DESIGN, MODELLING AND CONTROL OF INTER-VEHICLE DAMPING MECHANISMS FOR CONVOY TRANSPORTATION SYSTEMS

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Contents

List of Tables

List of Figures

Abstract

This work investigates and evaluates the use of a novel inter-vehicle damper concept which enables automated vehicles to operate as tightly packed convoys. A survey of current transportation methods shows the need for high performance systems to achieve better utilization of resources and infrastructure, higher capacity, and better efficiency. The work also introduces the convoy technology which fulfils these needs, substantially increasing vehicle throughput to that comparable with mass transportation systems. Additional system flexibility comes in the form of dynamic reconfiguration of convoys while en-route.

A technology review of close headway automated systems reveals that minimal headway systems are considered impractical at present due to the required control and communication specifications and passenger comfort and safety implications. The 'bridging dampers' introduced in this work successfully remove these constraints and become an integral part of the vehicles' control strategies, simplifying them considerably.

This research also identifies the potential degradation of ride quality due to the interactions between adjacent vehicles and undertakes a literature-based review of the human factors involved in the subjective perception of ride quality, resulting in comprehensive evaluation criteria and two sets of limits. The less stringent limit corresponds to degradation of ride quality while the higher one corresponds to hazardous situations for vehicle occupants.

The passenger comfort problem is addressed through the development of damping strategies which minimise the vehicle interactions affecting passengers but assist the drivetrain in maintaining sufficient position accuracy. Safety is ensured during catastrophic system failures through the rapid dissipation of collision energy in a destructive, yet controlled manner.

The control strategy and damper design principles are defined to suit different applications. An appropriately detailed model of a hybrid electric vehicle is used to simulate the convoys under a variety of conditions to test and verify the concepts and findings of this work. Comparisons with other short headway technologies show the benefits of using the bridging dampers in performance, reliability, safety and cost. This work resulted in the production of a MATLAB simulation toolbox which incorporates the convoy philosophy and provides the flexibility to evaluate these strategies and mechanisms in diverse situations and transportation systems.

The thesis concludes that the damping devices proposed will allow the operation of automated vehicles in convoys, that they allow better utilisation of resources and infrastructure and that the control system and communication requirements for convoy operation may be reduced without sacrifices in safety, comfort or accuracy.

Declaration

No portion of the work referred to in this thesis has been submitted in support of an application for another degree or qualification of this or any other university or other institution of learning.

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Dedication

To my beautiful wife Carla, for her love, support and understanding. I look forward to the next adventure in our journey through life together.

Chapter 1

Introduction

1.1 Introduction to the Thesis

This thesis is concerned with providing possible solutions to the challenges of short headway automated transportation systems. The work has been directed towards the development of intelligent inter-vehicle damping equipment which effectively allows zero headway *'convoys'* of automated vehicles whilst providing safety and simplicity to the system without compromising the ride quality. The project emanates from the system proposed in Perrott & Renfrew (1998) and follows two previous PhD projects concerning the modelling and development of drive control schemes for convoy systems (Ferreira-Rodriguez 2003), and the dynamics and control of light rail vehicle convoy systems (Sudin 2005).

Although the main feature of this project is the design of the inter-vehicle damping device, the innovative character of the proposed transportation system demanded an extensive amount of work dedicated towards defining and justifying the system's features and requirements relevant to the design of the damping device. The multi-vehicle dynamic problem is engaged largely from the control system's perspective taking into consideration human factors and limitations. The mechanical details of the damper are discussed in general terms, but left open to allow flexibility in the design of specific convoy systems.

Because of the nature of this project, which involves several vehicles and considerable infrastructure, the budget associated with manufacturing components and obtaining experimental results from them are prohibitive. Instead, simulation models are used to obtain results. A software toolbox which embodies the system's philosophy was developed for homogeneity of results and to allow researchers to experiment with different convoy configurations. The toolbox's architecture, models and methods are thoroughly discussed in this work.

1.1.1 Organisation of this work

This chapter gives an introduction to the project and defines the area of interest. The status of current ground transportation modes is described and the transportation crisis introduced. The chapter ends by outlining the aims of the project and describing the methodology adopted.

Chapter 2 is a comprehensive review of automated transit technology. It describes the state of the art technologies and techniques relative to the field and establishes terms and concepts used throughout the thesis. An extended analysis on control design and information architecture is given in chapter 3. The nomenclature for the rest of this work is there described and some of the proposed transport system features and requirements are justified.

In chapter 4 current research, standards and regulations related to human vibration and shock are reviewed extensively. It leads to the specification of a set of limits and rules for damper design and the production of comprehensive evaluation criteria which are essential for the assessment of different vehicle and convoy configurations. Such metrics fall into three main categories: Evaluation of ride quality and motion perception in steady state, evaluation of motion transients during convoy manoeuvres, and evaluation of safety and occupant protection.

The concept of *'bridging dampers'* is introduced in chapter 5. The convoy systems philosophy is made clear in this chapter and a conceptual design of the damper is proposed. The implications of introducing the damper in the control system are analysed with the methods derived in chapter 3. Five operating regimes thought to cover every possible scenario in convoy operations are identified in this section, resulting in the definition of requirements for the damping device. A discussion on different strategies and control schemes adopted by the vehicle's controller which affect the behaviour of the damper and the drivetrain is also included in this chapter. The concept is verified using a simplified model of vehicles performing the various operating modes. The results are benchmarked with simulations of identical vehicles under identical circumstances with the exception of the inter-vehicle damping devices.

Chapter 6 presents the application of the concept to road hybrid electric vehicles which are validated against a suitably detailed model. To simulate a realistic transport network, vehicles of different mass and performance are tested under environmental and dynamic conditions based on real journeys. The convoy is compared with an automated system that uses a traditional control scheme.

Finally, chapter 7 examines technical and economic issues raised by the introduction of the inter-vehicle devices proposed in this research. Results are discussed and conclusions drawn. The chapter then suggests future work in the area placing the project in the perspective of an ongoing research and development programme.

1.2 The Transportation Challenge

Motorised transportation has been evolving since the 19th century with great success. Modern societies have become highly dependent on the ability to move an ever increasing number of passengers and goods. Such is our need for transportation that our roads, rail lines, waterways and even our skies are constantly crowded. The capacity of current transportation systems has been largely overstretched due to the application of partial solutions and will soon reach its limits.

During 2002, 738 billion passenger kilometres were travelled using terrestrial transportation systems in Great Britain, 85% of which were made using cars or taxis, accounting for 80% of the road traffic (DfT 2003*b*). The occupancy rate for cars and taxis, however, was of only 1.56 passengers per vehicle. Due to the enormous number of vehicles using the road networks around the world in such an inefficient way, traffic congestion is said to be the most challenging unsolved problem in most cities. Every day millions of inhabitants find themselves trapped in crowded roads at rush hours. Figure 1.1 (DfT 2003*a*) shows the percentage of time spent within different speed bands for urban areas in Great Britain. It can be seen that up to 20% of the time on average may be wasted without even moving. Furthermore, it can be seen that up to 30% of the time may be spent moving at very low speeds (under 5mph). Increasing journey times result in hours of delay, producing stress to the travellers and significant losses in productivity.

Moreover, enormous amounts of fuel are wasted every day producing exhaust fumes and pollutants. The road transport industry alone used 37.49 million tonnes of petroleum during 2002, representing 77% of the petroleum used by all means of transport and nearly half of the total amount of petroleum used for energy and non-energy purposes in Great Britain during that year (DfT 2003*c*).

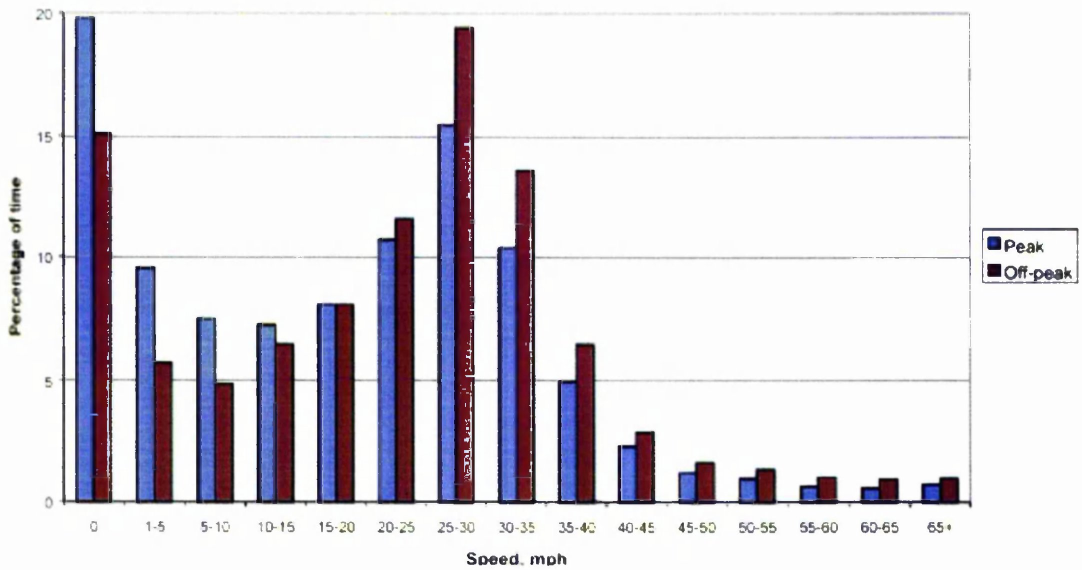


Figure 1.1: Traffic speed distribution in Great Britain. Source: DfT (2003*a*).

Road transport is also responsible for releasing 31 million tonnes of carbon dioxide to the atmosphere; nearly 25% of the total CO_2 emissions. These figures are expected to rise, reaching at least 39 million tonnes of carbon dioxide each year by 2005 and 46 million tonnes by 2020. In addition, city centres are running out of space to correctly park the always increasing number of vehicles. Parking restrictions are enforced in all major cities, and motorists usually spend a long time looking for a parking place or park their vehicles incorrectly. All these problems result in prohibitive parking fees and increased congestion.

Historic data from transport surveys show that the percentage of passenger kilometres travelled using public transport has fallen from around 60% to around 12% since the 1950's (DfT 2003*c*). That trend has been stopped in the last decade, but the number of vehicles in circulation increases every year and measures implemented in the past are no longer sustainable.

Yet, the private car is the preferred means of transportation for urban and suburban trips because of a very simple reason: the public transit systems just do not offer a service which will attract people away from their automobiles (Irving, Bernstein, Olson & Buyan 1978).

1.2.1 The future of transport

Compared with the general technological progress of the last century, the land transport industry did not change significantly until recent years. Emerging technologies have created more efficient engines and drives. Advanced transport systems, such as automated metros and people movers are already being used in several places, connecting cities with their surroundings and their airports. Car sharing programmes have been successfully implemented in certain environments. High velocity rail systems are becoming the norm in developed countries.

Changes in transportation demand respond to how modern societies evolve. It is thought that *'tele-working'* and *'e-commerce'* may reduce the total volume of daily travellers, but as the Department of the Environment & the Regions (1998) points out, it can complicate social and regulatory issues, as well as increasing the car travel from home. It also may encourage the movement out of towns into the countryside, resulting in more scattered population, and therefore, in less sustainable travel patterns.

The United Kingdom government's white paper called "A new deal for transport: Better for every one" (Department of the Environment & the Regions 1998) describes the new approach on transport policy adopted by the current government. It admits the urgent need for a radical change in the issue, formally abandoning the previous road construction policy launched in great scale by previous administrations (Goodwin 1999). Its main ideas are:

- The need to reduce the dependence on the car.
- The improvement of the public transport services, including better integration and multi-modal systems.
- The provision of more choices for travel.
- The need to give priority to healthier transport modes such as walking and cycling in the cities.
- The increase and redefinition of partnerships between public transport operators and authorities, resulting in better services for users.
- The provision of better information for public transport users and easier ticketing.

- The integration of transport with other aspects of Government policy.
- The creation of new institutions, such as the Strategic Rail Authority.

This white paper paved the way for the document "Transport 2010 the 10 year plan" presented on July 2000 by Department of the Environment & the Regions (2000, "Transport 2010 the 10 year plan"). This plan looks for public and private investment of 180 billion over the next ten years, aiming mainly to *"tackle congestion and pollution by improving all types of transport in ways that increase choice"*. The core strategy is to develop new approaches based on transport integration, public and private partnerships and the modernisation of the transport network in ways that make it bigger, better, safer, cleaner and quicker.

The 10 year plan clearly sets its targets, mostly focusing on better, more reliable public transport services (railway, underground and bus), less time delays, reduced harmful emissions, better management and maintenance of the network and increased walk and cycle trips. However, it includes little discussion on alternative transportation systems, the technologies to be used, or the possibility of automated transportation.

Such efforts indeed improve the capacity and reduce environmental impact of our transportation systems; however, core limitations have seldom been addressed. In order to have transportation systems that are more appealing than private cars, they must offer reliability, flexibility, integration, privacy and reduced waiting times and walking distances. They also must meet present and future capacity needs.

1.2.2 Automated transportation systems

Automated Transport Systems (ATS) are those which are not controlled by a human driver. A computer is in charge of driving the vehicle in a safe and efficient manner, avoiding obstacles and choosing the best way to arrive at its destination. ATS can be either mass transit or for individual use, and act as a demand responsive system or serve a specific route (McDonald & Vöge 2002). Often, the vehicles are isolated from conventional traffic and pedestrians, running on dedicated guide-ways. This track can lie on the ground, be elevated, subterranean or any combination of these modes. Automated Highway Systems (AHS) and their associated technologies (i.e. Adaptive or Intelligent Cruise Control, "Electronic

Tow-Bar", Platooning, etc.) are, arguably, an exception to the need for dedicated guide-ways; however, all systems known to the author require a specialised guidance system in a secluded lane or a human driver at least partially in charge of the required control tasks.

Individual transport systems

A commonly proposed form of individual ATS is the Personal Rapid Transit (PRT). It is an automated, demand responsive system consisting of several smaller vehicles travelling on dedicated guide-ways. The first projects involving PRT were introduced in the early seventies.

PRT systems typically employ smaller vehicles (e.g. 4-6 seats) and are intended to operate an on-demand point-to-point service without intermediate stops, thus providing some of the benefits of both the car and public transport. Current UK government policy considers PRT only as a 'niche' market for shopping complexes, theme parks, airports etc.

One example of PRT is the Urban Light Transport (ULTra). Its battery operated vehicles are intended to accommodate up to four passengers travelling at 40 km/h with 1 second headways. The track is to be partly elevated and partly at ground level and feature off-line stations (Lowson 2001). The city of Cardiff (UK) is the host for a 1 km test track. They plan to use the system with paying passengers by 2006.

Another well documented PRT system is SkyWeb Express, by Taxi 2000. The vehicles in this case would obtain the energy from the guide-way and use linear motors for propulsion. The guide-way would be elevated and designed to cause less visual intrusion. Each vehicle would carry up to three adults and travel at between 20 and 40 mph (*Skyweb Express* 2005).

The above systems are both to be publicly operated, but there are alternatives which allow the use of privately owned cars within the public system infrastructure. The idea is to be able to drive the car to a station to engage the vehicle to the public system, allowing it to take over control for the rest of the trip.

A project called Autoshuttle in Braunschweig, Germany is an example of such a system. It allows cars to drive into cabins, which are transported along a track using magnetic levitation, then drive out to continue their journey. A test track has been built at the technical University of Braunschweig (McDonald & Vöge 2002).

Another alternative project is the Rapid Urban Flexible. This system features a battery operated car that has a special slot to engage on a monorail track, from which it takes energy. A larger version can be operated as an automatic people mover (*RUF* 2005).

In addition to the previously cited examples, there are a vast number of other projects operating in leisure parks, airports, business complexes and car parking facilities. The operation principle is similar to that on a full system, but with a very limited range and impact on the transport industry.

Car sharing and car pooling programmes have been introduced in several cities since 1948 (Sefage in Zurich is an early example) (McDonald & Vöge 2002) with some success. This type of transport systems, together with demand responsive transport systems such as taxis, does not necessarily involve vehicle automation; however, they would benefit from it.

Mass transport systems

In addition to automated personal transportation systems, there are automated mass transportation systems, often called Automated People Movers (APM). These are the automatic versions of buses and trams, often relying upon technology which originated in the 1960s. According to McDonald & Vöge (2002), by June 2000, 23 APMs were operating at airports, 24 at leisure sites and 21 at institutional systems (subways, malls and universities). Today, the International Association of Public Transport (UITP) recognises 115 APMs in service around the world, 19 of them as driverless metros. Most of the people movers are smaller in scale and capacity than metros, and also include simple shuttle systems designed for distances of a few hundred meters.

Two well known early examples of APM systems are the one operating at Morgantown, USA since 1975, developed by Boeing, and the Light Automatic Vehicle (VAL) developed by Matra, in operation at Villeneuve d'Assq-Lille, France, since 1983 (Brader 1985). Both systems have an excellent record on safety and timing, and have proved that computer controlled systems are feasible and can be operated at competitive cost. The Docklands Light Railway is the best UK example of APM; having being inaugurated in 1987, it has being considerably extended and upgraded to current standards.

The growth in this form of transport around the world is evident. It is rapidly becoming mainstream as more governments are opting for this option and replacing older systems with automatic ones.

Automated highway systems

The Automated Highway System (AHS) was introduced in the General Motors Pavilion in 1939 during the World's Fair; however, the first research efforts were not conducted until the late 1950's by the Radio Corporation of America in collaboration with General Motors (Zworykin & Flory 1958, Gardels 1960). Later, the Ohio State University embarked on a long term research programme spanning from 1964 to 1980 on various aspects of automated highways. Many followed with similar programmes and significant progress was made in related areas (i.e. adaptive cruise control, platooning, information and control schemes, etc.). Technical limitations at the time forced the researchers to abandon the idea since it was deemed infeasible for any practical application; yet, research continued in related areas prompting the development of enabling technologies such as adequate sensors, communication protocols and computational tools. The AHS concept has been receiving renewed consideration in recent years as a means to reduce congestion and increase transport efficiency. Legal and liability issues have dampened the development of this mode of transport automation more than any other, forcing the implementation of the relevant technologies to be limited to driver assistance in commercial vehicles.

Freight transport

The freight transport sector has traditionally served the manufacturing and retailing industries, and is a highly competitive business. A healthy economy depends on the ability to distribute goods and services efficiently. Although there are only a few examples of vehicle automation in this area (mostly limited to manufacturing plants), some technologies such as traffic guidance and identification have been developed in order to improve the efficiency of the existing infrastructure. Such changes obey the developing needs introduced by new concepts in logistics management, such as multi-modal chain or just-in-time distribution.

The same technology and infrastructure utilised in personal ATSs can be used for freight operations, and the integration of these two areas in a single system may alleviate transportation problems in the future.

1.3 Proposed Research Area

The project which is described in this thesis is part of a broader research effort which intends to demonstrate the potential of automated close headway convoy systems to increase the capacity and flexibility of corridors such as conventional railway track, automated highways, guided busways or APM and PRT guideways. This strategy maximises infrastructure utilisation and throughput; however, as the vehicle separation is reduced, it places increasingly rigorous demands on the vehicles and requires extensive use of sensors, computing power and large communication networks. Additionally, control and traction equipment specifications required to achieve reasonable controllability and maintain or alter the relative positions of vehicles with respect to each other result in prohibitive costs for an operable system.

The aim of this project is to explore the application of inter-vehicle damping devices to close-headway convoy systems to enable zero spacing operation. The proposed damper would not only provide inherent safety in the case of catastrophic failure, but also assist the vehicle's control system and drivetrain, being an integral part of the control strategy. This permits the use of inexpensive equipment without compromises in safety or capacity. Since passenger comfort and ride quality can potentially be affected by the interactions among adjacent vehicles, it is required to develop suitable performance criteria taking into consideration human reactions to motion. The situations in which this technology is best suited are to be investigated, as are its limitations and infrastructure requirements.

The anticipated outcome of this work is to develop the required knowledge, tools and methodology for the design of zero headway ATSs irrespective of the peculiarities of different transportation modes, guidance systems or vehicle technologies.

1.4 Contributions of This Work

The main contribution of this work is the development of the 'bridging damper' concept to be the enabling technology for close headway convoy systems. The original damper from the system described by Perrott & Renfrew (1998) was a passive hydraulic system for which only a vague description and sketches (which were obtained through personal communication with the authors) existed. The system proposed here is a fully controllable device which not only can cope with emergencies but is also an integral part of the vehicles' control strategy also proposed in this work.

In addition to the theoretical information provided, the results have been verified through the development of simulation models such as those described in chapter 6. The models and modelling techniques have been made into a convoy simulation toolbox for the development and design of future systems.

Additional contributions of this work include:

- An extended analysis of the influence of different control parameters for close headway systems (see chapter 3). The analysis applies to a variety of control strategies including forward-looking only and bidirectional controllers.

- An analysis of safety and ride quality requirements for convoy systems (see chapter 4).

Chapter 2

Review on Close Headway Transportation Systems

2.1 Introduction

Current motorised ground transportation modes require large and costly infrastructure developments and the commitment of vast resources which remain largely underexploited due to the limitations inherent to each mode. For instance, the current simultaneous track usage for both rail and motorway is approximately 5% at maximum throughput, which represents a significant waste of land and investment (Perrott & Renfrew 1998). Congestion of both networks results in reduced capacity and, hence, increased journey times, which in turn, lead to inefficient usage of energy. The evidently increasing degree of automation being applied to all modes of modern transportation systems implies that there will also be an increased ability to manage resources and organise transit through the networks by reducing the impact of human unpredictability. This organisational ability conferred to a central network computer would reduce congestion, also achieving better transport integration through improved scheduling; thus enhancing the network's passenger throughput.

An ATS as described above in which automatic vehicles operate independently from each other and travel uniformly spaced at safe distances between them to avoid collisions would be a definite improvement from the current situation. Yet, the infrastructure would still remain under utilised. Close headway formations have been widely proposed in order to take full advantage of transport automation. Said "platooning" strategies allow increased simultaneous utilisation of track, resulting in improved traffic capacity while maintaining appropriate safety levels (Perrott & Renfrew 1998, Shladover 1978, Yanakiev & Kanellakopoulos 1996, Canudas de Wit & Brogliato 1999, Alvarez & Horowitz 1999). In addition, platoon configurations are appropriate when driving a string of vehicles with a single driver (electronic tow-bar), to reduce drivers' stress and fatigue in urban driving (stop-and-go situations) or when fuel efficiency is a factor, as aerodynamic drag is greatly reduced when operating at close inter-vehicle separation (Parent 2002, Canudas de Wit & Brogliato 1999, Michaelian & Browand 2000).

2.2 Vehicle Control Schemes

Research related to control techniques for ATS has led to the development of two main frames of reference in relation to the positioning of vehicles along the string. Point following (also known as synchronous model (Brader 1985) or moving cell (Peppard 1974)) requires that each vehicle follows a well defined target independently of the states of other vehicles. Its simplicity of implementation and stability make it attractive for system designers; however, it is inferior in terms of capacity, flexibility and adaptability compared to vehicle following schemes (also known as the asynchronous model (Brader 1985) or relative motion measurements (Peppard 1974)) (Shladover & Desoer 1991). On the other hand, possible flow instability, control complexity and implementation issues forced early researchers to switch to point following techniques (i.e. the researchers at Ohio State University abandoned the idea of vehicle following due to lack of adequate sensing equipment (Fenton & Nayhan 1991)). That trend has been reversed thanks to the availability of current sensor and communication technology, enabling increasingly shorter headways to be used, and hence making possible the application of platooning techniques.

The idea of platoon configuration is not new. Over 35 years back, Levine & Athans (1966) worked on the design of an optimal linear feedback system to regulate the position and velocity of a densely packed string of vehicles travelling at high speed. Although a proper definition of platoon was not introduced in this work, the authors acknowledged the serious restrictions imposed by the required sensing and communication capabilities in a long string.

The platooning concept was formalised shortly afterwards in the works of

Shladover (1978). The author suggested a group of consecutive vehicles maintaining minimal separation (30 cm. to 60 cm.) between them, controlled by a special vehicle-follower control system. The model consisted of a mass with a propulsion system having a first order lag and included a jerk limiter in the loop. Jerk and acceleration limits recommended for passenger comfort were found to be extremely restrictive, as they tended to produce instability along the string.

2.2.1 String stability

The concept of string stability was defined by Peppard (1974) as the property of the system to attenuate disturbances as they propagate along the string of vehicles. This original notion was somewhat intuitive in the sense that the necessary condition to achieve string stability was that the ratio of the transfer functions describing the deviations from the desired position of two adjacent vehicles, $|G_i(j\omega)/G_{i-1}(j\omega)| \leq 1$, for all ω. This study concluded that a PID feedback control using only velocity error and separation measurements can be designed to exhibit string stability if both forward and rearward separation measurements are employed.

String stability is further discussed in the works of Swaroop & Hedrick (1996), where formal mathematical definitions of l_∞ *string stable*, *asymptotically (exponentially) string stable* and l_p*-string stable* are introduced. In general, the p-norm $|\mathbf{x}|_p$ of an n-dimensional vector $\mathbf{x} = \langle x_1, x_2, \cdots, x_n \rangle$ is defined as:

$$|\mathbf{x}|_p = \left(\sum_i |x_i|^p \right)^{1/p}. \tag{2.1}$$

For an interconnected system $\dot{x}_i = f(x_i, x_{i-1}, \cdots, x_{i-r+1})$, the state $x_i = 0$ is l_p string stable if for all $E > 0$, there exists a δ such that

$$\|x_i(0)\|_p < \delta \Leftrightarrow \sup_t \left(\sum_1^\infty |x_i(t)|^p \right)^{\frac{1}{p}} < E. \tag{2.2}$$

Strict string stability (the complete notion of it where $p = \infty$) is considered to be a particular case of the definition in equation 2.2. Additionally, the state $x_i = 0$ is asymptotically string stable if it is string stable (in the l_∞ sense) and $x_i(t)$ tends to zero asymptotically.

In the specific context of strings of automated vehicles, the definition of string stability is applied to the spacing error propagation transfer function. If $G(s)$ is such a transfer function and $g(t)$ its equivalent in the time domain, then the necessary and sufficient condition for string stability is that its corresponding l_∞ induced norm remains less than one at all times (i.e. $\|g(t)\|_\infty < 1$ for all values of t).

It is important to note that $\|G\|_\infty$ is the frequency domain equivalent of the l_2 induced norm, (also known as root-mean-square (rms) norm), which is related to the energy of the system (Antoulas 2001). A system which is only l_2-string stable causes attenuation of the spacing error energy along the string, but not attenuation of the spacing error itself; it is therefore, a weaker condition than l_∞ (Canudas de Wit & Brogliato 1999). Swaroop, Chien & Ioannou (1994) showed that if the impulse response $g(t)$ is positive at all times it is possible to use frequency domain analysis to guarantee l_∞ string stability, in which case, the condition $\|g(t)\|_\infty < 1$ can be replaced by the equivalent condition

$$\|G\|_\infty < 1 \text{ if } g(t) \geq 0 \text{ for all } t; \tag{2.3}$$

otherwise time domain analysis must be used.

Although the formal definition of string stability implies the uniform boundedness of the states of an infinite system if the initial conditions are bounded, it is relevant to vehicle following schemes even if a finite number of vehicles is present. The reason for this is that the stability of the platoon must be independent of the number of vehicles. This ensures that actuator saturation does not occur in the presence of perturbations and that ride quality does not deteriorate as a result.

An additional interesting conclusion from the definition of string stability is that it can also be ensured for general interconnected systems if they present sufficiently "weak coupling" (Swaroop & Hedrick 1996). This condition means that there is a trade-off between traffic friendliness in terms of capacity and vehicle regulation performance. It also means that point following control strategies are string stable provided that the individual vehicles are stable. This is because point following strategies can be viewed as infinitely weak interconnected systems.

Early research reported in Shladover (1978) and Garrard, Kornhouser, MacKinnon & Brown (1978) opened the discussion on two major aspects of vehicle controller design which influence string stability. The former pointed out the relevance of providing information about the leading vehicle to the rest of the platoon, which requires vehicle to vehicle communications, while the later showed that speed dependent spacing can be used to ensure string stability at the expense of reduced capacity. These design aspects can be classified in the following two categories:

- Inter-vehicle spacing policies.
- Vehicle controller information strategies.

Inter-vehicle spacing policies

There is a variety of spacing policies which researchers working on vehicle following schemes have explored since the concept of ATS was introduced. The most obvious reason for introducing a well defined spacing policy is, naturally, to keep the vehicles at safe distance between them. The exact definition of *safe distance* has been widely debated. Hajdu, Gardiner, Tamura & Pressman (1968) defined a safety factor K as:

$$K = \frac{\textit{Minimum vehicle to vehicle distance}}{\textit{Stopping distance at emergency deceleration}}. \tag{2.4}$$

Early transportation planners considered the *brick wall criterion* as the safest possible solution. This criterion dictates that vehicles are kept at such distance from each other that if the leading vehicle stops instantaneously, the following vehicle would be able to stop without colliding (i.e. $K > 1$). Fixed block signalling used widely in current railway systems still relies on this criterion. Moving cell systems and road vehicles on the other hand, constantly violate this criterion in recognition that vehicles do not stop instantaneously. This approach yields better vehicle throughput and is only limited by the reaction times of drivers and vehicles and their deceleration capacity.

There is, however, another important reason for studying spacing policies in vehicle automation. The spacing policy generates a dynamic model of the ideal vehicle performance during station keeping (Olson & Garrard 1979). Such a model must exhibit string stability. The general consensus is that string stability is not achievable when the desired inter-vehicle spacing is constant even if the complete set of dynamic states of the preceding vehicle only are available at any time. The reason for this is that the poles and zeros of the transfer function which represents this spacing strategy are coupled in a way which prevents them being positioned to satisfy the conditions for string stability. Take, for example, the magnitude of the transfer function derived by Yanakiev & Kanellakopoulos (1996) for this case:

$$|G(s)| = \left| \frac{\frac{c}{m}s + \frac{k}{m}}{s^2 + \frac{c}{m}s + \frac{k}{m}} \right|, \tag{2.5}$$

where m is the vehicle's mass and k and c are the position and velocity error gains respectively. Both errors are relative to the preceding vehicle exclusively. By substituting $s = j\omega$ in equation 2.5 it can be concluded that $G(j\omega) > 1$ at frequencies in the range $0 < \omega < \sqrt{2k/m}$; therefore, string stability is not attainable regardless of the choice of k and c.

A second spacing policy which has been studied extensively is the *"constant time headway"* strategy. The desired inter-vehicle separation is made to vary linearly with velocity so that time headway is constant. This can be accomplished by selecting the desired separation to be $\delta = hv + \delta_0$, where h is the time headway, v is the vehicle's velocity and δ_0 is a constant minimum separation. Garrard et al. (1978) and Chien & Ioannou (1992.) showed that this kind of speed dependent spacing yields string stability. Using this policy Yanakiev & Kanellakopoulos (1996) found the magnitude of the transfer function to be described by:

$$|G(s)| = \left| \frac{\frac{c}{m}s + \frac{k}{m}}{s^2 + \frac{c+kh}{m}s + \frac{k}{m}} \right|. \tag{2.6}$$

The time headway introduces additional damping acting between the reference coordinate system and the vehicle (as opposed to between two adjacent vehicles). In this case the poles and zeros can be positioned independently in a suitable manner to achieve string stability. The condition for $|G(j\omega)| < 1$ for all $\omega > 0$ is $c > \frac{2m - kh^2}{2h}$. This condition was already reported in previous studies (Ioannou & Xu 1994). Although this technique suffers from reduced vehicle throughput it is suitable in applications where no communications are available and each vehicle relies on its own sensing equipment to determine the states of its predecessor. Intelligent cruise control is one example of such systems.

A variation of the constant time headway in which the term h is made to vary linearly with the relative velocity between the vehicle and its predecessor (i.e. $h = h_0 - c_h v_r$, where $h_0 > 0$ and $c_h > 0$ are constants and v_r is the relative velocity between the vehicle and its predecessor) has also been explored in the literature (Yanakiev & Kanellakopoulos 1995). This has the advantage of allowing tighter spacing than the constant headway controller since the value of h_0 to ensure string stability is much smaller than the constant h. This is especially true for vehicles with low actuation to weight ratios such as heavy goods vehicles. It is important to note that the value of h must be restricted to stay in a well defined range through saturations to avoid instability or gaps that are too large. Looking for an example, Yanakiev & Kanellakopoulos (1996) state that the magnitude of the transfer function using this strategy after linearising around the operation point (all the vehicles travelling at the desired speed v_d) is given by:

$$|G(s)| = \left| \frac{\frac{c+kc_hv_d}{m}s + \frac{k}{m}}{s^2 + \frac{c+kh_0+kc_hv_d}{m}s + \frac{k}{m}} \right|. \tag{2.7}$$

The condition to guarantee string stability is then $c > \frac{2m-kh0^2-2kh_0c_hv_d}{2h_0}$, which indeed allows better traffic performance than with fixed time headway thanks to the additional term $2kh_0c_hv_d$.

In early research projects (Olson & Garrard 1979) a *"constant safety factor"* spacing strategy was considered. The desired spacing varies quadratically with speed so that separation is proportional to the safe stopping distance. The desired spacing can be defined by $\delta = \frac{K_sv^2}{2A_e}$, where K_s is the safety factor and A_e is the minimum guaranteed emergency deceleration. This non-linear technique also yields adequate string stability, but is seldom considered in modern research, probably due to the poor traffic friendliness as compared to other spacing policies.

Information strategies

As in the case of spacing policies, there have been numerous studies into the effects of the kind and quality of information made available to each member of a group of automated vehicles.

Shladover (1978) described a vehicle following controller which relies on the use of the *preview information* from the first vehicle in the platoon to complement the information from the immediate predecessor. The author used the velocity of the leading vehicle to introduce an absolute velocity feedback loop with gain c_l, which has a similar effect in the string stability to the constant time headway approach. In fact the only change in equation 2.6 is the substitution of the term

$\left(\frac{c+kh}{m}\right)s$ in the denominator for $\left(\frac{c+c_l}{m}\right)s$, giving a transfer function with magnitude

$$|G(s)| = \left| \frac{\frac{c}{m}s + \frac{k}{m}}{s^2 + \frac{c+c_l}{m}s + \frac{k}{m}} \right| \tag{2.8}$$

Similarly, the necessary condition for string stability in this case becomes $c > \frac{2km-c_l^2}{2c_l}$. It would appear that these two control methods perform equally well in terms of string stability and spacing regulation, yet, in practice, they imply opposite design considerations. For instance, under the constant time headway policy each vehicle can obtain the information it requires by itself, but requires comparatively large inter-vehicle spacing. Conversely, accessing the leader's velocity requires constant communications but the spacing is irrelevant for the control design, allowing better capacity.

The technique just described can be extended such that the leader transmits its acceleration to the rest of the vehicles along with its velocity (Sheikholeslam & Desoer 1990). This modification allows better space regulation and is especially beneficial to vehicles with significantly slow actuation time constants (see section 3.5 of this work).

