Complex Embedded Automotive Control Systems
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Executive Summary

The aim of WP 1.1 is to develop controllers capable of preventing untripped vehicle rollover accidents. Untripped rollover events arise from driver actions such as excessively harsh manouvering at high speeds, as opposed to tripped rollover accidents which occur when a vehicle hits an obstacle after it has begun skidding. Tripped rollovers can be prevented through the use of ESP (Electronic Stability Programs) which prevent skidding. New controllers must be designed to prevent untripped rollovers.

The tendency of a vehicle to roll over depends on the loading conditions, primarily characterised by the mass and the height of the center of gravity. The controller should be able to prevent rollover under all loading conditions, while mainaining reasonable performance.

Work package 1.2 is concerned with the design and implementation of an automatic collision avoidance controller. The objective of the controller, as described in the project proposal, is to track the vehicle along a prescribed trajectory in order to avoid an impending collision. Possible scenarios for collision avoidance are to automatically start an evasion/dodging manoeuvre to avoid hitting an obstacle in cases when braking alone is not sufficient anymore; and to automatically control the attitude of the vehicle in such a way that the impact is least harmful in cases when the crash has become unavoidable.

This report summarised the requirements on functionality and performance imposed on the controllers being developed for rollover mitigation and collision avoidance.

1 WP 1.1: Controller Specifications for Rollover Prevention

This section describes the function and performance requirements of rollover prevention controllers. The reader may refer to Deliverable D1: Vehicle Control for Active Safety - Modelling for description of the vehicle dynamics and models referred to here.

1.1 Functional Requirements

The essential function of the rollover controller is to prevent driver-induced (untripped) rollover accidents by means of various actuators. The system should be capable of preventing rollover for a range of loading conditions, and should not restrict vehicle performance more than necessary.

1.1.1 Actuators

Potential actuator choices include braking systems where brake pressures can be individually assigned, steer-by-wire systems and active suspension systems. Within the context of this project the primary interest is using the brakes as actuators. In Deliverable D11 an experimental vehicle intended for research in rollover prevention is described. This vehicle is equipped with electronic brake force distribution (EBD) which allows brake torques at each wheel to be individually assigned. In this way braking commands can be used to directly influence both the longitudinal velocity u and the yaw rate $\dot{\psi}$.

1.1.2 Available Inputs

It is assumed that the following vehicle states are available, either through measurement or estimation:

State	Description
u	Longitudinal velocity
v	Lateral velocity
$\dot{\psi}$	Yaw rate
$\dot{\phi}$	Roll rate
ϕ	Roll angle

Additional signals that are assumed to be available include:

Signal	Description
a_y	Lateral acceleration
δ	Steering angle
μ	Coefficient of friction
ω_i	Wheel angular velocities

1.1.3 Controller Outputs

Vehicle modelling is often performed by considering resultant forces and moments acting on the vehicle chassis (see Deliverable D1). These forces are derived from forces at each of the tires. Controllers based on such models will therefore have outputs expressed as forces. The inputs to the brake actuators are pressures, so a conversion between brake pressure and force at the tire contact point is required. This can be done with knowledge of the parameters of the brake actuators, the dynamic rolling radius of the tires, and an estimate of the road-tire friction coefficient.

1.1.4 Loading Conditions

As previously mentioned, the controller must be capable of preventing rollover for a range of different loading conditions. A 'worst-case' approach is however not desirable here since this would lead to conservatism in performance. A change in loading conditions is primarily characterised by changes in the mass m and vertical position of the center of gravity h. Moments of inertia around the vehicles axes, I_{xx}, I_{yy} and I_{zz} will also change. In order to obtain better performance, it is possible to estimate the loading condition of the vehicle and use these estimated parameters in the control laws and switching schemes.

1.1.5 Modes of Operation

Under normal driving conditions the controller should not be active. A detection scheme (discussed later) should be used to activate the controller when a rollover event is likely. In addition to this on/off mode changing, there exist several possible modes of operation during a rollover event.

- All four wheels retain contact with the road surface. This is the 'preferred' mode of operation since all four tires can be used to influence the vehicle's motion.

