Integrated Automotive Control
Robust Design and Automated Tuning of Automotive Controllers

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Ph.D. Thesis - wp1
Literature research to the state-of-the-art of integrated automotive control in the automotive industry, focusing on Integrated Clutch Control (ICC) and Adaptive Cruise Control (ACC).

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Nomenclature

\( a \) acceleration / deceleration (m s\(^{-2}\))
\( \dot{a} \) jerk (m s\(^{-3}\))
\( b \) damping (Ns m\(^{-1}\))
\( d_h \) (relative) distance headway (m)
\( E_d \) dissipated energy (J)
\( F_{cl} \) clutch actuator force (N)
\( F_y \) lateral forces (N)
\( i_d \) final drive or differential ratio (\(-\))
\( i_g \) gearbox or transmission ratio (\(-\))
\( J \) moment of inertia (kg m\(^2\))
\( J \) generally, a control strategy’s objective function
\( k \) stiffness (N m\(^{-1}\))
\( t \) time (s)
\( t^* \) the time total clutch lock-up is reached (s)
\( T_{cl} \) torque transmitted by the clutch system (Nm)
\( T_e \) engine output torque (Nm)
\( t_h \) time headway (s)
\( u \) actuator input control signal
\( v \) longitudinal velocity (m s\(^{-1}\))
\( x \) generally, a system’s state
\( x_{cl} \) clutch closing path (m)

\( \dot{\omega} \) rotational acceleration / deceleration (rad s\(^{-2}\))
\( \ddot{\omega} \) rotational jerk (rad s\(^{-3}\))

Super- and subscripts

\( \wedge \) filtered signal
\( 0 \) initial state
\( b \) brake system
\( cl \) clutch system, often related to the passive clutch plate
\( d \) desired
\( e \) engine
\( g \) gearbox
\( gs \) gearshift
\( h \) host vehicle
\( l \) load
\( ref \) reference
\( s \) shafts (propeller and drive shafts)
\( set \) user-defined setpoint
\( t \) target vehicle
\( th \) throttle
\( v \) vehicle

Greek letters

\( \alpha_{th} \) throttle actuator input
\( \Delta_v \) relative speed (m s\(^{-1}\))
\( \varepsilon \) error
\( \theta \) torsion or rotation angle (rad)
\( \theta_b \) backlash size (rad)
\( \mu \) friction coefficient (\(-\))
\( \omega \) rotational velocity (rad s\(^{-1}\))
\( \omega_{e,i} \) engine idle speed (rad s\(^{-1}\))
\( \omega_{sl} = \omega_e - \omega_{sl} \) slip speed between the clutch plates (rad s\(^{-1}\))
Definitions, acronyms and abbreviations

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<th>Definition</th>
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<tr>
<td>(C)ACC</td>
<td>Cooperative Adaptive Cruise Control</td>
</tr>
<tr>
<td>(C)FCA</td>
<td>Cooperative Forward Collision Avoidance</td>
</tr>
<tr>
<td>(C)FCW</td>
<td>Cooperative Forward Collision Warning</td>
</tr>
<tr>
<td>ABS</td>
<td>Anti-lock Braking System - “Anti Blokkeer System”</td>
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<tr>
<td>AHS</td>
<td>Automated Highway System</td>
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<td>AMT</td>
<td>Automated Manual Transmission</td>
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<td>ASL</td>
<td>Application Speed Limiter</td>
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<td>CAN</td>
<td>Controller Area Network</td>
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<td>CAS</td>
<td>Collision Avoidance System</td>
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<td>CC</td>
<td>Cruise Control</td>
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<td>CSC</td>
<td>Curve Speed Control</td>
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<td>DP</td>
<td>Dynamic Programming</td>
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<td>DSC</td>
<td>Downhill Speed Control</td>
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<td>ECU</td>
<td>Electronic Control Unit</td>
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<tr>
<td>EDC</td>
<td>Electronic Diesel Control</td>
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<td>EMS</td>
<td>Energy Management System</td>
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<tr>
<td>ESP</td>
<td>Electronic Stability Program</td>
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<td>FOC</td>
<td>Follower Control</td>
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<tr>
<td>FTS</td>
<td>First Tier Supplier</td>
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<tr>
<td>GPS</td>
<td>Global Positioning System</td>
</tr>
<tr>
<td>HDV</td>
<td>Heavy Duty Vehicle (“Lastkraftwaagen”)</td>
</tr>
<tr>
<td>I/O</td>
<td>system Input / Output</td>
</tr>
<tr>
<td>IACC</td>
<td>Intelligent Adaptive Cruise Control</td>
</tr>
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<td>ICC</td>
<td>Integrated Clutch Control</td>
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<tr>
<td>ISA</td>
<td>Intelligent Speed Adapting system</td>
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<td>IVHS</td>
<td>Intelligent Vehicle Highway System</td>
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<td>LOC</td>
<td>Longitudinal Control</td>
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<tr>
<td>LQ(R)</td>
<td>Linear Quadratic (Regulator)</td>
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<td>LQG</td>
<td>Linear Quadratic Gaussian optimal control</td>
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<tr>
<td>MIMO</td>
<td>Multiple Input, Multiple Output</td>
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<td>MPC</td>
<td>Model Predictive Control</td>
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<tr>
<td>OEM</td>
<td>Original Equipment Manufacturer</td>
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<td>PID</td>
<td>Proportional, Integrating, Differentiating</td>
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<tr>
<td>QFT</td>
<td>Quantitative Feedback Theory</td>
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<tr>
<td>RQV</td>
<td>“Regler Quer Verstellung” (Speed control of diesel engines)</td>
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<tr>
<td>S&amp;G</td>
<td>Stop-&amp;-Go</td>
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<tr>
<td>SISO</td>
<td>Single Input, Single Output</td>
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<tr>
<td>SSC</td>
<td>Set Speed Control</td>
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<td>TCS</td>
<td>Traction Control System</td>
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<td>VSE</td>
<td>Vehicle Stability Enhancement system</td>
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Preface

Within the context of the project 'Robust design and automated tuning of automotive controllers', a literature research to the state-of-the-art of integrated automotive control in the automotive industry is performed. The project aims to develop methods and algorithms for robust low-complexity controller design and algorithms and software tools for tuning of controller parameters with minimal effort, e.g. automated tuning, focussing on control systems in the field of commercial vehicle. Results are validated at DAF Trucks and TNO via case studies in the area of the driveline controllers, such as the clutch control (automated gearbox) and (adaptive) cruise control.

This report presents the results of the literature research, focussing on Integrated Clutch Control (ICC) and Adaptive Cruise Control (ACC). This corresponds to workpackage 1 (wp1) as defined in the Project Proposal, see Appendix [B].
Chapter 1

Introduction

1.1 Today’s automotive control

Starting in the 1980s, North-American and European automotive industries copied the supply relationship management construction of the Japanese industry (see Figure 1.1). The Original Equipment (or vehicle) Manufacturers (OEMs) selected fewer so-called First Tier Suppliers (FTSs) for their components. The suppliers became responsible for larger component subsystems of high technological level, designed in co-development with the OEMs.

The make-and-deliver-to-order trend of the 1990s made the OEMs focus on core competencies. The increasing component complexity in combination with the strong time-to-market requirements led to delegation of even the development of main subsystems to specialized FTSs (Lung 2003, Richter & Ernst 2006).

Figure 1.1: Schematic representation of the upward supply chain for OEMs and the corresponding suppliers of entire (sub)systems, so-called tier suppliers. In (a) the supply chain in general is shown. In (b) the supply chain corresponding to the automotive industry in the Netherlands is shown (Wismans 2007).

Recent developments in software and electronics have led to new functionality potential. Using software and electronics, new functionality can be implemented more quickly and easily than mechanically. Today’s market demands for new features are thus often resolved with electronic or software solutions. The main contributors to the exponential growth in electronic components,
software and controllers are driveability and performance requirements and legislation on safety, environment and fuel consumption. Analogously, controller complexity has increased significantly. Due to the outsourcing of the development of the main subsystems to specialized suppliers, today’s responsibility for software development and subsystem knowledge commonly resides with the suppliers (Ward & Fields 2000, Heinecke et al. 2004, Richter & Ernst 2006).

Part of the potential for new functionality and control optimization encompasses the integration of various subsystems. Integration of engine and transmission controls, improved emission controls, further integration of the Energy Management System (EMS) functionality and integration of electronic throttle and brake control are examples of today’s research areas. As the character of the vehicle is determined more and more by software, OEMs again want to be more involved in the definition of control software (Kelling & Heck 2002, Coelingh et al. 2002).

However, various component’s control systems have become complex black box systems for the OEMs. Complexity of components and the corresponding electronic architectures often prohibit further improvements and exploration of possibilities for integration of subsystems. Much research to standardization of component I/O is done by FTSs (Lung 2003, Balluchi et al. 2005). The desired involvement in control software development of OEMs nevertheless entails increased tuning effort. Hence, application of intelligent or automatic controller design and tuning techniques to decrease tuning times is demanded (Magnor et al. 2005). The demand for cost and time reduction in product development in combination with the increased subsystem complexity, makes integration of various components a major challenge for today’s OEMs (Heinecke et al. 2004).

1.2 Integrated automotive control

Consider a vehicle as a system that can be divided into several vehicle subsystems. A vehicle subsystem, e.g., the engine, comprises a physical device or component including all electronics and software, which contributes to various vehicle functions. A vehicle function then is defined as the separately identifiable aspects of a vehicle’s behaviour, e.g. longitudinal acceleration (Schilke et al. 1988).

The control system of a subsystem can be represented by a three-layer structure (see Figure 1.2). The high level or supervisory control part comprises firstly the identification of the situation via sensor diagnostics. Secondly it incorporates the mode selection leading to control objectives for the upper level control. The upper level (also called middle level) or subsystem control part incorporates the actual intelligent control. The controller should be adaptive to the present situation in order to perform optimal in a complete envelope of working conditions. This control part is commonly based on modeling of the vehicle and determines setpoints and trajectories for the lower level control part. The lower level or component control part comprises component specific controllers (Wills et al. 2001).

Integrated control in the sense of this study specifically aims at coordination and optimization of upper level controllers to improve performance and enable the mapping of new functionality onto various subsystems. Coordination comprises integrated controller design as well as suitable attenuation of conflicting control actions, which originate from different subsystems.

1.3 Research objective and report outline

This research studies the benefits and possibilities of an integrated approach in controller design from an OEMs perspective. As first example case studies, the integration of engine and transmission controls in an Integrated Clutch Controller (ICC) and the integration of electronic throttle and brake controls in an Adaptive Cruise Control (ACC) system are considered. Focus lies on robust low-complexity controller design. This forms a basis for a subsequent study to techniques and methods for automated tuning of automotive controllers. See the Project Proposal in Appendix B for a complete overview of the project.

The project is executed in association with DAF Trucks N.V., Eindhoven, the Netherlands and
1.3. RESEARCH OBJECTIVE AND REPORT OUTLINE

Vehicle, driver, environment, sensors, etc.

High level, supervisory control

Identification and arbitration
situation analysis and event detection
arbitration and mode selection

Intelligent control
stable, working condition dependent
subsystem specific

Setpoints, trajectories
component specific
situation specific

Upper or middle level control

Lower level component control

Figure 1.2: Schematic representation of an integrated automotive control system. The Vehicle, driver, environment, sensors, etc. block is shown in more detail in Figure 1.3.

Figure 1.3: General representation of the vehicle, environment and driver interaction with each other and an example automotive control system. The vehicle conditions encompass amongst others the current vehicle speed and acceleration, the vehicle mass, etc. Immediate control settings encompass for example the throttle and brake pedal and the steering wheel. Finally, strategic control settings encompass for example on / off switching of cruise control functionality and switching between various adaptive cruise control modes, e.g. sportive, normal or comfortable.

TNO Automotive, Helmond, the Netherlands. As a major truck manufacturer, DAF contributes with extensive OEM knowledge. Hence, a truck is considered as a research example throughout this study. TNO Automotive is a large research institute dedicated to the automotive industry. TNO contributes with state-of-the-art and in-depth knowledge about specific today’s automotive functionality.

An overview of today’s main automotive control systems, the corresponding electrical embedding and a general view of the potential of integrated control in the automotive industry is given.
in Chapter 2. In Chapter 3, the state-of-the-art of ICC respectively ACC are discussed.
Chapter 2

Automotive control

The electronic architecture of a modern vehicle consists of a complex network of ECUs, sensors and actuators. The electronics and software interact with the physical environment, the system mechanics, through sensors and actuators. This is called a mechatronic system. The integration of software, electronics, sensors, actuators and mechanics is the key aspect of a mechatronic system.

In this chapter the main mechatronic systems in a vehicle, which will be referred to as automotive control systems, are presented (Section 2.1). Furthermore, the network architectures connecting these systems and the current trends in research to automotive control are discussed in Sections 2.2 and 2.3.

2.1 Automotive control systems

The amount of electrical and software components has taken an enormous growth during the past decennia. At first, mainly additional luxury was incorporated. Today, in an increasing degree electric and software components are used in combination with sensors and actuators to integrate different mechanical components. Performance improvements and additional functionality are thus obtained.

The enormous growth in electrical and software components has enabled an enormous increase in functionality and performance. The other way around however, research to new functionality and increased performance has stimulated this growth as well. Analysts estimate that today more than 80 percent of all automotive innovation stems from electronics (Leen & Heffernan 2002). In a modern vehicle, several automotive control systems can be distinguished. The main systems in a modern truck are listed below.

- Engine
- Transmission
- Brake system
- Retarder
- Exhaust after treatment
- Suspension system
- Active steering system

Besides these main systems, several smaller control systems, so-called body systems are present. Examples are dashboard instruments, front and rear lights, electrical windows, seats, doors, AC system, etc. Generally, body systems are less crucial regarding safety and overall functionality of the vehicle and hence not regarded as separate control systems. The control systems are connected
via databusses and point-to-point wiring and both read and send measurements or estimates from
observers from and to other systems.

In the past decades, the term x-by-wire is adopted by the automotive industry to indicate
electrically controlled automotive systems. By-wire actually refers to mechatronic systems in
which the communication and control is enabled electronically, thus steering actuators that control
the actual mechanical system. X-by-wire is a general term that covers any application with by-
wire functionality where the ‘x’ is a referral to the actual application. In e.g. a brake-by-wire
control system, the information transfer from the brake pedal to the braking actuator is handled
electronically. Hydraulic, pneumatic or electric-hydraulic actuators perform the actual actuation.

In the late 1970s, brake-by-wire and throttle-by-wire systems, i.e. Anti-lock Braking System
(ABS), and Electronic Diesel Control (EDC), were introduced. During the past decades these
systems have been extended, improved and elaborated from many perspectives and nowadays
they are standard applications in modern vehicles. In the 1990s, the first commercially available
automated manual transmissions became available and since 2000 the steer-by-wire systems have
become the current challenge for the automotive industry (Larses 2005).

Today, almost all automotive control systems comprise by-wire functionality, i.e. the transmis-
sion control system comprises clutch-by-wire and gearshift-by-wire functionality, engine control
throttle-by-wire functionality, the retarder control system comprises brake-by-wire functionality
and the active steering system steer-by-wire functionality. In fact, today’s main automotive control
systems originate from by-wire functionality requirements.

Besides the advantages the control of by-wire systems has (equal to advantages of automotive
control systems: firstly the potential for implementation of advanced and exact control algorithms
and integrated control via communication between systems. Secondly, some mechanical com-
ponents become superfluous, thus simplifying the mechanical design and increasing flexibility of
a system.), disadvantages are the diminished reliability with respect to mechanically controlled
systems, which introduces safety issues. This is one reason why steer-by-wire is not introduces
yet.

2.2 In-vehicle networks and electronic architectures

Numerous electronic network architectures and communication standards to connect a vehicle’s
electronic equipment have been developed during the past decades. Several architectures and
standards have been developed specifically for automotive applications, while others are adopted
by the automotive industry. In this section an overview of the communication standards used in
the automotive industry and the main aspects of automotive network architectures are discussed.

2.2.1 Communication standards

Communication between subsystems is enabled by point-to-point wiring and communication data
busses. In the past, point-to-point wiring was the standard means of connecting one element
to another. Due to the exponential increase in electronics however, the use of an increasing
amount of discrete wiring came to an end. Added wiring increased vehicle weight, weakened
performance and decreased reliability, limiting expanding vehicle functionality. Beginning in the
1980s, centralized and distributed networks replaced point-to-point wiring. The advantages of data
busses are 1) cost, weight and space savings, 2) increased reliability as the number of connections
is decreased, 3) a centralized fault diagnosis, enabling detection of transmission errors, 4) enabling
easy implementation of new (integrated) control functionality, which requires communication with
different subsystems. Hence, data busses really enable implementation of integrated controllers
(Leen & Heffernan 2002).

Within typical domains of a vehicle, e.g. the powertrain, the chassis, the body, etc., the ECUs
are interconnected via specific communication busses. In this way electronic or control networks are
formed. These networks or busses are connected via so-called gateway nodes, allowing to exchange
messages across a system’s borders. Commonly, a vehicle incorporates high-speed Controller Area
2.2. IN-VEHICLE NETWORKS AND ELECTRONIC ARCHITECTURES

Networks (CANs) for critical high-performance functions such as powertrain control and low-speed CANs for serving devices like door locks and tail light clusters. A second distinction can be made between these CANs and specific dashboard and media networks.

An overview of the most widespread and currently available network protocols is given in Table A of Appendix A. A distinction is made between 1) network protocols that are developed specifically for certain automotive applications and that are widely adopted by the automotive industry nowadays, 2) upcoming protocols that are developed specifically for certain automotive applications, 3) protocols from other markets that are upcoming in the automotive industry and 4) some further ‘smaller’ protocols that are less widespread in the automotive industry (e.g. Nolte et al. 2005, Lind et al. 1999).

The CAN is one of the first, most widespread and most enduring automotive control networks. It was developed in the mid-1980s by Bosch. During the last decades the CAN bus in combination with the CAN protocol has proven to be a reliable and robust mechanism for in-vehicle communication. The main specifications of CAN are the limited bandwidth and the non-deterministic communication behaviour.