An interesting special case of the original strategy is to completely disregard the velocity of each vehicle's immediate predecessor (but still use its relative position information) and get relative velocity feedback only from the platoon leader (Eyre, Yanakiev & Kanellakopoulos 1997). Although it may seem prejudicial at first, a closer examination of the transfer function in equation 2.8 reveals that setting $c = 0$ removes the zero; thus improving the magnitude response and ensuring positive impulse response, regardless of the choice of the other parameters.

A further modification of the original strategy is to transmit the leader's desired velocity instead of the actual one (Eyre et al. 1997). Since the spacing error propagation transfer function is identical to the original, the obvious result is that, presumably, this technique requires less frequent data exchanges. However, there are additional benefits: In the absence of perturbations, the inter-vehicle spacing errors will always be zero once the platoon reaches steady state, despite the inputs applied to the leader because all the vehicles receive the same command at the same time as the leader. Moreover, this is also the case when actuation delays are included (provided that all vehicles are identical and that communications are perfect). This modification is, in fact, a way of coordinating manoeuvres without introducing perturbations to the system.

The *bidirectional* strategy used originally in Peppard (1974) offers another way of achieving string stability. This scenario involves vehicles receiving relative velocity and relative position information about both their immediate predecessor and immediate follower. The technique was later investigated by Chein, Zhang & Cheng (1995), concluding that close spacing is achievable and performance is improved compared to time headway techniques. A clear advantage is that transmission of data between vehicles is not necessary since the relevant information can be obtained autonomously by each vehicle. To arrive at this conclusion, the authors consider only acceleration manoeuvres for a small number of vehicles. Later studies into this topic (Yanakiev & Kanellakopoulos 1996, Eyre et al. 1997, Zhang, Kosmatopoulos & Ioannou 1999) reveal the limitations of the technique. In fact, the strict definition of string stability (Swaroop & Hedrick 1996), implies that it is impossible to achieve l_∞ string stability. Because of the *interconnectedness* of this system, each vehicle is affected not only by its preceding vehicles as in *forward-looking* only scenarios, but also by its following vehicles. The resulting spacing error propagation transfer function is therefore a recursive one. Since the last vehicle (vehicle n in a platoon with n vehicles) is unique in the string because it only looks forward (Yanakiev & Kanellakopoulos 1996), the magnitude of its transfer function is

$$|G_n(s)| = \left| \frac{\frac{c}{m}s + \frac{k}{m}}{s^2 + \frac{2c}{m}s + \frac{2k}{m}} \right|. \tag{2.9}$$

The magnitude of the transfer functions of the other vehicles is then

$$|G_{n-i}(s)| = \left| \frac{G_n(s)}{1 - G_n(s)G_{n-i+1}(s)} \right|, i = 1 \ldots n-1. \tag{2.10}$$

Clearly the conditions for $|G_i(s)| < 1$ for all $\omega > 0$ become complicated requiring numerical analysis. Yanakiev & Kanellakopoulos (1996) indicate that the necessary and sufficient condition is that $c^2/km > C_i$, where C_i is a constant whose value increases as the number of vehicles increases. It is however certain that the steady state gains $G_{n-i}(0) < 1$ for every vehicle in the platoon for any finite number of vehicles. Since the poles and zeros in equation 2.9 cannot be placed independently, positive impulse response is unattainable. As a result, only l_2 string stability can be assured through the use of frequency domain analysis.

A further investigation of the effects of the different information strategies on string stability is reported in (Swaroop & Hedrick 1999). It is reported that making available the information of a reference vehicle yields the best robustness of the system. If it is not available or the delays are too large, splitting the platoon into miniplatoons is beneficial.

Information and spacing policy combinations

There are in the literature multiple examples of innovative combinations of spacing policies and information strategies to get the advantages of both. For instance, as in the forward-looking scenario, positive impulse response can be achieved in the bidirectional scenario by introducing a time headway. Yanakiev & Kanellakopoulos (1996) proposed three possible constant time headway spacing policies in the bidirectional scenario, but it has been confirmed (Eyre et al. 1997) that varying only the desired spacing from the preceding vehicle as a function of velocity (i.e. the same strategy as described in section 2.2.1) yields the best results, by easing the conditions for string stability. Information from the leader can, presumably, be used to achieve the same results.

Another variation of the forward looking scenario in combination with a constant time headway policy is proposed by *Two-Vehicle Look-Ahead Convoy Control Systems* (2004). It investigates a situation in which each vehicle looks not only to its immediate predecessor, but two or more vehicles in front. The authors modelled and simulated a string of vehicles with a two vehicle look-ahead controller achieving good performance in the presence of large perturbations within the platoon.

2.2.2 Vehicle controller architecture, design and implementation

Olson & Garrard (1979) discussed controllers which generate acceleration or jerk commands, concluding that jerk controllers develop instabilities even at large headways when saturation of the jerk and acceleration limiter is reached, thus favouring controller designs which generate acceleration commands.

Canudas de Wit & Brogliato (1999) recognise that the definition of string stability does not automatically ensure collision avoidance among the vehicles, and only a few authors (Li, Alvarez & Horowitz 1997, Alvarez & Horowitz 1999) have addressed these problems. In their paper, Canudas de Wit & Brogliato (1999), redefined the problem by incorporating restrictions using state-space constraints.

They proposed a control law that openly looks for vehicle collisions (possibly using inter-vehicle dampers), as they help the dissipation of excessive energy and can be effectively used to ensure stability.

Studies by Rajamani, Choi, Law, Hedrick, Prohaska & Kretz (2000) used hierarchical control architecture design applying the sliding surface method of controller design described by Slotine (1991). It was possible to design the upper level controller for the desired acceleration

$$\begin{aligned}\ddot{x}_{i\_des} &= (1-C)\ddot{x}_{i-1} + C\ddot{x}_l - (2\xi - C(\xi + \sqrt{\xi^2-1}))\omega_n\dot{e}_i \\ &\quad -(\xi + \sqrt{\xi^2-1})\omega_n C(v_i - v_l) - \omega_n^2 e_i. \end{aligned} \tag{2.11}$$

by defining a sliding surface which depends on the relative position and velocity errors e_i and $\dot{e}_i$ respective to the preceding vehicle and the relative velocity from the leader x_l, and ensuring that the vehicle states converge to the defined sliding surface. The control gains to be tuned were the weighting of the lead vehicle's speed and acceleration (C), the damping ratio (ξ) and the controller bandwidth (ω_n). The lower level controller is in charge of producing the inputs for the drivetrain to produce the desired acceleration. Swaroop & Hedrick (1996) proved the robustness of this upper level controller to lags introduced by the lower level controller. This work acknowledged the difficulties caused by gear changes under automatic control. To prevent the large error build up and actuator saturation that are caused by consecutive gear shifts, the powertrain control module was modified so that gear shift points became purely a function of the engine speed instead of engine speed and throttle position. Also, the value of w_n was scheduled to be dropped to 50 % of its value during the gear shift, to return gradually to its original value.

2.2.3 Sensor and communication considerations

A typical information structure for a close headway automated system is shown in figure 2.1. Only the links essential for the control of vehicles are shown, with the emergency links depicted in discontinuous lines. Naturally, the emergency links must be available separately and at all times. The links between the section controllers and the central network can be a physical link such as optic fibre or cable, but the communications to platoons need to be wireless. The two possibilities available are:

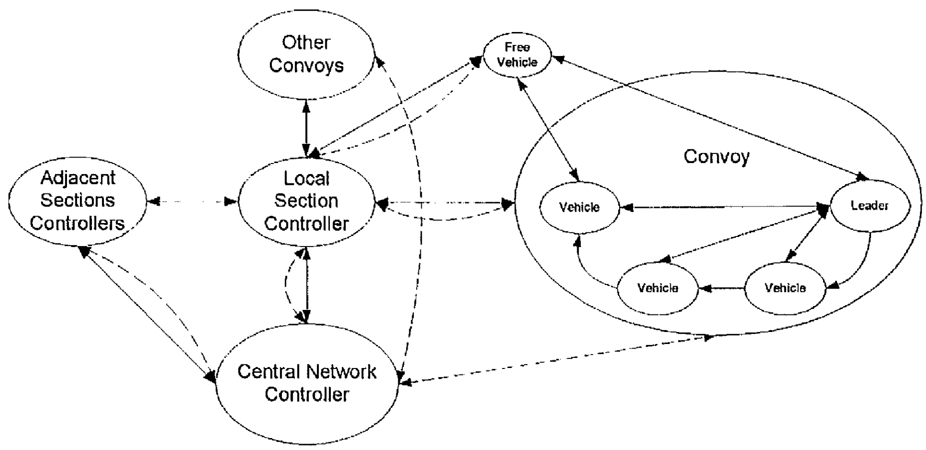


Figure 2.1: Typical information structure

1. *Cooperative vehicles* which communicate their information to the rest of the vehicles.

2. *Independent vehicles* which gather the information relevant to them through the use of sensors.

Both systems would produce identical results if communications and sensing were perfect.

2.2.4 Communications

Researchers have proposed a variety of vehicle to vehicle communication technologies, including infrared rays (e.g. Fujii, Hayashi & Nakagata 1996) and radio based systems such as Wireless Token Ring Protocol (WTRP) and WaveLAN (i.e. IEEE 802.11 protocol) (e.g. Ergen, Lee, Sengupta & Varaiya 2003). Recent approaches tend to prefer the radio based systems to avoid sensitivity to dirt, weather conditions and misalignment. Interference between platoons is a problem with radio based systems and there is no guarantee that the information will reach its target on time. Liu, Mahal, Goldsmith & Hedrick (2001) showed that delays and drop information packets normally introduced by communication networks destroy string stability if the vehicle controllers are triggered by the reception of data from either the lead or the predecessor vehicle's information, hence requiring that all the vehicles are synchronised to update their controllers at the same time. Additionally, wireless communications can leave the system vulnerable to electronic attacks.

A considerable bandwidth is consumed even if sensors are used to measure the preceding vehicle's range and range rate. This is especially true for large capacity systems in which a number of platoons operate close by. It is estimated that the maximum acceptable delay in the spacing regulation layer is one order of magnitude smaller than the main time constant of the vehicle dynamics (Sachs & Varaiya 1993). In (Shladover & Desoer 1991), inter vehicle communications are modelled as a time delay of 0.01 s. It was thought that it would take 40 bits for the communication of the speed and acceleration of the preceding vehicle and the platoon leader at adequate accuracy, but a further 60 bits were required for error detection and other information required for safe operation. Communication of the preceding vehicle's information was used instead of range sensing because of the better accuracy and reduced possibility of noisy measurements, especially acceleration. The bandwidth required, though, was calculated at 10 000 b/s per vehicle under ideal conditions. A directional (optical) communication system in which each vehicle stored and passed the information along the string was then developed.

2.2.5 Sensing equipment

There is a trade-off between ride quality and spacing regulation due to the inaccuracy of current range and range-rate sensors. Ultrasonic sensors are widely available in the automotive industry, but limited to ranges around 1 to 3 meters due to the atmosphere's absorption properties. Additionally, they provide insufficient information about the target object and can be affected by echo. They are therefore more suitable to near environment detection at low speeds (i.e. parking assistance). Laser scanners produce accurate measurements for ranges well over 100 m at rates between 10 to 50 Hz but suffer from inability to detect objects out of the scanning plane. Moreover, their susceptibility to bad weather (fog, rain and snow), dusty conditions and contaminants in their optical system renders them unusable for normal travel scenario.

Currently the most widely available range and range rate sensing technology for transportation purposes is the microwave radar. The typical Frequency Modulated Continuous Wave (FMCW) automotive radar is able to measure distances at ranges around 100 m with resolution around 0.78 m with accuracy of 0.11 m in range and resolution of 2.8 m/s and accuracy of 0.39 $/s$ in range rate (Russell, Crain, Curran, R., Drubin & Miccioli 1997). They provide sufficient information about the target and can track multiple targets (David 1997). This type of radar can be found in modern production vehicles equipped with Adaptive Cruise Control (ACC). The noise content of the output signal of this type of sensor (particularly range-rate measurements) requires to be filtered with a low-pass filter with time constant of the order of 200 ms (Rajamani & Shladover 2001). This time lag, combined with the pure time delay caused by the extensive signal processing required (i.e. multiple target discrimination, reflection elimination, etc.) limit the usable bandwidth of this sensor. Additionally, long range (79 GHz band) FMCW radar is unable to produce reliable measurements at short separation of the order of 1 to 3 meters. Short range microwave radars (24 Ghz band) with superior accuracy characteristics are currently under development to be used in several automotive applications (Parent 2002) and have been exempted from wireless telegraphy licensing (TSO 2005) . Doppler radar produces relatively clean rate measurements, but is unable to detect objects which have low relative velocity.

Vision based range sensing has also been proposed for ATSs (Broggi, Bertozzi, Fascioli, Guarino Lo Bianco & Piazzi 2000, Aufrere, Marmoiton, Chapuis, Collange & Derutin 2001, Bertozzi, Broggi, Cellario, Fascioli, Lombardi & Porta 2002). This technology has been successfully tested under carefully controlled conditions. It is potentially the best suited for obstacle detection in mixed traffic contexts because of the wide field of view and the ability to detect smaller objects as well as pedestrians or obstacles with unusual or unforeseeable shape (Franke, Gavrila, Gorzig, Lindner, Puetzold & Wohler 1998). Visual systems are convenient where pedestrians are expected since no potentially harmful radiation emission is necessary. In addition to the susceptibility to bad weather, limited visibility conditions and contaminants on the lenses, the main limitation of vision systems for use in ATSs is the computational power required to process the large amounts of information generated (around 7 million pixels per second is typical) without introducing unreasonably long time delays. This is especially true for complex uncontrolled conditions with a multitude of textures, colours, objects and lighting conditions, where object detection algorithms tend to be unreliable.

2.3 Limitations of Short Headway Systems

In early research (Pue 1978) it was considered that at time headways under three seconds, a fundamental kinematic constraint arises. Essentially, the minimum allowable spacing is a function of the trailing and preceding vehicle velocities, and the future manoeuvre capability of each vehicle (i.e., the preceding vehicle decelerating while its follower approaches at a higher velocity). Consequently, the requirement for the vehicle response at such short headways would be inconsistent with comfort criteria, since the system's response to initial conditions can lead to violations of such criteria. Additionally, merge manoeuvres can cause unacceptable high jerk and accelerations if applied to constant gain controllers (Chiu, Stupp & Brown 1977). As a result, explicit kinematic constraints were considered necessary. Carbaugh, Godbole & Sengupta (1998) studied the safety and capacity of different strategies of transport automation and concluded that short headway platoons are effective in increasing the capacity of the system while reducing the severity of collisions, however, the probability of their occurrence increases as the spacing is reduced.

Shladover (1991) examines the question of effects of noise, jerk and acceleration limits, sampling time and time constant effects on vehicle performance. It concludes that noise has a noticeable effect on jerk and acceleration, but position and velocity regulation are hardly affected. Acceleration and Jerk limits proved restrictive, possibly causing oscillations under severe manoeuvres. The study also proves the convenience of introducing jerk limits to the commands. In addition, it was found that controller design is extremely sensitive to slower propulsion dynamics (i.e. internal combustion vehicles) requiring relative acceleration feedback to augment performance and maintain stability. The minimum safe no collision headway has also been found particularly sensitive to the vehicle's reaction delay (Fenton 1979).

To overcome difficulties associated with non-homogeneous vehicle response capabilities, Bae & Gerdes (2000) investigated the use of reference models which are consistent with the capabilities of each vehicle and command modification. In this architecture, the leader collects information from the rest of the platoon members and broadcasts the parameters of a reference model. Additionally, the commands are modified so that they are achievable for every platoon member.

The adaptiveness of this strategy permits adequate inter-vehicle space regulation, but there is an increased risk of adversely affecting the overall platoon performance, thus requiring larger manoeuvring distance and inter-platoon spacing, limiting the overall throughput.

2.4 Examples of Projects Incorporating Close Headway Systems

Early ATSs studies predicted that the bulk of development necessary to deploy actual automated systems would take place during the 1990s, but a reduction in funding and interest from governments resulted in a long delay to the original predictions (Bender 1991). However, renewed interest in the 1990s have started a number of development programs during the last decade.

PATH (Partners for Advanced Transportation and Highways) (PATH 2005) is a consortium involving several universities, governmental institutions and private companies from the state of California, U.S. It is administered by the Institute of Transportation Studies (ITS), University of California, Berkeley, in collaboration with Caltrans. One of its main interests is Advanced Vehicle Control Systems (AVCS) (see Horowitz & Varaiya (2000)). They have succeeded in forming an eight vehicle platoon consisting of standard road cars on the interstate highway near San Diego (I-15) at a fixed distance of 6.5m (DEMO '97). The controller was able to maintain the longitudinal distance as well as the lateral control . They have also demonstrated the use of the technology with heavy duty vehicles and 'bus-trains' (Ching-Yao 2003).

The European CHAUFFEUR program implemented close-headway 'electronic tow-bar' operations on two Mercedes trucks during its first phase of research (1996-99).CHAUFFER II, led by Daimler-Chrysler and partners IVECO and Renault V.I., among others further developed this work by implementing platooning techniques for several vehicles and made advances in legal and liability implications (Chauffeur 2005).

Renault I.V., in collaboration with The Laboratoire d'Automatique de Grenoble among others, have been working in a project called PlaTooN which aims at a semi-automatic guidance system where drivers are partially assisted by some automatic features such as automated vehicle following (L.A.G. 2005).

The CyberCars (Cybercars 2005) project has as its principal objective to bring all the European actors of this field together in order to test and exchange best practices, share some of the development work and progress faster in the experiments. This project has considered the use of convoys and some of its partners have already worked with them (i.e. RUF and Serpentine S.A) with some success (McDonald & Vöge 2002). The final project presentation was made in Antibes, France, in 2004 where they demonstrated the technologies developed including platooning using visual servoing.

2.5 Chapter Summary

The relevant literature concerning vehicle longitudinal automation for short headway systems was reviewed in this chapter. Capacity, infrastructure utilisation, fuel economy, safety, convenience and driver stress reduction were identified as positive features of short headway automated transportation systems. The main complications in the development of such transportation schemes encountered in the existing literature include communication and sensing issues, flow stability, concerns over compromised ride quality and the increased risk of minor collisions among adjacent vehicles.

String stability is regarded as a key property of vehicle following systems. The major factors affecting it are the kind of information made available to each platoon member about the states of nearby vehicles and the inter-vehicle spacing policy adopted. Requiring that the position error propagation along the string of vehicles exhibits positive impulse response has two main advantages; precluding oscillations in the string and allowing the use of frequency domain analysis to ensure string stability.

It is widely accepted that string stability is impossible to achieve under constant spacing with information only from the preceding vehicle's states. Bidirectional control strategies can only ensure l_2 string stability, which is a weaker condition than string stability.

Chapter 3

Extended Analysis of Close Headway Spacing Strategies

3.1 Introduction

A comparison of inter-vehicle spacing strategies and the conditions which yield positive regulation of spacing error within a string of vehicles is provided in this chapter. This analysis is an extension of the principles studied in the previously reviewed literature. The aim is to better understand the effects of the design parameters of vehicle controllers and imperfections of real systems (i.e. delays, noise, lags and saturations) to prove the necessity of bridging dampers and study the situations where they are beneficial.

Simplified linear representations of convoy vehicles are used to design the upper level controller in a hierarchical vehicle following control structure. This approximation makes possible the analysis of control laws of convoys involving several vehicles. Previous research developed by collaborators of this work (Ferreira-Rodriguez, Renfrew & Brunn 2003) suggests that it is acceptable to simplify the drives of electric vehicles using a first order lag drivetrain if jerk and acceleration limitations are present. The highly non-linear models on which this assumption rests are used in following chapters to verify the conclusions derived from this analysis. The controller nomenclature used is described in table 3.1. Any coherent set of units may be utilised in this analysis; however, S.I. units will be used throughout the reminder of this thesis.

Variable	Definition
n	Number of vehicles in the convoy,
m_i	mass of the ith vehicle,
l_i	length of the ith vehicle,
τ_i	time constant of the ith vehicle's drivetrain,
x_i	ith vehicle's position,
$\dot{x}_i$ or v_i	ith vehicle's velocity,
$\ddot{x}_i$ or $\dot{v}_i$	ith vehicle's acceleration,
$\dddot{x}_i$	ith vehicle's jerk,
x_L	leading vehicle's position
F_{Wi}	traction force at the ith vehicle's wheels,
u_i	ith vehicle's controller output (demanded force),
δ_i	desired separation between vehicles i and $i-1$,
δ_{0i}	desired velocity independent inter-vehicle separation,
h_i	time headway between vehicles i and $i-1$,
e_i	ith vehicle's position deviation from δ_i
α', β' and γ'	forward looking controller constants
α'_r, β'_r and γ'_r	backward looking controller constants (bidirectional scenario)
ϵ', κ' and ζ'	"leader looking " controller constants

Table 3.1: Convoy nomenclature.

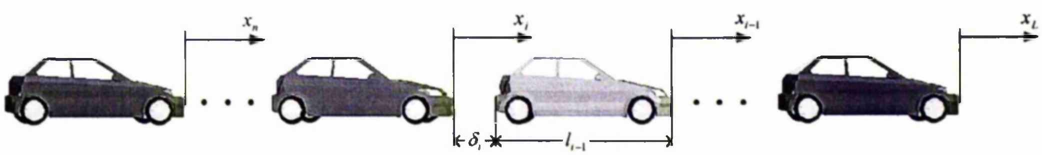


Figure 3.1: Convoy configuration.

3.2 Analysis Framework

Consider the scenario depicted in figure 3.1. A convoy of n closely spaced vehicles is following a leader L. Each vehicle acquires the information it requires about its surrounding vehicles trough a combination of sensors and communication links. The spacing regulation controller is designed in accordance with section 2.2. The parameters which have been considered relevant for this work are: relative acceleration, velocity and position gains with respect to neighbouring vehicles and to the leader, time headway and drivetrain's time constant. This study considers first the forward-looking scenario and secondly, the bidirectional scenario.

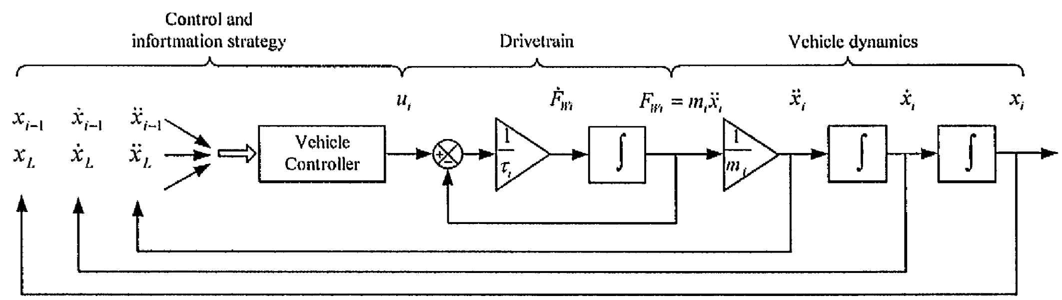


Figure 3.2: Forward Looking Linear Model.

3.3 Forward Looking Scenario

A simplified vehicle model featuring the first strategy is illustrated in figure 3.2. The vehicle controller receives information from its immediate predecessor and the platoon leader only. It is assumed that the powertrain behaves linearly as a first order lag with time constant τ_i, obeying the dynamic equation:

$$\dot{F}_{Wi} = \frac{u_i - F_{Wi}}{\tau_i}. \tag{3.1}$$

Its output acts to accelerate the equivalent mass of the vehicle which includes inertia from the rotating components within the vehicle's powertrain. No other forces are considered at this stage. The motion equation is:

$$F_{Wi} = m_i \ddot{x}_i. \tag{3.2}$$

Substituting equation 3.2 in 3.1 produces the system describing equation:

$$\dddot{x}_i = \frac{u_i - m_i \ddot{x}_i}{\tau_i m_i}. \tag{3.3}$$

The nose-to-tail spacing policy can be defined as

$$\delta_i = \delta_{0i} + h\dot{x}_i, \tag{3.4}$$

and a vehicle following controller action which considers all the relevant variables can be defined as

$$\begin{aligned} u_i \;=\;& \gamma_i'\left(\ddot{x}_{i-1}-\ddot{x}_i\right)+\beta_i'\left(\dot{x}_{i-1}-\dot{x}_i\right)+\alpha_i'\left(x_{i-1}-x_i-\delta_i-l_{i-1}\right) \\ &+\zeta_i'\left(\ddot{x}_L-\ddot{x}_i\right)+\kappa_i'\left(\dot{x}_L-\dot{x}_i\right)+\epsilon_i'\left(x_L-x_i\right). \end{aligned} \tag{3.5}$$

Substituting equation 3.5 in equation 3.3 and rearranging produces the following equation of motion:

$$\begin{aligned} &\tau_i m_i \dddot{x}_i+\left(m_i+\gamma_i'+\zeta_i'\right)\ddot{x}_i+\left(\beta_i'+\kappa_i'+\alpha_i' h_i\right)\dot{x}_i+\left(\alpha_i+\epsilon_i'\right)x_i \\ &-\gamma_i'\ddot{x}_{i-1}-\beta_i'\dot{x}_{i-1}-\alpha_i' x_{i-1}=\zeta_i'\ddot{x}_L+\kappa_i'\dot{x}_L+\epsilon_i' x_L-\alpha_i'(\delta_{0i}+l_{i-1}). \end{aligned} \tag{3.6}$$

Note that the vehicle immediately behind the leader is different from the rest since its immediate predecessor is also the leader, and its equation of motion is:

$$\begin{aligned} &\tau_1 m_1 \dddot{x}_1+\left(m_1+\gamma_1'+\zeta_1'\right)\ddot{x}_1'+\left(\beta_1'+\kappa_1'+\alpha_1' h_1\right)\dot{x}_1+\left(\alpha_1'+\epsilon_1'\right)x_1 \\ &=\left(\zeta_1'+\gamma_1'\right)\ddot{x}_L+\left(\kappa_1'+\beta_1'\right)\dot{x}_L+\left(\epsilon_1'+\alpha_1'\right)x_L-\alpha_1'(\delta_{0i}+l_L). \end{aligned} \tag{3.7}$$

Both equations 3.6 and 3.7 can be divided by $\tau_i m_i$ to produce

$$\begin{aligned} &\dddot{x}_i+\overbrace{\left(\frac{1}{\tau_i}+\gamma_i+\zeta_i\right)}^{a_{3i-2}}\ddot{x}_i+\overbrace{\left(\beta_i+\kappa_i+\alpha_i h_i\right)}^{a_{3i-1}}\dot{x}_i+\overbrace{\left(\alpha_i+\epsilon_i\right)}^{a_{3i}}x_i \\ &-\gamma_i\ddot{x}_{i-1}-\beta_i\dot{x}_{i-1}-\alpha_i x_{i-1}=\zeta_i\ddot{x}_L+\kappa_i\dot{x}_L+\epsilon_i x_L-\alpha_i\left(\delta_{0i}+l_{i-1}\right) \end{aligned} \tag{3.8}$$

and

$$\begin{aligned} &\dddot{x}_1+\overbrace{\left(\frac{1}{\tau_1}+\gamma_1+\zeta_1\right)}^{a_1}\ddot{x}_1+\overbrace{\left(\beta_1+\kappa_1+\alpha_1 h_1\right)}^{a_2}\dot{x}_1+\overbrace{\left(\alpha_1+\epsilon_1\right)}^{a_3}x_1 \\ &=\left(\zeta_1+\gamma_1\right)\ddot{x}_L+\left(\kappa_1+\beta_1\right)\dot{x}_L+\left(\epsilon_1+\alpha_1\right)x_L-\alpha_1\left(\delta_{01}+l_L\right), \end{aligned} \tag{3.9}$$

where the constants $\alpha_i, \beta_i, \gamma_i, \epsilon_i, \kappa_i$ and ζ_i represent $\frac{\alpha_i'}{\tau_i m_i}, \frac{\beta_i'}{\tau_i m_i}, \frac{\gamma_i'}{\tau_i m_i}, \frac{\epsilon_i'}{\tau_i m_i}, \frac{\kappa_i'}{\tau_i m_i}$ and $\frac{\zeta_i'}{\tau_i m_i}$.

By defining

$$\begin{aligned} X_1 &= x_1, & X_{3i-2} &= x_i, \\ X_2 &= \dot{x}_1-b_1 x_L & X_{3i-1} &= \dot{x}_i-b_{3i-2}x_L \\ &= \dot{X}_1-b_1 x_L, & &= \dot{X}_{3i-2}-b_{3i-2}x_L, \end{aligned}$$

$$
\begin{aligned}
X_3 &= \ddot{x}_1 - b_1\dot{x}_L - b_2 x_L & \qquad X_{3i} &= \ddot{x}_i - b_{3i-2}\dot{x}_L - b_{3i-1}x_L \\
&= \dot{X}_2 - b_2 x_L, & &= \dot{X}_{3i-1} - b_{3i-1}x_L,
\end{aligned}
\tag{3.10}
$$

$$
Y_1 = x_L - x_1 - \delta_{01} - l_L, \qquad Y_i = x_{i-1} - x_i - \delta_{0i} - l_{i-1} \tag{3.11}
$$

and

$$
\begin{aligned}
b_1 &= \gamma_1 + \zeta_1, & \qquad b_{3i-2} &= \gamma_i + \zeta_i, \\
b_2 &= \beta_1 + \kappa_1 - a_1 b_1, & b_{3i-1} &= \beta_i + \kappa_i - a_{3i-2}b_{3i-2}, \\
b_3 &= \alpha_1 + \epsilon_1 - a_1 b_2 - a_2 b_1, & b_{3i} &= \alpha_i + \epsilon_i - a_{3i-2}b_{3i-1} \\
& & &\quad - a_{3i-1}b_{3i-2}
\end{aligned}
\tag{3.12}
$$

for $i = 1 \ldots n$, the convoy can be fully represented by

$$
\begin{bmatrix} \dot{X}_1 \\ \dot{X}_2 \\ \dot{X}_3 \\ \vdots \\ \dot{X}_{3i-2} \\ \dot{X}_{3i-1} \\ \dot{X}_{3i} \\ \vdots \\ \dot{X}_{3n-2} \\ \dot{X}_{3n-1} \\ \dot{X}_{3n} \end{bmatrix}
=
\begin{bmatrix}
0 & 1 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\
0 & 0 & 1 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\
-a_3 & -a_2 & -a_1 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\
\vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\
0 & 0 & 0 & \cdots & 0 & 1 & 0 & \cdots & 0 & 0 & 0 \\
0 & 0 & 0 & \cdots & 0 & 0 & 1 & \cdots & 0 & 0 & 0 \\
\cdots & \alpha_{3i} & \beta_{3i} & \gamma_{3i} & -a_{3i} & -a_{3i-1} & -a_{3i-2} & 0 & 0 & 0 & \cdots \\
\vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\
0 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 1 & 0 \\
0 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 1 \\
\cdots & \cdots & 0 & 0 & 0 & \alpha_{3n} & \beta_{3n-1} & \gamma_{3n-2} & -a_{3n} & -a_{3n-1} & -a_{3n-2}
\end{bmatrix}
\begin{bmatrix} X_1 \\ X_2 \\ X_3 \\ \vdots \\ X_{3i-2} \\ X_{3i-1} \\ X_{3i} \\ \vdots \\ X_{3n-2} \\ X_{3n-1} \\ X_{3n} \end{bmatrix}
+
\begin{bmatrix}
b_1 & 0 \\
b_2 & 0 \\
b_3 & -\alpha_i\left(\delta_{01} + l_L\right) \\
\vdots & \vdots \\
b_{3i-2} & 0 \\
b_{3i-1} & 0 \\
b_{3i} & -\alpha_i\left(\delta_{0i} + l_{i-1}\right) \\
\vdots & \vdots \\
b_{3n-2} & 0 \\
b_{3n-1} & 0 \\
b_{3n} & -\alpha_i\left(\delta_{0n} + l_{n-1}\right)
\end{bmatrix}
\begin{bmatrix} x_L \\ 1 \end{bmatrix},
\tag{3.13}
$$

$$\begin{bmatrix} Y_1 \\ \vdots \\ Y_i \\ \vdots \\ Y_n \end{bmatrix} = \begin{bmatrix} -1 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ \cdots & 1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & \cdots \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ \cdots & \cdots & 0 & 0 & 0 & 1 & 0 & 0 & -1 & 0 & 0 \end{bmatrix} \begin{bmatrix} X_1 \\ X_2 \\ X_3 \\ \vdots \\ X_{3i-2} \\ X_{3i-1} \\ X_{3i} \\ \vdots \\ X_{3n-2} \\ X_{3n-1} \\ X_{3n} \end{bmatrix} + \begin{bmatrix} 1 & -\delta_{0L} - l_L \\ \vdots & \vdots \\ 0 & -\delta_{0i} - l_{i-1} \\ \vdots & \vdots \\ 0 & -\delta_{0n} - l_{n-1} \end{bmatrix} \begin{bmatrix} x_L \\ 1 \end{bmatrix}. \tag{3.14}$$

Since the convoy becomes an interconnected system in which each vehicle is affected by all its predecessors, the transfer functions representing the inter-vehicle spacing deviations from δ_{0i} $(G_i(s) = Y_i(s)/X_L(s))$ become increasingly complicated and pointless to calculate. The spacing error propagation transfer function $\left(G_{\overline{i,i-1}} = G_i/G_{i-1}\right)$, on the other hand, provides information regarding the stability and general behaviour of the system and will be used to design the controller constants. The aforementioned transfer function can be simplified considerably if the constants are assumed to be identical for all the vehicles, in which case it becomes

$$G_{\overline{i,i-1}} = \frac{G_i}{G_{i-1}} = \frac{\gamma s^2 + \beta s + \alpha}{s^3 + \underbrace{\left(\frac{1}{\tau} + \gamma + \zeta\right)}_{a_1} s^2 + \underbrace{(\beta + \kappa + \alpha h)}_{a_2} s + \underbrace{(\alpha + \epsilon)}_{a_3}} \tag{3.15}$$

and

$$G_{\overline{1,L}} = G_1 = \frac{s\left(s^2 + \frac{1}{\tau}s + \alpha h\right)}{s^3 + \left(\frac{1}{\tau} + \gamma + \zeta\right)s^2 + (\beta + \kappa + \alpha h)s + (\alpha + \epsilon)} \tag{3.16}$$

for $i = 2 \ldots n$ and $i = 1$ respectively.

3.3.1 String stability

As discussed in section 2.2.1, it is required that the magnitude of equation 3.15 is smaller than one at any frequency to ensure error attenuation, and that its impulse response is positive. Additionally, although not essential for string stability, it would be beneficial to avoid oscillations in the spacing error response to preserve passenger comfort at short headway.