- One wheel loses contact with the road. This is usually the rear wheel on the inside of the turn. In this case the controller must be modified to reflect the fact that an actuator has been lost. This can for example be done by modifying a control allocation problem.

- Two wheels lose contact with the ground. This is an extreme stage of rollover. Since the vehicle dynamics change in this mode (see Deliverable D1), another controller is necessary [1], [2].

The control strategy must be capable of detecting these different modes and switching between them.

1.1.6 Switching

Some mechanism is required for switching the controller on and off, and for changing between different controllers depending on the operating mode. For on/off switching, some form of rollover indicator measure is required. This could be based on lateral

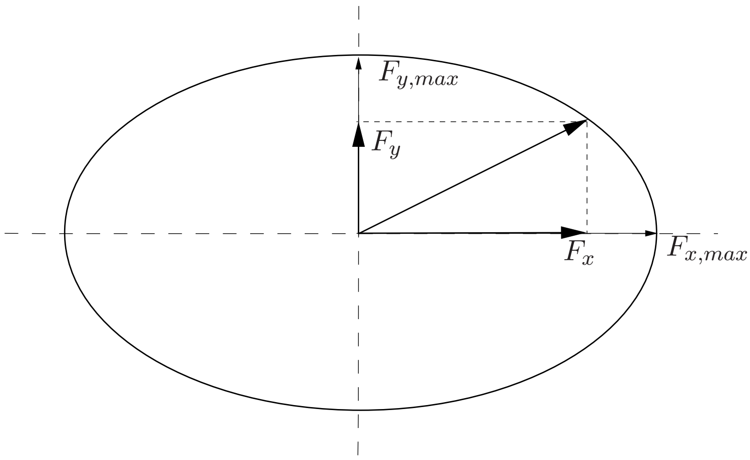

Figure 1: Friction ellipse

acceleration, roll angle, roll energy stored in suspension components, or steering angle measurements. For switching between different control modes, it is necessary to detect when a wheel loses contact with the road surface. This could be done by using wheel angular velocity and braking information, for instance. A key feature of the switching is that stability of the system must be maintained. To this end the theory of hybrid systems could be applicable.

1.1.7 Constraints

The rollover mitigation problem is characterised by a number of constraints, primarily involving the forces at the tire contact patch. These constraints are summarised by the so-called 'friction ellipse', illustrated in Figure 1. This implies that the resultant tire force must lie within an ellipse, defined by the maximum available lateral and longitudinal forces. The ellipse is described by the equation:

$$\left(\frac{F_y}{F_{y,max}}\right)^2 + \left(\frac{F_x}{F_{x,max}}\right)^2 = 1 \tag{1}$$

In fact, the resultant force is constrained to lie in one quadrant of the ellipse, since longitudinal forces must be negative (corresponding to braking), and the lateral force has a prescribed direction (determined by the sign of the tire slip angle α). The maximum longitudinal force is determined by the normal force F_z and the coefficient of friction μ. The maximum lateral force is given by the so-called 'Magic Formula' [3]. The reader is referred to Deliverable D1 or [8], [7] for further information on tire models.

Additional constraints arise from the actuators. The brakes have a number of performance constrains, outlined in Table 1.

1.1.8 Parameters

A number of parameters (or their estimates) are assumed to be available. These are summarised in Table 2

Maximum pressure	200	bar
Time delay	10	msec
max. pressure build-up time 0-50 bar	200	msec
max. pressure build-up time 0-100 bar	500	msec
max. pressure reduction rate	-1000	bar/s

Table 1: Brake actuator constraints

Parmeter	Symbol	Unit
Mass	m	kg
Height of CoG	h	m
Moment of inertia, x axis	I_{xx}	kgm^2
Moment of inertia, y axis	I_{yy}	kgm^2
Moment of inertia, z axis	I_{zz}	kgm^2
Dist. from CoG to front axle	a	m
Dist. from CoG to rear axle	b	m
Track width front	l_f	m
Track width rear	l_r	m
Roll stiffness	c_ϕ	Nm/rad
Roll damping	k_ϕ	Nms/rad

Table 2: Parameters

1.2 Performance Requirements

The purpose of the controller is to keep the vehicle within certain rollover margins. There are two types of rollover margins, static and dynamic [11]. In the static case, the roll acceleration is assumed to be zero. This corresponds to a slow increase in lateral acceleration. It is possible to define many different rollover margins, and a common way to define a margin is when both wheels on one side of the vehicle lose contact with the ground. It is possible to derive simple limits on lateral acceleration, for example from analysis of Figure 2. However, such limits are of limited use since they tend to represent upper bounds on the actual limit (more complex modelling usually gives lower values of the limit).

