Contributions to Primary, Secondary and Integrated Traffic Safety

Habilitation Thesis

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Abstract

This habilitation thesis deals with various aspects of traffic safety. It first provides a holistic overview of traffic safety, including its social consequences, and a discussion of the traffic system, which consists of the human, the vehicle and the environment. Drawing on the relevant literature, traffic safety is classified into primary, secondary and integrated safety, and a brief overview of accidentology is presented.

The *kind of impact* classification is used to summarise contributions to secondary safety, with a particular focus on frontal, side, rear impact and rollover research. The ensuing section then lists and categorises several different traffic safety systems. Due to the large number of traffic safety systems, it is necessary to prioritise based on market introduction, which is accomplished through a retrospective case study of fatal traffic accidents in Austria in 2003 *(RCS-TUG study).* The distinguishing feature of this study is its in-depth investigation of the pre-collision phase of more than 200 accidents. By conducting a detailed analysis of the dynamics of the involved vehicles, the benefit of traffic safety systems is assessed. The results are in turn compared to findings from the literature and a prioritisation is proposed. This study makes a valuable contribution to aspects of primary safety. Finally, the thesis describes an approach for the integration of primary and secondary safety systems through an appropriate algorithm (*Integrated Safety Controller ISC*). It predicts parameters of an oncoming collision and calculates (pre-fire) trigger times and force levels of an adaptive restraint system.

The present habilitation thesis summarises contributions of the author to the scientific subject *Automotive Engineering*, of which traffic safety is an important aspect. The thesis includes a summary of key publications, as well as innovative scientific research (RCS-TUG and ISC). An additional introductory portion describes the scientific subject and deals with mobility, vehicle subsystems and components, and automotive history, present and future. A description of the activities of other institutes provides an insight into the breadth of the subject, and the author's contributions to research and education in the field of *Automotive Engineering* are listed.

Kurzfassung

Die vorliegende Habilitationsschrift befasst sich mit verschiedenen Aspekten der Verkehrssicherheit. Dabei führt sie mit einem umfassenden Überblick in die Thematik ein. Dieser beinhaltet die sozialen Folgekosten die sich aus Verkehrsunfällen ergeben, sowie das Verständnis des Gesamtsystems Verkehr bestehend aus Mensch, Fahrzeug und Umwelt. Wie in der Literatur beschrieben, werden die Begriffe aktive, passive und integrierte Verkehrssicherheit eingeführt und ein kurzer Überblick in die Unfallforschung gegeben.

Die Einteilung nach *Aufprallarten* wird verwendet um Beiträge des Autors in der passiven Sicherheit zusammenzufassen. Diese liegen in Forschungstätigkeiten in den Bereichen Frontal-, Seiten- und Heckaufprall sowie dem Fahrzeugüberschlag. Im Folgenden werden Sicherheitssysteme aufgezählt und kategorisiert. Die Vielzahl unterschiedlicher Systeme verlangt nach einer Priorisierung für deren Markteinführung. Dies geschieht durch eine retrospektive Untersuchung tödlicher Verkehrsunfälle in Österreich des Jahres 2003 (*RCS-TUG* Studie). Die Besonderheit liegt dabei in der Detailuntersuchung der Vorkollisionsphase durch Berechnung der Fahrdynamik der beteiligten Fahrzeuge. Dabei werden über 200 tödliche Unfälle detailliert untersucht und die Wirksamkeit unterschiedlicher unfallvermeidender Systeme bewertet. Die Ergebnisse werden mit der Literatur verglichen und ein Vorschlag zur Priorisierung von aktiven Systemen angeführt. Dabei ist diese Studie als Beitrag zur aktiven Sicherheit einzustufen. Schließlich beschreibt die Habilitationsschrift einen Ansatz zur Integration von aktiven und passiven Systemen. Dabei errechnet ein im Detail beschriebener Algorithmus (*Integrated Safety Controller ISC*) geeignete Auslösezeiten und Rückhaltekräfte eines adaptiven Rückhaltesystems.

Die vorliegende Habilitationsschrift fasst Beiträge des Autors zur Weiterentwicklung des wissenschaftlichen Faches *Fahrzeugtechnik* zusammen, wobei die Verkehrssicherheit einen wesentlichen Aspekt darstellt. Die Schrift beinhaltet eine Zusammenfassung von Schlüsselpublikationen wie auch neuartige Forschungserkenntnisse (RCS-TUG und ISC). Ein zusätzlicher einführender Teil beschreibt das wissenschaftliche Fach, wobei Mobilität, Systeme und Komponenten im Fahrzeug sowie Vergangenheit, Gegenwart und Zukunft des Automobils beleuchtet werden. Aktivitäten anderer Forschungseinrichten beschreiben die Breite des Faches und Beiträge des Autors in Hinblick auf Forschung und Lehre werden aufgezählt.

Acknowledgement

The present habilitation thesis is the result of my scientific research works which started with my diploma thesis at the Institute of Mechanics at the Graz University of Technology (1994). This was followed by a dissertation, which was finalised in 1998. I could learn a lot from my doctoral thesis supervisor, Prof. Hermann Steffan, as well as as from my colleagues Dr. Bertram Geigl, Heinz Hoschopf, Dr. Andreas Moser, Dr. Kurt Steiner and Dr. Stephan Winkler.

In 1998, I joined MAGNA STEYR Fahrzeugtechnik (former known as Steyr-Daimler-Puch-Fahrzeugtechnik), where I had the opportunity to contribute to vehicle research and development. From this company, I would like to especially acknowledge the support of Toros Akgün, Wilhelm Breitenhuber, Manuel Harzheim, Wolfgang Körner, Anton Reisenhofer, and Charly Stocker, to name just a few.

Due to my interests in scientific teaching, I joined the Institute of Automotive Engineering at the Graz University of Technology in 2007. I would like thank the head of the institute, Prof. Wolfgang Hirschberg, for his professional support and trust in my abilities. He granted me the freedom to perform basic research while still guiding me in the right direction. Professor Hirschberg manages to foster a warm and friendly atmosphere, which makes the institute a truly enjoyable place to work.

The scientific research carried out at the institute would not have been possible without the aid of my graduate students, Ignasi Pijoan Bori, Gashawbeza Abebe Degefie, David Puerta Llama, Rüdiger Rohm and Daniel Wallner, and the remarkable programming skills of our trainee, Michael Sieberer. From the institute's staff, I would like to acknowledge the aid of Dr. Daniel Kollreider (who also shares my interest in music), as well as Andreas Podlipnig and Erich Erhart. A debt of gratitude is also owed to our inexhaustible secretary, Petra Schützenhofer, for keeping my organisational chaos in check, and to Mike Phillips for his support with the English for this thesis. I would also like to mention my colleagues from the Virtual Vehicle Competence Center, Robert Cresnik, Dr. Thomas Heubrandtner, Thomas Jost, Dr. Christoph Lind, Dr. Christian Ruff, Dr. Andreas Rieser, Wolfgang Schimpl and Harald Schluder, for their contributions to projects we carried out together. In addition, I offer a special thanks to Dr. Bernd Fachbach, who has shared my professional career at Graz University of Technology, MAGNA Steyr and now in the Virtual Vehicle Research and Test Center, and has been my ever-dependable source of valuable advice.

Finally, I would like to thank my family and friends, who endured the occasional neglect occasioned by this thesis and nevertheless supported me whenever necessary. And of course my deepest gratitude goes to my wonderful wife, Ursula, who has been my steady source of support and encouragement for 20 years now and has given me the most valuable gift I can imagine, our daughter Johanna.

"The computer can't tell you the emotional story. It can give you the exact mathematical design, but what's missing is the eyebrows."

Frank Zappa

Contents

Abbreviations

ACC	Automatic Cruise Control
ADAS	Advanced Driver Assistance System
ALE	Arbitrary Lagrangian Euler method
ANN	Artificial Neural Networks
APROSYS	Advanced PROtection SYStems (EC project)
ATD	Anthropometric Test Device (synonym for crash test dummies)
AWD	All Wheel Drive
C1	First cervical vertebra
CAE	Computer Aided Engineering
CAx	Computer Aided methods
CFC	Channel Filter Class
CFD	Computational Fluid Dynamics
CFR	Constant Force Restraint system
CKF	Cubature Kalman Filter method
CoG	Centre of Gravity
CRC	Continuous Restraint Control
CVIS	Cooperative Vehicle Infrastructure Systems (EC project)
CVT	Continuous Variable Transmission
CWS	Collision Warning Systems
DAS	Driver Assistance System
E/E	Electrics and Electronics
EACS	European Accident Causation Survey
EC	European Community
ECU	Electronic Control Unit
EES	Energy Equivalent Speed
EKF	Extended Kalman Filter method
EMA	Automatic Evasive Manoeuvre Assistant
FARS	Fatality Analysis Reporting System (in-depth US accident database)
FCD	Flexible Collision Deflector
FCWS	Forward Collision Warning Systems
FEM	Finite Element Method
FMVSS	Federal Motor Vehicle Safety Standards
FWD	Front Wheel Drive
GIDAS	German In-Depth Accident Study
GST	Global System for Telematics (EC project)

HGV	Heavy Goods Vehicles
HMI	Human-Machine Interface
I/O	Input/Output
IHR	Inflatable Head Restraint
ISC	Integrated Safety Controller
ISC-C	Integrated Safety Controller-Collision Module
ISC-I	Integrated Safety Controller-Integration Module
ISC-O	Integrated Safety Controller-Occupant Module
ISC-P	Integrated Safety Controller-Pre-collision Module
ISC-V	Integrated Safety Controller-Vehicle Module
ISO	International Organisation for Standardization
IV-NIC	Intervertebral Neck Injury Criterion
KF	Kalman Filter method
LDW	Lane Departure Warning
LKA	Lane Keeping Assist
MBS	Multibody System
NASS	National Automotive Sampling System
NASS-CDS	National Automotive Sampling System - Crashworthiness Data System (in depth accident database, representative sub-sample from NASS-GES)
NASS-GES	National Automotive Sampling System - General Estimates System (US motor vehicle accident statistics)
NHTSA	National Highway Traffic Safety Administration (US authority)
NIC	Neck Injury Criterion
NIF	Neck Injury Factor
ODE	Ordinary Differential Equation
OEM	Original Equipment Manufacturer
PBA	Predictive Brake Assist
PENDANT	Pan-European Co-ordinated Accident and Injury Database (EC project)
PF	Particle Filter
PMHS	Postmortem human subjects
PReVENT	Preventive and Active Safety Applications (EC project)
RISER	Roadside Infrastructure for Safer Roads (EC project)
ROLLOVER	Improvement of rollover safety for passenger vehicles (EC project)
RCS-TUG	Retrospective Case Study - Graz University of Technology
RWD	Rear Wheel Drive
SAE	Society of Automotive Engineers
SOP	Start Of Production
SUV	Sports Utility Vehicle
T1	First thoracic vertebra
TRACE	Traffic Accident Causation in Europe (EC project)
TSS	Traffic Safety Systems
TTC	Time-to-Collision

TTF	Time-to-Fire (triggering of safety system)
TTP	Time Triggered Protocol
UKF	Unscented Kalman Filter method
US-NCAP	United States New Car Assessment Program (crash test program for consumer information)
V2I	Vehicle-to-Infrastructure Communication
V2V	Vehicle-to-Vehicle Communication
VRU	Vulnerable Road User
VS90	Vehicle Safety 90 (traffic accident database)
VSS	Vehicle Safety System
WHO	World Health Organisation
ZEDATU	Zentrale Datenbank Tödlicher Unfälle (in-depth accident database)

Symbols

Coordinate systems

x_e, x_o, x_g, x_f, x_r	Longitudinal axis of ego-vehicle, obstacle, global (inertial) system, front axle, rear axle
y_e, y_o, y_g, y_f, x_r	Lateral axis of ego-vehicle, obstacle, global (inertial) system, front axle, rear axle
$\mathcal{O}_g$	Origin of global coordinate system
$\mathcal{O}_e, \mathcal{O}_o$	Origin of ego-vehicle and obstacle
$\mathcal{O}_f, \mathcal{O}_r$	Origin of front and rear axle coordinate system
$\mathcal{L}_e$	Origin of ERS on ego-vehicle (laser-scanner)

Parameters and constants

a_F, b_F, c_F, d_F	Main dimensions of Flexible Collision Deflector(FCD)
a_E, c_E	Parameter for EMA trajectory
c, c_e, c_i, c_o	Stiffness parameter (collision model) for ego-, i-th obstacle and obstacle vehicle
c_D	Air drag coefficient
$d_{e,0}, d_{o,0}, d_{i,0}$	Initial distances
f	Focal length, degree of freedom
$f_{e(i)}$	Index for time step in the future, ego-vehicle (obstacle i)
f_{TTC}	time step at calculated TTF
g	Gravitational acceleration
h	Drop height (inverted drop test)
i	Index of obstacle
k	Index for time step in the past, ego-vehicle
l	Index for time step in the past, obstacle
$l_{e(i)}$	Length of ego-vehicle (obstacle i)
m	Index for measurement signal
m_{car}	Number of registered cars in VS90 [Ver94]
m_e	Mass of ego-vehicle
m_o, m_i	Mass of (ith) obstacle
$m_O, m_{O,k}$	Mass of (k-th) occupant
m_{Germ}	Total number of registered cars in VS90 [Ver94]
n	Length of dynamic state vector
n_{neck}	Number of neck injuries in VS90 [Ver94]

n_{VS90}	Total number of rear impacts in VS90 [Ver94]
$p_{O,k}$	Position of k-th occupant w.r.t dashboard
r	Retina radius, index of ERS measurement signal
$r_{dyn,f(r)}$	Dynamical tyre radius at the front (rear) axle
$r_{dyn,i}$	Dynamical tyre radius of wheel i
r_0, r_s	Unloaded and static tyre radius
s	Index of s-th occupant
$v_{e,0}, v_{o,0}, v_{i,0}$	Initial velocity of ego-vehicle and (i-th) obstacle
$w_{e(i)}$	Width of ego-vehicle (obstacle i)
$x_{rel,max}$	Maximal available displacement for occupant ride-down inside the vehicle during collision
A_0, A_1, A_2, B_1, B_2	CFC filter constants
A_e	Projected frontal area of ego-vehicle
CFC	Channel Filter Class parameter
$F_{e(i)}$	Number of future states of ego-vehicle (obstacle i)
I	Number of detected obstacle
$J_e, J_{f(r)}$	Moment of inertia of ego-vehicle, front (rear axle)
K	Number of time steps in the past
L_1	"Low" Radar range
L_2	"Long" Radar range
M	Number of vehicle measurement signals
O_e	Offset of ego-vehicle to obstacle
R	Obstacle radius, Number of measurement signals of ERS
S	Number of considered occupant
ΔW	Overlapping width between ego-vehicle and obstacle
$W103_i$	Width of vehicle i

Variables

$a, a_{e(o)}$	Acceleration of ego-vehicle (obstacle)
a_{CFC}	Filtered acceleration data
$a_{s,max}$	Maximum acceleration of s-th occupant
$a_{s,mean}$	Mean acceleration of s-th occupant
a_{max}	Maximum acceleration
a_{raw}	Unfiltered acceleration data
a_{rel}	Relative acceleration between C1 (first cerebral vertebra) and T1 (first thoracic vertebra)
	Relative acceleration between vehicles
$c_{sx,f(r)}$	Lateral stiffness of front (rear) axle
$corr$	Correction factor
f_d	Limit frequency
k_{Hyst}	Hysteresis for collision model
$m_{O,s}$	Mass of s-th occupant

$n_{A,S}$	Number of avoided fatal accidents of safety system S
$n_{Ev,S}$	Number of evaluated fatal accidents of safety system S
$n_{P,S}$	Number of cases of safety system S, which has potential to prevent fatalities
$p_{O,s}$	Position of s-th occupant
$p_{e(o)}$	Pedal position of brake/throttle of ego-vehicle (obstacle)
$s_{x,i}, s_{f(r)}$	Longitudinal tyre slip of wheel i, front (rear) axle
$s_{y,i}$	Lateral tyre slip of wheel i
$\Delta s_{e(o)}$	Displacement (crush) of ego- (obstacle) vehicle
t	Time
v, v_x	Longitudinal vehicle velocity
v_{rel}	Relative velocity between C1 and T1
	Relative velocity between ego-vehicle and obstacle
${}_{e(g)}v_{x,e(i)}, v_{e(o)}$	Longitudinal velocity of ego-vehicle (obstacle i) in ego-vehicle (global) coordinate system
$v_{x,ek}$	Longitudinal velocity of ego-vehicle e at time step k
$v_{x,il}$	Longitudinal velocity of obstacle i at time step l
$v_{x,i}$	Longitudinal tyre velocity of wheel i
${}_{e(g)}v_{y,e(i)}$	Lateral velocity of ego-vehicle (obstacle i) in ego-vehicle (global) coordinate system
$v_{y,i}$	Lateral tyre velocity of wheel i
$v_{y,ek}$	Lateral velocity of ego-vehicle e at time step k
$v_{y,ek}$	Lateral velocity of obstacle i at time step l
$\Delta v, \Delta v_{e(o)}$	Relative velocity between vehicles/obstacles
x_P	Longitudinal distance between P and Q
$x_{0,c}$	Number of vehicles in control group without a safety system S
$x_{0,p}$	Number of vehicles in case (pertinent) group without a safety system S
$x_{1,c}$	Number of vehicles in control group (model year 1)
$x_{1,p}$	Number of vehicles in case (pertinent) group (model year 1)
$x_{2,c}$	Number of vehicles in control group (model year 2)
$x_{2,p}$	Number of vehicles in case (pertinent) group (model year 2)
$x_{c,e}$	Displacement of ego-vehicle's CoG during collision
$x_{c,o}$	Displacement of obstacle vehicle's CoG during collision
${}_{e(g)}x_{e(i)}$	Longitudinal position of ego-vehicle (obstacle i) in ego-vehicle (global) coordinate system
Δx	Relative distance between vehicles/obstacles
	Relative distance between eye and obstacle
$x_{i,0}$	Number of vehicles i without a safety system 0 in accident database
$x_{i,1}$	Number of vehicles i in time period 1 in accident database
$x_{i,S}$	Number of vehicles i without a safety system S in accident database

x_{rel}	Displacement of occupant inside the vehicle during collision
$x_{S,c}$	Number of vehicles in control group with safety system S
$x_{S1,p}$	Number of vehicles in case (pertinent) group with safety system $S1$
$x_{S1,c}$	Number of vehicles in control group with safety system $S1$
$x_{S2,p}$	Number of vehicles in case (pertinent) group with safety system $S1$
$x_{S2,c}$	Number of vehicles in control group with safety system $S1$
$x_{S,p}$	Number of vehicles in case (pertinent) group with safety system S
$xe_{i,2}$	Expected number of accident involvement of vehicles i in time period 2
$xe_{i,S}$	Expected number of accident involvement of vehicles i with safety system S
$xo_{i,2}$	Observed number of accident involvement of vehicles i in time period 2
x_e	Longitudinal coordinate of EMA trajectory
x_{ek}	Longitudinal position of ego-vehicle e at time step k
x_{il}	Longitudinal position of obstacle i at time step l
x_{Vm}	m-th vehicle measurement signal
Δx_b	Distance for braking intervention
Δx_c	Relative displacement of CoG of ego- and obstacle vehicle during collision
Δx_w	Warning distance
$\Delta x_{w,stat}, \Delta x_{w,mov}$	Warning distance for stationary and moving obstacles
$y_{i,0}$	Number of exposure of vehicles i without a safety system 0
y_e	Lateral coordinate of EMA trajectory
y_{ek}	Lateral position of ego-vehicle e at time step k
y_{il}	Lateral position of obstacle i at time step l
${}_{e(g)}y_{e(i)}$	Lateral position of ego-vehicle (obstacle i) in ego-vehicle (global) coordinate system
$y_{i,1}$	Number of exposure of vehicles i in time period 1
$y_{i,2}$	Number of exposure of vehicles i in time period 2
$y_{i,S}$	Number of exposure of vehicles i without a safety system S
y_M	Desired lateral position (EMA)
y_P	Lateral distance between P and Q
A_S	Avoidance of fatal accidents by safety system $S[\%]$
CR_i	Control ratio of vehicle i
E_S	Effectiveness of safety system S
$E_{S1,add}$	Effectiveness of safety system $S1$ in addition to $S2$
Ea_S	Adjusted effectiveness of safety system S
$F_{app,e(o)}$	Applied force at ego-vehicle and obstacle
$F_c, F_{c,n}$	Collision force, of n-th vehicle
$F_{c,corr}$	Corrected collision force

$F_{c,b}$	Collision force measured at barrier load cell
$F_{x,i}, F_{x,f(r)}$	Longitudinal tyre force of wheel i, front (rear) axle
$F_{\mu,x,f(r)}$	Longitudinal tyre force of front (rear) axle, scaled with grip μ
$F_{\mu,y,f(r)}$	Lateral tyre force of front (rear) axle, scaled with grip μ
$F_{y,i}$	Lateral tyre force of wheel i
$F_{z,i}$	Vertical tyre force of wheel i
F_B	Braking force
G	Weight force
$IR_{i,0}$	Involvement rate of vehicle model i without a safety system 0
$IR_{i,1}$	Involvement rate of vehicle model i in time period 1
$IR_{i,S}$	Involvement rate of vehicle model i with a safety system S
$M_{D,f(r)}$	Driving or braking moment at the front (rear) axle
$M_{R,f(r)}$	Rolling resistance moment at the front (rear) axle
NIF	Neck Injury Factor
NIC	Neck Injury Criterion
P_S	Potential of safety system S to prevent fatalities [%]
R	Corner radius
$R_{1,S}$	"Crude" odds ratio of safety system S
R_2	Correction of odds ratio
R_D	Air drag
R_S	"Corrected" odds ratio of safety system S
$R_{S1,add}$	"Added" odds ratio of safety system $S2$ added with $S1$
$RF_{AB,s}$	Restraint force, airbag, s-th occupant
$RF_{SB,s}$	Restraint force, seatbelt, s-th occupant
$RR_{i,S}$	Risk ratio of vehicle i with a safety system S
$RRa_{i,S}$	Adjusted risk ratio of vehicle i with a safety system S
$T_{f,e(i)}$	Time interval in the future, ego-vehicle (obstacle i)
$T_{p,e}$	Time interval in the past, ego vehicle
$T_{p,i}$	Time interval in the past, obstacle i
ΔT	Time step
$\Delta T_{f_{e(i)}}$	Time interval at time step f for ego-vehicle (obstacle i)
ΔT_k	Time interval at time step k
ΔT_l	Time interval at time step l
$TTF_{AB,s}$	Time-to-Fire, airbag, s-th occupant
$TTF_{SB,s}$	Time-to-Fire, seatbelt, s-th occupant
TTC	Time to collision
$V_{x(z),f(r)}$	Intersecting forces in longitudinal and vertical at wheel centre
$W_{x(y)}$	Longitudinal (lateral) wind forces
α_F	Angle between deflector and coupling bar (FCD)
$\alpha_{f(r)}$	Front (rear) axle tyre slip angle
β	Side slip angle
γ	Longitudinal road inclination
$\varepsilon, \varepsilon_{e(o)}$	Coefficient of restitution, of ego- (obstacle) vehicle

$\sigma_{a_{x(y)}}$	Standard deviation of longitudinal (lateral) acceleration
$\sigma_{v_{x(y)}}$	Standard deviation of longitudinal (lateral) velocity
σ_{ω_z}	Standard deviation of yaw rate
φ_F	Angle between longitudinal axis and coupling bar (FCD)
ϕ_1	Aperture low range Radar
ϕ_2	Aperture long range Radar
ψ_{ek}	Heading angle of ego-vehicle e at time step k
ψ_{il}	Heading angle of obstacle i at time step l
ψ_F	Angle between lateral axis and fictitious line between P and Q (FCD)
${}_{e(g)}\psi_{e(i)}$	Heading angle of ego-vehicle (obstacle i) in ego-vehicle (global) coordinate system
θ	Pitch angle
μ, μ_i, μ_e, μ_o	Grip, grip at wheel i, grip at ego-vehicle e and obstacle o
ω_i	Wheel rate of wheel i
$\omega_{z,e(i)}$	Yaw rate of ego-vehicle, obstacle i
$\omega_{x,ek}$	Yaw rate of ego-vehicle e at time step k
$\omega_{z,il}$	Yaw rate of obstacle i at time step l
ω_D	Difference between reference and observed yaw rate
ω_R	Reference yaw rate
ω_t	Yaw rate threshold
ω_O	Observed yaw rate
ρ	Air density
τ_B	Loss of time until full braking
τ_R	Driver reaction time

Vectors

${}_e\mathbf{a}_{e,f_e}$	Vector of predicted passenger cell pulse
$\mathbf{k}$	Vector of centrifugal forces
$\mathbf{ic_s}$	Vector of injury criteria, s-th occupant
$\mathbf{p}_e(i)$	Parameter vector ego vehicle (obstacle i)
$\mathbf{q}$	Vector of applied forces
$\mathbf{r}_s$	Input vector for the restraint system, s-th occupant
$\mathbf{u}(t)$	Control input to vehicle
$\mathbf{v}(t)$	Disturbances from environment
$\mathbf{v}_k$	Measurement noise
$\mathbf{v}_{f(r)}$	Front (rear) wheel velocity vector
$\mathbf{v}_e$	Ego-vehicle velocity vector
$\mathbf{w}_k$	Process noise
$\mathbf{x}_{c,e}, \mathbf{x}_{c,o}$	State vector of ego-vehicle and obstacle during collision
$\mathbf{x}, \mathbf{x_e}(t)$	State vector of ego-vehicle
$\mathbf{x}_k$	State vector at discrete time step k
$\mathbf{x_{ek}}$	State vector $\mathbf{x_e}$ of ego-vehicle at time step k
$\mathbf{x_{ir}}$	State vector $\mathbf{x_i}$ of obstacle i at time step r
$\mathbf{x_{E,i}}(t)$	State vector $\mathbf{x_E}$ of obstacle i
${}_e\mathbf{x_{O,s}}$	Parameter vector of s-th occupant
${}_e\mathbf{x_V}(\mathrm{t})$	Measurement signals from vehicle on-board sensing system
$\mathbf{x}_k$	State vector at discrete time step k
${}_g\mathbf{y}$	Position vector in global coordinates
${}_e\mathbf{z}$	Generalised velocity vector in vehicle coordinate system

Matrices

$\mathbf{A}$	Transition matrix relating $\mathbf{x}_k$ to $\mathbf{x}_{k+1}$
$\mathbf{M}$	Mass matrix
$\mathbf{Q}$	Covariance matrix
$\mathbf{T}_{ge}$	Transformation matrix ego-vehicle to global coordinate space
${}_e\mathbf{X}_i$	Matrix including K dynamic states of I obstacle (vehicles)
${}_e\mathbf{X}_{e,k}$	Matrix of stored (past) dynamic states of ego-vehicle
${}_e\mathbf{X}_{i,l}$	Matrix of stored (past) dynamic states of obstacle i
${}_g\mathbf{X}_{i,l}$	Matrix of stored (past) dynamic states of obstacle i
${}_g\mathbf{XF}_{e,f_e}$	Matrix of predicted (future) dynamic states of ego-vehicle
${}_g\mathbf{XF}_{i,f_i}$	Matrix of predicted (future) dynamic states of obstacle i

Part I.

Automotive engineering

1. Introduction to automotive engineering

With the present thesis, the author applies for the *venia docendi* in the scientific subject "Automotive Engineering", which deals with *road vehicles*. Road vehicles are classified according to DIN 70010, [Nor01] Fig. 1.1, and are among the most complex industrial products. The product is not only complex, but is also produced in quantities ranging from low to high volumes and, since these days it is a consumer or investment good, faces high pressure on production costs. It has to fulfil many diverse requirements, which are often competing against each other. Due to the complexity and diversity of the product, numerous scientific subjects are involved in its development and production.

Fig. 1.2 provides an example of the complex web of interrelations between some important scientific subjects and applications in automotive engineering. Due to this complexity, automotive engineering requires the multidisciplinary cooperation of experts from many different sciences. This is true for both the automotive industry, as evidenced by complex corporate organisational structures, and for higher education. At universities, the teaching of automotive engineering is carried out with different strategies:

- **Complete coverage of the subject**
  To meet this objective, either different scientific subjects join together with a focus on automotive engineering[1], or the instruction focuses on a basic level.

[1] A typical example is RWTH Aachen.

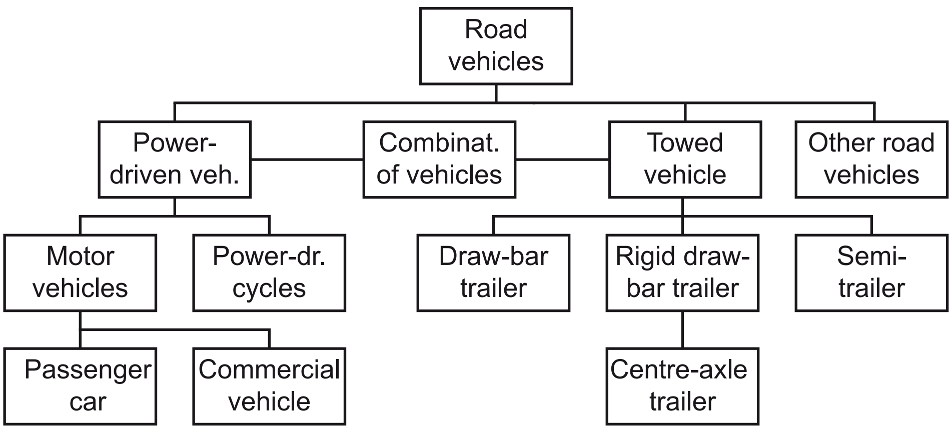


Figure 1.1.: Classification of road vehicles according to DIN 70010, [Nor01]

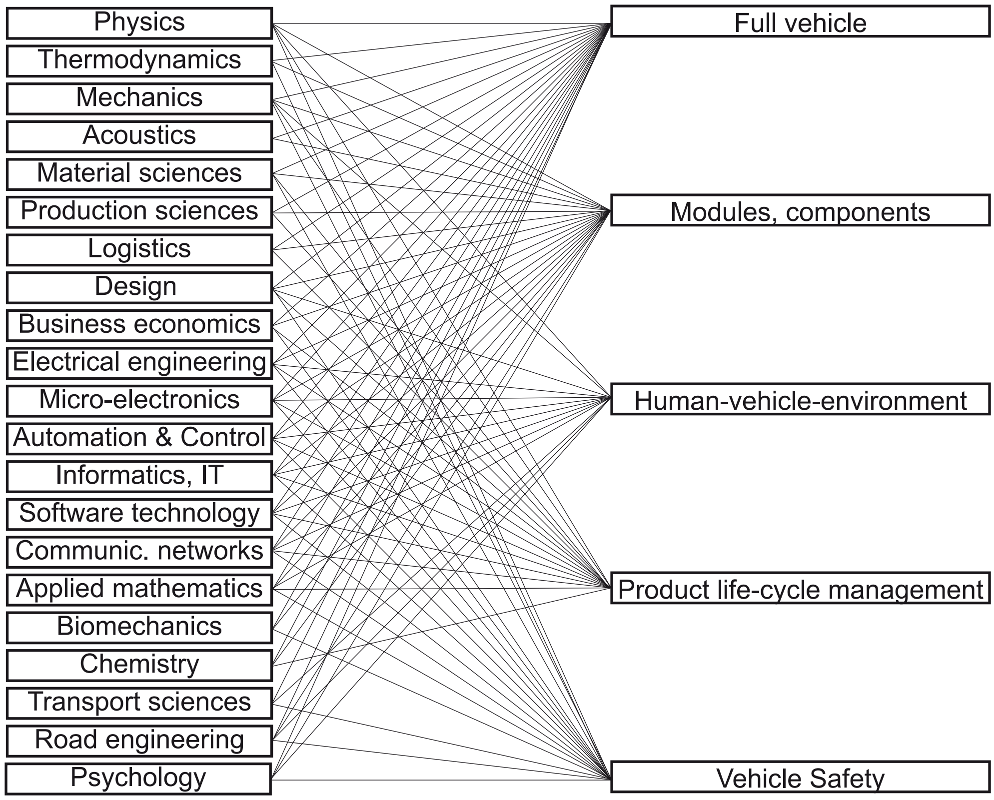


Figure 1.2.: Relationships between some important scientific subjects and applications in automotive engineering

- **Specialisation**
  The institute chooses a specific topic on which it focuses. This specialisation complements introductory courses and is the institute's main field of research.

- **Embedding in complementary courses**
  Frequently, the different scientific subjects mentioned in Fig. 1.2 are available at the university. Students gain their abilities in automotive engineering by attending courses at different institutes which are integrated into a major course. This is the current situation at the Graz University of Technology, with its specialised masters course "Automotive engineering and safety". Here, basic and specialised lectures at the core institutes Institute of Automotive Engineering and Vehicle Safety Institute are complemented with specialised courses from other institutes, [Gra07].

The following section summarises important aspects of automotive engineering to enable the primary goal of the automobile: *individual mobility*.

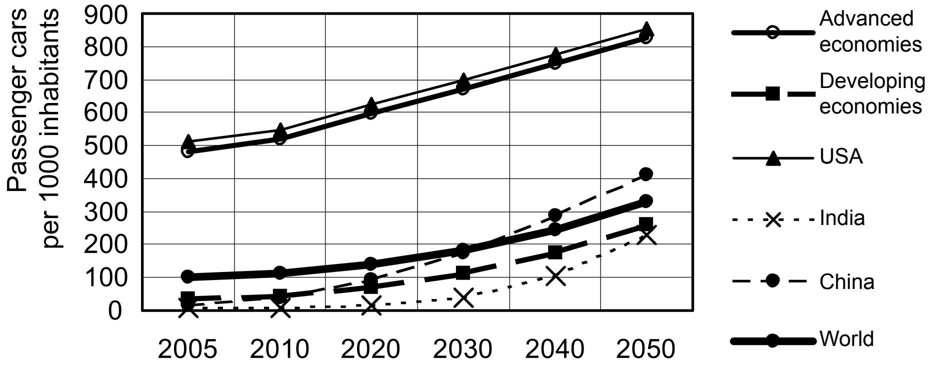


Figure 1.3.: Forecast in motorisation, adapted from [CMO08]

1.1. Mobility and motorisation

For covering distances, mobility is a basic need of the human being. Throughout history, mankind has continually improved mobility by developing advanced means of transportation. Future scenarios for the increase in world population and wealth through the industrialisation of emerging countries are based on sufficient supplies of energy and mobility. Energy consumption is expected to increase by 50% by 2030 and by 100% by 2050 [She09]. However, increasing wealth and income are related to growing distances travelled by human beings. The main factor contributing to this trend is individual mobility, in particular as made possible by passenger cars. Today, about one billion road vehicles are registered worldwide, which breaks down to 700 million passenger cars and 250 million commercial vehicles. A further increase to 1.5 billion passenger cars in 2030 and 2-3 billion in 2050 has been forecasted, [CMO08]. Fig. 1.3 shows the projected growth of individual traffic by the increase of passenger vehicles per 1000 inhabitants. This figure shows both the current domination by developed countries and the future increase caused by rising mobility in emerging countries. The challenge will be to ensure mobility while simultaneously managing the related consequences, which are related to energy resources, emissions and climate change, transport capacity and traffic safety.

1.2. Global traffic system

The road vehicle is part of a complex interacting system, which consists of the human driver, the environment and the vehicle itself, this interrelation is described in more detail in chapter 5. Research in automotive engineering focuses on the vehicle, nevertheless it is important to consider the whole system to improve the overall performance with respect to the challenges of today's mobility.

1.2.1. Vehicle, subsystems and components

Road vehicles consist of different subsystems or modules, which are usually delivered by specialised system suppliers (Tier-1 suppliers). The main task of the OEM in vehicle development is the integration of these subsystems to meet the product requirements on the full-vehicle level. The subsystems consist of components which can be delivered by further suppliers (Tier-2 suppliers). This section provides a brief overview of vehicle subsystems.

