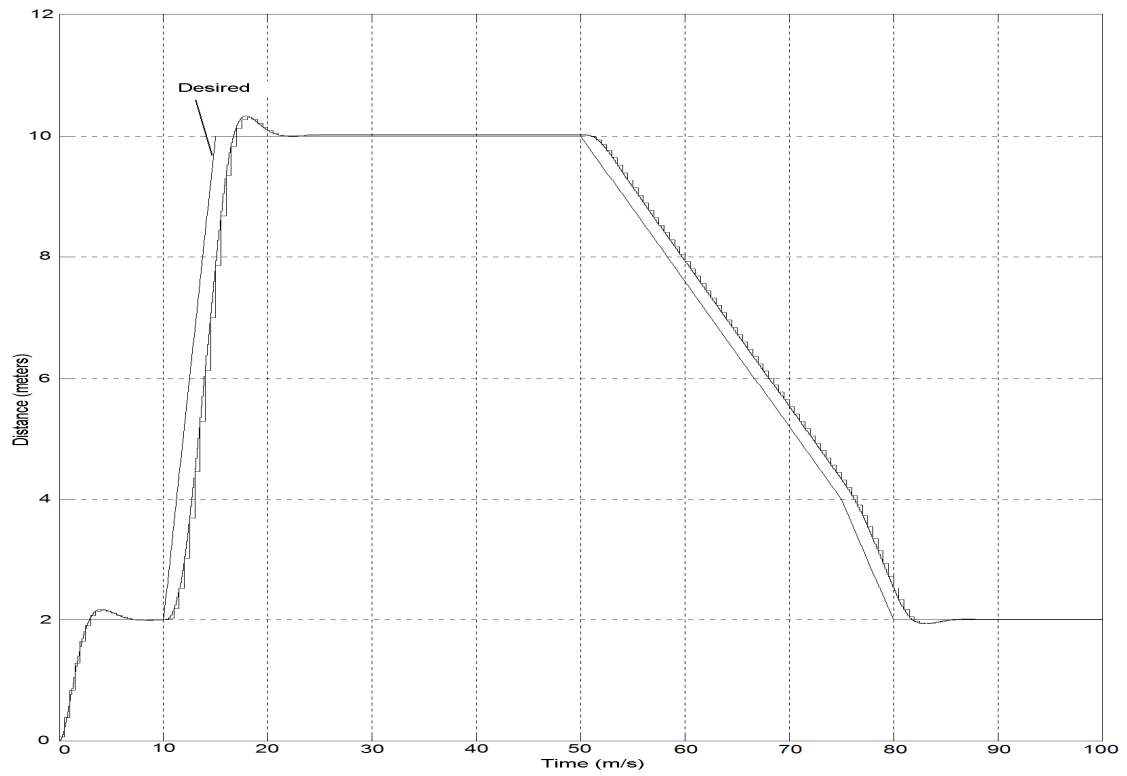


Figure 4.4 2-vehicle Simulation with Different Sampling Period

These simulations show the similar results for both sampling periods. The results proved that the discrete control system is equivalent to a type 1 system in the continuous domain. This conclusion is true because the original Laplace transfer function had been multiplied by term $[1-e^{(-Ts)}]/s$. Therefore, in Figure 4.4, when there is the ramp-function input, the error occurs.

4.1 Longitudinal and Cruise Control Simulation

Figure 4-6 shows a simple illustration of an ACC controller in a vehicle. A lidar or radar is usually connected to the vehicle to measure the relative speed and relative distance. Its output is calibrated and subjected to signal conditioning before being converted into a digital signal. The digital signal is processed by the adaptive cruise control(ACC) system realized as a digital controller, whose output is then fed back again to the engine or brake system to control throttle angle or brake torque finally control the vehicular speed.