Another method for defining rollover margins is through energy considerations [1], [9],[2]. Here, a critical energy is derived, which represents the miminum necessary roll energy required for the vehicle to roll over. The condition is then that the vehicle's total roll energy (both kinetic and potential) must be less than this critical energy value.

1.2.1 Steering Angle Ramp

To measure the stationary rollover margin the steering angle ramp manoeuvre is used. At a constant speed of 80km/h the steering wheel angle is increased continuously and, thereby, starting in the linear range of lateral dynamics the vehicle is slowly driven toward the roll-over critical range. The gradient of the steering wheel angle at this manoeuvre is 5deg/sec. The steering input for this manouver is

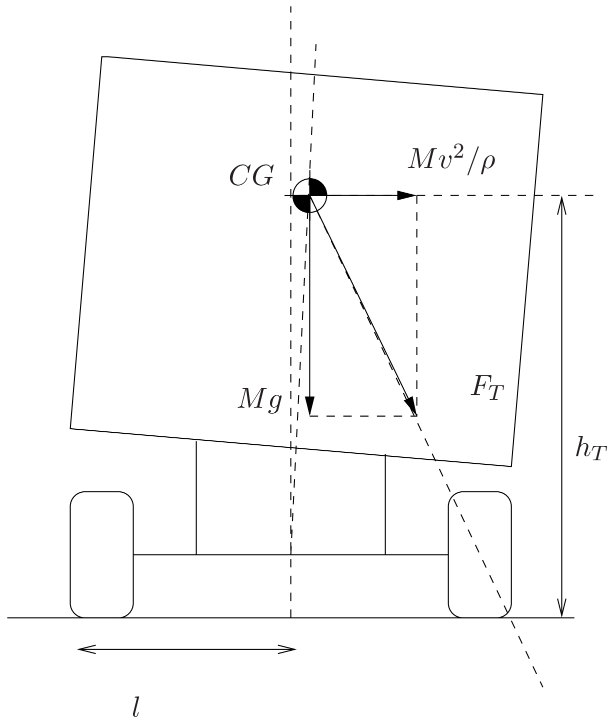

Figure 2: Rollover margin derivation

illustrated in Figure 3.

1.2.2 J-Turn

The J-turn is a simple step in the steering wheel angle driving the vehicle towards the physical limits. This manoeuvre can cause a roll over of vehicles with critical load. The speed of the vehicle just before the step input to the steering wheel angle is 60 mph (approx. 96 km/h). After releasing the accelerator pedal the steering wheel angle is increased at a rate of 1000deg/sec until it reaches 8 times the value δ_{stat} (the steer angle which is necessary to achieve 0.3g stationary lateral acceleration at 50mph (approx. 80km/h) . The steering input for this manouver is illustrated in Figure 4.

1.2.3 Fishhook

The so-called Fishhook manouver, like the J-Turn, is a dynamic manoeuvre. The speed before the start of the steering manoeuvre should be 50 mph (apprx 80 km/h). Similarly to the J-turn the accelerator pedal is released during the manoeuvre. The sequence of events in a fishhok manouver is:

- The steering wheel angle is increased at a rate of 720deg/sec up to 6.5 times δ_{stat}
- This value is kept for 250ms