Conditions for critically damped or over damped response

It is known from linear systems theory that the necessary and sufficient condition for a linear system to present critically damped or over damped behaviour is that its transfer function poles are strictly real and negative. It is also known that the poles of the characteristic polynomial of equation 3.15 are stable (i.e. have negative real part) if the coefficients a_1 and a_3 are positive and $a_2a_1 > a_3$. The proof is simple and can be found in texts such as Hurwitz (1964) and Gantmacher (1959).

The location of the poles of the characteristic polynomial of equation 3.15 can be obtained by defining $Q = p^2/4 + q^3/27$, where $p = (2a_1^3 - 9a_1a_2 + 27a_3)/27$ and $q = (3a_2 - a_1^2)/3$ for real coefficients a_1, a_2 and a_3. If $Q < 0$, then there are three distinct real poles (*CRC Handbook of Mathematical Sciences* 1987). Let $\rho e^{i\theta}$ be the polar representation of $-p/2 + i\sqrt{-Q}$; then the pole positioned closer to the imaginary axis is $s_1 = 2\rho^{1/3}\cos(\theta/3) - a_1/3$, and the other poles are $s_{2,3} = -\rho^{1/3}\left(\cos(\theta/3) \pm \sqrt{3}\sin(\theta/3)\right) - a_1/3$. This pole distribution is represented in figure 3.3. The magnitude ρ is given by

$$\rho = \sqrt{\frac{p^2}{4} - Q} = \sqrt{\frac{-q^3}{27}}, \tag{3.17}$$

and therefore,

$$\rho^{1/3} = \sqrt{\frac{-q}{3}} = \frac{1}{3}\sqrt{-3a_2 + a_1{}^2}. \tag{3.18}$$

Figure 3.4 represents the volume in the three-dimensional space formed by a_1, a_2 and a_3 which satisfies the condition $Q < 0$. It is worth mentioning that the surface representing $a_2a_1 = a_3$ (i.e. $s_1 = 0$) lies to the left of the volume as viewed here, thus confirming that in fact $Q < 0$ is a more stringent condition than $a_2a_1 > a_3$. It can be concluded then, that the necessary and sufficient condition for stable, non-oscillatory response of the system represented by equation 3.15 is that $Q < 0$ and that the coefficients a_1, a_2, a_3 are real and positive.

Positive impulse response

According to Swaroop & Niemann (1996) a set of necessary and sufficient conditions for a stable, strictly proper, minimum phase transfer function to have positive impulse response is that:

1. The dominant pole must be real.

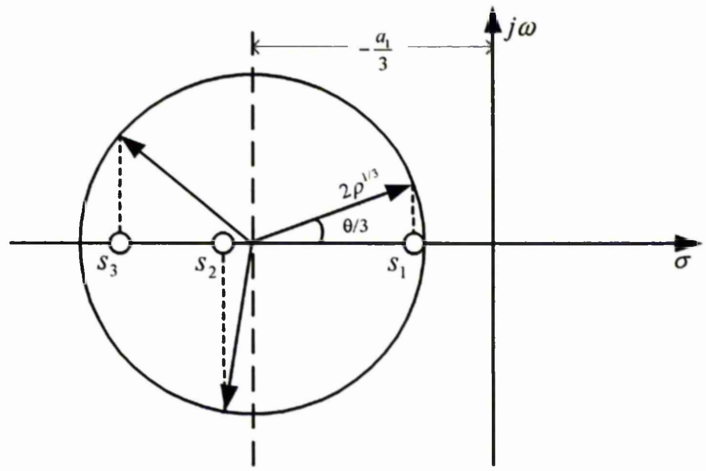


Figure 3.3: Pole placement

2. The real part of the right-most zero must lie to the left of the dominant pole.

The first condition has been discussed in section 3.3.1. The second condition, on the other hand, requires the knowledge of the position of the transfer function zeros. From the pole distribution in figure 3.3, it is possible to determine that the left-most position that the dominant pole can take, regardless of the value of a_3, is

$$s_1 = -\frac{a_1}{3} + 2\rho^{1/3}\cos(\frac{\pi}{3}), \tag{3.19}$$

or, substituting equation 3.18,

$$s_1 = -\frac{a_1}{3} + \frac{1}{3}\sqrt{-3a_2 + a_1^2}. \tag{3.20}$$

This value corresponds exactly with the boundary surface facing to the positive values of a_1 from figure 3.4. It is also the rightmost position that any non-dominant pole can take and defines the local minimum of the characteristic polynomial of equation 3.15. Furthermore, assuming that the conditions of section 3.3.1 are met, the positions of the poles on the real axis, s_1, s_2 and s_3, satisfy the equation

$$s_1^3 + a_1 {s_1}^2 + a_2 s_1 + a_3 = 0. \tag{3.21}$$

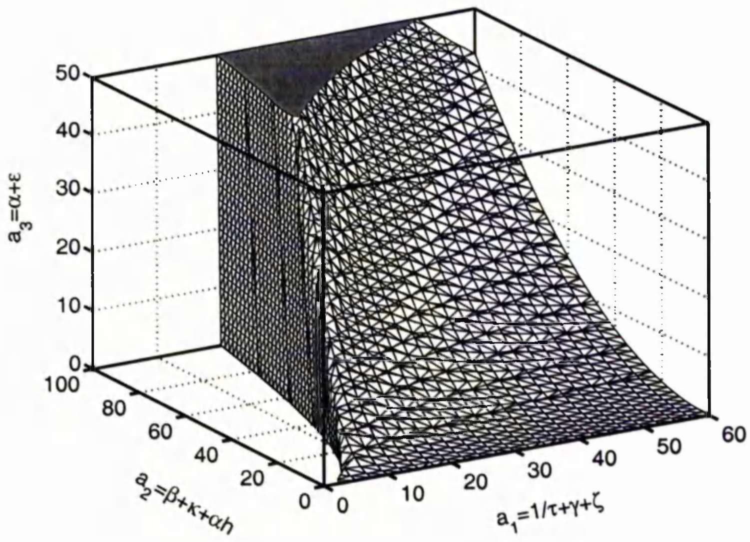


Figure 3.4: Volume of coefficients of equation 3.15 that satisfy $Q < 0$.

This equation takes the form of a plane in the space a_1, a_2, a_3 which intersects the boundary surfaces $Q = 0$ on the plane $3{s_1}^2 + 2a_1 s_1 + a_2 = 0$, one of which roots is equation 3.20. It is worth pointing out that if the real part of the zero is to the left of the local minimum (i.e. equation 3.20) on the real axis it will also be to the left of the dominant pole, making $Q < 0$ a more stringent condition. Therefore, positive impulse response can be obtained for any zero $z_{\Re} + jz_{\Im}$ by choosing any combination of values of a_1, a_2 and a_3 contained within the intersection of the spaces defined by ${z_{\Re}}^3 + a_1 {z_{\Re}}^2 + a_2 z_{\Re} + a_3 < 0$ and $Q < 0$. Figure 3.5 shows this intersection for a right-most zero with real part $z_{\Re} = -0.2$. Evidently, as the zero moves to the left, causing the plane ${z_{\Re}}^3 + a_1 {z_{\Re}}^2 + a_2 z_{\Re} + a_3 < 0$ to move upwards, it allows a wider choice of coefficients for equation 3.15 which satisfy condition 2 for positive impulse response.

A system with non-minimum phase complex zeros can also present positive impulse response if all the minimum phase zeros satisfy the conditions above and the imaginary part of the non-minimum phase zeros is sufficiently large to prevent them causing an increase in the number of sign changes of the impulse and step responses (Swaroop & Niemann 1996). These systems are referred to as *"weakly non-minimum phase"* systems and give origin to the "sufficiently weak coupling" mentioned in section 2.2.1. These systems are unsuitable for close headway vehicle

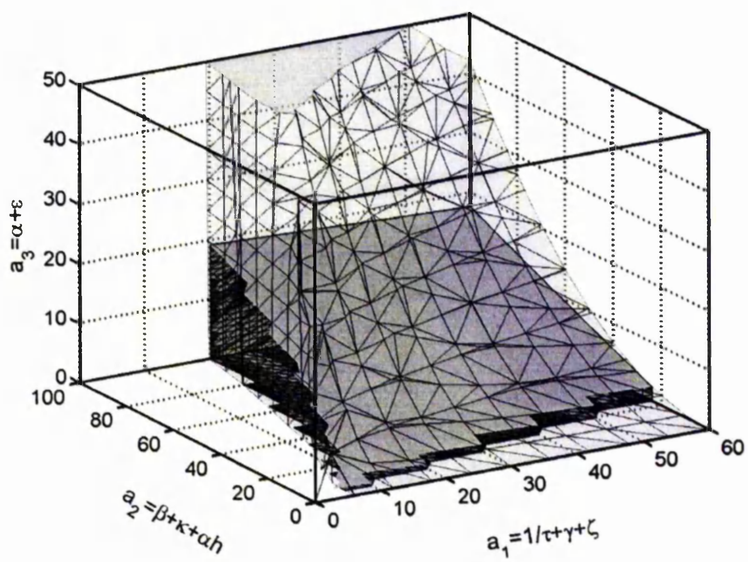


Figure 3.5: Intersection between $-0.008 + 0.04a_1 - 0.2a_2 + a_3 < 0$ and $Q < 0$ which ensures positive impulse response of equation 3.15 when the right-most zero has a real part $z_{\Re} = -0.2$.

following applications because they suffer from poor spacing regulation and excessive phase lag. Additionally, in the system investigated here, the non-minimum phase elements can only result from an unstable loop in the controller by choosing negative gains for the relative states with respect to the preceding vehicle. This choice of parameters is unacceptable for the application because it is not robust and could encourage collisions between vehicles or excessive separations which would later result in even worse collisions.

Conditions for spacing error attenuation

Once stable non-oscillatory behaviour and positive impulse response are obtained, it is possible to ensure spacing error attenuation through evaluation of the magnitude in the frequency domain of the error propagation transfer function given by equation 3.15. It is required that the said magnitude is less than 1 for all frequencies. Put into more rigorous terms, $G_{\overline{i,i-1}}$ is string stable if $Q < 0$, ${z_{\Re}}^3 + a_1 {z_{\Re}}^2 + a_2 z_{\Re} + a_3 < 0$ and $\left|G_{\overline{i,i-1}}(j\omega)\right| < 1$ for all frequencies ω. The magnitude $\left|G_{\overline{i,i-1}}(j\omega)\right|$ is:

Coefficient sign	Possible local minima	Comment
$t > 0$, $u > 0$	-	no positive real roots
$u < 0$	$\omega_{c2} = \frac{\sqrt{-3t+3\sqrt{t^2-3u}}}{3}$	always real
$t < 0$, $u > 0$	$\omega_{c2,c3} = \frac{\sqrt{-3t\pm3\sqrt{t^2-3u}}}{3}$	real if $t^2 > 3u$

Table 3.2: Possible minima of $P(\omega)$

$$\left|G_{\overline{i,i-1}(j\omega)}\right| = \left|\frac{\gamma\,(j\omega)^2 + \beta\,(j\omega) + \alpha}{(j\omega)^3 + a_1\,(j\omega)^2 + a_2\,(j\omega) + a_3}\right|, \tag{3.22}$$

or

$$\left|G_{\overline{i,i-1}(j\omega)}\right| = \left|\frac{-\gamma\omega^2 + \alpha + j\beta\omega}{-a_1\omega^2 + a_3 + j\left(-\omega^3 + a_2\omega\right)}\right|; \tag{3.23}$$

therefore, to achieve error attenuation the following condition must be met:

$$\sqrt{\frac{\left(-\gamma\omega^2+\alpha\right)^2 + \left(\beta\omega\right)^2}{\left(-a_1\omega^2+a_3\right)^2 + \left(-\omega^3+a_2\omega\right)^2}} < 1 \tag{3.24}$$

for all positive values of ω. Rearranging, the condition becomes:

$$\omega^6 + \underbrace{\left(a_1^2 - \gamma^2 - 2a_2\right)}_{t}\omega^4 + \underbrace{\left(a_2^2 - \beta^2 + 2\gamma\alpha - 2a_1a_3\right)}_{u}\omega^2 + \underbrace{\left(a_3^2 - \alpha^2\right)}_{v} > 0. \tag{3.25}$$

Since the principal coefficient of the polynomial on the left of the inequality is positive, the condition is met if the polynomial $P(\omega) = \omega^6 + t\omega^4 + u\omega^2 + v$ has no real positive roots; hence, v must be positive and the local minima of $P(\omega)$ must also be positive. The roots of $P'(\omega)$ are:

$$\omega_{c1} = 0, \tag{3.26}$$

$$\omega_{c2,3,4,5} = \pm\frac{\sqrt{-3t \pm 3\sqrt{t^2-3u}}}{3}. \tag{3.27}$$

Only the positive real roots of $P'(\omega)$ have the potential of being a local minimum of $P(\omega)$, which leaves the three distinct possibilities shown in table 3.2.

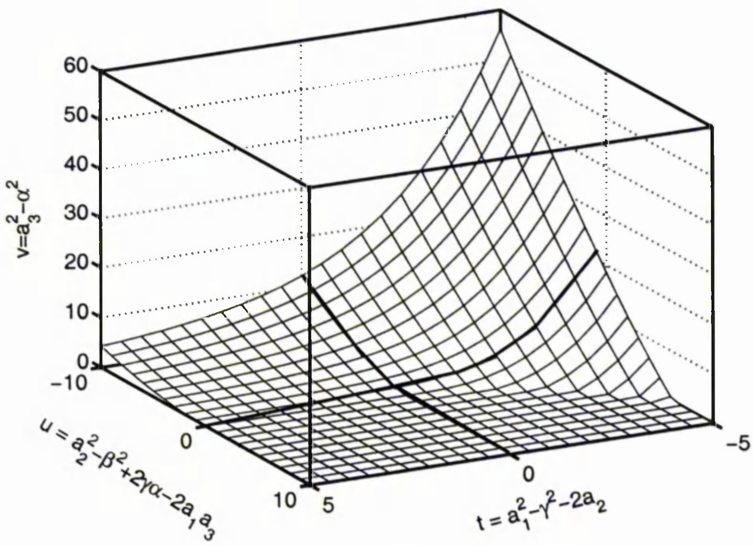


Figure 3.6: Minimum values of v to achieve spacing error attenuation.

Evaluating $P(\omega)$ at both critical points ω_{c2} and ω_{c3} produces

$$P(\omega_{c2}) = \frac{2}{27}t^3 - \frac{2}{27}\sqrt{t^2 - 3u}\,t^2 - \frac{1}{3}tu + \frac{2}{9}\sqrt{t^2 - 3u} + v \tag{3.28}$$

$$P(\omega_{c3}) = \frac{2}{27}t^3 + \frac{2}{27}\sqrt{t^2 - 3u}\,t^2 - \frac{1}{3}tu - \frac{2}{9}\sqrt{t^2 - 3u} + v. \tag{3.29}$$

Equation 3.28 results in a smaller value than equation 3.29 in the range where equation 3.29 is relevant ($t, 0$ and $u > 0$), hence, $P(\omega_{c2}) > 0$ is a more stringent condition.

In summary, the system described by equation 3.15 is l_∞ string stable if:

1. It is stable and non-oscillatory ($Q < 0$).

2. It has positive impulse response (${z_\Re}^3 + a_1 {z_\Re}^2 + a_2 z_\Re + a_3 < 0$).

3. v is positive and

4. $P(\omega_{c2})$ is positive when u is negative or when t is negative and $t^2 > 3u$.

Figure 3.6 shows the minimum values that v must take to ensure spacing error attenuation.

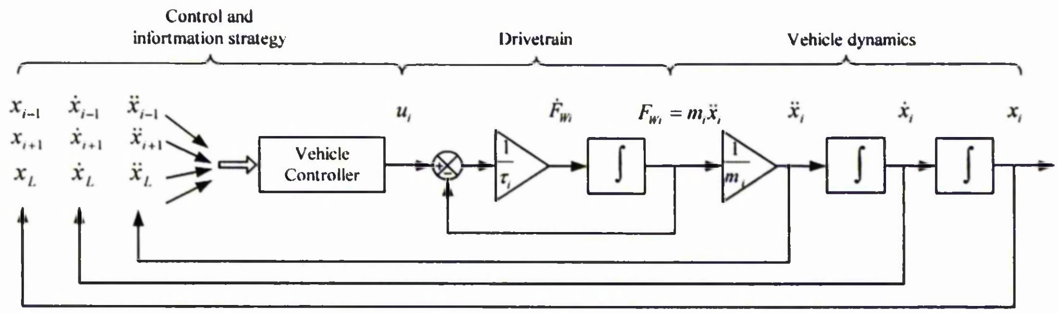


Figure 3.7: Bidirectional Linear Model.

3.4 Bidirectional Looking Scenario

The bidirectional scenario has often been dismissed in the existing literature because it is not possible to obtain positive impulse response only by using the information of the immediate preceding and following vehicles. The additional information links and the recursive nature of the transfer functions make bidirectional systems less desirable for system designers. They are interesting, nonetheless, in the context of convoy vehicles where bridging dampers can be represented by these bidirectional links. Figure 3.7 shows a convoy member similar to the one in section 3.3 with the addition of the information from its immediate follower.

The controller action equation 3.5 can be extended for the bidirectional case as follows:

$$\begin{aligned} u_i &= \gamma_i' (\ddot{x}_{i-1} - \ddot{x}_i) + \beta_i' (\dot{x}_{i-1} - \dot{x}_i) + \alpha_i' (x_{i-1} - x_i - \delta_i - l_{i-1}) \\ &\quad + \gamma_{ri}' (\ddot{x}_{i+1} - \ddot{x}_i) + \beta_{ri}' (\dot{x}_{i+1} - \dot{x}_i) + \alpha_{ri}' (x_{i+1} - x_i - \delta_{i+1} - l_{i-1}) \\ &\quad + \zeta_i' (\ddot{x}_L - \ddot{x}_i) + \kappa_i' (\dot{x}_L - \dot{x}_i) + \epsilon_i' (x_L - x_i) . \end{aligned} \tag{3.30}$$

To avoid conflicts between the controllers of adjacent vehicles, δ_i is only defined for the vehicle immediately ahead but δ_{i+1}, the desired separation between the ith vehicle and its follower must also be available. The bidirectional system can be characterised in the state space in a similar manner to the forward looking system through the matrix equation

$$\begin{bmatrix} \dot{X}_1 \\ \dot{X}_2 \\ \dot{X}_3 \\ \vdots \\ \dot{X}_{3i-2} \\ \dot{X}_{3i-1} \\ \dot{X}_{3i} \\ \vdots \\ \dot{X}_{3n-2} \\ \dot{X}_{3n-1} \\ \dot{X}_{3n} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\ 0 & 0 & 1 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\ -a'_3 & -a'_2 & -a'_1 & \alpha_r & \beta_r & \gamma_r & 0 & \cdots & 0 & 0 & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ 0 & 0 & 0 & \cdots & 0 & 1 & 0 & \cdots & 0 & 0 & 0 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 1 & \cdots & 0 & 0 & 0 \\ \cdots & \alpha_{3i} & \beta_3 & \gamma_{3i} & -a'_{3i} & -a'_{3i-1} & -a'_{3i-2} & \alpha_{r\,i} & \beta_{r\,i} & \gamma_{r\,i} & \cdots \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 1 & 0 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 1 \\ \cdots & \cdots & 0 & 0 & 0 & \alpha_n & \beta_n & \gamma_n & -a'_{3n} & -a'_{3n-1} & -a'_{3n-2} \end{bmatrix} \begin{bmatrix} X_1 \\ X_2 \\ X_3 \\ \vdots \\ X_{3i-2} \\ X_{3i-1} \\ X_{3i} \\ \vdots \\ X_{3n-2} \\ X_{3n-1} \\ X_{3n} \end{bmatrix}$$

$$+ \begin{bmatrix} b_1 & 0 \\ b_2 & 0 \\ b_3 & -\alpha_i\left(\delta_{01} + l_L\right) \\ \vdots & \vdots \\ b_{3i-2} & 0 \\ b_{3i-1} & 0 \\ b_{3i} & -\alpha_i\left(\delta_{0i} + l_{i-1}\right) \\ \vdots & \vdots \\ b_{3n-2} & 0 \\ b_{3n-1} & 0 \\ b_{3n} & -\alpha_i\left(\delta_{0n} + l_{n-1}\right) \end{bmatrix} \begin{bmatrix} x_L \\ 1 \end{bmatrix}, \tag{3.31}$$

$$\begin{bmatrix} Y_1 \\ \vdots \\ Y_i \\ \vdots \\ Y_n \end{bmatrix} = \begin{bmatrix} -1 & 0 & 0 & \cdots & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ \cdots & 1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & \cdots \\ \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ \cdots & \cdots & 0 & 0 & 0 & 1 & 0 & 0 & -1 & 0 & 0 \end{bmatrix} \begin{bmatrix} X_1 \\ X_2 \\ X_3 \\ \vdots \\ X_{3i-2} \\ X_{3i-1} \\ X_{3i} \\ \vdots \\ X_{3n-2} \\ X_{3n-1} \\ X_{3n} \end{bmatrix} + \begin{bmatrix} 1 & -\delta_{0L} - l_L \\ \vdots & \vdots \\ 0 & -\delta_{0i} - l_{i-1} \\ \vdots & \vdots \\ 0 & -\delta_{0n} - l_{n-1} \end{bmatrix} \begin{bmatrix} x_L \\ 1 \end{bmatrix}, \tag{3.32}$$

where the constants $a'_{3i} = a_{3i} + \alpha_{r\,i}$, $a'_{3i-2} = a_{3i-2} + \gamma_{r\,i}$ and $a'_{3i-1} = a_{3i-1} + \beta_{r\,i}$. Again, if the constants for every vehicle are considered to be identical, the propagation transfer functions are simplified. The last vehicle in the string is unique because it has no follower and its backward constants have no effect in the response. This makes it a forward looking only vehicle with a spacing error propagation transfer function similar to equation 3.15 with the only difference being the substitution of coefficients a_1, a_2 and a_3 for their bidirectional counterparts a'_1, a'_2 and a'_3. The transfer functions for the rest of the vehicles, on the other hand, depend on all the vehicles behind. $G_i(s)$ takes the following recursive form:

$$G_i(s) = \frac{G_n(s)}{1 - G_n(s)G_{i+1}(s)\frac{\gamma_r s^2+\beta_r s+\alpha_r}{\gamma s^2+\beta s+\alpha}}, \tag{3.33}$$

where $G_n(s)$, the last vehicle's transfer function, is the 'seed' transfer function. $G_i(s)$ is similar to the transfer functions obtained in previous studies (Yanakiev & Kanellakopoulos 1996), with the difference that the model in the literature considers a perfect drivetrain, no information from the leader or variable headway and identical forward and backward constants.

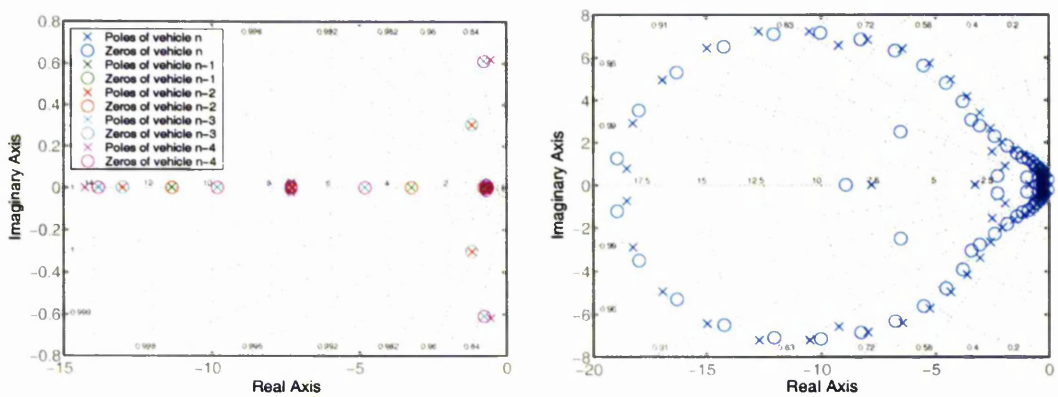
3.4.1 String stability

The placement of poles and zeros of transfer function 3.33 follow a well defined pattern in which the original zeros are common to all vehicles, but each iteration places additional zeros where the dominant poles of the previous vehicle are. There are no common poles to all vehicles, but every iteration places a real pole or pair of complex poles to the right of the dominant pole or poles and a pole or pair of poles to the left of the left-most poles of the previous iteration. The rest of the poles and zeros are placed in groups around the position of the real poles of the original vehicle at a distance which increases with each iteration. Figures 3.8(a) and 3.8(b) show the pole-zero maps for the last five vehicles and for the 20th vehicle from the back respectively in a system similar to the one described by Yanakiev & Kanellakopoulos (1996) (i.e. infinitely fast actuation force and identical forward-looking and backward-looking gains) with the following transfer function for the last vehicle:

$$G_n(s) = \frac{4s + 2.5}{s^2 + 8s + 5}. \tag{3.34}$$

Eventually, for a sufficiently large number of vehicles, poles cross the imaginary axis leading to instability. That happens for a number of vehicles well above the maximum number of vehicles that can achieve string stability. For this example system instability occurs after 35 vehicles, but the maximum number of vehicles that can achieve string stability is six. This was determined by assessing each iteration individually.

As discussed in chapter 2, the system above only produces l_2 string stability due to the coupling of poles and zeros; however, information from the leader or



(a) Last five vehicles

(b) Twentieth vehicle from the last

Figure 3.8: Pole-zero map for iterations of equation 3.33 using equation 3.34 as the seed.

time headway can be used in the case of the bidirectional system in the same manner as in the forward looking case. If the last vehicle meets the conditions for positive impulse response derived in section 3.3 it is possible to ensure positive impulse response for a finite string of vehicles. The additional requirement is that the back-to-front spacing error propagation of the string also meets the same requirements as the front-to-back propagation (equation 3.15). Back-to-front spacing error propagation is defined as:

$$G_{\overline{i-1,i}} = \frac{\gamma_r s^2 + \beta_r s + \alpha_r}{s^3 + a_1' s^2 + a_2' s + a_3'}. \tag{3.35}$$

The effects on stability and dynamic performance due to the choice of the backward looking constants in relation to the forward looking constants can be analysed by defining a *"bidirectional ratio"*.

3.4.2 Bidirectional ratio

The bidirectional ratio η is the relation between the polynomials formed by the forward and backward gains. It is defined as:

$$\eta = \frac{\gamma_r s^2 + \beta_r s + \alpha_r}{\gamma s^2 + \beta s + \alpha}. \tag{3.36}$$

There are four possibilities for the value of η:

Seed transfer function $G_n(s)$	η	Backward gains
$\frac{9s^2+40s+10}{s^3+20s^2+100s+20}$	1	$\alpha_r = 10$, $\beta_r = 40$, $\gamma_r = 9$
$\frac{4.5s^2+20s+5}{s^3+20s^2+100s+20}$	3	$\alpha_r = 15$, $\beta_r = 60$, $\gamma_r = 13.5$
$\frac{13.5s^2+60s+15}{s^3+20s^2+100s+20}$	1/3	$\alpha_r = 5$, $\beta_r = 20$, $\gamma_r = 5$

Table 3.3: Transfer functions for different scalar values of η

1. The forward and backward gains are identical; $\eta = 1$.
2. The forward gains are larger than the backward looking gains by the same proportion; η is a scalar smaller than one.
3. The backward gains are larger than the forward gains in the same proportion; η is a scalar larger than one.
4. The forward and backward gains form distinct polynomials; η is a function of s.

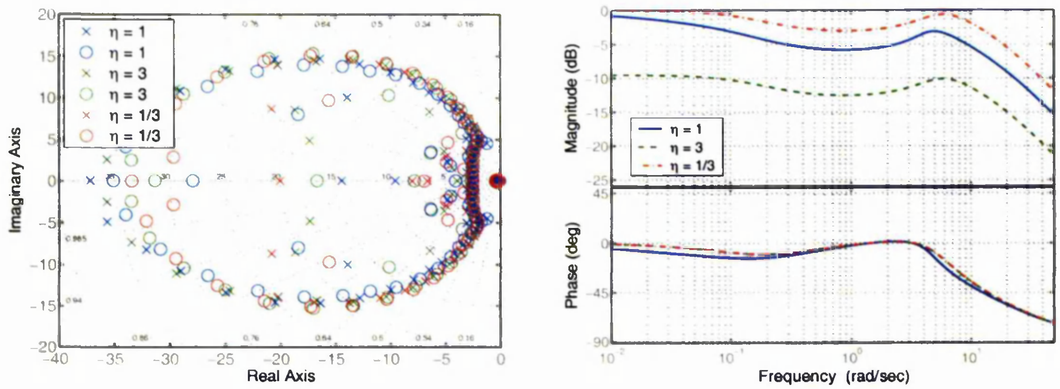
Scalar bidirectional ratio

A scalar value of η produces the same pole and zero arrangement for both forward-to-backward and backward-to-forward seed transfer functions. It is only necessary to ensure that the conditions for error attenuation are met by the two transfer functions. Figure 3.9(a) shows the pole-zero map of the fifteenth iteration of the transfer functions with different values of η in table 3.3.

The behaviour of all three systems is similar. All three show l_∞ string stability and their gain and phase plots are similar, except for a gain shift (figure 3.9(b)). The system with $\eta = 1$ is symmetrical because disturbances propagate in the same manner in both directions. The systems with $\eta = 3$ and $\eta = 1/3$ are in fact dynamically identical; they are only seen from the other end of the convoy.

Bidirectional ratio as a function of s

In this case, the zeros of the transfer functions for both propagation directions are different. It is therefore important to ensure that both propagation directions



(a) Pole-zero map

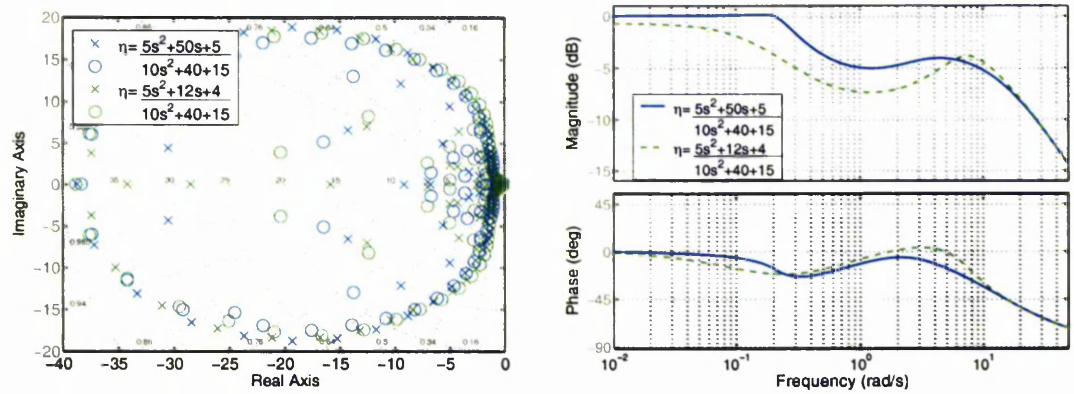
(b) Bode plots

Figure 3.9: Characteristics of the fifteenth iteration using the equations from table 3.3

Seed transfer function $G_n(s)$	η	Backward gains
$\frac{10s^2+40s+15}{s^3+20s^2+100s+20}$	$\frac{5s^2+50s+5}{10s^2+40s+15}$	$\alpha_r = 5$, $\beta_r = 50$, $\gamma_r = 5$
$\frac{10s^2+40s+15}{s^3+20s^2+100s+20}$	$\frac{5s^2+12s+4}{10s^2+40s+15}$	$\alpha_r = 4$, $\beta_r = 12$, $\gamma_r = 5$

Table 3.4: Transfer functions for η as a function of s

comply with the string stability criteria. Figure 3.10(a) shows the pole-zero maps of the fifteenth iteration using the seed transfer function in table 3.4. The seed transfer function meets the string stability criteria, but the first backward-to-forward transfer function has a zero to the right of the dominant pole; thus, it shows no positive impulse response. The iterations using such arrangement produce complex dominant poles, causing the system to show l_2 string stability only. On the other hand, the second backward-to-forward transfer function has positive impulse response, and subsequent iterations produce real dominant poles ensuring l_∞ string stability until the damping becomes insufficient for the number of vehicles. The bode plots of the fifteenth iteration of both cases are shown in figure 3.10(b).



(a) Pole-zero map (b) Bode Plot

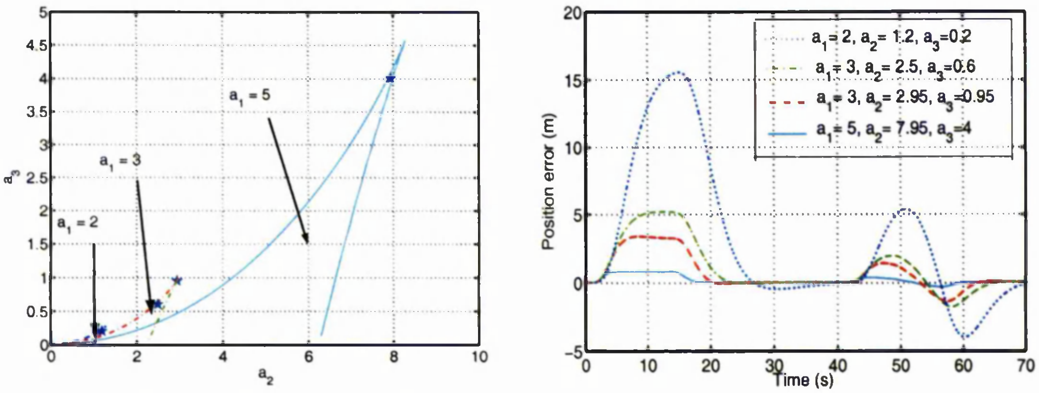
Figure 3.10: Characteristics of the fifteenth iteration using the equations from table 3.4

3.5 Controller Design

The previous sections in this chapter are dedicated almost exclusively to the study of string stability and other response characteristics associated with it. Although essential in automated vehicle following schemes, they are indication of the spacing error propagation only and are not related to the vehicles' capability to maintain their relative positions or to avoid dangerous collisions in the presence of transients and disturbances. Both capabilities should be considered to be the highest priority in the design of close headway automated systems since they govern the system's safety and maximum capacity.

3.5.1 Controller gains

In the idealised systems described here, as in the case of traditional higher order systems (each vehicle being a system of $3i$th order responding to the convoy leader's motion), transient response and disturbance rejection are improved by increased gains up to the limit where oscillation or system instability is reached. Figure 3.4 is the first indication of the conflict between individual vehicle performance (distance keeping and safety) and string stability. For a given drivetrain time constant (τ) there is a limited choice of values for the relative position and relative velocity gains represented by a_3 and a_2 respectively ($\alpha_i h_i$ is a velocity gain relative to the ground) or a'_3 and a'_2 in the bidirectional case. Usually, for the values of τ expected in real systems, the maximum values of a_3 and a_2 are



(a) Acceptable values of a_2 and a_3 for different values of a_1.