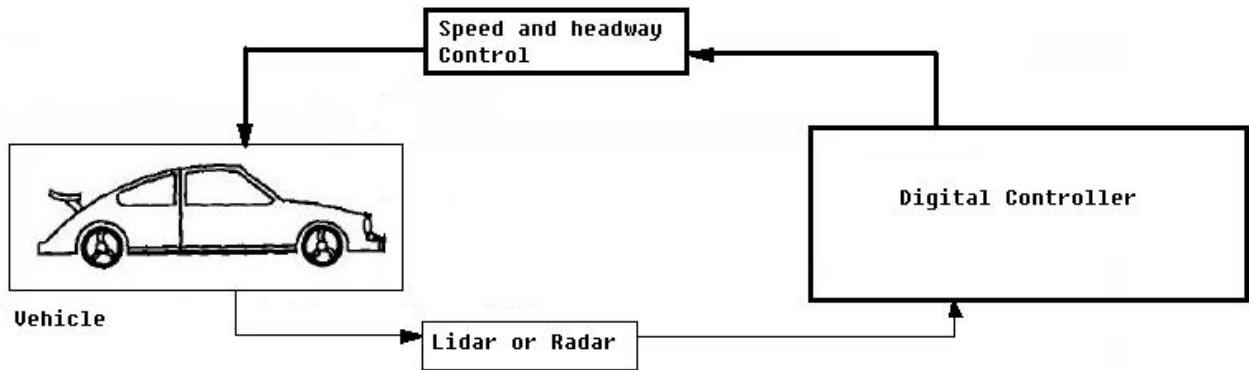


Figure 4.5 Speed and Headway Control Process

In this section, simulations of a four-car platoon under a closed loop control are presented. The mathematical model of the car is given by the two state car model (drop the air mass rate) described in section 3.1. The simulated speed, following distance, acceleration, and required torque results were presented in Figure 4.7 to Figure 4.9, and they are based

on the controller designed by impedance relations. The maneuver used in the simulation is a typical velocity profile from 0 to a cruise speed then decrease to a slower cruise speed.

Also the simulation a two-vehicle group using adaptive cruise control strategy is shown in Fig 4.10. This simulation was still based on the controller designed with impedance but it will have a simple switching law to switch the subject vehicle to CC or ACC regarding to the existence of the lead vehicle.