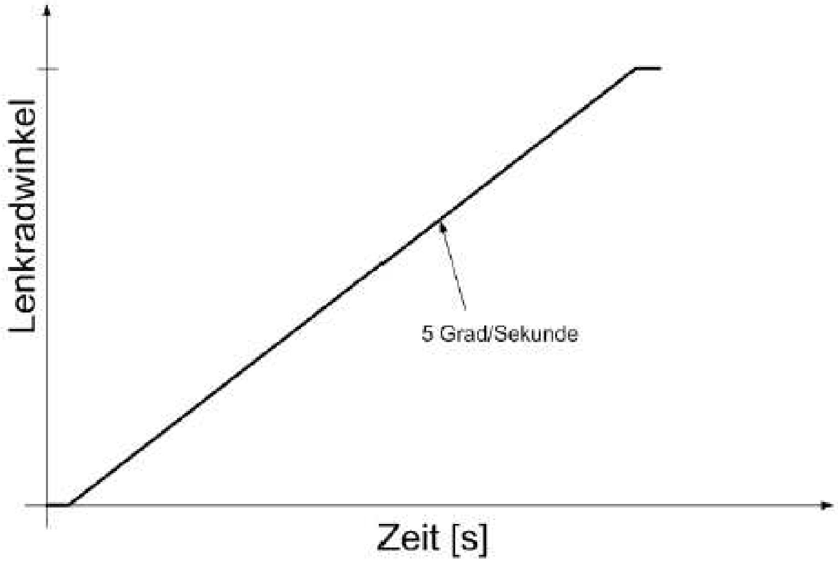

Figure 3: Steering angle ramp manouver

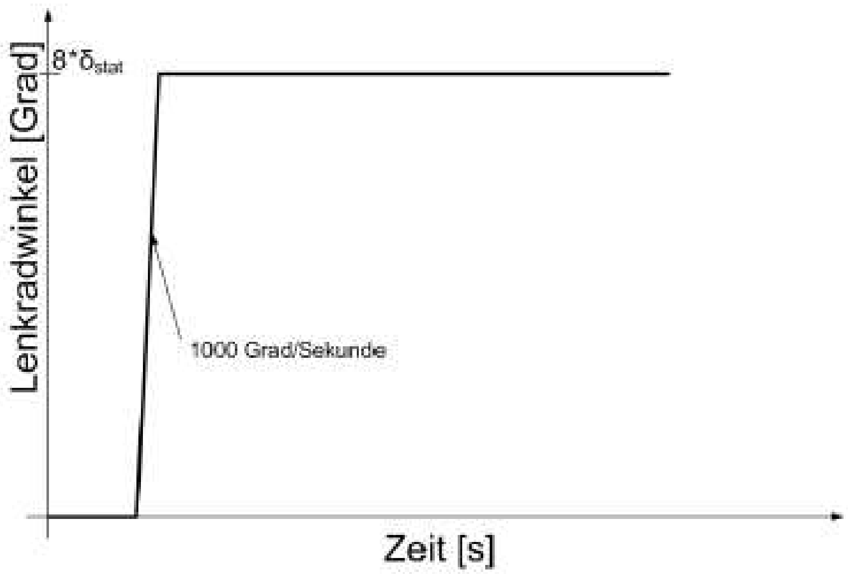

Figure 4: J-turn manouver

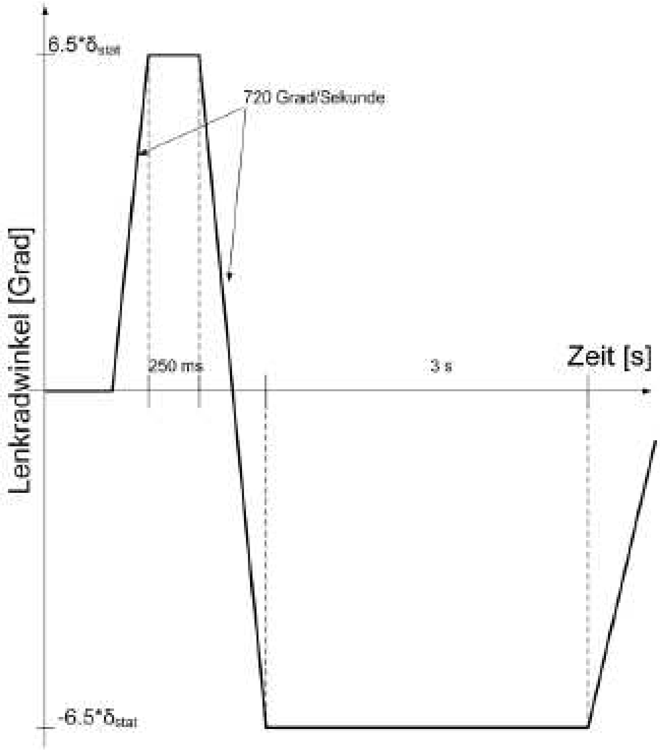

Figure 5: Fishhook manouver

- After that the steering wheel is turned into the opposite direction at a rate of 720 deg/sec until it reaches -6.5 δ_{stat}
- The value is kept for 3 seconds. Afterwards the steering wheel is turned 0 degrees again
- There is no specification on how the the retrun to 0 degrees is carried out

The steering input for the Fishhook manouver is illustrated in Figure 5.

1.2.4 Roll Rate Feedback Fishhook

A modified version of the Fishhook manouver, known as the roll rate feedback Fishhook manouver, can also be specified. This manouver is optimized for producing maximum vehicle roll. The sequence of events is similar to the standard Fishhook, but the second steering angle change is performed only after the roll rate becomes small (that is, when the roll angle reaches its maximum). The manouver is performed as follows:

- The steering wheel angle is increased at a rate of 720deg/sec up to 6.5 times δ_{stat}
- This value is held until the roll rate drops below 1.5 deg/sec

- After that the steering wheel is turned into the opposite direction at a rate of 720 deg/sec until it reaches -6.5 δ_{stat}
- The value is then held

1.3 Robustness

An important element of the control system is robustness to errors in state and parameter estimates. Robustness implies that modelling and estimation errors up to a certain magnitude should result in bounded output errors. Since a large number of states, parameters and signals must be estimated in the rollover control problem (examples include roll angle, mass, height of center of gravity and coefficient of friction), there are many sources of uncertainty. The controller must be capable of retaining vehicle stability in the presence of the uncertainties arising from the respective estimation schemes used.

2 WP 1.2: Collision Avoidance

This section of the report describes the control system specifications defining the requirements that are to be met in the later experimental validation of work package part 1.2. The work described in this section is primarily undertaken at the University of Glasgow.

2.1 Work package scope