A vehicle typically contains two or three distinctive CANs operating at different transmission rates. A low-speed CAN running at a baud rate less than 125 kbps usually manages a vehicle’s so-called comfort electronics like seat and window movement controls and other interfaces. Control applications that are not real-time critical use these low-speed networks. Furthermore, a high-speed CAN runs more real-time-critical functions such as the engine management, ABS, and cruise control. Due to electromagnetic radiation and the required corresponding shielding, a baud rate of about 500 kbps is the maximum speed (Leen & Heffernan 2002).

Today’s expanding by-wire systems, utilizing electronics rather than mechanical or hydraulic means, require more specialized and highly reliable control networks. Furthermore, the increase in new functionality requiring integration of subsystems as well as the complexity of the resulting networks (expressed in terms of number of ECUs and the number of messages being exchanged), show the limitations of the currently used CAN protocol. Hence, recent research focuses on dependable, high-data rate network protocols for specific advanced-control applications, e.g. Byteflight (2000), LIN (2000), Flexray (2002), TTCAN (2001), etc. See Table A of Appendix A for a more detailed list of protocols and specifications. Availability of data, reliability of data and data bandwidth are the main aspects of research to new communication protocols (Stroop & Stolpe 2006, Leen & Heffernan 2002).

Furthermore, research to entertainment and multimedia systems for automotive applications has taken a huge flight in the past decades. Many network protocols specifically aiming at these applications are developed, i.e. D2B, Bluetooth, Mobile Media Link (Nolte et al. 2005, Lind et al. 1999), MOST (1998), etc. See Table A of Appendix A for a more detailed list of protocols and specifications.

2.2.2 Electronic architectures

Electronic architectures comprise two main aspects. Firstly the control functionality, i.e. the control algorithms, and secondly the control hardware, representing the physical components enabling control of a system, i.e. the sensors, actuators, ECUs, wiring, data busses, etc. Whereas the control hardware is physically restricted to a certain system, the corresponding software incorporating the control functionality may be located elsewhere (Gander et al. 2000, Persson 2004).

Three main architectures can be distinguished. The so-called stand-alone architecture comprises an electronic architecture in which each mechatronic system has its own control unit. The engine for example incorporates the the Engine Management System (EMS) and the gearbox incorporates the Transmission Control Unit (TCU). The architecture physically integrating the control units into the corresponding systems is the so-called mechatronic module control architecture. It resembles the functional distribution of the stand-alone architecture. However, system specific functionality is integrated in a mechatronic module, thus establishing a physical link between the control unit and the corresponding system. This is the most widespread architecture at the moment. Finally, the full system control architecture focuses on centralized control of
all mechatronic systems. All functionality is integrated in one control unit and no physical inte-

gration between this control unit and the corresponding systems is present. Much research is

performed highlighting the advantages and disadvantages of the different architectures. See for

example Gander et al. (2000), Giovanardi et al. (2005).

Until recently, a stand-alone architecture comprising subsystem specific functionality in com-

bination with separate ECUs was common. Today, mechatronic module control architectures

combining mechatronic subsystems with a network topology are used more and more. These sub-

systems are nevertheless still developed separately and independently mostly, leading to integration

problems. To further enable and exploit the potential of integrated control, a transition to full

system control architectures or mechatronic module control architectures with jointly developed

subsystems will be demanded in future (Larses 2005).

2.3 Areas of interest

In the past decennia, the four main issues in research to automotive control systems have been

emissions, safety, comfort and traction. Major achievements in the area of active vehicle safety

control include ABS braking and traction control systems. Much research is done to active con-

trol of roll-over dynamics, vehicle stability controllers, integration of endurance brakes besides

the already present friction brakes and driver assistance systems. Driver assistance systems for

example comprise automated or adaptive cruise control, vehicle tracking, ultimately targeting at

platooning of trucks, active steering and the nowadays often already standard automatic transmis-

sion systems. Considering heavy duty trucks much research is done to intelligent highway systems

(IHWS), thus enabling increased safety and fuel economy as well as reduced congestions, etc.

Decreasing emissions and fuel consumption remains one of the main research topics. Concern-

ing control, much research is done in the field of exhaust after treatment systems, near zero

exhaust emissions and fuel economy by optimisation of diesel engines. Research to hybrids and

the application of CVT’s for medium duty trucks also shows promising research results.

In recent years, improvement of the so-called driveability or comfort and performance require-

ments have become increasingly important. Besides issues as the indoor environment and thermal

comfort in a vehicle, the driveability and performance requirements comprise for example dy-

namic behaviour of the vehicle, detection and suppression of critical dynamic vehicle states, active

suspension systems and driver assistance systems.

In recent years, the potential of cooperating devices, or integrated control, is explored success-

fully and many improvements in functionality and performance are thus attained. For example in-

tegration of the Automated Manual Transmission (AMT) and the engine system enables improved

control of longitudinal comfort when driving away. The AMT controlling the engine rotational

speed when driving away may thus overcome undesirable longitudinal oscillations in the driveline.

Hence improving longitudinal driving comfort. The integrated character of the control systems

mentioned, gives rise to research often comprising several areas of interest. For example an active

suspension system not only increases comfort, but also enables increased safety when it is used to

prevent roll-over. As a result, modern control systems although often developed separately still,

will have to be developed jointly in future.
Chapter 3

Integrated clutch control (ICC)

3.1 Introduction

Traditionally, OEM’s primary focus is fuel economy, while complying to legislation on emissions. Today’s increased application of electronics and software components in the automotive industry, enables active control of certain driveability characteristics. While for example increasingly stringent demands on emissions have to be followed by all OEMs, driveability characteristics are used more and more by OEMs to distinguish themselves. Driveability characteristics encompass requirements concerning acceleration and vehicle handling as well as minimization of undesired driveline oscillations.

The problem of excitation of undesired driveline dynamics is mostly attributed to the clutch system, specifically the clutch engagement process. Moreover, clutch control and engine control encompass the main control functionality to overcome undesired driveline oscillations (Szadkowski et al. 1995). Focussing on today’s trucks, the clutch system is part of the Automatic Manual Transmission (AMT). Accordingly, AMT control has become of major interest in improving the driveability of a vehicle and active control of driveability characteristics has become an important aspect in recent years (Persson 2004). For example Mercedes claimed driver comfort to be their main focus besides the traditional fuel economy of their vehicles, when introducing their new transmission (Kern 2006). Table 3.1 provides an overview of the aspects AMT control comprises and the main research topics correlated to it.

<table>
<thead>
<tr>
<th></th>
<th>gear selection, shifting</th>
<th>clutch control</th>
<th>engine control</th>
<th>vehicle dynamics</th>
</tr>
</thead>
<tbody>
<tr>
<td>fuel consumption and impact on the environment</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>prohibition of driveline oscillations</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>fast acceleration and gear shifting</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
</tbody>
</table>

Table 3.1: The four main aspects of AMT control and the three main research topics related to these aspects. This research targets at prohibition of driveline oscillations. Assuming given vehicle dynamics, this implies focussing on clutch control and engine control.

With respect to torque-converter systems, which incorporate much more damping, in particular powertrain systems incorporating a dry plate clutch suffer problems with undesired driveline oscillations. In combination with vehicle dynamics as backlash, shaft flexibilities, clutch facing
material characteristics and a non-smooth clutch engagement process due to e.g. bad or no synchronization between the clutch plates, this forms a basis for undesired driveline oscillations. These oscillations are triggered by transient effects in drive-away dynamics, gear shifting and tip-in or tip-out manoeuvres, i.e. brusque accelerator pedal pressing or releasing. Generally, attenuation of the first resonant mode produces longitudinal oscillations of the vehicle in a region of 2 to 8 Hz. Especially these oscillations are experienced as unpleasant by the driver, as the natural frequency of the human upper body part lies in the same frequency range. Hence, the handling of such oscillations and the corresponding resonances form the basis for driver comfort. Many specific control challenges are present, i.e. vehicle drive-away feeling, undamped behaviour during shifts and tip-in / tip-out manoeuvres (Sun & Hebbale 2005).

Commonly, the engine, the AMT and the other driveline subsystems including their corresponding control systems are developed separately. However, integrating control of e.g. the engine and the clutch system have shown to be effective in diminishing the amount of induced driveline oscillations (Fredriksson & Egardt 2000). As a result, the focus of powertrain control has shifted in recent years from control of the individual powertrain components to control of the complete powertrain, so-called integrated powertrain control. In the underlying study, integrated powertrain control is referred to as a control system considering the engine in combination with the driveline as one system. Hence, integrated powertrain control is not aiming at centralizing all controllers, but increases and exploits cooperation and communication between several control systems with separate functionality. This enables improvements in the field of driveability characteristics when combined with modern control technology. System integration thus has become the key for successful transmission development in recent years (Sun & Hebbale 2005).

This study focusses on optimization of the clutch engagement process to minimize excitation of driveline oscillations by exploration of the possibilities of integrated clutch control. Integrated clutch control is an example of integrated powertrain control, specifically aiming at control of the clutch engagement process by integration of the clutch control system and engine control system. Targeting at actual implementation on market ready vehicles, an OEM as well as a tier 1 supplier will be involved in the project.

### 3.1.1 The clutch engagement process

The clutch system considered is a dry clutch system. Generally, the automated dry clutch system consists of two plates that can be moved together by an actuator that exerts a force $F_{cl}$ on the so-called passive or pressure clutch plate or disk, see Figure 3.1. The plate is connected to the rest of the vehicle driveline via the clutch output shaft. The other, so-called active or friction plate is connected to the crank shaft of the engine. As the clutch engages, the plates are pushed together by the actuator. When the plates touch, a torque $T_{cl}$ is transmitted from the engine to the rest of the driveline and the speed difference between the active and passive clutch plates $\omega_{sl} = \omega_e - \omega_{cl}$ diminishes. If the speeds of the two plates are equal, $\omega_{sl} = 0$, clutch lock-up is reached and the engine is connected directly to the rest of the driveline. In Figure 3.2 an example plot of the engine speed $\omega_e$ and the clutch passive disk speed $\omega_{cl}$ during the clutch engagement.
The minimum engine speed is bounded by the idle speed $\omega_{e,i}$. At time $t = t^*$ actual clutch lock-up is reached.

Figure 3.2: The figure shows an example synchronization between the engine output speed $\omega_e$ and the rest of the driveline via the clutch passive disk speed $\omega_{cl}$ during the clutch engagement process when starting from standstill. $\omega_{e,i}$ represents the starting idle speed of the engine. Total clutch lock-up is reached at $t = t^*$.

3.1.2 Outline

To start with, an introduction to the problem of undesired longitudinal vibrations and oscillations is given in Section 3.2. Next, the concept of integrated clutch control incorporating the state-of-the-art in AMT control and a general formulation of the control objectives is discussed (Section 3.3). An overview of solely engine control to damp or minimize excitation of driveline oscillations is given in Section 3.3.2. This is followed by an overview of the research performed in the field of clutch control in Section 3.4, specifically focussing on various control strategies. Finally, some conclusions concerning integrated clutch control are given, which will be used as a start for future research (Section 3.5). In general, specific attention is paid to heavy duty vehicles and related problems as a truck is used as a basis for an example case study.
CHAPTER 3. INTEGRATED CLUTCH CONTROL (ICC)

3.2 The powertrain and undesired driveline dynamics

This study focusses on the powertrain of a heavy duty vehicle incorporating an Automatic Manual Transmission (AMT). An AMT typically incorporates a gearbox in combination with a dry plate or lock-up clutch system, both operated by either electro-mechanic, electro-hydraulic or electro-pneumatic actuators. The driveline in combination with the engine will be referred to as the powertrain. The main parts of a powertrain are the engine, the clutch system, the gearbox, the propeller shaft, the differential, the driving shafts and the wheels, see Figure 3.3.

Figure 3.3: Schematic representation of the powertrain of a truck with a dry clutch and an AMT.

As mentioned in the introduction, the powertrain incorporates dynamics, which cause undesired longitudinal oscillations. These undesired phenomena are referred to as shunt, shuffle and clutch judder and will be discussed in more detail in Section 3.2.1. The undesired vehicle dynamics forming the basis for these undesired phenomena generally can be divided into two parts. On the one hand, passive vehicle mechanics as backlash, shaft flexibilities and bad clutch facing material characteristics can be defined (see Section 3.2.2). On the other hand, a non-smooth and badly or non synchronized clutch engagement process causes undesired driveline oscillations. Sections 3.3 to 3.4 focus on minimizing the effect of these undesired phenomena exploring the possibilities of active, integrated control of the clutch engagement process.

3.2.1 Undesired driveline oscillations

Longitudinal motion is dominant in characterizing driveability aspects regarding driver comfort. Important factors characterizing the vibrations that cause effect on the human body are intensity, frequency, direction and exposure time (Schnetzler et al. 2000). In this way several phenomena can be distinguished in examining the undesired longitudinal oscillations and vibrations in an automotive powertrain.

Shunt and shuffle

An impact at the driveline, which is experienced as a sharp jerk by the driver, is called shunt. Shunt often is accompanied by a frequency, metallic noise, which is known as clonk or clunk and which originates from backlash causing components to thrash against each other. The longitudinal vibration and oscillation of the entire vehicle is referred to as shuffle. Shuffle is e.g. a result of oscillation of the flywheel against the wheels, thus oscillating the entire vehicle. So-called vehicle lurch represents a measure of longitudinal acceleration. In particular, attenuation of the first resonant mode produces unpleasant (2 – 8 Hz) longitudinal oscillations, as shoulders, stomach and upper trunk have their resonance frequencies in this interval (e.g. Lagerberg & Egardt 2002, Lagerberg & Egardt 2003b, Sun & Hebbale 2005).

The torsional flexibility between different mechanical parts, the mass moment of inertia of these parts, backlash and the low mechanical damping enable phenomena like shunt and shuffle (see Section 3.2.2). Researchers deal with shunt and shuffle from many perspectives. Besides mechanical solutions to improve damping of powertrain oscillations, the trend in recent years is to apply active control in damping and minimizing excitation of undesired dynamics.
Szadkowski et al. (1995) focus their research on the properties of dry clutch dampers in a heavy duty truck. They conclude that resonance problems in powertrains can not be solved satisfactorily by mechanical solutions alone. Furthermore, they conclude that excitation of undesired driveline dynamics is mostly attributed to the clutch system, which supports the trend to use active clutch control in minimizing excitation of these undesired longitudinal vehicle dynamics.

### Clutch judder

Friction-induced vibrations in the powertrain originating from slipping in the clutch, are referred to as clutch (engagement) judder (in German: *rupfen*). In fact, judder is a friction-induced vibration between masses with sliding contact. Concerning a dry friction clutch, the masses are formed by the masses of the engine and the rest of the driveline, \( J_e \) and \( J_d \) respectively, lumped to the active and passive plates of the clutch, (see Figure 3.1). The vibrations introduce undesired dynamic loads in the system, increase slip and wear effects on the friction surfaces and corrupt smoothness of the engagement.

The most important source of judder are clutch facing induced vibrations. This occurs when the facing materials have a negative rate of change of the friction coefficient with respect to the sliding velocity between the two clutch plates and is also called negative damping (see Section 3.2.2). Other potential sources of judder are (1) misalignments in the powertrain or poor pedal actuation that may induce fluctuating pressure between the sliding surfaces, (2) some thermo-elastic phenomena on the contact surfaces and (3) external torsional excitations from e.g. the engine or the gearbox, like gear backlash or tyre slip, in the presence of certain resonant conditions.

Qualitative simulation results by e.g. Centea et al. (1999) show that a positive gradient of the clutch friction coefficient is a priority for system stability and damping of torsional oscillations. Yet a negative gradient is not a necessary condition for judder to occur, as in practice, judder has been observed even with a positive gradient, depending on the manner in which the clutch is engaged by the driver. Considering the friction coefficient as a given parameter value, this again indicates the potential for improvement of active clutch control.

### 3.2.2 Sources of undesired driveline oscillations

In an automotive powertrain, several effects give rise to the undesired phenomena discussed in Section 3.2.1. These undesired effects are triggered by transient situations such as drive-away dynamics, gear shifting and tip-in or tip-out manoeuvres, i.e. brusque accelerator pedal pressing or releasing. The most significant effects can be divided into nonlinear effects, i.e. backlash and clutch facing dynamics, and (approximately) linear effects, i.e. shaft flexibilities and tyre longitudinal slip characteristics.

#### Shaft flexibilities

The drive shafts incorporate the main flexibilities of the powertrain. Pettersson & Nielsen (2003) shows that third order linear modeling of the powertrain, including the drive shaft flexibility captures the first main resonance of the powertrain. Hence, this modeling is sufficiently detailed for controller design, targeting at damping and minimization of driveline oscillations. Consequently, the effect of these flexibilities can be taken into account rather straightforward in the design of an appropriate controller (see Section 3.4).

#### Backlash

Backlash represents the play or lash in the powertrain. The gearbox generally is the main contributor to the total amount of backlash. Backlash is the main nonlinearity in the powertrain contributing to undesired driveline oscillations (Salle et al. 1999). In Figure 3.4 the backlash is represented schematically by an amount of play \( \theta_b \) in between the lumped mass of the load \( J_I \) and the gearbox. In some cases, the effect of winding and unwinding of the shafts due to their flexibility, is approached as a backlash problem as well (Lagerberg & Egardt 2002). Control algorithms specifically aiming at control of backlash are applied to cope with this winding and unwinding.
Particularly heavy duty trucks incorporating large engine torques and high load torques, give rise to relatively much winding of the shafts.

\[ J_{e,cl}, \omega_e, k_s, TL, \omega_v, \theta_b, J_l \]

Figure 3.4: Schematic representation of backlash of size \( \theta_b \) in a powertrain. \( k_s \) represents the torsional stiffness of the shafts, \( b_g \) the damping of the gearbox bearings.

Much work is done in approaching control systems with backlash. However, only few automotive applications were reported until Lagerberg & Egardt started their research to the subject several years ago. The main strategies in control of backlash can be divided into three types.

**Linear control**: control strategies that are robust enough to handle the backlash nonlinearity. A model of the uncertainty is implemented as a disturbance model and robust control design methods e.g. \( \mathcal{H}_\infty \) or QFT theory is used to design a robust and often rather conservative controller.