- **Vehicle Body**
  The vehicle body builds the trunk in the sense of a topological tree structure. The main parts are the structure and the interior.

  Structure. The vehicle body structure has to take up forces and moments during vehicle operation. It includes energy absorbing zones for collisions, carries the drive train, and provides the required interior space. The initial "frame" design (mounting the vehicle body on a frame) has been replaced by self-supporting unit vehicle bodies for passenger cars, whereas the frame type is still used in heavy goods vehicles and off-road cars, [STG07]. Today, the dominant design is a steel body made up of deep-drawn sheet metal connected by spot welds. To achieve lighter-weight designs, new technologies (e.g. high-strength steels or skeleton-type designs[2]) are becoming increasingly common.

  Vehicle interior. The vehicle interior represents an important interface between the vehicle and the human. Its design has to follow ergonomic requirements with respect to geometry and the human-machine interface. The geometric design of the interior has to respect anthropometry, seating positions, sufficient clearance and the feeling of the human in the vehicle. The human-machine interface has to support the driver in vehicle guidance. It is comprised of seat, instruments and vehicle control elements, [Gev07]. A further element of growing importance is *Infotainment*.

  As pointed out in chapter 5, the human driver performs three main tasks while driving: navigation, course planning and stabilisation. Modern navigation and communication systems seek to optimise support in the navigation task. In the future, dynamic driver information provided by improved Real Time Traffic Information (RTTI) will replace the current, rather static navigation technology. Both driver and passengers are increasingly equipped with multi-media solutions familiar from the personal computer. The design of the human-machine interface demands multidisciplinary development and the integration of psychology to adapt the technical system to the mental workload capacities.

- **Chassis**
  The chassis is the link between the road and the vehicle and determines driving behaviour and comfort, [HW09, Pau07]. It includes systems that generate or transfer

[2] An example is the AUDI space-frame concept.

applied forces and moments in the contact area between road and tyre. The main components are: wheels and tyres, brakes, suspension system, steering system and additional components[3].

The development of chassis components is supported by investigation of vehicle dynamics. Based on which vehicle motions are primarily considered, these dynamics are categorised as:

Longitudinal vehicle dynamics (Driving, climbing and braking performance)

Lateral vehicle dynamics (Driving behaviour)

Vertical vehicle dynamics (Comfort and dynamic wheel loads)

- **Drive train**
  The drive train generates and transfers propelling force from the propulsion unit to the tyre-to-road contact zone. In a conventional motor vehicle, it consists of an engine, a clutch or a hydrodynamic torque converter, a transmission with either fixed gear ratios or a continuous variable transmission (CVT), an axle drive with fixed gear ratio, and a differential transmission to compensate for the different wheel speeds of inner and outer wheels during cornering. Gaps between these components are closed by propulsion shafts. Occasionally, the transmission is complemented with All Wheel Drive (AWD) transmissions of different types.

  Currently, the dominant propulsion units are Otto- and Diesel-based internal combustion engines. These are gradually being joined by alternative power sources, which range from alternative fuels (e.g. natural gas, biodiesel and hydrogen) to hybrid technologies that combine electric and combustion engines and electric engines based on batteries or fuel cells, [Ger02]. These alternative power sources will be essential for managing the trade-off between increasing demand for mobility and the negative consequences of individual traffic.

- **Electric/Electronics (E/E)**
  Automotive E/E applications are related to engine and drive train, safety, comfort and communication. In [Rob07], the portion of E/E systems in the road vehicle's added value is assumed to have grown from 20% in 1995 to 35% in 2010. The electrification of mechanical and hydraulic systems is leading to mechatronic systems for vehicle dynamics control, Driver Assistance Systems (DAS), engine control, electric and hybrid drive trains and others. They complement conventional systems, such as lighting, ignition and starter engines. An automotive mechatronic system consists of sensors, Electronic Control Units (ECUs), actuators and the related connections for power supply and signal transmission.

[3]Pedals for brakes, clutch and throttle, steering column and wheel, vehicle dynamics control systems and Driver Assistance Systems

The next section, which is a summary of [Eck01, BS07], provides an overview of the historic evolution of the road vehicle and its components as well as an outlook on future trends.

1.3. Automotive history, present and future

In [Eck01], the history of the automobile is divided into the following phases: the prehistory of automobiles, the pioneering era (1885-1918), the automobile as an industrial product(1919-1945), the mass-produced automobile (1946-1979) and the automobile as a consumer good (1980 onwards).

1.3.1. The prehistory of automobiles

Mankind's first attempts to design means of transportation were characterised by the search for an appropriate power source, which was wind and muscle-powered (both by human and animal). The first reports of sledges, wheels and wagons dated back to the fourth millennium B.C. in different regions, such as in Iraq. The initial design of non-steered wagons with axles directly attached to the wagon evolved into hunting and war chariots. The art of wagon making spread to Europe around 2000 B.C. By 1000 B.C. the first designs of steered front axles appeared.

During the Roman Empire a road network was established to support expansion of the empire. A courier service with stations for horse changing was established around the birth of Christ. The Latin name “posita statio” is the origin for “postal” services. Road traffic stopped with the end of the Roman Empire, until the 15th century, when traffic again gained importance due to growing economies. The “coach” with a closed compartment and leather straps serving as suspension springs was probably invented in Hungary at this time. Cambered spoke wheels which prevented the wheels from running off the axle were introduced.

Steel-leaf springs appeared in the 17th century, which improved riding comfort. From the 17th century onwards, the development of coaches was mainly done in the United Kingdom, which had become the economic superpower of that era. The body design for coaches for human transport changed from a “frame” type to self-supported coachwork, while the frame type remained for commercial vehicles. The first braking systems with simple brake shoes were also introduced in the 17th century.

Scientific research in wagon building was initiated in the 18th century, the beginning of automotive engineering research. The ensuing centuries featured a search for an adequate power source, which began with Christiaan Huygens' (1629-1695) gunpowder machine and led to James Watt's (1736-1819) invention of the direct-acting steam engine. First developed in England, the steam engine led to the Industrial Revolution. Due to the size and weight of the steam engine, its initial use was limited to railways and ships. Nicolas

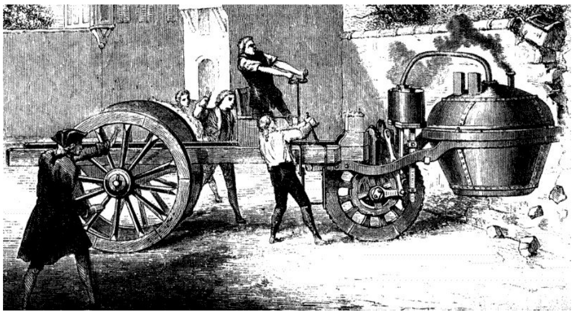


Figure 1.4.: The first historically mentioned traffic accident of the steam powered "Fardier á vapeur" of Nicolas Cugnot, reported 1804.

Cugnot (1725-1804) built the first self-propelled vehicle in 1769 with a smaller design of the steam engine. The improved "Fardier á vapeur" was said to have also experienced the first traffic accident in the form of a collision against a wall during a presentation in a military casern in 1771[4], Fig. 1.4. The assumed reason for the collision was the lack of vehicle safety systems - in that case, brakes.

The "Red Flag Act" of 1865, which limited the top speed of self-propelled vehicles to 6.4 kph, halted the evolution of vehicles in England. The further development of steam-powered engines was led by French engineers, who also developed the theoretical basis for combustion engines. Based on the work of Francois Isaac de Rivaz (1752-1828), Jean Joseph Etienne Lenoir (1822-1900) introduced a gas combustion engine, which he first installed into a vehicle in 1863. In 1876, Nikolaus August Otto (1832-1891) built a gas-fired, 4-stroke type single-cylinder engine, which was the basis of engines for self-propelled vehicles, [Sas62]. In 1886, the German Imperial Supreme Court decided against Otto and his attempts to defend his German Reich patent number 532 against other companies and thereby opened the way to fast developments in automotive engineering.

1.3.2. The pioneering era (1885-1918)

In Germany, at about the same time, Gottlieb Daimler (1834-1900) and his partner Wilhelm Maybach (1846-1929), as well as Carl Benz (1844-1929) were working on the challenge of designing a compact gasoline engine which would be suitable for road vehicles. The major problems were the preparation of a combustible fuel air mixture, ignition of the mixture and an engine running at higher speeds. They considered the four stroke principle, which was covered by Otto's patent at that time, the proper solution. "Surface carburetors" with floats to maintain a constant fuel level and air to fuel ratio, a timed vibrator type ignition (Benz) and an untimed hot-tube ignition solved the problems. Daimler and Maybach conducted successful test runs with a four-stroke gasoline engine in an experimental two wheeler (1885) and in a "motor-carriage" (1887), Fig. 1.5(a). In

[4]Mentioned in Cugnots obituary in 1804 with a later version of the vehicle.

(a) The Daimler/Maybach motor-carriage of 1887, [Hup10]
(b) The Benz patent motorcar of 1886, [Deu10]

Figure 1.5.: The first self-propelled road vehicles in automotive history

1886, Benz made his historic first test run in Mannheim, Germany with a self-propelled gasoline road vehicle, Fig. 1.5(b).

However, after gaining inspiration from the German automobiles in the 1889 Paris world exhibition, France again took the lead in the development of automobiles. The French company Panhard & Levassor introduced a drive train concept with a front engine and rear wheel drive, and brothers André and Edouard Michelin introduced pneumatic tyres in a Peugeot vehicle in 1895. The tyre, which had been invented by Robert William Thomson in 1845 (without further automotive application), was a breakthrough in automotive history and allowed for better riding comfort and higher velocities. John Boyd Dunlop (1840-1921) patented an air filled bicycle tyre in 1888, which led to bicycle races on the comparably good French roads, and later to the first automobile race in 1894 from Paris to Rouen. Meanwhile, Albert de Dion and Goerges Bouton concentrated on the development of gasoline engines and road vehicles and became an important engine and automobile manufacturer around 1900, with about 75 manufacturers in France alone.

Inspired by the French automobiles, several German companies switched from bicycle to automobile production, and automobiles were soon being manufactured in many European countries (except England, which still suffered under the effects of the Red Flag Act). The Austrian engineer Siegfried Marcus (1831-1898), who until 1968 was falsely considered the father of the automobile, introduced a magneto ignition in his Marcus car of 1889. It was similar to the ignition designed by Robert Bosch (1861-1942), which Bosch had successfully employed in a vehicle in 1897. Ferdinand Porsche (1875-1951) patented an in-wheel electric drive train and built the first hybrid drive train with the Austrian manufacturer Lohner in 1900. Despite the progress in automotive development, the social impact of automobiles was small at that time, since they were not affordable for people of the working and middle classes. In the following years, the European automobile development concentrated on the technological challenges. Maybach built the

Mercedes[5] 35 HP, a race car which carried many important inventions that were copied by other companies in the ensuing years.

The situation in USA was different from Europe. In 1895, there were only 80 automobiles in the country (50% gasoline, 17% electric and 17% steam propelled). Other power sources, such as spring power, acetylene and compressed air, were also used. In fact, steam and electric vehicles such as the Studebaker Electric or the Detroit Electric Brougham were in favour until about 1900. In 1879, the attorney George Baldwin Selden (1846-1922) filed a patent for an "auto buggy", which he did not build until 1906. The patent turned out to be a barrier for American automobile development until it was voided in 1911 by the Supreme Court.

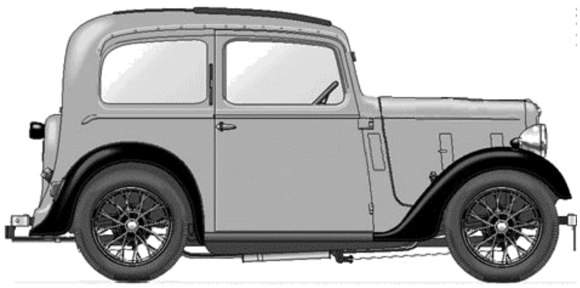
In contrast to Europe, the American development concentrated more on mass motorisation rather than on innovative technology. The automobile soon became one of the most complex industrial products, and Henry Ford (1863-1947) introduced his Ford Model T in 1908, Fig. 1.6(a). Due to high market demand, he adopted the assembly line concept in 1913 to cope with the product complexity and the need for cheaper vehicles. Starting with 200,000 units in 1913, about 1.8 million Model T's were produced by 1923. Henry Ford also viewed his employees as consumers and abandoned the typical industrial exploitation of workers. He paid better salaries and reduced the price of the Model T to make it affordable.

Other manufacturers followed Ford and introduced innovations to compete with Ford, who did not improve the Model T for 19 years, when he had to stop production in 1927. Mass production made the USA the number one world manufacturing country. This era was accompanied by the founding, merging and failure of many companies, such as General Motors, which was founded by William Crapo Durant (1861-1947) in 1908.

The development of gasoline-propelled commercial vehicles lagged behind the passenger car. In 1893, Panhard & Levassor built the first vehicle of this kind called the "camionette a pétrole", but steam powered vehicles proved to be more effective in the beginning. One pioneer was Heinrich Büssing (1843-1929), who established a plant for commercial vehicles in Braunschweig, Germany. The driving force for the development of trucks and busses were transportation firms in London that wanted to replace horse-drawn transport with automobiles. In addition, postal services were established, for example between Bad Tölz and Lenggries (Germany) in 1905. Nevertheless, the initial German leadership in commercial vehicle development was soon taken over by France and England, due to their larger market demands. The demand for transport during World War I stimulated the commercial vehicle business and led to the formation of companies like MAN, Magirus and others.

[5] The "Mercedes" brand was introduced for marketing reasons.

(a) Ford Model T ("Tin Lizzy") of 1924, [vB10]



(b) Austin 7 of 1935, [vB10]

Figure 1.6.: Popular cars in the early era of motorisation

1.3.3. The automobile as an industrial product(1919-1945)

World War I brought the European automotive industry to a standstill, whereas in the USA the development of motor vehicles continued. European manufacturers continued to rely on hand-made parts and components produced in-house, while in the USA a supplier industry was introduced which is analogous to today's production processes. After 1925, American brands exported vehicles to Europe, built up factories in Europe and bought European manufactures such as Vauxhall (1925) and Opel (1929), which were bought by GM. Edward Gowen Budd (1870-1946) introduced stamped steel members, which replaced the wooden frame-work and led to improved vehicle safety, higher torsional stiffness, shorter assembly times, simpler painting processes and drastic cost reductions. In 1931, Chrysler introduced "floating power engine mounts", which solved vibration and noise problems from the previous rigid attachment of the drive train.

André Citroën (1878-1935), Wilhelm von Opel (1871-1948), William Richard Morris (1877-1963), Herbert Austin (1866-1941) and Giovanni Agnelli (1866-1945) were the pioneers who had revived the European automotive industry by imitating Henry Ford's production methods. The first such automobile was the Citroën Type A model of 1919, and one of the most successful models was the Austin seven four seater, built between 1922 and 1939, Fig. 1.6(b). These first European models were the beginning of the small car concept, in contrast to the larger US models, which consumed more fuel.

After 1930, American dominance ended, and the USA and Europe were about equal. In particular, German production increased, due to Hitler's support of the automotive industry, which he considered suitable for his intentions. Hermann Föttinger (1877-1945) undertook successful test drives with a hydrodynamic transmission in a Mercedes in 1936, the Zahnradfabrik Friedrichshafen introduced silent helical-cut transmissions in 1929, and Mercedes introduced the independent front suspension in 1933. Independent suspensions improved ride comfort and driving behaviour and lowered the vehicle body.

In 1934, Earl S. McPherson (1891-1960) joined GM and invented the McPherson suspension by combining suspension spring and shock absorber. However, GM rejected the idea, and McPherson patented it for Ford in 1949, which introduced it in the Ford Consul. With models like the Lancia Lambda (1923) and Citroën Traction Avant (1934), frame-on-chassis design was replaced by self-supporting unit body construction. One important milestone was the improvement of roads with asphalt, paving blocks and concrete. Hitler, with his Reichsautobahn project, was pushing the German automotive industry. He also founded the Volkswagen factory and ordered the Porsche design bureau to design a passenger car for a larger customer group, the Volkswagen KdF. The KdF could only be ordered in advance by regularly pasting stamps into savings books. However, due to World War II, the civilian production of the KdF was stopped, and military vehicles were produced in the plant financed by those savings.

Commercial vehicles were improved through the introduction of shaft drives, pneumatic tyres, rear tandem axles (Goodyear, 1920 and Büssing, 1923) and hydraulic brakes (Krupp, 1926). However, the largest improvement was introduced by Rudolf Diesel (1858-1913) with the diesel engine. The first working experimental engine was built in 1897, but it took a long time until innovations such as injection pumps, governors and nozzles were introduced and solved the initial problems. By the end of the 1930s, diesel-propelled commercial vehicles above 3.5 tons dominated the European market, whereas American trucks continued to feature gasoline engines.

In World War II, the horse lost its importance as a means of transport and was replaced by vehicles. Various vehicles of different designs and operation fields were developed. Based on the KdF, Porsche designed the Volkswagen Type 62 and 82 (“Kübelwagen”) all-terrain vehicles, which, together with the corresponding American Jeep, were the most reliable vehicles. Passenger cars and trucks, artillery tractors, armoured scout cars, halftracks and other vehicles with improved off-road performance were introduced by the German regime, and others soon followed after the success of German “Blitzkrieg” tactics.

1.3.4. The mass-produced automobile (1946-1979)

After World War II, Europe recovered much more slowly than the USA, whose plants had not been damaged by the war. The public wealth in the USA led to larger cars than before, whereas Europe favoured small and mid-size cars. GM introduced glamorous automobile exhibitions to prepare the customers for the styling of future “dream car” models.

In 1959 middle-class sedan cars featured up to 240 hp. In Europe, micro-cars appeared, such as the ISO Isetta of 1953, but were soon replaced by the huge success of the Volkswagen Beetle, Fig. 1.7(a), and others. In 1955 the Citroën DS introduced a central hydraulic system for brakes, clutch, transmission and steering; it featured disc brakes and the hydropneumatic suspension, Fig. 1.7(b). The DS technology, together with its futuristic styling, made the model ahead of its time.

In 1948 Heinrich Nordhoff (1899-1968) became director of Volkswagen. With the introduction of customer service, steady improvements in quality and model development, rational manufacturing methods and universal applicability, Nordhoff made both the Beetle and the Volkswagen brand into one of the most successful ones. The economic success led to large exports to the USA and forced US manufacturers to imitate the concept. However, these US manufacturers did not realise that the success of the Beetle was due to its economy and reliability, and US imitations (e.g. the Chevrolet Corvair, 1960-69) therefore amplified the weaknesses of the design (a tendency to oversteer, missing of on-centre stability, sensitivity to crosswinds). The attorney Ralph Nader started a consumer campaign accusing the automotive industry of not improving the driving behaviour of these cars for profit reasons, [Nad65], which led to the creation of the Federal Motor Vehicle Safety Standards (FMVSS). The sales of the Corvair fell to 7% within four years.

Other competitors of the Beetle which maintained the standard FWD layout were the Citroën 2CV, which introduced radial tyres in 1949 as standard equipment, and the Mini (1959). The Mini, designed by Alec Issigonis (1906-1988), featured innovations such as a transverse four-cylinder engine, a transmission that shared the crankcase with the crankshaft, and an independent suspension. It provided space for four occupants despite its length of 3.05 m. The Mini concept became a standard that was followed by most of the today's built automobiles. The NSU Ro 80, presented in 1967, featured a revolutionary aerodynamic design and a Wankel type propulsion unit.

The negative consequences of motorisation (air pollution) were first recognised in American cities, however the automotive industry ignored this factor. In 1968, a law limiting exhaust emissions was passed in California. Innovations such as positive crankcase ventilation, exhaust gas recirculation and catalytic converters were introduced. Volkswagen featured electronic fuel injection (Bosch D-Jetronic) in the 1967 VW 1600 to comply with the US regulations. Driven by the first oil crisis in 1973, the USA instituted fuel-consumption requirements which had to be met by the American car models. The manufactures had to react on short notice with weight reduction, introduction of diesel engines and the development of new concepts, which was achieved through joint-ventures with Japanese competitors.

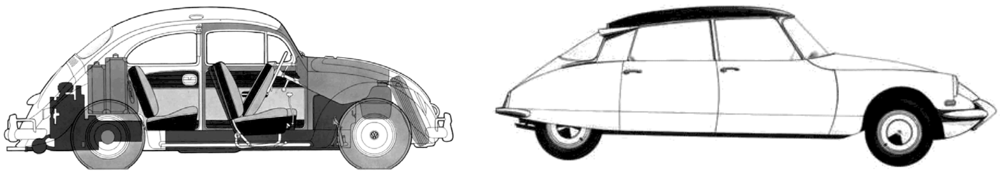
A second negative consequence was traffic safety. By 1965, US traffic fatalities had risen to about 50,000 per year. In 1965, the Oldsmobile Toronado was the first car to feature negative steering offset for course stabilisation. Knee bolsters and safety steering columns were introduced by the vehicle safety Pioneer Béla Barényi (1907-1997), who also patented the crush zone concept [Bar51], which has become state-of-the-art in today's passenger cars. Air suspension systems were first introduced by US manufacturers in 1957 and later refined into more reliable systems by Borgward and Mercedes (1960 and 1961 respectively). Although invented for other applications, the safety belt was introduced as an option in Nash (1949) and Ford (1955) automobiles, and as standard equipment by Saab in 1958. Based on previous designs, Nils Bohlin (1920-2002) invented the modern three-point seat belt, which was introduced by Volvo in 1959. Based on a 1953 invention by John W. Hetrick [Het53], Ford (1971) and GM (1973) introduced fleets with experimental airbags, and in 1974 GM introduced the Air Cushion Restraint System (ACRS) for the first time as an option in the Oldsmobile Toronado. The initial problems of reliably sensing frontal crash incidents and fast deployment times led to the withdrawal of the ACRS.

Between 1961 and 1969, most European brands and some American brands replaced RWD with FWD. In order to maintain its position as the largest European car manufacturer, Volkswagen introduced the Audi 80 and VW Passat in 1972 and 1973. It featured a semi-beam rear axle, braking circuits which were diagonally split, floating caliper disc brakes and a vehicle body which was designed with the aid of CAE methods. These models formed the basis of other VW models, such as the Golf (introduced in 1974). The Audi 50 (introduced in 1974 and then marketed as the VW Polo from 1975 onwards) featured safety measures such as collapsible longitudinal beams to form a crush zone, a rigid perimeter around the floorpan for side impact protection, and an instrument panel designed for head protection.

It took only 15 years to bring the Japanese automotive technology up to the same level as in Europe and the USA. Before 1961, Japanese manufacturers were imitating products such as the Austin 7 or Renault 4CV. After that, the Japanese began to develop cars of their own design, originally just for the Japanese market. Production methods were improved in terms of automation, and quality methods were introduced in the overall process. Lean production with a high level of out-sourcing and just-in-time deliveries of components and subsystems were the key elements of this fast progress. New vehicle categories appeared, such as the AWD pick-up (Subaru, 1978) and the mini-van (Mitsubishi Super Station Wagon, 1979). The success of the Japanese cars led to import limitations in Italy, France, UK, Spain, Portugal and the USA. Germany, which did not limit imports, profited by selling more cars. The Japanese reacted by manufacturing vehicles locally and developing more expensive cars, since the import limits were based on units and not on value. South Korean and South American manufacturers then took the opportunity and entered the smaller car market that had been partly vacated by Japan.

1.3.5. The automobile as a consumer good (1980 onwards)

Since World War II, the economic and social status of people, particularly in Europe, has continually improved. The road vehicle is increasingly becoming part of everyday life, as the cost of purchasing and operating a vehicle have become affordable for the majority. A reliably functioning vehicle and safe driving behaviour, even for inexperienced drivers,



(a) VW Beetle 1200 of 1954, [vB10]

(b) Citroën DS 19, [vB10]

Figure 1.7.: Cars in the era of mass motorisation

are considered a matter of course.

The role of electronics in fulfilling the complex demands of modern automobiles has grown steadily. Conventional coil ignition was first replaced by braker-controlled and later by electronic ignition. In 1978, the Citroën Visa introduced a braker-less semiconductor ignition, and in 1979 Bosch equipped the BMW 732i with Motronic, the first motor management system that improved emissions and fuel consumption. Today, electronics are found in transmission controls, governors for diesel engines, and injection systems that are replacing carburetors.

Driving behaviour and primary vehicle safety are improved by mechatronic systems, such as anti-lock braking systems, traction control, vehicle dynamics control, tire pressure monitoring, active roll stabilisers, actively controlled suspensions systems, electromechanic brakes and others. Numerous applications in driver information and Advanced Driver Assistance Systems (ADAS) are being introduced, ranging from car radio to Predictive Brake Assist systems. Communication between the road infrastructure and vehicles is the subject of current research and will bring many innovations for a comfortable and safe driving experience. Local area networks, such as the CAN bus, MOST, TTP and Flexray, help balance the demand for large amount of wiring harness.

The number of automotive requirements has risen steadily due to legal requirements, consumer protection tests and manufacturers' in-house requirements developed by analysing the needs of their target customers. [Pau07] lists the following main requirements:

- Vehicle Safety
- Comfort
- Quality
  - Reliability
  - Conservation of value
  - Look-and-feel
- Emissions

- Noise
- Recycling
- Driving performance and transport capacity
- Costs
  - Purchase
  - Operation
  - Maintenance
- Integration in overall traffic

These requirements are often at odds with each other. For example a high level of vehicle safety requires safety systems that are not used in standard vehicle operation and increase weight, costs and emissions. The customer determines the importance of the requirements, and since each customer is different, manufacturers offer different automotive solutions. Fig. 1.8 shows the evolution of vehicle categories from the sixties to today.

Even within the same make and model, a high level of individualisation is possible today. Whereas Henry Ford's Model T maintained the same design and black colour, today's customers can choose between colours of the vehicle outer surface and interior, engine type, seat types, special equipment and others. For commercial trucks, customisation has led to a situation in which basically no two vehicles are exactly alike. On the other hand, the innovation cycle of road vehicles has been significantly decreased in the last decades. A fast time-to-market is the key to survival in the rapidly changing automotive industry, which is characterised by globalisation and requirements for sustainable mobility. To meet these demands, manufacturers have adopted development processes such as platform strategy[6], carry over parts[7], construction kits[8], badge engineering[9] or production in licence[10].

Another important answer to decreasing innovation cycles is the increased use of Computer-Aided methods (CAx). Modelling and simulation, as well as the computation hardware and software tools, are being rapidly developed to cover new fields of application. CAx methods are being used in earlier stages of the design process, which enables shorter innovation cycles and better product quality (a process known as "front-loading"). Fig. 1.9 provides a qualitative depiction of the reduction in total development cost made possible by limiting the number of prototypes and prototype generations, as

[6] Components such as engine, drive train, chassis, main vehicle body structure, tank, exhaust, system, seat and cable harness form a platform where a different "hat" is placed upon. The hat includes the vehicle body without the platform related parts.

[7] The same part is used in different vehicle models.

[8] A vehicle subsystem such as the seat is built from standardised components fitting together.

[9] The same vehicle model with minor stylistic changes is sold at different brands.

[10] An OEM sells a phased out model to another OEM including tooling and manufacturing processes.

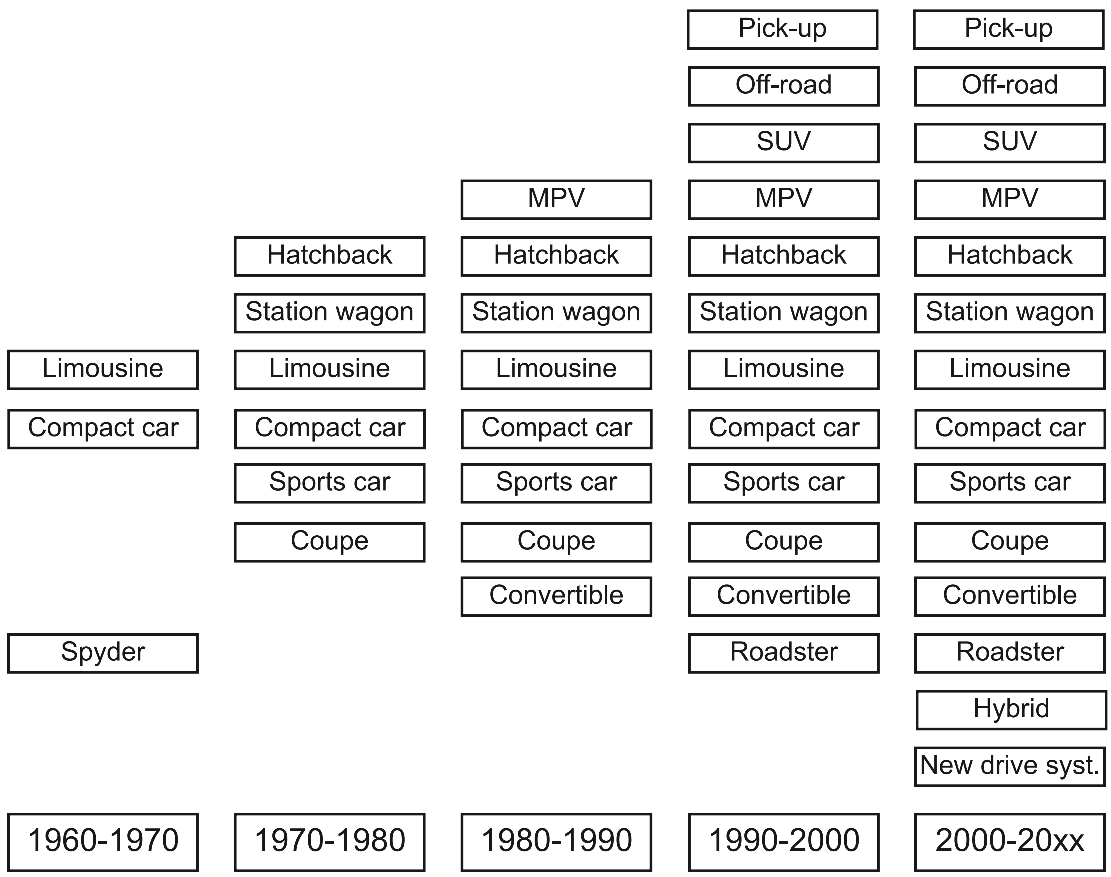


Figure 1.8.: Vehicle categories, source: MAGNA STEYR Fahrzeugtechnik

well as by cost savings due to a decrease in the number of particularly costly, late-stage design changes. This leads to a higher product maturity in earlier phases, and the start of production (SOP) is shifted towards a faster time-to-market.

1.3.6. The automotive future: Sustainable mobility

As pointed out above, the future challenge is to balance the demands for increased mobility and need to manage the related negative consequences. In [Fro09], the following primary conclusions about the evolution of future vehicle technologies are drawn:

- Changes in legislation (safety and emissions) and socio-economic aspects (urban infrastructure, ageing population) will influence the introduction of new vehicle models, especially in Europe.
- In the high-volume market, there will be a conflict between the need for light-weight vehicles for lower fuel consumption and the introduction of new functionalities and

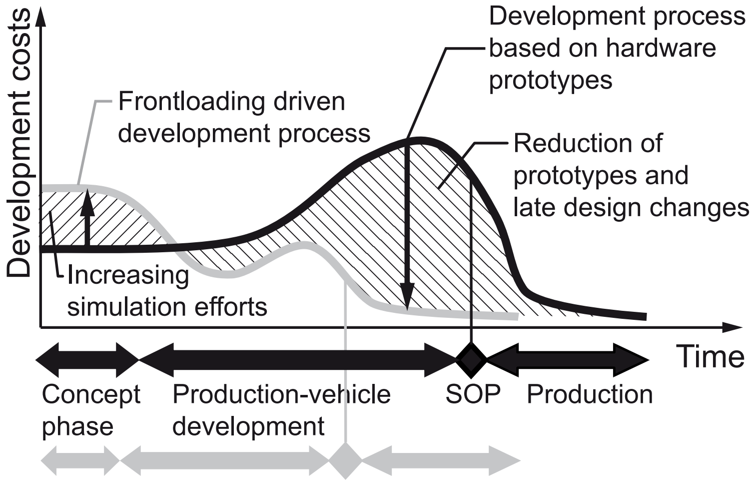


Figure 1.9.: Qualitative description of frontloading driven development processes

innovations.

- Vehicle technologies will be driven by consumer demands for safety and comfort and will be accommodated by an OEM brand philosophy.
- The key technologies for the future will be alternative power trains (e.g. electric vehicles) and less distracting and more affordable human-machine interfaces.

Important expected trends for future automotive technologies described in [Fro09] are:

- **Drive train.** A larger penetration of diesel engines is expected in the near future, along with a concentration on direct gasoline injection, downsizing of engine capacity, improved combustion, advanced turbo charging technologies and exhaust after treatment. Micro-hybrid systems will grow faster than other hybrid drive train concepts. These other hybrid concepts, together with other alternative power trains (e.g. hydrogen or electric engines), are not expected to dominate over the combustion engine before 2020. Automated manual transmission and dual clutch transmission are expected to be more important than manual transmission, due to comfort and fuel-consumption advantages.
- **Chassis.** Mechatronic systems for improved driving behaviour and primary safety will be promoted. The main technologies will be advanced ESC, which will become mandatory in Europe, active and semi-active suspensions systems, which will dominate all segments except the low-cost segments, X-by-Wire systems and light weight design. Electric power steering and anti-lock braking systems will enter almost all passenger car segments. Legal regulations will cause tire pressure monitoring systems and conventional Brake Assist(BA) to become standard equipment.

- **Vehicle safety.** The avoidance of collisions through Advanced Driver Assistance Systems is expected to be one of the most important technologies. In the near future, the focus will be on vehicle-based systems (e.g. Automatic Cruise Control and Predictive Brake Assist), which will later be complemented by vehicle-to-vehicle and vehicle-to-infrastructure communication. The variety of accident causes is being addressed by a growing number of ADAS, a topic which the following habilitation thesis describes in detail. Primary and secondary safety system will be integrated to provide enhanced functionalities. Secondary safety will concentrate on advanced restraint system, improved energy absorption coupled with lightweight design, and new concepts for the safety of electric vehicles.
- **Electric and Electronics.** In addition to the growth of mechatronic systems, a decreased separation between automotive and standard consumer electronics is expected to trigger the introduction of multi-media systems for driver information and passenger entertainment. Navigation based on digital maps and real-time traffic information will divert traffic by guiding vehicles on different routes. The European ecall initiative will provide faster crash notification and injury treatment. Advanced Human-Machine Interfaces (HMI) systems, such as voice recognition and touch screens, complemented with innovative lighting technologies, such as active front lighting, are expected to provide improved comfort and convenience. New bus systems (MOST, LIN, Flexray), which will enhance the current standard CAN bus technology, will help meet increasing demands for speed and bandwidth of data transmission.

Anyway, it should be mentioned that forecasts like the above should be considered more or less unreliable. It will be essential for the success of automotive companies to follow the right strategy.

2. Breadth of the subject automotive engineering

Scientific research in automotive engineering is covered by a large number of academic institutes. The largest centres for scientific automotive research have historically been based in Germany, the USA, Japan, France, Italy, the UK and Korea. In the USA alone, several dozen universities and colleges have research areas related to the automobile. However, automotive research is not limited to universities. The research departments of manufacturers and suppliers, universities of applied sciences, and research centres (e.g. the *Fraunhofer* institutes in Germany[1], TNO[2] in The Netherlands, VTI[3] in Sweden, the *Centro Ricerche FIAT*[4] in Italy or the *Virtual Vehicle Research and Test Center*[5] in Austria, to name a few) also make important contributions. The following section summarises only the German-speaking universities, which are members of the *Wissenschaftlichen Gesellschaft für Kraftfahrzeug- und Motorentechnik e.V. (WKM)*[6]:

- **RWTH Aachen**

  Institut für Kraftfahrwesen[7], headed by Prof. L. Eckstein. The institutes covers the complete area of automotive engineering with research areas in chassis, body, drive train, electronics, acoustics, driver assistance and strategy/process Development.