The scope of the project is restricted to the narrow technological issue of applying appropriate control inputs to the vehicle in order to achieve the objective. Commercial, legal and social issues are outside the scope of the project. Importantly, the project does not seek to address the decision making process that determines whether, or how, an evasive manouevre should be initiated. It is assumed that a decision has already been made to perform a manoeuvre and that the general nature of the manoeuvre has been pre-determined. Thus there is no attempt to integrate image recognition, radar, GPS or other technologies to determine whether the manoeuvre is appropriate. Nor does the project seek to determine how transfer of control authority from the driver should take place. Human factors considerations of interaction between the driver and warnings of the impending emergency manoeuvre are not considered here.

2.2 Schedule

Four milestones are defined in the schedule of work; these milestones are specified to occur 6, 12, 24 and 36 months after the start of the project.

2.2.1 Milestone 1 - month 6

In the first six months of the project, vehicle models were developed to facilitate the later controller design phase of the project. At milestone 1, an interim report was delivered which outlined the system model development which had occurred.

2.2.2 Milestone 2 - month 12

This interim report outlines the controller specification for the experimental validation which will occur in later phases of the project, together with an implementation of the system model developed in the previous phase of the project.

The model source code is included with this report as a zipped archive. Documentation for the model is included in HTML format, also as an additional zipped archive.

2.2.3 Milestone 3 - month 24

Detailed controller design using techniques being developed in work package 3, and evaluation of that controller against the CASCaDE Simulink model, will be presented in a report at milestone 3, 12 months hence.

2.2.4 Milestone 4 - month 36

Following successful demonstration of controller performance by simulation at milestone 3, experimental validation of the controller will be undertaken. Validation results from these experiments, along with reports of any redesign activity, will be presented at milestone 4 one year later.

2.3 Controller specification

2.3.1 Problem introduction

For the purposes of the project, it is assumed that a rear-wheel drive passenger vehicle with front wheel steering and independent braking is travelling at fairly high speed on a straight and level multi-lane carriageway, with a free lane to one side.