**Passive nonlinear or piecewise linear control**: the most common method is to switch between two (often linear) controllers. The first one is specifically tuned for the period when backlash is present. It has to be smooth enough not to excite the backlash too much and consequently is rather conservative. While the second one is optimized for the period when no backlash is present anymore.

**Active nonlinear control**: often the same principle as with passive nonlinear controllers is adopted. However, the first controller actively controls the backlash angle, enabling leaving the backlash as fast as possible in a smooth way. The second controller switches back to control of the wheel torque.

Based on an estimation of the backlash in automotive powertrains, Lagerberg & Egardt (2002) evaluate various control strategies, concluding that active nonlinear controllers have most potential for improved backlash control. The common solution used in the automotive industry today is to use linear, conservative controllers, robust enough to deal with the nonlinearity. A more sophisticated solution is to detect situations when the sign of the requested torque changes and accordingly use an passive or active nonlinear control strategy. However, these controllers tend to be less robust for variation in the backlash size as well as for measurement noise. Consequently good estimators for the actual size of and position in the backlash are needed, which accordingly, have to operate continuously and online.

Little is written on backlash estimation in rotating systems specifically concerning automotive systems. Lagerberg & Egardt (2003b) use switched Kalman filtering theory to determine an estimate for the size of and for the position in the backlash gap. The results are simulation based, however practical implementation is kept in mind as the backlash size is determined by the difference between the position measurements from the speed sensors at the engine and wheels (available by common ABS sensors). The estimate for the position has a higher bandwidth than the size estimator and follows the true position in the backlash gap with high accuracy. In (2003a), they increase the bandwidth of the position estimate by use of a specially developed event-based Kalman filter as pre-filter on the position measurement signal. In (2004) the simulation results are validated by experimental measurements and online estimations. No
3.2. THE POWERTRAIN AND UNDESIRED DRIVELINE DYNAMICS

control strategy is implemented yet however.

**Clutch facings**
The gradient of the friction coefficient of the clutch facings with respect to the slip speed between the two clutch plates plays a significant role in the stability of the clutch system. If the friction coefficient increases (or decreases) while the slip speed decreases (or increases), this gives rise to instable behaviour and vibrations, i.e. clutch judder and stick-slip. For e.g. ceramic facing materials, examples are known for which this effect occurs at low slip speeds, see Figure 3.5. In the range of 200 to 0 rpm, the friction coefficient \( \mu \) increases. This thus implies the excitation of clutch judder at clutch engagement (Bostwick & Szadkowski 1998).

![Figure 3.5: A typical example of the coefficient of friction for clutch facings of ceramic material.](image)

In the next steps, a description of the origination of the phenomena of clutch judder and stick slip is given in which \( \omega_{sl} = \omega_e - \omega_{cl} \) represents the slip speed in the clutch and \( \mu(\omega_{sl}) \) represents the \( \omega_{sl} \)-dependent friction coefficient of the clutch facings (see also Figure 3.1).

1. A constant slip speed \( \omega_{sl} \) and an accordingly constant coefficient of friction \( \mu \), yield a friction torque \( T_{cl} \) on the output shaft of the clutch. This torque causes a twist \( \theta_{cl} \) of the elastic clutch output shaft, i.e. the elasticity in the rest of the driveline.

2. When the clutch is closed, \( \omega_{sl} \) decreases. If \( \omega_{sl} \) decreases, while \( \mu \) increases, which implies a negative rate of change of the friction speed with respect to the slip speed, the equilibrium of the system will be disturbed and \( \theta_{cl} \) will increase, thus increasing \( T_{cl} \).

3. The increase of \( T_{cl} \) leads to an increase of \( \omega_{sl} \), thus decreasing \( \mu \) as well as \( \theta_{cl} \).

4. The system re-establishes an instantaneous equilibrium. However, \( \omega_{sl} \) is still forced to decrease by the closing of the clutch, so the next vibration cycle starts (see Figure 3.6).

5. Although friction forces generally dissipate energy from a vibrating system, in this case energy is actually fed into the system. Consequently, the self-excited vibrations are generated and magnified in amplitude, which is why this phenomenon is referred to as "negative damping".

6. Due to the magnification in amplitude, the slip speed becomes zero, \( \omega_{sl} = 0 \), at a certain point before total lock-up of the clutch is reached, see Figure 3.6. From this first occurrence of \( \omega_{sl} = 0 \), the motion represents a stick-slip phenomenon. The stick-slip phenomenon is associated with the nonlinear intermittent sliding and stichon (sticking) of these sliding contacts, due to the existence of this vibration at a very low slip speed.
Figure 3.6: Example of origination of clutch judder and stick-slip phenomena during the clutch engagement process.

Hence, the existence of clutch judder and stick-slip depends on the friction characteristics of the clutch facing, system dynamics and external rotational and normal forcing (Bostwick & Szadkowski 1998, Crowther et al. 2004, Centea et al. 1999).
3.3 Integrated clutch control in general

A brief overview of the state-of-the-art in AMT focussing on heavy duty trucks, is given in Section 3.3.1. The development of research to control of only the engine torque to provide damping and minimization of driveline oscillations is discussed in Section 3.3.2. After that, the general structure and specific control objectives of integrated clutch control are discussed (Sections 3.3.3, 3.3.4), forming a basis for the research discussed in Section 3.4.

3.3.1 State-of-the-art AMT control

Starting in the 1980s, the idea of implementing heavy duty trucks with AMTs and solving the corresponding control problems using advanced control techniques was initiated (Bader 1990). One of the first examples in literature of an application of advanced control techniques to improve driveability via improvement of the automatic clutch engagement controller in a truck is presented by Slicker & Loh (1996). In (1990) the transmission division of General Motors Corp. is one of the key-players in performing research to electronically controlled automatic transmission systems utilizing advanced technologies. Bender & Struthers (1990) discuss a transmission system incorporating a hydromechanical transmission subsystem and sensors to measure the throttle position, the engine speed and the output speed of the transmission. Using these measurements, the hydraulic control module is controlled.

In recent years, AMTs have become the commonly used transmission for heavy duty trucks. AMTs combine the advantages of manual transmissions, i.e. lightness, high efficiency and manual mode possibilities, with the advantages of automatic transmissions, i.e. the ease of use and driving comfort (Dolcini et al. 2006). The current key-players in this market are Allison Transmission (General Motors), American Axle, ArvinMeritor, Chicago Rawhide, Dana Corp., Eaton Corp., Federal-Mogul Corp., Holland Group, LuK Automotive Systems, Rockford Powertrain, The Timken Company, Transmission Technologies Corp. (TTC), Valeo Transmissions and ZF Friedrichshafen AG.

Originally, automated clutch management systems were developed to improve driver comfort. The systems aimed at easier operation of the clutch system and a quieter environment, positively influencing the safety of the vehicle as well (Nordgard & Hoonhorst 1995). Research to damping and minimizing excitation of driveline oscillations via cooperating engine and transmission control system started in the mid 1990s (Andersson & Johansson 1998). Nowadays, the clutch engagement process is evaluated by numerous variables, i.e. the amount of fuel consumption, the clutch facings wear during the process, the time to lock-up and the smoothness of the process. Modern AMT control systems incorporate a large variety of control functionality on different levels. The main functions comprise automatic gear selection, gear shifting and clutch control. The corresponding control functions take into account driver’s input, e.g. pedal position and gear shift preferences, as well as the truck’s working condition via e.g. load and road slope estimations. Consequently, modern AMT systems comprise rather extensive and complex control systems.

Today, specialized first tier suppliers incorporate most of this functionality in extensive AMT subsystems. E.g. most of the European key-players on the OEM market for heavy duty trucks use the AS (Automatic Shifting) Tronic system of ZF Friedrichshafen AG, which is one of the world’s largest independent suppliers of powertrain components and systems (see Table 3.2). The OEMs are challenged with the integration of all subsystems, i.e. the AMT, the engine, etc., in their vehicles, establishing certain driveability characteristics for specific working conditions. The OEMs try to distinguish themselves by incorporating their own preferences and requirements during integration of these systems. For example implementation of a special manoeuvring mode besides the standard driving mode in which the clutch is regulated more accurately in low gears and reverse via the accelerator pedal. In this way control characteristics of the automatic transmission are changed, such that they fit the specific manoeuvring working conditions. Furthermore, additional control functionality as e.g. the Hill-Start-Aid to improve driveability, comfort of the driver and fuel economy are implemented. The Hill-Start-Aid is used in uphill drive-away situations. The brakes are not released until the engine torque is sufficiently high to prevent rolling downhill.
### Table 3.2: Overview of the state-of-the-art in automatic transmissions for the key-players on the OEM market for heavy duty trucks in Europe.

<table>
<thead>
<tr>
<th>AMT system</th>
<th>DAF</th>
<th>Iveco</th>
<th>MAN</th>
<th>Mercedes</th>
<th>Renault</th>
<th>Scania</th>
<th>Volvo</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>AS-Tronic</td>
<td>EuroTronic II</td>
<td>Tipmatic (AS-Tronic)</td>
<td>EAS Telligent G231/211</td>
<td>OptiDriver (AS-Tronic)</td>
<td>Opticruise (AS-Tronic)</td>
<td>I-Shift GearTronic</td>
</tr>
<tr>
<td>extra manoeuvring mode</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>gear shift time / drive torque interruption time</td>
<td>0.2 – 0.6 s</td>
<td>0.4 – 0.6 s</td>
<td>0.2 – 0.6 s</td>
<td>0.4 – 0.8 s</td>
<td>0.4 – 0.8 s</td>
<td>0.2 – 0.4 s</td>
<td></td>
</tr>
<tr>
<td>engine break assist at gear shifting</td>
<td>✓</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>economy / power mode</td>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>eco-roll function</td>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>hill-start-aid</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>✓</td>
</tr>
</tbody>
</table>

This requires an integrated control approach, which has to be adopted by the OEMs. However, they are bounded by the limitations of the subsystems delivered by their suppliers. Due to the complexity of the AMT systems, the OEMs’ understanding of and the insight in the systems delivered by the suppliers, is diminished significantly. Furthermore, as more functionality is incorporated in the AMT subsystems, the optimization with respect to the rest of the powertrain becomes more difficult for the OEMs.

In Table 3.2, an overview of the state-of-the-art in automatic transmissions and their corresponding control systems for heavy duty trucks is given. Today’s transmission systems as presented in Table 3.2 all incorporate estimation of the load or mass of the vehicle as well as the road slope. Generally, the change in vehicle speed due to a certain change in engine torque is measured. This information is used in selecting the appropriate gear. Functionality as a kick-down function is incorporated in all systems and all systems pretend to optimize driver’s comfort and fuel economy, while minimizing wear of the clutch plates via an appropriate shift strategy. Furthermore, a clutch pedal is absent, while the clutch engagement process can be controlled via the gas pedal. Manual gear selection is still optional in most cases.

Table 3.2 confirms the current trend of OEMs buying their components from specialized suppliers as mentioned in the introduction. However, the I-Shift of Volvo, for example, is developed by Volvo itself. This enables optimal integration of the AMT system in the total powertrain. As a result, the I-Shift indeed is a benchmark system in the field of AMT systems at the moment (Zeitzen 2005). This emphasizes the beneficial opportunities in application of integrated control in automotive systems.

### 3.3.2 Engine control developments

Focus of this study lies on cooperative control of the engine and the transmission systems, i.e. Integrated Clutch Control. However, damping or minimization of driveline oscillations is the target rather than integrating the control of engine and transmission systems at all costs. In the same context, much research to solely control of the engine is performed as well in recent years. The same underlying modeling of the powertrain is used. A brief overview of this research is given in this section.

Analogous to research to cooperating engine and transmission control systems to damp or minimize driveline oscillations, research to solely engine control started in the mid 1990s (Andersson & Johansson 1998). After several studies to the effects of engine output torque and the driveline...
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nonlinear characteristics on the vehicle driveability, Mo et al. (1996), Richard et al. (1999) and Salle et al. (1999), amongst others, start with active engine control to achieve active damping of resonances in the driveline, thus improving the vehicle driveability. Next, e.g. Pettersson & Nielsen, working in cooperation with Scania, proceed with this idea. After extensive modeling of the driveline (Pettersson 1996), they started with improving the existing speed control of diesel engines, the so-called "Regler Quer Verstellung" (RQV), of a truck (1997), showing the benefits of relatively simple modeling and reducing driveline resonances. In (1997), (2000) they propose a novel strategy for controlling the engine to a state where neutral gear can be engaged. This strategy is used in accomplishing gear shifting by engine control, while actively damping driveline resonances. Validation via field trials with a Scania 144L truck is performed. In (2003) they derive a model-based state-feedback controller using LQG techniques to reduce driveline oscillations. Again, validation is done via field trials with a Scania 124L truck. The field validation shows the robustness of the proposed strategies for e.g. nonlinear engine torque limitations. Next, model based feed forward control in combination with LQ feedback control is proposed (Bruce et al. 2005). Focus lies on damping powertrain oscillations that are excited during a tip-in (step throttle input) on gear seven. Validation tests are now performed on a Volvo FH12 heavy duty truck. Fredriksson et al. (2002) investigate several linear powertrain control strategies in order to actively damp torsional oscillations in the driveline, thus improving the vehicle’s driveability. Validation tests with a Volvo FH-12 truck show the satisfactory working as well as the limitations of the controllers. The main drawback of the proposed methods is the focus on damping rather than prohibition of undesired driveline oscillations. Hence, this study focuses on one of the main causes of induction of these oscillations, the clutch engagement process.

3.3.3 General structure of ICC

As mentioned in the introduction, Integrated Clutch Control (ICC) is part of the AMT control. ICC comprises control of several functions in the powertrain, i.e. the engine, the clutch system and the gearbox, while focussing on the optimization of the clutch engagement process. In Figure 3.7 a schematic representation of a general implementation of ICC is given. Within the total hierarchical structure, one can discriminate between so-called supervisory, upper level and lower level control.

Supervisory control is used to anticipate at driver’s input, at environmental conditions and at the vehicle working conditions, thus specifying desired control actions and desired control behaviour. Most of today’s automotive control systems lack a distinctive supervisory control level. Commonly, subsystem specific robust control strategies that are able to cope with all working conditions, are used. As a result of recent developments in control of driveability characteristics and automation of mechanical systems however, interpretation of driver demands and driver behaviour have gained significant importance. The supervisory level often is used to translate driver demands and driver behaviour into specific control settings. Shen et al. (1997) for example, propose a rule-based controller for the supervisory of several controllers, controlling the process of clutch engagement under varying working conditions. And Ercole et al. (1999) design a neural network and corresponding fuzzy supervisor to deal with the driver’s behaviour.

The lower level control part comprises component specific, often built-in controllers. I.e. an AMT typically incorporates a gearbox in combination with a dry plate clutch system, both operated by either electro-mechanic, electro-hydraulic or electro-pneumatic actuators. Control of these actuators is actuator specific and thus commonly built-in in the corresponding software. Lower level control in the context of integrated clutch control comprises control of the throttle actuator, the clutch actuator and the automated gear shifting actuator. The closed loop dynamics of these actuators are often taken into account in the modeling of the powertrain dynamics instead of focussing on controlling them. Therefore, they are often referred to as lower level controllers.

This study focusses on the upper level control strategy, hence no specific attention to the supervisory level and lower level control parts will be given furtheron. However, within the context of smooth clutch engagement and damping of driveline oscillations, specific research to the design of lower level controllers is performed as well. Particularly the clutch actuator controller is often
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3.3.4 Upper level ICC

This study focuses on the design of an integrated upper level control strategy to optimize the clutch engagement process in minimizing excitation of driveline oscillations. In this section, the general structure of the ICC is discussed as well as the general objectives and evaluation variables.

The engagement control of dry clutches must satisfy different and sometimes conflicting objectives. In general, four different control objectives can be distinguished (e.g. Ümit Özgüner 2006).

1. Minimizing excitation of driveline oscillations, thus improving the vehicle’s driveability. This is closely related to the speed difference between the clutch driven plate and passive plate, the so-called slip speed, just before clutch lock-up. Commonly, requirements to this slip speed or the derivative of the slip speed just before clutch lock-up, are defined.

2. The dissipated energy during the clutch engagement process is related to the wear of the clutch facings. Hence, this amount of energy, i.e. the friction losses, should be minimized.

3. During the engagement process, direct coupling of the engine to the wheels is absent and thus no driving power is present. Hence, fast lock-up is required.

4. Engine stall or overdrive during the engagement process should be prohibited. Commonly, these objectives are included in the control system of the engine. However, if the clutch and

Figure 3.7: Schematic overview of the implementation of integrated clutch control, based on the general structure for integrated automotive control (Figures 1.2, 1.3, Section 1.2). This study focuses on the upper level control (the grey block) in which Integrated Clutch Control is located. The vertical black line represents the controller boundary, incorporating controller inputs on all control levels. A typical example of 1) an environmental condition is the road slope, 2) a strategic control setting comprises the driving mode, i.e. manoeuvring or normal mode, 3) an immediate control setting is the gas pedal actuation, 4) vehicle state parameters are e.g. the engine output torque $T_e$ and engine rotational velocity $\omega_e$.
engine control system are integrated, this should be taken into account in the design of the ICC as well.

Generally, various combinations of some of the above presented objectives or self-established objectives derived from these objectives are used. Hence, a way to standardize results of research to damping and minimizing excitation of undesired longitudinal oscillations has to be defined.

A general control structure for the upper level part of ICC is depicted in Figure 3.8. Figure 3.8(b) shows a simplified representation of Figure 3.8(a) which will be used furtheron. This general control structure enables a clear comparison of different control strategies. Some remarks to Figure 3.8 are given next.

- The Engine with the engine input torque $T_e$ and the output rotational velocity $\omega_e$ includes the nonlinear engine torque-speed mapping and several built-in controllers. These controllers are mostly lower level controllers. Generally, the corresponding dynamics are incorporated in the dynamic modeling of the corresponding actuators or the total subsystem dynamics, i.e. the engine dynamics.