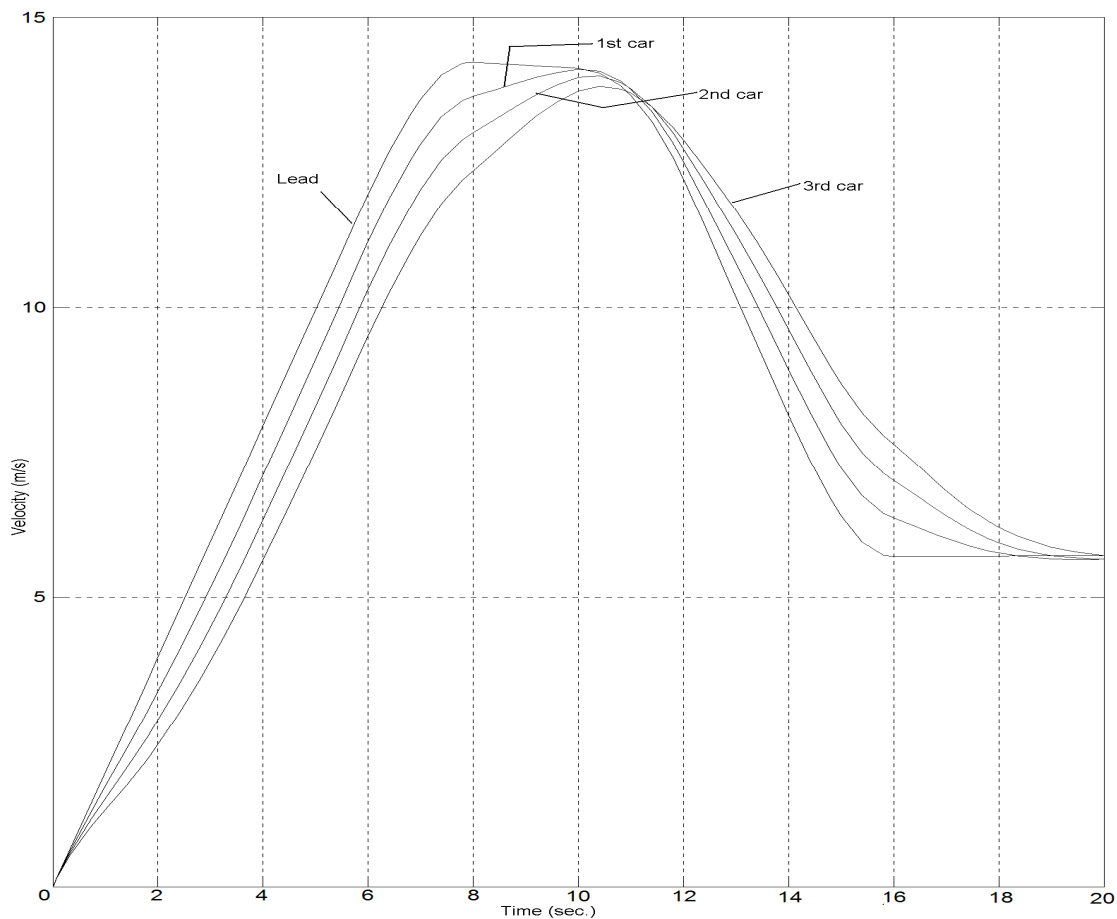


Figure 4.6 4-Car Platoon Simulation Results: Velocity