At some point, an evasive manoeuvre is triggered and the automatic controller will cause the vehicle to move into the adjacent lane in a controlled manner using the full potential of the vehicle dynamics, i.e. taking the vehicle to its physical limits.

Initial demonstration of the controller performance will be given by simulating a test manoeuvre (described in section 2.4.1) using a proprietary full vehicle model (CASCaDE).

Following a successful demonstration of the controller design by simulation, the controller will be implemented on a real vehicle and the same manoeuvre will be demonstrated on a test track.

2.4 Controller demonstration

2.4.1 Manoeuvre specfication

ISO standard 3888 specifies test track layouts for two severe lane-change manoeuvres for passenger cars. Part 1 [4] specifies the track layout for a double-lane change. Part 2 [5] specifies the layout for obstacle avoidance; this is similar to the part 1 definition but requires the maneouvre to be conducted within more tightly constrained limits.

Both parts of the standard use double lane changes to provide subjective information about the lateral handling qualities of the vehicle. For the collision avoidance

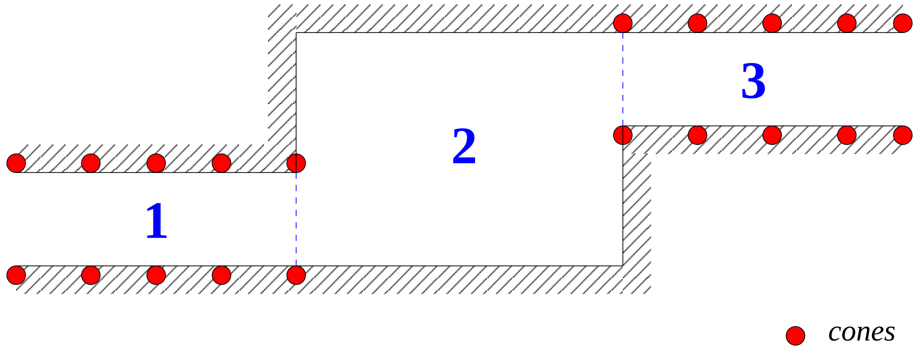

Figure 6: Test track layout for demonstration of severe lane change derived from ISO 3888-2:2002 *Passenger cars – Test track for a severe lane-change manoeuvre – Part 2: Obstacle avoidance*

Section	Length (metres)	Width (metres)
1	12.0	1.1 × vehicle width + 0.25
2	13.5	2.1 × vehicle width + 1.25
3	11.0	1.0 × vehicle width + 1.00

Table 3: Test track dimensions

problem being addressed in this project, subjective evaluation of the handling qualities of the vehicle is not required; nor is there any requirement for the vehicle to return to its original lane. After performing an emergency collision avoidance manoeuvre, the controller would be expected to do as little as possible and return control to the driver as soon as it is safe to do so. Consequently, the demonstration will perform only the first half of the manoeuvre specified in Part 2 of the standard.

2.4.2 Vehicle and environment specification

It is intended that this manoeuvre will be demonstrated on a DaimlerChrysler test track, using the existing Mercedes S-Class test car - “Technoshuttle” if it is available or a replacement vehicle if it is not. The full specification for this vehicle is provided by DaimlerChrysler in report D11 (also delivered at this milestone). The demonstration will take place on dry asphalt and it can be assumed that the friction coefficient between the wheels and the road is known to be approximately equal to a value of one.

Test track speed restrictions Safety concerns at the test track will limit the manoeuvre speed to a maximum of 100 km/hr. The recommended speed for performing the test described in ISO 3888-1 is 80 ± 3 km/hr so it is not anticipated that this limit will cause any difficulties.

Environmental disturbance The vehicle will be subject to natural disturbances during the manoeuvre, such as gusts of wind and road irregularities. However, there will be no disturbance of the steering wheel angle or wheel torques by the driver.

It is assumed that the driver is disengaged from control of the vehicle once the manoeuvre has been initiated.