- The Clutch system comprises the actual clutch plates as well as the clutch actuator, including the corresponding clutch actuator controller. This built-in lower level controller will not be taken into account furtheron. Generally, the corresponding dynamics are thus incorporated in the dynamic modeling of the clutch actuator or clutch system. In the control strategies discussed, a reference clutch displacement $x_{cl}^{ref}$, a reference actuator pressure $P_{cl}^{ref}$ or a reference clutch output torque $T_{cl}^{ref}$ is determined as the input for the controller. Figure 3.9 gives a more detailed representation of the Clutch system of Figure 3.8(a) in

- The engine speed $\omega_e$, the rotational speed of the clutch passive plate $\omega_{cl}$, the gear ratio $i_g$ and the driver’s input represented by the throttle angle $\alpha_{th}$ are generally used as inputs for the Integrated clutch controller. I.e. peripherals are neglected to this point, but may be
Figure 3.9: More detailed representation of the Clutch system of Figure 3.8(a) with $u_{cl} = x_{cl}^{ref} | F_{cl}^{ref} | T_{cl}^{ref}$, $y_{cl} = x_{cl} | F_{cl} | T_{cl}$ and $u_{cl,a}$ the actual input control signal for the clutch actuator.

added to the driver’s input. The lower level controller for the throttle actuator combining $i_g$ and $\alpha_{th}$ is included in the Integrated clutch controller. Furthermore, the controller consists of a part controlling the engine with the engine torque $T_e$ as output and a part controlling the actual opening and closing of the clutch via a prescribed position, force or torque (see the previous bullet).

- The Rest of the driveline contains the transmission, propeller and drive shafts, the differential, the wheels, the environmental conditions as the road slope and the load torque.

- All sensor dynamics as well as the estimators and observers needed to generate the desired system output signals are assumed to be included in the corresponding components. I.e. in general the engine speed $\omega_e$ and the clutch passive plate speed $\omega_{cl}$ are assumed to be known and are used to determine the slip speed $\omega_{sl} = \omega_e - \omega_{cl}$ in the clutch.

- Target slip speed and target engine speed are calculated taking into account pedal position (driver’s input), vehicle velocity, etc. Hence, these signals have to be considered as inputs to the integrated clutch controller. However, most research focusses on the design of the controller rather than the design of smooth target or setpoint profiles.
3.4 Research to integrated clutch control

In this section, a more detailed overview of research to ICC is given. Focus lies on the clutch engagement process, aiming at damping or minimization of excitation of driveline oscillations. To start with, an overview of the main research groups in the world performing key research to clutch control is given in Section 3.4.1. This is followed by an overview of specific research to ICC in the Sections 3.4.2 to 3.4.7.

3.4.1 Research groups

In Figure 3.10 the main research groups concerning (integrated) clutch engagement control, focussing on damping and minimizing excitation of undesired driveline oscillations are indicated.

Figure 3.10: Overview of the main research groups concerning (integrated) clutch engagement control focussing on damping and minimizing excitation of undesired driveline oscillations in the world.

In recent years, research groups of the Università di Siena, the Università degli Studi del Sannio and the Università degli Studi di Napoli in Italy (10) performed much research to the subject of clutch engagement control. In cooperation with Magneti Marelli, FIAT (10) and the ETH Zürich (2), Glielmo & Vasca, Bemporad, Borrelli, Glielmo & Vasca and Garofalo et al. worked on the identification, the (hybrid) modeling and the (optimal) control of the process. Ercole et al. of the Politecnico di Torino (12) also cooperate with Magneti Marelli, focussing on a more integrated approach. The research group of the Università di Modena (11) (Morselli et al., Zanasi et al.) adopts a different approach and focusses on the (POG) modeling of the vehicle and the driver as well as on a thoroughly design of setpoint profiles for the closed loop controller. Furthermore much research is done in these groups to optimal gear shift control strategies as well.

In Germany, recent work on control of automatic transmissions is done by e.g. Haj-Fraj & Pfeiffer and Treder & Woernle of the Technische Universität München and the Universität Rostock (16). Furthermore, Mercedes Benz, Siemens VDO Automotive and DaimlerChrysler AG (17) show interest in this research topic for a lot of years already. Bader (1990) and Nordgård & Hoonhorst (1995) as well as Giovanardi et al. (2005) discuss this subject specifically focussing on integrated (clutch) control.
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Much research to damping and minimization of excitation of undesired driveline dynamics, specifically focusing at heavy duty truck application, is done by research groups of the Chalmers University of Technology, the Jönköping University of Technology and the Linköping University in cooperation with Volvo and Scania (1), (9). I.e. modeling of heavy duty truck powertrains (Pettersson & Nielsen), integrated powertrain control (Fredriksson & Egardt), research to the phenomenon of backlash (Lagerberg & Egardt), model-based clutch controller design (Bruce et al.) and the objective evaluation of vehicle driveability (Persson) are some of their research topics. Due to the close cooperation with Volvo and Scania, experimental validations on heavy duty trucks can be performed. However, attention focuses on control of the engine to damp undesired driveline dynamics instead of optimization of the clutch engagement process.

In the USA, General Motors, Allison Transmission Div., Eaton Corp., Chrysler Corp. and Dana Corp. (7), (8) amongst others, are active in this field of research in cooperation with the University of Michigan and the University of Toledo. Their research comprises quite a lot of years as Bender & Struthers (1990) were some of the first ones to publish about AMT control, Slicker & Loh some of the first ones to actually apply robust control in control of the clutch engagement process, Szadkowski & Morford (2001) performed much research to the dynamics of driveline oscillations and recently Sun & Hebbale (2005) discuss the concept of integrated clutch control and David & Natarajan published a new clutch engagement control strategy based on the work done by the research group of Glielmo & Vasca in Italy (10).

With respect to other research groups, research on the subject of clutch control in Asia, i.e. Nissan, Isuzu, in cooperation with the Muroran Institute of Technology (3), Toyota (4) in Japan and the Jiao Tong University in Shanghai (6) in cooperation with Daimler Chrysler Corp., China, comprises relatively complex control techniques. Neural and fuzzy control solutions (Hibino et al. 1996, Hayashi et al. 1995, Zhang et al. 2002, Shen et al. 1997) and application of µ-synthesis to design a robust controller (Adachi et al. 2004) are proposed.

Research to the dynamics specifically during clutch engagement control, is performed all over the world. Besides the already mentioned research groups, some recent research specifically focusing on vehicle driveability, i.e. research to the phenomena of judder, stick-slip, shunt, etc., is done by Centea et al. (1999), University of Bradford (14) and Deur et al. (2006), University of Zagreb (18) working in cooperation with Ford and Crowther et al. (2004), University of Technology Sydney (5).

Within the region of Eindhoven itself (15), recently the Technische Universiteit Eindhoven has started this project in cooperation with DAF Trucks and TNO Automotive. The clutch control problem is regarded as a case study example, focusing on low-complexity design and automated tuning of automotive controllers. Furthermore, in recent years, several projects in cooperation with DriveTrain Innovations (DTI) are developed. I.e. Serrarens et al. (2004) and Kusters (2005) propose a control strategy specifically targeting at control of the clutch engagement process, based on the work of amongst others Glielmo & Vasca and Bemporad, Borrelli, Glielmo & Vasca (10).

3.4.2 Robust ICC

Considering variations in the working conditions to be disturbances, application of robust control in the design of integrated clutch control is of interest. Variations in the working conditions may encompass the variation in load, variation in the friction characteristics of the clutch, induced by aging deterioration, variation in the road slope, changes in gear or variation in engine torque related to driver’s input are meant.

Using Quantitative Feedback Theory (QFT), Slicker & Loh (1996) propose a robust double-loop feedback controller, see Figure 3.11. The inner loop is designed using QFT, while the outer loop contains a PID controller to control the inner closed-loop dynamics. The inner loop QFT controller assures robust stability and robust tracking. The pre-filter enables the design of smooth 2nd-order inner loop dynamics, assuring smooth clutch engagement. The outer loop PID controller is tuned to achieve exponential synchronization of \( \omega_{cl} \) and \( \omega_e \) to lock-up. The feed forward signal enables solving engine controllability problems as stall and flaring. Slicker & Loh specifically focus on smooth vehicle drive-away dynamics, i.e. smooth clutch engagement. No specific attention is paid
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Figure 3.11: Schematic representation of the control strategy proposed by Slicker & Loh (1996).

Adachi et al. (2004) propose a robust control strategy to attain robust performance for varying vehicle parameters (see Figure 3.12). I.e. the engine torque $T_e$ is treated as a disturbance. The slip speed $\omega_{sl}$ is controlled during the clutch engagement process, based on a reference slip speed $\omega_{sl}^{ref}$. The Reference model of Figure 3.12 filters the commanded setting for the slip speed $\omega_{sl}^{set}$ to determine a smooth reference signal $\omega_{sl}^{ref}$. The Robust controller is designed using $\mu$-synthesis and is combined with a Feed forward controller. An integral weighting function is applied in addition to the robust controller to eliminate steady-state errors. The feed forward controller is calculated based on the modeling of the system. Validation on a test vehicle showed too conservative behaviour of the controller. Consequently, two open-loop controllers are designed, taking over the control of the robust controller at the starting and ending parts of the engagement process. In this way, requirements for driveability and clutch durability are satisfied.

Hibino et al. (1996) are the first ones presenting a robust control strategy that is actually implemented in a commercially available automotive component. Although the controller is designed for regulating the slip speed of a torque converter system, the problem statement is comparable to that of the above examples. Hibino et al. state that the used small gain theorem cannot guarantee stability for transient characteristics in perturbed situations. Consequently, an additional frequency dependent disturbance filter is designed to attenuate the disturbances without sacrificing the performance of the $\mathcal{H}_\infty$ controller, see Figure 3.13.
3.4.3 Fuzzy and neural ICC

Neural control comprises a control system, which is tuned using experimental data. Based on measured input-output data, the system adapts its parameters to mimic the system’s behaviour as good as possible. The resulting (nonlinear) system can be converted to fuzzy logic rules, which then exhibits a qualitative representation of the measured input-output behaviour. This gives rise to parameter and variable estimation and learning of a driver’s intention, based on interpretation of measured signals and experience.

Automotive controllers have to cope with varying working conditions, dependent on the vehicle’s dynamics, the driver’s input and the environmental influences. Besides that, contradictory objectives as minimization of the clutch engagement time and clutch facings wear as well as optimization of comfort in the shortest time possible have to be fulfilled. Several researchers (e.g., Shen et al. 1997, Zhang et al. 2002) believe fuzzy and neural approaches to be a promising interface between the driver and the vehicle in e.g. the control of clutch engagement. Moreover, fuzzy logic is shown to work effectively in managing the problem of gear selection in AMTs by several researchers. E.g. Hayashi et al. (1995) use a neural-fuzzy approach in developing an optimal gear-shift controller. Fuzzy control is used to interpret the driver’s intention and estimate the load. Neural control is used to achieve the optimal gear for a complete envelope of working conditions.

Shen et al. (1997) amongst others, propose a rule-based controller for the supervision of several adaptive (intelligent) PD and PID controllers, controlling the process of clutch engagement under varying working conditions. A distinction is made between lower level control, which involves the clutch actuator and the throttle actuator controllers, and upper level control. Liu et al. (2002) focus on the lower level controller for the clutch actuator, while Zhang et al. (2002) focus on the design of the upper level clutch engagement controller. The adaptation mechanism of the upper level controllers proposed by Shen et al. (1997) adapts to the varying working conditions using fuzzy reasoning. Experimental results show the proper working of this mechanism.

The driver’s behaviour is fully defined in a parametric way by Ercole et al. (1999). The influence of parameter variations on the clutch engagement process concerning drive-away dynamics as well as gear upshifts are considered. A fuzzy supervisor is designed using cost functions, which take into account specific evaluation variables, i.e. the amount of dissipated energy, the amount of excitation of driveline oscillations and the total time the clutch engagement process takes. The supervisor prescribes optimal input parameters for the throttle valve opening derivative $\dot{\alpha}_{th}^{ref}$ and the clutch position derivative $\dot{x}_{cl}^{ref}$. However, the results could not be used directly in actual applications due to e.g. the large amount of computing power required.

3.4.4 Decoupled MIMO ICC

The main control objectives can be formulated in terms of the engine speed $\omega_e$ and the slip speed $\omega_{sl}$. The control outputs consist of the engine torque $T_e$ and the control signal for the clutch actuator $u_{cl}^{ref}$ (see Section 3.3.3). Hence, the control problem can be interpreted as a MIMO control problem, which can be decoupled using for example feedback linearisation techniques.
The control problem can then be formulated as

\[
\begin{pmatrix}
T_e \\
u_{cl}^{ref}
\end{pmatrix} = A(s) \begin{pmatrix}
C_e(s) & 0 \\
0 & C_{cl}(s)
\end{pmatrix} \begin{pmatrix}
\omega_e \\
\omega_{sl}
\end{pmatrix}
\]

(3.1)

with \(A(s)\) the decoupling controller and \(C_i(s)\) two decoupled linear SISO controllers. In Figure 3.14 a general overview of this control strategy is shown.

In order to be able to apply feedback linearisation techniques, the modeling of the system is of major importance. Garofalo et al. (2001) propose a piecewise linear LTI model of the clutch system to distinguish between the locked (or engaged) clutch phase and the slipping (or disengaged) clutch phase. This hybrid model is used to design a decoupling tracking controller. Control of \(\omega_e\) and \(\omega_{sl}\) is then achieved via tuning of two classical PID controllers, directly steering \(T_e\) and \(T_{cl}\). The reference signals \(\omega_{ref}^{e}\) and \(\omega_{ref}^{sl}\) are chosen such that engine stall is prohibited respectively smooth engagement is assured. Garofalo et al. use the derivative of the slip speed \(\dot{\omega}_{sl}(t)\) at the moment of lock-up, \(t = t^*\), to quantify the smoothness of the engagement process.

Serrarens et al. (2004) adopt the same strategy, applying PI instead of PID controllers for the two SISO controllers. The same evaluation variables are used. However, simulations show large peaks in the drive torque, i.e. the torque in the drive shafts. A modified controller is proposed, controlling the drive torque instead of the torque transmitted by the clutch system. A peak in the drive torque during the engagement process is prohibited, but the rest of the driveline exhibits undesired dynamics instead.

### 3.4.5 Optimal ICC

As mentioned in the introduction of this section, the ICC control objectives are contradictory. Hence, application of optimal control techniques including weighting functions seems appropriate. In Figure 3.15 a general structure of an optimal control strategy applied to the integrated clutch control problem is shown. The objective function \(J\) is minimized over the time of the engagement process till clutch lock-up at time \(t = t^*\). The results of the off-line minimization are saved in a look-up table (LU-table) so that they can be used on-line. \(J\) generally is a weighted function of the control output \(u_{cl}\) and the state \(x\) of the problem. The state \(x\) may contain e.g. the slip \(\omega_{sl} = \omega_e - \omega_{cl}\) in the clutch and / or the engine rotational speed \(\omega_e\).

Glielmo & Vasca (2000b), (2000a) proposes a Linear Quadratic (LQ) controller design, based on an optimal control strategy. To prevent the clutch normal actuator force \(F_{cl}\) from varying discontinuously and to be able to set constraints to this force, \(F_{cl}\) is included in the state vector of the system, \(x_3 = F_{cl}\). This yields \(u_{cl} = \frac{d}{dt} F_{cl}\), which requires an integrating action in series with the controller to determine a reference normal force \(F_{cl}\). The objective function to be minimized equals

\[
J = \int_0^{t^*} \left\{ q \omega_{sl}^2(t) + r u_{cl}(t)^2 \right\} dt
\]

(3.2)
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**Powertrain**

**Engine**

**Drive-line**

**Controller**

\[ u_{cl} = \gamma T x + \rho \]

**LU-table**

\[ x = \int_{0}^{\gamma(t)} f(x, u) \, dt \]

\[ \gamma(t), \rho(t) \]

**Controller**

\[ u_{cl} = \gamma T x + \rho \]

**Powertrain**

\[ J = \int_{0}^{t^*} \left\{ k \omega_{sl}(t) F_{cl}(t) \right\} \, dt \]

**Figure 3.15:** General structure of an optimal control strategy. The results of minimizing the objective function \( J \) are saved in a look-up table (LU-table). The state of the system \( x \) as well as the control input \( u_{cl} \) may either be single variables or vectors. Hence, the control problem may be regarded as SISO, MIMO or a combination of both.

with \( q \) and \( r \) constant weighting functions. The state \( x \) of the problem equals

\[ x = \begin{pmatrix} \omega_e \\ \omega_{sd} \\ F_{cl} \end{pmatrix} \]  

(3.3)

A look-up table containing the results of the objective function is used for simulations purposes. In evaluation of the controller, specific attention is given to driver comfort and dissipated energy \( E_d \). Driver comfort is related to the prevention of driveline oscillations, which are assumed to be dependent on the slip acceleration \( \dot{\omega}_{sl}(t) \) at the moment of clutch lock-up, \( t = t^* \). The dissipated energy is represented by

\[ E_d = \int_{0}^{t^*} \left\{ k \omega_{sl}(t) F_{cl}(t) \right\} \, dt \]  

(3.4)

A lower bound for the total time the engagement process takes, is obtained using simulation results with a so-called minimum time controller. I.e. within defined actuator saturation limits, the clutch is closed as fast as possible. With respect to this minimum time controller, good results are obtained with the proposed optimal control strategy considering the evaluated variables. Constraints on the control and state variables are not considered explicitly in the solution of the problem. This may lead to practical implementation problems.

Based on the hybrid modeling of Garofalo et al. (2001) and the before discussed control strategy proposed by Glielmo & Vasca (2000b), Garofalo et al. (2002) proposes a finite horizon LQ optimal tracking controller. The proposed objective function equals

\[ J = \frac{1}{2} \int_{0}^{t^*} \left\{ \varepsilon_x(t)^T Q \varepsilon_x(t) + u_{cl}(t)^T R u_{cl}(t) \right\} \, dt \]  

(3.5)

with \( \varepsilon_x = x - x^\text{ref} \), \( Q \) and \( R \) the weighting functions and \( u_{cl}(t) = (T_e + T_{cl})^T \) the control outputs (see Figure 3.16). The reference trajectories \( x^\text{ref} \) assure prevention of engine stall and engine flame-out as well as smooth clutch engagement. Again driver comfort is evaluated and related to the slip acceleration \( \dot{\omega}_{sl}(t^*) \) at clutch lock-up. Considering the driveline oscillations during the engagement process, the results of simulations using a 6th-order model in combination with the proposed LQ controller exceed those of the decoupled PID controllers of Garofalo et al. (2001). Experimental validation is lacking yet.