(b) Transient response and disturbance rejection for different controller gains.

Figure 3.11: Deviation from the desired position of a vehicle with $\tau = 0.5$ and forward-looking controller for different gain values.

a_1, a_2, a_3	ϵ	κ	ζ	α	β	γ	h
$a_1 = 2$, $a_2 = 1.2$, $a_3 = 0.2$	0.04	0.7	0	0.16	0.5	0	0
$a_1 = 3$, $a_2 = 2.5$, $a_3 = 0.6$	0.1	2	0.5	0.5	0.5	0.5	0
$a_1 = 3$, $a_2 = 2.95$, $a_3 = 0.95$	0.15	2	0.5	0.8	0.95	0.5	0
$a_1 = 5$, $a_2 = 7.95$, $a_3 = 4$	0.5	5	2	3.5	2.95	1	0

Table 3.5: Set of gains used to produce figure 3.11.

insufficient to be used in close headway systems under normal operation even in the idealised case used here. The remaining option is to increase the value of a_1 (or a_1') by means of the relative acceleration gains.

Figure 3.11(a) shows three sections of the volume $Q < 0$ from figure 3.4 at the planes $a_1 = 2$, $a_1 = 3$ and $a_1 = 5$. A vehicle equipped with a drivetrain represented by $\tau = 0.5$ can only take the values of a_2 and a_3 enclosed in the area representing the section at $a_1 = 2$ if no acceleration gains are used. Increasing the acceleration gains allows higher relative distance and velocity gains, hence, improving individual vehicle performance as demonstrated in figure 3.11(b). It shows the position error of a follower when the leader accelerates from rest to 20 m/s with a maximum acceleration of 1.5 m/s^2 and a maximum jerk of 2 m/s^3. Then it encounters a small hill with a maximum grade of 1/10 and a minimum grade of $-1/10$. The different values of a_1, a_2 and a_3 used in the experiment are expressed in figure 3.11(b) and shown in figure 3.11(a) as stars. The experiment was conducted using the Convoy Toolbox and is described in appendix A. The gains used in the experiment are shown in table 3.5.

It is worth noting that the conditions to which the vehicles are subjected in the experiment are considered mild manoeuvres expected under normal operation. Yet, the spacing error produced without the use of acceleration gains is incompatible with significant capacity and safety improvements, as it would require at least 16 m of spacing to perform a deceleration manoeuvre (acceleration and deceleration at the same rate produce the same amount of deviation but opposite direction). In the case of the simulation with $a_1 = 5$ the maximum deviation is less than 1 m under normal acceleration/deceleration, which suggest that it is possible to operate a similar system under this conditions with significant capacity improvement; however, it would not be able to cope with emergencies or abnormal conditions at close range without collisions.

In a real system, the implementation of real-time relative acceleration feedback presents complications. Relative acceleration estimation could be accomplished by differentiation of the relative velocity signal obtained through, for example, microwave radar. The typically poor signal-to-noise ratio of said measurement makes extensive filtering and band limiting unavoidable; hence, adding excessive time lag to the acceleration loop, which defeats the purpose of its inclusion in the system. Alternatively, acceleration measurements can be taken from acceleration sensors installed in each vehicle and broadcast to other vehicles in the same group (i.e. cooperative platoons) as discussed in section 2.2.4. This strategy introduces a considerably smaller delay in the acceleration loop, but if this link presented a fault, the string of vehicles would be left in an unstable state. Even if a safety critical wireless communication network with the required redundancy, robustness, bandwidth and response time was available and could be deployed in this environment, its cost would be elevated. It is worth noting that the leader's states can only obtained by the rest of the vehicles if they are broadcast, since in a line of vehicles it is not possible to take direct measurements.

A third method of obtaining acceleration measurements would be to use adaptive digital signal processing techniques to differentiate the relative velocity signal (Vainio 1999). Although the noise cannot be completely removed, using predictive filtering can significantly enhance the signal-to-noise ratio in digital differentiation. Also modern processors allow near to real-time operation. The major drawback with these techniques is that they are not robust to all kinds of random signals.

3.5.2 Balance between gains relative to the leader and to the adjacent vehicles

So far in this section the discussion has been limited to the denominator of the propagation transfer function (equations (3.15) and (3.33)), which does not distinguish between the gains relative to the leader and those relative to the adjacent vehicles. Figure 3.5 is the first indication of the relation between the sets of gains. In general, it can be said that increasing the acceleration gain relative to the predecessor moves the zero to the right, reducing the choice of values of a_2 and a_3 for a given a_1, while increasing the velocity gain relative to the predecessor increases the choice. This contradiction to the conclusion of Eyre et al. (1997) that disregarding the predecessor's velocity is beneficial applies only if the acceleration relative to the predecessor is used in the control loop. Removing the zeros by ignoring both gains also benefits the string stability in this case, but it would allow large relative velocities and accelerations to develop between adjacent vehicles, thus compromising safety.

The last constraint for string stability depicted in Figure 3.6 shows the importance of the gains relative to the leader and their interaction with the gains relative to the adjacent vehicles. These represent yet another indication of the trade-off between string stability, individual performance and safety.

Consider the value of v; although it cannot be negative, the highest it is and the farthest from its minimum string stable value, the best error attenuation along the string, therefore, it makes sense increasing the position gain relative to the leader (i.e. ϵ) since v responds better to it than to the position gain relative to the predecessor (i.e. α).

The value of t may be negative, potentially leading to regions in the space in figure 3.6 where large values of v are required to achieve string stability. In this case giving priority to the leader's acceleration gain (i.e. ζ) over the predecessor's acceleration gain (i.e.γ) is even more necessary than in the case of v. The reason for this is that acceleration signal is affected by noise in a greater measure than the position signal, forcing the acceleration gains to be kept to a minimum in order to avoid amplifying the noise excessively and injecting it in the acceleration loop, which has a smaller time constant than the position loop.

The value of u is similar to t in that it may lead to regions in the t, u, v space where a large value of v is necessary for stability. It also responds better to an increase to the time headway (i.e. h) or to the velocity gain relative to the leader (i.e. κ) than to that of the predecessor (i.e.β). Increasing α or γ also improves the value of u but may adversely affect the values of v or t.

3.5.3 Design Methodology

With considerations of the previous sections, it is possible to follow a systematic methodology for the selection of controller gains. The following sequence is suggested:

1. Estimate the value of τ from the drivetrain's time constant, the controller and sensor lags and delays.

2. Select the maximum permissible value of a_1 based on the method used to obtain the acceleration signal and on the characteristics of the control system and drivetrain.

3. Select positive values for a_2 and a_3 which fall within the volume $Q < 0$. The best vehicle performance, safety and system capacity are obtained by selecting the highest possible values of a_2 and a_3.

4. Select a value of γ, which must be positive and smaller than a_1, based on the qualities of the leader's and the predecessor's acceleration signals (which may have been obtained differently). Ideally the value of t should be around zero, which allows a more comfortable choice for the values of u.

5. Select a value of β between zero and a_2 which takes the value of u as close as possible to zero if $t > 0$ or to $u = t^2/3$ if $t < 0$. Assume that the term $2\gamma\alpha = 0$

6. Select the largest value of α between zero and a_3 which yields string stability. Notice that the term $2\gamma\alpha$ may not be zero, in which case a larger value of α may be possible. It may then be worth to recalculate u. If string stability cannot be achieved it is necessary to choose larger values for a_2 and a_3 and repeat the process.

7. Calculate ζ, κ, ϵ and, if time headway is used, h from the previously obtained values.

8. Ensure that the positive impulse response condition (i.e. figure 3.5) is met and modify accordingly by reducing γ or a_3 or by increasing a_2 within the limits of $Q < 0$ if necessary.

Once these parameters are fixed, it may be desirable to fine tune the system for example to get a more comfortable ride or to better adapt the system according to more specific considerations.

It is interesting to note that, in this linear system, vehicles of different mass can be accommodated since their gains are expressed per unit of mass. The limitation in real systems is that the maximum force available from the drivetrain cannot be compensated in the same manner. As for vehicles with different time constants (τ), it may be possible to adjust their controllers by using the acceleration gains to provide a uniform value of a_1. Perhaps the controller design could follow the described methodology for an average vehicle and adapt the gains as necessary for every vehicle's constant.

3.6 Chapter Conclusions

In this chapter, a simplified model of a string of vehicles was described. The relevant parameters for forward-looking and bidirectional controllers were identified and the conditions to achieve string stability were derived and analysed in depth. It is shown that in practical system design there is a conflict between the required gains for improved capacity and safety and those for string stability. It is also shown that for a given set of vehicles, it is possible to partially mediate the conflict through the use of sensing, communication, control and actuation equipment of higher specification, which results in increased overall system cost and complexity, but reduced robustness. Lastly, a controller design methodology, which would lead to the best compromise between comfort, safety and capacity, is suggested.

Chapter 4

Safety and Ride Quality; Human Responses to Vibration and Shock

4.1 Introduction

The nature of the system proposed in this work implies that frequent physical interactions between vehicles will occur. That, added to the conflict between passenger comfort and accurate automated vehicle following (documented in the literature reviewed in chapter 2) make understanding the factors which affect ride quality essential for the design of a successful system. It is crucial to ensure that the overall ride quality of automated convoy vehicles is consistent with comparable vehicles from current transportation systems. Similarly, safety must be at least equivalent to that of current systems. The increased risk of rear-end collisions must not increase the risk of occupant injury and protection must be guaranteed even in the presence of major system failures.

Although transport system designers have constantly changed the approach towards passenger protection as their knowledge evolves, ensuring safety has always been the paramount priority. The concept of acceptable ride quality has also been in the minds of transport system designers for a long time, but, unlike safety, for which specific rules and regulations exist, ride quality is a subjective interpretation and therefore impractical to regulate. The term "ride quality" was first defined in an investigation commissioned by the Electric Railway Presidents' Conference Committee in 1932 as *'the effect upon the passenger of repetitive car* *body movements or vibration which produce a movement of the entire passenger's body'* (Hirshfeld 1933).

Modern day passengers expect to be able to accomplish activities such as reading, writing, eating, drinking, sleeping or even walking while travelling. Their subjective evaluation of the ride quality is likely to be affected by the extent to which they can perform such activities. In some cases motion can be comfortable or even beneficial, or it can cause annoyance and injury; however, a specific limit for motion and vibration has not yet been set and transportation companies usually set their own limits.

The purpose of this chapter is to acquire awareness of the aspects which influence the passenger's perception of motion when travelling on board an automated vehicle which operates as a convoy member. A metric for assessing ride quality is proposed and motion limits are set. Additionally, vehicle safety and occupant protection are studied to understand what will be required from the bridging dampers to bring passengers to a safe stop in case of emergency situations involving strong collisions. An assessment method is described and recommendations for the design are made.

4.2 Motion Characteristics of Convoy Vehicles

The motion to which passengers in conventional vehicles are subjected has multiple causes. In addition to the normal vehicle acceleration and deceleration manoeuvres, suspension motion and lateral acceleration, there are also interactions between the track/road and wheel, changes in curve radii, joints and long-wave or localised track/road irregularities which contribute to the motion characteristics. In the case of fixed guideway vehicles, track alignment errors, switches and swaying must also be considered to fully describe their motion.

Automated vehicles are affected by additional longitudinal motions resulting from their controller dynamics and how they respond to disturbances and to nearby vehicles. If bridging dampers are used as proposed here, interactions between vehicles can be stronger making possible the transmission of vibration and motion along the string and may even contain small magnitude shocks, especially while performing entrainment and extrainment manoeuvres.

Since the object of the research reported here is the longitudinal control of vehicles, only motion in the axis parallel to the direction of travel is relevant. The five remaining rotational and translational coordinates are ignored, but reference to them is made only for completeness or comparison where pertinent in the reminder of this work.

It follows that the characteristics of concern for the design of the proposed system regarding ride quality can be divided into two broad categories:

- Quasi-static acceleration along the longitudinal axis caused by intentional manoeuvres to change velocity, including acceleration, coasting and braking.
- Longitudinal vibration originating from the interactions with adjacent vehicles through their controller dynamics as well as through the dampers.

Similarly, the issues of concern regarding safety can be classed into the following categories:

- Vibration with large values of acceleration which may cause a health hazard for passengers.
- Single events with large magnitude acceleration levels such as emergency decelerations and shocks.

4.3 Ride Quality

Human perception of ride quality is a subjective response to the surrounding environment represented by a complicated blend of physical, physiological and psychological changes. It not only depends on the motion characteristics of a journey, but also can be influenced by factors such as moisture, temperature, visual stimuli, acoustic noise and subject's experience and expectations.

The acceleration history of a vehicle is often used to evaluate the ride quality when investigating the performance of a particular controller set-up for automated transport systems; however, it may not accurately represent the discomfort caused by motion resulting from the vehicle's dynamic response.

4.3.1 Jerk and quasi-static acceleration

Humans are not directly sensitive to velocity, but only to the force that acts upon them. A passenger sitting in a cushioned seat with large backrest facing in the direction of travel would barely notice a relatively strong acceleration, but if there is no headrest discomfort would result from the large force produced by the neck to transmit the acceleration force. Seating passengers facing the direction opposite of travel require a greater effort to maintain upright, yet, standing passengers are of more concern since the force needs to be transmitted through smaller areas. They need to adjust their posture to compensate the force, especially when not in contact with other means of support than the floor of the vehicles that carry them. This requires muscular action and some sense of balance.

Another study from 1932, also commissioned by the Electric Railway Presidents' Conference Committee which led to the revolutionary 'PCC' streetcar, considered methods of acceleration of a rail vehicle for best comfort. Acceleration profiles include constant acceleration, constant jerk, linearly increasing jerk and non-linear jerk (Hirshfeld 1932).

It was recognised that the streetcar system required frequent stops and therefore ride quality was a major concern when starting the vehicles. If the vehicle is at rest, the acceleration must increase in some fashion to a limit value where it should be kept until the desired velocity is reached. The questions regarding comfort are:

- Should the increase in acceleration be along a straight line, or should it be concave or convex, and if so, what shape should it have?
- What value should be chosen for the highest constant acceleration?
- How long should the maximum acceleration be sustained?

The tests were carried out using a vehicle with a maximum acceleration of 3.65 m/s^2 and a change of acceleration of any value from 0.3 m/s^3 to over 15 m/s^3. Plots of the probability of maintaining balance in the different cases showed the best consistency of results, but these curves only showed the specific circumstances of the test. It is desirable to have a direct measure of the difficulty to maintain equilibrium under any combination of conditions that may be experienced in a real car. An index of disturbance of equilibrium was developed recognising that the stability of an individual is a function of two factors: the ability of the subject to preserve his equilibrium and the severity of the condition imposed. The tests showed no marked variations in performance attained by passengers of different sex, build, age and occupation.

The authors related the accumulated degree of disturbance Y_t to the fraction of passengers retaining equilibrium f_t with the equation

$$Y_t = -\log f_t \tag{4.1}$$

By fitting the experimental results, it was discovered that an expression of the form

$$Y_t = C_1 a^3 + C_2 a^2 t, \tag{4.2}$$

where a is the acceleration, t is the time and C_1 and C_2 are constants, gave the best agreement for the accelerations tested except for increasing accelerations with a Rate of Change of Acceleration (rococ) greater than 2.133 m/s^3 and for the jerk tests, where the subjects had no time to adjust to the motion. It is inferred that this equation holds for acceleration and deceleration equally.

For rococ greater than 2.133 m/s^3, it was found that

$$Y_t = C_3 a^2 \tag{4.3}$$

gives good agreement.

For the jerk test

$$Y_t = C_4 a_i t_i, \tag{4.4}$$

where a_i is the change in acceleration and t_i is the time between the beginning and the end of the jerk, is suitable.

By plotting the index of disturbance against the acceleration under different starting strategies, it was evident that for the same disturbance greater accelerations were attainable when greater rococ was used provided it stayed under 2.133 m/s^3. It was also notable that disturbance index does not grow rapidly as higher constant accelerations are used. The study also showed that giving the passengers a small acceleration as warning before accelerating to the maximum did not give any advantage and only meant a loss in time with respect to other starting methods.

Limits for jerk and quasi-static acceleration

Studies reviewed by Hoberock (1977) suggest that steady, non-emergency accelerations in the range of 1 m/s^2 to 1.5 m/s^2 are acceptable for public transport, where passengers may be standing as well as seating. Also, values of jerk larger than 3 m/s^3 are not likely to be acceptable. The Federal Railroad Administration regulations (FRA 2004), with regard to the suspension system suggest that the horizontal acceleration in steady state should not exceed 1 m/s^2, the maximum single event peak to peak vertical acceleration should not exceed 5.45 m/s^2 over a one second period and the maximum single event peak to peak horizontal acceleration should not exceed 3 $m/^s$ over a one second period.

The Automated Guideway Transit Technology (AGTT) program run by the Urban Mass Transportation Administration (UMTA), specified a goal service acceleration of 1.5 m/s^2 with maximum jerk of 2 m/s^3 for PRT systems (UMTA 1976). For road vehicles, where passengers are seating and restrained, acceleration could be up to $\pm 2\,m/s^2$ according to Frankel, Alvarez, Horowitz & Li (1994) and between $+3\,m/s^2$ and $-5\,m/s^2$ according to van Arem, de Vos & Vanderschuren (1997).

4.3.2 Vibration and ride quality

The human body is a complex dynamic system, consequently, accurately estimating its physical behaviour under vibrating conditions is a difficult task. Similar vibration in terms of frequency and magnitude can produce different responses in different environments. Motion experienced while driving an off-road vehicle may be described as not uncomfortable, although, a similar ride condition may be unacceptable when related to a coach journey.

Vibration can be categorised as 'deterministic' if its future behaviour can be precisely predicted from its previous behaviour, or as 'stochastic' or random if its behaviour has only some statistical properties. Deterministic vibration may be further subdivided in 'periodic' (e.g. sinusoidal) and 'non-periodic' (i.e. transient). Research (Griffin 1997) has been carried out to predict the responses of humans to sinusoidal vibrations. However, modern vehicle movement is more likely to produce exposures to stochastic and shock motion than pure sinusoidal movement. Some experiments (Kubo, Terauchi, Aoki & Matsuoka 2001, Griffin 1997) have shown that knowledge from human exposures to pure sinusoidal vibration can be helpful in predicting responses to random vibration.

Human response to vibrating environments

The mechanical impedance of the human body cannot be simplified to that of a pure mass (Griffin 1997, BSI 1987, Kubo et al. 2001, Smith 1999). Its physical reaction is highly dependent on the vibration frequency, amplitude and direction. Some authors have attempted to simplify the human body exposed to vertical vibration using several configurations of masses and viscoelastic elements (e.g. Kubo et al. 2001, Smith 1999), but knowledge of exposure to horizontal vibration is limited and predicting the responses of all the individuals in all circumstances is not yet possible. Physical characteristics such as transmissibility, mechanical impedance and apparent mass vary among individuals and can be significantly affected by small changes in posture or by muscular action. Physiological and psychological reactions to vibration are considerably more complex, making their prediction even more difficult.

Vibration exposures to horizontal vibration of seated subjects without backrest at frequencies below 1 Hz cause the upper body to sway, but can be controlled with muscular action. In the frequency range of 1 Hz to 3 Hz it becomes difficult to stabilise the body. At higher frequencies (above 10 Hz) the motion is only felt at the supporting surface. The use of backrests helps to stabilise the body at low frequencies, but may be the main cause of motion transmitted to the upper body at higher frequencies. In standing position the body becomes unstable at frequencies below 2 Hz, the x axis (see below for the definition of coordinate system) the being most sensitive.

In addition to the physical reactions, physiological changes are common human responses to environmental stressors, including vibration. Studies like those conducted by Kubo et al. (2001) showed that vertical vibration at certain frequencies may affect the subject's heart and respiration rates, blood pressure and salivation levels among other vital signs. Also, vibration may have effects on psychological reactions adversely affecting the performance and state of mind of the subject (Smith 1999).

The effects of human exposures to vibration and shock may be divided in four main groups:

- Interference with activities.
- Interference with comfort.
- Motion sickness.

- Interference with health.

Vibration evaluation

When the PCC streetcar investigation was commissioned, the original vibration experiments recorded motion amplitude, but acceleration was found to be more representative of the disturbance caused to passengers (Hirshfeld 1933). Since then, most researchers and standards agree in the adoption of acceleration as the preferred quantity for expressing vibration because of the convenience and relative ease of taking direct measures (e.g. Griffin 1997, BSI 1987, ISO 1997). The coordinate system used for measurement and report has also been subjected to similar treatment.

Measurement coordinate systems

Early methods of vibration evaluation suggested body-centred coordinate systems with the origin placed at the heart, and capable of rotating with the body. This configuration is not always well suited for characterising the vibration exposure because the origin cannot be exactly located and it is difficult to take measures at that point. In addition, passengers may move, rotate or simply change posture, seriously affecting the transmissibility of the body. Additionally, some experiments suggest that the relative position and motion between the lower limbs and the rest of the body play a crucial role in the severity of vibration and its effects on human subjects (Griffin 1997, Kubo et al. 2001).

More recent methods have placed coordinate systems centred at the interface with the body. BSI (1987) and ISO (1997) describe a convenient set of basicentric axes for seated, standing and prone persons. These standards suggests placing three translational and three rotational axes on the seat, three translational axes at the back and three translational axes at the feet. All placed at the interface formed by the body and the supporting surface in the midsagital plane. For standing subjects the only coordinate system must be placed between the feet and the supporting surface (i.e. floor). All coordinate systems are by convention right hand with positive x towards the front of the subject, positive y towards the left and positive z towards the head.

Vibration measurement

BSI (2001) specifies that measurement at the surface/body interface is the preferred method but may not be appropriate for the evaluation of fixed guide-way systems, especially if the contribution of the seat to comfort is not the primary concern. When not taking measurements at the body-seat interface, it is appropriate to do so at rigid points of the vehicle structure using the same reference system.

Transducers placed at one position have to be orthogonally positioned and be as close as possible to each other. According to BSI (1987), the sensitive axes of the transducers may deviate from the preferred axes by up to 20° where necessary, while ISO (1997) recommends a maximum of 15°. Comprehensive guidance on the instrumentation for the measurement of vibration exposure of human beings is given in (BSI 1991).

All the measuring procedures must be recorded and included in the report, as well as the specifications of the measuring equipment. The position of the vehicle in the convoy may affect the outcome of the experiment. Direction of travel and position of the vehicle in the convoy should be also recorded.

Frequency weighting

Human responses to vibration are highly dependent on the associated frequency. Early works (Hirshfeld 1933) suggested the use of "equivalent comfort contours", which are acceleration and frequency pairs that cause the same level of discomfort (Griffin 1997). More recently, the effects of frequency are commonly evaluated with "frequency weightings", which are analogous to the inverse of the equivalent comfort contours.

Experimental work has proven that multi frequency and random movement result in greater discomfort than the worst single component in the motion spectrum. Therefore, it is reasonable to evaluate the effect of motion in a direct manner if its frequency band is small. However, for evaluation of the effects of wider frequency ranges, the use of weighting curves has proved to be necessary and sufficient (Griffin 1997).

According to (BSI 2001), the frequency range of motion expected to adversely influence ride comfort in conventional rail vehicles includes 0.1 Hz to 2 Hz on roll motion, 0.5 Hz to 10 Hz in the lateral and longitudinal directions, and 0.5 Hz to 20 Hz in the vertical direction. Additionally, low frequency motions in the vertical direction in the range from 0.1 Hz to 0.5 Hz are likely to induce kinetosis (motion sickness). The standard (BSI 1987) provides the characteristics of six frequency weighting curves used to evaluate the different effects on vibration occurring in different directions. These curves are specified as band-limiting filters and frequency weightings. Details of the curves are provided in appendix B. Similar weightings are defined in ISO (1997).

BSI (1987) recommends the use of weighting W_d to assess the discomfort caused by x-axis vibration transmitted through the supporting surface of the seat or the floor in the case of standing persons. It is defined by the following transfer function:

$$W_d = \overbrace{\frac{3.948 \times 10^5 s^2}{s^4 + 888.5s^3 + 3.979 \times 10^5 s^2 + 1.403 \times 10^6 s + 2.494 \times ^6}}^{BandLimiting} \cdot \overbrace{\frac{25.13s + 315.8}{2s^2 + 39.89s + 315.8}}^{Weighting}. \quad (4.5)$$

The frequency weighting curve W_d is shown in Figure 4.1.

It is not always correct to treat the vehicle and seat as rigid bodies, so, in addition to the previous measure, W_c may be used for x-axis vibration on the backrest with a multiplying factor of 0.8 and W_b for the x- and y-axis vibration at the feet with a multiplying factor of 0.25.

Averaging methods

Root-mean-square averaging of the weighted accelerations can be effectively used to assess vibration and should always be recorded. However, the severity of motion containing high peaks and intermittent vibrations may be underestimated by this method. The ratio of the maximum instantaneous peak value of the frequency weighted acceleration signal to its rms is called 'Crest Factor'. It is defined as:

$$crest\ factor = \frac{weighted\ peak\ acceleration}{weighted\ rms\ acceleration}. \quad (4.6)$$

The rms value is the square root of the average of the square of the function over an interval of time. It is defined as:

$$rms = \sqrt{\frac{1}{T}\int_0^T a^2(t)dt}, \quad (4.7)$$

where T is the total time period in which the motion occurs and $a(t)$ is the frequency weighted acceleration.

Both standards describe alternative methods to evaluate the effects of human

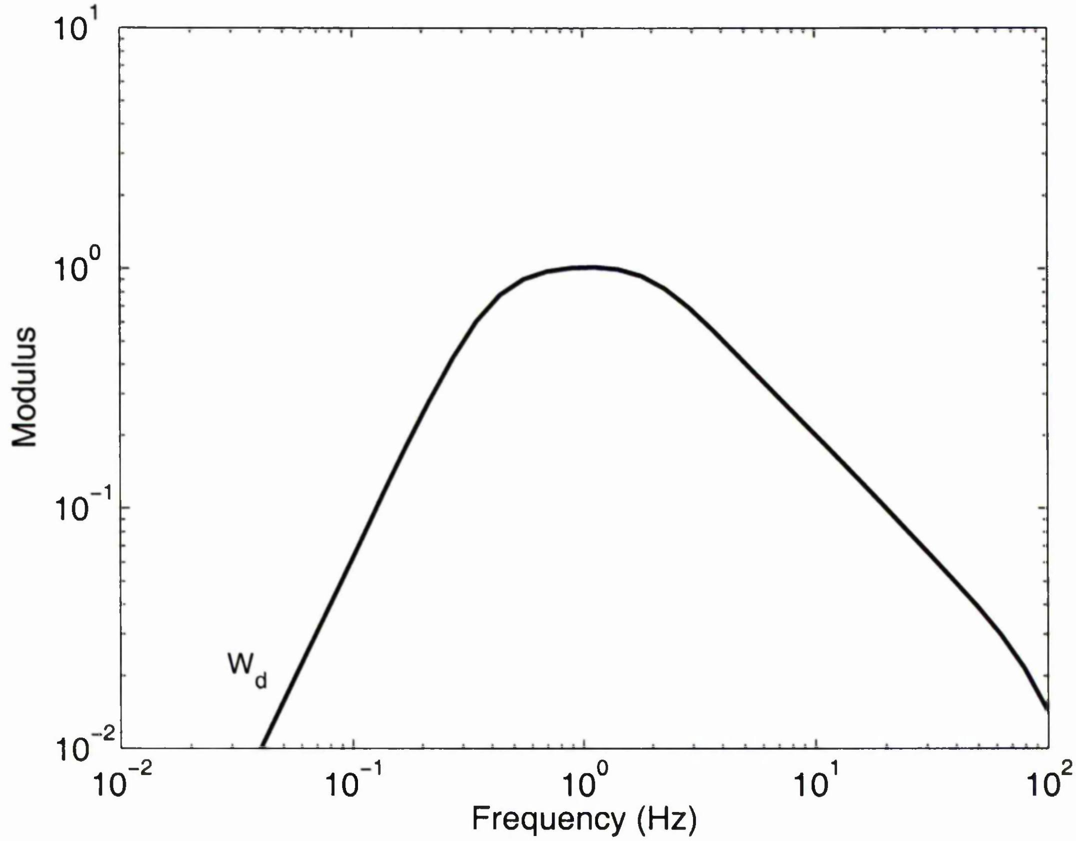


Figure 4.1: Frequency Weighting W_d from BS 6841 (1987)

exposure to motion with high crest factors. BSI (1987) recommends their use specially when evaluating vibration with crest factors above 6.0, whereas ISO (1997) does it when the crest factor is above 9.0. The root-mean-quad method, described in BSI (1987), is one way of overcoming the limitations of the rms procedure in evaluating the effects of vibration and repeated shock on comfort and perception when the motion is characterised by a high crest factor. The rmq value is the fourth root of the average of the fourth power of the function over an interval of time. It is defined as:

$$rmq = \sqrt[4]{\frac{1}{T}\int_0^T a^4(t)dt}. \tag{4.8}$$

The vibration dose value (VDV) is a more robust metric introduced by BSI (1987) as the preferred method of assessing the severity of all motions and later adopted by ISO (1997) as an alternative technique. It gives the magnitude of the motion with a 1 second duration which causes equivalent effects to humans as the motion that is being evaluated. It is given by the fourth root of the integral of the fourth power of the frequency weighted acceleration.

$$VDV = \sqrt[4]{\int_0^T a^4(t)dt}. \tag{4.9}$$

Its units are $ms^{-1.75}$.

As VDV includes time dependency, the cumulative effects of exposure to motion can be predicted, and empirical data shows that the exposure magnitude can be related to the actual subjective response in a closely linear way (Griffin 1997), so it is a good measure of the likely human reactions. Additionally, if shocks are present in the acceleration history, the VDV may be applied over a period of time which includes the events of interest. It can assess the effects of compound shocks and repeated shocks effectively, and the values can be compared to those obtained for vibration (Griffin 1997); however, it is not recommended for large magnitude, single shocks which may cause injury, such as a car crash. The use of VDV as a metric for vehicle movement in the convoy context has been already discussed (e.g. González-Villaseñor, Renfrew & Brunn 2003*a*, González-Villaseñor, Renfrew & Brunn 2003*b*).

When the crest factor is less than 6.0, (BSI 1987) suggests an alternative method to estimate the vibration dose value (ISO (1997) also mentions this method). This is an empirical way of evaluating the time dependency. The estimated Vibration Dose Value ($eVDV$) is defined as:

$$eVDV = \sqrt[4]{(1.4a)^4\, b}, \tag{4.10}$$

where a is the rms value of the frequency weighted acceleration and b is the duration in seconds. If the vibration exposure consists of N periods of different magnitude, the VDV for the total exposure should be calculated from the fourth root of the sum of the fourth power of individual VDVs (calculated or estimated).

Lastly, the 'running' rms evaluation method, described by (ISO 1997), takes into account occasional shocks and transient vibration by the use of a short integration time constant. The vibration magnitude is defined as a maximum transient vibration value ($MTVV$), given as the maximum value in time of $a(t_0)$, defined by:

$$a(t_0) = \sqrt{\frac{1}{T_c}\int_{t_0-T_c}^{t_0} a^2(t)dt}, \tag{4.11}$$

where $a(t)$ is the instantaneous frequency-weighted acceleration, T_c is the integration time constant and t_0 is the time of observation (instantaneous time). The $MTVV$ is defined as:

$$MTVV = \max[a(t_0)]; \tag{4.12}$$

the standard recommends using $T_c = 1$ second in measuring $MTVV$.

Discomfort caused by exposure to vibrating environments

Guidance on the evaluation of vibration and repeated shock with respect to discomfort and perception is provided in Clause 6 and Appendix C of BSI (1987). The frequency weightings applied to the vibration of the seat are W_b for the z direction and W_d in the x and y directions. Additionally, translational vibration occurring at the floor and at the backrest and rotation occurring at the base of the seat may be assessed. Details of the weightings and multiplying factors used for the additional directions are provided in appendix B. For standing persons the use of the same weightings applied to translational seat vibration is recommended.

Rms averaging should be used when possible in the assessment of vibration in regard to discomfort. If the motion contains a high crest factor, however, the alternative methods are better suited. VDV is also well suited for the evaluation of exposure to intermittent motion.

An average fit person can just detect vibration with weighted acceleration of 0.015 m/s^2 peak-to-peak. Additional approximate indications of the likely reactions to weighted vibration of different magnitudes encountered in public transport given in (BSI 1987) and (ISO 1997) in Appendix C are shown in table 4.1.

Ride quality may be considered good if the weighted acceleration does not exceed $0.2m/s^2$ rms, while bad ride quality is associated with weighted accelerations of about 1 m/s^2 rms (Griffin 1997).

As a comparison, a sinusoidal motion with weighted acceleration of $0.315m/s^2$ rms produces a VDV of $0.816\,m/s^{1.75}$ after an exposure time of 30 seconds, and $0.970\,m/s^{1.75}$ after 60 seconds, while motion with weighted acceleration of $1\,m/s^2$

Rms value	Reaction
$< 0.315\, m/s^2$	not uncomfortable
$0.315 - 0.63\, m/s^2$	a little uncomfortable
$0.5 - 1.0\, m/s^2$	fairly uncomfortable
$0.8 - 1.6\, m/s^2$	uncomfortable
$1.25 - 2.5\, m/s^2$	very uncomfortable
$> 2.0\, m/s^2$	extremely uncomfortable

Table 4.1: Approximate indications of the likely reactions to vibration in transport.

rms produces a VDV of 2.59 $m/s^{1.75}$ after 30 seconds and 3.08 $m/s^{1.75}$ after 60 seconds.

Vibration and its interference with activities

Activities most commonly considered in the evaluation of vibration include visual perception and the coordinated control of hand movement. BSI (1987) gives guidance on the way to evaluate the severity of motion in Clause 5 and appendix B (but ISO (1997) does not give guidance on this subject). It is mostly applicable to the vibration transmitted to the seated body through the seat surface. The weightings used are W_d for the x and y axes and W_g for the z axis. Guidance on time dependency is not given and the method is restricted to vibrating conditions that are approximately constant. A total manual control decrement can be calculated from the root-sum-of-squares of the weighted accelerations in all three axes, but the activities involved may be more sensitive to movement in one axis than in other.

As a general consensus of opinion, the standard suggests that where hand control is required to an accuracy of around 5 mm rms or 2.5 N rms the weighted acceleration in any axis should not exceed 0.5 m/s^2. For less accuracy the acceleration values could be increased in linear proportion. For visual activities the weighted acceleration should not exceed 0.5 m/s^2 rms if it is necessary to resolve detail which subtends less than 2 minutes of arc at the eye. For every increase by a factor of $\sqrt{2}$ in the size of the detail the vibration magnitude could be doubled.