2.4.3 Criteria for success

Actuator limits will restrict the control action that may be applied to the vehicle. However, the most significant constraint on vehicle performance will be that of the friction generated between the tyres and the road; the maximum acceleration achievable in the absence of aerodynamic propulsive devices is thus limited to approximately $\mu \times g$ metres per second squared, where μ is the coefficient of friction between the tyres and the road and g is the local constant of gravitational acceleration. Consequently, achievement of the controller objectives will be demonstrated by showing that the vehicle acceleration during the manoeuvre peaks close to this maximum value in the process of the vehicle tracking the prescribed trajectory while maintaining directional control.

2.5 Controller interfaces

2.5.1 Controller inputs

Following a prescribed trajectory with heading angle control requires accurate measurement of the longitudinal and lateral postion of the vehicle relative to the road, as well as the yaw angle. Measurement of the vehicle position on the test track will be assumed to come from GPS, however any other source of these measurements may be substituted if it should become desirable and practical to do so.

Accurate control of the vehicle will also require measurement of the body and wheel velocities and accelerations. The measurements of these parameters will be fused in the observer being developed as part of work package 4. Consequently the controller will be designed with the assumption that the observer is able to provide reliable and timely measurement data throughout the duration of the manouevre.

2.5.2 Controller outputs

The controller is to be designed such that it could be applied to a production vehicle with minimal modification. Consequently there will be no use of specialist actuation such as active suspension or independent four wheel steering. Five control actions will be available to the controller: front wheel steering and torque control of each of the four wheels.

The controller will output a desired steering angle which will be implemented by the existing steering wheel actuators on the test vehicle. The existing anti-lock braking system will be used to provide longitudinal wheel forces between the tyres and road as demanded by the collision avoidance controller.

2.6 Constraints

2.6.1 Actuator limits

The maximum force that can be generated by the friction between the wheels and road surface is $\mu \times F_z$ Newtons where F_z is the vertical load on the wheel. The longitudinal component of this force (in the wheel axis system) can be demanded from the existing ABS slip controller as long as it does not exceed the maximum available traction. The lateral component requires active control through the wheel steering angle, taking into account the tyre characteristics relating force to slip angle.

The steering system, under the control of a good driver, can achieve a steering wheel angle rate of up to 1000 degrees per second with a limit on the steering angle of approximately 50 degrees. It is not anticipated that these limits will be reached by the controller.

The longitudinal slip controller within the ABS has a bandwidth of approximately 15-18 radians per second, with the possibility of wheel lock occurring within approximately 40 milliseconds. Brake pressure can be increased at a rate of 500 bar per second and reduced at a rate of 2000 bar per second.

Overall, the dynamics of the wheels operate on a significantly faster timescale than the body dynamics which are of interest in controlling the vehicle trajectory.

2.6.2 Timing

The brake controller operates at a sample rate of 50 Hertz, with a delay of 20 milliseconds. The steering actuator control is faster and operates at 100 Hertz, but has a longer delay of 40 milliseconds.

2.7 Controller architecture

The high level conceptual architecture is shown in figure 7. A nominal trajectory is chosen which the car is required to follow. This nominal trajectory can be predefined to match any desired profile; in production use, however, some management system would have to select an appropriate profile based on the environment in which the vehicle finds itself - a swerve to the left would be inadvisable if there is a sheer drop to the left of the vehicle, or if another vehicle is driving alongside the car on that side.

The trajectory chosen by [6] uses the error function

$$y_d = 1.5\,\mathrm{erf}(x(t)/11.3); x(0) = -45m, x(t_f) = 45m$$

where y_d is the desired trajectory and $x(t)$ is the distance as a function of time.

A very similar profile can be obtained using a function of the form:

$$y_d = \begin{cases} y_0 & \text{if } x(t) < x_0 \\ y_1 & \text{if } x(t) > x_1 \\ y_0 + \frac{y_1 - y_0}{2}\left(1 + sin(\theta)\right) & \text{otherwise.} \end{cases}$$