David & Natarajan (2005) further elaborate on the concept of optimal control based on LQR methods for the design of two clutch engagement strategies. The first strategy equals the design
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Figure 3.16: Schematic representation of the MIMO optimal control strategy proposed by Garofalo et al. (2002).

proposed by Glielmo & Vasca (2000b) and is used as a benchmark for the results of the second strategy. In the second strategy they add the evaluation variables for driver comfort and energy dissipation to the objective function used in the first strategy. Consequently, the acceleration of the position difference $\dot{\omega}_{sl}$, and the energy dissipation during the engagement process, represented by $E_d = \omega_{sl} T_{sl}$, are also weighted in calculating an optimal solution. The resulting objective function equals

$$J = \frac{1}{2} \int_0^\infty \left\{ \alpha \omega_{sl} T_{cl} + \beta \omega_{sl}^2 + \gamma \dot{\omega}_{sl}^2 + r u^2 \right\} dt$$

with $\alpha$, $\beta$ and $\gamma$ so-called performance index constants and the derivative of the clutch actuator torque $u_{cl} = \frac{dT_{cl}}{dt}$ the new control input. The controller is used in series with an integrating action to determine a reference clutch actuator torque $T_{cl}$. $\frac{dT_{cl}}{dt}$ instead of $T_{cl}$ is used as control output, as this should better mimic driver behaviour. In practice, the driver used to control $T_{cl}$ by changing $\frac{dT_{cl}}{dt} F_{cl}$ via the clutch pedal, actually controlling $\frac{dT_{cl}}{dt}$. The performance index constants are tuned to achieve desired controller behaviour. In this way, it is possible to select a specific vehicle drive-away clutch engagement profile for specific situations, e.g. driving away uphill with a fully loaded vehicle. The state $x$ equals

$$x = \left( \begin{array}{c} \omega_c \\ \theta_{cl}/i_{cl} - \theta_w \\ \omega_{cl} \\ \omega_w \\ T_{cl} \end{array} \right)$$

The proposed control strategy is validated via simulations using a model of a class 6 truck, which is validated experimentally.

Zhang et al. (2002) compare the results of fuzzy controllers with those of an optimal control strategy. Feedback linearisation techniques are applied to a model of the engine to be able to design an adaptive observer for the engine conditions i.e. $\omega_e$ and $\dot{\omega}_e$. In combination with this observer, a PID controller is designed, prohibiting engine stall and flame-out. The smoothness of the clutch engagement process is quantified by the shock intensity of the vehicle $\mid \ddot{a} \mid \leq 10 \text{ m s}^{-2}$ and the rotational jerk of the clutch passive plate $\ddot{\omega}_{cl}$ rather than the acceleration in the rotational slip in the clutch $\dot{\omega}_{sl}$. Furthermore, a time weighting coefficient $W$ is related to the amount of dissipated energy during the process and thus to the wear-out of the clutch plates. $\ddot{\omega}_{cl}$ as well as $W$ are taken into account in the objective function used in formulating the optimal control
Lower and upper bounds for the optimization results are determined by requirements concerning wear-out of the clutch facings and driver comfort respectively. The result of the optimization comprises trajectory solutions for the (rate of the) optimal friction torque $T_{cl}$ and $\frac{d}{dt}T_{cl}$. The optimal value for $W$ is determined online using the adaptive engine observer.

Kusters (2005) solves the optimal clutch engagement control problem by application of Dynamic Programming (DP). The DP algorithm, based on the principle of optimality, computes the optimal control input. The engine torque is regarded as a constant disturbance, while the load torque is regarded as a disturbance dependent on $\omega_{cl}$. The problem is formulated in the discrete time domain and the corresponding objective function equals

$$ J = \sum_{i=1}^{N} \left\{ r \Delta u + \alpha \omega_{cl}^2 + \beta (\omega_e - \omega_{ref}e)^2 \right\} + \gamma \dot{\omega}_{sl} $$

(3.9)

with weights analogous to David & Natarajan (see Equation 3.6). The state vector $x$ equals

$$ x = \begin{pmatrix} \omega_e \\ \omega_{cl} \end{pmatrix} $$

(3.10)

In combination with a setpoint $T_{ref}^{cl}$ for the clutch actuator as control output, the resulting control structure is a combination of those proposed by Garofalo et al. (see Figure 3.16) and the general optimal control structure shown in Figure 3.15. Computational limits prohibit the calculation of a controller containing setpoints for both the clutch actuator and the throttle actuator. Simulation results are evaluated by the total time the process takes, the size of $\dot{\omega}_{sl}(t^*)$ and the minimal engine speed to prohibit engine stall.

(Dolcini et al. 2006) combine an optimal control structure with an open-loop control structure to minimize the amount of time the process takes. They split up the clutch engagement process into two parts. During the first part, an open-loop control structure using a look-up table is used, aiming at fast clutch engagement. For the second part, an optimal control strategy is designed, assuring smooth final clutch lock-up. The used state vector equals

$$ x = \begin{pmatrix} \omega_{sl} \\ \dot{\theta}_s \\ \theta_s \\ T_e \\ T_{cl} \end{pmatrix} $$

(3.11)

with $\theta_s$ the torsion in the transmission shafts. In order to determine this variable, observers are derived and tested online. The corresponding quadratic objective function equals

$$ J = \int \left( x^T Q x + u^T R u \right) \, dt $$

(3.12)

with $u = \frac{d}{dt}T_{cl}$ the control output. Practical implementation on a Clio II AMT 1.5dCi prototype gives good results, showing robustness for vehicle parameter variations as well.

### 3.4.6 Model predictive ICC

The LQ problems formulated in the previous section, in general do not incorporate constraints on the control and state variables. Hence, Bemporad, Borrelli, Glielmo & Vasca (2001), (2001) propose a Model Predictive Control (MPC) strategy that does incorporate these constraints. The
3.4. RESEARCH TO INTEGRATED CLUTCH CONTROL

MPC strategy is based upon the before mentioned optimal control LQ formulations of Glielmo & Vasca and Garofalo et al. (see Section 3.4.5). The corresponding objective function equals

\[ J = \sum_{k=0}^{N_y-1} \varepsilon_{t+k|t}^T Q \varepsilon_{t+k|t} + \delta u_{t+k}^T R \delta u_{t+k} \]  

(3.13)

with \( \varepsilon_{t+k|t} \) the prediction error of the model, \( \delta u_t = u_t - u_{t-1} \) the increment of the input signal, \( Q \), \( R \) the weighting matrices and \( N_y \) representing the prediction horizon. The state vector \( x \) equals

\[ x = \begin{pmatrix} \Omega_e \\ \omega_{sl} \\ T_e \end{pmatrix} \]  

(3.14)

The control output incorporates the torque transmitted by the clutch \( T_{cl} \) and the engine torque \( T_e \). In reducing the control law to a piecewise linear affine function of the state vector, off-line computation of the MPC controller is possible (analogous to the look-up tables of the optimal control problems, Section 3.4.5). Analogous to Glielmo & Vasca (2000b), performance evaluation of the controller is done via evaluation of the amount of dissipated energy and comfort. Simulation results show good closed loop performance in combination with admissible controller complexity. After application of the MPC theory to clutch engagement control, Bemporad, Borrelli, Glielmo & Vasca successfully apply the same technique to automatic gear shifting control (Bemporad et al. 2003).

Following the MPC strategy proposed by Bemporad, Borrelli, Glielmo & Vasca, Kusters (2005) designs a comparable controller for the clutch engagement process. Based on a hybrid model distinguishing three phases in the engagement process, a hybrid controller is calculated using the following objective function (see also Equation 3.9).

\[ J = \sum_{k=1}^{N} \left\{ \| \alpha \omega_{sl} \|_2 + \| \beta (\omega_e - \omega_{ref}^e) \|_2 + \| \gamma (\omega_{sl} \dot{\omega}_{sl}) \|_2 \right\} \]  

(3.15)

with \( \gamma = \gamma(\omega_{sl}) \), the latter cost is only taken into account at the end of the engagement process, just before clutch lock-up. The resulting controller first assures minimization of \( \omega_{sl} \), after which smooth lock-up is assured via minimization of \( \dot{\omega}_{sl} \).

3.4.7 Smooth clutch engagement profiles

Whether the clutch engagement process is valued smooth or not, strongly depends on the used target or setpoint speed profiles. Often, setpoint profiles for the slip speed \( \omega_{ref}^e \) and / or the engine speed \( \omega_e^e \) are used. Considering research to ICC, generally little attention is paid to the design of these profiles. However, some research to the total gear shifting process does include this design aspect. Although often not specifically focussing on damping and minimization of driveline oscillations, the design of smooth clutch engagement profiles is discussed to some extent. An overview of the most relevant research concerning the design of smooth profiles for the clutch engagement process is presented in this section.

Based on the hybrid modeling of Garofalo et al. and the optimal controller for smooth engagement developed by Glielmo & Vasca (see Section 3.4.5), a control strategy for gear shifting is proposed by Glielmo et al. (2004), (2006). The gear shifting process is split up in 5 regions. For each region of the process an optimal controller is developed. One of the control objectives is to restrict slip acceleration \( \dot{\omega}_{sl} \) at the moment of clutch lock-up. This is related to clutch engagement smoothness and accordingly, to the excitation of undesired driveline oscillations. A model of a FIAT STILO is used for simulations, which shows smooth and quick gear shifting results. As for each region of the process a specific optimal controller is developed, further developments as controller design specifically targeting at the gear shifting process in trucks or optimization specifically focussing on damping of shunt and shuffle phenomena, may be possible as well.
Analogous to Glielmo et al., 5 different states in gear shifting are determined by Fredriksson & Egardt (2000). The I-Shift of Volvo operates in the same way (Volvo 2006). Starting with (0) decreasing throttle, (1) disengaging the (dog) clutch and (2) putting the transmission into neutral gear. (3) The engine rotational speed $\omega_e$ is synchronized with the rotational speed of the transmission $\omega_t$, accounting for the new gear ratio $i_g$. This is done using either the engine brake or extra throttle. Next, (4) the gear is changed and (5) the (dog) clutch is re-engaged using an algorithm or control law for reduction of driveline oscillations or application of torque to prevent oscillations is applied. By means of integrated powertrain control, Fredriksson & Egardt aim at minimization of gear shift times, exhaust emissions, oscillation excitation and noise and wear on powertrain components.

Morselli et al. (2002) specifically focus on the generation of smooth acceleration profiles for $\omega_e$. The problem of shunt is addressed, focussing at gear shift operations in order to improve the driver’s comfort. Based on the gear shift specifications concerning the time required for a gear shift $t_{gs}$ and the maximum allowable acceleration / deceleration in combination with a (subjective) driver’s model, a smooth acceleration profile using optimal control theory is calculated. This acceleration profile is linked to the engine torque and assures smooth longitudinal driver dynamics, thus prohibiting driver oscillations during gear shifting. In combination with their research to modelling of a vehicle’s powertrain (Zanasi et al. 2001), this may form a basis for good clutch engagement control. Zanasi et al. (2001) address the aspect of controller design for the clutch engagement process already briefly. Assuming the control system to determine a smooth reference signal $\omega_e^{ref}$, the straightforward control law becomes

$$x_{cl}^{ref} = x_{cl}^0 + k (\omega_e^{ref} - \omega_e) \text{sgn}(\omega_{sl}) \quad (3.16)$$

(see Figure 3.17). No specific attention is given to damping or prohibition of driveline oscillations yet.

Figure 3.17: Schematic representation of the clutch control scheme proposed by Zanasi et al. (2001).
3.5 Conclusions and recommendations regarding ICC

Section 3.3 shows the dependency of OEMs on specialized FTSs concerning e.g. AMT systems and the design of the corresponding control systems. Application of integrated control strategies, e.g. ICC, combining control functionality of several subsystems, provides potential benefits and comprises today’s challenges for OEMs. This has led to significantly increased tuning effort for the OEMs. Combination of such automotive control problems with modern control techniques, may provide solutions for this increased effort, e.g. automated tuning. Enabling ICC requires close cooperation of OEMs and their FTSs. Automated tuning requires focussing on well-structured, low complexity controller design.

Based on the overview of research to ICC presented in Section 3.4, some early conclusions regarding the applicability of different control strategies may be drawn. Application of optimal control strategies showed the most promising results to this point. Application of robust control strategies generally resulted in too conservative controller designs. Decoupling the MIMO system using feedback linearisation techniques has shown to be an effective basis for further controller design. The same holds for splitting up the process in several phases via piecewise linear modeling.

Generally, the proposed controller strategies provide many tuning possibilities. In presenting the results however, most attention is paid to the actual design of the controller rather than to the tuning of the resulting controller. Only little research exploits the possibilities for online or even automatic tuning.

A standard way to value results of the research to the control of undesired driveline dynamics and more specifically the research to ICC, is lacking yet. In Section 3.3.3 the commonly used control objectives are defined. Most of the research presented in Section 3.4 comprises a combination of these objective or some related evaluation variables. Furthermore, some constraints on the control and state variables have to be taken into account in the solution of the problem, which is generally not done.

Based on these conclusions, a case study involving further research to ICC, targeting at damping or minimizing excitation of driveline oscillations is initiated in cooperation with DAF Trucks N.V., Eindhoven, The Netherlands. Focus lies on the integrated approach in designing the controller, i.e. considering the dynamics of the powertrain as a whole instead of examining separate subsystems. The crucial points in the dynamics and / or the constraints in the present controller settings and control loops have to be identified. Possibilities for improvement of the current control concept have to be examined and compared to suggestions for a (re)design of the controller concept. This comparison has to be based on a set of general evaluation variables, focusing on robust and low-complexity controller design. A suited control strategy has to be selected and implemented to verify its effectiveness. In addition, further research to automated tuning techniques has to provide solutions for optimal tuning with minimal effort.
Chapter 4

Automated Cruise Control, ACC

4.1 Longitudinal control (LOC)

Over 90% of the road accidents are caused by drivers’ mistakes. Automation of automotive systems is regarded as a possible solution to decrease this number. Specifically active control of the longitudinal dynamics of the vehicle, assisting the driver, provides good opportunities. Active safety systems enable accident avoidance whereas passive safety systems only provide reluctance of the effects of accidents. Automated Longitudinal Control (LOC) of the vehicle can be regarded as an active safety system. Today’s electronically controlled, so-called x-by-wire systems provide wide flexibility in the tuning of longitudinal vehicle dynamics via software. This enables increased potential for application of active safety systems. Moreover, means for increased passive safety are created as well by e.g. the redundancy of the steering column when applying a steer-by-wire system (Aparicio et al. 2005).

Longitudinal control of a vehicle can be divided into conventional and intelligent control (see Figure 4.1). Conventional controllers as Cruise Control (CC), Application Speed Limiters (ASL) or Downhill Speed Control (DSC), do not incorporate detection of a vehicle’s surroundings, i.e. detection of other vehicles or possible obstacles or the curvature in the road. Intelligent controllers on the other hand, take advantages of this detection. This enables new longitudinal control functionality as vehicle following or collision avoidance and potential for adaptation of conventional controllers to specific traffic situations (Aparicio et al. 2005, Güvenç & Kural 2006).

Influencing and controlling the longitudinal vehicle dynamics has been a research topic of interest for several decades until now. Simultaneous to the exponential increase in possibilities due to the developments in software and electronics, the development of research programs to active longitudinal control as driver assistance system has taken an enormous growth in Europe, Japan and the US. Initially, attention focused on Intelligent Vehicle Highway Systems (IVHS) or Automated Vehicle Highway Systems (AHS), i.e. highways equipped with sensors to communicate with the vehicles present, and cooperative control strategies, i.e. control systems requiring inter-vehicle communication. However, these systems require much funding for costly modifications to the present infrastructure and a consistent policy support with general standards. These financial and practical limitations caused short-term research to change focus to more feasible, stand-alone and non-cooperative solutions. Since the 1990s, significant progress in software and electronics has enabled actual implementation of so-called 1st generation driver assistance systems on commercially available vehicles (Figueiredo et al. 2001, Gehring & Fritz 1997, French et al. 1994, Liang & Peng 1999).

Examples of today’s commercially available systems providing the driver with information, warning and operational support and based on detection of a vehicle’s surroundings, are Adaptive Cruise Control (ACC), Forward Collision Warning (FCW) and keep-in-lane systems (see Figure 4.1). 1st generation systems are only allowed to moderately decelerate and accelerate the vehicle or to warn the driver by audible or visual signals rather than actually intervene a driver’s action. In
CHAPTER 4. AUTOMATED CRUISE CONTROL, ACC

Conventional control

Manual operation (driver in control)

Longitudinal speed control systems

Detecting preceding vehicles

1st gen. adaptive cruise control (ACC)

Detecting vehicle surroundings (radar, GPS)

2nd gen. intelligent control

Communication with preceding vehicle

2nd gen. adaptive cruise control (ACC)

Cooperative CACC and CFCW CFCA systems

Collision avoidance systems (CAS, FCA)

Comm. with environment (AHS, IVHS)

Intelligent speed adapting systems (ISA, IACC)

Figure 4.1: Longitudinal speed control systems. Detection of road geometry incorporates e.g. GPS information or radar sensor data identifying the surrounding obstacles and road curvature. Estimation of the road slope is not incorporated in the road geometry as presented in this figure, as this is a commonly measured variable in today’s control systems. Road following and keep-in-lane systems are actual lateral control or warning systems. However, longitudinal control as an active safety system may be incorporated in these systems as well.

this way, potential hazardous situations due to unnecessary automatic braking are prohibited. This however, requires driver intervention in critical situations. After implementation of 1st generation systems by several OEMs, research to 2nd generation systems has grown exponentially in recent years. Higher levels of deceleration and acceleration are allowed, enabling the development of Collision Avoidance Systems (CAS) instead of collision warning systems. The driver has to intervene and take-over control in less situations, providing a further degree of automation (Rohr et al. 2000, Anouck Renée Girard & Hedrick 2001). Furthermore, actual implementation of IVHS systems and cooperative control systems including inter-vehicle communication, may become feasible in near future. Extensive research is performed to e.g. platooning of HDVs. Platooning requires inter-vehicle communication, which restricts widespread implementation for passenger vehicles to this point, but may be of interest for fleets of HDVs (Lu et al. 2004, Gehring & Fritz 1997).