  Lehrstuhl für Verbrennungskraftmaschinen[8], headed by Prof. S. Pischinger. The Institute covers engine topics like innovative engine constructions and R&TD of efficient, clean combustion processes.

- **Technische Universität Berlin**

  Fachgebiet Verbrennungskraftmaschinen[9], headed by Prof. B. Wiedemann. The institute engages in basic and application research related to the optimisation of efficiency and power density while maintaining low costs for production and maintenance, as well as drivability.

[1] Available at http://www.fraunhofer.de, accessed on 23 January 2010
[2] Available at http://www.tno.nl, accessed on 15 April 2010
[3] Available at http://www.vti.se, accessed on 15 April 2010
[4] Available at http://www.crf.it, accessed on 23 January 2010
[5] Available at http://www.vif.tugraz.at, accessed on 23 January 2010
[6] Available at http://www.wkm-ev.de, accessed on 15 April 2010
[7] Available at http://www.ika.rwth-aachen.de, accessed on 7 January 2010
[8] Available at http://www.vka.rwth-aachen.de, accessed on 7 January 2010
[9] Available at http://www.vkm.tu-berlin.de, accessed on 7 January 2010

Fachgebiet Kraftfahrzeugtechnik[10], headed by Prof. V. Schindler. The Institute focuses on full-vehicle concepts, secondary vehicle safety, energy resources and development methods.

- **Ruhr-Universität Bochum**

  Lehrstuhl für Verbrennungsmotoren[11], headed by Prof. W. Eifler. The institute focuses on test benches for engine technology.

- **Technische Universität Braunschweig**

  Institut für Fahrzeugtechnik[12], headed by Prof. F. Küçükay. The institute concentrates on the fundamental topics of automotive engineering with applications in drive train, chassis, body, brakes and tyres.

  Institut für Verbrennungskraftmaschinen und Flugtriebwerke[13], headed by Prof. P. Eilts. The main topics of the institute are the improvement of the combustion process and mixture preparation in gasoline and Diesel engines.

- **BTU Cottbus**

  Lehrstuhl für Fahrzeugtechnik und -antriebe[14], headed by Prof. P. Steinberg. The focus of the institute is thermal management, lubrication, engine technology and the interaction between driver, vehicle and environment.

- **Technische Universität Darmstadt**

  Fachgebiet Fahrzeugtechnik[15], headed by Prof. H. Winner. The institute focuses on the realisation of new potentials for vehicle safety through mechatronic systems. The specific topics are Driver Assistance Systems, motorcycle safety and the mechatronically controlled chassis.

  Fachgebiet für Verbrennungskraftmaschinen[16], Prof. C. Beidl (following Prof. G. Hohenberg). The institute focuses on experimental test methods, exhaust aftertreatment, development methods for engines and research on hybrid drive trains.

- **Technische Universität Dresden**

  Lehrstuhl für Verbrennungsmotoren[17], headed by Prof. H. Zellbeck. The main research topics are the transient behaviour of combustion engines and energy management on the full-vehicle level.

[10] Available at http://www.kfz.tu-berlin.de, accessed on 7 January 2010
[11] Available at http://www.rub.de/lvm, accessed on 7 January 2010
[12] Available at http://www.iff.tu-bs.de, accessed on 7 January 2010
[13] Available at http://www.ivb.tu-bs.de, accessed on 7 January 2010
[14] Available at http://www.tu-cottbus.de/fahrzeugtechnik, accessed on 7 January 2010
[15] Available at http://www.tu-darmstadt.de/fzd, accessed on 7 January 2010
[16] Available at http://www.verbrennungskraftmaschinen.de, accessed on 7 January 2010
[17] Available at http://tu-dresden.de/iad, accessed on 7 January 2010

Lehrstuhl für Fahrzeugmechatronik,[18] headed by Prof. B. Bäker. This institute conducts research on energy and information management, E/E architectures and safety-relevant mechatronic systems, and test and evaluation of E/E systems.

Lehrstuhl für Kraftfahrzeugtechnik,[19], interim head Prof. H. Zellbeck. The main topics are driving behaviour and comfort on full-vehicle level, concepts, technologies and components of vehicles.

- **Helmut-Schmidt-Universität / Universität der Bundeswehr Hamburg**

  Institut für Fahrzeugtechnik und Antriebssystemtechnik, Antriebe[20], headed by Prof. W. Thiemann. The institute focuses on specialised topics in engine technology such as evaporation of oil in the combustion chamber and exhaust gas measurement and analysis.

  Institut für Fahrzeugtechnik und Antriebssystemtechnik, Fahrzeuge[21], headed by Prof. M. Meywerk. Seven chairs related to automotive engineering have joined to cover topics in acoustics, drive train, CAE-methods, chassis, production science and driving simulators.

- **Universität Hannover**

  Institut für Technische Verbrennung[22], headed by Prof. F. Dinkelacker. The main topics of the institute are related to combustion technology. Future topics will involve multi-jet injection and alternative fuels.

- **Technische Universität Ilmenau**

  Fachgebiet Kraftfahrzeugtechnik[23], headed by Prof. K. Augsburg. The research areas of the institute are related to chassis and braking technology, objective and subjective investigations of the driver-vehicle interface, internal combustion engine technology, development of test facilities and acoustic investigations of gears.

- **Universität Kaiserslautern**

  Lehrstuhl für Verbrennungskraftmaschinen[24], headed by Prof. R. Flierl. The institute focuses on special topics in engine technology, such as valve trains, CFD simulation and catalytic converters.

- **Universität Karlsruhe**

  Institut für Kolbenmaschinen[25], headed by Prof. U. Spicher. The institute has two main research areas: engine technology (with a focus on combustion) and pump technology.

[18] Available at http://tu-dresden.de/iad, accessed on 7 January 2010
[19] Available at http://tu-dresden.de/iad, accessed on 7 January 2010
[20] Available at http://www.hsu-hh.de/thiemann, accessed on 8 January 2010
[21] Available at http://www.hsu-hh.de/meywerk, accessed on 8 January 2010
[22] Available at http://www.itv.uni-hannover.de, accessed on 8 January 2010
[23] Available at http://www.tu-ilmenau.de/site/kft/, accessed on 8 January 2010
[24] Available at http://www.mv.uni-kl.de/vkm/, accessed on 8 January 2010
[25] Available at http://www-ifkm.mach.uni-karlsruhe.de, accessed on 8 January 2010

Institut für Fahrzeugtechnik[26], headed by Prof. F. Gauterin. The institute focuses on methods and processes for the complex development of automobiles. A second focus is on tyre research.

- **Otto-von-Guericke-Universität Magdeburg**

  Institut für Maschinenmesstechnik und Kolbenmaschinen[27], headed by Prof. H. Tschöke. The main research topics are Otto and Diesel engine technology including modelling and simulation, alternative fuels, pumps and compressors, exhaust aftertreatment and measurement techniques in acoustics.

- **Technische Universität München**

  Lehrstuhl für Fahrzeugtechnik[28], headed by Prof. M. Lienkamp (following Prof. B. Heißing). The manifold research areas are processes and tools in the automotive development process, driver assistance and control systems, vehicle dynamics and design, mobile off-road machinery, and the interaction between driver and vehicle.

  Lehrstuhl für Verbrennungskraftmaschinenfootnote[29], headed by Prof. G. Wachtmeister. The institute focuses an alternative drive train, engine technology (Otto, Diesel, gas and hydrogen), hybrid drive trains, exhaust aftertreatment and modelling/simulation.

- **Universität Rostock**

  Lehrstuhl für Kolbenmaschinen und Verbrennungsmotoren[30], headed by Prof. H. Harndorf. The three main research topics are alternative fuels, combustion and large diesel engines.

- **Universität Stuttgart**

  Institut für Verbrennungsmotoren und Kraftfahrwesen, Verbrennungsmotoren[31], headed by Prof. M. Bargende. The institute focuses on combustion processes, exhaust gas analysis, engine acoustics and mechanics.

  Institut für Verbrennungsmotoren und Kraftfahrwesen, Kraftfahrwesen[32], headed by Prof. J. Wiedemann. The research areas of the institute are vehicle aerodynamics and thermal management, vehicle acoustics and vibration, automotive technology and driving dynamics, interdisciplinary projects, high-performance computing and wind tunnel operation.

[26] Available at http://www.kfzbau.uni-karlsruhe.de, accessed on 7 January 2010
[27] Available at http://www.uni-magdeburg.de/ims/, accessed on 8 January 2010
[28] Available at http://www.ftm.mw.tum.de, accessed on 7 January 2010
[29] Available at http://www.lvk.mw.tum.de, accessed on 8 January 2010
[30] Available at http://www.lkv-rostock.de, accessed on 8 January 2010
[31] Available at http://www.ivk.uni-stuttgart.de, accessed on 7 January 2010
[32] Available at http://www.ivk.uni-stuttgart.de, accessed on 8 January 2010

Institut für Verbrennungsmotoren und Kraftfahrwesen, Kfz-Mechatronik[33], headed by Prof. H.-C. Reuss. The institute focuses on automotive electronics and related software development.

- **Technische Universität Wien**

  Institut für Verbrennungskraftmaschinen und Kraftfahrzeugbau[34], headed by Prof. B. Geringer. The main research topics are combustion processes, exhaust aftertreatment and the development of engine components.

- **Technische Universität Graz**

  Institut für Verbrennungskraftmaschinen und Thermodynamik[35], headed by Prof. H. Eichlseder. Research focuses on engine research (working process, simulation and analyses, design, combustion systems), thermodynamics and emission research.

  Institut für Fahrzeugtechnik[36], headed by Prof. W. Hirschberg. The core competencies of the institute are vehicle dynamics in theory and experiment, tyre research and modelling, vehicle simulation (vehicle, control systems, roadway, driver), alternative drive trains, chassis and suspension technology, commercial vehicle technology, driver assistance systems, integrated safety and parametrised CAD design.

  Institut für Fahrzeugsicherheit,[37] headed by Prof. H. Steffan. The institute's research areas are accident reconstruction, biomechanics and development methods for vehicle safety (experimental and simulation). Research on innovative vehicle safety system complements these activities.

- **ETH Zürich**

  Laboratorium für Aerothermochemie und Verbrennungssysteme[38], headed by Prof. K. Boulouchos. The institute focuses on the fundamentals of chemically reactive flows, analysis of single-phase and multi-phase questions, and application-oriented optimisation of combustion systems.

[33] Available at http://www.ivk.uni-stuttgart.de, accessed on 8 January 2010
[34] Available at http://www.ivk.tuwien.ac.at, accessed on 8 January 2010
[35] Available at http://fvkma.tu-graz.ac.at, accessed on 8 January 2010
[36] Available at http://www.ftg.tugraz.at, accessed on 8 January 2010
[37] Available at http://www.vsi.tugraz.at, accessed on 8 January 2010
[38] Available at http://www.lav.ethz.ch, accessed on 8 January 2010

3. Contributions of the author to the subject automotive engineering

Teaching at an University of technology is research based. Graduates need not only state-of-the-art knowledge, but must also be aware of the latest research results. This is especially true in the rapidly developing subject of automotive engineering. This chapter describes the teaching activities, primary research projects and scientific contributions of the author to demonstrate his competence in research-guided teaching.

3.1. Teaching activities

Beginning in 2007, the author broadened his knowledge of automotive engineering as an assistant lecturer at the Institute of Automotive Engineering. The following lectures were prepared and conducted in cooperation with other lecturers:

- **Practical Course on Matlab/Simulink** (MATLAB Tutorium Fahrzeugdynamik, LV 331.100[1])
  This tutorial is a computer-based practical course designed to teach the specific skills needed for modelling and simulation in vehicle dynamics using the software package Matlab/Simulink®. It teaches basic knowledge, starting with the handling of variables, arrays and matrices, and continuing with the building of plots and programming of Matlab scripts and functions. Computational issues related to numerics are discussed briefly, and the basics of numerical time integration are explained. The course is supplemented with basics in graphical programming using the Simulink toolbox. The students are prepared for the follow-up course described below. The author contributes eight lecture units related to the basics of Matlab.

- **Modelling and Simulation in Vehicle Dynamics** (Modellbildung und Simulation in der Fahrzeugdynamik, LV 331.094)
  This lecture is a combined theoretical and practical course. It starts with a theoretical overview of appropriate methods. The practical part deals with the application of tyre models and uses test cases to show how to parametrise them using the model TMeasy, [HRW07]. Additional parts of the practical lecture deal with the preparation of longitudinal, lateral and vertical vehicle dynamics models. The author contributes eight lecture units dealing with the introduction to tyre models and their parametrisation.

[1] The LV number mentioned in brackets refer to the Graz University of Technology curriculum and can be accessed by [Gra10].

- **Project Supervision**
  Together with other lecturers, the author supervises the following project works for students:
  - Design project in automotive engineering (Projekt konstruktiv Fahrzeugtechnik, LV 331.007)
  - Work project in automotive engineering (Projekt Fahrzeugtechnik, LV 331.010)
  - Bachelor project in automotive engineering for Mechanical Engineering (Bachelor-Projekt MB, LV 331.011)
  - Bachelor project in automotive engineering for Mechanical Engineering and Business Economics (Bachelor-Projekt WIMB, LV 331.012)

  These projects cover topics related to automotive engineering, such as vehicle functions, production- and cost-compatible layout, design and documentation of selected automotive vehicle parts, components or assemblies using CAx-methods.

- **Introduction to Automotive Engineering** (LV 331.088)
  This basic lecture is a compulsory lecture for students of the master course *Production, Science and Management* at Graz University of Technology. It is held in English in cooperation with the Institute of Vehicle Safety (headed by Prof. H. Steffan) and teaches the fundamentals of modern automotive engineering. The topics covered are: mobility and automotive history; current developments in passenger and freight traffic, globalisation, environmental and climatical aspects; development processes and basics in CAx; basic knowledge in vehicle physics, with a focus on longitudinal, lateral and vehicle dynamics; main vehicle components (functions, requirements and applications); introduction to vehicle safety: active, passive and integrated vehicle safety. The author contributes 16 lecture units dealing with all topics mentioned, except vehicle safety.

- **Automotive Engineering and Vehicle Safety** (Fahrzeugtechnik und -sicherheit, LV 331.070)
  This content of this lecture, which is held in German, is comparable to the lecture explained above. However, it is geared towards bachelor students in *Mechanical Engineering* and *Mechanical Engineering and Business Economics*. The purpose of the lecture is to provide knowledge about automotive engineering to students who do not intend to specialise in that topic in their master study. Again, the author contributes 16 lecture units.

- **Integrated Vehicle Safety** (Integrierte Fahrzeugsicherheit, LV 333.052)
  This lecture is an elective course for students of *Mechanical Engineering* and *Mechanical Engineering and Business Economics* and is conducted in cooperation with the Vehicle Safety Institute. It supplements the lectures of the Institute of Automotive Engineering and the Vehicle Safety Institute which focus on primary (active) safety and secondary (passive) safety, respectively. The topics covered are the function, benefit, requirements and evaluation of integrated traffic systems

(e.g. vehicle dynamics control); Advanced Driver Assistance Systems (ADAS); Vehicle-to-Infrastructure (V2I) and Vehicle-to-Vehicle communication (V2V); and adaptive restraint systems. Aspects of X-by-Wire technologies and Environment Recognising Systems (ERS) are discussed. The lecture is rounded out with a practical demonstration of a laser- scanner ERS and an excursion to the "Highway[3]" V2I test track of MAGNA STEYR Fahrzeugtechnik at an highway near Graz. The author contributes 16 lecture units.

- **CAx in Automotive and Engine Technology** (LV 313.066)
  This lecture is held in English in cooperation with the Institute for Internal Combustion Engines and Thermodynamics (headed by Prof. H. Eichlseder) and the Institute of Automotive Engineering. It provides knowledge in modern CAx methods for automotive engineering and combustion engine development. The author contributes 2 lecture units dealing with the basics of CAx methods in primary and secondary vehicle safety.

3.2. Research projects

As an university assistant, the author was involved in several research projects related to basic research, and as a former employee of the automotive industry, he also participated in research projects related to automotive applications. These research activities contributed to improved simulation methods as well as product innovations. The following section describes four selected projects to which the author made significant contributions and which are further explained in the main portion of the following habilitation thesis.

- **EC R&TD project Whiplash I - Reduction of neck injuries and their societal costs in rear-end collisions** (1996-2000)
  The project, carried out by a consortium of nine European partners from science and industry, was one of the first attempts to investigate whiplash injuries and evaluation methods on a broad basis. It included research in biomechanics to derive performance corridors[2] for anthropometric test devices (ATD's) based on a literature review and on experiments with volunteers and cadavers. Prototypes for ATD's were developed and investigated for safety assessments in rear-end collisions. Modelling and simulation methods for design optimisation were developed, as well as test methods and performance criteria for assessing the safety potential of seat and head restraint systems. Some of the foundations of the current state-of-the-art evaluation methods (e.g. a generic sled test) were laid in this project.

- **EC R&TD project Improvement of Rollover Safety for Passenger Vehicles** (2002-2005)[3]

[2]Performance corridors are requirements for injury responses of ATD's exposured at standardised calibration tests.
[3]Available at http://www.vsi.tugraz.at/rollover/, accessed on 8 January 2010

The objective of this project was to develop a holistic view of the *rollover* accident load case. Fifteen scientific and industrial partners cooperated on this project, ranging from system suppliers to OEMs. It included an analysis of rollover accidents in Europe and the USA on an in-depth, statistical level of investigation. Typical rollover scenarios for passenger vehicles in Europe were derived. Vehicle and occupant performance in these typical scenarios was investigated in simulation and experiment. The cause of injury was analysed, and efficient rollover test methods in simulation and experiment with corresponding performance criteria were investigated. Emphasis was put on the integrity of the vehicle structure and the proper performance of the restraint system.

- **Austrian Kplus research project A1-c5 Sliding Collision** (2002-2006)
  For years, frontal accidents with narrow lateral offset were a topic of research, but this has never led to requirements for occupant protetion. Funded by the Austrian Kplus research program, this project, carried out by the Virtual Vehicle Research and Test Center and MAGNA STEYR Fahrzeugtechnik, focused on protective systems for this load case. Basic research was carried out to define a relevant evaluation method of the system based on in-depth accident analysis. The performance of an appropriate protection system was demonstrated in simulation and experiment.

- **Austrian Kplus research project A1-c3 Out-of-Position** (2002-2006)
  US legislation forced vehicle manufactures to introduce advanced airbags which are less aggressive to occupants not located in the design position for airbags (e.g. children, adolescents and small females). The development of low-risk airbag deployment was initially based on experiments, which was time-consuming and expensive. This project focused on simulation methods on the full-vehicle level, which included improvements in the simulation of the airbag (folding of airbag fabric and gas dynamics during deployment), the occupant (airbag loading of head, neck and thorax) and the airbag cover (opening of tear seams).

3.3. Scientific contributions

The author's scientific contributions to automotive engineering have been documented in several publications, including a book section, peer-reviewed journal articles, reviewed and not re-viewed conference papers and posters, research reports, patents and technical presentations. The author has been asked to review articles in *Automobiltechnische Zeitschrift (ATZ)* and various conferences (*icrash 2006, EUROSIM 2007, IMETI-International Multi-Conference on Engineering and Technological Innovation 2009 and 2010*). In 2007, he chaired the special session “Increased Predictability of Crash Models” during the *6th EUROSIM Conference on Modelling and Simulation.* In 2008 he co-chaired and co-organised the first *Grazer Symposium Virtuelles Fahrzeug*[4], which deals

[4]Available at http://www.gsvf.at, accessed on 9 February 2010

with research on CAx methods in automotive engineering. In 2010, he was heading the scientific session (peer-reviewed contributions) for this annual conference. In 2007, he coordinated the technical description of the *Virtual Vehicle Research and Test Center*'s successful application for an Austrian K2 competence centre. In 2008, he was appointed the scientific head of the centre's working area *D (Mechanics)*. His duties there include the strategic positioning of the area, assisting in project proposals, reviewing the publications and project reports of about 50 employees in the area, and taking part in project controlling. He is also a member of the research centre's program committee, which evaluates the scientific relevance of project proposals.

The contributions in the following habilitation thesis include:

- Investigations of narrow offset frontal crashes and protection systems
- Simulation methods for low-risk deployment of advanced airbags
- Methods for side impact simulation and testing
- Basic research in the biomechanics of whiplash injury
- Investigation of whiplash protection systems and their automotive application
- Basic research passenger car rollover
- Assessment of the benefit of vehicle safety systems
- Methods for the control of adaptive restraint systems

4. Outline of the habilitation thesis

The following habilitation thesis consists of four main chapters and summarises the auhor's contributions to primary, secondary and integrated safety.

Chapter 5 introduces to the topic by describing the important aspects of vehicle safety. It starts with the societal consequences of traffic accidents and a definition of the three elements involved in traffic: driver, vehicle and environment. The terms primary (active), secondary (passive) and integrated safety are defined, and a brief overview of accidentology is provided. The introductory chapter ends with a classification scheme for traffic accident given in [Kra08].

Chapter 6 uses the aforementioned accident classification into *kind of impacts* (i.e. frontal collisions, side impact, rear impact and rollover) in order to summarise the author's contributions to secondary vehicle safety. A broad spectrum is mentioned, which ranges from accidentology, biomechanics and the development of safety systems to virtual and experimental development methods for vehicle safety.

Chapter 7 deals with the broad topic of traffic safety systems[1]. Although a detailed description of these systems (which is available in the relevant literature) has been omitted, different classification schemes for traffic safety systems are mentioned. The next section discusses the potential benefits of traffic safety systems. After an introduction to evaluation methods, results from an extensive investigation, the "Retrospective Case Study of Graz University of Technology (RCS-TUG)", are presented[2] and compared with results from the relevant literature. The RCS-TUG study relies on the simulation of vehicle dynamics in the pre-collision phase and is therefore considered the primary safety contribution of the author.

Chapter 8 describes the step of the author towards the integration of vehicle safety. This chapter describes an approach for controlling an assumed adaptive restraint system with respect to the accident situation and size and the position of the occupant. The approach is based on the prediction of the anticipated deceleration of the passenger compartment in a straight, full-frontal collision. The prediction uses data from environment recognising systems (ERS) and integrates functionalities of ADAS (e.g. ABS and Predictive Brake Assist) with OCS (Occupant Sensing Systems) and an adaptive restraint system.

[1]This chapter is not restricted to vehicle-based safety systems.
[2]Some of the results are published for the first time.

Part II.

Habilitation thesis

5. Introduction

5.1. Societal consequences of traffic accidents

With increased traffic, the risk of traffic accidents and related consequences has been increasing. Traffic accidents and their consequences have become an important factor for society. Worldwide, in the year 2004 traffic accidents reached number 9 on the list of the 136 main categories of disease and injury [MBF08]. The 1.3 million fatalities accounted for 2.2% of all deaths. The study projected an increase in global deaths due to accidents of 28% by 2030, with the major contributing factor being an increase in fatal road accidents from 1.3 million to 2.4 million. The main reported reason is increasing motorisation in emerging countries. In 2030, road accidents will be the number five reason for mortality. The same study reported 24.3 million severe injuries[1] resulting from traffic accidents in 2004, which would place traffic accidents at number 9 worldwide and number 6 in Europe in the leading causes of disease. However, by 2030 injuries are predicted to reach number 3 on this list and account for 4.9% of all injuries. In Europe, the costs to society resulting from traffic accidents are reported to be about 160 billion €, which represents 2% of the Gross National Product (GNP) [Off03][Pet00]. For this reason, the European Commission has started an initiative to halve the number of fatalities by 2010 and is contributing to this effort through the harmonisation of penalties for traffic violations and the promotion of new technologies to improve road safety [Off01].

5.2. Driver, vehicle and environment

A comprehensive discussion of improvements in road safety requires an understanding of the complex interaction between the three elements in road traffic: the human being, the vehicle and the environment, see Fig. 5.1. Drivers guide their vehicle using the vehicle control inputs. They are supported in this task by the Human-Machine Interfaces (HMIs). Depending on the driving conditions, they also receive feedback from the vehicle by forces and motions transferred by the vehicle interior (e.g. seat), vehicle controls and the restraint system. The vehicle reacts to the driver input, but also to outer forces from the environment, such as the road, wind or obstacles. Drivers take in information about the environment via their sense organs, mainly from optical (approximately 80 to 90%, [AB09]) and acoustic information. As part of the traffic situation, the environment influences both the vehicle and the driver. In literature (e.g. [Don82, Ber70, Mic85]), the driving task is often classified in three different levels:

[1] Medical treatment required

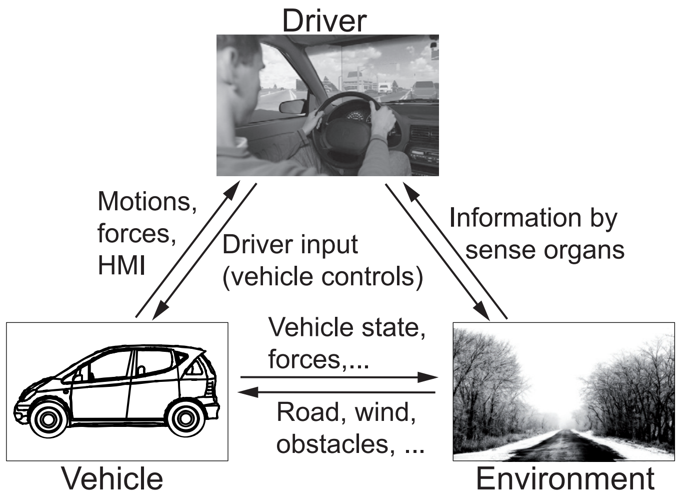


Figure 5.1.: Interaction between driver, vehicle and environment

- Navigation
- Course planning in longitudinal and lateral direction
- Stabilisation

This classification is based on a publication by [Ras83] which postulates a general psychological model for target-oriented human tasks in a working process. It differentiates between

- Knowledge-based behaviour
- Rule-based behaviour
- Skill-based behavior

Knowledge-based behaviour occurs when human beings face new and complex situations for which they have not been previously trained. Based on their prior knowledge, humans imagine different alternatives that may be suitable for the desired aim before selecting the most appropriate and reacting.

Rule-based behaviour takes place in situations which have already been experienced several times before. The most effective alternative is chosen based on a set of rules the human has obtained in previous training.

The third level is *skill-based behaviour.* Here, the reactions are characterised as reflexes which are initiated without awareness and require a previous, long-term learning process.

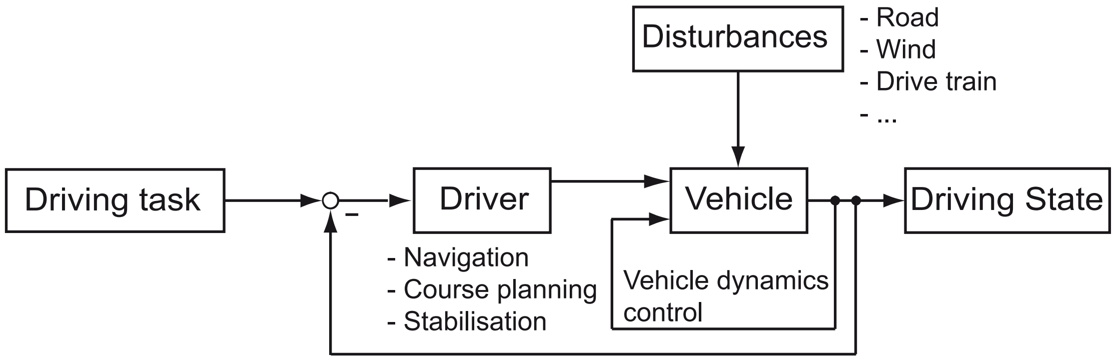


Figure 5.2.: Control block diagram of the driving task, adapted from [HW09]

In the adaption of this model by [Don82], the human being decides upon a route of travel through the *navigation* task. He/she has to take into account several aspects, including the duration, intention and interim stops of his travel, the safety of the travelled road and other factors, [Grü05]. When drivers navigate in an unknown region, their mind act on the knowledge-based behaviour level and plans the route. In other situations, the navigation task is more or less already fulfilled, and only monitors the route in time discrete steps. On the *course planning* level, drivers plan the course for their travel and guide the vehicle to follow it. They have to take into account the course of the road, its present condition and the traffic situation. This includes the longitudinal (reference velocity) and lateral guidance (reference path or trajectory) of the vehicle. Vehicle course planning can be defined as an open-loop control [Don09]. Finally, on the *stabilisation* level the driver reacts to environmental disturbances that have led to deviances from the planned vehicle course. In control theory, this can be considered as a closed-loop control.

The driving task described above is also often illustrated as a control block scheme, as illustrated in Fig. 5.2, [HW09]. The intended driving task is controlled by the driver on the three levels discussed above. The vehicle reacts to the driver inputs with a dynamic state that is influenced by disturbances from the environment. This leads to the actual driven course, which is controlled by the driver (outer closed loop) and by autonomous systems of the vehicle (inner closed loop).

Compared to the model of [Ras83], course planning and stabilisation tasks occur on the rule-based and skill-based behaviour levels, depending on the driver's acquired competencies. A novice driver will face rule-based behaviour more often than an expert driver, see Fig. 5.3. According to [Don09, WHW09], the driver is forced into knowledge-based behaviour in critical driving situations. If there is not enough time to decide on the best behavioural alternative (high velocity, small distance to obstacle), the risk for accident is imminent. In Fig. 5.3 the time for decision is called Time-to-Collision (TTC) to obstacles. TTC is considered decisive for the driver action. This categorisation is especially useful when categorising Driver Assistance Systems (DAS).

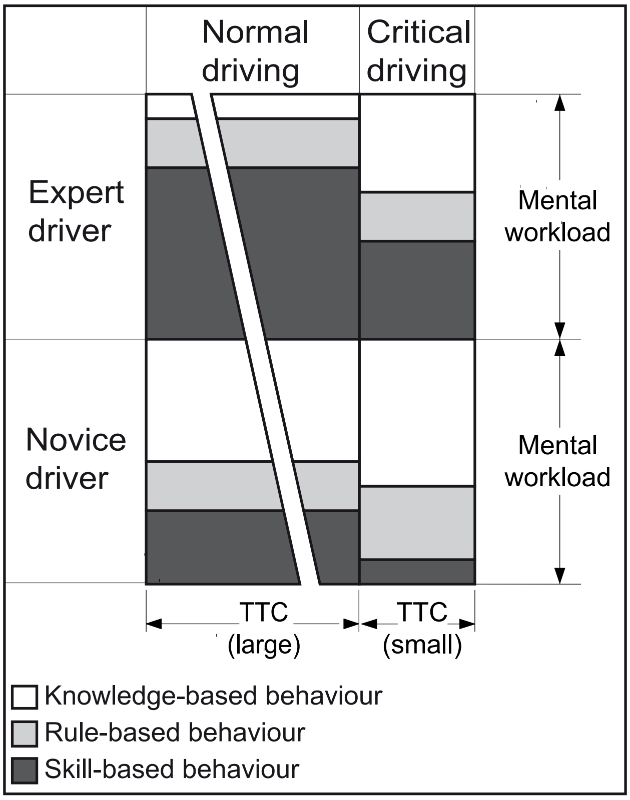


Figure 5.3.: Qualitative description of the fraction in mental workload according to the model of Rasmussen, [Ras83]

5.3. Primary, secondary, tertiary and integrated traffic safety

Traffic safety is frequently classified into active and passive safety [Kra08]. The recent trend is to replace these terms with the more precise terms primary and secondary safety, which will be used throughout this thesis. Some publications (e.g. [LBH+08]) even define tertiary safety for post-crash treatment, see Fig. 5.4. Primary safety is related to measures used to avoid accidents or at least to decrease collision severity. Secondary safety is related to measures used to mitigate the consequences to the human in an unavoidable accident. Finally tertiary safety refers to the rescue system and immediate injury treatment. Fig. 5.4 depicts some examples of these measures. For decades, these measures have been developed separately, thereby improving the overall traffic safety. In order to counter the increase in traffic accidents predicted by the WHO, the integration of primary, secondary and tertiary measures, in context of this thesis defined as “Integrated Traffic Safety”, is believed to hold greater potential for improvement. In literature, Integrated Safety is also sometimes defined as interaction of vehicle based safety systems in the immediate pre-collision phase.

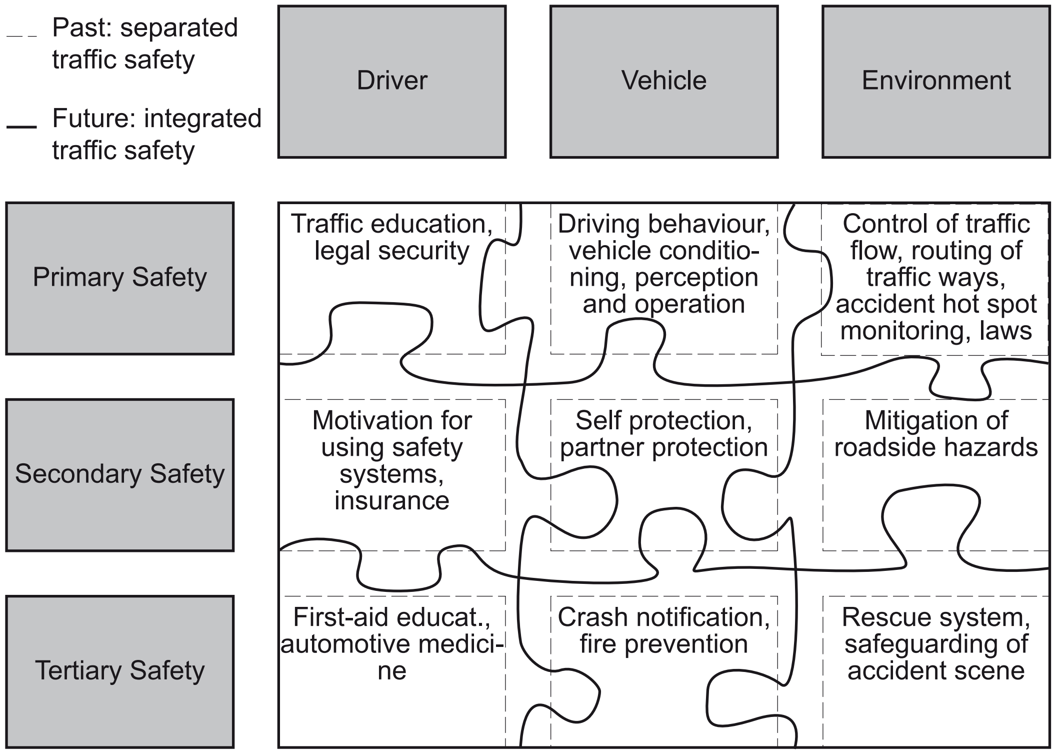


Figure 5.4.: Aspects of traffic safety and examples for safety measures, adapted from [Kra08]

5.4. Accidentology

Accident research is the basis on which countermeasures to traffic accidents are developed [LBH+08]. Fig. 5.5 depicts the history of accidents, injured persons and fatalities on European roads from 1997 to 2006. Using a linear trend, the European goal of halving the number of traffic fatalities to 25,000 will not be accomplished until 2015 (R^2=0.98). Fig. 5.6 depicts the difference between injured persons per accident and fatalities per accident. This figure shows that the number of injured persons per accident (approximately 1.3) is quite constant over time[2]. On the other hand, the number of fatalities per accident is consistently decreasing[3], indicating improved occupant protection at high collision severity and/or decreased collision severity.

The following sections provide a brief overview of accident research.