where

$$\theta = \pi\left(\frac{x(t) - x(0)}{x(1) - x(0)} - \frac{1}{2}\right)$$

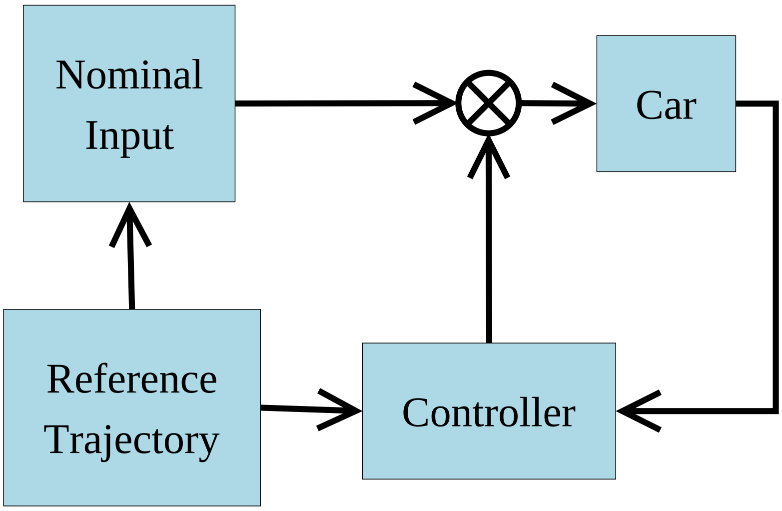

Figure 7: High level conceptual architecture

and (x_0, y_0) and (x_1, y_1) are the start and finish co-ordinates of the manoeuvre, respectively. Although similar to the erf-defined function, this profile is not asymptotic to the start and finish trajectories, but intercepts them cleanly. This should make it easier to define appropriate derivatives to execute a clean manoeuvre. For the demonstration, the start and finish co-ordinates are defined such that the trajectory does not exceed the boundaries of the coned area described in section 2.4.1.

Having defined a reference trajectory, an inverse model of the vehicle is used to obtain a feedforward controller to steer the vehicle to follow the path.

The output of the feedforward controller is then applied to the vehicle. A feedback controller can then correct for disturbances by adjusting the steering angle and the braking torque on each of the wheels. This feedback controller requires the reference input to be provided to it so that deviations from the path can be calculated.

The design of these lower level controllers is discussed briefly in the interim report describing work package 3. A more detailed description will be given in the interim report at milestone 3.

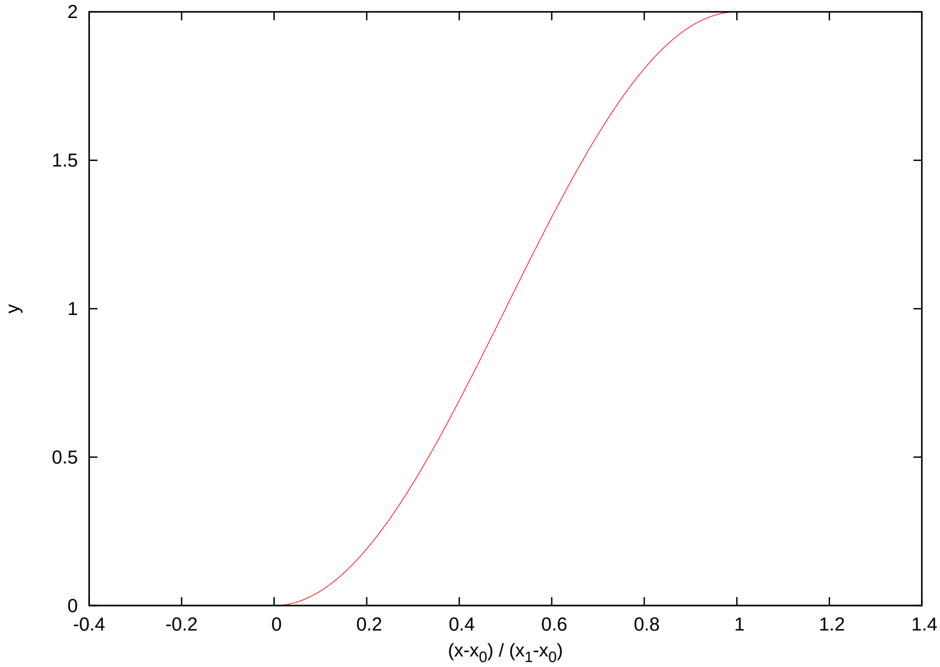

Figure 8: Nominal trajectory

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