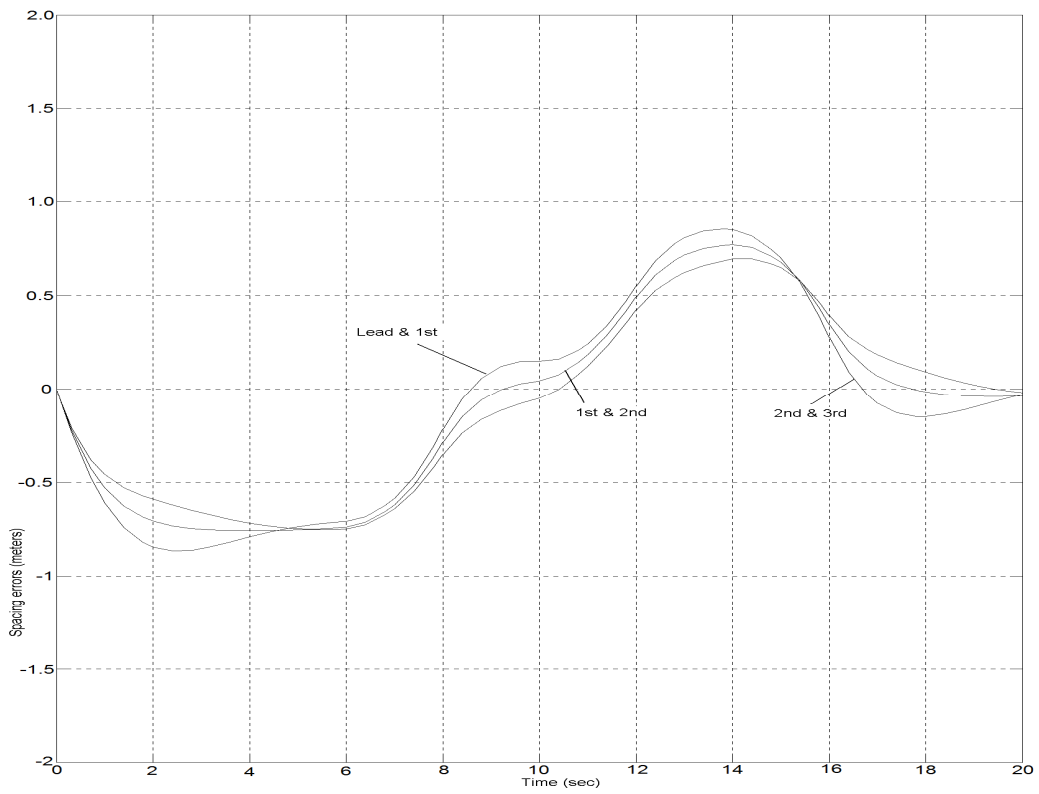
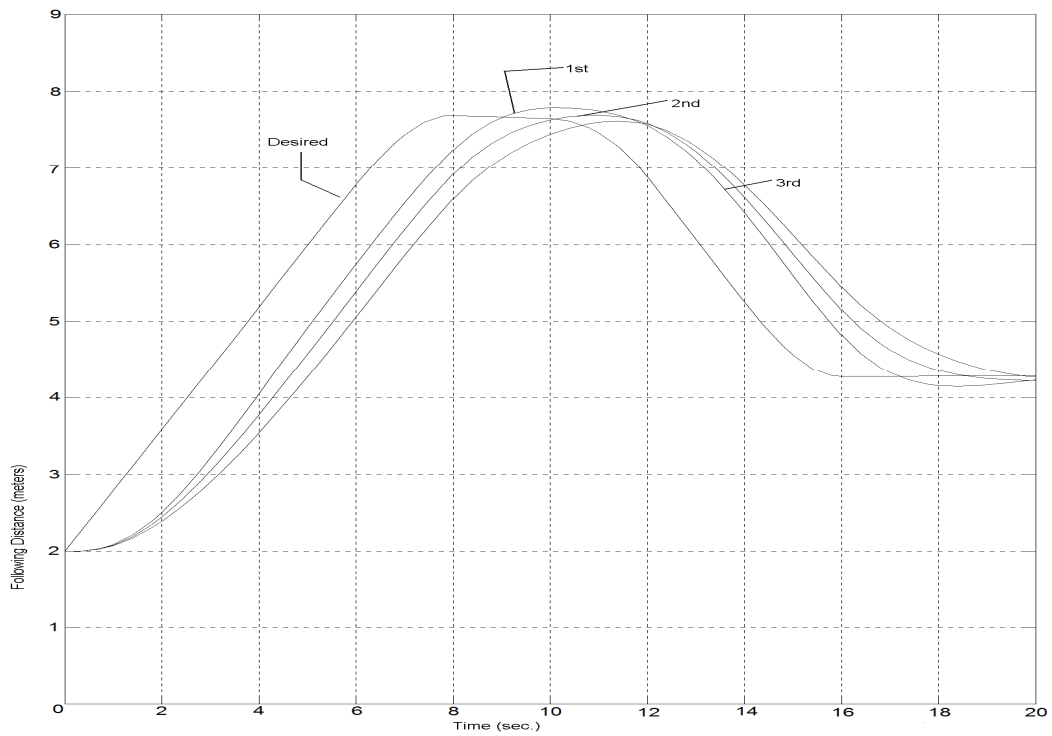