Motion Sickness

Motion sickness is mainly caused by low frequency motion. Nausea, sweating, pallor, and even vomiting are the most common symptoms of motion sickness. All normal persons can feel sick if exposed to a suitable stimulus. A common theory that explains the occurrence of motion sickness implies that it arises from conflicting signals of the acceleration originated from visual, vestibular, somatosensory cues Griffin (1997). Other researchers (J. & Bles 1999) suggest that the change of relative direction of gravity with respect to the body may have great influence in the motion sickness incidence. The symptoms of motion sickness are increased with time. Over several hours adaptation to the motion may occur.

A standard procedure of evaluation for the motion sickness is included in the standards; however, it is restricted to motion in the vertical direction due to limitations in the available data. The weighting curve W_f is recommended by this procedure for the evaluation of motion sickness.

The Motion Sickness Dose Value $MSDV$ is given by the square root of the integral of the square of the frequency weighted z-axis acceleration.

$$VDV = \sqrt{\int_0^T a^2(t)dt}. \tag{4.13}$$

It is worth mentioning that the $MSDV$ cannot be related or compared with the motion evaluation methods described in previous sections. The reason is that the only weighting curve used is W_f, and therefore, it is only defined for low frequency motion. The percentage of persons who may vomit, according to both standards, can be approximated by the relationship:

$$\% \text{ of persons who may vomit} = Km \times MSDVz, \tag{4.14}$$

where Km is a constant (1/3 for a mixed population).

This approximate value is based on exposures lasting for between 20 minutes and 6 hours, and with the prevalence of vomiting up to 70%.

In addition to the vertical vibration in the frequency range between 0.1 Hz and 0.5 Hz known to cause motion sickness, there is firm evidence (Griffin 1997, Vogel, Kohlhaas & Baumgarten 1982) suggesting that low frequency horizontal vibration may also be strongly nausogenic; in particular, that associated with repeated braking experienced while travelling in surface vehicles.

4.3.3 Recommendations for improving ride quality

It is recommended to maintain the acceleration as smooth as possible in order to maximise the agility of the vehicles (Hirshfeld 1932). Also, reducing or eliminating the parasitic movements of the vehicle helps in decreasing the severity of the disturbance due to starting and stopping phenomena. That is, vehicles must be as perfectly sprung and vibration free as possible and should run on a nearly perfect surface in order to preserve comfort during acceleration or deceleration manoeuvres.

Seat

The amount of motion transmitted to the passenger is greatly affected by the seat. A good seat design should prefer an erect position, with a backrest capable of attenuating the motions at higher frequencies but rigid enough to help stabilise the body at low frequencies. The design of the cushion is also very important, as it may amplify the effects at low frequencies. Some authors (Griffin 1997, Dreyfuss 1967) have suggested using suspension systems and seat cushions with natural frequencies close to those encountered in walking or gentle jogging (around 1 Hz -1.5 Hz) as they are well known to the body and may cause less discomfort.

Environment

In order to enhance the ride quality of a certain transportation system, it is advisable to control the passenger's surroundings. Maintaining a good ventilation, temperature and illumination levels at all times may reduce the overall perception of discomfort caused by motion and vibration. Noise is well known to worsen the annoyance caused by vibrating environments, so it should be maintained at the minimum possible level.

Motion sickness incidence can be reduced by minimising oscillations in the range of 0.1 Hz 0.5 Hz. Providing a clear view of distant objects and comfortable headrests, which reduce the movement of the head relative to the moving environment, may be beneficial. The effects of being able to look at the preceding vehicle when travelling in a convoy must be investigated, as this oscillating motion may produce adverse effects.

4.4 Passenger Safety

Motion can adversely affect the health of living beings. Tissue damage, joint injury, and even death, is likely to happen at relatively high accelerations. The lumbar spine and the connected nervous system, the neck-shoulder, the digestive system, the female reproductive organs, the peripheral veins, and the vestibular system are the most affected by whole body vibration and shocks. However, a definitive limit cannot be set because of the variability of subjects and situations in which motion may occur. Also, the health changes may take several years to develop, making their observation more difficult.

4.4.1 Human tolerance to acceleration

The highest level of sustained acceleration in transportation applications is commonly associated with emergency braking situations. The limiting factor for this value, apart from friction, is the ability of passengers to survive the event uninjured. The main factors affecting this ability are the event duration and the type of restraint provided (the highest deceleration survived voluntarily is around 440 m/s^2 facing forward with full restraint, attributed to Dr. John Paul Stapp in 1948).

For transportation applications, (Hoberock 1977) suggests a maximum emergency deceleration of 3.9 m/s^2 is acceptable for public transport applications. The AGTT program also specified the same value for PRT systems (UMTA 1976). This value is typical for hard braking of normal motor vehicles on pavement; skilled braking may be as high as 6.8 m/s^2. Some system designers even claim to be able to safely achieve emergency decelerations as high as 60 m/s^2. For comparison, some fairground rides may safely induce accelerations as high as 39 m/s^2 in the z direction and around 10 m/s^2 in the x direction for short exposure times.

4.4.2 Vibration and health

There are few experiments concerned with adverse effects of vibration on health. However, accelerations in excess of 9.81 m/s^2 are a cause of concern and magnitudes as high as 98.1 m/s^2 have been survived, but injury may be expected (Griffin 1997).

One way of evaluating the severity of the exposure of humans to vibration is described in (BSI 1987) in clause 4 and in appendix A. The weighting applied to the vibration occurring on the supporting surface in the z direction is W_b and in the x and y directions W_d. For the x axis on the backrest the weighting used is W_c.

For exposures of constant magnitude and crest factor below 6.0 the rms of the weighted acceleration should be determined over a period of at least 60 seconds by linear integration. Otherwise, the VDV procedure should be used. The total VDV of the combined environment may be determined by the fourth root of the sum of the fourth power of the vibration dose values in each axis.

Although there is no set limit, (BSI 1987) states that vibration magnitudes with a VDV in the region of 15 $m/s^{1.75}$ causes severe discomfort and going further increases the risk of injury.

ISO (1997) describes an evaluation method in which the rms acceleration is determined for each axis and assessed independently. Weighting W_d is used for the x and y axes with a multiplying factor $k = 1.4$, while the weighting applied to the z axis is W_k with factor $k = 1$. Guidance on the effects of vibration on health is given in annex B of the standard, which is informative only. It assumes that responses are strongly related to energy, and suggests that two exposures are equivalent when:

$$a_1\sqrt{T_1} = a_2\sqrt{T_2} \tag{4.15}$$

A 'caution zone' has been plotted using this equation. It indicates that exposures below it have not been clearly documented, and values over that zone are likely to cause injury. This recommendation is based on exposures of 4 hours to 8 hours and may not be useful for different exposure durations. The standard also indicates that some studies suggest a time dependence according to:

$$a_1\sqrt[4]{T_1} = a_2\sqrt[4]{T_2} \tag{4.16}$$

Also, a caution zone has been plotted for this equation. It only matches the previous zone in the 4 hours to 8 hours region. The energy-equivalent vibration magnitude corresponding to the total duration of exposure when two or more

G vector	Posture	Restraint	dv (m/s) (critical velocity change)	Pulse duration (s)	a (m/s^2) (plateau acceleration)
$x+$	Seated	Full	24	0.06	400
$x+$	Seated	Lap and shoulder	9	0.04	200
$x+$	Seated	Lap	9	0.06	150
$x+$	Standing	Holding rail	1.8	0.6	3
$x+$	Standing	None	0.8	0.7	1.1
$x-$	Seated	Lap	24	0.06	400

Table 4.2: Shock tolerance of the human body

periods are taken into account can be calculated by:

$$a_e = \sqrt{\frac{\sum a_i^2 T_i}{\sum T_i}} \tag{4.17}$$

or

$$a_e = \sqrt[4]{\frac{\sum a_i^4 T_i}{\sum T_i}} \tag{4.18}$$

According to this standard, for some environments such as when the crest factor is greater than 9, the previous methods may underestimate the effects of the vibration. For those cases, the VDV and running rms methods may be used.

4.4.3 Responses to mechanical shock

The total tolerance of the human body to shocks depends on many factors such as the posture of the body, restraint provided, shock direction, magnitude, duration and even on the shape of the waveform. Some early studies attempted to determine the maximum tolerable magnitudes of shocks with idealised shapes assuming different postures (Griffin 1997). Some relevant results are shown in table 4.2.

Realistic shocks never have simple shapes, and their magnitude may be difficult to determine, as well as their duration. Some researchers have suggested various methods to overcome such limitations, including the use of shock polygons, shock spectra, and the application of simple mechanical models to predict spinal injury. All these methods may be useful in certain circumstances; however, they are likely to underestimate the effects in other situations. The methods are not applicable to the assessment of vibration.

4.4.4 Occupant protection

During a vehicle collision the occupant continues to travel without significant velocity change and only stops when it is forced to do so by the interior of the vehicle or any restraining devices. This is known as *secondary collision* and is the one that causes injuries. Hence, knowledge of the interior of the vehicle is essential to assess injuries (Evans 2002). Commonly used injury criteria include the Head Injury Criterion (HIC) proposed by Versace (1971), Neck Injury Criteria and Thoracic Injury criteria (Eppinger, Sun, Bandak, Haffner, Khaewpong, Maltese, S., Nguyen, Takhounts, Tannous, Zhang & Saul 1999, NHTSA 2004).

The HIC is the most widely used criterion for railway vehicle crashworthiness; it is computed according to the following expression:

$$HIC = max\left[\frac{1}{t_2 - t_1}\int_{t_1}^{t_2} a\,(t)\,dt\right]^{2.5}(t_2 - t_1)\,, \tag{4.19}$$

where t_2 and t_1 are any two arbitrary times during the acceleration pulse. The acceleration is measured in g's and time is measured in seconds. The National Highway Traffic Safety Administration (NHTSA) proposed to limit the time interval to 36 milliseconds. Later, a 15 milliseconds time interval was also introduced. Tyrell, Severson & Marquis (1995) described a procedure to relate HIC to life threatening injuries. It showed that a HIC of 1000 produces a value of 18% life threatening injuries. This is also the limit established by NHTSA for HIC_{36}.

4.5 Evaluation Criteria for Automated Convoy Systems

The evaluation procedures reviewed in this chapter require taking direct measurements from the assessed vehicles during operation or during the events of interest. Additionally, it is assumed that detailed information about the vehicle's dynamic and physical properties is available. In the case of the proposed convoy system, which is not specific to a single transportation mode and for which vehicles and infrastructure do not exist, simulated measurements must suffice. Since only motion in a single dimension is considered in the simulations, the motion profile is not complete and cannot be compared with physical systems. However the simulation measurements can be utilised to compare the performance of the controllers, strategies and mechanisms object of this report.

The use of vibration dose value is adequate for the assessment of ride quality since the acceleration signal can be obtained from the simulation models. Also, the type of motion expected from convoy systems, which includes sustained acceleration/deceleration periods, vibration and occasional shocks, makes the VDV preferable to pure or weighted acceleration history or rms averaging.

The desired motion characteristic for the simulated convoy vehicles is that the longitudinal contribution to discomfort is minimal. If good ride quality is associated with an overall motion of 0.2 m/s^2 rms, equivalent to a VDV of 0.616 $m/s^{1.75}$ after 60 seconds, it would be ideal if the contribution to discomfort due to vehicle interactions is kept below an equivalent motion of 0.1 m/s^2 rms or a VDV of 0.31 $m/s^{1.75}$ after 60 seconds under normal operation. The 'not uncomfortable' boundary of 0.315 m/s^2 rms, or an VDV of 0.970 $m/s^{1.75}$ after 60 seconds is acceptable for entrainment manoeuvres for the vehicle executing it, but should not affect the rest of the convoy in the same measure. Vibration dose value should, under no circumstances exceed 15 $m/s^{1.75}$ under normal operation, since injury may occur.

The service acceleration and deceleration depend on the system being considered and on the permitted position of passengers. The chosen design values for acceleration and jerk limits are chosen as follows:

- Public transport carrying standing passengers should not exceed a service acceleration of $\pm 1\,m/s^2$. Emergency deceleration should be limited to $-1.5\,m/s^2$.

- Vehicles where passengers are seating facing either way should not exceed a service acceleration of $2\,m/s^2$ and $-2.5\,m/s^2$. Emergency deceleration should be limited to $-4\,m/s^2$ for unrestrained passengers or $-6\,m/s^2$ for restrained passengers.

- Jerk no greater than 2 m/s^3 under normal operation

Assessment of passenger protection during collisions also requires detailed information about the vehicles including structural design, dimensions, interiors, restraints available. There is no provision for this in the simulation models developed. It is possible to evaluate the square of the relative velocity between vehicles at the time of collision as a comparative measure of safety. The square of the collision velocity can be indicative of the severity of collisions since it is linearly related to the energy interchanged during the event and can be related to the injuries sustained by passengers (Carbaugh et al. 1998) in a more precise way than pure acceleration in the case of these simulations.

4.6 Conclusions

In this chapter the effects of motion and shock upon humans are reviewed. Emphasis is made on the aspects of motion thought to be more relevant to close headway convoy systems, where vehicle interactions may have a detrimental effect on passenger comfort.

The selected metric for ride comfort is the vibration dose value for its robustness and its nearly linear relation to human reactions to motion and vibration. The desired limits for convoy operation are set for normal cruise and for convoy manoeuvres. Additionally, jerk, acceleration and deceleration limits are set for normal operation and for emergency procedures for different types of vehicles and transportation modes.

For passenger safety, the selected evaluation metric is the square of the collision velocity between vehicles since it is an indicative of the energy interchanged during the collision.

Chapter 5

Convoy Concept and Bridging Dampers

5.1 Introduction

Up to this point, this work has identified the following:

- The limitations of current transportation systems
- The proposed vehicle automation technologies which attempt to solve them
- The shortcomings of said technologies
- The limitations imposed due to passenger safety and comfort criteria.

In this chapter the use of bridging dampers in short headway ATSs is discussed. First, the *'convoy'* concept is described; its principal characteristic being the role of the damping equipment as an integral part of the control system under normal operation, in addition to the traditional passenger protection role often documented in the existing literature regarding inter-vehicle devices, which is reviewed next. A functional description of the damping device is given at the end of the chapter. The proposed design is supported by simulations using a simplified vehicle model, and the implications of using the damper as part of the control system are studied.

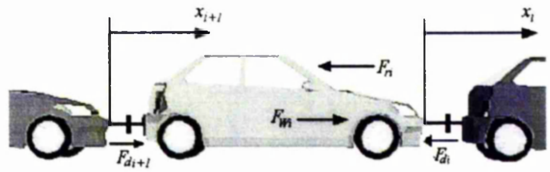


Figure 5.1: Convoy vehicles with bridging dampers

5.2 Convoy Concept

A *'convoy'* is formed by bringing a group of adjacent self-powered vehicles to be effectively in contact with each other through a non-coupling bridging damper as shown in figure 5.1. The vehicles remain self sufficient in terms of propulsion and control despite being in physical contact. The convoys would have to be kept apart at sufficient distance between them so that no collisions between neighbouring convoys are possible. For the sake of flexibility of the system, vehicles must be able to merge frequently to and from other lines, hence the necessity of dynamic reconfigurations of the convoys. Off-line stations would ensure non stop journeys even if speeds are low near high station density areas. Aerodynamic drag reductions between 40% to 50% for the inner vehicles in a convoy are possible at close vehicle spacing with adequate body design, allowing substantial energy savings (Zabat, Stabile, Frascaroli & Browand 1995).

Highly responsive vehicle control systems and powerful drivetrains often associated with short headway ATSs are not required for convoy operation because, having the dampers as backup, they are not the sole safety critical actuator in charge of spacing regulation. Moreover, the inter-vehicle dampers can effectively assist the control system to mitigate positioning or sensor errors, in situations of limited traction or in the presence of external disturbances such as wind gusts or the initiation of gradient by distributing loads among the vehicles. The physical contact between vehicles through the bridging dampers and the inherent safety they provide at short range permit the use of simpler range sensors, which, in turn, require less signal conditioning and data processing. The result is a faster, more robust regulation which is also economically viable, and which is not entirely reliant on wireless links.

5.2.1 Requirements for Inter-Vehicle Damping Devices

It is necessary to ensure that ride quality is preserved despite the fact that physical contact between vehicles also permits the transmission of shocks and vibration. This is achieved by avoiding high frequency components at the time when the buffers come into contact and isolating frequencies that are known to be annoying.

Damper force must be precisely controlled in order to minimise comfort degradation when corrections that require the intervention of the dampers are needed. Perturbations must be rejected and must not produce instability among the convoy; furthermore, they must be attenuated as they propagate along the string of vehicles. In addition, the energy dissipated by the damper must be minimal under normal operation.

The author identified four manoeuvres to be performed by vehicles operating in convoys, corresponding to four main operation regimes of the bridging dampers in which they complement the convoy control system to add safety without compromising ride quality in the most energy efficient manner. Additionally, an emergency mode is proposed. It takes place in the case of severe collisions among adjacent vehicles and requires the rapid dissipation of vast amounts of energy in a short time interval.

First operation mode: Cruise

This mode can be considered as the 'normal' operation mode since no action is being executed. Under this mode, vehicles should be able to reject relatively small disturbances without affecting comfort. Jerk and acceleration limitations, as well as technical considerations restrict the maximum magnitude of perturbations that can be attenuated by the control system and drive system alone. The damper, then, is required to reduce the effects of larger disturbances ensuring the correct position error regulation, but without affecting the ride quality when its contribution is not needed.

Second operation mode: Convoy changing velocity

Assuming that the convoy performs a planned change in velocity, the acceleration of the convoy can be matched to that of the slowest vehicle, in which case the manoeuvre is executed smoothly. The biggest safety threat occurs when vehicles perform close to their design limit. In this case, it is when vehicles decelerate at the emergency rate. Unplanned separations of vehicles are also a potentially hazardous scenario, since large relative velocities can develop. The dampers should ensure that the relative velocities are kept to a minimum under large loads and that the rejoining manoeuvre is smooth and without shock if the necessity arises.

Third operation mode: Vehicle joining the convoy

The joining manoeuvre can be performed at either end of the convoy, however, because of the nature of the controller considered here, it makes sense to perform it from the aft end. The critical factor under this operating mode is the amount of kinetic energy that needs to be removed so the vehicles remain together after contact. It will limit the maximum velocity at which vehicles can join the convoy. Theoretically, the convoy would travel at constant velocity while the joining vehicle accelerates to the approaching velocity. At a distance determined by the jerk and acceleration limits it starts it's deceleration so that its velocity is the same as the convoy by the time that contact happens. The accuracy of the incorporated position and velocity tracking systems is not expected to be enough to perform the manoeuvre to perfection, and external perturbations may be potentially dangerous at close range. The bridging dampers must be prepared to cope with those errors while producing the least possible disturbance to occupants.

Fourth operation mode: Convoy separation

In order to take different routes, a convoy needs to be separated on two or more smaller units. Sudden deceleration of the leading vehicles may cause collisions if no correcting action is performed. The bridging damper must provide safety when executing the manoeuvre until the critical separation point is reached. If vehicles make contact after the separation, the dampers must reduce the shock while reducing the relative velocity error.

Fifth Operation Mode: Severe Collisions Between Adjacent Vehicles

This operation mode is to provide adequate occupant protection in the event of severe rear end collisions among vehicles of the same platoon which may cause longitudinal accelerations well above those experienced during normal operation. In the event of severe collisions, the bridging dampers are required to dissipate vast amounts of energy in a short period of time. Current safety devices used in modern transportation systems achieve this by introducing crumple zones and devices which dissipate energy by permanent deformation of elements and structures designed for the purpose.

5.2.2 Lane Capacity

By reducing the inter-vehicle spacing to zero, capacity and infrastructure utilization are increased. Consider a convoy of identical vehicles of length l separated by a constant time headway h_v travelling at a constant lane velocity v_l . The lane capacity, expressed as vehicles per minute (vpm), can be defined as the inverse of the time that each vehicle 'slot' (i.e. the vehicle's length l plus the distance from its predecessor) takes to cross over a fixed point:

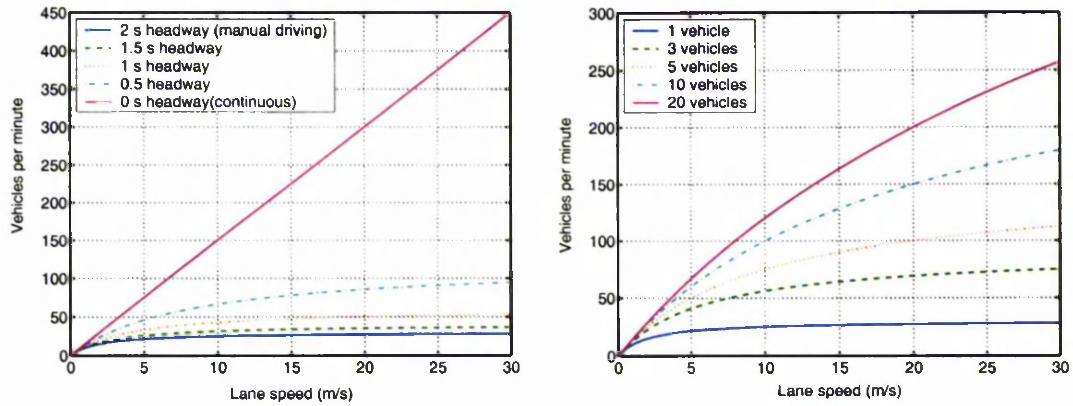
$$\text{capacity} = \frac{1}{h_v + \frac{l}{v_l}} \times 60. \tag{5.1}$$

Figure 5.2(a) shows plots of the capacity against lane velocity at different headway times ranging from 0 seconds, which correspond to a continuous string, to 2 seconds, which correspond to manual driving (*The Highway Code* 2004). The vehicle length is assumed to be $l = 4\ m$. According to this model the maximum capacity for manual driving (i.e. 2 s headway at 30 m/s) is 28.125 vpm. In order to obtain capacity twice that of a manual system, an automated system would require to operate at a time headway of 0.933 s instead of 2 s at a lane speed of 30 m/s. On the other hand, the maximum capacity under the 2 s headway policy at any positive speed is 30 vpm; a 7% increase from the maximum theoretical capacity of manual driving. In other words, capacity increases substantially with reduced inter-vehicle distance whereas a significant increase in velocity has a limited effect.

The definition of capacity presented above is flawed essentially in the fact that is impractical to implement a continuous string of equally spaced vehicles. Applying the same definition of capacity as above to the convoy concept results in the following equation:

$$\text{capacity} = \frac{n}{(n-1)\, h_v + h_c + \frac{n \cdot l}{v_l}} \times 60, \tag{5.2}$$

where n is the number of vehicles in a convoy and h_c is the time headway between convoys. Figure 5.2(b) shows this new definition of lane capacity for different



(a) Lane capacity at different headway times (b) Lane capacity at different convoy sizes

Figure 5.2: Theoretical lane capacity for autonomous driving

convoy sizes. The convoy headway (h_c) remains constant at 2 s for consistency while the vehicles within the convoy are in contact ($h_v = 0$). In this situation the vehicle throughput can be nearly doubled as compared to manual driving by using two vehicle convoys (53 vpm) and by using 20 vehicle convoys at the same speed, can reach 257 vpm; over 9 times the capacity of a manual driving lane.

5.2.3 Convoy Information Structure

The information structure is such that communications are minimised, requiring a central control unit which exchanges information with each vehicle at the station so it can plan its route and communicate it back before departing. The central control sets the lane speeds and desired convoy separation for each section of the track, communicating it by track-side markers. No further communication is required between the vehicle and the central unit unless the network conditions change drastically while enroute, in which case an update is given.

Leader's states information

The wireless link required to continuously broadcast the leader's states information to the following vehicles in traditional systems can be substituted in a convoy system by the information of a *"virtual leader"*. This virtual leader generated internally in each vehicle is unaffected by external disturbances; therefore, it can be a stronger reference for the coordination of vehicles than a real leader. An example of the virtual leader is shown in figure 5.3. The input to the virtual leader is the velocity requirement generated by the central and local controllers. The outputs are the leader's acceleration, velocity and position profiles with jerk and acceleration limits applied. If the vehicles are identical and the calculation of virtual leader profiles is identical for all the vehicles, their response is also identical and no relative position error occurs without external perturbations; thus enabling the convoy to comply with the requirements for the second mode of operation (section 5.2.1). This behaviour contrasts with the situation of systems following a real leader, where vehicles' response to the leader's commands depends on their position within the string (e.g. figure 3.11).

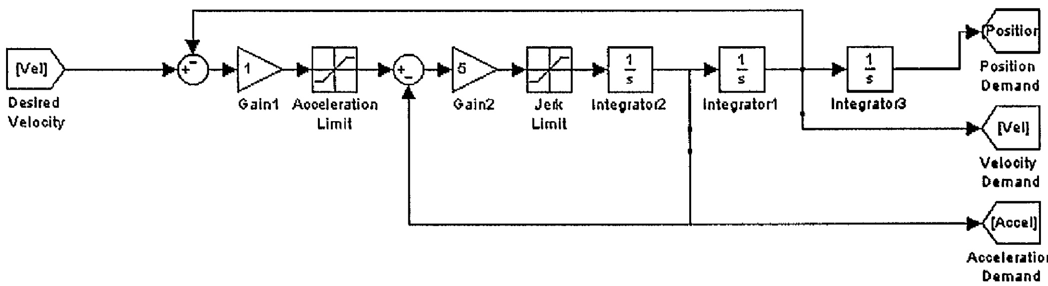


Figure 5.3: Simulink model of the "virtual leader"

Although the virtual leader strategy is not exactly an inertial navigation system, it suffers from similar drift characteristics, requiring frequent recalibration and synchronisation and is therefore not suitable to substitute wireless communications in traditional close headway systems. To demonstrate this statement the simulation in section 3.5.1 was repeated applying the virtual leader strategy to the system with the lowest gains (e.g. the system with values $a_1 = 2$, $a_2 = 1.2$ and $a_3 = 0.2$). In this case, the desired separation between vehicles was set to 0.6m. Figure 5.4 shows the results of the simulation when the leaders are identical in both vehicles (continuous line), when the following vehicle's virtual leader presents drift produced by integration constants 20% higher than its predecessor (i.e. $Gain1 = 1.2$ and $Gain2 = 6$) and no dampers are present (dashed line) and when the same drift as before occurs but a passive damper is present (dash-dot line).

The results show that indeed, with no drift the two vehicles are perfectly coordinated and no deviations from the desired separation occur. With integration drift, however, the vehicles collide when the deviation from the desired separation reaches -0.6 m, before the acceleration to 30 m/s is completed; this is despite the fact that acceleration and jerk limits are identical for both vehicles. With the use of dampers the deviation is minimal and is eventually eliminated. The characteristics of the damper used and the modelling of the collision event are detailed in section 5.4.

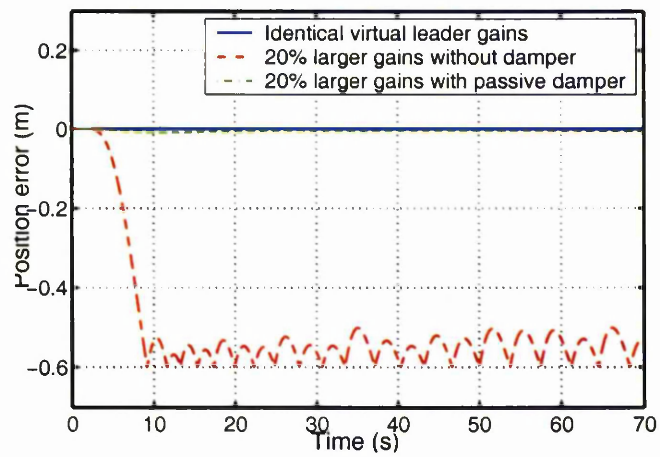


Figure 5.4: Effects of virtual leader's drift. Collisions may occur if the virtual leader strategy is implemented without the aid of dampers.

Preceding vehicle's information

During convoy operations, when adjacent vehicles are in physical contact, each vehicle is able to measure the relative states from its predecessor through the damper itself by using a suitable range and range rate sensor such as a quadrature encoder. When not in contact, but at close range, conventional laser or ultrasound can be used for the same purpose. This choice of simpler sensing equipment as compared to other close headway systems is possible for convoy systems because lower gains are necessary due to the better coordination and the additional actuation capability provided by the dampers. With these simpler short range sensors data processing requirements are also reduced since the single target of known characteristics is directly in front requiring only a single narrow beam for measurement and avoiding multiple target acquisition and parasitic reflections.

Further information requirements

The scheme described above permits safe operation of convoys without relying heavily on communications; however, enhancements are still possible through the use of non safety-critical data links. For instance, merge and separation manoeuvres are only possible if a given amount of coordination between vehicles exists. When a vehicle within the convoy requires to stop at an off-line station, for example, the surrounding vehicles should allow sufficient spacing to do it safely.

Also, since the vehicles in each convoy may respond differently due to load or wear, all should match the response of the weakest one. The leader would require every vehicle to inform of its maximum achievable velocity, acceleration and actuation time constant upon joining. The leader would then produce a control policy for the convoy containing the information of the weakest vehicle and broadcast it occasionally to ensure all the convoy members have the updated information. The vehicles can then adapt the acceleration gains (γ and ζ) to obtain the required value of a_1. Also the position and velocity gains need to be adjusted in accordance with the broadcast control policy. Finally, information about the virtual leader characteristics and states can be broadcast in the same message.

This communication link must be wireless, but unlike other platoon systems, stability and safety do not depend on the timely and constant delivery of information. Furthermore, the bandwidth consumed is minimal in comparison and the link can be shared with other services such as updates from the central network, toll collection and passenger information. Because of the low data rate required robust addressing protocols checks and coordination can be implemented making interference between convoys no longer an issue. A secure emergency channel would still be essential to ensure safe operation in the worst cases.

5.2.4 Safety

Short headway systems may offer additional safety advantages that are neglected if collision calculations are performed assuming a fixed unyielding barrier. If the automated vehicles are secluded from other traffic then collisions would only occur among similar vehicles in terms of both, mass and geometry, which simplifies the design of the external protection devices. More importantly, the relative velocity between the vehicles is not allowed to grow to dangerous levels thanks to the small inter-vehicle spacing.

In addition to these factors, inelastic collisions between vehicles of similar mass result in smaller sudden changes in velocity relative to the inertial frame of reference (i.e. the ground) than collisions against a fixed body. Consider, for example, a perfectly inelastic collision between two identical vehicles, one travelling at the lane speed of $v_1 = 20\ m/s$ followed by the other travelling at $v_2 = 25\ m/s$ at the time of impact. Conservation of momentum ($m_1 v_1 + m_2 v_2 = (m_1 + m_2) v_f$) indicates that the original velocity difference of 5 m/s produces a final velocity $v_f = 22.5\ m/s$ of the two vehicles. In that case the energy that needs to be dissipated by the vehicles' structures ($E_d = 1/2(m_1 v_1^2 + m_2 v_2^2) - 1/2(m_1 + m_2) v_f^2$) per unit of mass is 6.25 J/Kg. The drivetrain would have then the task of removing the exceeding 2.5 m/s over the lane speed (106.25 J/Kg). Against an unyielding wall however, the same 5 m/s original overspeed requires four times as much energy (25 J/Kg) to be dissipated entirely by the structure of one vehicle. This is equivalent to a vehicle to vehicle collision of twice the relative speed (10 m/s).

Collision velocity is strongly linked to both emergency deceleration and to the deceleration experienced by a failed vehicle. The bulk of references reviewed list failed vehicle decelerations between 5 m/s^2 and 10 m/s^2 with fewer citing optimistic values in the region of 2.5 m/s^2. The reason behind these values is the limited force that can be provided by the wheels. Only collisions with stationary objects with much greater mass than the vehicles themselves could produce greater decelerations; an unlikely event if the automated lane is secluded from other traffic (Shladover 1979). The third critical factor which determines collision velocities is the reaction time of vehicles to an emergency situation. This parameter is extremely uncertain as it depends on each implementation. Early figures considered in the literature range from 0.05 s to 1.3 s, usually compromising sensing decision making and actuation times (Shladover 1979). Newer developments, particularly computer control, allowed shorter times; Brader (1985) calculated a minimum reaction time of 0.3 s for computer controlled vehicles. Modern systems with electric actuators and today's off-the-shelf computers should allow faster reaction times. As a comparison, an average driver takes 0.7 s of to react to an emergency situation and a typical automotive brake system takes additional 0.5 s of actuation.

Figure 5.5 shows the effects of inter-vehicle spacing ($x-$axis) and different lane speeds (various lines) on mean vehicle to vehicle collision velocity 5.5(a) and distribution 5.5(b) within a group of 20 consecutive vehicles when a catastrophic failure occurs to the leading vehicle. For the experiment, the worst case scenario failure deceleration is 5 m/s^2 while the emergency deceleration is 0.4 m/s^2. The reaction time was 0.5 s. It was assumed that once that the collision between the first and second vehicle occurred, both would continue moving together at the velocity resulting from the inelastic collision and would decelerate at the failure deceleration. The Matlab script and Simulink model used in the experiment are shown in appendix C. Capacity of the system under the same conditions is shown in figure 5.5(c). The distance between convoys was in this case calculated to produce an unitary K safety factor, hence avoiding collisions between different groups of vehicles.

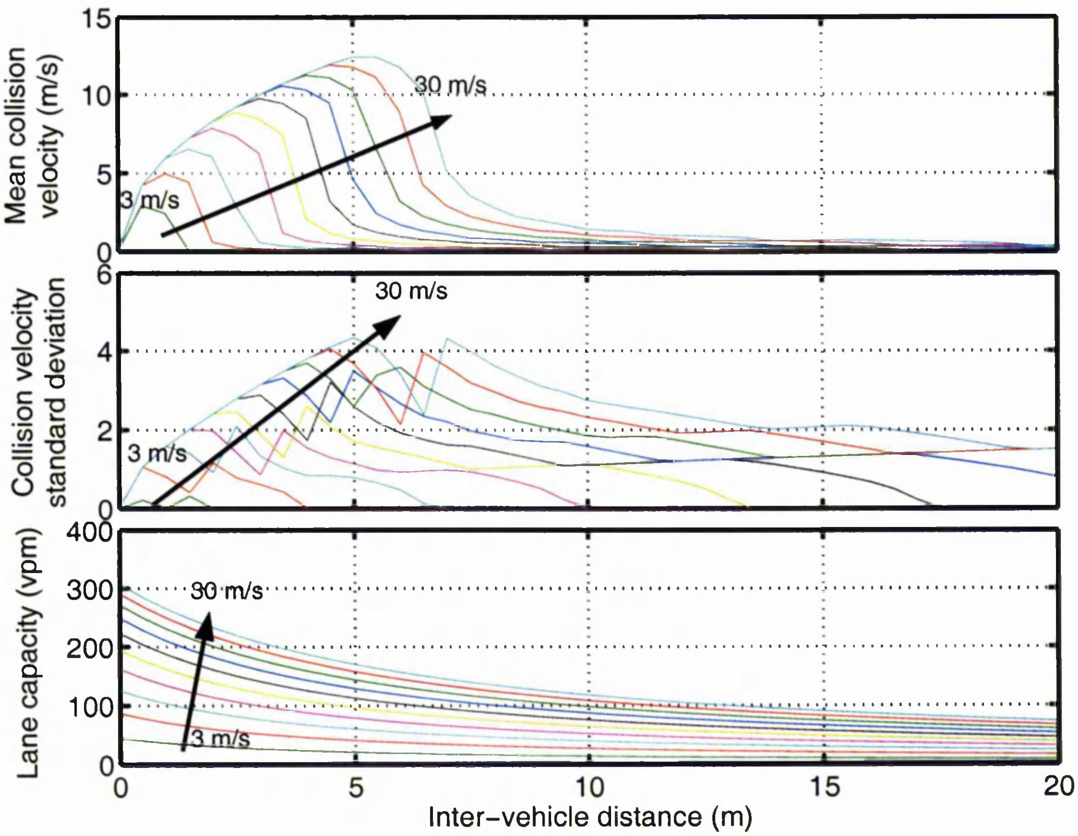


Figure 5.5: Collision velocities and distribution under catastrophic failure at different inter-vehicle spacings and lane velocities.