[2]Mean value $\bar{x}$=1.34, standard deviation σ^2=0.01
[3]Coefficient of determination R^2=0.98

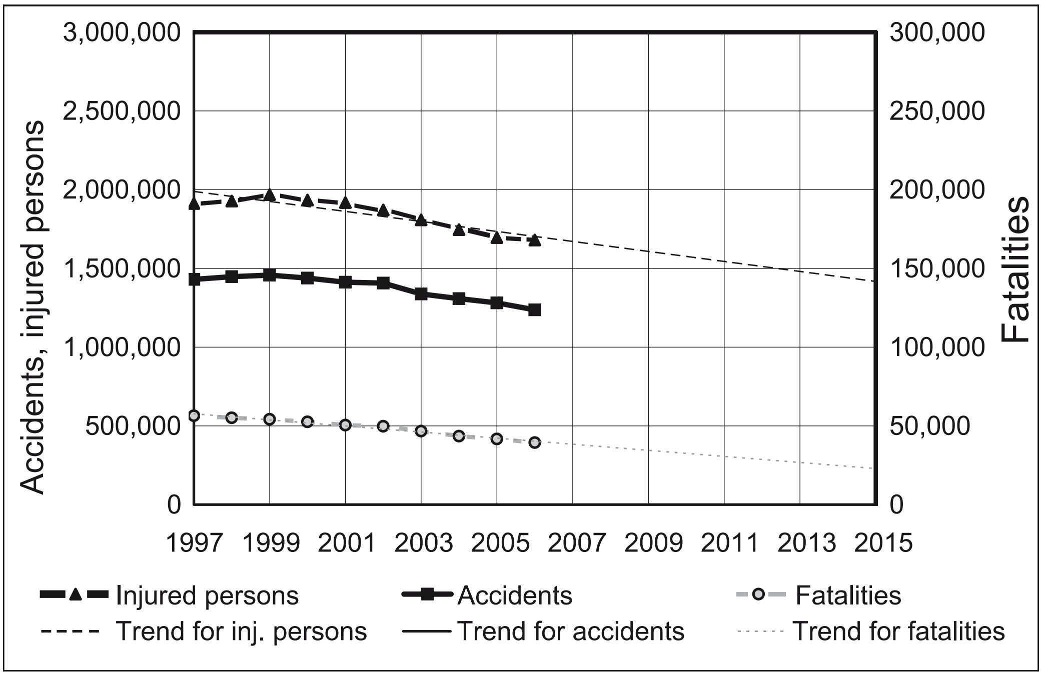


Figure 5.5.: Accident, fatalities and injured Persons in EU 25, [LBH+08]
The lag of time between analysed data (2006) and 2010 is explained by unavoidable delays in accidentology caused by data collection and analysis. For reasons of comparison, the territory of EU 25 was used throughout the analysis.

5.4.1. Accident phases

Traffic accidents can be divided into different phases. Fig. 5.7 provides a qualitative description. The lower part of the figure shows which vehicle safety system apply to which phases.

- Normal driving situation
  The driving situation is non-critical with respect to the dynamic state of the vehicle and approaching obstacles. In this phase, ADAS observe and act in a “comfort mode”. A typical example is Automatic Cruise Control (ACC).

- Critical driving situation
  The transition to critical driving situation requires that ADAS switch from comfort to safety functionalities. The target is to avoid accidents or minimise collision severity either by supporting the driver or by stabilising the vehicle. Intervening and assisting ADAS for longitudinal and lateral vehicle dynamics act in this phase, thereby avoiding oncoming collisions. Typical examples are Collision Warning Systems (CWS) or Electronic Stability Control (ESC).

- Pre-collision
  As the “point of no return” is reached, safety systems are activated to reduce the

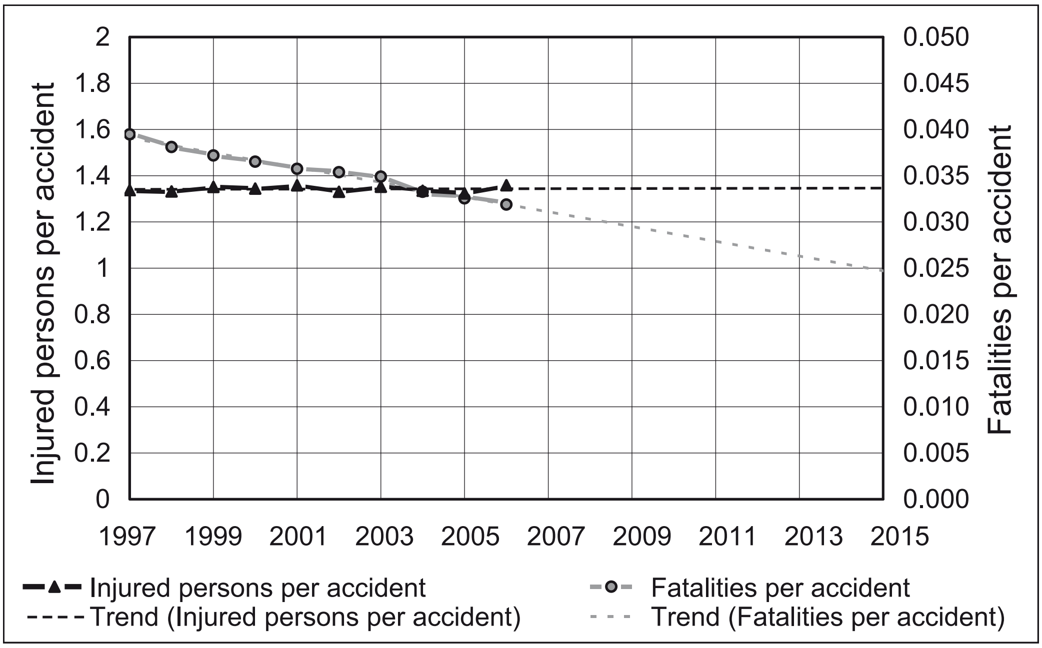


Figure 5.6.: Fatalities and injured Persons per accident in EU 25

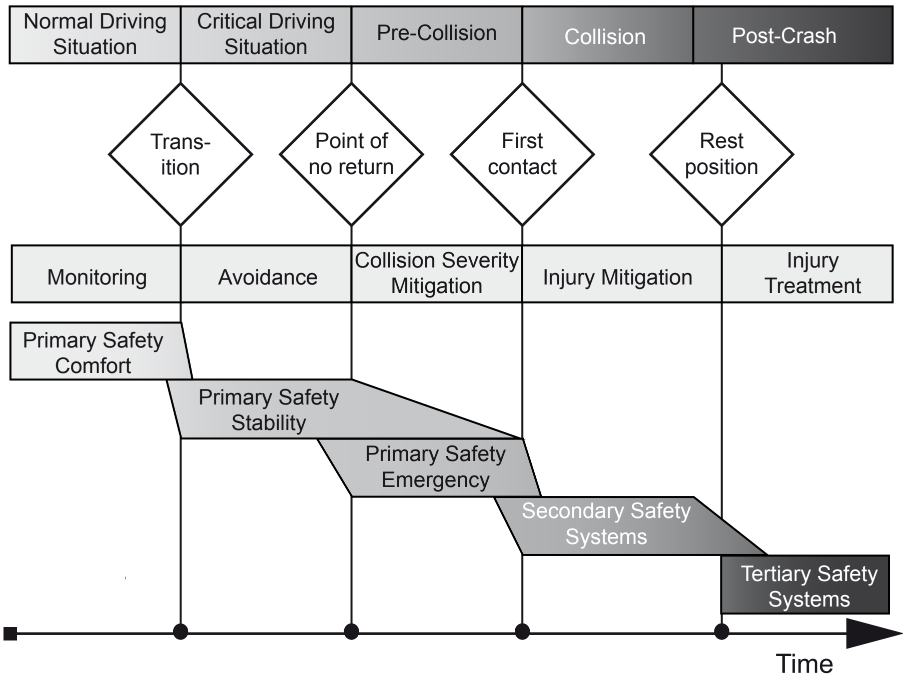


Figure 5.7.: Phases in traffic accidents

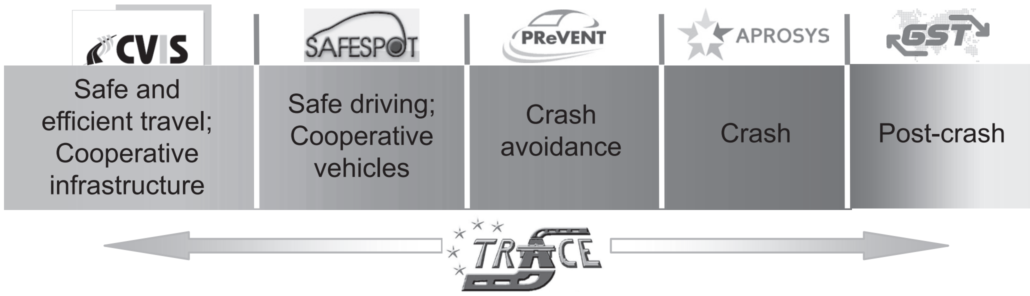


Figure 5.8.: Overview of EC projects related to Integrated Safety, [Eur09e]

collision severity until the collision partners come into contact (time “zero”). Adaptive and pre-fired restraint systems are typical examples. The necessary interaction of different systems for such a restraint system will be described in detail in chapter 8 of the thesis.

- Collision
  As time zero is passed, secondary safety systems minimise the consequences for occupants and other vulnerable road users. Examples are seat-belt systems or pop-up hoods. After the collision partners part, they move into their rest positions.
- Post-crash
  Post-crash systems address the treatment of injuries by initiating the rescue chain. Ecall and Automatic Crash Notification are typical examples.

In support to its white paper policy, the European Commission has initiated several supporting projects. Fig. 5.8 shows a classification of important projects related to Integrated Traffic Safety. The illustration is related to the accident phases: CVIS [Eur09b] and Safespot [VDV07] focus on cooperative infrastructure and vehicles; PReVENT [SMI+08] deals with accident avoidance; APROSYS [Eur09a] deals with secondary safety; and GST [Eur09c] handles post-crash treatment. These projects are based on the TRACE project [Eur09e], which aimed to identify and assess traffic safety technologies, as well as to develop an etiology[4] of road accidents.

5.4.2. Accident causation

An understanding of traffic accidents and mechanisms operating during the collision is the key to improving secondary safety, while an understanding of accident causation is the key to collision avoidance and collision severity mitigation. It forms the basis of ADAS development. Several investigations have analysed the causation of traffic accidents. In principle, the research can be divided into two areas. On the one hand, in-depth databases, which feature detailed accident descriptions, usually limited case numbers, and often restrictions to localised areas, lend themselves to detailed analysis.

[4]Study of causation or origination

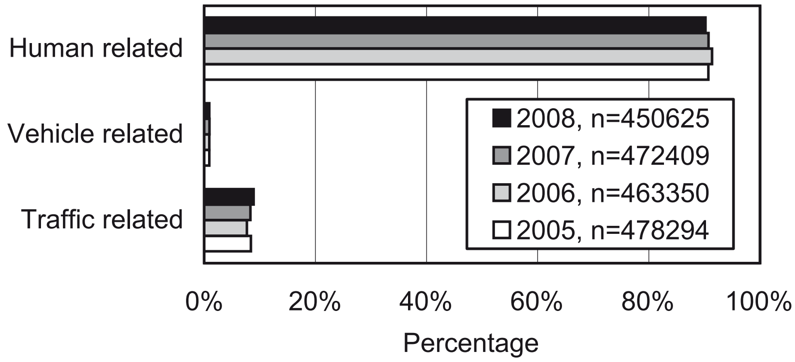


Figure 5.9.: Main categories of driver error, adapted from [Sta09]

On the other hand, larger databases, which are often based on police reports, allow for the investigation of high case numbers, but with a limited amount of detail.

In Germany in 2007 [Sta08], the police recorded 2,335,005 traffic accidents. In 409,529 of these cases, the accident causation was classified when the record was made. Of these 91% were caused by factors related to the human being, and 87% to driver errors. This distribution of causations remains more or less constant over time, see Fig. 5.9. Fig. 5.10 shows the main categories of driver error. Unlike previous years, in which excessive speed was the dominant cause of accidents, in 2007 the main causes of accidents were incorrect behaviour during turns, turn-overs, reversing, starting and merging into traffic (16%). Speeding and right-of-way violations were the next most common causation factors, with 15% for each category. The results of this study are limited by the accuracy of the police report and its level of expertise. Nevertheless, the statistics suggest that the focus should be on driver support in critical driving situations.

A risk factor investigation was done with an in-depth accident database based on fatal traffic accidents in Austria in 2003, [Tom07]. Once again, human factors dominate (46.6%). However, in contrast to the aforementioned German studies, which covered all accidents, infrastructure and vehicle-related factors were more important in the Austrian study, which was limited to fatal accidents, see Fig. 5.11.

A more detailed analysis of accident causation can be found in [MBMvdH08]. A research team acting at the site of the accidents developed an in-depth accident database. Approximately 3,000 cases were investigated between 2000 and 2005. Since a combination of factors often contribute to an accident, the 54 contributing factors indexed in the database were grouped into 10 main classes, see Fig. 5.12. The principal causation factors were driver inattention (54.4%) and aggressive driving (27.6%).

A comprehensive study of accident causation was done within the EC project TRACE, [Eur09e]. In the TRACE report, [EF07] accidents are seen as malfunctions of the system

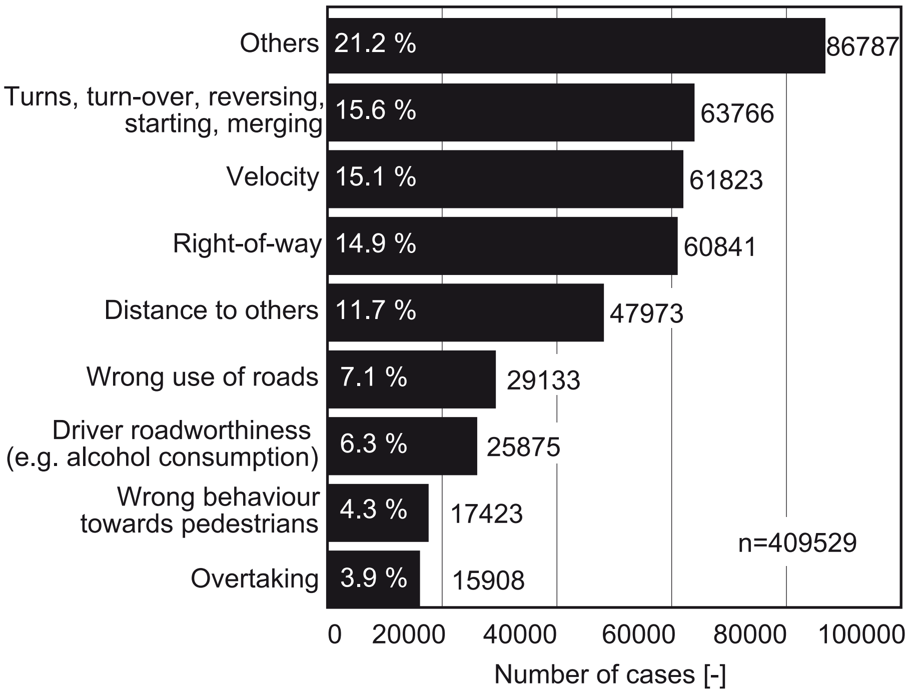


Figure 5.10.: Main categories of driver error, adapted from: [Sta08]

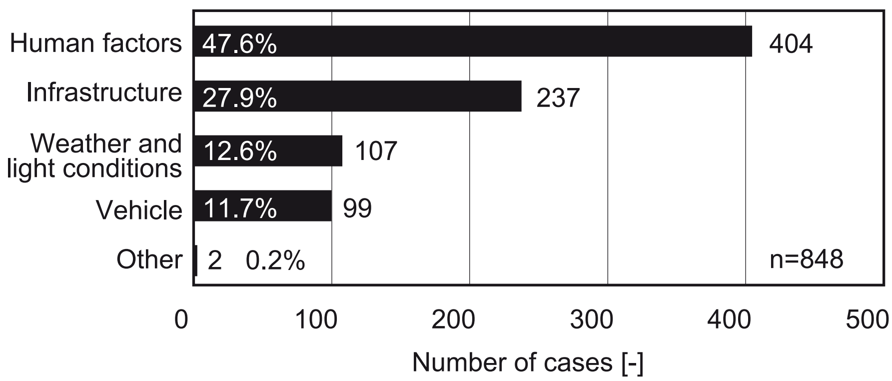


Figure 5.11.: Main categories of risk factors, adapted from: [Tom07]

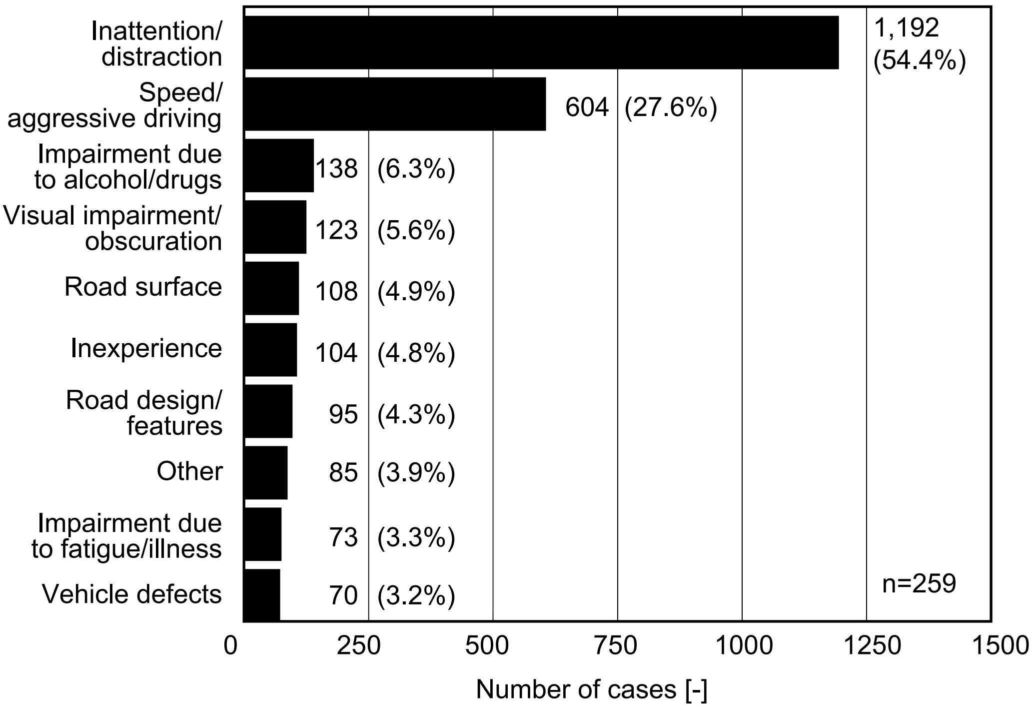


Figure 5.12.: Main categories of driver error, adapted from: [MBMvdH08]

human-vehicle-environment. Therefore, the study developed better methods for understanding the process that leads to accidents. However, at the time of writing this thesis, detailed results have not been released.

To summarise, it is difficult to compare research studies dealing with causation factors of traffic accidents, since the studies use different databases, different methods and different definitions of risk factors. However, across the research, the dominant factor in traffic accidents is the human being, who, in certain conditions, will benefit from systems that support him/her in the driving task.

5.4.3. Classification of road accidents

In order to study accidents in detail, a classification is necessary to compare the incidence of certain aspects. For example, the incidence of head injuries in fully overlapped car to car collisions could be of interest, in order to decide if this is a relevant aspect to develop protective measures. Within the literature, there have been different attempts to classify accidents. A consistent categorisation was described in [Kra08], which will be used throughout this thesis, see Fig. 5.13. Road accidents are primarily differentiated by *kind* and *type*.

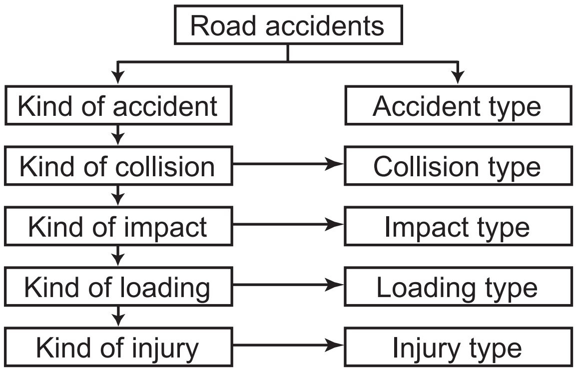


Figure 5.13.: Classification of road accidents, adopted from: [Kra08]

The *kind of accident* is based on the road users involved (commercial vehicles, passenger vehicles, motorised two-wheelers, bicycles, pedestrians and single vehicles), while the *accident type* is related to the cause of the accident. In [Sta08], the following accident types are defined: driving accident, accidents caused by the turning of road users, intersection accidents, accidents caused by crossing pedestrians, accidents involving stationary vehicles and accidents between road users moving along the same road.

The *kind of collision* classifies collisions with respect to the road users involved, if only two objects (including obstacles) are included. Multiple collisions, which account for about 10% of all accidents ([Sta08]), are excluded. One example of *kind of collision* is commercial-to-passenger-vehicle collision. The classification of collisions uses the same categories of road users as those used in the "kind of accidents" classification. In contrast to kind of accidents, double counts are avoided. As an example, a single "car-to-car" kind of collision would count as two incidents of a "car" kind of accident. The *collision type* classifies collisions according to the impact location, with separate classifications for commercial-to-passenger-vehicle, passenger-to-passenger-vehicle, and passenger-vehicle-to-obstacle collisions.

Kind of impact describes the accident situation for each object. It includes front, side, rear and rollover kind of impact. *Impact type* is a further classification which describes the exact location and severity of the deformation.

Kind of loading describes the loaded body regions (head, neck, thorax, upper extremities, abdomen, pelvis and lower extremities). The *loading type* is based on both the seating position (driver, passenger, rear passenger left/middle/right) and whether or not the seatbelt is fastened.

The last level is the injury level. The *kind of injury* categorises injuries to soft tissues, organs, ligaments/tendons and vascular system. Finally, the *injury type* classifies direct,

indirect loading and inertial loading, as well as hyperextensions/flexions.
A detailed description of the classification scheme and accompanying accident statistics can be found in [Sta08]. The next chapter summarises the author's contribution to secondary vehicle safety with respect to the *kind of the impact*, namely

- frontal impact,
- side impact,
- rear impact,
- rollover.

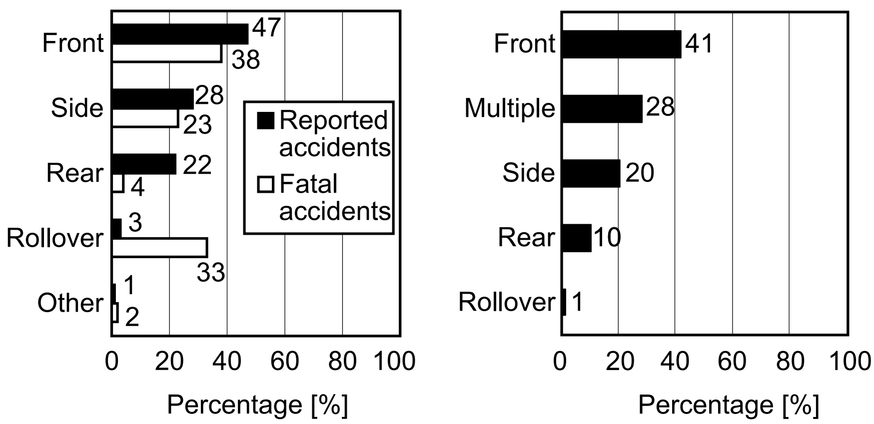
6. Contributions to secondary vehicle safety

Fig. 6.1 depicts the results from accident research into *kind of impact*. It should be noted that these numbers differ due to differences in the considered country, the methodology used to assess and classify the data, the year of the accident sample, the size and representativeness of the sample and other factors.

6.1. Frontal impact

Frontal impact is the most frequent kind of impact, see Fig. 6.1. In [BH04] primary damage to the front of the vehicle accounts for about two thirds of all accidents in the sample. A frontal impact is characterised by the following parameters, Fig. 6.2:

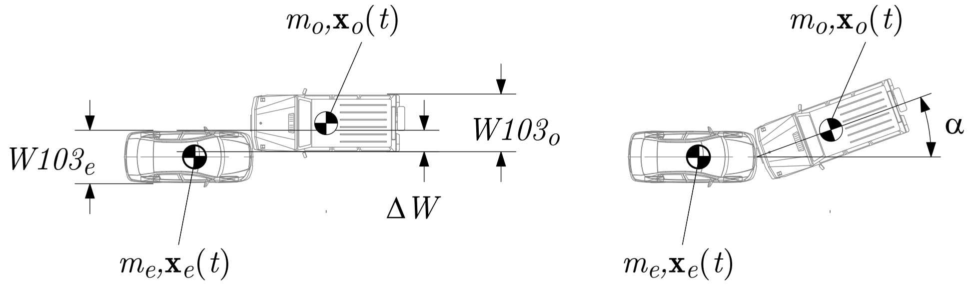
- **Ego-vehicle**
  Mass m_e and dynamic $n \times 1$ state vector $\mathbf{x}_e(t)$
- **Obstacle**
  Mass m_o and dynamic $n \times 1$ state vector $\mathbf{x}_o(t)$ for an obstacle vehicle
- **Offset**
  The percentage of the overlapping vehicle width O_e,



(a) FARS 2003 and NASS GES 2003 data, [Gug06]

(b) GIDAS data, [BH04]

Figure 6.1.: Findings in distribution of kind of impact



(a) Definition of offset ΔW (b) Definition of impact angle α

Figure 6.2.: Definition of offset and angle in a frontal impact

$$O_e = \frac{\Delta W}{W103_e} \cdot 100[\%] \ , \tag{6.1}$$

of the ego-vehicle with the collision opponent influences the deformation characteristics and direction of the collision impulse. $W103$ refers to the vehicle width defined by standard DIN 70020-1 [Nor93].

- **Angle of impact**
  The angle of impact α also influences the results with respect to deformation characteristics and direction of the collision impulse.

Frontal impacts have been extensively investigated in the field of secondary vehicle safety. In the USA, current legislation mandates manufacturer testing of straight and oblique frontal impact against a rigid barrier (FMVSS 208 and US-NCAP). The tests are carried out with full overlap using belted and unbelted crash test dummies of different body size. In Europe, ECE R94 [Uni07] and the consumer test EURO-NCAP [Eur09f] require a 40% overlapped straight frontal impact. Recent ratings in US-NCAP and EURO-NCAP show that an increased occupant protection in frontal impact has been achieved in these load cases. Nevertheless, frontal impacts with narrow offset are not covered by the aforementioned crash requirements. A review of accident investigations [ESWS07, SNZ09] revealed that frontal collisions with narrow lateral offset show significant incidence and risk for injuries, Table 6.1 shows the findings. In [SNZ09] it was summarised that the findings in the literature suggest that about 25% of serious frontal crashes include offset lower than 40%. A possible explanation for this could be the so-called “hang-on collision”, which is discussed in the next paragraph.

6.1.1. Frontal collision with narrow lateral offset

In a frontal collision with narrow lateral offset, the wheels on the impacted side can get caught, which prevents the vehicles from sliding off each other (a “hang-on collision”). A significant portion of the vehicle’s kinetic energy loads the foot-well area via the load

Table 6.1.: Reported significance of narrow lateral offset frontal accidents

Ref.	**Method**	**Result**	**Accident sample**	**Database**
[Hob91]	Accident analysis	In 27% no involvement of longitudinal beam	Fatal frontal accidents	UK
[OLZP94]	Accident analysis	In 22% offset lower than 33%	Fatal frontal accidents	USA
[SJH+94]	Accident analysis	30% equivalent to less than 30% offset	Frontal accidents	Germany
[LHB04]	Accident analysis	34% no deformation of non-impacted longitudinal beam	Fatal frontal accidents	Sweden

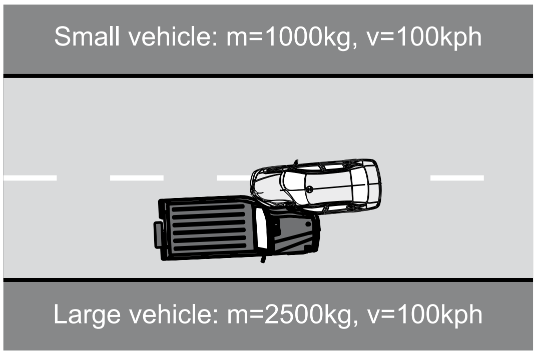
path at the wheels. Vehicle body structures such as the longitudinal beams are not involved in the absorption of collision energy since they are located outside the main load path. The occupant compartment collapses, causing severe injuries by intrusion of the passenger cell and high accelerations, especially to the driver.

Fig. 6.3 qualitatively depicts the outcome of a frontal collision with 25% offset and two vehicles with different mass classes [WFBS01]. Note the large difference in velocity change Δv and Energy Equivalent Speed EES[1] for the large and the small vehicle. A high risk of fatal injury for occupants of the small car exists. Previous research has shown that this type of frontal collision can be prevented by transforming the hang-on collision into a "sliding collision", [Win01, WFBS01, Sch00], see Fig. 6.3 (c,d). A large reduction of Δv and EES for occupants of the small car (i.e. 74% and 39% respectively) is calculated in the example.

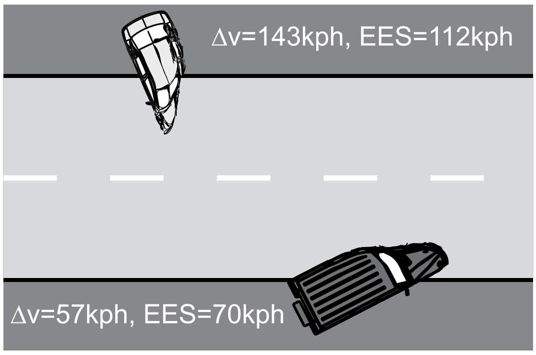
In [Win01, SE05] protective systems that are able to transform hang-on into sliding collisions were investigated. Since no established laboratory tests were available, a proposal for an appropriate configuration for narrow offset frontal impacts was developed [ESWS07]. The research was based on accident investigations with different accident databases. Methods ranging from statistical evaluation to in-depth accident analysis and reconstruction were used to define a straight frontal crash configuration that features vehicles with an impact speed of 56 kph each and a lateral offset O of 17%. In [ESF08] this work was continued and the complex car-to-car test was transformed into a collision against a rigid barrier. Fig. 6.4 shows the bottom view of a Finite Element Method (FEM) simulation using two models of a Ford Taurus. Both deformation behaviour and vehicle measurements were in acceptable agreement.

Based on the previous work of [Win01], several concepts to prevent the hang-on collision were developed [EHP+05, SE05, SBS+06, SSE05]. The most promising concept, the so-called "Flexible Collision Deflector (FCD)", was developed by aid of FEM crash simulation. It was built and successfully tested on a full-vehicle system level in a simpli-

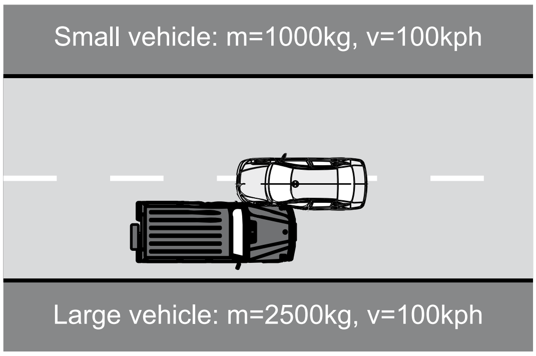
[1]Deformation energy W_{def} expressed as velocity EES using the equation $W_{def} = m \cdot EES^2/2$



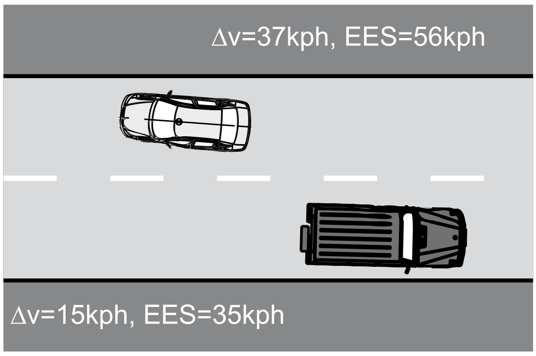
(a) Hang-on collision, impact phase



(b) Hang-on collision, rest position



(c) Sliding collision, impact phase

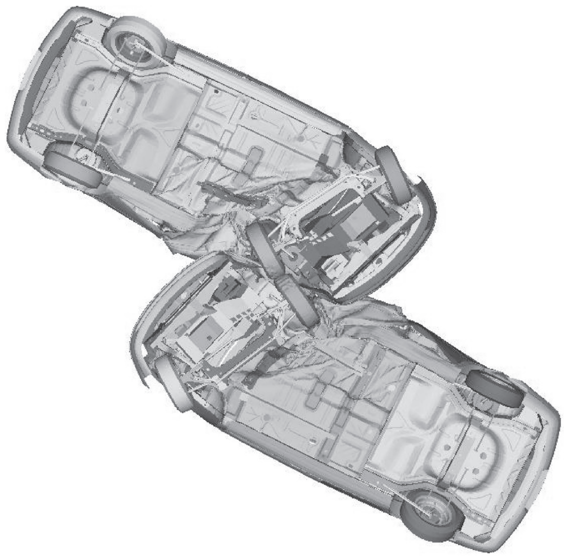


(d) Sliding collision, post-impact phase

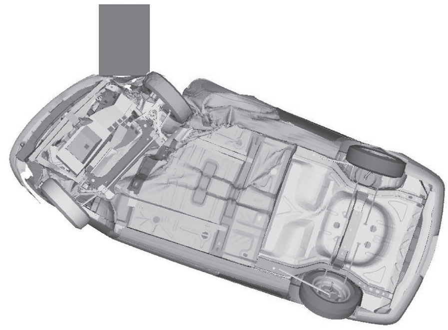
Figure 6.3.: Example of a frontal impact with narrow lateral offset, adapted from [WFBS01]

Example of a frontal impact with narrow lateral offset of two passenger cars with different mass classes (small car: 1000 kg, large car: 2500 kg), the offset is 25 %, the relative impact velocity is 200 kph. Figures (a) and (b) depicts the vehicles during impact and in rest position in a hang-on collision, (c) and (d) the same for a sliding collision. Δv and EES for the cases (b) hang-on collision and (d) sliding collision are given.

fied car-to-car crash test with two times 56 kph collision velocity. Fig. 6.5(a) illustrates the principal design. A rigid deflector designed to prevent the catching of the wheels is connected to the vehicle's longitudinal beams via a coupling bar and a telescope. The deflector (length d_F) is connected to the telescope by a translational joint, the telescope (initial length b_F) is fixed to the longitudinal beam of the vehicle. Since it is not mounted on the crash box, the system is protected against low-speed accidents. Deflector and longitudinal beams are connected to the coupling bar by pivot joints. The length of the coupling bar is c_F, and the distance between the pivot joints of the longitudinal beam is a_F. The kinematic mechanism defined by these parts and the respective pivot joints yield the desired trajectory of the outer point P of the deflector (6.2). This allows the complete system to be packaged within the vehicle design surfaces, while the deflector can be moved out during the collision. Fig. 6.5 (b) shows a possible application in a

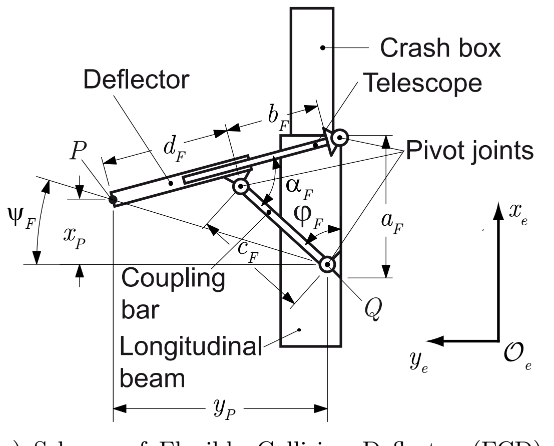


(a) Car-to-car hang-on collision, [ESF08]

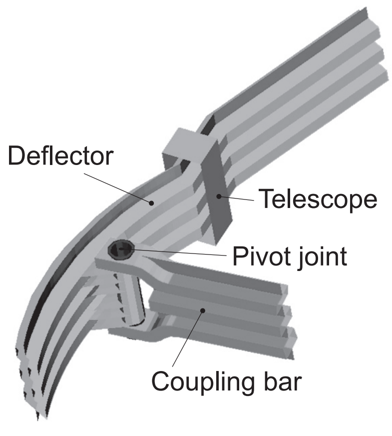


(b) Barrier hang-on collision, [ESF08]

Figure 6.4.: Comparison of hang-on collisions (FEM simulation)
The left figure depicts the bottom view of a FEM simulation using two Ford Taurus models, the right picturefigure shows the respective collision into a rigid barrier [ESF08]. The deformation behaviour is comparable.