Implementation of active safety systems as ACC requires integration of several subsystems, demanding thorough cooperation of OEMs and FTSs. An increasing amount of possibilities to tune the longitudinal vehicle dynamics is provided. However, often attention focuses on the development of new functionality, whereas optimal tuning and low-complexity controller design are regarded as issues of minor importance. Hence, tuning requires more and more time, limiting fast implementation (Amberkar et al. 2000, Rohr et al. 2000, Liubakka et al. 1993).
4.1. LONGITUDINAL CONTROL (LOC)

4.1.1 ACC: integrated longitudinal control

ACC systems encompass control of the engine, the automatic transmission and the braking system. Hence, ACC in an example of an integrated control system as defined in the introduction of this study (see Chapter 1). Commonly, ACC works in a wide range of velocities, making it necessary to change gears during ACC operation. In vehicles with manual transmissions, acceleration and deceleration capabilities are limited by the current gear, or if the clutch is disengaged, the ACC system may either temporarily or permanently be switched off. Hence, ACC is preferably combined with an automatic transmission.

As today’s ACC systems are often additional features for luxurious vehicles, a modular design of the system is preferable. However, ACC systems in general have distributed functionality in different Electronic Control Units (ECUs). I.e. the conventional Cruise Control functionality, which is often incorporated in the engine control unit, in general is also used by the ACC system. Generally, the actual ACC system incorporating the radar and the ACC specific algorithms are embedded in a specific ACC ECU. This module is connected via the CAN databus to the rest of the vehicle, including the interface to the driver, see Figure 4.2 (Winner et al. 1996).

![Figure 4.2: Schematic representation of the integrated ACC system, with $u_i$ a combination of the actuator control signals to the throttle $u_{th}$, to the brake system $u_b$ and to the gearshift actuator of the AMT $u_g$.](image)

Commonly, ACC systems are divided into a general part, the upper level controller, determining the behaviour of the ACC system, and a vehicle-specific lower level controller, controlling the longitudinal vehicle dynamics. This enables combination of the general part with different types of powertrains. Hence, a large variety of ACC systems is possible without the need to redesign the complete controller.

This study provides an overview of today’s ACC systems, focusing on the structural design of the system. I.e. which control parts can be identified, what are the objectives of each control part and which approaches are used to design the various parts. In combination with a following study focusing on modern control solutions for optimal, low-effort tuning and low-complexity controller design, this may provide directions for robust design and optimal tuning of an ACC system.

4.1.2 Outline

To start with, an introduction to the ACC system is given, discussing its potential and the corresponding research areas (Section 4.2). Next, the general structure of an ACC system is discussed in Section 4.3. The design of the two main parts, i.e. the so-called outer or upper level and inner or lower level control loops, are discussed in more detail in Sections 4.4 and 4.5. Finally, conclusions are drawn and a case study is formulated concerning the (re)design of an ACC system (Section 4.6).
4.2 Introduction to ACC

Generally, Adaptive Cruise Control (ACC) stands for the automatic adaptation of a vehicle’s longitudinal speed $v_h$ to the speed of a predecessor $v_t$, keeping a predefined distance, the so-called desired distance headway $d^d = d^d_h$ between the vehicles. If no predecessor is present, a cruise control function comparable to conventional Cruise Control (CC) is applied. The ACC-equipped following vehicle is called the host vehicle. The predecessor vehicle is called the target vehicle (see Figure 4.3).

Figure 4.3: Representation of the main variables of an ACC system. The speed and acceleration are represented by $v$ and $a$ respectively. The subscripts $h$ and $t$ indicate the following host vehicle and the target vehicle. The relative velocity equals $\Delta v = v_t - v_h$. Often the subscript $h$ is omitted. $s^d = d^d_h$ represents the desired distance between the two vehicles, the so-called desired distance headway. $s = d_h$ represents the actual vehicle separation. The minimum distance between the two vehicles is equals $s_o$.

ACC is an enhancement of conventional Cruise Control (CC) systems. Nonadaptive cruise control is designed to maintain a vehicle speed, which is set by the driver. The driver is required to apply the brakes when a slower preceding vehicle is encountered and has to change the set speed manually if desired. Adaptive cruise control adjusts the speed of the host vehicle automatically to safely follow a target vehicle with a possibly varying speed. ACC makes use of a sensor detecting the presence of a predecessor and measuring the relative distance $d_h$ and the relative speed $\Delta v$ between the two vehicles. Hence, concerning the situation of following a predecessor, CC can be regarded as open-loop control, whereas ACC proposes a closed-loop solution.

Adapting the speed of a vehicle to the demands of the traffic flow is assumed to improve safety, comfort and efficiency. Furthermore, road capacity will increase as a result of speed harmonization. Smoother driving will also reduce fuel consumption and emissions (Vahidi & Eskandarian 2003, Germann & Isermann 1995). Hence, much research is performed to the various possibilities the application of ACC systems provides. ACC comprises the automatic adaptation of a vehicle’s speed. ACC thus can be seen as an active, intelligent and driver assisting system autonomously adjusting a vehicle’s speed. As a result, the acronym ACC is interpreted in many ways: Active, Assisting, Adaptive, Automatic, Autonomous or Intelligent Cruise Control are found in literature, each focusing on a specific aspect of the ACC system. Next, an overview of the most important aspects of the implementation of ACC systems and of the currently available systems is given.

Traffic safety, traffic flow and comfort

Much research has been performed to the impact of application of ACC systems on the driver’s behaviour. ACC is designed to reduce driver’s workload. On the one hand greater concentration of the driver on other driving tasks is allowed, thus increasing safety. On the other hand however, drivers may become lazier and thus less alert, decreasing safety (Lu et al. 2004).

Hence, driving simulators in a virtual environment rather than under real highway traffic conditions are developed to be able to test driver’s behaviour in specific situations. A variety of driving simulators specifically targeting at evaluation of ACC systems has been built. The VEHIL
of TNO, the Renault simulator and the Fokker Control Systems (FCS) are the current state-of-the-art simulators (Güvenç & Kural 2006). However, practical implementation tests provide the most reliable results.

Besides increasing traffic safety, ACC is often hypothesized to improve traffic capacity and efficiency. However, it is not yet sure if it really improves traffic capacity as drivers’ behaviour changes with implementation of ACC as well. Besides that, simulation studies to quantify the change in traffic flow characteristics that ACC brings are difficult to compare. Different driver and ACC models are used. Hence, much research to the impact of ACC equipped vehicles on a mixed traffic is performed. Modeling attempts utilizing neural networks and fuzzy logic seem to give most satisfying results (Marsden et al. 2001, Chakroborty & Kikuchi 1999, Xu & Sengupta 2003).

Furthermore, general traffic flow characteristics vary significantly over the world. I.e. tests in the US showed the usefulness of the ACC system even without active brake control, whereas in Europe active braking is definitely required in application of ACC systems (Winner et al. 1996).

Field tests show that tracing with an ACC system may provide considerable reduction in the variation of acceleration compared to manual following. This indicates a potential comfort gain for the driver as well as environmental benefits (Marsden et al. 2001, Gehring & Fritz 1997).

**Platooning traffic and string stability**

Platooning, which is a future variation on today’s ACC systems, will influence traffic safety and traffic flow significantly. Platooning can be regarded as ACC with minimal inter-vehicle distances and with several vehicles. This may be especially effective for groups of HDVs, traveling long distances with the same speed. Driver’s workload and fatigue may be reduced greatly and accidents may be avoided. Furthermore, following HDVs can reduce aerodynamic dragging force of the follower vehicles by reducing the inter-vehicle distance and thus a better fuel economy can be achieved (Lu et al. 2003), (2003, 2004).

In fact, platooning is a today’s continuation of the AHS and IVHS research ideas developed before the 1980s. After an initial head start of the Japanese automotive industry, the US followed with e.g. the PATH program, which is an extensive today’s research program focussing on AHS and IVHS systems and cooperative control of all vehicles on a highway (Tsugawa 1999, Swaroop 1997). Much research has been performed to the impact of platooning on the traffic flow density (Rajamani & Zhu 2002, Lu et al. 2004).

A control problem arising with implementation of the platooning functionality is the demand for string stability. A platoon of vehicles is said to be string stable if, under no other excitations, the range errors decrease as they propagate along the vehicle stream. I.e. if the transfer function from the range error of a vehicle to that of his following vehicle has a magnitude less than or equal to 1, it is string stable (Swaroop 1997), (1999), (2001). (Sheikholeslam & Desoer 1990) show that for a platoon of vehicles with a constant inter-vehicle spacing, cooperative control, i.e. vehicle-to-vehicle communication, is a necessary condition. This is a practical limitation preventing implementation of platooning functionality yet. However, practical tests are already performed and evaluated (Gehring & Fritz 1997).

This study focusses on so-called 1st-generation ACC systems, specifically focusing on the controller structure. Today, various OEMs have implemented ACC systems in their vehicles already, which may be used as examples (see Section 4.2.1). A general lack of defining appropriate control objectives, converting driver’s general behaviour into deterministic ACC behaviour, has to be resolved. Hence, after a short overview of the state-of-the-art implementation examples of today’s ACC systems, the general structure of today’s ACC systems is discussed in detail.

**4.2.1 State-of-the-art ACC systems**

Several OEMs have implemented ACC systems in their vehicles already. Daimler Chrysler’s Mercedes-Benz was the first one to introduce their adaptive cruise control system in October 1998 in their S-Class vehicles. Toyota Lexus was the second one to put their so-called Dynamic Laser Cruise Control (CLCC) system into the market. Jaguar launched their system in 1999 and
have implemented it in several vehicles today. BMW introduced their active cruise control system 7-Series in 2000 and from then on implemented it in their 3 and 5-Series as well. Other OEMs equipping their most luxurious vehicles with adaptive cruise control systems are Volkswagen, Audi, General Motors Cadillac, Mitsubishi and Nissan (called intelligent cruise control), Toyota and the 2007 Volvo S80 will have one as well (Prestl et al. 2000, Anouck Renée Girard & Hedrick 2001, Xilinx 2006).
4.3 General structure of ACC

The structure of an ACC system may vary largely. However, in general the structure can be divided into three generic parts (see Figure 4.4). The first one includes the sensor and the corresponding situation assessment, resulting in the identification of a target vehicle. Secondly, the outer or master loop determines the actual behaviour of the ACC system. And thirdly, the inner or slave loop comprising the actual longitudinal controller of the vehicle, determines the vehicle response and controls the appropriate vehicle actuators. Decoupling the engine and brake control tasks from the overall vehicle control task, i.e. speed tracking or vehicle following, enables a generic design of the outer control loop applicable to any type of vehicle with its vehicle specific inner loop controller. The outer loop and inner loop are coupled via switching logic. Considering the terminology of integrated control as proposed in the Introduction (Section 1), the outer and inner control loop represent the upper level and lower level controllers respectively (Prestl et al. 2000, Widmann et al. 2000, Gerdes & Hedrick 1997, Persson et al. 1999, Lu & Hedrick 2003).

4.3.1 Sensor, target identification

The environment detecting sensor in combination with the target identification are the basic key components of ACC systems. Hence, much research to sensor selection and processing of the corresponding data is performed. In this study however, focus lies on the actual controller design rather than selection of the best sensors and identification algorithms. Consequently, only a short
overview of the key aspects is given in this section.

The sensor is used to detect the environment of the vehicle (connection 3 in Figure 4.4). The main sensor requirements are 1) the information required by the ACC upper outer control loop, i.e. the relative position and the relative speed of preceding vehicles, 2) the lateral position or the angle relative to the host vehicle’s longitudinal axis of the preceding vehicles to be able to select the relevant target vehicle and 3) robustness against bad weather conditions, e.g. fog, rain, snow, etc.

Requirements for the sensor are based on the control objectives of the ACC system. I.e. angle resolution, angle accuracy and sensor range determine the minimum allowable clearance between two successive vehicles. Consequently, several commercially available products are available today and much research into new products is performed. I.e. radar, laser, optical vision, ultrasonic or infrared systems with or without Doppler range-rate information, with varying angle resolution, angle accuracy, sensor range, etc. (Widmann et al. 2000).

Commonly, today’s ACC systems use radar-based systems. Future’s enhanced ACC functionality however, requires new techniques. Combination of systems, i.e. multi-sensor fusion, is one of today’s area’s of research. ACC stop-&-go for example, requires information about object extents close to the vehicle as well as ‘general’ ACC information about preceding vehicles (Venhovens et al. 2000, Dellaert et al. 1998).

Identification of a target vehicle based on the sensor data, is the rather complicated following step. This process can be divided into 3 main parts.

1. Situation assessment: objects in the range of the sensor(s) are determined and the yaw rate, acceleration and speed data of the host vehicle (connection 6 in Figure 4.4) are processed to determine the current situation. This differs considerably per sensor type and may be enhanced by sensor and data fusion.

2. Predict future state: based on the situation assessment results and possibly other IVHS, AHS systems, the roadway curvature is estimated and the future path of the host vehicle and the objects in the sensor range are determined. Depending on the situation, this is regarded as a difficult function. E.g. distinguishing between changing lanes and cornering of the target vehicle. In the latter case no action has to be taken, while in the first case the ACC system switches from FOC to SSC mode (Miyahara 2003).

3. Threat assessment: finally, objects in the same lane as the host vehicle, with lower velocity and within the maximum range, are determined. From these objects a target vehicle is selected and its relative distance and relative velocity with respect to the host vehicle are determined (connection 7 in Figure 4.4). If no threat is present, no target vehicle is determined and conventional cruise control will be applied by the ACC system.

4.3.2 Outer, master control loop

The outer or master control loop determines setpoints for the acceleration and deceleration of the vehicle. Based on the outcome of the target vehicle identification and the driver’s settings, an appropriate ACC control mode is chosen and the corresponding control objectives are determined. E.g. when faster vehicles cut-in in the lane in front, ACC has to ignore the abruptly decreased distance to his predecessor. However, in case slower vehicles cut-in in the lane in front, ACC has to prescribe a deceleration. The target vehicle identification provides data about the relative velocity and distance between the host vehicle and the target vehicle (connection 7 in Figure 4.4). A desired acceleration or deceleration $a_d$ is determined by the chosen control mode (connection 8 in Figure 4.4).

In determining the desired acceleration or deceleration of the vehicle, the outer control loop determines the behaviour of the ACC system. The control objectives and control modes should
approach a driver’s behaviour as closely as possible to gain acceptance and thrust of the driver. The outer control loop preferably is independent of the host vehicle’s configuration. Actual longitudinal control of the vehicle is performed by the inner control loop and information about the host vehicle’s current state is incorporated in the target vehicle identification data. In this way, a generic outer control loop can be combined with a vehicle specific inner control loop.

The design of the outer control loop is discussed in more detail in Section 4.4.

### 4.3.3 Inner, slave control loop

The inner or slave control loop comprises the actual longitudinal control of the vehicle and thus is vehicle (configuration) specific. Generally, the inner control loop consists of two separate parts. Firstly, arbitration functionality determines whether brake control or throttle control is required. Accordingly, the desired acceleration $a^d$ is limited and a set acceleration $a^{set}$ for the longitudinal controller is determined. Secondly, taking into account the current acceleration or deceleration data of the vehicle (connection 6 in Figure 4.4), a longitudinal vehicle controller is used to determine a control signal $u_i$ for the corresponding actuator, i.e. the throttle actuator $u_i = u_{th}$, the brake actuator $u_i = u_b$ or the actuator for automatic gear changing $u_i = u_g$. The inner control loop thus has to be able to cope with changing operating conditions of the host vehicle via e.g. online parameter estimation. The connections are made via the CAN-bus of the vehicle.

Generally, the engine is seen as the torque actuator. A setpoint signal is sent to the throttle / injection actuator of the engine $\alpha^{set}_{th}$, resulting in an engine torque $T_e$. Accordingly, a setpoint signal is sent to the gear shifting actuator $i^{set}_g$. Specifically in case of HDVs, the brake system may consist of several systems. Accordingly, a vehicle-specific controller has to be designed. Kampung et al. (2005) describe the design of such a controller, specifically aiming at implementation on a HDV. The design of the inner control loop is discussed in more detail in Section 4.5.

### 4.3.4 Driver and general settings

The ACC interface for the driver is an important aspect of the ACC system (connection 1 in Figure 4.4). As discussed in Section 4.3.2, the ACC system is designed to act as natural as possible in the sense of driver’s behaviour. As driver’s behaviour is very subjective, it is important that the ACC system provides clear information feedback about its operating conditions to the driver. In this way, the driver is able to supervise the system and intervene when necessary. Hence, an important aspect of research describing actual implementation of ACC systems in commercial vehicles describes the interface to the driver (e.g. Prestl et al. 2000). Generally, the ACC interface comprises information about the value of the target speed and desired time headway, the actual acceleration or deceleration values, the control mode the ACC system is using, i.e. ‘common’ cruise control or target vehicle following, and an emergency signal, often acoustic in combination with visual warning signs (Winner et al. 1996).

Furthermore, based on the present behaviour of the vehicle and the environment (connections 4 and 5), the driver supervises the ACC system and has to be able to intervene in all situations (connection 2 in Figure 4.4). Generally, the driver commands manual on / off functionality, manual setting of the target speed and the desired time headway and intervention of the ACC system by application of the brakes or the gas pedal. Generally, a desired time distance between the vehicles, the so-called target time headway of 1.0 to 1.5 s is reasonable and the driver is able to command the system in discrete steps, e.g. 1.0, 1.2 or 1.6 s (Prestl et al. 2000). The minimum time headway is limited by the maximum allowable braking deceleration value, the sensor accuracy and the speed of data processing.