Figure 4.7 4-Car Platoon Simulation Results: Following Distance and Errors

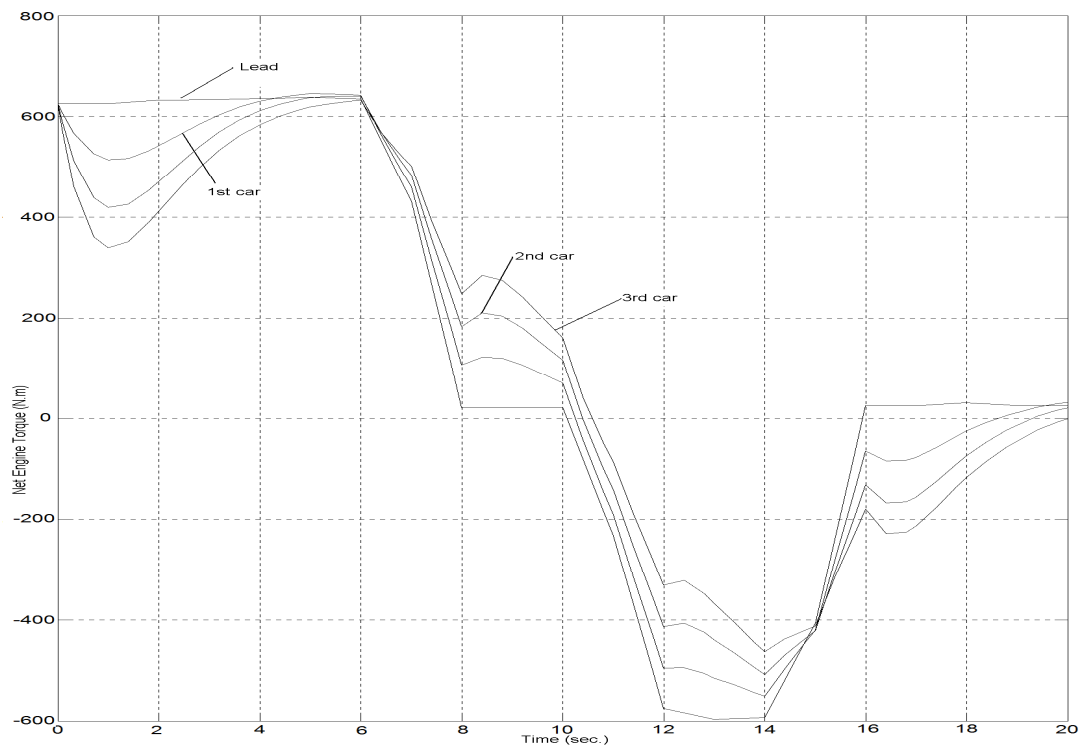
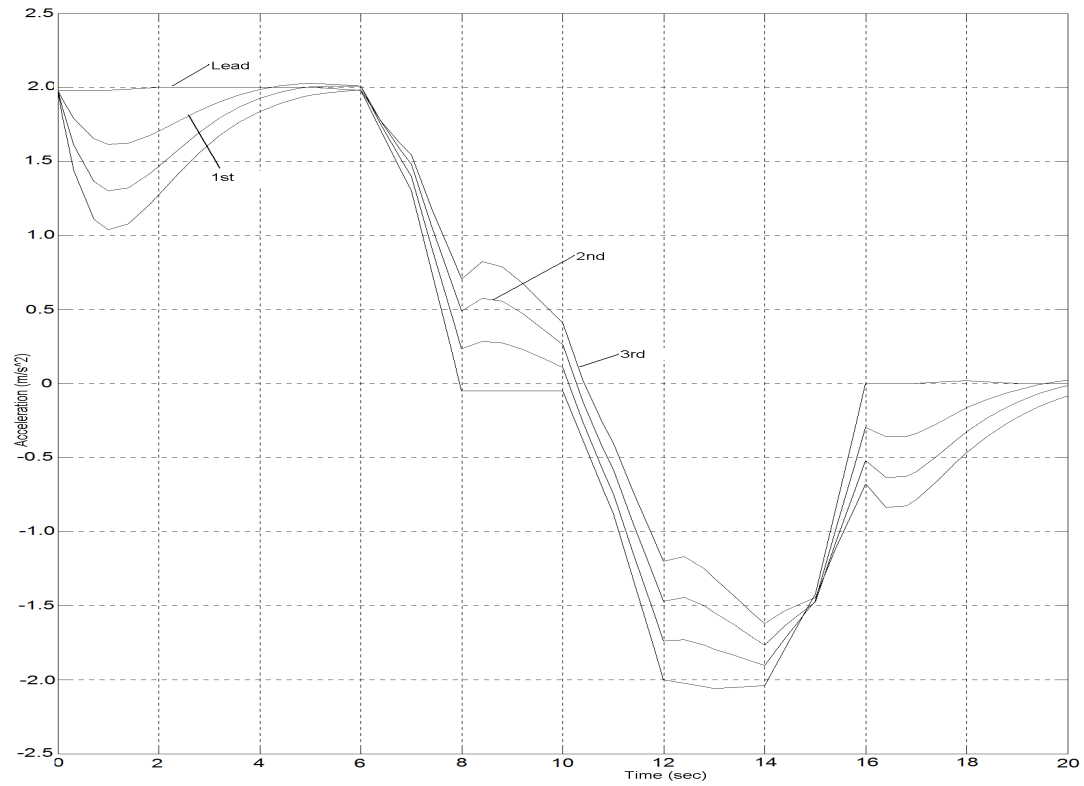


Figure 4.8 4-Car Platoon Simulation Results: Net Torque & Acceleration

In this simulation, all subject vehicles have the initial speed of 0 m/s. The lead vehicle start to move at ($t = 0$ s) with an initial acceleration of 2 m/s^2 . The subject vehicles start to accelerate at $t = 0$ too. The lead vehicle start to decelerate at $t = 10$ second and finally go to zero at 16 second.

Figure 4.6 shows the different speeds of 4 vehicles in this platoon. It shows that the speed error start to propagate along the platoon. When the desired speed tends to maintain a constant, the following speeds will start to approach the desired value. Therefore, the distance headway will reach a constant in steady state ideally. The Figure 4.7 shows that the following distance also tends to approach a constant when the speed is steady, but the distance errors are still not insignificant. Figure 4.8 shows the acceleration and net torque profile of 4 vehicles. The net torque of each vehicle matches the curves of acceleration.