(a) Scheme of Flexible Collision Deflector (FCD), [SE05]



(b) Application of FCD in vehicle, [ESW+06]

Figure 6.5.: Flexible Collision Deflector (FCD)

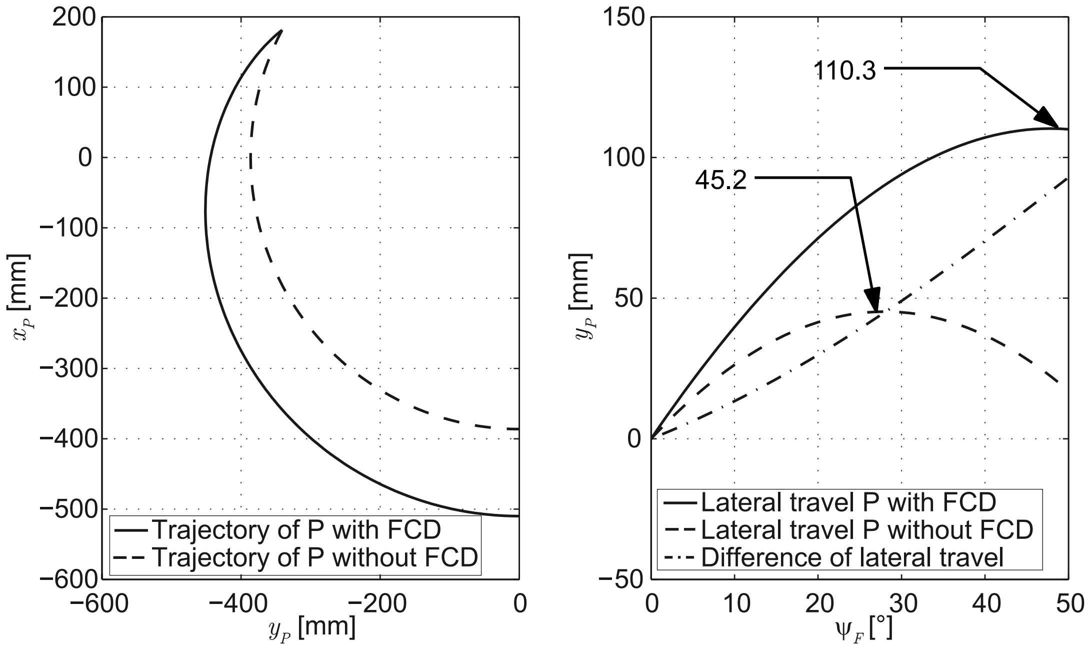
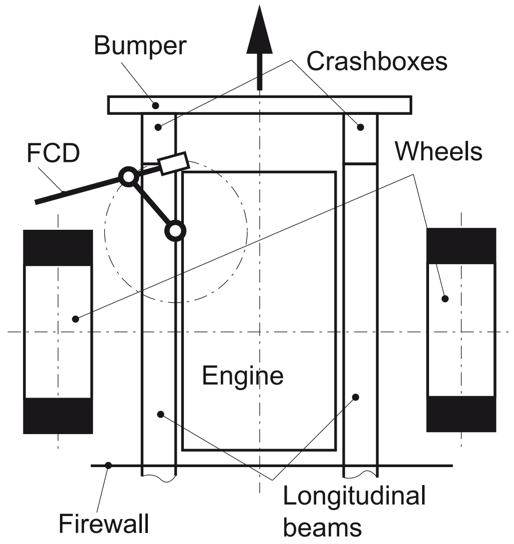


Figure 6.6.: Kinematics of point P with and without the Flexible Collision Deflector

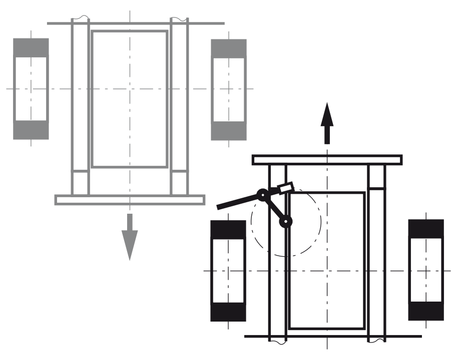
vehicle. The kinematical relations for the shown mechanism read

$$\begin{aligned}
\alpha_F &= \arctan \frac{a_F - c_F \ \cos\varphi_F}{c_F \ \sin\varphi_F} \ , \\
b_F(\varphi_F) &= \sqrt{a_F^2 + c_F^2 - 2 \ a_F \ c_F \ \cos\varphi_F} \ , \\
y_P &= (d_F + b_F) \ \cos\alpha_F \ , \\
x_P &= a_F - (d_F + b_F) \ \sin\alpha_F \ .
\end{aligned} \tag{6.2}$$

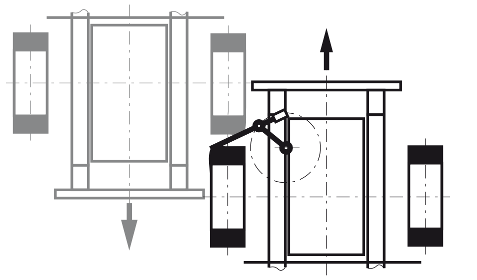
Fig. 6.6 depicts the motion of point P with the FCD mechanism compared to a circular path without the FCD. The chosen parameters were a_F=340 mm, c_F=310 mm, d_F=200 mm and $\varphi_F(0)$=31°. These parameters were obtained by maximising the lateral travel y_P of point P while avoiding intersections with other parts of the target vehicle (Ford Taurus). The maximum lateral travel of point P with the FCD system is 110 mm compared with 45 mm without FCD, i.e. an increase of 65 mm in the lateral travel. Fig. 6.7(a) schematically depicts the integration in the vehicle, (c) through (e) shows how the FCD works on full vehicle level in a frontal car-to-car impact with narrow offset (b). The collision opponent contacts the FCD system (c) and pushes the deflector (d). The kinematic mechanism supports that movement so that the deflector is positioned along the wheel of the ego-vehicle. The deflector deforms the wheels of the collision opponent during the collision such that a plane is achieved that leads to a sliding-off motion of both vehicles (e).



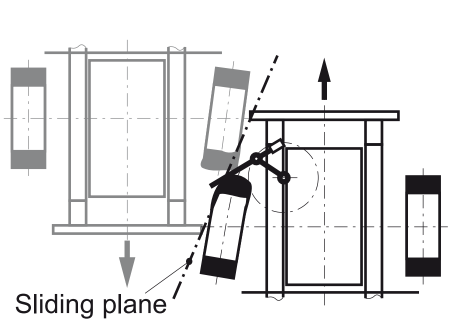
(a) Scheme of FCD on full vehicle level



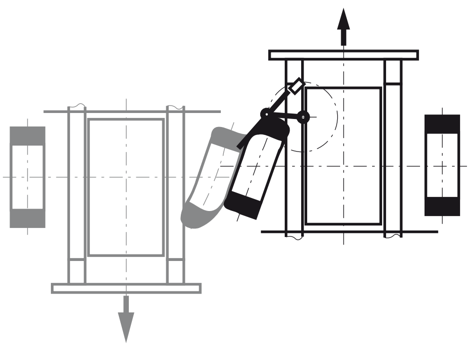
(b) Scheme of car-to-car frontal impact with narrow offset



(c) Wheels of opponent vehicle contact and activate FCD of ego-vehicle



(d) FCD is pushed out, wheels are deforming, a sliding plane is established



(e) Vehicles are sliding off each other

Figure 6.7.: Operating principle of Flexible Collision Deflector, adapted from [SE05]

6.1.2. Low-risk air bag deployment

Beside the deformation characteristics of the vehicle structure, the restraint system, comprised of the seat belt and the frontal airbag system, represents an additional important protection system in frontal collisions. Chapter 8 will present a method for controlling an adaptive restraint system based on the actual accident situation.

Regarding airbag functionality, legal requirements exist in the USA for belted and unbelted occupants (FMVSS 208, [Nat07][2]). However, the unbelted requirements for occupant protection led to the introduction of so-called "full-size" airbags, which were able to restrain unbelted occupants according to the legal requirements established in standard 208. The performance of full-size airbags was limited to laboratory crash tests with average sized adult dummies in standard seating position. In belted cases, the belt system takes up to 70% of the restraint forces, so full-size airbags capable of restraining the occupant without a belt system have to provide higher restraint forces in a shorter time. This turned out to be harmful to occupants who are outside the scope of the requirements (i.e. children, adolescents and small adults in a position close to the airbag). Accident research [KVC05] revealed non-accident-related fatalities caused by the airbag system. Therefore, the US legislation was modified in order to bring less aggressive airbag systems on the market while maintaining the unbelted crash requirements.

The modified legislation on advanced airbag systems offered car manufacturers several options for complying with the new requirements. One of the options was the so-called "low-risk deployment" of the airbag system. Crash test dummies of occupants at risk (children, adolescents and small adults) are positioned near the airbag, and injury responses due to airbag deployment in a static test set-up have to be passed. The issue was to find a compromise between in-position (crash tests with dummies in normal seating position) and out-of-position (airbag deployment tests) requirements. To meet these requirements, new airbag technologies were introduced (e.g. dual-stage inflators, new airbag folding technologies). The development was initially conducted exclusively on an experimental basis, which led to numerous tests and cost-intensive airbag systems. Packaging in the cockpit was also a problem, since the location of the airbag was one of the key elements for designing low-risk airbags. The alternative of numerical simulation was not useful in the beginning because of several problems:

- **Simulation models of crash test dummies**
  Numerical MBS and FEM models of dummies were initially developed for in-position load cases, so a direct loading of the airbag could not be reproduced with comparable dummy injury responses.

- **Simulation models of the airbag**
  Airbag models were also developed for in-position load cases. Dummy to airbag

[2] United States Code, Code of Federal Register Title 49, Chapter 301 (Motor Vehicle Safety) Part 571, Federal Motor Vehicle Safety Standard 208 (occupant crash protection)

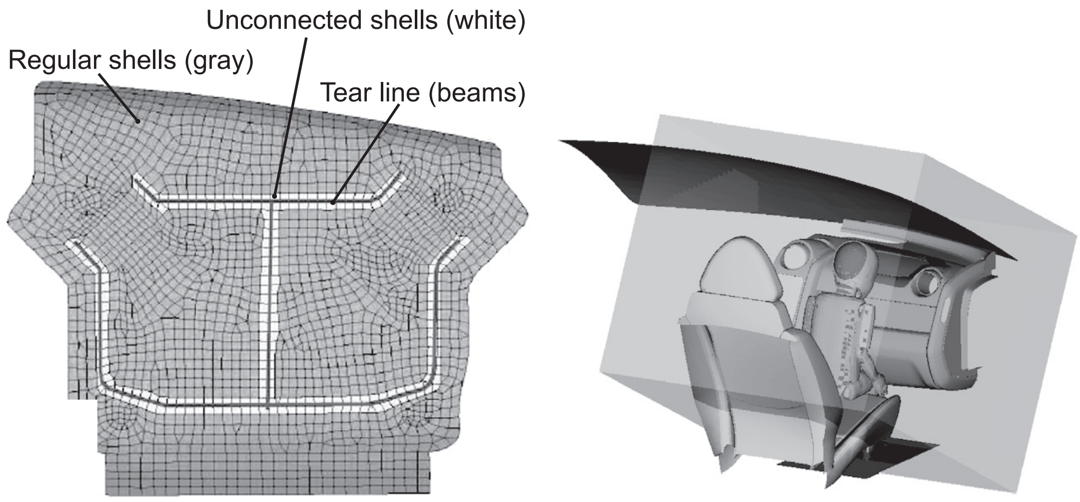
interaction requires a detailed simulation of the airbag deployment, including Computational Fluid Dynamics (CFD). The unfolding process during deployment requires the simulation of the actual folding in the airbag module.

- **Simulation model of the airbag cover**
  Since the previous technology of visible airbag covers was increasingly being replaced by invisible covers, the tearing process of plastic components had to be simulated. The opening behaviour of tear seams has a significant influence on the deployment process. Failure of weakened plastic components (tear seams) was a task that had not been previously conducted with numerical simulation.

In [RJE07] the complete process of airbag deployment for low-risk requirements was investigated. In accordance with the classical V-model approach, see also Fig. 6.10, the full system deployment test was divided into subsystem tests of the dummy, airbag and cockpit (left branch of the V-model). Next, the subsystem of the dummy was divided into relevant components (head and thorax). The airbag subsystem was then divided into airbag inflator, fabric, folding and deployment, and finally the cockpit subsystem was divided into airbag cover and other relevant parts. Consequently, these components, subsystems and system tests were simulated and the simulation model was sequentially validated in ascending order of complexity (right branch of V-model). New FEM dummy models available on the market were verified using impact tests in areas loaded by the airbag. The airbag model was enhanced with new available simulation methods for airbag deployment, in particular by Computational Fluid Dynamics (CFD) with the Arbitrary Lagrangian Euler method replacing the standard uniform pressure simulation, [HF04]). The folding of the airbag fabric was simulated according to the folding process during manufacturing, which required many changes in the airbag model. Here, the material properties of the airbag fabric and the parameters of the airbag contact model were of particular importance.

A method for simulating the opening of an invisible plastic airbag cover with tearing was introduced. The approach included a model of the plastic part and the textile coating using FEM shell elements. Material properties were carefully selected from component tests. The tearing behaviour was simulated by separating the shell meshes along the tear line. In reality, this tear line consists of small holes drilled by a laser manufacturing process. Since the resulting small elements of the FEM mesh make it impossible to model the laser holes with an FEM mesh, the separated shell meshes were connected by beam elements. A model of the failure of the beam element was developed and parameterised with experimental data from tearing tests using plastic parts from the cockpit, Fig. 6.8(a). Fig. 6.8(b) shows the setup of a low-risk deployment test with a three-year-old dummy on the passenger side.

The final results showed acceptable prediction of the dummy injury responses but called for further improvements of the method (e.g. a more numerically stable CFD simulation and the need for improved airbag inflator models). During low-risk deploy-



(a) FEM model of invisible airbag cover with tear seam, [RJE07]

(b) FEM simulation of a low-risk deployment test, [RJE07]

Figure 6.8.: Simulation of low-risk airbag deployment
The left figure shows the FEM model of an invisible airbag cover. The plastic parts are modelled with standard shell elements (gray elements). At the tear seam, the shells are not connected to each other (white elements). Beams with special failure properties, which represent the failure of small laser drilled holes, connect the white elements at the tear seam. The right figure shows the set-up of a low-risk deployment simulation. The dummy, airbag and cockpit, which have been verified on the component and subsystem levels, are assembled to form a system simulation on vehicle level.

ment, the inertial forces from the loaded dummy produce inflator characteristics that are different than those produced in a standard deployment. Therefore, a pyrotechnical simulation of the inflator is needed. The extensive time needed to prepare and validate the model was also a major problem, which has to be addressed in future work.

6.2. Side impact

Although the incidence of side impact (20 to 30%) is less than that of frontal impact, see Fig. 6.1 and 6.9, side impact is more harmful. The portion of costs related to side impact is approximately 50% higher than its portion of incidence [Kra08]. This indicates that injuries in side impacts are more severe than those from other kinds of impacts. The higher injury severity can be explained by the direct loading of the occupant by the vehicle structure. In side impacts, the kinetic energy of the impacting vehicle must be mainly absorbed by the lateral vehicle structure, which offers much less space than in frontal impacts. Since an intrusion of the lateral structure is basically unavoidable in high-speed side impacts, the dummy is loaded not only by inertial forces, but also by the lateral vehicle structure. Depending on the target market of a vehicle and the in-house requirements of OEM's, up to about twenty high-speed load cases have to be considered for side impact [EMWF07]. The resulting amount of necessary full-vehicle prototypes has led to advanced simulation methods with improved predictability of dummy injury

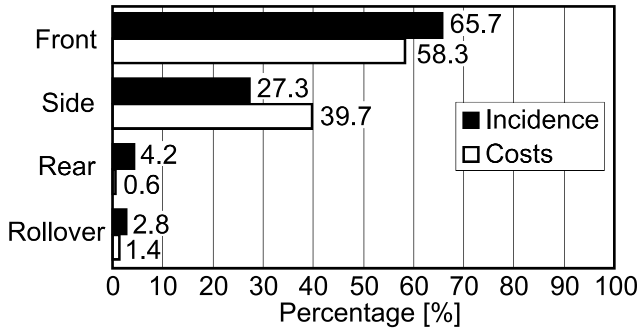


Figure 6.9.: Incidence of kind of impacts and related costs of consequences, [Kra08]

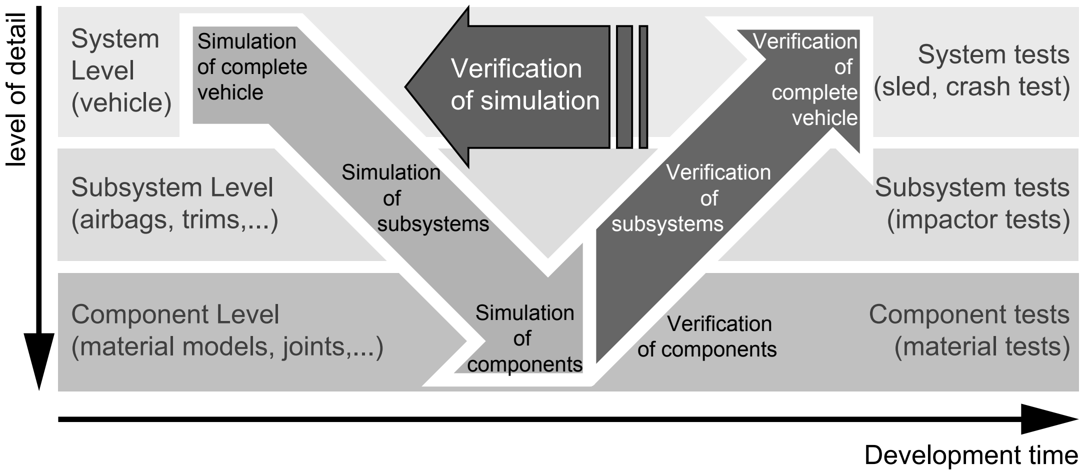


Figure 6.10.: Scheme of V-model for vehicle development
The figure shows a scheme of a vehicle development process using simulation and experiments, based on the classical V-model approach. Examples for side impact development are given.

responses. [EMWF07] presented a development process in which, based on a modified V-model approach, the dummy injury responses are assessed with numerical simulation on a full-vehicle level supported by simulation and verification experiments on the subsystem and component levels, Fig. 6.10.

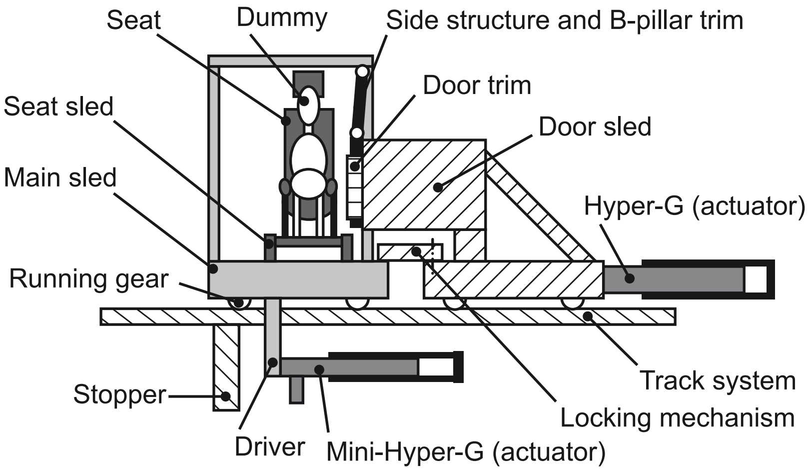
The overall crash performance is assessed by calculating the dummy injury responses on the full-vehicle (system) level. The FEM model of the vehicle structure is enhanced with subsystems relevant for the dummy injury responses (e.g. seat, door trim, belt system, lateral airbag system, dummy, detailed model of the side door). Detailed FEM simulation models of these subsystems are prepared in order to assess their prognosis quality. Relevant components of these subsystems (e.g. inflator or airbag fabric) are extracted and modelled again in detail (left branch of the V-model). The right branch of the V-model represents experiments of growing complexity ranging from component to subsystem and system level. These experiments are performed primarily to provide data for verification of the simulation. Modifications to the simulation models that are necessary to meet the verification experiments are transferred stepwise back to the full-vehicle simulation. Thus, the overall validation of the system simulation is improved without the need for a crash test on vehicle level in earlier phases of the development. Typical examples for component tests are tank tests[3], or airbag impactor tests[4] on subsystem level.

The key element in this development process is verification on the system level. For this purpose a sled rig was designed [EWHF08]. Fig. 6.11 depicts the operating principle. The sled rig consists of a main sled, a door sled and a seat sled. Except for the door, the lateral vehicle structure is mounted on the main sled. Hinges in the B-pillar allow bending of this part. The main sled reproduces the overall lateral motion of the impacted ego-vehicle with an open-loop-controlled actuator (Mini-Hyper-G). The door sled carries the side door, including the door trim. Its motion (intruding lateral vehicle structure) is achieved by a closed-loop-controlled actuator (Hyper-G). A seat sled is mounted on the main sled and carries the seat and the dummy. It allows for a relative motion of the seat being pushed by the intruding door sled. This sled differs from current state-of-the-art side impact sleds because it is not designed to reproduce the dummy injury responses in every body region with high accuracy, but rather serves as a verification tool for the full-vehicle FEM simulation. It eliminates the reproduction of the complex deformation of the vehicle structure and focuses instead on a single point of intrusion (door sled). This point is preferably the area with the highest dummy loading, which is often the rib area.

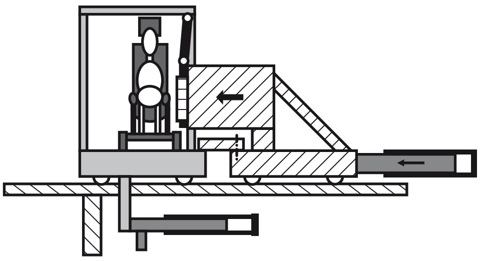
Theoretically, no pre-tests should be needed to tune dummy injury responses for a baseline test, due to the closed-loop control of the door sled motion and the definition of the sled set-up by a pre-simulation of the sled experiment. The operating process is:

- FEM side impact simulation on full-vehicle level
  Verified components and subsystems from component and subsystem testing are implemented in the standard crash simulation model including the body-in-white.
- FEM simulation of the side impact sled, Fig. 6.12 (a)
  Relevant subsystems are included in the model. Parameters of the sled set-up are tuned until the global dummy responses meet the full-vehicle simulation.
- Sled tests with parameters from the sled simulation, 6.12 (b)
- Validation of the sled simulation until the results of the sled tests are achieved.
- Transfer of modifications to the full vehicle model and optimisation of the overall side impact performance.

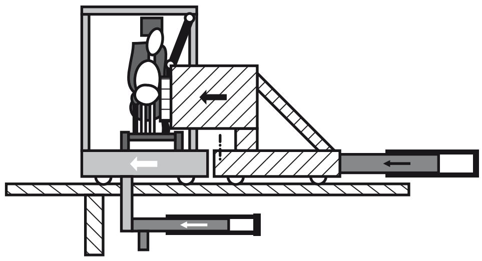
[3]Deployment of an airbag inflator in closed, rigid environment (tank) with measurement of temperature and pressure in the tank for deriving data of the gas mass-flow.
[4]Impactor or pendulum tests on the deployed airbag to quantify the restraint forces.



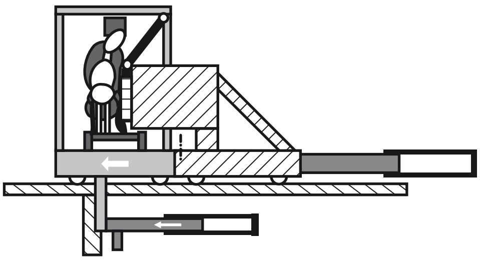
(a) Scheme of sled rig



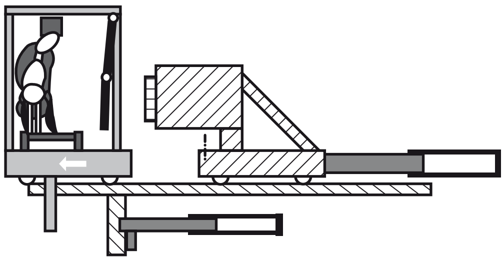
(b) Door sled starts to move



(c) Main sled starts to move



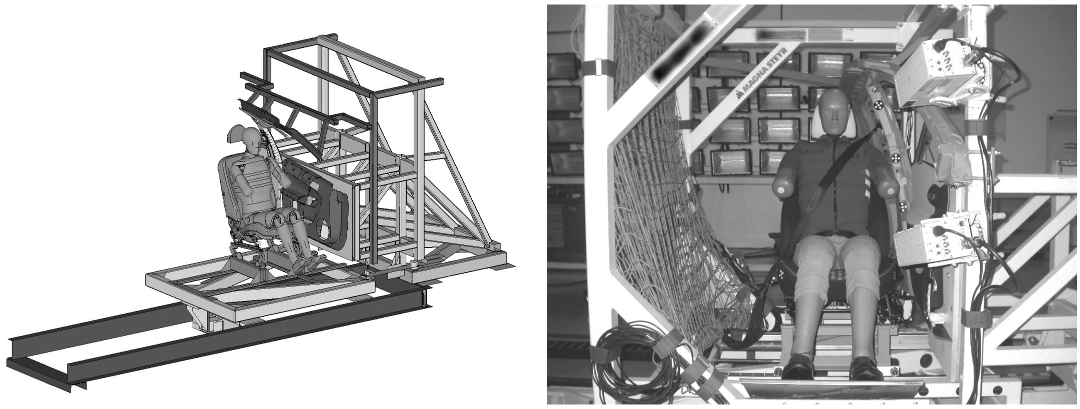
(d) Door sled stops



(e) Mini Hyper-G stops

Figure 6.11.: Operating principle of the sled rig

In (a) the main components of the side impact sled rig are denoted. From the initial position (b) the door sled starts to move and loads dummy and seat (c), powered by the Hyper-G actuator. The door sled motion represents the intruding lateral vehicle structure. Simultaneously, the locking mechanism between door and main sled is released. Powered by the Mini-Hyper-G actuator, the main sled starts to move which represents the lateral motion of the complete vehicle. The door sled stops which corresponds to the end of side structure intrusion (d). Finally, the Mini- Hyper-G actuator is decoupled and the main sled moves into rest position (e).



(a) Simulation model of sled rig, [EWHF08] (b) Side impact sled rig, [EWHF08]

Figure 6.12.: Verification on system level by sled rig

- Verification of the full-vehicle simulation model with a crash test and release of parts for production.

The development process was demonstrated by the successful production-vehicle development of a compact passenger car without a prototype generation. In principle, the development process was confirmed, but testing also revealed that, in particular, the accuracy of the FEM simulation with respect to the vehicle structure has to be improved and complemented with component and subsystem tests of the lateral vehicle body to verify the intrusion characteristics.

6.3. Rear impact

According to accident research, rear end collisions are frequent, but do not have a high risk of severe injury, [RSAG08, SSS+95]. The reasons for this are:

- The mean collision severity is low.
  The classic rear impact occurs in the same driving direction of the involved vehicles, resulting in lower collision velocities as compared to frontal collisions.

- Intrusions do not affect the occupant.
  In contrast to side impact, in which intrusion of the vehicle structure is the most critical cause of injuries, intrusions that affect the occupants are rare in a rear impact.

- The risk for ejection of the occupant outside the vehicle is low.
  Rear impacts seldom lead to occupant ejections, which is one of the major problems of rollover accidents.

- The restraint system is usually always active.
  The restraint system for the rear impact is the vehicle seat and head restraint, which are usually in place and fulfil minimum legal requirements.

Nevertheless, rear impacts cause comparatively high costs to society resulting from minor to mid-severe injuries. These injuries are often neck injuries. Since the symptoms of these injuries are manifold [Eic98, SSS+95], they cannot be classified as single injury, but have rather been assigned the term "Whiplash Associated Disorders"[5] with a scale from 0 to IV [SSS+95][6]:

- 0: No complaint about the neck, no physical sign(s),
- I: Neck complaint of pain, stiffness or tenderness only, no physical sign(s),
- II: Neck complaint and musculoskeletal sign(s),
- III: Neck complaint and neurological sign(s),
- IV: Neck complaint and fracture or dislocation.

A recent consensus paper [RSAG08] suggested a return to the term "whiplash injury" because of its widespread use and the removal of grades 0 and IV from the scale, in order to avoid misunderstandings and misinterpretations in clinical research. [RSAG08, SSS+95] reported 60 to 85% of traffic injuries are related to whiplash injuries, identifying it as the most frequent injury and cause for long-term disabilities in traffic accidents. Due to difficulties with reliable diagnosis and observation of long-term disabilities, exact figures for the cost to society are difficult to derive. Incidences of whiplash are estimated at 1 per 1000 inhabitants for the western world, but for certain countries this number can go as high as 7 per 1000 inhabitants [SSS+95]. Unresolved problems include the lack of a reliable diagnosis and differences in injury compensation systems, which can lead to possible insurance fraud [Eic95, EGM+96]. In [RSAG08] it was found that the incidence of whiplash and the cost to society are a function of the compensation system in the country considered. Nevertheless, in [SSS+95] the societal costs in the USA were estimated to an amount of 29 billion USD per year.

Whiplash injuries are not only related to a variety of clinical symptoms. In addition, the biomechanical causation cannot currently be explained by a single mechanism. The biomechanics of whiplash have been investigated by numerous research groups. The kinematics of head and neck can be divided into [Eic98, MEGS99, Gei97]:

- **Initial position**, Fig. 6.13 (a)
  The ego-vehicle is impacted in the rear, no motion of head and neck is observed. The human spine is located in its natural position (lordosis).

[5] The term "whiplash" was introduced in 1928 by H.E. Crowe in a symposium on traffic accidents. He used the term not for a specific injury, but for the typical motion of the cervical spine.

[6] Other terms used in literature are: whiplash, soft tissue injury of the neck, distortion of the cervical spine, acceleration injury of the cervical spine.

(a) Start (b) Straightening (c) S-shape (d) Extension (e) Rebound

Figure 6.13.: Head neck kinematics in rear impact, [Eic98]
Starting from the initial position with the natural curvature of the spine (lordosis), the spine straightens due to a retraction movement of the unsupported head, while the torso is loaded by the seat backrest. The head retracts backwards until this motion is physiologically limited (S-shape). The spine then extends backwards, which is followed by a forward rebound motion, [Eic98].

- **Straightening of the spine**, Fig. 6.13 (b)
  A translational motion between the unsupported head and the thorax is initiated. In a high-speed-video analysis, this can normally be seen as a “ramping up” of the occupant with respect to the seat backrest.

- **Head retraction**, Fig. 6.13 (c)
  The translational movement results in a S-shape of the cervical spine until further motion is restricted by the anatomical range of motion.

- **Extension of the cervical spine**, Fig. 6.13 (d)
  The cervical spine extends backward until this motion is restricted by the backrest.

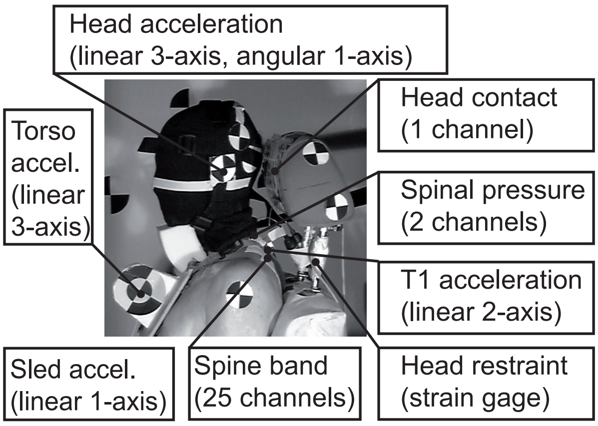
- **Rebound**, Fig. 6.13 (e).
  The whole occupant moves forward. Often, the thorax is restrained by the seat belt, which leads to a pronounced flexion motion of the head.

Although there are several hypotheses about the cause of whiplash injuries, the exact causation mechanism has not been completely clarified. Swedish research [SAH93, BSA$^+$96, BKA$^+$97] has suggested that pressure waves of the cervical spine fluid exerted by rapid acceleration of the spine lead to micro-lesions of the nerves in the spinal ganglia. A Neck Injury Criterion (NIC) was postulated, calculated by

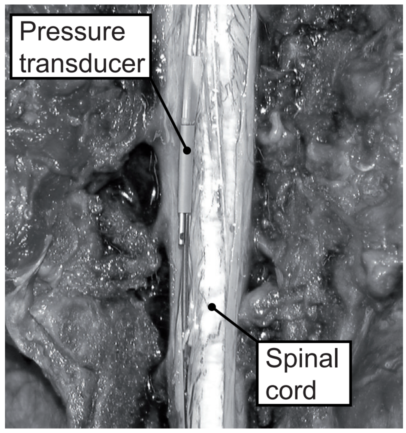
$$NIC(t) = a_{rel}(t) \cdot 0.2 + v_{rel}^2(t) \; . \tag{6.3}$$

Here, a_{rel} represents the relative acceleration in the longitudinal direction between the first cervical vertebra (C1) and the first thoracic vertebra (T1), while v_{rel} is the respective relative velocity between C1 and T1. The NIC value at maximum head retraction was the proposed injury criterion[7]. The “pressure wave” hypothesis, which was conducted using anaesthetised pigs, was confirmed by [ESG$^+$98, DLES00, EDS$^+$00]. These experiments were conducted with postmortem human subjects (PMHS) in a vehicle-comparable environment loaded with accelerations gained from accident research. Fig.

[7]The calculation of NIC has since changed slightly.



(a) Sensor location



(b) Pressure transducer

Figure 6.14.: Instrumentation of PMHS tests [EDS+00]

6.14(a) depicts the experimental setup. The subjects were instrumented with acceleration transducers on head, T1 and thorax, pressure transducers in the spinal canal and a spine band[8] for visualisation of the spinal curvature. In addition, sled and head restraint acceleration and head to head restraint contact were measured. Even without a cardiovascular circulation, the observed pressure peaks measured by transducers in the spinal cord, see Fig. 6.14(b), were in the same range as in the pig experiments.

In [ESG+98, DLES00] NIC was calculated for 28 experiments with postmortem human subjects and 70 volunteer tests. It was shown that NIC was sensitive to other proposed injury criteria. Injury thresholds of NIC for application in seat development were proposed, based on minor injuries in the volunteer tests and a severe injury in a PMHS test.

In [EKS99] the application of NIC for automotive development was demonstrated with Multibody System (MBS) simulations using a modified crash test dummy in a vehicle environment. A parameter study was done using different parameters that had reportedly influenced the outcome of whiplash. It was shown that NIC was sensitive to all of these parameters. Moreover, NIC proved to be a useful injury criterion using a dummy with only limited biofidelity[9].

Apart from NIC, further proposed whiplash injury mechanisms and criteria are hyperextension of the cervical spine [MP71], shear forces in the cervical vertebrae [Wal94], a combination of forces and moments loaded to the neck[10] (N_{ij},[KSE+98]), lesions to the

[8] 25 strain gages forming a flexible band were attached to the spine from head to pelvis

[9] Biofidelity is the comparability of responses of a human being and a crash test dummy in an impact.