The experiment shows that the combination of high lane speed and intermediate distance results in more dangerous mean collision velocities among the group than at either very short or very large distances. The standard deviation of the velocities is also a cause of concern since at intermediate distances the collision severity is distributed unevenly along the group. This, in combination with the relatively high mean velocity, implies that some vehicles sustain severe collisions while others remain virtually unaffected.

The operating point of automated highway and platoon systems generally has the combination of intermediate spacings and high speed. The proposed convoy systems operate at zero or very small spacings. The only time spent at intermediate spacings is during convoy reconfigurations, and, assuming that this time accounts for a small fraction of the overall journey, the probability of fatalities resulting from rear-end collisions is small as compared to other high capacity systems.

5.3 Antecedents of Inter-Vehicle Safety Devices

Inter-vehicle safety devices have been briefly described in a variety of close headway ATS studies (e.g. Shladover 1979, Garrard, Caudill & Rushfeldt 1976, Brader 1985). The justification for requiring them is the increased risk of collisions due to the proximity of adjacent vehicles, especially at fractional safety K factors. In addition, the traditionally conservative transportation regulations are likely to require such a safety feature if fractional K factor systems are to be licensed for public operation since collisions cannot be entirely avoided in all situations, and dynamic entrainment and extrainment manoeuvres considerably increase the risk of dangerous collisions as shown previously in section 5.2.4.

The principal purpose of the safety devices described in the literature is to protect the passengers from injury in case of rear-end collisions. A secondary purpose of the inter-vehicle devices mentioned in the literature is to provide protection to the vehicle (Garrard et al. 1976). This last feature is necessary due to the frequency of minor collisions which may not present any danger to passengers but could damage the vehicle's structure or unnecessarily activate non-reusable passenger protection devices such as airbags or crushable structures, which could in turn cause injuries

The best possible protection is achieved by bringing the colliding vehicles and, more importantly, the passengers within them, to zero relative velocity over the longest possible period of time (Tyrell & Perlman 2003). The limiting factor for the deceleration time is the maximum distance that the safety devices can collapse.

It is necessary to acknowledge that any inter-vehicle device can only influence the outcome of the primary collision (between the two vehicles) but any secondary collisions (between passengers and vehicle interiors) need to be controlled by suitable restraints, seats and vehicle interior surfaces. Furthermore, the bulk of documents reviewed in this chapter agree that the likelihood of significant passenger injury during a collision of a given impact velocity is more sensitive to the details of passenger restraint system and the compartment interior geometry than to the effects of any exterior impact absorbers (see Irving et al. (1978) for PRT harness system details and NHTSA (2004) for automotive harness systems); however, most authors also agree that external absorbers are essential to control the velocity of secondary collisions.

Protection against secondary collisions can be enhanced by reducing the *'throw distance'* and spreading the loads upon the passenger's body on large contact areas. Rear facing seats have been used exclusively in some systems such as the Japanese CVS to achieve emergency decelerations in the order of 20 m/s^2 by clamping the guideway rail (Ishii, Takemochi, Iguchi & Koshi 1976). In a rear-end collision situation, however, the direction towards which the seats face is of little relevance since the acceleration vectors of the two vehicles involved have opposite directions. Kirkpatrick, Schroeder & Simons (2001) showed that relatively minor head impacts result from secondary collision velocities as high as 6.7 m/s by using only the backrest of the seat in front to restrain passengers in rail coach collisions.

In a study regarding human factors and hardware design for passenger protection Wilkins & Hullender (1975) claim that shoulder belts and hydraulic pneumatic devices could potentially protect passengers during collisions at up to 500 km/h with 274 m/s^2 peak deceleration for 0.2 s within 0.38 m if passengers wore adequate restraints (possibly inadequate for general public use). Using conventional restrains, however, the same study concluded that the probability of injury is low during collisions below 50 km/h if an external shock absolver with 25 cm of stroke is used.

In a similar fashion, Garrard et al. (1976) used combinations of hydraulic shock absorbers and crushable structures to investigate crash survivability. This combination of smaller stroke backed up by crushable portion had been identified as a way to save weight without compromising safety (Mayne 1974). The study found that it is relatively easy to protect passengers (excluding standees) by the use of adequate interiors during collisions of up to 7.5 m/s against a fixed unyielding barrier. At these speeds the external damper mostly protects the vehicle.

At collisions of 15 m/s a combination of crushable structures, external damper and adequate interior design can be expected to produce non-life-threatening injuries. At collisions of 30 m/s it is impossible to protect passengers with any combination of conventional systems. It has been noted that the magnitude of the traction or braking force at the wheels does not affect significantly the outcome of medium and severe collisions since the inertial load is at least one order of magnitude greater than the maximum traction achievable through conventional friction methods. For milder collisions, Brader (1985) considered a bumper with 0.6 cm of stroke which could absorb a 3.5 m/s impact without exceeding 10 m/s^2.

5.4 Damper Design and Modelling

5.4.1 Normal Operation; Modes 1 to 4

Vertical suspension designs (e.g. Keum-Shik, Hyun-Chul & Hedrick (2002) and references therein) and inter-vehicle vertical suspensions for rail applications (see (e.g. Mei & Goodall 2002)) are often characterised by an elastic restoration force and a viscous resistive force. The bridging damper was designed in a similar manner since the combination is known to provide superior comfort and protection (Mayne 1974). The elastic constant (k') and the viscous constant (c') were optimised for minimum force application during the vehicle interactions. Peters (1997) states that, given an impact velocity (v_0) and a maximum stroke (x_m), the minimum impact force against a fixed barrier is obtained by choosing a damping coefficient $\varsigma = 0.404$ and a natural frequency $\omega_n = 0.6 v_0/x_m$. Assuming that the collision happens between vehicles of identical mass m and not against a fixed body, the natural frequency and damping coefficient are given by:

$$\omega_n = \sqrt{\frac{2k'}{m}} \quad \text{and} \quad \varsigma = \frac{c'}{2\omega_n}. \tag{5.3}$$

The resulting constants for minimum impact force are:

$$k' = \frac{0.3606\ m\ {v_0}^2}{2\ {x_m}^2} \quad \text{and} \quad c' = \frac{0.4851\ m\ v_0}{2\ x_m}. \tag{5.4}$$

Acceptable collision velocity was set at $v_0 = 3.5\ m/s$ and a maximum damper stroke of $x_m = 0.6\ m$. The maximum collision acceleration experienced by the vehicles under these conditions is 5.3 m/s^2.

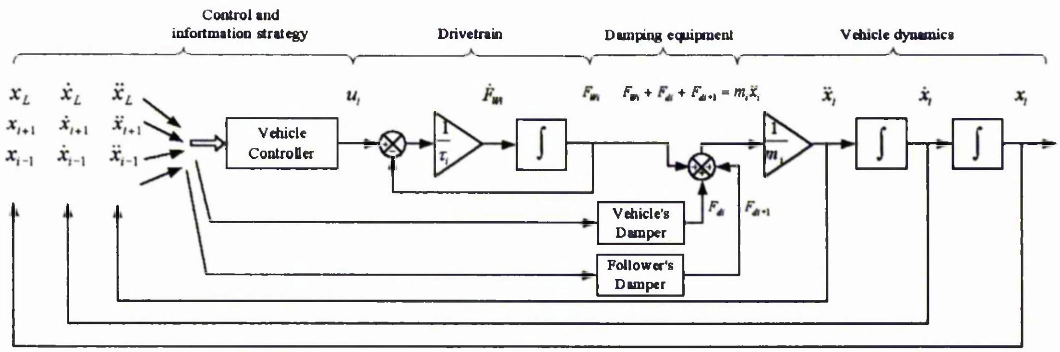


Figure 5.6: Linear model of convoy vehicle with damping equipment

With this arrangement the damper mounted on the ith vehicle can be represented as a piecewise system with the following characteristics:

$$F_{di} = \begin{cases} k'_i\,(x_{i-1} - x_i) + c'_i\,(\dot{x}_{i-1} - \dot{x}_i) & \text{iff } (x_{i-1} - x_i) < 0 \text{ and} \\ & k'_i\,(x_{i-1} - x_i) < -c'_i\,(\dot{x}_{i-1} - \dot{x}_i)\,; \\ 0 & \text{otherwise.} \end{cases} \tag{5.5}$$

The addition of the dampers to the vehicle model is shown in figure 5.6. When the damper forces are not zero, the describing equation that results from this modification is:

$$\begin{aligned} &\dddot{x}_i + \left(a'_{3i-2} + \tau_i c_i + \tau_i c_{i+1}\right)\ddot{x}_i + \left(a'_{3i-1} + \tau_i k i + \tau_i k_{i+1} + c_i + c_{i+1}\right)\dot{x}_i \\ &+ \left(a'_{3i} + k_i + k_{i+1}\right)x_i - \left(\gamma_i + \tau_i c_i\right)\ddot{x}_{i-1} - \left(\beta_i + \tau_i k_i\right)\dot{x}_{i-1} - \left(\alpha_i + k_i\right)x_{i-1} \\ &- \left(\gamma_{ri} + \tau_i c_{i+1}\right)\ddot{x}_{i+1} - \left(\beta_{ri} + \tau_i k_{i+1}\right)\dot{x}_{i+1} - \left(\alpha_{ri} + k_{i+1}\right)x_{i+1} \\ &= \quad \zeta_i \ddot{x}_L + \kappa_i \dot{x}_L + \epsilon_i x_L - \alpha_i(l_{i-1}). \end{aligned} \tag{5.6}$$

It is a similar equation to that of the bidirectional case and has been analysed in a similar way (notice that the constants k'_i and c'_i have been divided by m_i to produce k_i and c_i).

Passive damper

The use of a simple passive damper with these settings in convoys presents several problems; for instance, the obvious inability to adjust the reaction according to the magnitude of the impact to minimise discomfort. Also, it is impossible to select a combination of values for the controller which complies with the criteria outlined in section 3.5 for contact and no contact situations at the same time. This is the result from the comparatively large gains provided by the damper combined with its low damping ratio necessary for passenger comfort.

Another flaw of a passive system is that the viscous portion of the damper's force can cause frequent shocks due to the sudden application of force at the time when the vehicles initially come into contact at a given velocity (González-Villaseñor et al. 2003*b*). Pre-load of the elastic element also produces the same undesirable effect and should be avoided.

A third complication arises from the bounce produced when the vehicles separate and have to rely entirely on the controller and drivetrain to return to stability. This bounce also compromises safety because of the intermediate distances that may result.

Typical buffers and motion arrestors generally are a form of non-linear passive dampers. Normally the size of the discharge orifice changes as a function of deflection in an attempt to reduce the initial shock while maintaining the same average force. These type of dampers are also unsuitable because the bounce and low damping ratio characteristics are exacerbated near the operation point.

Semi-active damper

In order to overcome the limitations cited above, it is therefore desirable to achieve a variable response from the damper to adapt to different conditions of the journey. Typical semi-active devices allow changes in the viscous portion of the force to produce the desired response according to a suitable control law such as the 'skyhook' strategy introduced by Karnopp, Crosby & Harwood (1974) for vehicle suspensions. This strategy allows for a fictional damper to be inserted between the vehicle's body and the stationary sky to suppress the vehicle's body motion (Kitching, Cole & Cebon 2000, Keum-Shik et al. 2002).

Typically, semi-active dampers are electro-hydraulic devices, although the use of variable viscosity fluids (electro and magneto -rheological fluids) (Petek 1892) and electromagnetic devices have been studied (Karnopp 1989).

It is possible to reduce the constant k_i of the elastic element and compensate for it via the controller. The resulting device may have adequate response when the relative velocity is sufficient but is unable to restore sufficient energy to the system since the process becomes mostly dissipative. The inability to control the elastic portion of the reaction results, therefore, in either the same deficiencies as the non-linear passive arrestors or in insufficient force to separate the vehicles in the presence of persisting low frequency disturbances such as a prolonged downward slope.Hence, a semi-active device in which only the viscous constant can be varied is unsuitable for this purpose.

Active damper

Active devices have also been successfully used in vertical suspensions (Lin & Kanellakopoulos 1997, Yue, Butsen & Hedrick 1989, Stein & Ballo 1991). Fully active systems produce the control force with a separate actuator unit typically hydraulic or pneumatic but also electromagnetic systems have been used with success (Jones 2005). External energy input and amplifiers are required, resulting in higher cost and weight of the complete system. Although active systems generally outperform semi-active systems, this increased cost means that more attention is given to semi-active suspension systems.

Active dampers provide a suitable solution for the problem investigated here since they can produce the response required by the control law despite the dynamic states of the vehicles. However, the increased cost and energy waste are not coherent with the high efficiency and reasonable cost characteristics desired for this project.

Quasi-active damper

The solution that the author presents is the 'quasi-active' device shown in 5.7. In the compression stroke (downwards in the figure) hydraulic fluid from the bottom chamber of the damper is forced to the upper chamber through the check valve in the piston. This check valve presents no restriction in that direction. The fluid displaced by the connecting rod is evacuated from the cylinder through the port in the upper chamber and then forced through the variable discharge orifice which allows adjustments to the viscous constant in a similar manner as a semi-active device. The fluid then fills the reservoir pushing the gas to the pressure repository. A diaphragm is necessary to separate the gas and the hydraulic fluid to avoid dissolved gas and bubbles trapped in the hydraulic circuit, which would deteriorate its damping performance and bulk modulus (Audenino & Belingardi 1995).

On the extension stroke the pressure in the reservoir pushes the hydraulic fluid through the second check valve towards the lower chamber of the cylinder and forcing the damper to extend due to the difference in area presented by the piston in the upper and lower chambers. Adequate damping control is retained

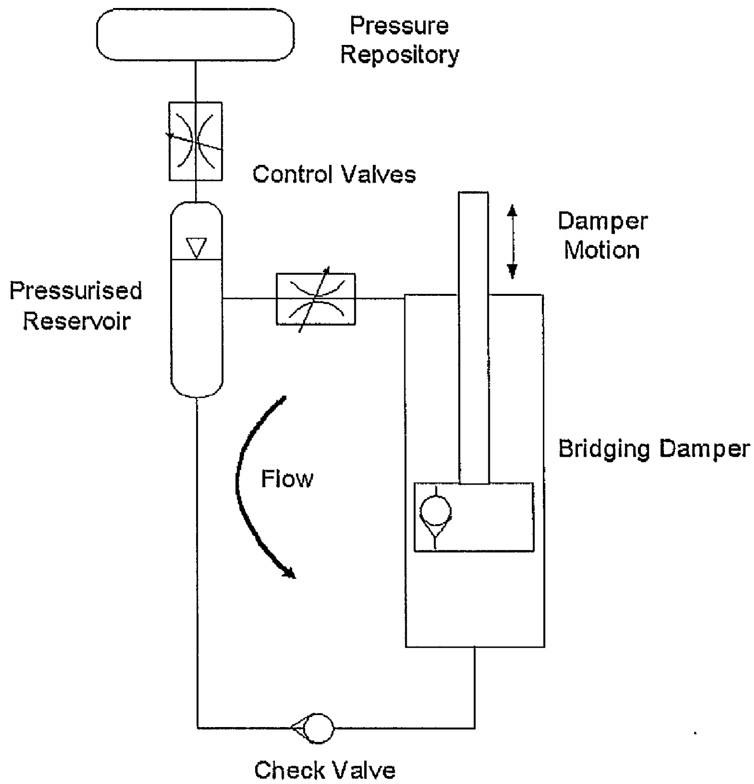


Figure 5.7: Damper design concept

thanks to the check valve in the piston by only allowing the upper chamber to be evacuated through the variable orifice.

The elastic constant is regulated by controlling the gas pressure in the reservoir. The gas under pressure is sourced from the gas repository, which may be shared with the vertical suspension members if a similar control method is adopted. This is the energy storage element in the system since pressure is restored to the repository during each compression stroke while inhibiting the bounce. A similar device for heavy vehicle suspensions, with exception of the elastic element, has been previously studied (Kitching et al. 2000) with encouraging results.

Figure 5.8 shows the model of the damper designed to be used with the Convoy Toolbox. It can be used to simulate the passive damper, semi-active damper and quasi-active damper. Constants k and c represent the maximum values obtainable through the control system for that particular design. These values are calculated according to equation (5.4) (The value of c is adjusted below). Control of the actual values of the constants is obtained by multiplying them by outputs of the spool valve block, which presents a first order lag with time constant of 0.1 s. Cubic stiffening was not used in any of the simulations. No fluid dynamic losses are considered and the effects of compressibility and inertia of the hydraulic fluid and piston assembly were modelled. Simulation showed that these last two did not contribute significantly to the performance of the device provided that the fluid and piston inertial mass was under 5% of that of the vehicle. This is because of the low power required as compared to traditional suspensions. Therefore they were discarded from further simulations. The mechanical fuse block is described in section 5.4.3. The logic blocks provide the means to discern when the damper has made contact and when the limit of stroke has been reached, after which point the deformation of the mechanical fuse commences.

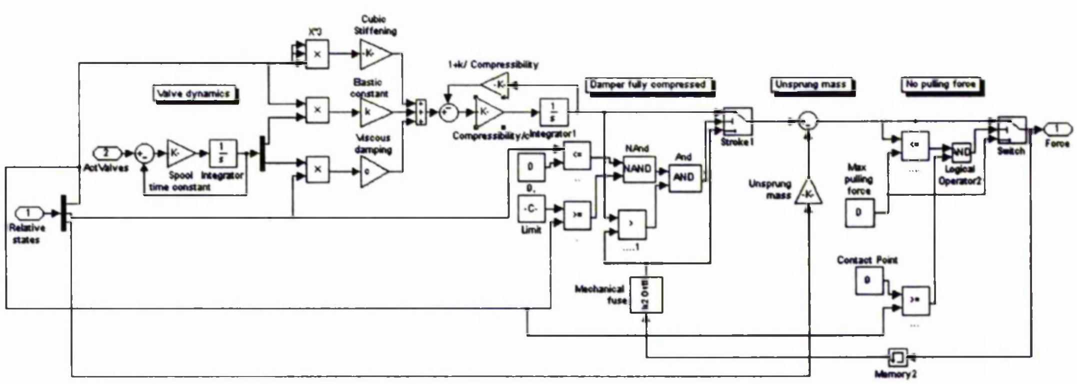


Figure 5.8: Simulink model of the damper used in the simulations

5.4.2 Damper Control Law for Convoy Operation

To satisfy the requirements from section 5.2.1, the control law governing the damper's response must be such that:

1. Reactions to small perturbations near the point of contact are weak so they do not propagate through the damper's reaction and do not cause discomfort (operation mode 1).

2. Dampers are not allowed to fully collapse, even if it means sacrificing ride quality in the presence of long duration compressive disturbances (operation mode 1).

3. Vehicles do not bounce after contact (operation modes 1 and 3).

4. Shocks at the time of contact are avoided (operation mode 3).

5. Vehicle's response does not continuously rely on other vehicles' drivetrains by pushing them or being pushed by them.

6. Protection is provided when the possibility of strong collisions exists (operation modes 3, 4 and in preparation for 5).

When the damper deflection, d, is $0 > d > -0.5x_m$ (notice t hat d is negative under compression) and the relative velocity is negative (compressing motion), the control law is such that the optimal values of K and C are continuously calculated with equation 5.4 using the current relative velocity (i.e. $\dot{x_i} - \dot{x_{i-1}}$ in place of v_0 and the distance remaining before reaching $-0.5x_m$ (i.e. $d - 0.5x_m$) instead of x_m. The compressive motion is, hence, treated according to its severity by attempting to stop it within the same distance ($0.5x_m$) with the smallest force possible. Simulations suggest that this approach effectively addresses the first design objective. It also has the advantage of creating a pseudo-equilibrium range at relative velocities around zero; effectively a *'buffing range'*.

Objective number 2 can be met by increasing the constants K and C to their maximum value(i.e. from equations 5.4 once the damper is compressed beyond $-0.5x_m$. The damper then resumes its normal operation of comfort keeping when the dampers re-extend at least to $-0.5x_m$.

In order to meet objective 3 it is possible to reduce the value of the elastic constant K to its minimum when in extension motion and deflection is in the range $0 > d > -0.5x_m$, so that the damper induced bounce is inhibited. The value of C is of little importance because of the low velocity, but it may be worth keeping it at its maximum in preparation of a collision.

To further preserve ride quality when contact first occurs and the portion of the control law regarding objective 1 is about to start, it is possible to reduce the viscous constant C to avoid the initial shock. Meanwhile, the elastic constant is momentarily set at the maximum so safety is maintained in case of malfunction. It was found through experimentation that a value of the altered viscous constant $\hat{C}$ that depends on the deflection d following the relationship

$$\hat{C} = \begin{cases} -10\, d \cdot K & \text{iff } -10\, d \cdot K < C \text{ and} \\ C & \text{iff } -10\, d \cdot K \geq C \end{cases} \tag{5.7}$$

could adequately meet objective 4. The drawback of this implementation is that the effective power dissipation per stroke is reduced and the calculated constants fail to stop the motion before $0.5x_m$ is reached. This is easily compensated for by increasing the effective value of the viscous constant by 20% to achieve the same average power dissipation (i.e. $1.2c$).

A major cause of concern when introducing the bridging dampers to convoy systems is that vehicles would have the opportunity to persistently transfer their loads to their adjacent vehicles when it is not essential for safety or stability purposes. Initial experiments showed that compressive loads were transferred to the front vehicles. This was especially evident with permanent perturbations when constant braking effort was needed, such as noticeable speed reductions and downward slopes. In such situations the leader ended carrying almost the entire weight of the group. This is unacceptable because it not only causes excessive wear, but also causes the entire convoy's position accuracy and ability to stop promptly to be destroyed.

Transient perturbations such as a joining vehicle coming into contact with the convoy at excessive speed cause unnecessary degradation of comfort because of the same phenomenon as above. In this case the control system unloads the vehicle's drivetrain when the relative velocity is suddenly reduced by the damper after contact. The offending vehicle then transfers a greater portion than strictly necessary of its inertial load through the damper, thus causing greater discomfort. The offended vehicle behaves then in a similar way allowing the perturbation to reach further into the string, causing yet more disruption to passengers of other vehicles.

The opposite event, in which a vehicle loses velocity due to perturbation or failure, presents a situation that is not only acceptable, but also desirable since it promotes progressive deceleration of the following vehicles in a safe manner giving opportunity to recognise an emergency through the virtual leader's reference. In such case, once the emergency is declared, the damper makes possible for a vehicle or group of vehicles to push the failed one towards the next station at reduced speed.

The previous characteristics, adverse to objective 5, cannot be addressed directly on the damper control without robbing it the ability to perform its principal duties. It is more sensible to address the problem by complementing the drivetrain control by demanding from it to produce a reasonable fraction of the force produced by the damper. Naturally, the exact number depends on the specific implementation, but it has been found from experiments that around 20% of the damper's force is sufficient to avoid unloading but does not contribute to discomfort.

To eliminate the transmission of permanent loads an integral part was added, but found to be inadequate as it tended to either be too fast as to cause large bounces or excessively slow to be useful. Furthermore, it was exceedingly sensitive to parameter variations on the controllers. The reason for its failure is the non-linearity at the equilibrium point since there was no suitable way of determining when it should stop growing, when it should be reset and to which initial condition. The solution implemented was to progressively inject a force from a value of 0 to a value similar to the damper's force but with a viscous constant $C_m = 2.5C$ to discourage bounce. The demand to the drivetrain should end upon separation but this second portion should also be removed progressively.

The damper control law and complementary drivetrain control were modelled in Simulink and the resulting diagrams are showed in figures 5.9 and 5.10 respectively. They were designed to be used with the Convoy Toolbox and were used in all the simulations. The control outputs to the damper's valves are expressed as a fraction of the values of k and c to achieve the desired K and C. They have a range from 0.1 to 1.0 and from 0.1 to 1.2 respectively.

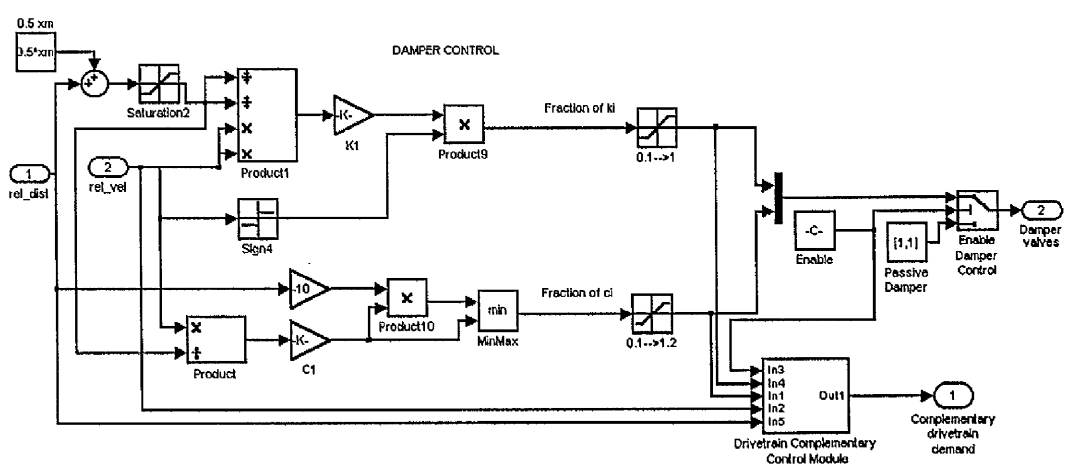


Figure 5.9: Simulink model of the damper control module

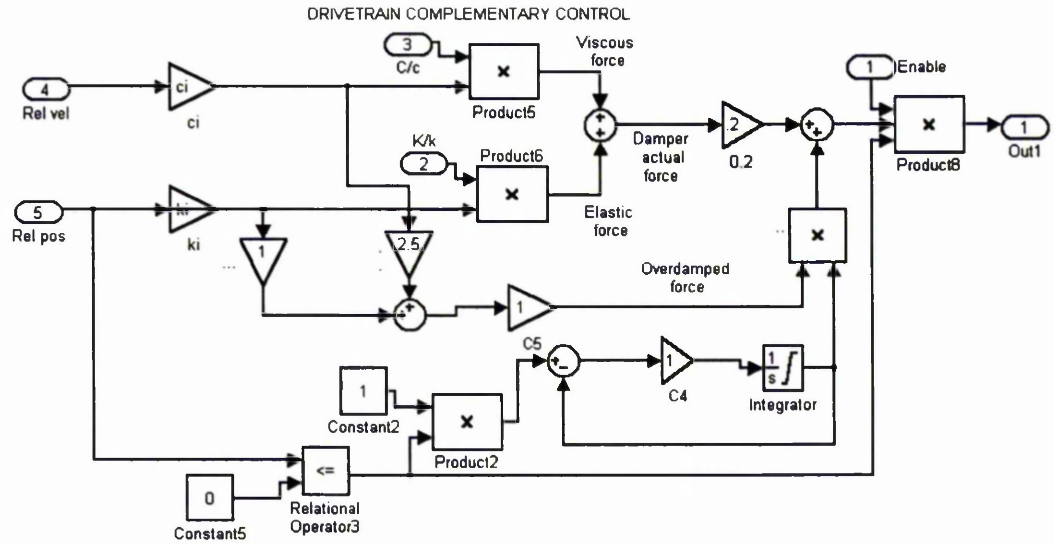


Figure 5.10: Simulink model of the complementary drivetrain control module

5.4.3 Severe Collisions, Mode 5

Since the beginnings of close headway transportation research a wide variety of materials and technologies for crushable or otherwise non-reusable elements have been investigated for their application as passenger protection mechanisms against collisions. From hollow elastomer cubes and elastomer extrusion devices (Tundermann 1975) to aluminium honeycomb structures (Brown, Weinstock & Rossetos 1976), the main objective is the same; to dissipate the collision energy in a controlled manner with the best possible efficiency. Energy absorption efficiency is optimal if the force/deformation characteristic has the largest possible area for a given maximum force (square shaped) (Shladover 1979). It is also highly desirable that the devices are lightweight and cost effective.

Figure 5.11 presents a typical force/deformation characteristic during impact often desired for crushable structures designed for collision survivability. The curve presents high initial peak load followed by significantly lower load, which should be approximately constant. Use of similar curves for rail (Tyrell & Perlman 2003) and automotive (Witteman & Kriens 1998) applications is widespread practice. In automotive applications typical decelerations for mild collisions are in the range 200 m/s^2 to 300 m/s^2 for durations of around 60 ms. Typical loads can be between 200 kN to 600 kN.

One implication of the force/crush characteristic shown in figure 5.11 is that

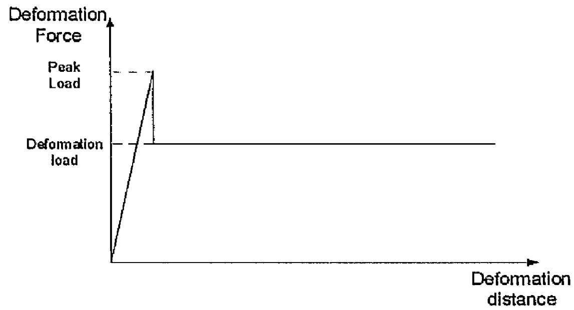
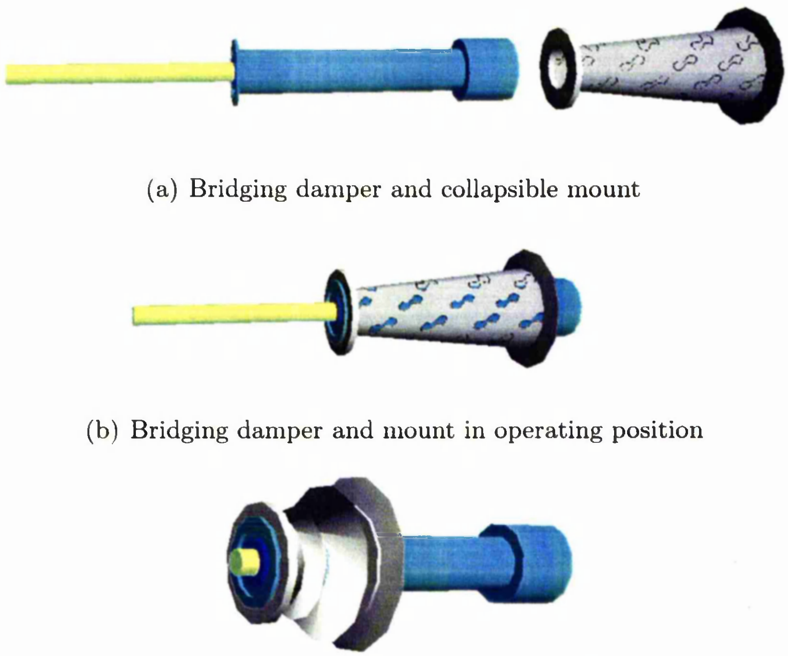


Figure 5.11: Force/deformation characteristic of typical crushable structures for land vehicle collisions

the crush will be focused on the colliding vehicles. If two convoys, which are made up of cars with the same crush characteristics collide, at best only the two colliding vehicles will crush (Tyrell 2001). If progressive crush load was allowed the collision would be better distributed among several vehicles preserving the interior volume but also increasing the secondary collision velocity (Tyrell & Perlman 2003).

Unlike railway coaches, the smaller convoy vehicles have low length to width ratios. This geometry discourages vehicles from engaging each other (bending onto each other) during collisions because the shorter lever arm requires greater lateral force component to initiate any lateral deflection (Tyrell 2001). If the convoy vehicle is twice as long as it is wide, the lateral component would require to be at least 50% of the longitudinal force for the vehicle to deflect. A railway car would typically need 17% of the longitudinal force to turn. Also, unlike road vehicle collisions, impacts in convoy operation are rear-end and full overlap only. These are advantageous characteristics of as the collision dynamics can be better predicted allowing for simpler, cheaper and lighter solutions. The only drawback is that lateral collisions at junctions can be potentially more dangerous than longitudinal ones (McGean & Lutkefedder 1973).

If the collision velocity is in excess of that for which the damper was designed, further protection is provided by a mechanical fuse in the form of a conical perforated tube into which the damper is mounted (Figure 5.12(a). The tube should be calibrated to produce a constant force of around 180 N to 200 N per kilogram of vehicle mass once past its yielding point. This ensures that collisions of 15 m/s against a fixed unyielding barrier or 30 m/s against a similar vehicle can be survived if the device is allowed to collapse 0.6 m. The sustained deceleration



(a) Bridging damper and collapsible mount

(b) Bridging damper and mount in operating position

(c) Bridging damper and mount after severe collision

Figure 5.12: Bridging damper and mechanical fuse mount assembly

would remain under 200 m/s^2. A similar device is documented by Mochidome, Masukawa, Suzuki, Kashiwa, Sadamitsu & Kouno (2003) to be in use for the Singapore Crystal mover. It allows for collisions of a 15 tonne vehicle at $3m/s$ and a deformation of 0.5 m to be survived by standing passengers. Further protection could be provided by a carefully calibrated explosive discharge in the damper mechanism at the time of impact to momentarily boost its energy absorption potential and possibly prompting milder secondary collisions with the interiors.

The refinement required to accurately model the collision response of a complete vehicle requires a large number of elements (around 50,000 to 250,000). This combined with the small time step (approximately 5 μs) of the simulations would require extensive computing time and resources and was considered out of the scope of this thesis. Rigid bodies are used throughout the simulations and the dynamics of the deformation are modelled simply by producing the peak initially and switching to the constant force. This means that only the first collision of each vehicle is valid.

5.5 Conclusions

The convoy systems concept was presented in this chapter, being the inclusion of the damper into the control system its main difference from comparable close headway systems. The requirements for such device were outlined and four normal operating modes were identified. These include cruise, acceleration/deceleration manoeuvres, vehicle merging and separation. An emergency mode in the event of collisions was also outlined. It was shown that lane capacity could be 9 times greater than that of manual driving systems by using twenty vehicles per convoy while the probability of dangerous collisions was reduced. was shown to be increased. The unique information structure which features a virtual leader was made possible thanks to the introduction of the damping equipment. Literature relevant to the damping equipment was reviewed and a quasi-active system was proposed. It is the preferred technology for this application. The non-linear control law was detailed and verified through simulation for which Simulink models were produced, as was the physical characteristics of the damper and its crushable mount.