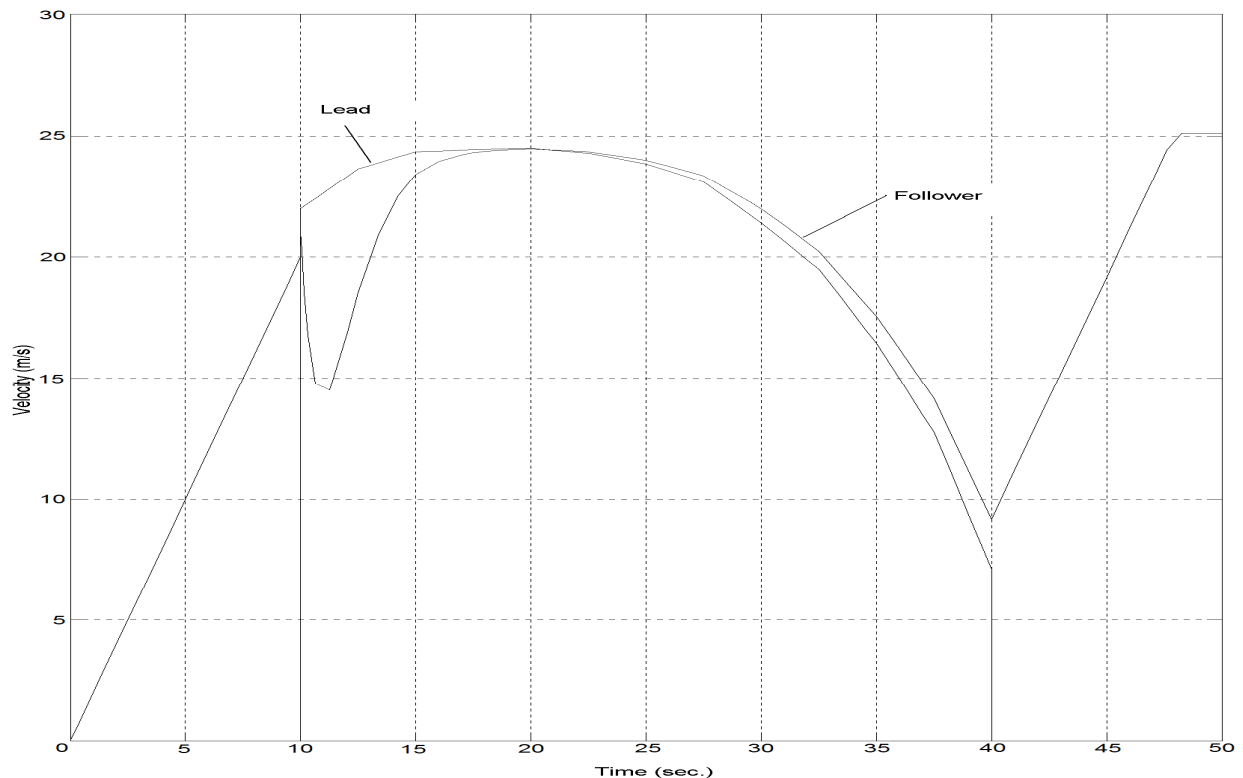


Figure 4.9 Adaptive Cruise Control Simulation: Velocity

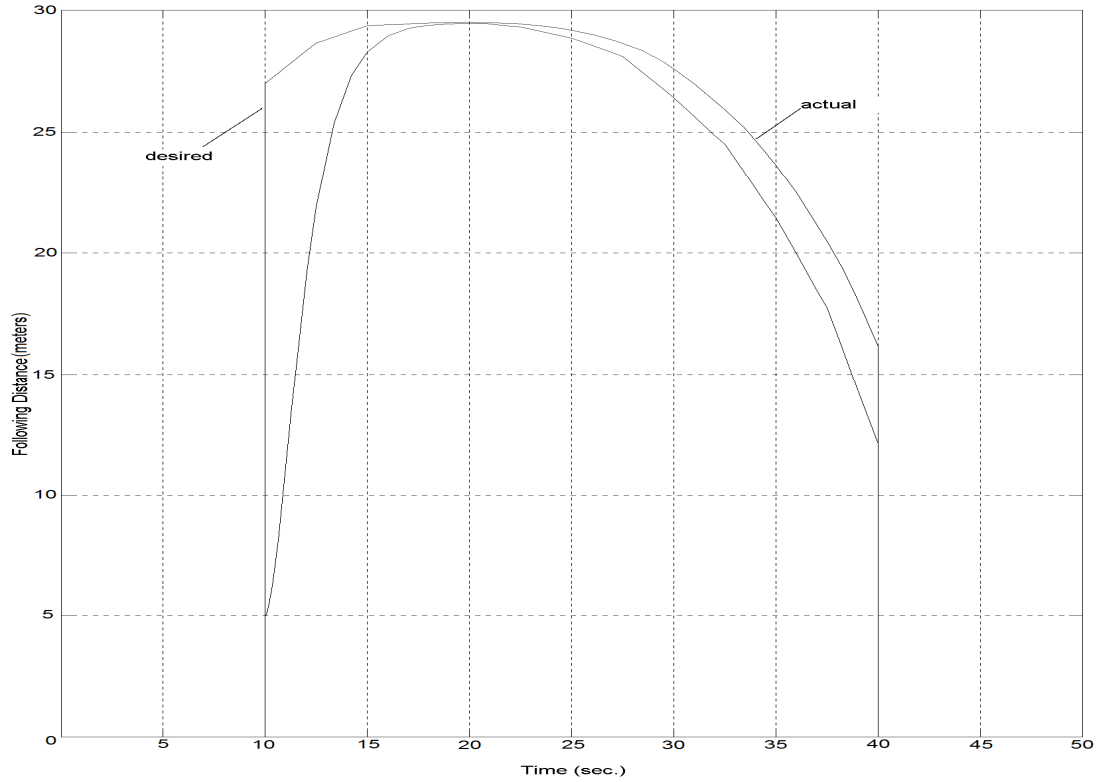


Figure 4.10 Adaptive Cruise Control Simulation: Following Distance

In this simulation, the subject car has a constant acceleration of $2 \text{ [m/s}^2\text{]}$ from the beginning and a desired speed of 25 [m/s] . It starts to accelerate at $t = 0$ to reach the desired speed when there is no leading vehicle. After 10 seconds ($t = 10\text{s}$) a leader car cuts in front of the subject car only 3 m in distance. The desired following distance for the subject car is: $x_d = 1.0 \cdot v + 5.0$. Since 3 m is much less than X_d at this point, the controller performs an emergency transition (skipping the CACC mode) from a conventional cruise control (CC) mode to a following mode (ACC). At $t = 40\text{s}$, the leader car moves out of the subject car's lane and the subject car transits back to CC mode and accelerates to its desired speed.

In Figure 4.9 and 4.10, the preceding acceleration (a_p) was used as an input. The gains are same with the 4-car platoon simulation. It has the satisfactory results for speed control. Except for the errors that occurred when the lead car cuts in and left, the general errors are acceptable. But still it has noticeable distance errors under this control strategy.

CHAPTER 5: CONCLUSIONS

An adaptive cruise control design based on the conceptual impedance relations has been proposed. A guidance model involving the spring and damper has been discussed. The controller leads to globally stable closed-loop systems. Due to the headway choice, which depends on an unknown state variable, i.e. subject vehicle speed, the stability of the closed-loop system can not be determined directly. The transfer function of the system using distance headway as input and relative distance as output is used to discuss the stability issues. The distance headway depends on lead vehicle speed in general. However, if the preceding speed is constant, the distance headway approaches a constant too in steady state. This design can be implemented for ACC. In this thesis, a simple three states vehicle model was used to achieve this.