[10] N_{ij} was introduced for high neck loading in high-speed impacts and not especially for whiplash injuries.

facet joints of the vertebra [KOI$^+$02], head velocity during rebound [KKL$^+$95], dynamic three-dimensional intervertebral motion beyond physiological limit (IV-NIC, [IPTM06]), lower neck load (LNL, [HSF$^+$03]), displacement of the head relative to T1 (NDC, [VD01]) and again a combination of forces and moments loaded to the neck modified for whiplash injuries (N_{km}, [SMWN02]). Evaluation of neck injury criteria was also carried out in [LCCM05] which found a direct relationship between seat backrest angle and risk for injury. A close correlation between LNL, N_{km}, neck shear forces and N_{ij} was found in sled experiments with the BioRID II dummy. NIC did not show the same correlation, but the authors could not draw definite conclusions from the experiments[11].

For years, the automotive industry was forced to develop whiplash prevention systems without knowing the exact injury causation and with no standardised evaluation tools. In 1995, a procedure was developed to evaluate seat designs with respect to risk of whiplash injuries [EGM$^+$96]. The incidence of whiplash injuries in different car seats was calculated a posteriori from accident statistics using (6.4). In this equation,

$$NIF = \frac{n_{neck}}{n_{VS90}} \cdot \frac{m_{Germ}}{m_{car}} , \tag{6.4}$$

the Neck Injury Factor (NIF) describes the incidence of whiplash injury for a specific passenger car model; n_{neck} is the number of neck injuries in the accident database “Vehicle Safety 90 (VS90)” [Ver94] for a specific car model; n_{VS90} is the total number of rear impacts in VS90 database; m_{Germ} is the total number of registered cars in Germany in 1990 and m_{car} is the number of registered cars of the considered vehicle model in Germany (1990). In addition, dynamic sled tests were performed with different seats from series production of popular passenger cars. Due to the lack of biofidelic human manikins, volunteers at low-impact speeds were used. A ranking system was applied after test evaluation using geometric position of the head restraint and dynamic measurements of the volunteers. Fig. 6.15 presents the correlation between accident statistics and the dynamic testing, which indicates that the seat design is the main influencing factor.

Recently, EURO-NCAP introduced requirements for car seats with respect to whiplash, [vREA$^+$09] that include the basic concepts of [EGM$^+$96]. Although it is stated that the injury mechanism is not yet clarified, a best-practice procedure was used based on the assumption that appropriate seat and head restraint design will reduce loading to the neck, and thereby the likelihood of whiplash injury. The procedure consists firstly of a static test assessing the geometric position of the head restraint with respect to the horizontal and vertical distances to the head, similar to [EGM$^+$96]. The second element is a dynamic generic sled test using the BioRID dummy, which has been specially designed for straight rear-end collisions in low- to mid-crash severity [DSF$^+$98]. The sled test is performed using three different acceleration pulses to avoid single point optimisation. The evaluation criteria are a combination of the state-of-art research described above.

[11] The authors suggested that the usage of Bio-RID II outside of the designed range of seating postures could be the reason for the non-linear correlation of NIC to the seat backrest angle.

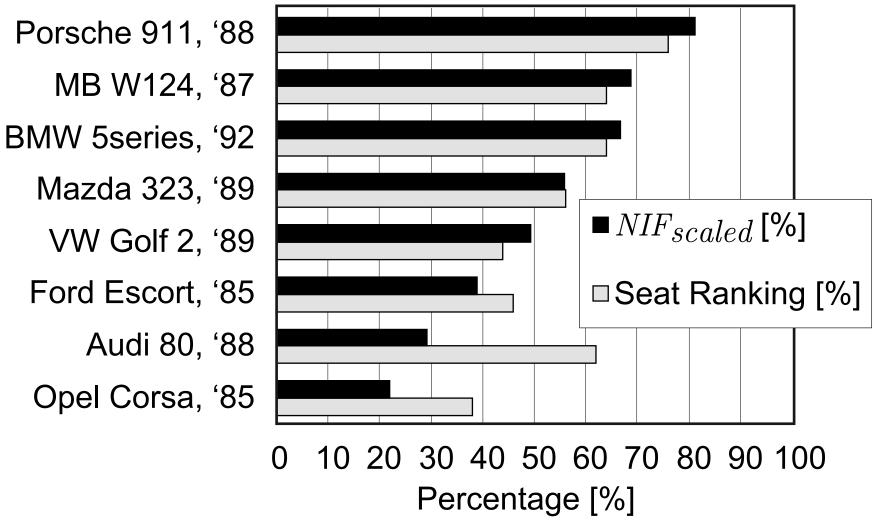


Figure 6.15.: Results of early investigations of the influence of seat design on whiplash injury, [EGM+96]

For comparison between NIF derived from accident statistics and the experimental determined seat ranking, the best NIF value was scaled to 100% (NIF_{scaled}).

The total assessment is scaled to a rating between 0 and 4, which has been added to the total EURO-NCAP rating scheme. To meet these requirements, the automotive industry has developed several different concepts for whiplash protection systems [vREA+09]. "Passive seat" designs use energy absorbing foams; "reactive head restraints" are passively activated by inertial forces of the occupant; "reactive seats" involve the whole seat design for energy absorption; and "proactive head restraints" are moved to a protective position by means of an actuator. Fig. 6.16 depicts results of an Euro-NCAP whiplash test series from 2009. It shows that a good score can be achieved with all of the designs described above. The mid range rating of proactive head restraints can be explained by the fact that the geometric evaluation was done in the design and not in the activated head restraint position, which is comfort oriented. This shortcoming of the procedure is currently under review.

Early concepts for proactive head restraints were published in [BHSE98, Eic98, SGM+96]. Fig. 6.17 depicts an example of a so-called Inflatable Head Restraint (IHR). The left figure shows a rear impact sled test with a modified dummy[12] for which the head restraint was replaced by an inflatable device. The right figure shows the same situation with the baseline head restraint at 100 ms after collision start. The Inflatable Head Restraint not only closes the gap between head restraint and head, it also reduces acceleration of the head-neck region during the collision. The result is a reduced relative motion between head and neck and reduced injury responses in that region. The effectiveness of the IHR

[12] Hybrid-III 50th percentile dummy with a so-called TRID neck for improved biofidelity in rear impact, [TRBJ96].

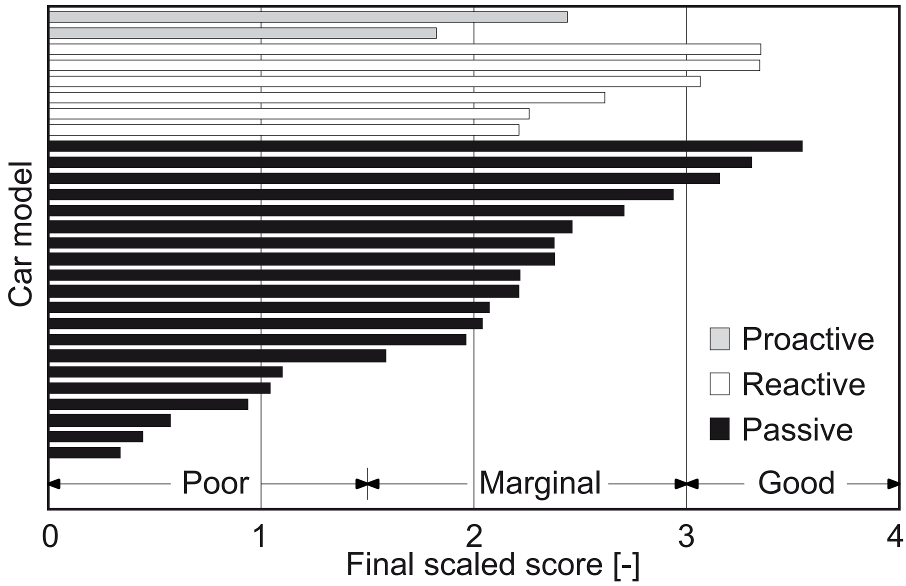
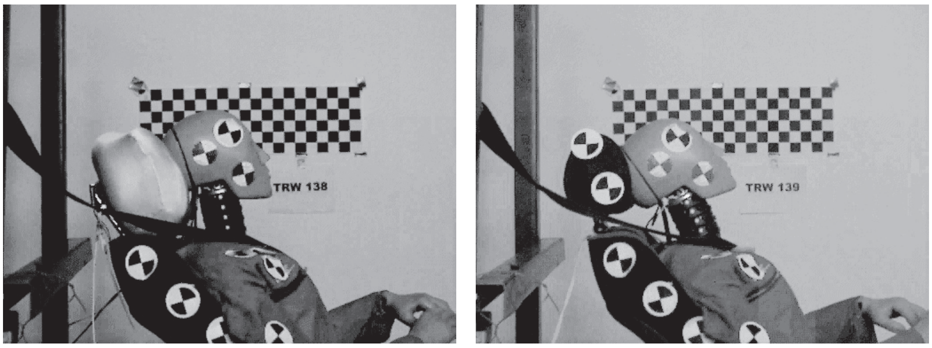


Figure 6.16.: Results of recent EURO-NCAP whiplash seat tests, adapted from [vREA+09]

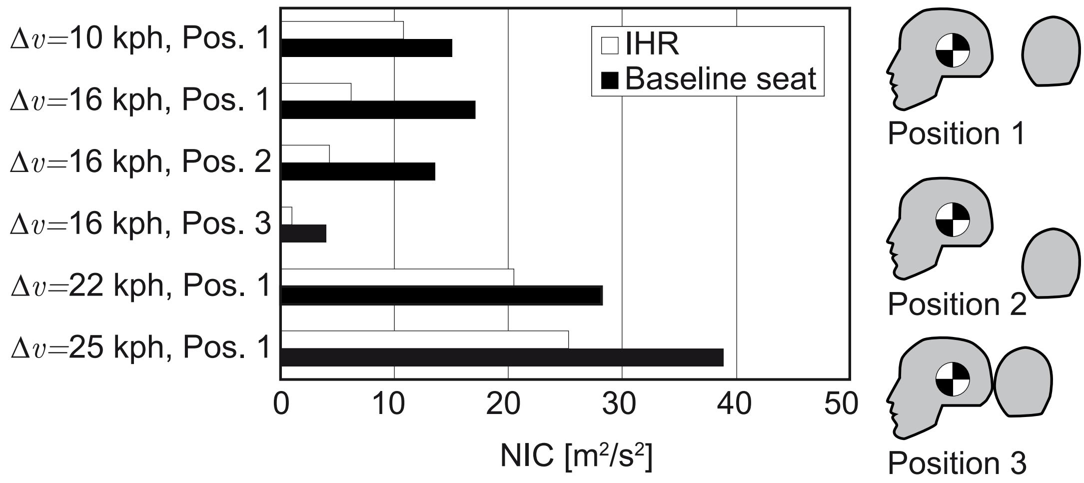


(a) With IHR

(b) Without IHR

Figure 6.17.: Comparison of dummy kinematics with and without a proactive head restraint.

The figure compares dummy kinematics at 100ms after collision start in generic rear impact sled tests with identical test set-ups. In the left figure the standard head restraint is replaced by an experimental Inflatable Head Restraint (IHR). Relative motion between head and torso is significantly reduced compared to the baseline test (right side).



(a) Results of experimental tests, [Eic98]

(b) Definition of head restraint position, [Eic98]

Figure 6.18.: Reduction of NIC by Inflatable Head Restraint (IHR)
The left figure shows an example of the reduction of injury risk by the IHR; the depicted injury criterion is NIC. Sled experiments were conducted at different collision severities (velocity change Δv) and with different initial position of the head restraint. The same seat design was used in all experiments; the only difference was the replacement of the standard head restraint (baseline tests) with the IHR.

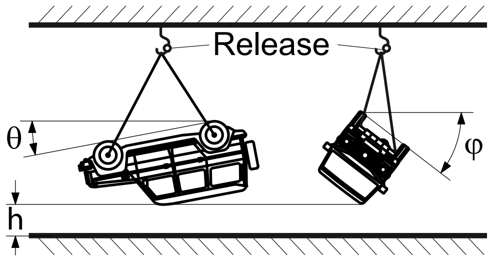
was demonstrated in several test series with dummies and volunteers. Fig. 6.18 provides an example of the reduction of NIC in sled tests at different loading severities (velocity change Δv) and head restraint positions. The benefit of inflatable head restraints was also demonstrated in [LC00, LC03].

6.4. Rollover

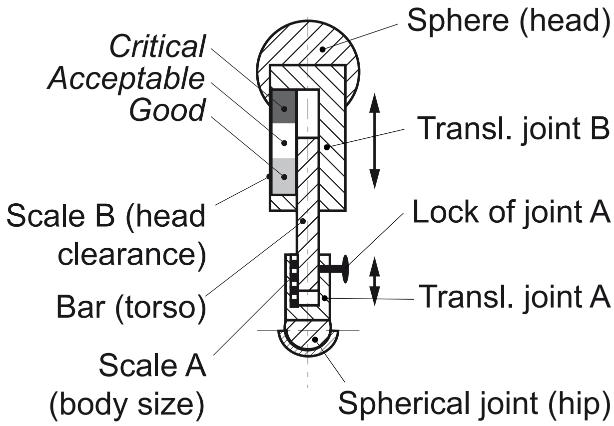
With respect to Europe, the EC project "Rollover" [Eur06] investigated this type of collision in detail. Rollover is involved in approximately 5-15% of all accidents [GSLF04, SMBF09]. The difference is related to the investigated accident database and the analysis method. Approximately one third of these cases are single rollover events, and two thirds take place in more complex accident scenarios involving multiple impacts. In these cases, the severity of the first impact is usually more important than the rollover itself. It was found that four distinct injury causation mechanisms are involved, [GS06]:

- **Localised Injury**
  Localised injuries caused by a direct impact to a body part,

- **Global Injury**
  Remote and diffuse injuries caused by a direct impact to a body part,



(a) Inverted drop test, [BBH+06]



(b) Head clearance assessment, [Eic04]

Figure 6.19.: Proposed evaluation of roof strength in rollover
The left figure shows the proposed test configuration (modified SAE J 996 inverted drop test). Tests are performed at two pitch angles ($\theta = \pm 10°$) and two roll angles ($\phi = 130°$ and $\phi = 170°$). The height was defined with $h = 0.5\ m$ resulting in an impact velocity of 20 kph. For simplicity, the combined position including pitch and roll angle is not depicted. The right figure depicts the "Rolland" device used to assess the head clearance after the inverted drop test as a performance criterion for the inverted drop test.

- **Indirect Load Injury**
  Load-based injuries produced by indirect loading associated with an impact to another body,
- **Crush Injury**
  Crush-based injury produced by crushing of the body part between deformed vehicle structures and/or outside structures.

In Europe there is no legal requirement for roof crush to limit crush-induced injuries, whereas the US legal requirement in standard FMVSS 216 requires the vehicle to pass a static test in which a rigid plate is pressed on the roof of a passenger vehicle in a certain position. In [MCC+05] it was suggested that the FMVSS 216 test set-up is inadequate for simulating real-life accidents and should be revised to represent worst case loading conditions. In [DEBP04] it was shown that requirements for roof crush can be assessed in a reproducible and repeatable manner by using an inverted drop test based on a modified SAE J996 [Soc80, BBH+06] procedure, Fig. 6.19 (a). Furthermore, this drop test can be efficiently used in FEM crash simulation without modifications to the standard FEM crash model. In [DEBP04] it was also shown that dynamic rollover tests, such as the corkscrew or the FMVSS 208 lateral rollover, are improper methods for assessing the performance of the vehicle body due to the weak reproducibility. Additionally, FEM crash simulation models would have to be enhanced for numerical stability and efficient processing time. Performance criteria for this inverted drop test were proposed by [LE06], and a procedure for reliably assessing this performance criteria with a simple human manikin called "Rolland" was developed [Eic04], Fig. 6.19(b). The manikin consists of a spherical joint located in the seat reference point (SRP, hip point). A bar representing the torso is connected to this spherical joint via a translational joint A. The body size of the manikin, which can be adjusted with this joint, is metered on scale A. The torso carries a sphere representing the head, and both parts are connected by translational joint B. After the inverted drop test, the manikin is placed in the vehicle, fixing the spherical hip joint at SRP. While maintaining contact of the head sphere to the intruded roof, the whole device is rotated by sliding the translational joint B in and out. The minimum value read on scale B is the head clearance as a measure of the survival space and is rated *Good*, *Acceptable* or *Critical*.

7. Traffic safety systems and their benefit potential

7.1. Traffic Safety Systems (TSS)

As described in chapter 5, countermeasures for traffic accidents can operate on the primary, secondary or tertiary safety level of the involved traffic element (human, vehicle or environment). The number of different traffic safety systems has become large, which has led to different attempts to categorise them. Detailed descriptions of traffic safety systems can be found in the relevant literature, e.g. [WHW09, BADS08, Kra08], and is therefore omitted from this thesis. In [BADS08] a brief overview of 161 systems is provided. For consistency reasons, this thesis will use the same terminology, in which safety systems are classified according to:

- **Primary, secondary and tertiary safety**
- **Location of installation**
  Passenger car, heavy goods vehicles, motorcycles and road
- **Functionality**
  Braking systems, handling, restraints, ...

In [Kom08] another categorisation of ADAS systems is described. Here, categorisations are based on which level they operate on (i.e. navigation, course planning or stabilisation) and the levels of complexity[1] and autonomy[2]. For orientation, the 161 traffic safety systems listed in [BADS08] are classified according to the following criteria (see Tab. A.2 in the Appendix for a complete list):

- **Phase of accident**
  - Primary safety
  - Secondary safety
  - Tertiary safety
- **Complexity of the system**
  Based on [Kom08], a rating from 0 to 5 was applied to the ten sub-items listed below; the mean value is the system complexity.

[1] Complexity of the traffic situation.
[2] Degree of responsibility taken by the vehicle.

  - Rating 1: very low complexity
  - Rating 2: low complexity
  - Rating 3: mid complexity
  - Rating 4: high complexity
  - Rating 5: very high complexity

  The rated subitems are:

  - Driving velocity
  - Curviness of the road
  - Number of road users
  - Variety of road users
  - Number/type of disturbances (obstacles, road conditions, etc.)
  - Road conditions
  - Unpredictability of the situation
  - Consequences for system failure
  - Number and complexity of traffic rules
  - Environmental conditions

- **Autonomy of the system**
  A rating from 1 to 5 is applied to classify the autonomy of the system (not the driving task), see also [Kom08] and [She92]:
  - Rating 1: driver controls, system assists
  - Rating 2: system controls, driver executes sub-tasks
  - Rating 3: system controls, driver monitors
  - Rating 4: system controls, driver takes control in emergency situations
  - Rating 5: system controls autonomously

- **Type of vehicle**[3]
  - Passenger car
  - Heavy goods vehicle
  - Motorcycle
  - Infrastructure

- **Intervention strategy**
  - Information: The driver is informed in uncritical driving situations
  - Warning: In critical driving situations the driver is warned

[3] Classification according to [BADS08]

  - Control: Stabilisation of the driver's planned course
  - Intervention: Autonomous intervention of the vehicle in critical driving situations, may be different from planned course
  - Multiple interventions

- **Level of Execution**[4]
  - Navigation
  - Course planning
  - Stabilisation

- **Main driver Benefit**
  - Comfort: Support of driver in annoying/bothersome driving tasks in uncritical driving tasks.
  - Safety: Collision avoidance, mitigation or injury risk reduction.

- **Classification of functionalities** according to TRACE project [BADS08]

Fig. 7.1 shows the level of complexity versus the level of autonomy of all systems listed in Tab. A.2 [EWss]. This data shows that most systems can be placed in one of two groups: one group (group A) works highly autonomously, but with a minor to moderate level of complexity, while the other group (group B) has low autonomy but works at different levels of complexity. This shows two distinct tendencies of traffic safety systems: They do not take driver behaviour into account, but rather restrict their functionality to situations with defined system boundaries (group A); or they assist the driver in any imaginable driving situation without taking control (group B). The reasons for this may be explained by the following [EWss]:

- **Legal restrictions**
  According to the Vienna convention on road traffic [Uni68], autonomous vehicle control is only allowed when the driver has control of his/her vehicle or when the vehicle is at least controlled according to the driver's intentions. The latter is more easily applied in well-defined situations, such as in skidding, where ESC stabilises the vehicle to follow the planned course of the driver.

- **Product security and liability**
  Traffic safety systems are obliged to improve traffic safety. Malfunctions and unintended vehicle control have to be avoided. The liability concerns of the vehicle manufacturer often lead to systems that support the driver, rather than taking control.

- **Reasons for traffic accidents**
  As explained above, reasons for traffic accidents are varied, complex and multiple.

[4] Only applicable for primary safety systems

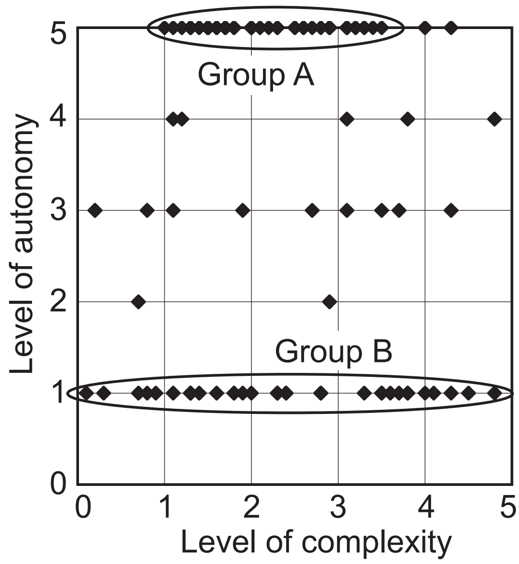


Figure 7.1.: Complexity and autonomy of traffic safety systems

> Therefore, many different systems have been developed to cover a large portion of real accidents. The safety systems can range from comparatively simple systems, such as a seat belt reminder, to more complex ones such as Collision Warning Systems (CWS).

The large number of safety systems raises the question of which systems are most beneficial with respect to real accidents. The next section describes methods for evaluating the safety potential and introduces results of investigations of the author, which are then compared to the literature.

7.2. Methodology for investigation of traffic safety system potentials

Assessing the benefit of traffic accident systems provides the basis for developing their functionalities and, as the variety of the systems listed in Tab. A.2 shows, for prioritising them in terms of market introduction. Two main methods for investigating the benefit have been established: A priori (i.e. evaluating systems before they are introduced) and a posteriori (i.e. evaluating systems after they have been introduced) [KPE^{+}06].

7.2.1. A priori effectiveness evaluation

A priori investigations analyses accidents that have already happened to determine what portion could have been avoided or for which portion could the collision severity have been decreased. The advantage is that the systems do not have to be already developed and introduced to the market. The disadvantage is that several assumptions with respect to functionality and fleet penetration have to be made. In the TRACE project, five different methods were selected for a priori effectiveness assessment [KPE+06]:

- **Target population times efficiency approach.**
  This method is useful when the functionality of a future system is undefined. First, the maximum efficiency of the investigated system is determined by identifying accidents where the system could be beneficial. The technological limitations of the system are then taken into account, often through expert prediction. One example of a technological limitation is the capability of the system to identify the traffic situation (e.g. object classification of radar-based environment recognising systems).

- **Automatic case-by-case analysis**
  For this method to work, scientists must be able to describe exactly how a system functions and integrate this function into a simulation model for a specific accident. A set of accident scenarios is calculated with and without the analysed system. The effectiveness is the difference between the two simulations.

- **Case-by-case analysis within database**
  Similar to the automatic case-by-case analysis, this method requires modelling of the accident and the analysed system. However, actual accidents out of an in-depth accident database are used. In a further step, reduction of injury risk can be estimated from the number of accidents avoided or reduced injury severity, and the data can be projected from the in-depth database with limited case numbers to national statistics.

- **Artificial Neural Networks (ANN)**
  Artificial Neural Networks are mathematical models that describe complex relationships between input and output through a network of processing elements called neurons. For example, in [YCB99] this approach was used to detect driving patterns with less chance of causing fatal or severe injury accidents.

- **Decision tree model**
  Decision trees are mathematical models consisting of sequences of branching operations. This method is used in similar applications for traffic safety systems evaluation as the ANN approach. In [CAP04] this approach was used to analyse important factors influencing fatal injuries, such as excessive speed or driver impairment by alcohol.

7.2.2. A posteriori effectiveness evaluation

A posteriori investigations assess the effectiveness of traffic safety systems after their introduction to the market by investigating accident databases. The literature mentions several methods, in [PRCZ07] the two relevant methods are described:

- **Comparison between observed and expected number of involved vehicles** This method is similar to the method for a posteriori assessment of head restraints described in [EGM+96], see section 6.3. In [FLTB97] it was used to evaluate Anti-Lock Braking System (ABS) effectiveness. It requires several steps. First, two sets of vehicle fleets including different vehicle make and model i are selected, one with a specific safety system (S) and the other without (0). Next, the different involvements of the two set of vehicles in the accident database with respect to e.g. fatal accidents is observed, $x_{i,0},\ x_{i,S}$. Additionally, the exposure of the vehicles in terms of the amount of cars registered or kilometers driven is observed, $y_{i,0},\ y_{i,S}$. Then the involvement rate IR for both sets is calculated by

$$\begin{aligned} IR_{i,0} &= \frac{x_{i,0}}{y_{i,0}} \ , \\ IR_{i,S} &= \frac{x_{i,S}}{y_{i,S}} \ . \end{aligned} \tag{7.1}$$

Next, the expected number of accident involvement $xe_{i,S}$ for vehicle i equipped with S is calculated by

$$xe_{i,S} = IR_{i,0} \cdot y_{i,S} \ . \tag{7.2}$$

Finally, a so-called risk ratio $RR_{i,S}$ is calculated by

$$RR_{i,S} = \frac{x_{i,S}}{xe_{i,S}} \ . \tag{7.3}$$

A risk ratio RR of 1 means no effectiveness of the analysed system. Below 1 it is beneficial, and above 1 it would increase the investigated accident type (e.g. fatalities). However, the results can be influenced, since different time periods are analysed (i.e. before and after introduction of the safety system). In [FLTB97], it was proposed to correct this by the relations

$$\begin{aligned} IR_{i,1} &= \frac{x_{i,1}}{y_{i,1}} \ , \\ xe_{i,2} &= IR_{i,1} \cdot y_{i,2} \ , \\ CR_i &= \frac{xo_{i,2}}{xe_{i,2}} \ , \\ RRa_{i,S} &= \frac{RR_{i,S}}{CR_i} \ , \end{aligned} \tag{7.4}$$

where $IR_{i,1}$ is the involvement rate of the considered vehicle i in time period 1 with $x_{i,1}$ and $y_{i,1}$ representing the number of involvements in a certain accident scenario and the number of registered vehicles respectively. Symbol $xe_{i,2}$ is the expected number of involvements of vehicle type i in time period 2 and $xo_{i,2}$ is the actually observed number. Symbol $y_{i,2}$ is the number of vehicles in time period 2, and CR_i is the control ratio which expresses if there is a difference due to the fact that the vehicles in time period 2 are newer than those in 1. CR_i is computed regardless of the considered safety system S. Finally, the adjusted risk ratio $RRa_{i,S}$ of vehicle model i with respect to safety system S is calculated. Obviously, to achieve statistically significant results, the investigated database has to be large, since the subsamples with respect to a specific make, model and the safety system can be low compared to the total number of cases in the database.

- **Evaluation of relative crash risk (odds ratio method)**
  In the EC project TRACE, this method was primarily used to perform a posteriori assessment of the effectiveness of safety systems [PRCZ07]. Introduced in [Eva99], it is based on a comparison of case and control groups. First, two samples are selected, vehicles equipped (indexed S) and unequipped (indexed 0) with a safety system. Then a pertinent case group indexed p and a control group indexed c are selected. The case group must include accidents where the analysed safety system could have shown benefit. A typical example is "loss of control" accidents, which could have been avoided with ESC. Naturally, the accident database has to include the related coding for this accident category. The control group must include a sample for which the safety system should be of low influence, e.g. dry roads when analysing ABS [Eva99].

The "crude" odds ratio $R_{1,S}$ is calculated by

$$R_{1,S} = \frac{\frac{x_{S,p}}{x_{S,c}}}{\frac{x_{0,p}}{x_{0,c}}} , \tag{7.5}$$

where $x_{S,p}$ and $x_{0,p}$ are the numbers of equipped and unequipped vehicles in pertinent cases (case group), and $x_{S,c}$, $x_{0,c}$ the numbers of vehicles in the control group. The effectiveness E_S reads:

$$E_S = (1 - R_{1,S}) \cdot 100 \; [\%] , \tag{7.6}$$

and describes the portion of avoided accidents in the case group which is a subset of the accident database depending on the investigated system. In order to correct the database bias caused by comparing different model years, the odds ratio can be corrected by the ratio R_2,

$$R_2 = \frac{\frac{x_{2,p}}{x_{2,c}}}{\frac{x_{1,p}}{x_{1,c}}} , \tag{7.7}$$

where $x_{1,p}$ and $x_{2,p}$ are the numbers of pertinent cases of model year 1 and 2. Variables $x_{1,c}$ and $x_{2,c}$ are the respective cases in the non-pertinent control group. Variable Ea_S reads:

$$Ea_S = (1 - R_S) \cdot 100 \; [\%] , \tag{7.8}$$

and represents the adjusted effectiveness of safety system S, where R_S holds:

$$R_S = \frac{R_{1,S}}{R_2} . \tag{7.9}$$

The drawback in this approach is the selection of non-pertinent cases in the database. The assumption of non-effectiveness of ABS on dry roads in [Eva99] was not supported by other studies, e.g. [HHJ98]. In [PRCZ07] this was avoided by selecting the whole database as the control group.

Another drawback of the odds ratio approach is the assumption that only the selected specific safety system influences the results. In reality, other external variables may affect the analysis, such as the gender and age of the driver[5]. In [PC06] it was proposed to consider the influence of external variables by extending the method using an "adjusted odds ratio" approach based on logistic regression. A detailed description of the method can be found in [PRCZ07]. It basically assumes a linear model for the influence of the external variables.

An advantage of the odds ratio method is that it allows for investigation of multiple safety systems, for example vehicles equipped with ABS which are enhanced with ESC. The effectiveness $E_{S1,add}$ of safety system $S1$ in addition to $S2$ is calculated by the additional odds ratio $R_{S1,add}$. It compares the number of vehicles equipped with safety system 1 in pertinent and non-pertinent accident scenarios $x_{S1,p},\ x_{S1,c}$ to those of safety system 2 $(x_{S2,p},\ x_{S2,c})$. Thus, it holds

$$\begin{aligned} R_{S1,add} &= \frac{\frac{x_{S2,p}}{x_{S2,c}}}{\frac{x_{S1,p}}{x_{S1,c}}} \ , \\ E_{S1,add} &= (1 - R_{S1,add}) \cdot 100\ [\%] \ . \end{aligned} \tag{7.10}$$

The following section describes the methodology for assessing the effectiveness of safety systems that was used in [ET08, ETRH09]. The method is an a priori assessment of the "case-by-case analysis within database" type.

7.3. Safety benefit of ADAS in fatal crashes in Austria 2003

As reported in the literature, the results of investigations into the effectiveness of ADAS systems have several drawbacks.

- **Level of detail in statistical accident databases**
  These accident databases often lack detailed information on the course of the accidents. Statistical evaluation can be performed by means of global data, such as "road departure of single vehicle", which indicates a benefit of Electronic Stability Control (ESC).

[5] For example, it could be possible that older male drivers prefer to buy cars equipped with ESC compared to young females.

- **Number and representativeness of in-depth accident databases**
  Analysis of the pre-collision phase requires time-consuming accident reconstructions and a high level of detail regarding the documentation of the accident in the database. This limits the number of cases and the significance of results, especially when comparing different safety systems. Often, in-depth databases are restricted to local areas, which makes it difficult to draw broader conclusions, such as on an European level.

- **Comparability of potentials on different systems**
  Most investigations in the literature focus on a specific system, for example the Electronic Stability Control. A comparison of the benefit of different *systems* based on results from different *databases* investigated with different *methodologies* is usually not possible.

To overcome these drawbacks, a retrospective case study by means of numerical simulation was performed [ET08, ETRH09]. In the following section, this study will be referred to as the "RCS-TUG"[6] study.

7.3.1. Methodology of RCS-TUG study

7.3.1.1. Database

The database used was ZEDATU, the in-depth accident database of fatal traffic accidents in Austria, [TS06, Tom07]. In November 2009, the database contained about 700 cases. For ADAS potential analysis, the year 2003 was selected, since it was the best-documented year, containing approximately 60% of all cases (514 out of 848 cases with 931 fatalities). With 763 database arrays per accident, ZEDATU is one of the most detailed in-depth databases. The arrays are based on the standardised STAIRS protocol [VLM$^+$99], supplemented by additional arrays developed in the EC projects PENDANT, RISER and ROLLOVER. From these 514 cases, 217 cases were selected, and the reconstruction of the complete accident course was carried out. It starts at the moment of transition from a normal to critical driving situation and lasts until the rest position after the collision of the involved vehicles. The following selection criteria were used:

- The collision phase of the case had already been reconstructed by an accident reconstruction expert from the point of first contact.
- The level of detail of the accident documentation allows for the reconstruction of the pre-collision phase within an acceptable level of accuracy.

7.3.1.2. Reconstruction of pre-crash phase

The reconstruction of the accident was carried out by numerical simulation using the specialised software package PC-Crash, [Ste09a]. For simulation of the pre- and post-crash

[6]Retrospective Case Study of Graz University of Technology

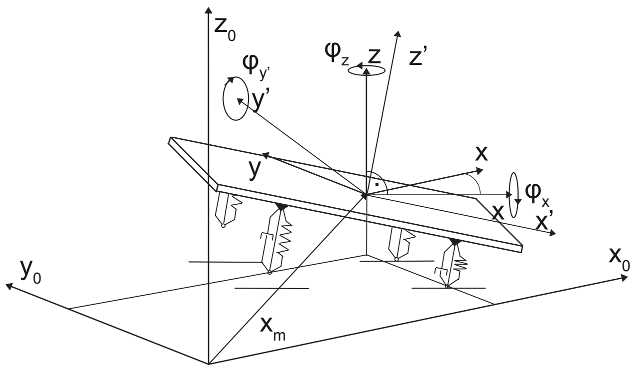


Figure 7.2.: Modellisation of vehicles in PC-crash [Ste09a]

vehicle dynamics, it includes a discrete three-dimensional kinetic forward simulation with an explicit time integration of the involved object's kinematics. Objects are modelled as individual rigid bodies. Tyre forces, air resistance, wind, gravity and trailer coupling forces can be applied as external forces. For the collision phase, both momentum and forced-based collision models are provided. Vehicle parameter can be obtained by a database that covers almost the complete vehicle fleet. Vehicles are discretised as double-track models with Voigt-Kelvin elements representing the suspension system for each wheel, see Fig. 7.2. Different tyre models are provided for the description of the tyre to road contact.

In the RCS-TUG study, emphasis was put on the pre-collision phase. This requires a more accurate modelling of the vehicle dynamics than classical accident reconstruction. In particular, when assessing the benefit of vehicle stability control systems (ABS, ESC), it is essential to calculate tyre forces with sufficient accuracy. Therefore, an optional tyre model was chosen in PC-Crash, which is based on a former TMeasy approach [HRW03, HRW07, WH07]. TMeasy is a semi-physical[7] tyre model and provides nonlinear modelling of longitudinal, lateral and vertical tyre forces and moments. Reduced longitudinal and lateral tyre forces due to combined loading as well as the influence of tyre camber angle, rolling resistance and pneumatic trail are taken into account. Road data (grip potential and road topology) is taken from the PC-Crash environment. Since information about make and condition of the tyres is normally unavailable in the accident database, parameters of a typical passenger car tyre were chosen, Fig. 7.3. Tyre forces of wheel i in longitudinal F_x and lateral direction F_y are calculated depending upon longitudinal slip s_x, slip angle α, vertical wheel load F_z and grip potential μ between tyre and road, Fig. 7.4. The basis functions for the tyre forces read:

$$\begin{aligned} F_{x,i} &= f(F_{z,i}, s_{x,i}, \mu_i) \ , \\ F_{y,i} &= g(F_{z,i}, s_{y,i}, \mu_i, \gamma_i) \ , \end{aligned} \tag{7.11}$$

[7] Most parameters of the model have physical meaning and can be derived directly from tyre test data.