Chapter 6

Model and Simulation of a Convoy of Hybrid Electric Vehicles

Current trends in automotive research and development indicate that an increased degree of automation is anticipated in future vehicle designs, implying also an increased commonality and greater regulation of vehicle specifications. Although present developments aim towards vehicle features for enhanced safety and driveability and superior driver assistance, future technology is expected to gradually replace human drivers with fully automated systems. The automotive industry is also facing increasing demands for emissions and energy waste reductions and for an increased utilisation of alternative fuels and propulsion systems. At the same time, road transport authorities around the world are pressed to solve congestion, to increase urban mobility and to provide safer roads.

The use of Hybrid Electric Vehicles (HEVs) is seen by many automotive manufacturers as an interim measure until the technologies which will ultimately replace fossil fuels are mature enough for mass production. The technologies that make HEVs possible, as is the case of technologies which would allow fully automated driving, have being evolving rapidly in recent years. Yet, further developments are required to make the technologies affordable in order to achieve adequate market penetration. The author envisions a similar time-scale for both, the widespread use of vehicle automation and the achievement of a substantial market share for HEVs.

In addition to the numerous benefits attributed to HEVs in terms of fuel economy and performance, they are particularly well suited for a practical application of the convoy concept. Combined, both technologies in a single transport system, they prompt an interesting possibility; the possibility of introducing privately owned vehicles which are capable of being manually driven on conventional roads, but also can be linked to automatic networks in high traffic areas such as city centres, where the vehicles can be automatically parked. This dual mode system would allow a door to door service with the advantages of private cars but without their worse disadvantages. Even public services could drive to areas where it is not feasible to build automatic guideways and be linked to the automated networks near the city centres, bringing flexibility, convenience and efficiency to the public transport industry.

The highly responsive hybrid drivetrains allow superior manoeuvrability and position regulation than typical thermal engines. Furthermore, the vastly automated traction control and power management modules are an ideal base for interfacing with the convoy controllers and with the infrastructure. In the long term, it is likely that the knowledge and developments from HEVs will be incorporated to the next generation of "petroleum-free" automobiles.

In the present chapter a HEV model is developed and used to form a simulated convoy. This realistic vehicle model is used to verify the implementation of the convoy systems philosophy. The damper design and its control law are tested under a variety of conditions which attempt to mimic portions of real life journeys. The convoy system is compared to an equivalent system which does not feature inter-vehicle dampers.

6.1 Vehicle Model

The HEV modelled in this chapter to perform as a member of a convoy is a small hybrid passenger vehicle with a drive-train that consists of a Westinghouse AC induction motor and converter rated at 75 kW continuous and a Geo 1.0 l internal combustion engine rated at 41 kW connected in parallel configuration (i.e. either or both of them drive the transmission at one time). Electric power is supplied by 26 12 V lead-acid batteries rated at 26 Ah which supply 300 V at 70 kW. The calculated mass of the whole vehicle is of 1350 kg. Its drag coefficient is 0.335 and has $2m^2$ frontal area. The original HEV was studied by using ADVISOR (ADvanced VehIcle SimulatOR), which is an HEV specific simulation toolbox for MATLAB designed to assist in testing different vehicle configurations using standard and custom driving cycles or test procedures. It was distributed freely by The National Renewable Energy Laboratory (NREL) from the U.S. Department of Energy until July 2002, when AVL was granted the license to further develop it for commercial release.

The ADVISOR model's subsystems are often validated against real parts and the detail level allows researchers and designers to analyse their individual performance under different conditions and settings; however, they are unsuitable for the simulation of convoys where multiple vehicles are simulated simultaneously without much use for the details of individual parts. After careful consideration and study of the original model, it was decided that a simplified model, which was compatible with the requirements and aims of this investigation, could be extracted. The typical block diagram of a model generated by the Convoy Toolbox is shown in figure 6.1.

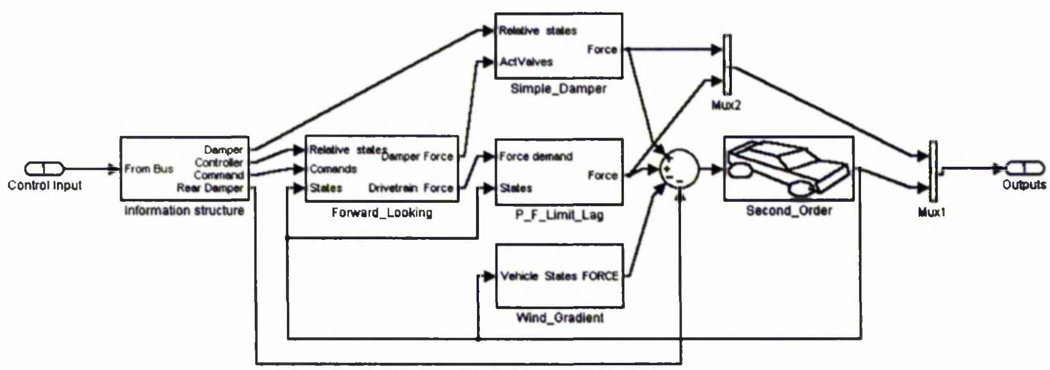


Figure 6.1: Simulink model of a convoy vehicle generated by the Convoy Toolbox

6.1.1 Vehicle dynamics

The base of the simplified model is the Newtonian dynamic equation. Since this is a model of a real vehicle, it is essential that it includes the effects of losses and resistive forces in addition to the forces considered in previous chapters. The equation of motion from previous chapters is modified accordingly as follows:

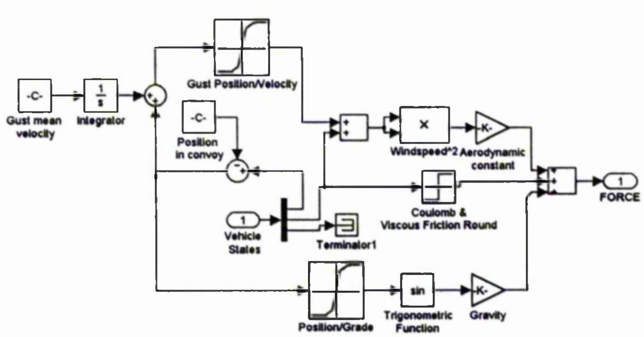
$$m_i \cdot \ddot{x}_i = F_{Wi} - F_{ri} - F_{di} + F_{di+1}, \tag{6.1}$$

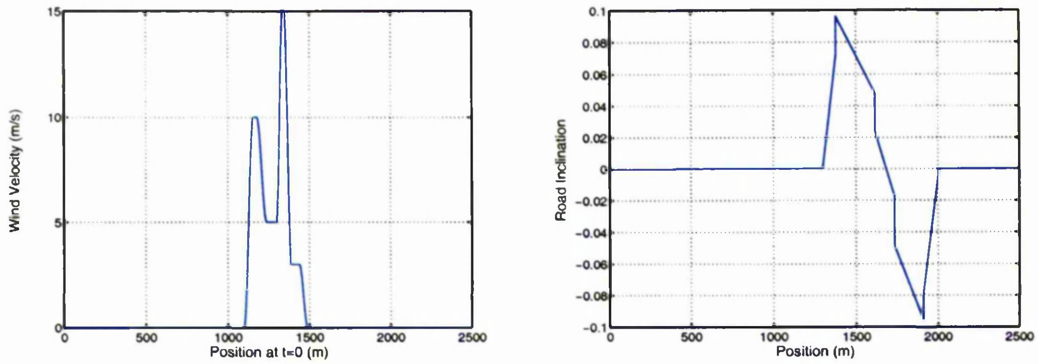
where m_i is the mass, x_i is the vehicle's position, F_{Wi} is the force given by the drive-train at the wheels, F_{ri} represents the resistive forces, F_{di} is the force exerted by the vehicle's damper and F_{di+1} is the force exerted by the following vehicle's damper.

The resistive forces considered for the simplified model include dry friction, viscous friction, aerodynamic drag and the force resulting from road grade. They are given by the expression

$$F_r = mg(C_f + C_v\dot{x})\cos\phi + C_a\left(\dot{x} - v_{wind}\right)^2 + mg\sin\phi, \tag{6.2}$$

where C_f and C_v are constants representing the dry and viscous friction coefficients respectively. C_f has the same sign as $\dot{x}$, and is zero for $\dot{x} = 0$. C_a is a constant that depends on the shape of the vehicle, the air density and, in the case of closely spaced vehicles, on the vehicle's position within the convoy (Zabat et al. 1995). The aerodynamic constant can be calculated from $C_a = 0.5\rho\, C_D\, A$; A being the frontal area, ρ the air density and C_D, the drag coefficient. The environmental wind speed is represented with v_{wind}, and the road angle is ϕ. The previously mentioned reference suggests a reduction in C_D in the region of 40% to 50% for the inner vehicles at close spacing and around 25% for the last vehicle. The value of C_D of each vehicle in the convoy model is adjusted accordingly. The Simulink model of the resistive forces is shown in figure 6.1.1. The look-up tables are used to extract the wind velocity and inclination at any given point on the track from a file containing the journey details. Their output depends on the position of each individual vehicle within the convoy.





(a) Wind gust profile at $t = 0$.

(b) Road gradient profile.

Figure 6.2: Typical journey characteristics for simulation with the Convoy Toolbox.

6.1.2 External Perturbations

A common representation of wind gusts is the 'Discrete Gust Model', which has a velocity profile of the '1-cosine' shape described in (Smith & Adelfang 1998). The main defining parameters for such representation are the gust length and the magnitude of v_{wind}. It is assumed that the gusts tend to have shorter length near the ground as a result of turbulence. A wind velocity profile was defined for the simulation using four consecutive discrete gusts of magnitudes 10 m/s, 5 m/s,15 m/s and $3m/s$, with lengths of 30 m, 30 m, 20 m and 25 m respectively. The direction of these gusts is opposite to the direction of travel of the vehicles. On a multi-vehicle scenario, each vehicle encounters the wind gust at different positions since the air moves along the road. To simulate this without deforming the profile, the gust was made to travel in opposite direction to the convoy at its average speed. In the simulation the gust starts at 1250 m from the vehicles at $t = 0$. Figures 6.2(a) and 6.2(b) show the wind gust profile at $t = 0$ and the road inclination profile respectively.

6.1.3 Drivetrain Model

An essentially first order drivetrain model with power and torque limitation and asymmetric acceleration and braking was designed. It works by calculating the power demand at any given velocity, it is then compared with the maximum power rating and limited accordingly. The available force is calculated and limited to stay within the limits of traction. The resulting force is then fed through a first order lag. The model allows for a different braking and propulsion powers, although the force remains limited by friction. The rest of the logic blocks are to avoid operation in reverse. The Simulink model is shown in figure 6.3. The

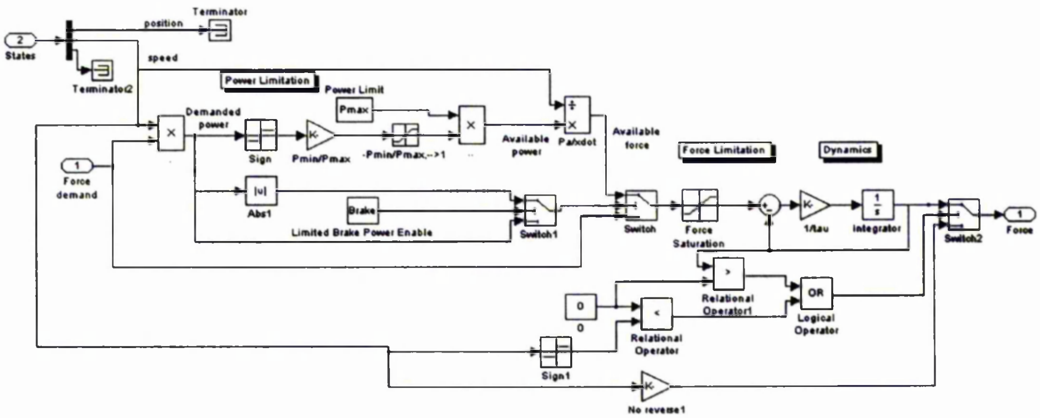
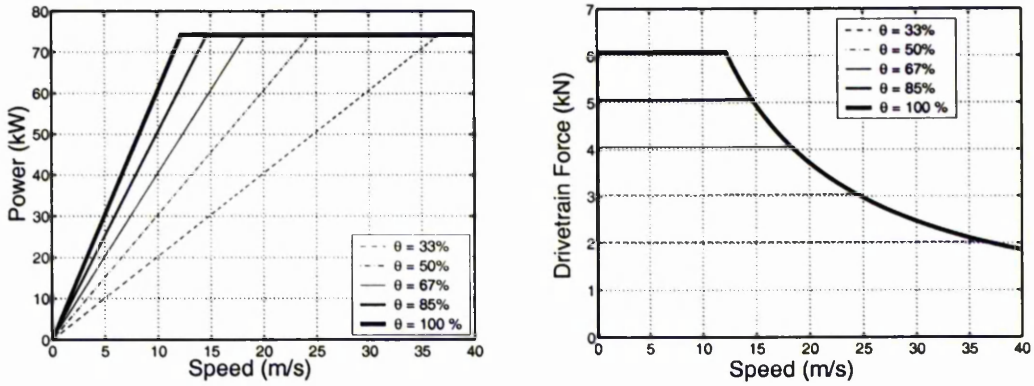


Figure 6.3: Simulink model of the HEV for use with the Convoy Toolbox

power and force characteristics are shown in figures 6.4(a) and 6.4(b) for different throttle settings θ. Braking power was set at 135 kW since mechanical brakes are available in this vehicle.

6.2 Model Tuning and Validation

The model was tuned (figure 6.5) using an acceleration test and the Urban Dynamometer Driving Cycle (CYC_UDDS) to match the power, acceleration and speed of the ADVISOR model. These tests permitted to extract the values of viscous and dry friction coefficients. The numbers that best matched the ADVISOR model were $C_f = 12.15$ and $C_v = 0.27$. More importantly, the drivetrain's time constant was found to be $\tau = 0.1$. Figures 6.5(a) and 6.5(b) show the acceleration test, which confirms that the convoy vehicle exhibits the same time constant, average acceleration and power output as the ADVISOR vehicle, though the transients due to gear changing are omitted from the simpler model. This was done principally to avoid excessive switching in the simulation of the convoy in order to reduce the calculation time. Also, because gear changes are performed by the HEV's traction control, they are done smoothly and consistently without great changes to the vehicle's velocity. Figures 6.5(c) and 6.5(d) demonstrate that the convoy and ADVISOR vehicles follow the urban driving cycle in the same manner.



(a) Power at different throttle (θ)

(b) Force at different throttle (θ)

Figure 6.4: Model Drivetrain Characteristics at different throttle angles (θ)

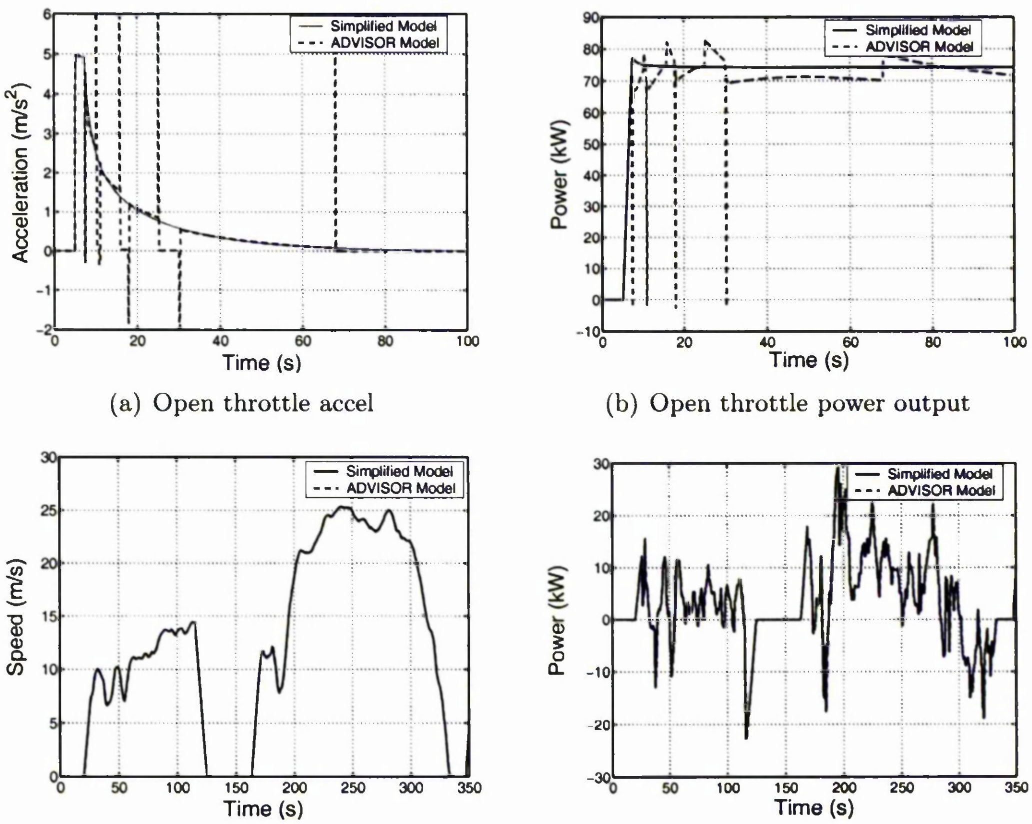
6.3 Convoy Model

6.3.1 Information model

Wireless leader information communications are modelled as a 0.1 s zero order hold. Achieving this information rate is considered challenging using current technology for large networks. For systems with bridging dampers, only the speed demand sent to the virtual leader is known. When vehicles are in contact through the damper, it is assumed that relative speed and position from the predecessor can be readily sensed from the damper. In the event of loss of contact, the model assumes that these quantities will be estimated from a short range sensor, which is modelled with the addition of band limited white noise to the sensed signal. It is worth noting that the seed for this noise was random and different for each vehicle. The radar sensor required by the vehicles without dampers was treated in the same way, but simulations showed that vehicles where more sensitive to it, which forced the noise power to be limited to 0.001 with a sample time of 0.12 s.

6.3.2 Controller design

The chosen controller for the situation was of the forward-looking type. I was assumed that the control system and basic sensors would contribute to the lag



(a) Open throttle accel

(b) Open throttle power output

(c) UDDS speed

(d) UDDS power output

Figure 6.5: Model tuning and validation to match ADVISOR for HEV.

Table 6.1: Chosen constant values

	Without damper	With damper
α	2.5	0.14
β	22	1.5
γ	6	0
ϵ	30	0.4
κ	10	5
ζ	1.5	0
τ	0.4	0.4
h	0	0
δ	0.6m	0
k'	-	8.2818 kN/m
c'	-	1.9101 $kN; s/m$
x_m	0.6m	0.6m

characteristic of the torque production by additional 0.1 s, but also it was assumed that the filtering necessary for the radar signal would contribute a further 0.2 s to the total (see section 2.2.5), producing a final time constant of $\tau = 0.4s$ for the vehicles without damper. This last reduction in performance is not necessary for the convoy system since it does not include a radar, but was kept artificially to compare vehicles with identical characteristics. Further performance improvements are possible to the convoy system if this constraint is removed and controller gains recalculated.

The controller and damper constants are given in Table 6.1. The values were calculated using the methodology described in section 3.5.3 and tuned for best performance. The vehicles without the damping equipment require considerable higher gains to maintain spacing than the vehicles with bridging dampers assisting the control system.

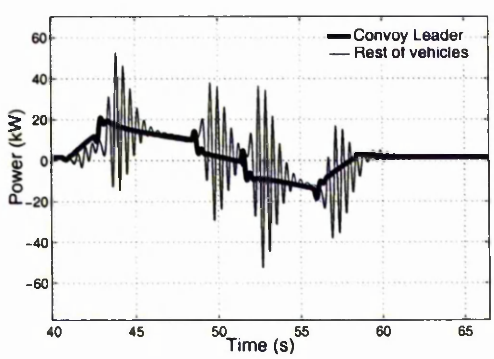
6.4 Research results

Figures 6.6(a) and 6.6(b) show power applied and acceleration respectively for the operation of convoy vehicles at close headway (60cm) without a bridging damper during a segment of a 100 second journey in which the vehicles are travelling at

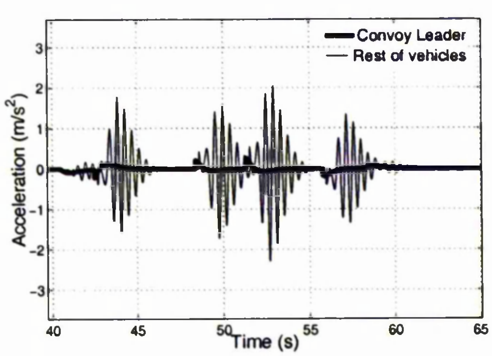
20 m/s and encounter a change in gradient ranging from 10% to -10%. High control gains are necessary to ensure adequate position regulation; also, due to string stability issues higher relative acceleration gains (γ and ζ) are required. The system is therefore highly susceptible to noise. It is observed that high power peaks occur and the resulting VDV is 2 $m/s^{1.75}$ for the eighth vehicle in the convoy during the 100 seconds of journey. Figures 6.7(a) and 6.7(b) show the results for the same grade change when the vehicles are separated by passive dampers. Lower gains may be applied and power peaks are considerably reduced. However, the leading vehicle is required to do most of the braking and the highest VDV is 0.5 $m/s^{1.75}$ for the second vehicle over the same journey.

figures 6.8(a) and 6.8(b) show the corresponding results for a quasi-active damper which is controlled in collaboration with the drive system. Lower gains are again applied and power peaks are negligible, allowing a lower rating of installed power plant. The VDV of between 0.25 $m/s^{1.75}$ and 0.4 $m/s^{1.75}$ for all the vehicles is also excellent.

The application of the wind gust (27 s to 37 s) and a change of grade (40 s to 60 s) also produce transient position errors. figures 6.9(a), 6.9(b) and 6.9(c) show the effects on vehicles without dampers, with passive dampers and with quasi-active dampers. The plots confirm that both the passive and quasi-active dampers are effective in maintaining inter-vehicle separation even with the lower controller gains, but the whole convoy's performance (represented by the leader's relative position) suffers in the case of the passive damper. The quasi-active damper on the other hand produces only a minimal deviation of the convoy leader's relative position. Figure 6.10(a) shows drive power during emergency braking

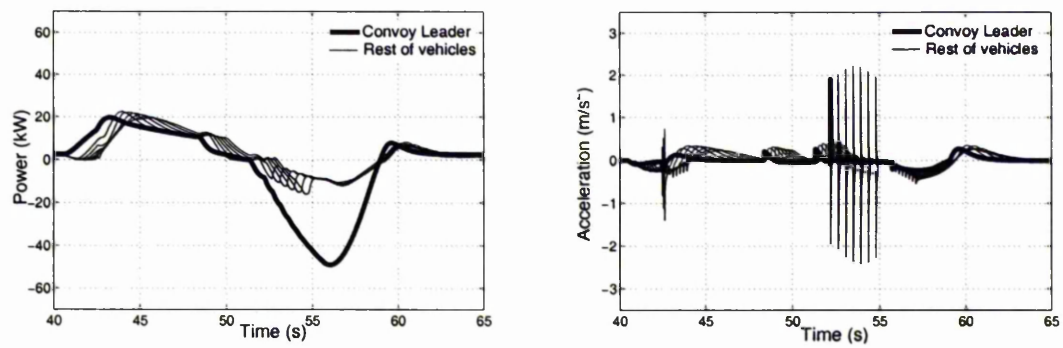


(a) Drivetrain power



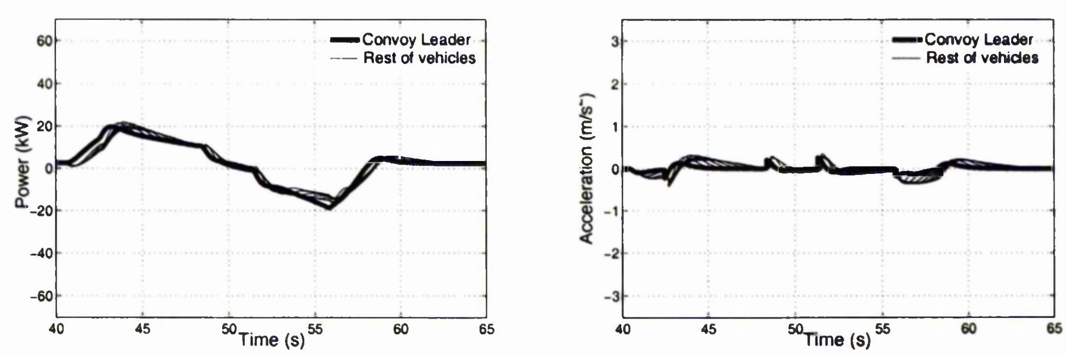
(b) Acceleration

Figure 6.6: Convoy vehicles with no damper



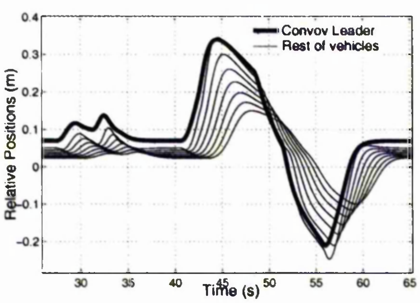
(a) Drivetrain power (b) Acceleration

Figure 6.7: Convoy vehicles with passive damper

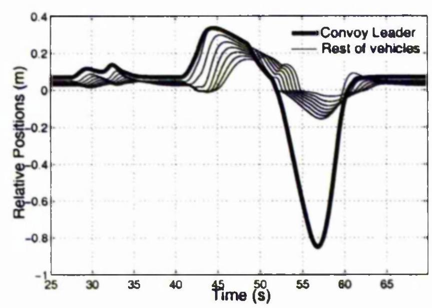


(a) Drivetrain power (b) Acceleration

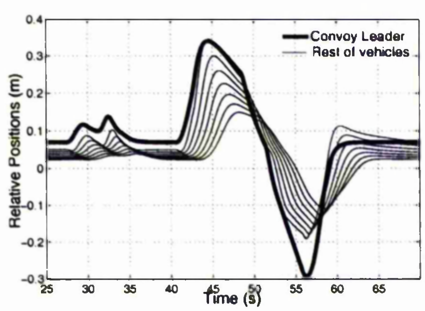
Figure 6.8: Convoy vehicles with quasi-active damper



(a) No damper



(b) Passive damper



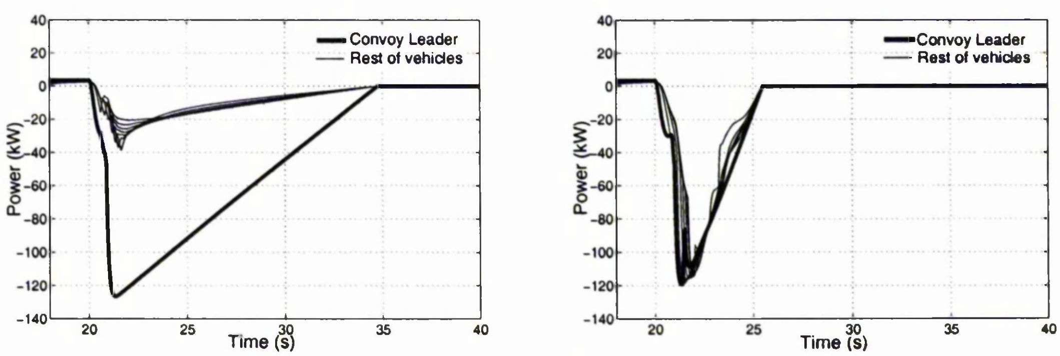
(c) Quasi-active damper

Figure 6.9: Position error due to wind and gradient

for a convoy fitted with passive dampers. It is assumed that no braking signal is available for simultaneous broadcast to all vehicles. Due to the lags/delays inherent in the system, most of the retardation power is applied by the leading vehicle and the convoy takes 15 seconds to come to rest. The control gain for position relative to the preceding vehicle is greater than that for the broadcast 'leader' signal to ensure string stability. With active dampers, the power required is shared between the vehicles (figure 6.10(b)) and the convoy is brought to rest in 6 seconds. This reinforces the benefit to safety of the active damper. In both cases, the maximum damper compression was found to be similar. For a system without dampers, destructive collisions occur.

6.5 Conclusions

The simulations described in this chapter have demonstrated that there is good correspondence between the characteristics of a typical parallel hybrid electric vehicle and the requirements of a convoy vehicle system using an inter-vehicle bridging damper. A simple simulation model has been derived, tuned and validated by comparison with an ADVISOR HEV model using the Urban Dynamometer Driving Schedule as reference. It has been shown that, for ideal conditions, vehicles can be controlled acceptably at low (around 1 m) spacing, though with compromised safety and comfort. However, with realistic communication and sensor delays, the peak power required from the vehicle drive train is excessive. In addition to reducing inter-vehicle spacing to virtually zero, the bridging damper has



(a) Power during deceleration with passive dampers

(b) Power during deceleration with quasi-active dampers

Figure 6.10: Power during emergency deceleration

been shown to produce the following results:

1. Removal of a high peak torque/power requirement through the improved management of available energy. The power plant size is thus reduced, and the operating regime of an HEV is optimised.

2. Simplified control is acceptable.

3. Communication/Sensing for normal in-lane running is limited to the adjacent vehicle.

4. Safety in the event of system failure is assured by the presence of a compressible inter-vehicle device.

Preliminary assessment indicates that an electro-hydraulic or electromagnetic device would be of sufficiently small size and mass to be acceptable in a road transport application. The future development of such a device would be an 'enabling technology' for economical, high-capacity road vehicle systems with enhanced safety.

Chapter 7

Conclusions, Evaluation and Suggestions for Further work

7.1 Conclusions

This thesis has introduced and investigated the concept of a 'bridging damper', which is interposed between contacting vehicles in an automatically controlled convoy. The research area is potentially broad, covering any form of semi-autonomous or fully automated vehicle operating on a highway or guideway, for example Automated Highway Systems or Personal Rapid Transit. The operation of vehicles which are effectively in contact implies that the highway or guideway infrastructure may be utilised at 100% of capacity. However, there are practical limits to the length of any convoy, as space must be allowed for merging vehicles into the system and for a change of control strategy during demerging of vehicles which are departing on to other routes. The analysis has also shown that instability may occur once a large number of vehicles exists in the convoy. However, the basis of control is largely linear and a more sophisticated controller, perhaps of variable structure, may extend the maximum convoy length if this should be desired. The thesis has reviewed some of the technologies and systems proposed for solution of 'the transport problem' and summarised government papers intended to tackle the problem.

In chapter 2, the control strategies of the leading 'string control' techniques are considered in detail and it is shown that a string or convoy of vehicles may follow each other at close headway under certain conditions. It is also noted that the 'brick wall' stopping criterion does not apply to realistic road transport, though railway signalling still retains this measure. Further consideration of published control strategies, some of which use variable structure control, reveals the degree of communication required, taking into account the information needed from preceding and possibly following vehicles and also sensory information. It is noted that, although radar systems for cruise control are well established in the commercial market, legal considerations limit the feature to 'driver assistance', with the driver being in overall command at all times. A number of demonstration systems intended to improve capacity, fuel economy, safety, convenience and driver stress are reviewed but it is noted that the major difficulties in achieving high capacity lie in obtaining the inter-vehicle communication density required. No system has yet been able to achieve its aims in a full-scale, realistic transport environment.

In chapter 3, an extended analysis of the vehicle-following scenario has been carried out. A largely linear model of a vehicle has been used to permit the computation of a large convoy and it may be considered that this aspect departs from a more realistic vehicle situation. However, results have been verified as far as is possible using two criteria.

1. The response of the model of an automated electric vehicle has been compared with that from a very detailed simulation, incorporating all the elements, of the electric drive, carried out by a previous worker and tuning has been executed to match the responses.

2. The response of the parallel hybrid-electric vehicle model has been tuned to match that of a corresponding vehicle from the established ADVISOR program widely used in the automotive industry, taking the vehicle through the Urban Dynamometer Driving Cycle and comparing results until these matched. However, gear changing has not been included and therefore these transients are omitted from the results.

The models used were also similar to those employed by previous workers in the AHS Program in the United States.

Chapter 3 also analyses the vehicular interactions within a convoy in which each vehicle receives only information on the status of the preceding vehicle and the convoy leader and a matrix representation of the set of interactions is developed. However, the main point of interest in stability studies is the propagation of spacing errors within the convoy. Broadly, if the error is reduced as it passes along the convoy, then the convoy will be stable. Considerations to accommodate vehicles of varying performance are proposed but remain to be thoroughly verified through further work. The conditions for positive impulse response, attenuation of error and stable, non-oscillatory response are identified in terms of both pole placement and volume space in which stable operating conditions may be located.

An important area which was identified early in the research is the comfort of vehicle occupants in an automated transport system. This is not a new area of research, as empirical consideration was given by means of extensive tests during design of the American PCC streetcar in the 1930s. Some theoretical studies have since been made and chapter 4 summarises the development of these and introduces the effects of motion on the human body. The movement of a vehicle may take place in three dimensions, with yaw, roll and pitch added, and criteria, including international standards, are reviewed. However, for the control of operation of a convoy, longitudinal acceleration and vibration are of prime importance. The chapter reviews weighting factors for these quantities and concludes that the Vibration Dose Value is the most suitable of current measures for assessment of bridging damper performance. On the basis of published research, VDV limits are proposed for performance of a convoy and are applied in later chapters. Although the limits are to some extent subjective, they are based on what research currently exists and are intended as a measure to show the difference between control strategies.

The characteristics of convoy systems are described in chapter 5. The main characteristic that identifies convoy systems is the use of a, inter-vehicle 'bridging damper' to complement the control system by bringing the vehicles to be in physical contact. This allows a simpler control strategy which results in a more comfortable ride and increased safety and reliability. The design considerations are proposed for the quasi-active device introduced. Four principal operating modes are identified and a non-linear control law tailored to meet the requirements of each of them. A fifth mode is identified as a total system failure in which severe collisions happen. Passenger protection is afforded thanks to a mechanical fuse in the form of a perforated conic tube over which the damper is mounted. The simulation models of the proposed device and control law are briefly described.

In chapter 6 an HEV model is employed in assessing the effect of an inter-vehicle device. The resistive forces considered in the vehicle model include dry friction, viscous friction, aerodynamic drag and the force resulting from road grade. Environmental wind speed is included and is based on the published 'Discrete Gust Model', which has a velocity profile of the '1-cosine' shape, the main defining parameters being the gust length and magnitude. In studies of the response of the vehicles, disturbances in the form of wind gusts and a rising and then falling gradient were applied. The simulations, which were carried out using SIMULINK, use control laws in which each vehicle knows the position and speed of the preceding vehicle and the position, speed and acceleration of the leader. Inter-vehicle communications are modelled as a time delay of 0.1s. On the basis of previously published research on AHS, these delays would be considered challenging even with current technology for large networks. Simulations have been carried out for three forms of convoy.