Also, this control law coupled with an impedance guidance model is discussed. This model uses relative speed and spacing measurements from the vehicle ahead only. The controller itself guarantees vehicle stability but this platoon system does not have good simulation results. The control system in discrete domain is discussed too. Using the z transform can discretize the continuous control system in to a digital control system. The response of the digital system is tested with two different signal groups. The simulation of different sampling periods shows the similar results and these results proved that this system in the discrete domain is equivalent to a type 1 system in the continuous domain.

5.1 Future Work

Based on the problems discussed in the previous sections, there are still a number of new challenges for this adaptive cruise control system. Some of these challenges are listed below:

- (1). Full implementation

The simulation of the control system with full implementation is crucial for testing the ability of ACC system. In order to make the proposed system work for real vehicle, the complete modeling for the target vehicle is required

(2). Sampling effects

The major problem with the use of sensors in practical control system is that the selection of the sampling frequency on the basis of system bandwidth will result in information loss due to sampling. Therefore, researches are needed to discuss how to lower this kind of losses

(3). Sensor issues

One way to reduce the ill-effects of sampling is using a very high sampling rate. However, a high sampling rate requires sensors with high update speed and computers with high computational speed. Thus, what sensor sampling frequency is appropriate for a particular vehicle is also a problem

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CURRICULUM VITAE

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