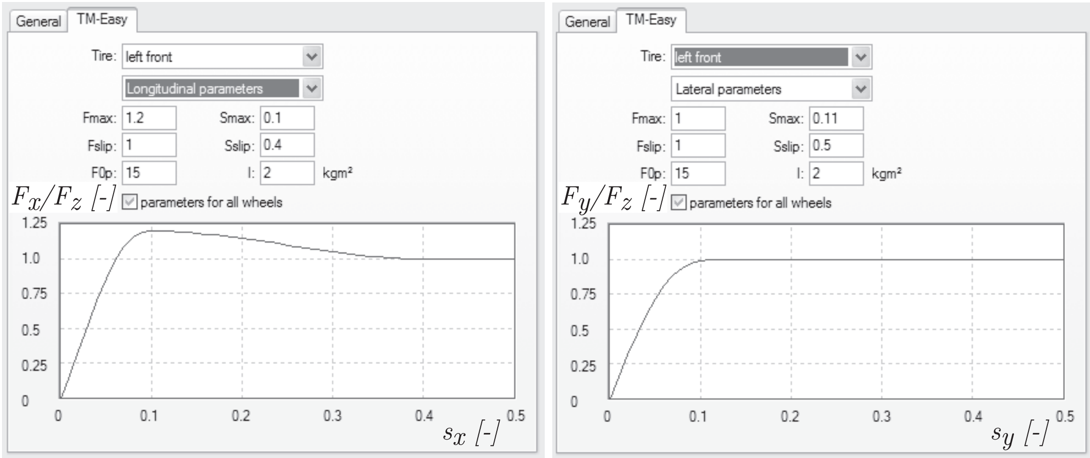


Figure 7.3.: Tyre parameter in RCS-TUG study

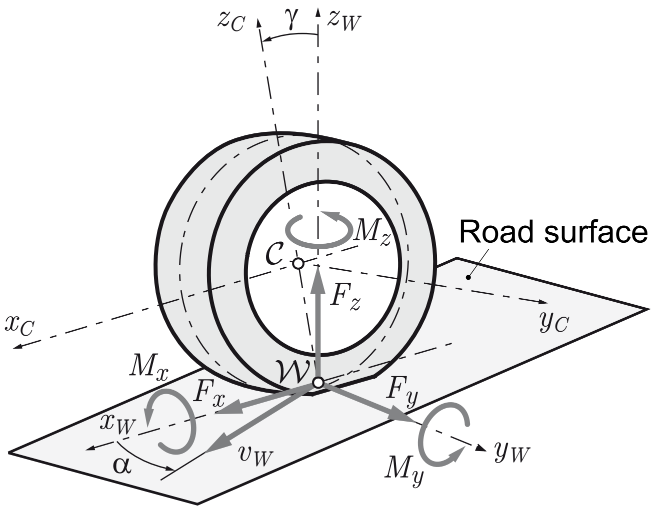


Figure 7.4.: Definition of tyre forces and moments (ISO 8855), adopted from [HRW07]

Coordinate system $\{\mathcal{C}, x_C, y_C, z_C\}$ is fixed at the wheel centre $\mathcal{C}$ and aligned with the wheel orientation. Coordinate system $\{\mathcal{W}, x_W, y_W, z_W\}$ is fixed at surface contact point $\mathcal{W}$ and normal to the road surface. Forces F_x, F_y and F_z are tyre forces with respect to $\mathcal{W}$. Slip angle α is the angle between the wheel velocity vector v_W and the x_W axis. Camber angle γ is the angle between z_C and z_W axis. Moment M_y for a free rolling wheel results from an eccentrical force application point of F_z, and M_z results from an eccentrical force application point of F_y.

where the following slip definitions hold:

$$\begin{aligned}
s_{x,i} &= \frac{r_{dyn,i} \cdot \omega_i - v_{x,i}}{r_{dyn,i} \cdot \omega_i} \ , \\
s_{y,i} &= \frac{2 \cdot \alpha_i}{\pi} \ , \\
\alpha_i &= \arctan(\frac{v_{y,i}}{v_{x,i}}) \ .
\end{aligned} \tag{7.12}$$

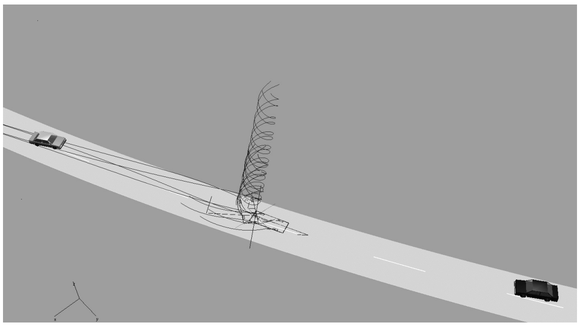
In (7.12), $r_{dyn,i}$ is the dynamic tyre radius, ω_i the rotational and $v_{x,i}$ the translational wheel velocity. Although a separate tyre parametrisation for each vehicle would be preferable and would increase the accuracy of the simulation, the level of accuracy that can be obtained by accident reconstruction does not require this level of detail, which would demand experimental data for each tyre.

After selecting a case, a plausibility check of the reconstruction of the collision phase was done[8]. Next, the point of first contact was defined as the starting point of the simulation, and a backward simulation was carried out until the beginning of the critical driving situation. This was usually about 3 seconds before first contact, which is also related to the capabilities of environment recognising systems and their collision prediction functionalities. The backward simulation within PC-Crash can be performed as a kinetic simulation comparable to the forward simulation. In case of pure sliding, the equations of motion are integrated as in the forward simulation but using reversed tyre forces. In the case of a normal driving situation, a predictor-corrector algorithm is used, where a simulation with a negative time-step (predictor loop) allows for the calculation of the initial conditions for another forward simulation (corrector loop). After performing the backward simulation, the transition from standard to critical driving situation was defined as the new starting point of the simulation.

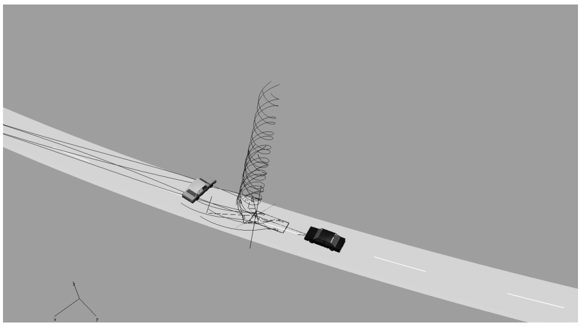
Finally, the pre-collision phase was optimised with respect to available expertise, testimonies, accident marks and other available information. As an example, Fig. 7.5 depicts the course of an accident which was reconstructed with the method described above. For unknown reasons, a passenger car (vehicle 1) leaves its designated driving lane, Fig. 7.5(a). The ensuing emergency braking of vehicle 1 takes place with different grip between left and right wheels. This difference in grip could be reconstructed because only one lane was cleared of snow, according to the available police report. The resulting yaw moment causes skidding, Fig. 7.5(b), and the oncoming passenger car (vehicle 2) cannot avoid the accident, Fig. 7.5(c). Fig. 7.5(d) depicts the vehicles in rest position.

The accuracy of the accident reconstruction of the collision phase simulation is within certain limits related to the quality of the accident documentation and the experience and elaborateness of the reconstructing person [WR00]. Methods for taking this into

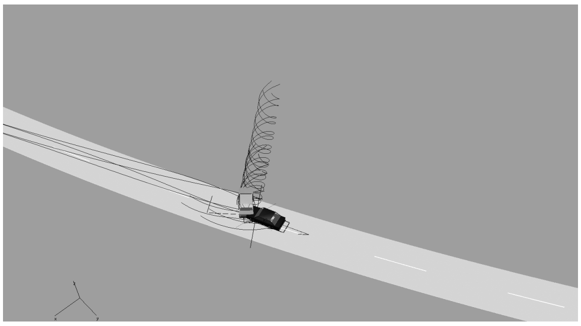
[8] Only cases with a pre-existing accident reconstruction of the collision phase by an accident reconstruction expert (e.g. from court cases) were selected.



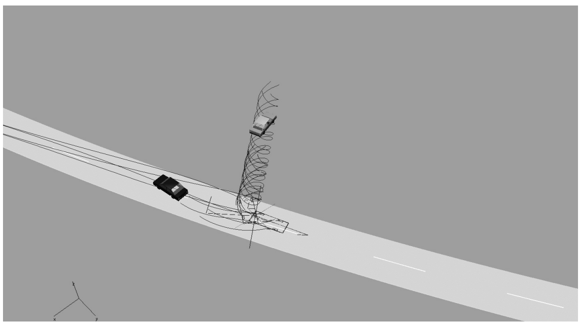
(a) t=0 s, vehicle 1 (left) leaves lane



(b) t=1.5 s, skidding of vehicle 1 due to μ_{split} braking



(c) t=2.6 s, vehicles in collision position



(d) t=4 s, vehicles in rest position

Figure 7.5.: Numerical simulation of ZEDATU case TUG-4154101

Due to unknown reasons, the vehicle on the left side leaves its designated driving lane, the driver brakes and - due to different friction on both lanes resulting from incomplete snow clearance - the vehicle starts to skid (typical μ_{split} situation)[9] account in numerical simulation are based on statistics and described in studies such as [WU07] and [Bat08]. The accuracy of the pre-collision phase is even less, since driver behaviour, which differs between human beings, has to be taken into account. The result of the simulation in the RCS-TUG study is an acceptable but not perfect analysis of the complete course of the accident for the present application: the investigation of traffic safety system benefits.

7.3.1.3. Simulation of vehicle-based safety systems

The following section summarises the implementation of *intervening* vehicle-based safety systems (VSS) into the accident reconstruction software. For a more detailed description refer to [Roh09, ETRH09].

- **Anti-Lock Braking System (ABS)**
  ABS systems prevents tyres from locking during braking. This results in shorter braking distances and allows lateral forces for steering manoeuvres by limitation of longitudinal tyre forces. ABS was taken into account by activating the existing ABS functionality in the accident reconstruction software [SM01]. Using the nonlinear tyre model TMeasy, the longitudinal brake force $F_{x,i}$ for each wheel i is reduced to 60 % of the grip potential provided by the vertical force F_z and the coefficient of friction μ,
  $$F_{x,i} = 0.6 \mu F_{z,i} \ , \tag{7.13}$$
  when a certain longitudinal slip $s_{x,i}$, see (7.12), is exceeded. The reduction of brake force in combined lateral tyre loading ($\alpha_i \neq 0$), see (7.12), is considered by the superposition model of [HRW03]. The default parameters of the PC-Crash software were not changed, since a vehicle-specific optimisation of the ABS function was not reasonable.

- **Electronic Stability Control (ESC)**
  ESC systems prevent skidding of the vehicle by observing the planned path of the driver and applying a stabilising yaw moment in the case of excessive under- or oversteering. The yaw moment is achieved by selective braking forces on different wheels (i.e. the inner rear wheel for under-steering and the outer front wheel for over-steering). The used accident reconstruction software provides pre-installed functionality for ESC [MS08]. This ESC model applies a wheel selective braking force F_B according to
  $$\begin{aligned} F_B &= |\omega_D \cdot k| \ , \\ \omega_D &= \omega_R - \omega_O \ , \\ \omega_R &= \frac{v_x}{R} \ , \end{aligned} \tag{7.14}$$
  where ω_D is the difference between the reference yaw rate ω_R and the observed ω_O, as measured by on-board sensors. The "control factor" k scales the intervening brake force. The reference yaw rate ω_R is calculated using the measured driving

velocity v_x and the intended corner radius R of the driver, see [MS08]. It is assessed from steering angle δ, driving velocity v_x, grip μ and maximum lateral acceleration $a_{y,max}$.

The software provides three parameters to adjust the ESC control algorithm:

- **Cycle time**
  It describes the time between two cycles for the ESC intervention. The default parameter 0.05 s remained unchanged.
- **Yaw rate threshold** ω_t
  This parameter initiates an ESC intervention whenever it is exceeded.
- **Control factor** k
  This factor is used to adjust the magnitude of the intervening braking force (see above).

Vehicle manufactures use different intervention strategies for ESC. The focus is on either driving stability, handling agility or a compromise between the two. In contrast to the ABS approach, the default parameters were modified in the RCS-TUG study [Roh09]. The objective was to identify whether or not different ESC strategies have a significant influence on the accident outcome. Thus, three different intervention strategies were used:

- ***ESC***: The parameters were set to achieve a compromise between a stable yaw control and agile handling characteristics.
- ***ESC cons.***: The focus of the yaw control was on stability (i.e. the ESC strategy was conservative).
- ***ESC sport.***: The focus was on agility (i.e. the desired driving behaviour was sportive).

This was achieved by tuning the parameters ω_t and k in a specific driving manoeuvre related to lateral vehicle dynamics. Since there are currently no standardised testing procedure for ESC systems, a proposal to assess ESC related driving behaviour described in [Zan07] was used. The assessment is based on a simultaneous evaluation of the maximum lateral acceleration $a_{y,max}$ and the integral of the side slip angle I_β according to

$$I_\beta = \int \frac{\beta^2}{|\beta|} dt \ . \tag{7.15}$$

The chosen manoeuvre was an open-loop pulse-steer-input manoeuvre, similar to ISO 7401, [Tec03]. The vehicle is driving in a straight direction at a constant test velocity of 90 kph. A sudden defined turning of the steering wheel according to Fig. 7.6 induces a yaw moment which results in skidding without ESC. Fig. 7.7(a) depicts the trajectories of the centre of gravity for an exemplary vehicle (BMW 530d, initial speed $v_0 = 90\ kph$). The steering input as depicted in Fig. 7.6 was applied, and start and end position of the vehicles after 4 s are shown. The arrow in

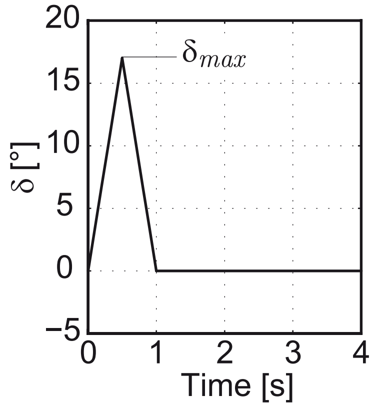


Figure 7.6.: Steering angle δ vs. time in pulse-steer-input driving test

The tested vehicle is accelerated to the desired test velocity v_x (90 kph in the RCS-TUG study); then the steering wheel is turned within 0.5 s with constant rate until the desired maximum steering angle δ_{max} at the wheel is achieved (equivalent to 50m corner radius in the RCS-TUG study). The steering angle is then reduced again at constant rate to zero during 0.5 s.

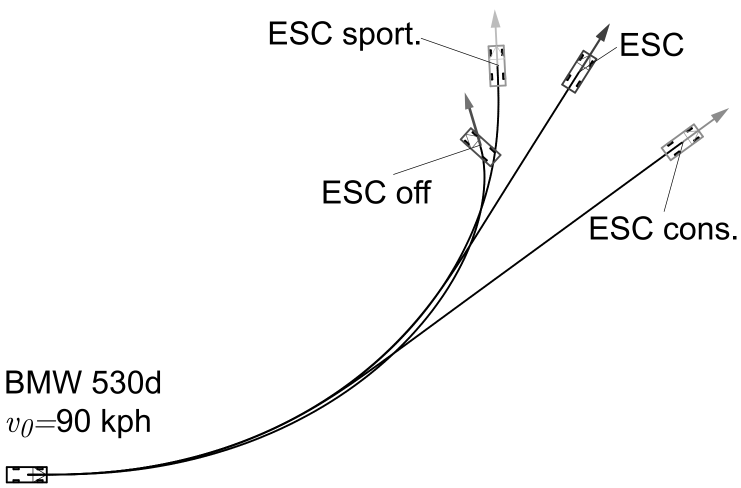
Table 7.1.: ESC parameter

Vehicle	Yaw rate threshold ω_t	Control factor k
ESC	0.1	0.6
ESC sport.	0.2	0.4
ESC cons.	0.1	0.9

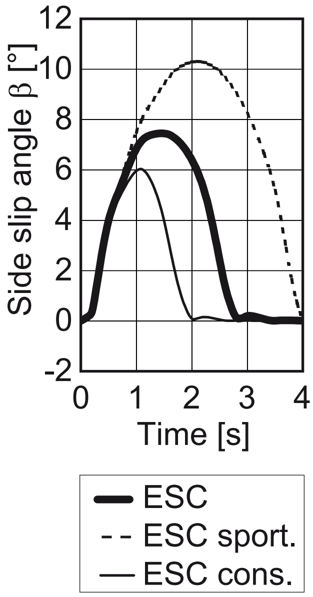
the end position denotes magnitude and orientation of the vehicle velocity vector. With ESC off, the vehicles skids, as expressed by the side slip angle β, which is the difference between the orientation of vehicle longitudinal axis and the velocity vector. ESC stabilises the vehicle with different resulting trajectories, and after 4 s the side slip angle is already reduced to zero, see Fig. 7.7(b).

Tab. 7.1 lists the chosen parameters for the three different ESC interventions. These parameters were derived by applying the driving manoeuvre described above to different vehicles in PC-Crash. It turned out that the same parameter setting could be chosen for different vehicles and vehicle types. Therefore, the identification of ESC parameters was not repeated for each vehicle in ESC-relevant cases of the ZEDATU database during the analysis of the pre-collision phase.

In [Zan07], the assessment of ESC systems was done by drawing the maximum value of the lateral acceleration $a_{y,max}$ and the side slip angle integral I_β in a common diagram. The intervention based on sportive, normal and conservative is related to the position of these values in the diagram. Higher numbers of the side slip angle integral mean higher agility, while lower numbers of the maximum



(a) Vehicle trajectories with different ESV parameter settings



(b) Results for side slip angle β for different ESC parameters

Figure 7.7.: Results for different ESC parameter settings with vehicle BMW 530d, pulse-steer-input manoeuvre with $v_0 = 90\ kph$

Fig. 7.7(a) shows trajectories of the vehicle centre of gravity with different ESC parameter settings, (b) shows the related side slip angle time histories.

lateral accelerations mean higher stability. The values of Tab. 7.1 were derived by analysing the vehicle trajectories and the side slip angle time histories, see Fig. 7.7. Fig. 7.8 shows the results for a C- and a D-segment passenger vehicle using the parameters of Tab. 7.1.

- **Predictive Brake Assist (PBA)**
  Predictive brake assist supports the driver in emergency braking. The driving lane in front of the ego-vehicle is observed by environment recognising systems. In the case of possible collision with an obstacle, PBA assists the driver. The support strategy on production cars differs. It ranges from Collision Warning Systems (CWS), autonomous braking with reduced braking force to autonomous braking at low velocity (city traffic). The modelling of the Predictive Brake Assist in accident reconstruction is complex, since firstly most production systems take the driver reaction into account; secondly the function of the environment recognising system has to be modelled; and thirdly there are different intervention strategies on the market. The following section summarises the used approach in the RCS-TUG study. More information is provided in [ET08, ETRH09, Roh09].

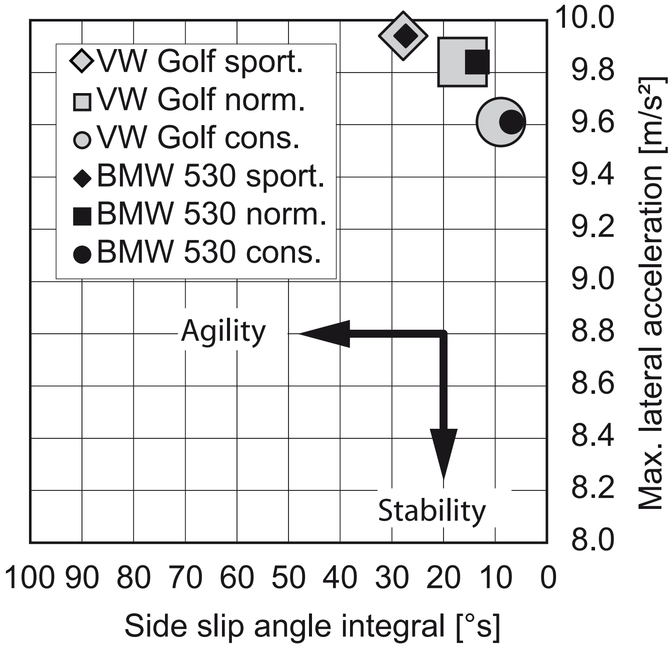


Figure 7.8.: Assessment of different ESC parameter settings
Results for side slip angle integral and maximum lateral acceleration for different ESC parameter settings for two vehicles. Since the results of different vehicles and vehicles types were similar, the ESC parameter settings of Tab. 7.1 were used throughout the RCS-TUG study, [Roh09].

Driver behaviour

The driver behaviour was modelled by defining three possible reactions to the intervention of PBA, which covers the range of possible driver reactions.

- a: The driver reacts with 0.8 s reaction time to an Human-Machine Interface (HMI) warning.
- b: The driver reacts with 0.8 s reaction time to a partial autonomous braking.
- c: The driver does not react, the system operates autonomously.

The choice of 0.8 s for a mean driver reaction time corresponds with findings in literature, e.g. [SS05][10].

Environment Recognising System (ERS)

An idealised radar-based ERS was modelled in PC-Crash by evaluating if an obstacle is within an ERS range as depicted in Fig. 7.9. The conditions for obstacle detection were:
- An object is in the same driving lane.
- The object is within the ERS range and can be tracked at least 100 ms.

[10] In [SS05] a reaction time of 0.82 s in emergency braking situations for the 50th percentile driver was found in driver simulator tests.

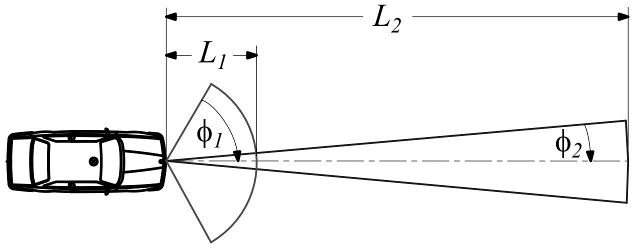


Figure 7.9.: Radar-based environment recognising system for Predictive Brake Assist [ETRH09]
Objects are detected within the depicted areas:
Low range Radar: aperture angle $\phi_1 = \pm 60°$, range $L1 = 30\ m$
Long range Radar: aperture angle $\phi_2 = \pm 5°$, range $L2 = 150\ m$

Table 7.2.: PBA intervention strategies in RCS-TUG

Abb.	Intervention	Driver reaction	Remark
PBA-A-a	Daimler	Driver reaction after warning	Equivalent to autonomous braking
PBA-A-b	Daimler	Driver reaction after braking	
PBA-A-c	Daimler	No driver reaction	
PBA-B-a	Audi	Driver reaction after HMI warning	Equivalent to intervention PBA-A-a
PBA-B-b	Audi	Driver reaction after braking	
PBA-B-c	Audi	No driver reaction	

- The ERS operates without system or environmental based faults.

Intervention strategy
To evaluate different systems two different intervention strategies were evaluated, see also Tab. 7.2:

– **System A** (comparable to Daimler PreSafe Brake [BBD+09]): 2.6 s before the anticipated collision an HMI warning occurs. On non-reaction, a braking sequence with about 50% of the braking potential is initiated 1 s later.

– **System B** (comparable to Audi Brake-Guard [MG07]): Instead of the autonomous braking of system A, a short abrupt braking sequence with a 5 kph velocity reduction is applied. The background of this intervention strategy is to safely attract the driver to the emergency braking without risking a false braking.

- **Idealised Evasive Manoeuvre Assistent (EMA)**
EMA systems add an evasive driving manoeuvre in cases where an emergency

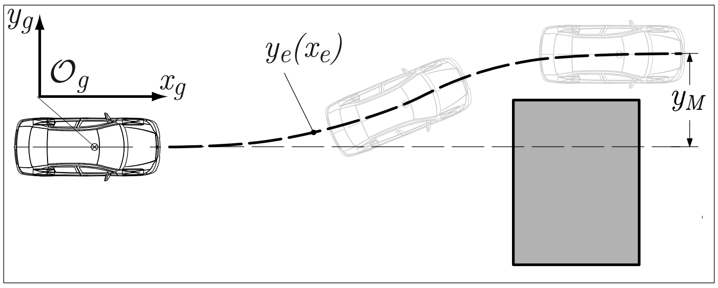


Figure 7.10.: Evasive path in PRORETA [IBB+09]

braking does not avoid the collision. This means that the steering control has to be taken over by the vehicle. Evasive manoeuvres require that the vehicle reliably detects whether sufficient and safe space is available or not. EMA have not been introduced to the market yet due to product liability and legal restrictions. A pilot study was done in the project PRORETA, which also formed the basis for the intervention strategy in the RCS-TUG study [IBB+09]. It has to be noted that in the RCS-TUG study the required space for the evasive manoeuvre was taken into account. Whenever a collision could not be prevented by a PBA system and space for the evasive manoeuvre was availabke, an evasive path was planned and added to the numerical simulation. In the simulation, the vehicle tries to follow this path. Fig. 7.10 illustrates the evasive path as given by

$$y_e = \frac{y_M}{1 + e^{-a(x_e - c)}} \ , \tag{7.16}$$

see [IBB+09]. Symbols x_e and y_e describe the ego-vehicle position in the global coordinate system $\{\mathcal{O}_g, x_g, y_g\}$ with the origin $\mathcal{O}_g$ at the vehicle CoG at the beginning of the EMA manoeuvre ($t = t_0$). The desired lateral distance compared to a non-steered manoeuvre is y_M. Parameters a and c are calculated from maximum lateral acceleration, actuator dynamics and the allowed lateral jerk [ISS08].

7.3.2. Results of RCS-TUG study

In total, 217 cases were reconstructed, selected from the 514 cases of the ZEDATU database according to the criteria of [Roh09]. Fig. 7.11 shows the representativeness of the RCS-TUG sub-sample. It can be seen that the RCS-TUG sample is proportionally lower[11] in single-vehicle accidents when compared with the Austrian national statistics of 2003 [HPK04] and the ZEDATU database. This is explained by the fact that single vehicle accidents are frequently not investigated in detail by legal authorities due to the lack of legal actions. Consequently, accidents with pedestrians and in oncoming traffic are proportionally higher in the sub-sample. For this reason, weighting factors, see Tab.

[11] Approximately half of the expected number.

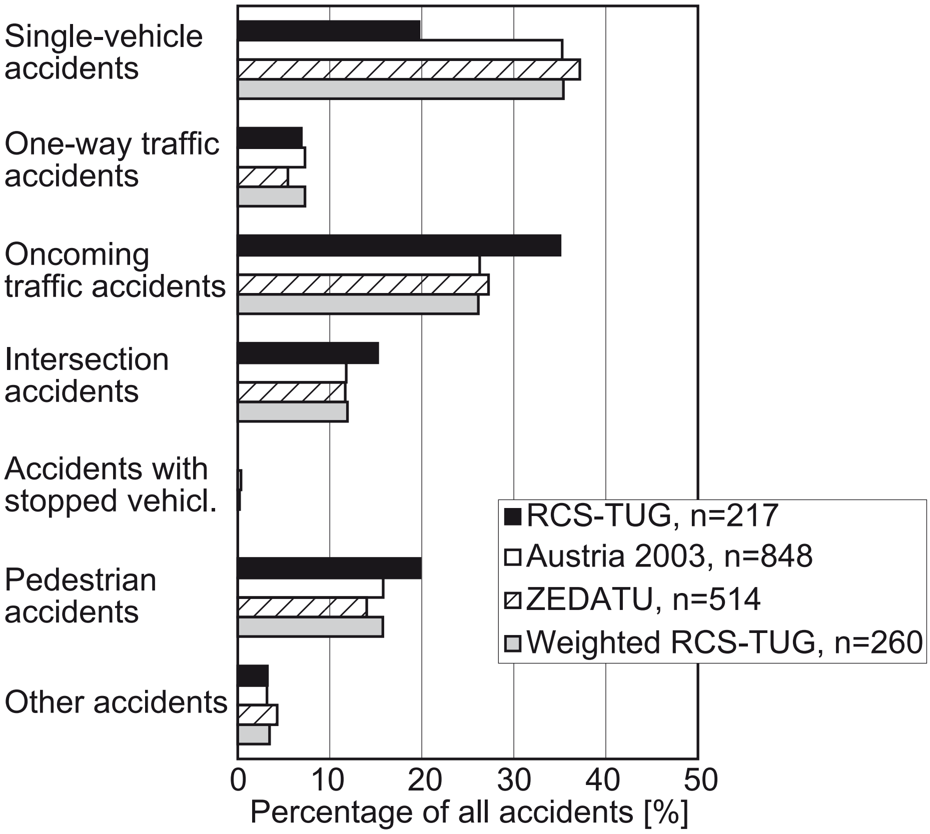


Figure 7.11.: Representativeness of RCS-TUG sample
Depicts the number of cases in different accident types according to [HPK04]. Compares cases from the RCS-TUG sub-sample of the ZEDATU database, the national statistics (full sample), ZEDATU database and the corrected RCS-TUG sub-sample using the weighting factors from Tab. 7.3.

7.3, were introduced. Single accidents were weighted first, and then other accident type numbers were adjusted to meet the fractions of the national statistics.

The next step was to evaluate the potential benefits of safety systems. In total, 43 systems were investigated. In principle, two groups were analysed for this evaluation: systems that can be *simulated* and systems where the safety potential is (subjectively) *estimated* by evaluation of the pre-crash simulation. The reason for this approach is the driver in the loop, which can react independently. Therefore, only systems where no driver reaction is required or where the driver reaction can be assessed were simulated.

- **Simulated safety systems**
  Anti-Lock Braking System (ABS), Electronic Stability Control (ESC) and automatic Evasive Manoeuvre Assistant (EMA) were taken into account. Additionally Predictive Braking Assist systems (PBA) with different intervention strategies and driver reactions were considered, Tab. A.1. Their intervention strategy

Table 7.3.: Number of cases in RCS-TUG

Accident type	**RCS-TUG** n=217	**Weighting Factor**	**Weighted** n=260
Single-vehicle accidents	43	2.14	92
One-way traffic accidents	15	1.27	19
Oncoming traffic accidents	76	0.89	68
Intersection accidents	33	0.94	31
Accidents with stopped vehicles	0	1.00	0
Pedestrian accidents	43	0.95	41
Other accidents	7	1.29	9
Total	**217**	**1.21**	**260**

was implemented in the accident reconstruction software, details can be found in [ET08, ETRH09, Roh09].

- **Estimated safety systems**
  Tab. A.3 in the appendix provides a list of the 39 safety systems, for which the safety potential was estimated, and detailed descriptions and the method for evaluating the safety potential of each system can be found in [Roh09].

The *simulated* safety systems were implemented in the numerical simulation from the starting point (transition from normal to critical driving situation) in each of the 217 selected cases. Whenever the accident was prevented by adding the safety system S to the simulation, the case was rated "avoided". The percentage of avoided cases A_S reads

$$A_S = \frac{n_{A,S}}{n_{ev,S}} \cdot 100[\%] \;, \tag{7.17}$$

where $n_{A,S}$ is the number of avoided accidents by safety system S and $n_{ev,S}$ the number of evaluated cases of safety system S. When the collision severity was significantly reduced by at least 10% of the vehicle velocity change Δv, the case was rated "potential"[12]. The percentage of avoided cases P_S reads

$$P_S = \frac{n_{P,S}}{n_{ev,S}} \cdot 100[\%] \;, \tag{7.18}$$

where $n_{P,S}$ is the number of significantly mitigated accidents by safety system S and $n_{ev,S}$ the number of evaluated cases of safety system S.

The same procedure was used for the *estimated* safety systems, but based on subjective evaluation of the pre-collision phase. Again, "avoidance" A_S and "potential" P_S were rated separately. In case of the estimated systems, A_S was rated when the investigator could state that the safety system S would have prevented the fatal accident, and P_S

[12] Potential to prevent the fatal injury.

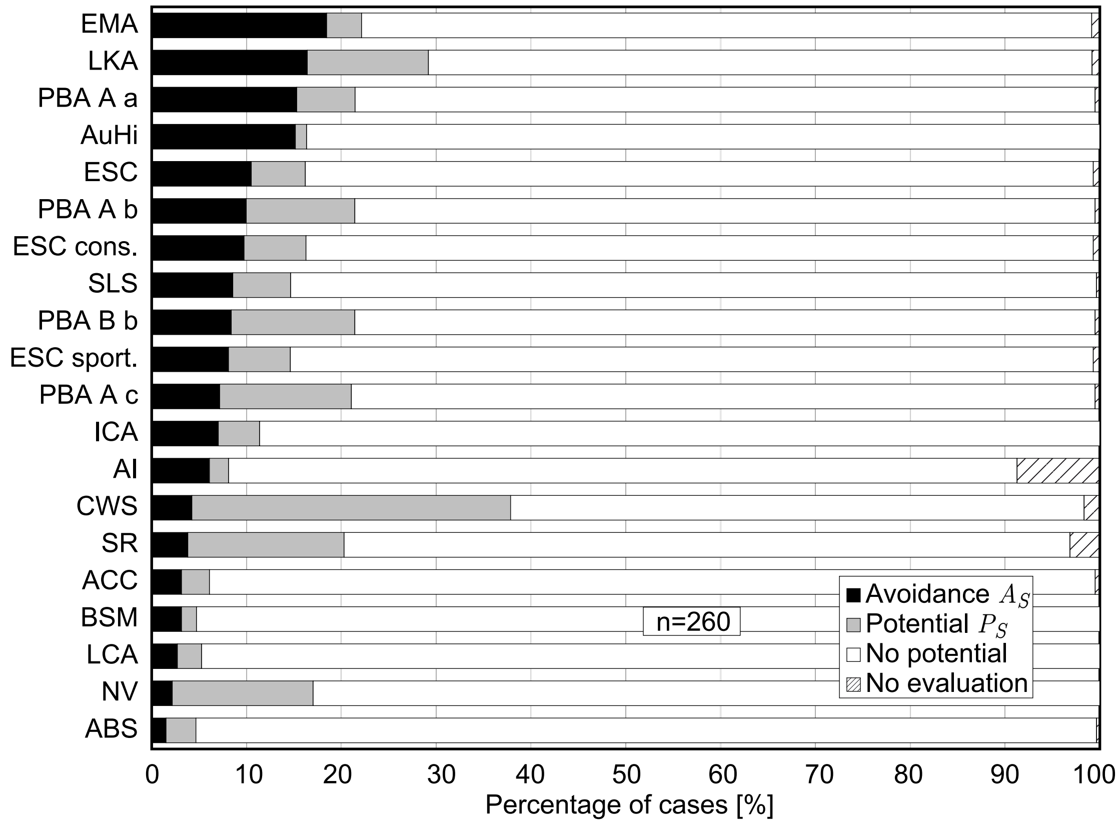


Figure 7.12.: Safety potentials of safety systems in weighted RCS-TUG, ranked by avoidance A_S of fatal accidents

was rated when the safety system had the potential to avoid the accident, depending on driver reaction. As an illustrative example, an accident which had been undoubtedly caused by an alcohol-impaired driver was rated "avoided", A_{AI}, for an Alcohol Interlock (AI); an accident where a CWS could have prevented the accident by an appropriate driver reaction was rated "potential", P_{CWS}. Figs. 7.12 and 7.13 depict the evaluated safety benefit of the 20 best systems investigated, and the complete result can be found in Tab. A.4 in the appendix. Additionally, the results for all systems in the RCS-TUG and the weighted RCS-TUG sub-sample are depicted, again separately for "avoidance" A_S and "potential" P_S, see Figs. A.1, A.2, A.3, A.4 in the appendix.