1. A convoy operating at small spacing with no inter-vehicle bridging damper device. All the control signals mentioned previously are available. This scenario corresponds to a conventional 'platoon' as employed by AHS researchers.

2. A convoy operating in contact with a simple passive spring and damper fitted between vehicles. In this case, only the speed demand sent to the leader is known, as information on the preceding vehicle may be obtained from the damper itself. In the event of temporary loss of contact, short-range radar may be used. The resumption of contact and normal information would typically correspond to the measurement error transient mentioned above.

3. A convoy operating in contact with a 'quasi-active' (controllable) bridging damper fitted between vehicles. The damper cannot 'pull', as there is no coupling between vehicles and hence certain degrees of freedom are lost. Again, only the speed demand sent to the leader is known, the remaining information on the preceding vehicle being obtained from the damper itself.

The results for the convoy without an inter-vehicle bridging damper show that high control gains are necessary to achieve stable convoy operation and hence a realistic system would be particularly susceptible to noise from measurement equipment or external sources. Although the VDV is considered tolerable, the response is notable for the high peaks required from the vehicle power plant in order for each vehicle to maintain station without collision. The peak power implies that the installed power plant must be required to deliver considerably more power than the average value required. This implies that a vehicle in a conventional AHS platoon or its corresponding vehicle on a guideway system will have a larger, more expensive and heavier power plant than necessary. In the case of an internal combustion engine and, to a lesser extent an electric drive, efficiency will be substantially below that achieved at maximum output. It must be noted, however, that this work assumes that energy storage, such as in a parallel HEV, is available to meet transient demands; a purely internal combustion vehicle is not considered. A further result from the convoy without dampers is that an emergency deceleration outside the normal convoy operation specifications may result in collisions.

Where a passive inter-vehicle damper is applied, there is considerable reduction in peak power requirements, although acceleration peaks during wind gusts would be expected to reduce occupant comfort. However, over the 100-second test, the VDV is still less than for the system without dampers. The reduction in installed power now optimises the HEV operating regime. No account was taken in the simulation of the vehicle mass reduction implied by the reduced power requirement, i.e. the same power plant was employed throughout. A further problem which was identified is the inequality in vehicle braking effort. When an emergency braking demand is applied, the convoy leader has to provide most of the braking effort. This is considered to potentially compromise safety, although the existence of the damper will absorb destructive energy during a collision.

For the convoy fitted with quasi-active bridging dampers, the power peaks and accelerations are further reduced and the VDV is considered very acceptable. It is also noted that braking effort during emergency deceleration is effectively shared among the vehicles.

The principal conclusion of the simulations, which include realistic communication and sensor delays, are that a vehicle convoy, operating in contact but separated by a bridging damper, can be controlled in a stable manner with considerably improvements in comparison with previous platooning research. The peak power required from the vehicle drive train may be considerably reduced through improved management and exchange of the available energy. The power plant size and vehicle mass are thus reduced and fuel efficiency is expected to improve. The reduction of inter-vehicle spacing to virtually zero maximises infrastructure utilisation and may, for example, allow autonomous-vehicle systems to approach the capacity of mass transit. As outlined in earlier chapters, the control strategy is considerably simplified through the need for information on only the virtual leader target speed and data from the preceding vehicle. If the latter is obtained directly from bridging damper movement, only a single broadcast signal may, in the extreme, be needed. This greatly reduces the communication and control requirements of previous platooning research. The existence of a compressible device will inevitably improve safety in the event of system failure. It is therefore considered that the bridging damper concept may be classed as an automated system and not simply as a driver aid. The computation of the VDV regulates the quality of ride and will prevent the control system from applying vibrations which would introduce motion sickness.

7.2 Evaluation

The research contained in this thesis has made significant contributions to the design and control of 'bridging dampers'. The improvements made to the original 'bridging damper' concept mentioned in Perrott & Renfrew (1998) are considered not as the basis of a marketable product but as an 'enabling technology' which can release the potential of other research. For example, the work may inform the UK Intelligent Highways' initiative. The thesis has presented results which demonstrate that limitations inhibiting full-scale automated highways may be resolved. However, the results have been obtained purely by simulation and no practical validation has taken place. The strengths of the work are considered to include the efforts taken to validate the simplified vehicle models, minimise control gains to alleviate noise problems and accommodate measurement errors. However, the parameters of the damper are preliminary, measurement noise has not been studied and a number of 'idealised' routes have been taken. For example, all vehicles have been assumed identical and the damper response has been considered instantaneous.

The work is believed to open up opportunities for novel transport automation systems with enhanced safety and greatly increased capacity. The removal of the severe constraints imposed by massive inter-vehicle communications provides potential for full-scale system operation. However, there are many obstacles which would need to be overcome. The social desire for individual road vehicles with widely varying performance and fronts uncluttered by bridging devices is important. Current vehicle occupant psychology would also not favour a contact convoy and considerable re-education in addition to legal and social barriers would need to be overcome. Furthermore, the application of intelligent bridging dampers to vehicles and the introduction of even a simple beacon road transmitter would require considerable investment. With these considerations, it seems more likely that a demonstration of the bridging damper concept would first appear on a light rail system of some description, with technology transfer to other modes at a later stage. The use on light rail would be expected to apply to an autonomous-vehicle system on a reserved, grade-separated guideway, as applied in theme parks, airports etc. rather than conventional street-running tramways. The current government view does not consider a PRT-type system as viable for normal urban use. While most autonomous-vehicle systems such as airport shuttles do not require high capacity, instances may arise, for example in stadia or theme parks, where additional vehicles may be added to a 'contact train' at peak periods.

7.3 Suggestions for further work

Since the research so far has been undertaken using SIMULINK, the most pressing need appears to be for a hardware demonstration and evaluation of the bridging damper. The device itself resembles an active suspension damper and one way forward would be to equip such a device with control equipment, measure its parameters and response and check the effects on the simulation. A form of dynamometer rig may also be set up to represent the interaction between two vehicles with the damper interposed. For this work, a D-Space system would be very helpful, as the relevant control and dynamic elements of the simulation could be downloaded to interface with the hardware equipment. Also, as previously noted the Bose Corporation has evaluated a prototype electromagnetic suspension and the design of a similar device as a bridging damper promises fast dynamic response.

While the convoy system reduces energy usage, mainly due to lower front-end air resistance, energy savings also occur due to improved and smoother traction control and a higher point on the efficiency profile. As the bridging damper and the vertical suspension systems both may include energy storage, the benefits of linking these through a common reservoir should be evaluated. Any reduction in energy storage and retrieval or additional energy recovery will again result in energy savings to be added to the above.

From the simulation viewpoint, vehicles of varying type and loading should be tested to establish the variation in damper performance required and the maximum allowable range of the parameters involved. In this aspect, partnerships with vehicle manufacturers or other bodies able to provide data should be sought. One area not so far considered is the additional mass of the damper and control gear. It is thought that this will be much less than the variation in vehicle weight due to occupancy but the topic should nevertheless be checked.

In addition to means of continuing the present line of research, 'spin off' benefits from the bridging damper research may be considered. Firstly, the analysis based on VDV may be extended to inform the design of a novel active suspension. Secondly, an active device for a train of railway vehicles which are not all independently powered would remove the 'snatch' apparent even with automatic couplers.

Within the simulation itself, there is considerable scope for detailed work, some of which may be undertaken via smaller individual projects. Firstly, the parameters should be reviewed and checked prior to testing of alternative controllers, e.g. sliding mode. More advanced alternatives such as neural-based systems may also be attempted to enhance the limit on convoy length due to stability considerations. The potential of the system to deal with failure modes may also be evaluated by arranging a vehicle power failure, for example, where the following vehicles in the convoy require to push it. A safe strategy should also be devised where the failed vehicle is at the rear and drops off the back of the convoy; the vehicle would then have to be 'picked up' by the following convoy. A related situation occurs where a vehicle or group of vehicles demerge from a convoy. The information from the vehicle in front disappears and a different control regime must be established. This is similar to the loss of contact mentioned earlier; relying purely on radar will degrade the safety-critical nature of the system. Where vehicles merge, distance and other variables will have to be acquired by the incoming vehicle and a following convoy may experience a reduced headway. Various management systems for merging have been reported, but obtaining a 'safe' rather than optimal technique represents more of a challenge.

It was noted earlier that public acceptance will be a barrier to a 'bridging damper AHS'. It is suggested that a study be carried out in partnership with academic psychologists working in the transport and driver attitude field, preferably with good links to industry and governmental organisations. For the system to find any form of acceptance in future, it is important to have support from as many relevant organisations as possible. Recent attempts to generate support from the motor industry have not been successful. At the time of writing this thesis, negotiations with the Department for Transport and the Highways Agency are in progress.

One of the most important aspects of development of the bridging damper concept is the techno-economic analysis of the potential for the device in the 'real' market place, either in guided or road applications. Although this facet is lacking in this thesis, it would be necessary to establish both the potential energy savings suggested above and a hardware demonstration prior to consulting industry for costing data, as volume and small-order production costs might vary considerably. It would also be important to have identified the intended market, again based on the application proposals in previous paragraphs.

In conclusion, the author hopes that his work will lead to considerable developments in future and wishes every success to his succeeding researchers in their endeavours.

Appendix A

Convoy Toolbox

The Convoy Tool box was designed to produce consistent simulations of convoys so different vehicle models, control techniques, information strategies or damper models can be analysed. The program builds the vehicle models from a large number of user selectable subsystems contained in a library. All the properties of each subsystem can be edited by the user to accommodate the largest possible freedom of design. Figure A.1 shows the vehicle building interface. Once a vehicle is built, it can be positioned in any position within a convoy (figure A.2), so it is possible to simulate vehicles with different characteristics in a single convoy. The number of vehicles is also selectable.

Figure A.3 shows the convoy used in chapter 3. The vehicles shown have a linear first order lag drivetrain with time constant $\tau = 0.5$s. The control system is forward-looking only and no communication delays or lag are considered. Also, no friction or aerodynamic drag are considered in this example. The damper also is used here either. The input to the system is the desired velocity profile, desired individual separation profile and the journey conditions such as wind speed and road gradient.

The simulation (figure A.4) window can be used to manage and compare the data generated by multiple simulations with different conditions and settings. The vehicles can be edited and the changes are tracked in the output file. The outputs can be plots or an animation representing the convoy in motion. This helps the designer to gather spacial awareness about the resulting motion.

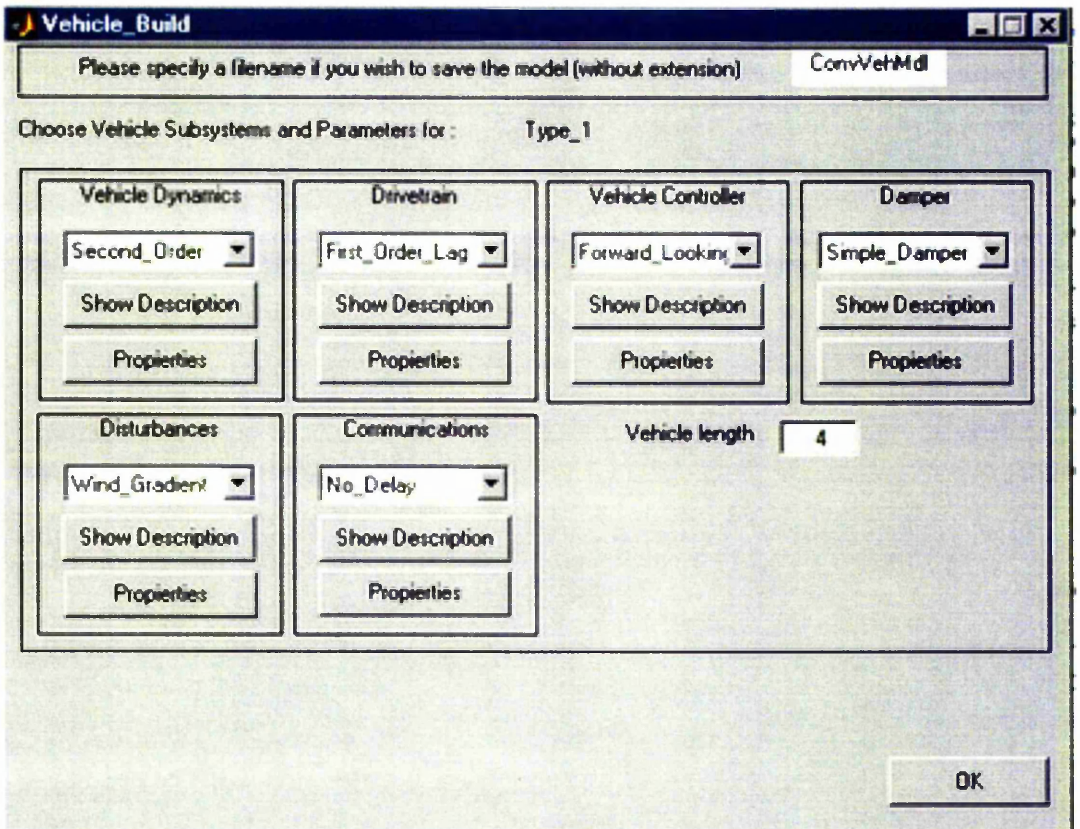


Figure A.1: Convoy Toolbox vehicle build tool

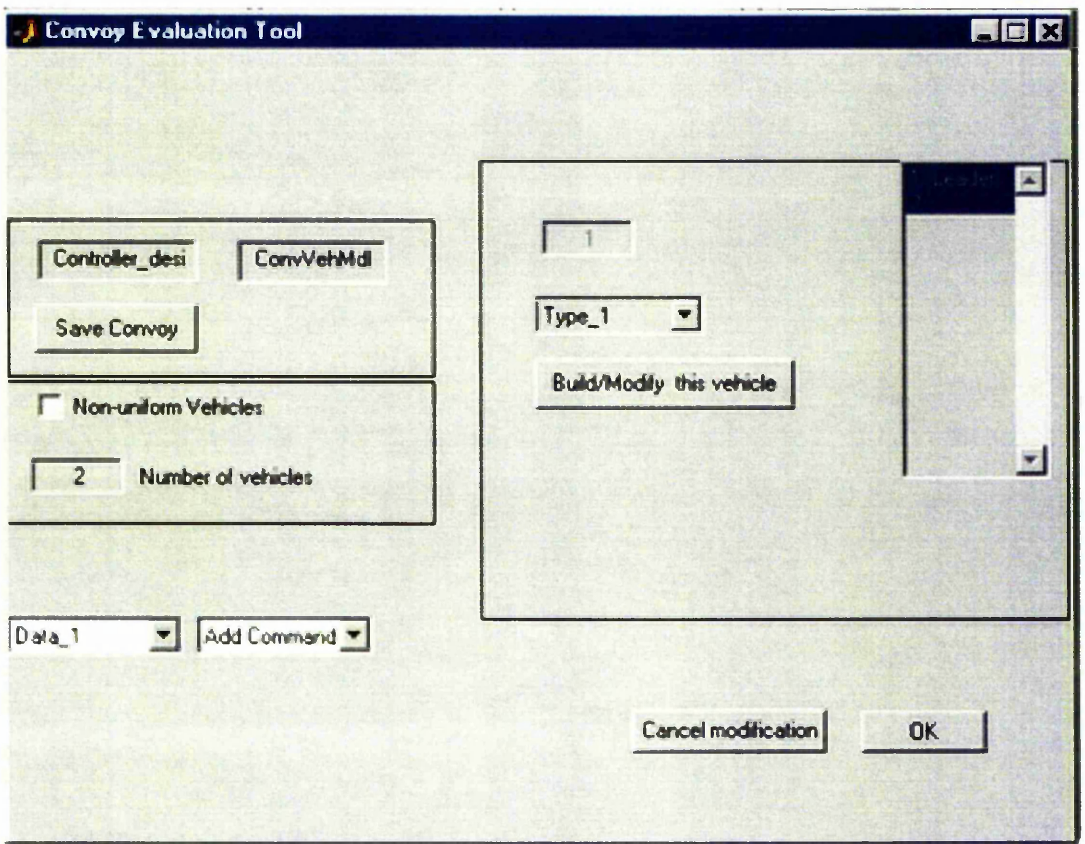


Figure A.2: Convoy Toolbox convoy build tool

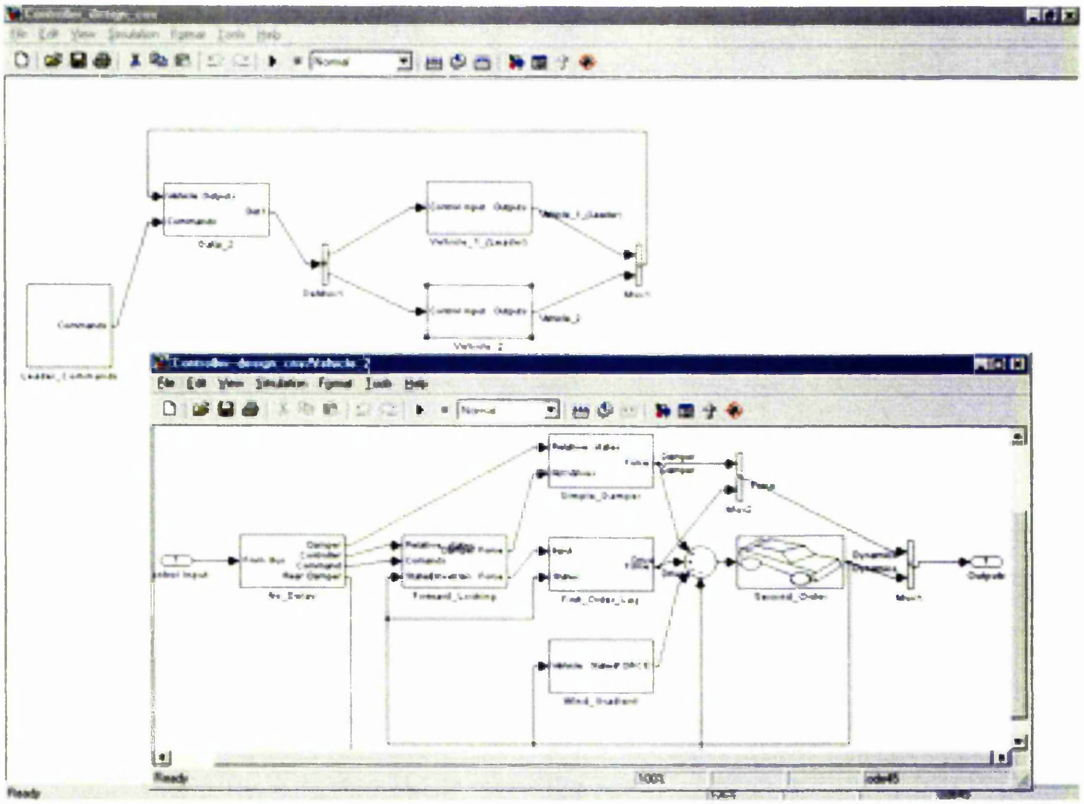


Figure A.3: Convoy Toolbox model

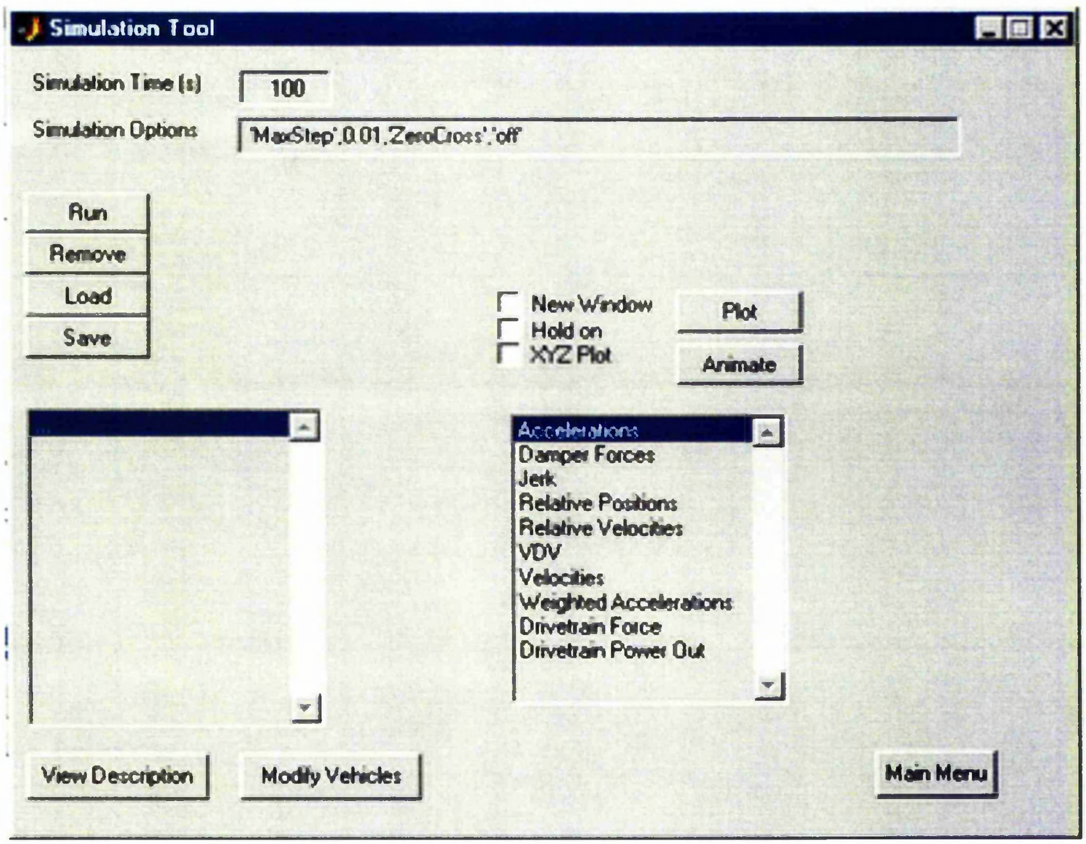


Figure A.4: Convoy Toolbox simulation tool

Appendix B

Frequency Weighting Curves From BS 6841:1987

Weightings W_b, W_c, W_d and W_e are used over the frequency range 0.5 Hz to 80 Hz; they have high- and low-pass band-limiting filters at 0.4 Hz and 100 Hz respectively. Weighting W_f is to be used over the range 0.1 Hz to 0.5 Hz and its filters are set at 0.08 Hz and 0.63 Hz W_g may be used over the range 1 Hz to 80 Hz and its filters are set at 0.8 Hz and 100 Hz.

The band-limiting filters are given by:

$$H_b(s) = \frac{4\pi^2\ {f_2}^2 \cdot s^2}{\left(s^2 + \frac{2\pi f_1}{Q_1} \cdot s + 4\pi^2 {f_1}^2\right)\left(s^2 + \frac{2\pi f_2}{Q_1} \cdot s + 4\pi^2 {f_2}^2\right)}. \tag{B.1}$$

The frequency weightings W_c, W_d, W_e, and W_g are given by:

$$H_w(s) = \frac{(s + 2\pi f_3)}{\left(s^2 + \frac{2\pi f_4}{Q_2} \cdot s + 4\pi^2 {f_4}^2\right)} \cdot \frac{2\pi K\ {f_4}^2}{f_3}. \tag{B.2}$$

and they may be realised by two pole filters. The weightings W_b and W_f are given by:

$$H_w(s) = \frac{(s + 2\pi f_3)\left(s^2 + \frac{2\pi f_4}{Q_2} \cdot s + 4\pi^2 {f_5}^2\right)}{\left(s^2 + \frac{2\pi f_4}{Q_2} \cdot s + 4\pi^2 {f_4}^2\right)\left(s^2 + \frac{2\pi f_6}{Q_4} \cdot s + 4\pi^2 {f_6}^2\right)} \cdot \frac{2\pi K\ {f_4}^2\ {f_6}^2}{f_3\ {f_5}^2}, \tag{B.3}$$

where $H(s)$ is the transfer function of the filter (s is the Laplace operator), values of f_n ($n = 1$ to 6) designate resonance frequencies, Q_n ($n = 1$ to 4) designate selectivity and K is a constant gain. Their values can be found in table B.1. Figure B.1 shows the moduli of the frequency weighting curves.

Weighting	Band-limiting			Frequency weighting							
	f_1	f_2	Q_1	f_3	f_4	f_5	f_6	Q_2	Q_3	Q_4	K
	Hz	Hz		Hz	Hz	Hz	Hz				
W_b	0.4	100	0.71	16	16	2.5	4	0.55	0.90	0.95	0.4
W_c	0.4	100	0.71	8	8	–	–	0.63	–	–	1.0
W_d	0.4	100	0.71	2	2	–	–	0.63	–	–	1.0
W_e	0.4	100	0.71	1.0	1.0	–	–	0.63	–	–	1.0
W_f	0.08	0.63	0.71	∞	0.250	0.0625	0.10	0.86	0.8	0.8	0.4
W_g	0.8	100	0.71	1.5	5.3	–	–	0.68	–	–	0.42

Table B.1: Characteristics of band-limiting and band-pass filters for frequency weightings

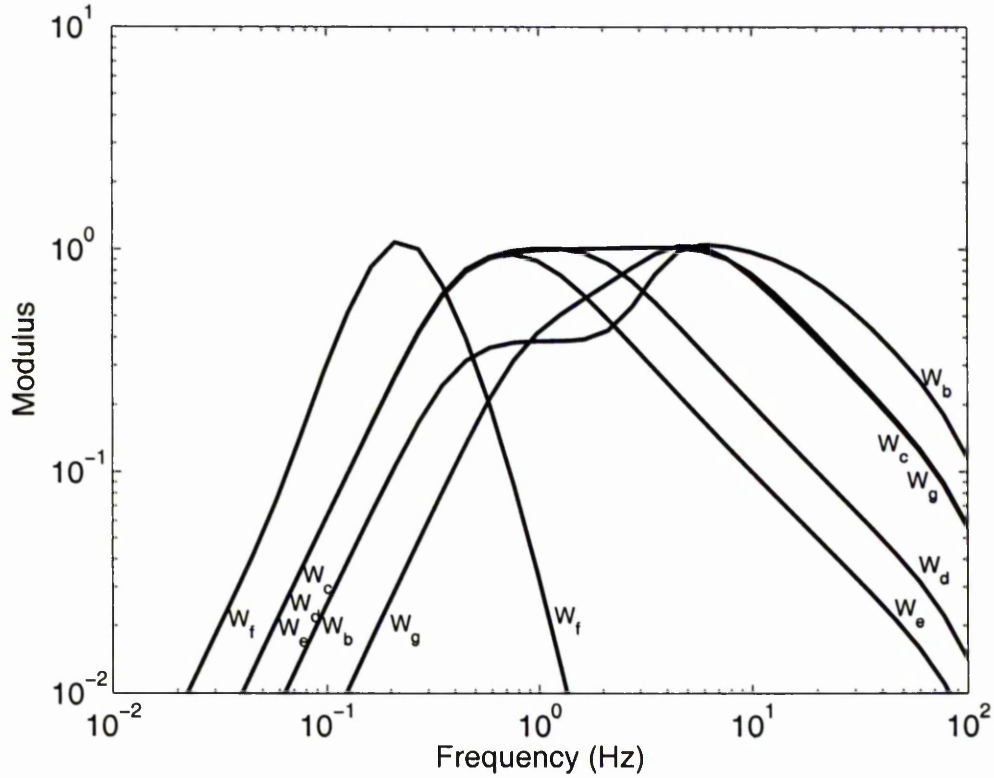


Figure B.1: Moduli of the frequency weightings with band-limiting filters from BS 6841: 1987

The frequency weightings applied to the vibration of the seat regarding passenger comfort are W_b for the z direction and W_d in the x and y directions. Additionally, translational vibration occurring at the floor and at the backrest and rotation occurring at the base of the seat may be assessed using the weightings shown in table B.2. For standing persons the use of the same weightings employed for translational seat vibration is recommended.

Motion direction	Weighting applicable	Multiplying factor
Rotation x axis on the seat	We	0.63
Rotation y axis on the seat	We	0.4
Rotation z axis on the seat	We	0.2
x axis on the backrest	Wc	0.8
y axis on the backrest	Wd	0.5
z axis on the backrest	Wd	0.4
x axis at the feet	Wb	0.25
y axis at the feet	Wb	0.25
z axis at the feet	Wb	0.4

Table B.2: Weightings applicable to additional directions of motion for seating persons

Appendix C

Calculation of Collision Velocities in Close Headway Operation

Impacts.m

```
nv=20; %number of vehicles
domax=20; %original max distance
step=.5;
af=-5;
ae=-4;
RT=.5; %reaction time
l=4;% Vehicle length
%calculate impact at different spacings
Capconcat=[];
for j=0:domax/step
    do=j*step;
    voo=[0:3:30]; %Original velocity vector
    vo=voo; %Followers' velocity at time t
    vol=vo; %Leader's velocity at time t
    capacity(j+1,:)=nv*vo./((nv-1)*do+vo.^2/...
    (-2*ae)-vo.^2/(-2*af)+RT*vo+nv*l)*60;

    %Calculate impact of all vehicles
for n=1:nv-1
    sim('impacts')
    rd=Pv1-Pv2+do; %relative distance
```

```
    rd(find(rd<=0))=NaN;%remove data after impact
    [dd,i]=min(rd);%find collision point
    Colvel(n,:)=diag(Vv1(i,1:size(i,2))- ...
    Vv2(i,1:size(i,2)))';%Collision velocity
    vo=diag(Vv2(i,1:size(i,2)))';%New initial
    %velocity for the nth impact
    vol=diag((n*Vv1(i,1:size(i,2))+...
    Vv2(i,1:size(i,2)))/(n+1))';%new velocity of
    %colided vehicles
    vol(find(vol<0))=0; %remove data after
    %vehicles stop
    vo(find(vo<0))=0;
end
Colmean(j+1,:)=mean(Colvel,1);
Colsqmean(j+1,:)=mean(Colvel.^2,1);
Costdvar(j+1,:)=std(Colvel,1);
Capconcat=[Capconcat,capacity(j+1,:)];
end
[Capsort,Index]=sort(Capconcat);
rows=ceil(Index/size(capacity,2));
cols=(Index-(rows-1)*size(capacity,2));
dist=[0:step:domax];
for ii=1:size(cols,2)
    Colsort(1,ii)=Colmean(rows(ii),cols(ii));
    Colstdsort(1,ii)=Costdvar(rows(ii),cols(ii));
    vels(1,ii)=voo(cols(ii));
    dists(1,ii)=dist(rows(ii));
end

%produce figures
figure
subplot('position',[.1 .69 .8 .27]);
plot(dist,-Colmean);
grid on
subplot('position',[.1 .395 .8 .27]);
```

```
plot(dist,Costdvar);
grid on
subplot('position',[.1 .1 .8 .27]);
plot(dist,capacity);
grid on

nv=20; %number of vehicles
domax=20; %original max distance
step=.5;
af=-5;
ae=-4;
RT=.5; %reaction time
l=4;% Vehicle length
%calculate impact at different spacings
Capconcat=[];
for j=0:domax/step
    do=j*step;
    voo=[0:3:30]; %Original velocity vector
    vo=voo; %Followers' velocity at time t
    vol=vo; %Leader's velocity at time t
    capacity(j+1,:)=nv*vo./((nv-1)*do+...
    vo.^2/(-2*ae)-vo.^2/(-2*af)+RT*vo+...
    nv*l)*60;
    %Calculate impact of all vehicles
for n=1:nv-1
    sim('impacts')
    rd=Pv1-Pv2+do; %relative distance
    rd(find(rd<=0))=NaN;%remove data after
     %impact
    [dd,i]=min(rd);%find collision point
    Colvel(n,:)=diag(Vv1(i,1:size(i,2))-...
     Vv2(i,1:size(i,2)))';%Collision
     %velocity
    vo=diag(Vv2(i,1:size(i,2)))';%New
    %initial velocity for the nth impact
```

```
    vol=diag((n*Vv1(i,1:size(i,2))+...
    Vv2(i,1:size(i,2)))/(n+1))';%new
    % velocity of colided vehicles
    vol(find(vol<0))=0; %remove data
    % after vehicles stop
    vo(find(vo<0))=0;
end
Colmean(j+1,:)=mean(Colvel,1);
Colsqmean(j+1,:)=mean(Colvel.^2,1);
Costdvar(j+1,:)=std(Colvel,1);
Capconcat=[Capconcat,capacity(j+1,:)];
end
[Capsort,Index]=sort(Capconcat);
rows=ceil(Index/size(capacity,2));
cols=(Index-(rows-1)*size(capacity,2));
dist=[0:step:domax];
for ii=1:size(cols,2)
    Colsort(1,ii)=Colmean(rows(ii),...
    cols(ii));
    Colstdsort(1,ii)=Costdvar(rows(ii),...
    cols(ii));
    vels(1,ii)=voo(cols(ii));
    dists(1,ii)=dist(rows(ii));
end

%produce figures
figure
subplot('position',[.1 .69 .8 .27]);
plot(dist,-Colmean);
grid on
subplot('position',[.1 .395 .8 .27]);
plot(dist,Costdvar);
grid on
subplot('position',[.1 .1 .8 .27]);
plot(dist,capacity);
```

```
grid on
```

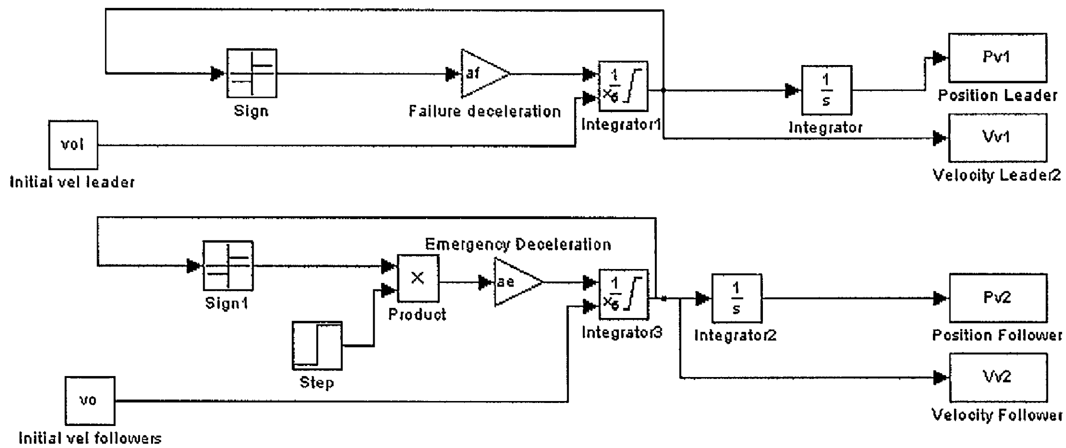


Figure C.1: Simulink model 'impacts.mdl'

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