With respect to the maximum deceleration and acceleration values allowed, a clear distinction is present between 1st generation and 2nd generation ACC systems. 1st generation systems comprise moderate deceleration and acceleration values. Generally, a maximum deceleration of about 2.5 m s$^{-2}$ and 1.0 m s$^{-3}$ for the maximum deceleration rate of change (this compares to 0.1 g per second change) are allowed. Hence, no emergency braking is possible. However, the driver is never surprised and never gets overstrained. For the same reasons and out of comfort reasons,
the maximum acceleration is limited to about 1.0 m s\(^{-2}\). 2\(^{nd}\) generation systems, including ACC stop-&-go and FCA systems will comprise emergency braking. Hence, deceleration values up to 5 m s\(^{-2}\) are required (Winner et al. 1996).
4.4 Outer, master control loop

As mentioned in Section 4.3.2, the outer or master control loop of a general ACC system consists of two main parts. The first one encompasses the selection of an appropriate control mode and determination of the corresponding controller settings or control objectives. The second part comprises the actual control functionality of the ACC system, e.g. cruise control or target vehicle following. Figure 4.5 shows a general representation of this setup of the outer control loop. In the following sections, first the various ACC control modes are discussed. Next, the arbitration between the modes and finally the filtering part mimicking driver’s behaviour are discussed. In literature, actual tuning of the designed controllers and filters often is said to be done via trial-and-error. Besides several adaptation mechanisms to determine varying vehicle parameters, little to nothing can be found about automatic tuning.

4.4.1 ACC control modes

The ACC control modes incorporate two general modes, i.e. conventional cruise control and target vehicle following. Besides that, several additional, optional control modes, i.e. Stop-&-Go control and curve speed control may be included. 1th generation ACC systems only allows moderate acceleration and deceleration of the vehicle, whereas 2nd generation systems allow increased acceleration and deceleration commands. Consequently, 2nd generation systems will be suitable for use in suburban areas, where the relative speed and the relative distance are relatively small and additional functionality as stop-&-go control can be implemented.

Set Speed Control (SSC)
The Set Speed Controller (SSC) is a standard mode of the ACC system functionality. SSC is based on conventional Cruise Control (CC). However, longitudinal control of an ACC system encompasses more possibilities than conventional CC as ACC is not restricted to the throttle actuator only. In this way, a certain desired speed \( v_d \) can still be kept when driving downhill for example.

Based on a driver’s setting for a desired set speed \( v_d \) and the measured actual speed of the vehicle \( v_h \), SSC determines an appropriate acceleration or deceleration \( a_d \) to control the vehicle speed. The inner loop controller uses the throttle, brake and automatic transmission actuators to actually control the vehicle.

In order to control the vehicle as smooth as possible from a driver’s perspective, the resulting setpoint acceleration or deceleration should be as human-alike as possible. This is bounded by several general constraints.

- bounding of the acceleration: \( a_{\text{min}}^+ \leq a^+ \leq a_{\text{max}}^+ \)
- bounding of the deceleration: \( a_{\text{min}}^- \leq a^- \leq a_{\text{max}}^- \)
• minimization of jerks for driver comfort: $|\dot{a}|$ should be kept small

General rules for the design of these acceleration setpoints however, are not available. Commonly, final tuning is done via trial-and-error. This however, may result in a very subjective controller behaviour.

**Following Control (FOC)**

Following Control (FOC) encompasses the main functionality of an ACC system. When a target vehicle is detected, the host vehicle adapts its speed automatically to the speed of this preceding vehicle. Commonly, several scenarios are distinguished, determining the settings for the controller (see Section 4.4.2).

The control objective of FOC is twofold. Firstly, the host vehicle should adapt its speed $v_h$ to the speed of the target vehicle $v_t$. This is limited by a maximum speed $v_{h\text{max}}$, set by the driver. Secondly, the relative distance or headway $d_h$ between the vehicles should be adapted to a desired headway $d_{d\text{h}}$, also set by the driver. Commonly, the driver has to set a desired time headway $t_{d\text{h}}$, which than is converted to a desired distance headway by the ACC system. This is done by controlling various actuators, i.e. the throttle actuator, the brake system actuator and the gearshift actuator (Prestl et al. 2000).

Analogous to the SSC design, tuning is generally done via trial-and-error and no generic rules or objective evaluation variables are defined yet. Only some general constraints are defined: the controller is constraint by bounded acceleration and deceleration values $a_{\text{min}} \leq a \leq a_{\text{max}}$ and for driver comfort, jerks should be minimized, i.e. $|\dot{a}|$ should be kept small.

Examples of objective evaluation variables for example may be

- the distance divergence, which is the ratio of the actual distance headway to the desired distance headway, $\rho_d = d_h/d_{d\text{h}}$. Or minimization of the error in desired relative distance between the target and host vehicles: $\epsilon_{d_h} = d_h - d_{d\text{h}}$. For safety, $\epsilon_{d_h}$ should be forced positive.

- the relative speed of the driver’s vehicle to the target vehicle $\epsilon_v = v_t - v$ should diminish (asymptotically) to 0. I.e. $\epsilon_v, \epsilon_{d_h} \rightarrow 0$ for $t \rightarrow \infty$.

Furthermore, today’s ACC systems are commonly designed to handle a fixed desired time headway $t_{d\text{h}}$, which can be set by the driver. This leads to a desired distance headway $d_{d\text{h}} = d_h + t_{d\text{h}} v(t)$. However, research to ACC incorporating a variable time headway shows promising results. Based on a minimum relative deceleration for example, a safe time headway and corresponding distance headway can be calculated automatically (Kamaris & Ioannou 1996). Other algorithms are proposed by (e.g. Yanakiev et al. 1998, Swaroop & Huandra 1999).

**ACC Stop-&-Go control**

To improve driver comfort, ACC stop-&-go systems offering vehicle guidance at low speeds all the way down to zero velocity, e.g. in suburban areas or congested traffic on highways, are developed. Stop-&-go control is an additional ACC mode enabling the vehicle to actually come to a halt and start driving again. Several general properties of a traffic situation for which ACC stop-&-go is intended are 1) a limited maximum velocity, i.e. about 30 km h$^{-1}$, 2) frequent stops followed by intermittent starts and 3) all lanes occupied by vehicles and no possibility to choose the speed of driving. The diagram depicted in Figure 4.6 shows the range of application of ACC stop-&-go (Venhovens et al. 2000).

Generally, ACC is intended for highway use only and a minimum speed limit of 30 to 40 km h$^{-1}$ is imposed on it. Low speed accelerations, which may cause dangerous situations when for example a nearby pedestrian is not detected, are avoided by this minimum speed limit. Consequently, ACC system and sensor development has mainly focussed on following control at highway speeds and detection of preceding vehicles with a moderate relative distance (> 10 m).
ACC stop-&-go requires complete coverage of a vehicle surroundings, hence application of several sensors is demanded.

Furthermore, higher deceleration and acceleration values are required for following control with lower speeds and smaller relative distances. Hence, throttle and brake authority have to be increased significantly with respect to 1st generation ACC systems. Generally, twice the deceleration and acceleration commands of 1st generation systems is taken, i.e. $-5 \text{ m s}^{-2} \leq a_d \leq 2 \text{ m s}^{-2}$ (Riley et al. 2000).

An example of a ACC stop-&-go controller design is presented by (Venhovens et al. 2000). With a desired relative distance headway

$$d^d = d_{\text{min}} + v t_h$$

with $d_{\text{min}}$ the minimum vehicle spacing at zero speed, the resulting controller becomes

$$u = k_v \Delta v + k_d \Delta d - a_z + \tau$$

with $\Delta v = v_t - v_h$ and $\Delta d = d_h - d_{\text{d}}$. The parameters $k_v$ and $k_d$ are tuning parameters, $a_z$ represents disturbance accelerations caused by e.g. road slopes, wind, uncertainties of parameters or, in case of low driving velocities, by the engine idling controller, and $\tau = v_t - k_v d_{\text{d}} - d_{\text{d}}^2$ eliminates the dependency on the target vehicle behaviour. This controller structure is suitable for various traffic situations by appropriate tuning of $k_v$ and $k_d$ per situation. For several specific situations, e.g. stopping behind a vehicle without overshoot, other control actions may be required.

**Curve Speed Control (CSC)**

Curve Speed Control (CSC) is an additional ACC mode, limiting the acceleration or the speed of the vehicle when driving on a road with a curvature. CSC functionality is twofold. On the first place, depending on the curvature of the road, detection of a target vehicle is difficult. Consequently, it is likely a previously detected target vehicle is not detected anymore although it is still present or a new target vehicle is not detected in time. Limiting the acceleration of the vehicle may thus prevent brusque behaviour when a vehicle is detected suddenly, i.e. at a small relative distance.

Secondly, upper bounding of the lateral forces $F_y$, experienced by the driver may be desired. This is also known as lateral acceleration control, providing comfortable lateral acceleration while driving through curves. The maximum value for lateral acceleration is typically in the range of 2.5 to 3.0 m s$^{-1}$. An Electronic Stability Program (ESP) system measuring the yaw rate of a
vehicle is already common in today’s vehicles. Determination of the lateral forces generally thus is fairly simply. These forces depend on the curvature of the road as well as the longitudinal velocity of the vehicle. Hence, speed limitation of the speed of the vehicle may be required (Aparicio et al. 2005, Widmann et al. 2000).

Frontal Collision Warning or Avoidance (FCW / FCA)
Frontal Collision Warning or Avoidance (FCW / FCA) systems sometimes are regarded as specific ACC functionality, but more often as separate systems. Consequently, no further attention will be paid to these systems at this point. The main objective of FCW or FCA systems is to start braking the vehicle in an early stage already, when an obstacle is detected ahead. In this way the driver or the ACC system is assisted in smoothly slowing down and maximum deceleration forces can be decreased (e.g. Tsugawa 1999).

4.4.2 Control mode arbitration and ACC behaviour settings
Commonly, several scenarios or traffic situations are distinguished to evaluate the working conditions for the ACC controller. Generally, these scenarios are linked directly to the various ACC modes of the system. On the one hand, arbitration between the ACC functionality or ACC modes (see Section 4.4.1) is demanded. On the other hand, reaction time, a setpoint for the host vehicle’s speed, maximum acceleration and deceleration values and an appropriate distance headway have to be determined per scenario. The outer control loop of the ACC system determines the behaviour of the vehicle in determining acceleration and deceleration setpoints (see Figure 4.5). Hence, this control loop should be tuned in such a way that its behaviour corresponds to human behaviour as much as possible in order not to distress the driver (Güvenç & Kural 2006).

Arbitration
Focussing on the general ACC modes, arbitration between SSC and FOC is required. When the host vehicle is driving steadily in SSC or FOC mode and a target vehicle disappears or a target vehicle with an moderate relative distance \( d_h \gg d_{h_{\text{min}}} \) and relative velocity \( v_t \approx v_h \), arbitration between the two main ACC control modes, SSC and FOC, is relatively simple. When a third vehicle cuts-in between the host and the target vehicle, direct reaction is required, however not too brusque. In case of other ACC functionality as e.g. stop-&-go control, the vehicle has to be able to come to a halt completely, without colliding with the preceding vehicle.

(Zhang & Ioannou 2004) propose a straightforward switching algorithm, utilizing SCC if no object is detected and utilizing FOC otherwise. (Persson et al. 1999) enable the control mode generating the lowest acceleration. In order to prohibit shattering between the two control functions, in general a relative-distance dependent boundary layer and a corresponding delay or reaction time are assigned to the switching rules. I.e. depending on the actual distance headway \( d_h \), the controller switches between SSC and FOC or not. (Widmann et al. 2000) propose a diagram containing switching surfaces and a boundary layer to switch between FOC and SSC assuring smooth switching behaviour without shattering between the two controllers.

Behaviour settings
In transient traffic situations, the relative velocity or the relative distance between the host and target vehicle may become significant. In general, filtering of these variables is required in transient situations to prohibit brusque reaction of the ACC system and to react as human-alike as possible to the present situation. Widmann et al. (2000) and Yansakiev et al. (1998) propose a nonlinear gain \( F_{\varepsilon_v} \) to filter the relative velocity \( \Delta v \). Zhang & Ioannou (2004) and Ioannou & Xu (1994) propose a nonlinear filter to smoothen the velocity trajectory of the vehicle. The filtered speed trajectory \( \hat{\varepsilon} \) is then used by the FOC. In Figure 4.7 the nonlinear filter used by Zhang & Ioannou and Ioannou & Xu is shown. The function \( f(z, \ldots) \) may represent a nonlinear function (Zhang & Ioannou 2004) or may be omitted (Ioannou & Xu 1994).

Widmann et al. (2000) propose a linear ‘gain scheduling’ controller providing robustness and stability in designing a controller for the FOC mode. Using several nonlinear filters, they establish
4.4. OUTER, MASTER CONTROL LOOP

Figure 4.7: Nonlinear filter to smooth the velocity setpoint, as used by Zhang & Ioannou (2004).

system behaviour closely mimicking actual human driving behaviour: 1) a filter to scale $t^d_h$ in order to be able to respond to large speed differences adequately, 2) a feedforward filter to be able to respond to a cut-in manoeuvre of a target vehicle, 3) a filter to limit jerks and 4) a filter to limit speed-up times. To be able to quickly respond in transient situations, the gain scheduling controller is able to exhibit various levels of under-damped behaviour in case of transients. In the case of steady-state following, the controller exhibits critically-damped behaviour, which assures suppression of target vehicle speed-up (or slow-down) oscillatory behaviour. Standard linear pole-placement techniques are used to design the various parts of the gain-scheduling controller.

Fritz (1999) designs a FOC, targeting at minimization of the relative distance between the target and the host vehicle. In fact, a 2nd-generation ACC system is designed. Instead of designing a cooperative controller, they double the allowed deceleration values, i.e. values up till $5 \text{ m s}^{-2}$ instead of $2.5 \text{ m s}^{-2}$. Specific attention is paid to feedback linearization of the inner loop controller, resulting in a linear model as a basis for the outer loop FOC controller. The acceleration as well as the jerk of the target vehicle are regarded as disturbances. Experimental tests with a Mercedes-Benz truck show that an additional feedforward loop is not needed. Online pole placement is used, taking into account the play in the driveline.

(Germann & Isermann 1995) use a neural network to learn the subjective driver demands, which are then implemented in a fuzzy controller to formulate the comfort requests. Separate fuzzy controllers for FOC and SSC are developed. Stop-&-go functionality is implemented as well. No experimental tests are executed, but simulation results show satisfactorily working.

Persson et al. (1999) design a controller based on intuitive reasoning. They focus specifically on driver comfort, which is incorporated in the inner loop longitudinal vehicle controller. A FOC is designed based on a tunable gain $Q$ representing the danger of a situation and a factor $F$ to make the deceleration fairly constant, which is assumed to be comparable to driver’s behaviour. Based on several scenarios, tuning is done by trial-and-error. The resulting controller is complex and incorporates a lot of parameters.
4.5 Inner, slave control loop

As mentioned in Section 4.3.3, the inner or slave control loop of a general ACC system consists of two main parts. The first one encompasses arbitration between throttle or brake actuation, depending on the desired vehicle acceleration. The second part comprises the actual longitudinal vehicle control, controlling the throttle actuator, brake system actuator and the gear shifting actuator. Furthermore, limiting of the acceleration or deceleration commands from the outer control loop may be required, see Figure 4.8. In the next sections, both parts are discussed in more detail.

![Figure 4.8: Schematic representation of a general inner control loop of an ACC system.](image)

### 4.5.1 Throttle, brake actuator arbitration

Unnecessary switching between throttle and brake control has to be avoided. Hence, switching rules for the actuation of the throttle and the brake actuator have to be designed. Furthermore, switching rules for the usage of friction brakes in combination with engine compression brakes have to be designed analogously.

A general set of switching rules for the actuator control is presented by Zhang & Ioannou (2004) for example:

**In set speed control mode**:

- If $u > 0$, where $u$ is the control output of one of the control functions FOC, SSC, etc., the throttle actuator is controlled.

- If $u_0 \leq u \leq 0$, where $u_0 > 0$, neither the throttle actuator nor the brake system is actuated; the engine is operated as in idle speed (coasting).

- If $u < -u_0$, the brake system is actuated.

**In following control mode**:

- If $d_h > d_h^{max}$, where $x^{max} > 0$, the throttle actuator is controlled.

- If $d_h < d_h^{min}$, where $x^{min} > 0$, the brake system is controlled.
• If \( d_{\text{min}} \leq d_h \leq d_{\text{max}} \):
  - If \( u > 0 \), the throttle actuator is controlled.
  - If \( u_0 \leq u \leq 0 \), neither the throttle actuator nor the brake system is actuated; the engine is operated as in idle speed (coasting).
  - If \( u < -u_0 \), the brake system is actuated.

Furthermore, switching strategies are often based on a model of the longitudinal vehicle dynamics (e.g. Anouck Renée Girard & Hedrick 2001). Occasionally, arbitration is based upon the output of the longitudinal vehicle control, inverting the arbitration and control part (e.g. Fritz 1999).

Besides the traditional friction brakes, today’s brake systems of HDVs incorporate engine compression brakes. The latter ones often comprise delays due to the use of pneumatic or hydraulic subsystems. Furthermore, due to wear and overheating, the steady-state behaviour as well as the reliability may deteriorate. Hence, the compression brake system is preferable used in steady downhill and non-critical braking maneuvers. Analogous to the before presented actuator arbitration, switching rules for the braking system have to be designed as well. Druzhinina & Stefanopoulou (2002) propose a coordination scheme between friction and engine compression brakes. The objective is reduction of friction brake usage. For several braking maneuvers successful validation tests are performed on a class 8 Freightliner truck. In (2002) they refine the scheme taking into account the distance constraints from other vehicles in traffic, gear ratio adjustments and road grade.

4.5.2 Actuator control

As discussed in Section 4.3, the actuator control encompasses the actual longitudinal vehicle control. The controller comprises the control of the x-by-wire systems of the powertrain, i.e. the throttle actuator, the brake (system) actuator and the gearshifting actuator of the AMT. Given the desired vehicle acceleration or deceleration, the state equations describing the dynamics of the vehicle are used to determine the appropriate brake and throttle torque. The behaviour of the ACC system is determined by the outer control loop, prescribing the desired acceleration of the vehicle. The longitudinal vehicle dynamics are controlled by the inner control loop (Persson et al. 1999).

Generally, a model of the longitudinal vehicle dynamics is designed, which is simplified via standard linearization techniques or using feedback linearization techniques. Based on the state equations describing the (simplified) vehicle dynamics, separate brake and throttle actuator controllers are then designed using traditional feedback and feedforward techniques (e.g. Lu & Hedrick 2005, Gerdes & Hedrick 1997, Fritz 1999). Furthermore, often an adaptation mechanism, determining varying parameters which influence the longitudinal vehicle dynamics, i.e. vehicle mass, road slope, etc., is used (e.g. Germann & Isermann 1995, Druzhinina et al. 2003, Bae & Gerdes 2003).

The longitudinal vehicle dynamics are vehicle specific. Hence, the inner loop controller, which is generally based on a model of these dynamics, is vehicle specific as well. In this way vehicle specific integration issues can be dealt with. E.g. Riley et al. (2000) discuss the integration of ACC inner loop brake commands with existing ABS (Anti-lock Brake System), TCS (Traction Control System) and VSE (Vehicle Stability Enhancement) systems.

(Anouck Renée Girard & Hedrick 2001) discuss implementation of cooperative control, which is a second generation adaptive control (see Figure 4.1). In case of throttle actuator control, the desired control torque \( T_{\text{dc}} \) is defined as

\[
T_{\text{dc}} = a^d_h + i_g (M_{rr} + h F_d + mgh \sin \theta)
\]

with \( a^d_h \) the desired acceleration of the follower vehicle, \( i_g \) the gear ratio, \( M_{rr} \) the rolling resistance moment, \( F_d \) the aerodynamic drag force and \( h \) the wheel radius. \( T_{\text{dc}} \) is related to the desired
throttle angle via an engine map. In case of brake actuator control, the desired torque is defined as

\[ T_{d} = -\frac{a_{n}^{d} + T_{ed}}{i_{g}} - M_{rr} - hF_{d} - mgh \sin \theta \]  

(4.4)

Now \( T_{d}^{b} \) is used to control the brake system.
4.6 Conclusions

Adaptive Cruise Control (ACC), encompassing control of the engine, the transmission and the brake system, is an example of an integrated control system as defined in the introduction (Chapter 1). Section 4.2 shows that the design of an ACC system can be regarded from many different points of view. Accordingly, much research is performed to these various aspects of the implementation of ACC systems, e.g. ACC as an active safety system, the resulting traffic flow characteristics, the possibilities for platooning, the impact on the driver behaviour, etc. This leads to different system design concepts.

In Section 4.3, a general structure for an ACC system is identified. The main parts comprise 1) the sensor and the corresponding sensor data processing, 2) the outer, master control loop and 3) the inner, slave control loop. The corresponding key objectives are 1) sensor range and reliability, 2) determination of the situation specific ACC system behaviour, which should be as human-like as possible and 3) the longitudinal control of vehicle specific dynamics.

Focus of this study lies on the outer, master control loop (Section 4.4). Commonly, the outer and inner control loop are considered separately. When focussing on the outer, master control loop, often little or no attention is paid to the inner control loop and the actual longitudinal control of the vehicle dynamics. However, the outer loop controller determining a desired acceleration or deceleration, is limited by the inner loop dynamics. Considering HDVs for example, the vehicle dynamics may be a limiting factor, whereas this may not be the case for luxurious passenger cars. Furthermore, today’s 1st generation ACC systems only allow moderate acceleration and deceleration values yet. This problem thus may manifest oneself with future 2nd generation systems.

Identification of all possible situations and translation of these situation and of human behaviour into general controller objectives and limitations, constitutes the main challenge of the outer control loop. Generally, the basis of an ACC system is rather simple. However, due to the situation or state-dependency of the controller settings, the final tuning of the system becomes complicated and time consuming. A general approach is lacking yet and tuning and controller design are generally based on trial-and-error.

Based on these conclusions, a case study is defined in cooperation with TNO Science and Industry, Business Unit Automotive, Helmond, The Netherlands, concerning the (re)design of an ACC Stop-&-Go system. TNO has developed several ACC systems with different purposes already, i.e. for passenger cars as well as for trucks and ‘standard’ systems as well as ACC Stop-&-Go systems.

To start with, explicit definition of the goals and control objectives concerning e.g. comfort, safety, legislation, etc., is required. Based hereon, a new concept for the design of an ACC Stop-&-Go system has to be developed, focussing on a low-complexity and robust controller design. In addition, further research to automated tuning techniques has to provide solutions for optimal tuning with minimal effort, which will be applied to this new concept.
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## Appendix A

### Network protocols in the automotive industry

The table in this Appendix provides an overview of the network protocols used in the automotive industry. A distinction is made between

1. network protocols that are developed specifically for certain automotive applications and that are widely adopted by the automotive industry nowadays;

2. upcoming protocols that are developed specifically for certain automotive applications;

3. protocols from other markets that are upcoming in the automotive industry;

4. some further 'smaller' protocols that are less widespread in the automotive industry;

<table>
<thead>
<tr>
<th>Network protocol</th>
<th>Year (dvlpmnt./implem.)</th>
<th>Development aim</th>
<th>Application field</th>
<th>Baud rate</th>
<th>Specifications</th>
<th>Developers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Communication Area Network (CAN) Byteflight</td>
<td>1985</td>
<td>first standardized fieldbus in the automotive industry to reduce vehicle wiring</td>
<td>low as well as high speed networks</td>
<td>1 MB s(^{-1})</td>
<td>message-oriented</td>
<td>Robert Bosch GmbH.</td>
</tr>
<tr>
<td></td>
<td>1996 / 2000</td>
<td>high performance network with respect to CAN</td>
<td>safety-related applications</td>
<td>10 MB s(^{-1})</td>
<td>integrated in FlexRay</td>
<td>BMW, ELMOS, Infineon, Motorola, Siemens and Tyco EC</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Audi, BMW, Daimler-Chrysler, Motorola, Volcano, Volvo and Volkswagen</td>
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<tr>
<td></td>
<td></td>
<td>link relatively low-speed networks to higher-speed networks like CAN</td>
<td>body and comfort subsystems</td>
<td>20 kB s(^{-1})</td>
<td>providing a master-slave protocol, i.e. unable to hack from the outside</td>
<td>More than 60 firm, including Ford, BMW, Daimler-Chrysler, General Motors</td>
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<tr>
<td>Local Interconnect Network (LIN)</td>
<td>1998 / 2001</td>
<td>interconnection of automotive multimedia and infotainment systems</td>
<td>multimedia and infotainment</td>
<td>25 MB s(^{-1})</td>
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**Upcoming protocols**

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<tr>
<th>Network protocol</th>
<th>Year (dvlpmnt./implem.)</th>
<th>Development aim</th>
<th>Application field</th>
<th>Baud rate</th>
<th>Specifications</th>
<th>Developers</th>
</tr>
</thead>
<tbody>
<tr>
<td>FlexRay</td>
<td>1998 / 2000</td>
<td>scope for x-by-wire applications and replace multiple current networks with one new protocol</td>
<td>by-wire applications</td>
<td>10 MB s(^{-1})</td>
<td>time-triggered</td>
<td>Many firms, including BMW, Daimler-Chrysler, Philips and Motorola. European Union’s Brite-Euram X-by-wire project</td>
</tr>
<tr>
<td>Time-Triggered Protocol (TTP)</td>
<td>1994 / 1998</td>
<td>scope for x-by-wire applications, however too costly and inflexible</td>
<td>by-wire applications</td>
<td>25 MB s(^{-1})</td>
<td>distributed TTP/C as well as master-slave TTP/A, time-triggered</td>
<td></td>
</tr>
<tr>
<td>Time-Triggered CAN (TTCAN)</td>
<td>1999 / 2000</td>
<td>time-triggered layer on top of CAN networks</td>
<td>by-wire applications</td>
<td>1 MB s(^{-1})</td>
<td>time- as well as event-triggered</td>
<td>Many of the major automotive and semiconductor manufacturers</td>
</tr>
<tr>
<td>ZigBee</td>
<td>2004</td>
<td>sensor network for monitoring and control purposes (air conditioning, heating, ventilation, lighting control, etc.)</td>
<td>wireless sensor networks</td>
<td>250 kBs(^{-1})</td>
<td>wireless</td>
<td>ZigBee Alliance (over 120 company members)</td>
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<tr>
<td>Protocols from other markets, upcoming in the automotive industry</td>
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</tr>
<tr>
<td>Bluetooth</td>
<td>1999</td>
<td>originally devised for Personal Area Networks (PAN) wireless applications multimedia and infotainment</td>
<td>3 MB s⁻¹ wireless, low-range (10 to 100 m)</td>
<td>Ericsson</td>
<td></td>
<td></td>
</tr>
<tr>
<td>FireWire (IDB-1394 or iLink)</td>
<td>1998</td>
<td>replacement of SCSI, connecting PC devices inter-vehicle communications (e.g. advanced drive assistance systems, user communications and information services, etc.) wireless multimedia and infotainment and active suspension systems</td>
<td>3.2 MBs⁻¹</td>
<td>Apple</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Universal Serial Bus (USB)</td>
<td></td>
<td>originally developed for the PC market</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Wireless Fidelity (Wi-Fi)</td>
<td></td>
<td>inter-vehicle communications (e.g. advanced drive assistance systems, user communications and information services, etc.) wireless multimedia and infotainment and active suspension systems</td>
<td>(see 'Development target')</td>
<td></td>
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<tr>
<td>Ultra Wide Band (UWB)</td>
<td></td>
<td>wireless multimedia and infotainment, collision-detection and active suspension systems</td>
<td>500 MB s⁻¹ higher bandwidth compared to Wi-Fi</td>
<td></td>
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<table>
<thead>
<tr>
<th>Further 'smaller' protocols</th>
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<tr>
<td>Domestic Digital Bus (D2B)</td>
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<tr>
<td>Mobile Media Link (MML Bus)</td>
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<tr>
<td>Safe-by-Wire</td>
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<tr>
<td>Motorola Interconnect (MI)</td>
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<tr>
<td>Distributed Systems Interface (DSI)</td>
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Appendix B

Project proposal

In this Appendix, the project proposal as formulated for this project is included. At the end of November 2006, the corresponding contracts were signed by all parties, i.e. TNO Automotive, DAF Trucks N.V. and Technische Universiteit Eindhoven. The underlying literature study is part of work package 1 (wp1) as described in the proposal.
Project proposal

Project title
Robust Design and Automated Tuning of Automotive Controllers

Project number

Starting / ending date
03/2006 – 02/2010

Partners

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Ph.D.-researcher
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E-mail / Tel
g.j.l.naus@tue.nl / +31 402474092

Project costs and financing

<table>
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<th>3</th>
<th>4</th>
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<td>56000</td>
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<tr>
<td>Contribution DAF*</td>
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<td>28000</td>
<td>28000</td>
<td>28000</td>
</tr>
</tbody>
</table>

* From year 2, these numbers will be adjusted with the CBS-index
1 Aim of the project
This project aims to develop methods and algorithms for robust low-complexity controller design and algorithms and software tools for tuning of controller parameters with minimal effort, e.g. automated tuning, focussing on control systems in the field of commercial vehicles. Results are validated at DAF Trucks and TNO via case studies in the area of the driveline controllers, such as the clutch control (automated gearbox) and (adaptive) cruise control.

2 State of the art
Control systems used in the automotive industry become increasingly complex due to the rising amount of electronic and software control components. This leads to an increasing number of interactions between components (direct interaction) as well as via the dynamics of the vehicle (indirect interaction). Moreover, performance improvements and extra features are often realised by suppliers via software, which are more or less black box systems to the OEM’s (Original Equipment Manufacturers) or, at best, lead to very complicated control structures. As a result the overview and the insight into the relevance of certain parametric and structural choices are difficult to obtain and a lot of effort is needed for the tuning of the controller parameters. However, both from process control and motion control applications, methods are available that exploit on-line estimation and optimization techniques for auto-tuning of controller parameters, including calibration. Also some results are known for fixed structure control design and design for robustness. In combination with a structured low-complexity controller design this may lead to a decreasing amount of effort needed for tuning of the controller parameters. As a first example, last year a M.Sc. project was done on auto-tuning for the cruise controller of a DAF truck.

3 Industrial application and valorisation of the project
The design, integration and tuning of control systems taking more and more time is an actual problem in the automotive industry. Consequently, suppliers and OEM’s like DAF Trucks N.V. as well as applied research organisations like TNO, benefit from the reduction of the required manpower for tuning. Moreover, this will eventually lead to performance improvement of the control systems because of increased insight in the controller structures and an increased amount of time left for optimization.

4 Project plan and description of tasks
The project is divided into five main work packages, which are described below.

4.1.1 Literature study into state-of-the-art automotive industry and industrial partner
To start with, an overview of and insight in the state-of-the-art of the automotive industry in general and the industrial partner specifically have to be obtained. The overview concentrates on the electronical and software components, controllers and tuning methods currently used as well as the accompanying typical problems, restrictions and interactions. Information is gathered at the industrial partners as well as in literature and will be presented in a report. Specific case studies such as the clutch control (automated gearbox), exhaust gas recirculation control (engine) and (adaptive) cruise control will be defined. M.Sc. students will start at DAF Trucks and TNO working on the case studies using state-of-the-art controller design and tuning techniques.
4.1.2 Literature study into control design and automated tuning, definition of goals and case studies

A literature survey to the relevant methods for robust low-complexity controller design and the possibilities for (automated) tuning is performed. Based on this literature survey, the goals for the case studies regarding robust low-complexity controller design and (automated) tuning will be defined.

4.1.3 Robust low-complexity controller design

In this phase of the project, methods for robust low-complexity controller design are developed and validated by means of application to the case studies. A typical work plan for the case studies is:

a. modelling of the particular controller structure as provided by the supplier,
b. analysis of the working and determination of the pro and cons of this controller structure,
c. proposition of a new, simplified controller structure using this insight into the specific application.

Based on the results of the case studies, an overview of the suitable and implemented methods and guidelines for robust low-complexity controller design is derived.

4.1.4 (Automated) tuning with minimal effort

Considering the developed robust low-complexity design of the controllers, algorithms and methods for automated tuning with minimal effort are developed. This eventually leads to software tools for practical implementation and use. The developed methods, algorithms and software tools are validated by the use of simulations and experiments via SIL, HIL and RCP. Relevant suppliers may be involved in the project as well. Concerning the work plan for the case studies this implies:

a. definition of easy-to-use auto-tuning algorithms for the new controller structure,
b. identification and analysis of robustness and performance properties under real-life experimental conditions,
c. development of specific software tools for practical implementation.

4.1.5 Finishing of the project and dissertation

5 Implementation plan including knowledge transfer plan towards the industrial partner

Actual implementation and knowledge transfer towards the industrial partners are met by

- Locating the Ph.D. researcher for one day a week at one or both the industrial partners if useful (depending on the project phase this can be otherwise),
- Two-monthly project meetings with all partners,
- Presentation and discussion of the results after each milestone at all partners concerned (i.e. DAF Trucks N.V., TNO, TU/e and possible relevant suppliers),
- Coaching of a number of M.Sc. students, performing case studies at the industrial partners,
- Written reports as a Ph.D. thesis, M.Sc. reports, etc.,
- Other possible means of output like scientific publications, possibly patents, workshops, demonstrations, etc.
### Milestones, deliverables and global timetable

Corresponding to the project plan, four milestones are defined, followed by a fifth one defining the conclusion of the project. Approximately twice a year actual results will be presented in the form of an achieved milestone or intermediate results and the goals of the next milestone or of the next intermediate results will be updated. A project Go/NoGo decision point is incorporated one year after the project starts. This Go/NoGo decision depends on the judgement of TNO and DAF towards the capacities of the Ph.D. student to be sufficient to accomplish the project in a manner that is satisfactory or sufficiently effective to them. In case this leads to a NoGo, TNO will inform the TU/e before the first of January 2007 via a substantiated writing. The first timetable shown below presents the milestones and provides a global overview of the contents of the project plan. The second and third timetables show case studies for M.Sc. students, which are defined as supporting projects at DAF and TNO respectively.

#### Table 1 – Milestones

<table>
<thead>
<tr>
<th>Project planning</th>
<th>2006</th>
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<th>2008</th>
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<td>Milestone 5</td>
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**Contents work package 1:** State of the art automotive industry and DAF

**Contents work package 2:** Literature low-complexity design and automated tuning

**Contents work package 3:** Robust low-complexity controller design

**Contents work package 4:** (automated) tuning with minimal effort

**Contents work package 5:** Finishing of the project and dissertation

#### Milestone 1 (March 2007): Results literature study, definition of case studies and Go/NoGo

- **Overview of the electronic and software components, the control structure, typical problems, restrictions and interactions and the tuning methods specifically used by the industrial partner.**
- **Report of WP1 literature survey involving the state-of-the-art electronic and software components, controller structures and tuning methods in the automotive industry.**
- **Report of WP2 literature survey involving relevant methods for robust low-complexity controller design and (automated) tuning.**
- **Finishing of the first case studies on clutch control and ACC.**
- **Definition of project goals and case studies for application and validation of WP3 and WP4.**
- **Ph.D. educational programme finished by Ph.D.-researcher (e.g. DISC courses).**
- **Project review and Go/NoGo decision.**

#### Milestone 2 (March 2008): Results robust low-complexity controller design

- **Validated applications of robust low-complexity controller design on the case studies.**
- **Report containing an overview of the methods and guidelines for low-complexity controller design specifically focusing on the case studies.**
- **First application(s) of automated tuning on the case studies.**
- **Project review.**
Milestone 3 (March 2009): First results (automated) tuning with minimal effort
- Application and first validation of automated tuning on the case studies.
- Report containing first results of the methods, guidelines and (eventually) software tools for automated tuning with minimal effort focusing on the case studies.
- Project review.

Milestone 4 (February 2010): Final results (automated) tuning with minimal effort and dissertation
- Application and final validation of automated tuning on the case studies.
- Report containing final results of the methods, guidelines and (eventually) software tools for automated tuning with minimal effort focusing on the case studies.
- Software tools for automated tuning.
- Conclusions of the project and dissertation.
Appendix A  Project costs and financing (excluding in-kind contributions)

Table A.1 – Project costs and financing

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<th>Costs</th>
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In-kind contributions

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