7.3.3. Discussion of RCS-TUG study

The following section discusses the results of the RCS-TUG study and compares them to investigations taken from the literature. Major literature reviews on the safety benefit of traffic safety systems were reported by [Org03, eSa05, BFRY07, LLP03], and the following section augments their findings with further studies. Tab. 7.4 lists the ten most

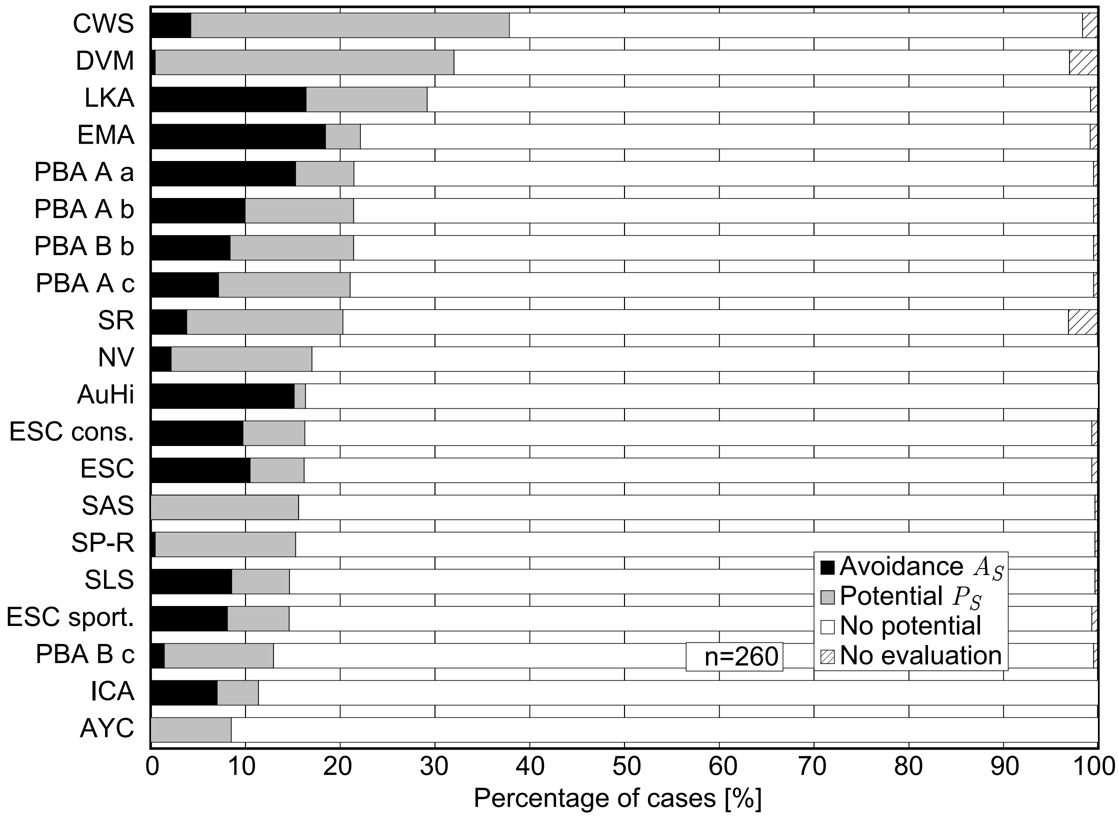


Figure 7.13.: Safety potentials of safety systems in weighted RCS-TUG, ranked by overall potential for prevention of fatalities $A_S + P_S$

effective systems based on the results of the RCS-TUG study. The following section discusses these systems in detail. The (subjectively) chosen selection criteria were:

Avoidance A_S *of fatal accidents* $> 5\%$
or
Overall potential to prevent fatalities $A_S + P_S > 20\%$

7.3.3.1. Evasive Manoeuvre Assistant (EMA)

Excluding *autonomous driving*[13], the most promising system with respect to collision avoidance is an automatic Evasive Manoeuvre Assistant (EMA). In 18% of all cases, EMA prevents the accident, and in 4% of all cases it reduces collision severity. Few studies on the benefit of EMA systems were found in the literature, Tab. 7.5. In [SS05] an approach for investigating the benefit of a combined emergency steering and

[13]Autonomous driving is a controversial topic and is not treated in this thesis. In the RCS-TUG study, autonomous driving was considered for only *one* involved vehicle, hence the potential is not 100%.

Table 7.4.: List of most promising safety systems according to RCS-TUG study

Abbr.	**Name of safety system**	**Avoidance of collision A_S [%]**	**Overall potential for fatality prevention $A_S + P_S$ [%]**
EMA	Evasive Manoeuvre Assistant	18.4	22.1
LKA	Lane Keeping Assist	16.4	29.2
PBA	Predictive Brake Assist (intervention A and B, all driver reactions)	3.8 to 15.3	13.0 to 21.5
AuHi	Automated Highway	15.2	16.3
ESC	Electronic Stability Control	8.1 to 10.5	14.6 to 16.2
SLS	Speed Limiting System	8.6	14.7
ICA	Intersection Collision Avoidance	7.0	11.4
AI	Alcohol Detection and Interlock	6.1	8.1
CWS	Collision Warning System	4.2	37.9
DVM	Driver Vigilance Monitoring	0.5	32.0

braking was shown. Fifty rear-end crashes taken randomly from the NASS/CDS accident database were studied with simulation-based accident reconstruction. The cases were weighted for occurrence in the complete database, and a PBA system was modelled. Instead of an automated steering/braking intervention, a driver model was developed. Based on driver simulator results the driver reaction was studied. Reaction times ranged from 0.32 s to 1.64 s. The 33rd (0.52 s), 50th (0.82 s), and 67th (1.10 s) percentile values were selected for the driver model. The driver model contained braking, steering, and braking in combination with steering. Emergency steering and braking sequences were modelled. Stochastic simulations with different driver behaviour were conducted, and the effectiveness of the system was calculated with probabilistic methods. Additionally, Δv reductions were compared with injury risk functions to assess the reduction of fatalities. A 38% reduction of fatal rear-end accidents resulting in a 40% fatality reduction were reported.

7.3.3.2. Lane Keeping Assist (LKA)

LKA systems have been introduced to the market both in commercial and passenger vehicles, e.g. the 2008 Volkswagen Passat CC. When evaluating the benefit of LKA, it was assumed in the RCS-TUG study that unintended leaving of the driving lane is successfully avoided. In reality, this depends on driver reaction and on the functionality of the LKA system, especially the lane recognition of video-based environment recognising systems. Today, the quality of lane recognition has not achieved this assumption, especially in bad weather conditions or where lane markings are missing or unclear. Successful

Table 7.5.: Reported benefit of EMA

Ref.	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	18.4% to 22.1%	Fatal accidents	ZEDATU, n=260
[SS05]	In-depth accident analysis	40% reduction of fatalities 38% avoidance of accidents	Fatal rear-end accidents Rear-end accidents	USA, NASS/CDS 2000-2001, n=50 USA, NASS/CDS 2000-2001, n=50

Table 7.6.: Reported benefit of LDW and LKA, [BFRY07]

Ref.	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	16.4% to 29.2%	Fatal accidents	ZEDATU, n=260
[LLP03]	Estimation	40% 2%	Fatal off-path accidents All accidents	Sweden
[ROGT01]	Estimation	30% [14] 1%	Fatal accidents All accidents	
[MIN+08]	Estimation	15.2% 8.9%	Injury accidents Fatal accidents	Europe Europe
[PS96]	Estimation	1%	Accidents on motorways	

lane recognition is often limited to highways and good weather conditions. Therefore, the result in RCS-TUG represents the optimal benefit of errorless LKA systems.

In the weighted RCS-TUG sample, LKA systems have the best potential for collision avoidance (16%) after the Evasive Manoeuvre Assistant. When considering the overall potential $A_S + P_S$ (29%), it is even the best system that intervenes automatically. Tab. 7.6 compares these results with other investigations on the benefits of LDW and LKA systems [BFRY07]. Most of the studies are estimations based on accident statistics with a low level of detail. For comparing the results of the RCS-TUG study, only [ROGT01] and [MIN+08] are useful, since they are related to (all) fatal accidents of their database samples. The results vary between 9 and 30%, as compared to 16% to 29% in RCS-TUG. Therefore, the findings of the RCS-TUG study as well as those available in the literature suggest that the benefit of autonomous LKA systems is currently underestimated.

7.3.3.3. Predictive Brake Assist (PBA)

Not surprisingly, in the RCS-TUG-Study the benefit of PBA depends on the driver reaction and the system used, Fig. 7.14. When the driver reacts with full braking and

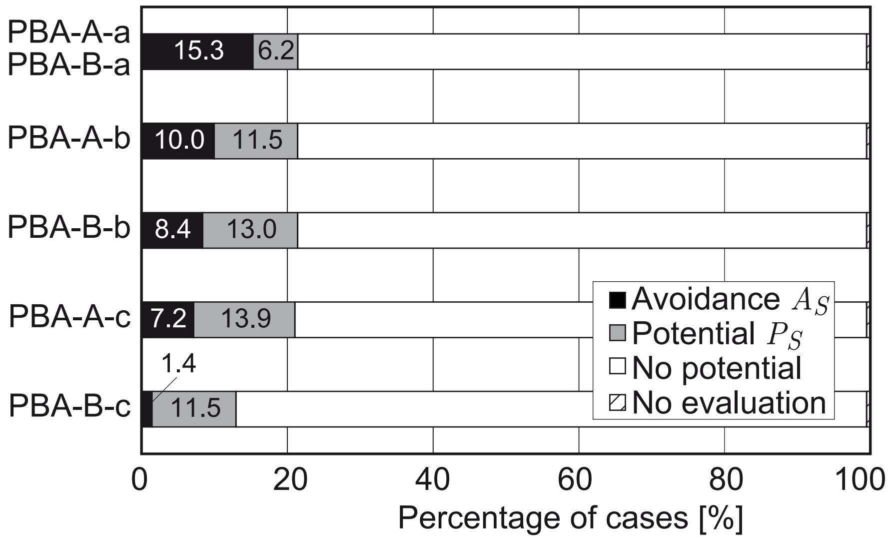


Figure 7.14.: Benefit of PBA (weighted RCS-TUG study, n=260)

a reaction time of 0.8 s to the system warning (Intervention "PBA-A-a"), 15.3% A_S of fatal accidents can be avoided. A further potential of 6.2% P_S exists for lowering the collision severity. PBA-A-a is comparable to an autonomous braking with full brake forces starting at a TTC of 1.8 s. PBA systems show less efficiency with later driver reactions (b and c) as well as with a reduced intervention severity "B" (brake jerk) instead of a semi-automatic brake assist with 50% brake pressure, "A".

Tab 7.7 shows the results of PBA benefits from literature. In [PFBC05] a reduction of fatalities in the range of 6.5% to 9% was estimated. However, these numbers are related to a conventional Brake Assist (BA)[15] and to a PBA that initiates autonomous braking.

In [BBZ06] a comparable method to the RCS-TUG study to assess the potential of BA and PBA was developed. The investigation is based on reconstruction of the pre-collision phase using data from 4,500 cases of the German In-Depth Accident database (GIDAS). A virtual prototype of the ADAS (BA and PBA) was used to calculate the difference in collision velocity, followed by a statistical method to assess the injury risk for different road users. The study did not differentiate between different PBA intervention strategies or the reaction of the driver. The numbers between the RCS-TUG study and [BBZ06] are not fully comparable, since RCS-TUG did not calculate the risk for injury based on the collision severity, and GIDAS also includes injury accidents.

In summary, the RCS-TUG study suggests that a detailed analysis shows a greater po-

[15]Conventional Brake Assist supports the driver by applying full braking pressure when an emergency braking is detected by the gradient of brake pedal activation.

Table 7.7.: Reported benefit of Predictive Brake Assist (PBA)

Ref.	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	16.4% to 29.2%	Fatal accidents	ZEDATU, n=260
[PFBC05]	Accident investigation	6.5 to 9% for BA	Fatalities in traffic accidents	French National injury accident census, n=917
[BFRY07, LLP03]	Estimation	20% for PBA 9% for PBA	Fatal multi vehicle accidents Fatal off-path accidents	Sweden
[BBZ06, Bus05]	Accident investigation	3% for BA 6% for PBA	Fatal accidents	GIDAS, n=4500
[SS05]	Accident investigation	44% for PBA	Fatal rear-end accidents	NASS 2000-2001, n=50

tential benefit than previously reported in the literature. However, a perfectly operating PBA system was anticipated, see section 7.3.1.3.

7.3.3.4. Automated Highway (AuHi)

Generally speaking, a technical system operating in well-defined system boundaries is better suited to take over human tasks than in a complex situation. Reduction in the number and complexity of situations in the system human-vehicle-environment can be especially achieved by autonomous driving on highways [Kom08]. ACC and LKA systems are already able to take over longitudinal and lateral vehicle guidance within certain system boundaries related to highway traffic.

Automated highway traffic has been a topic of intensive ongoing research. The technologic feasibility has already been demonstrated in the Californian PATH project [TRZ98], EC projects CHAFFEUR 1 and 2 [FGSB04, Fri99], the German KONVOI project [FHTP06] and the ongoing EC project HAVE-IT [Eur09d]. Benefits of Automated Highway Systems with respect to cost savings for improved traffic flow and decreased fuel consumption are reported in [Ioa97, Ben91]. For improvements in traffic safety, [CGS98] includes an estimation of 16% reduction of rear-end collisions on highways. The analysis is based on a comparison of manual and automated driving in a hard-braking emergency (rear-end collisions only). Frequency and collision severity were taken from accident statistics. The RCS-TUG study found the total potential to prevent fatal accidents at 16.3%, which is surprisingly high. It has to be taken into account that a failure-free automated system was anticipated on *all roads* with separated driving lanes, and not only on highways.

Table 7.8.: Reported benefit of Automated Highway Systems (AuHi)

Ref.	**Method**	**Result**	**Accident sample**	**Database**
Weighted RCS-TUG	In-depth accident analysis	16.3%	Fatal accidents	ZEDATU, n=260
[CGS98]	Estimation	16%	Rear-end collisions on highways	US accident statistics

7.3.3.5. Electronic Stability Control (ESC)

In the RCS-TUG study, three different ESC interventions were analysed. The potential for avoidance and collision mitigation is approximately the same for a conservative and a standard intervention, with 11% and 7% respectively. The sportive ESC tends to provide less potential, with 8% and 7%. Therefore, a total potential between 5% and 18% was found in RCS-TUG.

In the literature, several studies have investigated the potential of ESC. Tab. 7.9 lists important investigations. Further studies can be found e.g. in [BFRY07]. The investigations differ in methods, selection of database samples, countries and evaluation criteria. Most investigations concentrate on the reduction of single-vehicle accidents, where the major safety benefit of ESC is to be expected.

A comprehensive study was done in [KL06], which concluded that a complete fitment of ESC in the German car fleet would reduce the number of accidents with injuries by about 7 to 11%. The reduction in fatalities (car occupants) was estimated at 15 to 20%. In a 2004 study, IIHS [Ins06] concluded that ESC has the potential to reduce the fatality risk in 43% of all fatal crashes in the US, which is the highest reported ESC potential. In the US, a high portion of fatal rollover accident [Gug06] is observed. The high proportion of SUVs with increased risk for rollover in the car fleet is a possible reason for this at which ESC fitment is especially beneficial. In [Ins06] a reduction in risk for single-vehicle accidents of 33% for cars and 49% for SUVs was reported. Similar findings for higher ESC benefits in SUVs are reported in [Fer07].

Table 7.9.: Reported benefit of Electronic Stability Control (ESC)

Ref.	**Method**	**Result**	**Accident sample**	**Database**
Weighted RCS-TUG	In-depth accident analysis	8 to 18%	Fatal accidents	ZEDATU, n=260
[KL06]	Literature Review	7 to 11%	All accidents	Several

Continued on next page

Table 7.9 – continued from previous page

Ref.	Method	Result	Accident sample	Database
		15 to 20%	Fatal accidents	
[SN08]	Statistical	27% 68%	Single-vehicle accidents (cars) Single-vehicle accidents (SUVs)	Australia
[Fer07]	Literature review	30% to 50% 50% to 70%	Fatal single-vehicle accidents (cars and SUVs) Fatal single-vehicle accidents (SUVs)	Several
[Fer07]	Literature review	70% to 90% 17% to 38%	Fatal Rollover vehicle accidents (cars and SUVs) Multiple-vehicle accidents (cars and SUVs)	Several
[KSL05]	Statistical (odds ratios method)	32.4% to 54% 55.5% to 77.9%	ESC sensitive cases Fatal ESC sensitive cases	German Federal Statistical Office 1998-2002
[LTKK06, LTKK05]	Statistical (induced exposure methods)	16.7% +/- 9.3%. 21.6% +/- 12.8%	All accidents except rear-end coll. All accidents except rear-end coll.	Sweden 1998-2004
[Ins06]	Statistical	11.2%	Multiple-vehicle frontal accidents	USA state files from 1998-2002
[Bah05]	Statistical	52.6%	Single-vehicle frontal accidents	USA state files from 1998-2002
[SPCF01]	Statistical	18% 34%	Injury Accidents Fatal Accidents	EACS, n=1,647
[Far04, Far06]	Statistical	7% 34%	All accidents Fatal accidents	US police-reported crashes
[Tho06]	Statistical (odds ratios method)	3%	All accidents	National accident statistics of Great Britain, n=8951

Continued on next page

Table 7.9 – continued from previous page

Ref.	Method	Result	Accident sample	Database
		15%	Fatal accidents	
[Ins06]	Statistical	43%	Fatal accidents	US
[Ins06]	Statistical	32%	Fatal multiple-vehicle accidents	US
		56%	Fatal single-vehicle accidents	
[Dan04]	Statistical	35%	Single-vehicle accidents (cars)	US State and FARS database
		67%	Single-vehicle accidents (SUVs)	
End of table				

In summary, the results of the RCS-TUG study are confirmed and predict a decrease of about 15-20% in fatal accidents in Europe. In fatal accidents, the intervention strategy of ESC systems (sportive to conservative) has only a small influence. Nevertheless, this might be different when looking at all accidents, including accidents with injuries.

7.3.3.6. Speed Limiting System (SLS)

Excessive speed is the second main human error leading to traffic accidents (15% according to [Sta08]). The RCS-TUG study found an avoidance of fatal accidents by $A_S = 8.6\%$ and a overall potential of $A_S + P_S = 14.7\%$ by restricting the speed to the legal limit, which is in accordance with these results. SLS can have different operating principles. The speed limit can be assessed by vehicle based systems such as digital maps and navigation system or by road infrastructure via V2I communication. Another option is area-wide speed control. The speed limit can be reported to the driver or the vehicle can automatically reduce the speed. In the RCS-TUG study the technological functionality was not considered.

A cost-benefit analysis was done in [CT05], which reported a reduction of 37% of fatal accidents for SLS systems. These numbers were derived by applying empirical methods for accident risk based on driving velocity to UK accident data. The study did not investigate individual accidents. Results from field tests with speed observation report reductions of injuries and fatalities between 20 and 50% in selected areas, Tab. 7.10. In summary, the 15% reduction observed by RCS-TUG study is on the lower limit of studies from the literature. Since it is based on actual accident reconstruction, and not on statistical methods or field tests, it should provide more reliable data for benefit assessment.

Table 7.10.: Reported benefit of Speed Limiting Systems (SLS)

Ref.	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	8.6% to 14.7%	Fatal accidents	ZEDATU, n=260
[Elv97]	Field test	20%	Injury accidents	Norway field test
[GHSR04]	Field test	33% 40%	Injury accidents in observed areas Fatalities in observed areas	UK
[AL05]	Field test	25% 50%	Injury accidents in observed areas Fatal accident in observed areas	Sweden, 340 km of observed road
[CT05]	Estimation	20% 37%	Injury accidents Fatal accidents	UK injury accident data

Table 7.11.: Reported benefit of Intersection Collision Avoidance (ICA)

Ref.	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	7.0% to 11.4%	Fatal accidents	ZEDATU, n=260
[Org03] [Org03]	Estimation Estimation	50% 46%	Intersection accidents Intersection accidents	USA Japan
[BFRY07]	Literature review	29%	All accidents	

7.3.3.7. Intersection Collision Avoidance (ICA)

According to [Tom07], 11.8% of fatal accidents in Austria in 2003 were "intersection" accidents. In the RCS-TUG study, ICA systems were predicted to avoid fatal accidents by $A_S = 7\%$, with an overall potential of $A_S + P_S = 11.4\%$. [Org03] found that 50% of intersection crashes in the US and 46% in Japan could be avoided. The German INVENT project suggested 29% accident avoidance (cited in [BFRY07]). Further reliable data is rare because of the variety of possible solutions of ICA systems, which can either be vehicle based, infrastructure only or co-operative between infrastructure and vehicle, Tab 7.11. The RCS-TUG study did not distinguish between different ICA solutions.

7.3.3.8. Alcohol Interlock (AI)

The prevention of traffic accidents caused by alcohol-impaired drivers with automatic alcohol interlocks has been a field of research for years [BM04]. Field tests and interlock programs have been carried out mainly in the US, but also in Sweden, Canada and other countries. Manipulation and tampering of the systems have been reported as a major

Table 7.12.: Reported benefit of Alcohol Interlock Systems (AI)

Ref.	**Method**	**Result**	**Accident sample**	**Database**
Weighted RCS-TUG	In-depth accident analysis	6.1% to 8.1%	Fatal accidents	ZEDATU, n=260
[eSa05]	Estimation	1.1% 6.3%	All accidents, 70% fleet penetration and 25% effectiveness of the system All accidents	Europe
[LLP03]	Estimation	18% 1% to 5%	Alcohol related fatal accidents Fatal accidents	Sweden
[ROGT01]	Estimation	9% Reduction of collision severity	All accidents	Several

topic for their effectiveness.

The weighted RCS-TUG study calculates an avoidance of fatal accidents of $A_S = 6.1\%$ with an overall potential of $A_S + P_S = 8.1\%$. The study evaluated cases with proven accident causation by alcohol abuse (6.1%) and cases were causation by alcohol was likely (2.0%). A reliable functionality of the system, i.e. not manipulated by the driver, was assumed. In [eSa05] a 6.3% reduction in all accidents with a reliable system and full market penetration was predicted. An avoidance of 1.1% in all accidents was estimated, assuming 70% fleet penetration and 25% reliability of the system function (technical tolerance and bypassing). Results reported in literature, Tab. 7.12, correspond well with the findings of the RCS-TUG study.

7.3.3.9. Collision Warning System (CWS)

In contrast to Predictive Brake Assist (PBA) with (semi-)autonomous emergency braking intervention, collision warning systems provide acoustic, optic or haptic feedback to the driver in case of anticipated collisions. Often, CWS are limited to Forward Collision Warning Systems (FCWS), because of the availability of environment recognising systems from ACC and the relative simplicity of the collision prediction algorithms in the longitudinal direction, see chapter 8.3.2.

With $A_S + P_S = 37.9\%$, the RCS-TUG study shows the highest total potential for mitigation of fatal traffic accidents in Austria for a vehicle-based safety system, when correct system function is assumed. In contrast, avoidance A_S of fatal accidents was only rated at 4.2%. This result shows the main disadvantage of CWS, which only assists

Table 7.13.: Reported benefit of Collision Warning System (CWS)

Ref-	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	4.2% to 37.9%	Fatal accidents	ZEDATU, n=260
[KSAT05]	Estimation	57%	Rear-end accidents	Several
[Nat97]	Estimation	50%	Rear-end accidents	USA, police-reported accidents

and does not intervene. The warning strategy is a key issue for the benefit of CWS [JLC08]. Different drivers prefer warnings at different moments prior to collisions, and false alarms and alarms which are annoying to the driver could lead to deactivation of the system. In [ZM03] three uncertainties in FCWS are mentioned:

- Uncertainty in state estimation of objects by sensor errors,
- Driver reaction time (time between warning and driver reaction),
- Maximum deceleration depending on braking system, vehicle dynamics control and tyre-to-road grip.

According to findings in [ZM03], the importance of these uncertainties depends on the traffic scenario. In rear-end collisions involving stationary objects, the first two uncertainties dominate, while in close following in heavy traffic the accuracy of the sensor system becomes more important.
The findings in the literature, Tab. 7.13, regarding the benefit of CWS are rare and cannot be compared directly to the RCS-TUG study (Tab. 7.13). According to the RCS-TUG study, the potential for CWS is significant, but an efficient system would require a sophisticated warning strategy involving the preferences and the actual vigilance of the driver, as well as a reliable environment recognising system and suitable collision prediction algorithms.

7.3.3.10. Driver Vigilance Monitoring (DVM)

The literature estimates that about one third of all traffic accidents are related to driver inattention, Tab. 7.14. Driver inattention can be a result of fatigue or distraction due to other driving tasks (mobile phone operation, etc.) and can be the primary or secondary cause of the accident, along with other causation factors.

In [Ake00] a consensus status of an international expert group was compiled, which cited fatigue as the largest identifiable and preventable accident cause (15% to 20% of all accidents). In [YRH03] an extensive literature survey concerning driver distraction was conducted. Approximately 10% of all accidents were related to driver distraction. In the RCS-TUG study, DVM was defined as a definitely reliable and driver-accepted ADAS. When the driver is not able to fulfil his/her tasks, the HMI informs the driver

Table 7.14.: Reported benefit of Driver Vigilance Monitoring (DVM)

Ref.	Method	Result	Accident sample	Database
Weighted RCS-TUG	In-depth accident analysis	0.5% to 32%	Fatal accidents	ZEDATU, n=260
[eSa05]	Estimation	35% 2.9%	Fatigue-related accidents with 70% fleet penetration of DVM All accidents with 70% fleet penetration	
[BFRY07]	Literature review	4% 24%	Single-vehicle accidents Fatal vehicle accidents	
[RCF+99]	Estimation	10% to 15% 10%	Fatal and injury accidents on highways Fatal and injury accidents on rural roads	
[BFRY07]	Literature review	3%	Fatal accidents	Japan
[Ake00]	Consensus statement	15% to 20%	All accidents	
[Kul97]	Literature review	4% to 20%	All accidents	Several
[SRSR01]	Accident investigation	8.3%	All accidents	USA, Crashworthiness Data System 1995-1999
[Gor07]	Accident investigation	9.7%	All accidents	Police reported accidents New Zealand (CAS) 2002-2003, n=20,808

about his/her inattention and the possible risks, and in case of non-reaction, the vehicle is decelerated [Roh09]. Due to the unknown reaction of the driver, a safe avoidance of only 0.5% was assessed, whereas the total potential reported as about 32%.

7.3.4. Summary of potential analysis of traffic safety systems

The findings in the RCS-TUG study provide comprehensive information for the prioritisation of traffic safety systems. It is a unique study considering the direct comparison of safety systems on the same representative database. The assessment of potential benefit was based on in-depth accident reconstruction of the pre-collision phase.

In particular, the current state of the art seems to be underestimating the potential benefit of LKA and systems for automated driving on separated driving lanes. EMA systems show the highest potential, but they are also the most visionary and complex system in terms of market introduction.

One important assumption of the RCS-TUG study is the intentional consideration of perfectly operating ADAS, which provides data of their maximum safety potential. Relativising factors that take into account technological limitations and vehicle fleet penetration can be applied any time. Also, the study did not consider possible risk compensation by a human driver relying on the capabilities of the technical system. Real safety benefits of ADAS will logically be below to these reported data. Future work will include assessment of combined systems (e.g. ABS+ESC+PBA+LKA) and a more detailed investigation of benefits in different types of vehicles and collisions.

The following section will concentrate on adaptive restraints. The benefit of this system could not be evaluated by the RCS-TUG study due to the high crash severity and the unknown occupant position at time of restraint system activation.

8. Control of adaptive restraint systems

In chapter 7 the benefit potentials of different traffic safety systems working independently have been investigated. This chapter which build one focal point of the present thesis, takes the next step by introducing an approach for integrating functionalities of some specific safety systems in order to optimise the global goal, which is to minimise the risk of injury. The described application is an improved functionality of an adaptive frontal impact restraint system with respect to the actual collision scenario and the occupant.

This is done by development of a control module called “Integrated Safety Controller” (ISC), in which the force levels and activation times of an adaptive restraint system are predefined based on information on the oncoming collision and the occupant. Reference values for these force levels are generated in order to minimise the average acceleration of the occupants. The method takes into account the actual crash severity using a forecast of the acceleration behaviour of the passenger cell. It is based on the prediction of collision speed, mass and stiffness of the opponent and ego-vehicles.

The development of the ISC is part of the author’s long-term project, Fig. 8.1. The project started with simulation-based development, “ISC passive offline”. In this context, passive means no active control of the vehicle dynamics with respect to collision avoidance; only the adaptive restraint system is taken into consideration. “Offline” means the model does not need real-time performance at this stage, although this requirement is kept in mind for later vehicle application. This stage is the status of the project at the

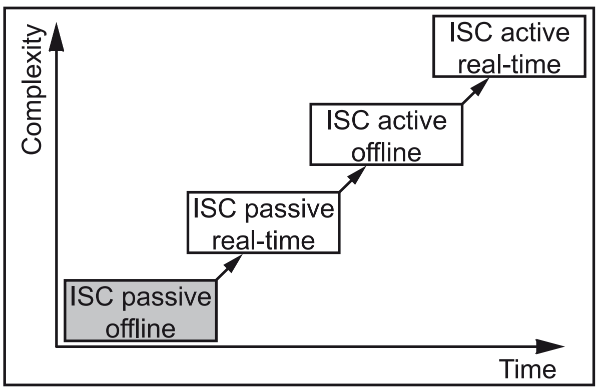


Figure 8.1.: Evolution of the ISC model during the project

time of the submission of the thesis, which is marked in grey in Fig. 8.1. The next project stage will be "ISC passive real-time", which will involve application in a demonstrating prototype hardware.

An additional goal will be the integration of the primary (active) safety systems for collision avoidance and mitigation, "ISC active". This will further improve overall performance with respect to injury risk and will allow to integrate the vehicle's safety systems by a common system architecture. A similar trend in vehicle dynamics control is currently seen in the development of "Global Chassis Control" systems, [Ros09]. With an increasing number of systems influencing vehicle dynamics[1], the intervention strategies of these individual systems might come into conflict without a higher control strategy. An additional advantage is that it makes several separate Electronic Control Units (ECU) obsolete, thereby reducing costs and improving package space and system weight.

The current status of the ISC was verified through off-line simulation. The following sections describe only this part of the project.

8.1. The Integrated Safety Controller ISC

The main idea of ISC is a pre-fired constant force restraint (CFR) system, which is adapted to the accident and the occupants of the ego-vehicle. Collision mitigation by primary safety systems can be incorporated by forecasting the collision parameters considering ADAS interventions. The approach is based on the prediction of the passenger cell acceleration pulse of the ego-vehicle and pre-setting[2] and pre-firing[3] of an adaptive occupant restraint system. This is based on a forecast of the collision speeds, masses and stiffness of the colliding vehicles, which are input parameters for the prediction of the passenger compartment acceleration ("pulse"). The predicted pulse, together with information about occupant weight and position, are the necessary input parameters for the adaptive restraint system.

After a literature review and patent search, this approach was patented in [WE09]. However, the search report from the patent office recently revealed a partial overlap with [Bac97]. Although the approach described in [Bac97] is similar to ISC, it does not cover the prediction of the acceleration pulses based on prior knowledge of collision severity and the crush characteristics of the involved vehicles derived from crash test or simulation data, see section 8.5. The level of detail of the methods patented in [Bac97] are not comparable to the presented ISC approach. Furthermore, it is important to recognise that the novelty of ISC is the holistic approach and the combination of several state-of-the-art methods with the "virtual deformation spring method", which is described below.

[1] Examples are Electronic Stability Control (ESC), Active Front Steering (AFS), Active Rear Steering, Torque Vectoring, Active Stabilisers, Active Body Control (ABC).

[2] Pre-setting in the context of this thesis is the setting of the restraint force level prior to impact.

[3] Pre-firing in the context of this thesis is the activation of the restraint system prior to impact.

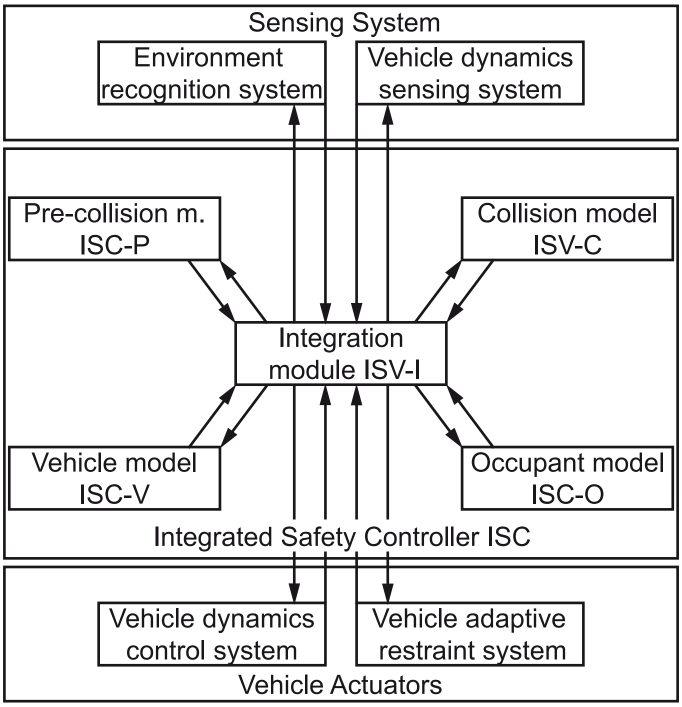


Figure 8.2.: System architecture of the Integrated Safety Controller ISC

Fig. 8.2 depicts the system architecture of ISC. It is organised in a modular structure and consists of five separate models:

- **Integration Module (ISC-I)**
- **Pre-collision Model (ISC-P)**
- **Vehicle Model (ISC-V)**
- **Collision Model (ISC-C)**
- **Occupant Model (ISC-O)**

Depicted in the centre, the Integration Module (ISC-I) exchanges input and output data with the other ISC modules, as well as with the sensing and the actuator system of the vehicle. It can be considered as the master module. In principle, the five separate modules of ISC could be integrated into a single one, but by using a predefined format for the exchanged I/O data, the modules could be developed separately and also exchanged for different full-vehicle applications. The open architecture allows for the integration of different solutions from other institutions and system suppliers.

ISC-I exchanges data with the vehicle sensing system and the vehicle actuator system. The sensing system consists of the environment recognising system (ERS) and the sensing system for vehicle dynamics. The actuator system consists of ADAS systems on course planning and stabilisation level (primarily ABS ,ESC, PBA, LKA). The modular structure of ISC-I allows for two options: interpret data only from the vehicle sensing system and calculate a suitable activation strategy for the adaptive restraint system ("passive model", see above), or to take into account directly the vehicle dynamics control ("active model"). Depending on the application, the ERS data can be interpreted by the ERS itself or by an additional module within the pre-collision model.

In principle, the development of ISC solely requires current state-of-the-art technology with respect to the hardware. Suitable ERS are available in series production or at least as prototypes, which is also true for the secondary vehicle safety systems[4]. Matlab/Simulink® software was chosen for the realisation of the complete model, due to its ability to generate real-time codes that can be run on automotive ECUs.

8.2. Control block chart of ISC

Fig. 8.3 depicts the control block chart of the complete traffic system, including the ISC approach. Typically, the scheme of the system has got input-output structure, where the vehicle is in the centre of the system. It starts with the planned course of the driver (reference course). Driving represents a closed-loop control, in which the dynamical state $\mathbf{x}_e(t)$ of the ego-vehicle is continuously fed back to the driver. The difference between reference and observed course is entered into the driver block. The ADAS (e.g. PBA, LKA) receive input from the "ERS (Env. recognising syst.)" block and the "Vehicle on-board sensing" block and adds an additional vehicle control input. Since the ISC approach does not necessarily require ADAS, the line is dashed. The vehicle responds with its dynamic state $\mathbf{x}_e(t)$ to the common control input of the driver and ADAS, $\mathbf{u}(t)$, and disturbances $\mathbf{v}(t)$ from the environment. The dynamic state ${}_e\mathbf{x}_V(t)$ of the vehicle is measured in the "Vehicle on-board sensing" block, with respect to the ego-vehicle fixed coordinate system denoted with e, and reads:

$$ {}_e\mathbf{x}_V(t) = [x_{V1}(t)\ x_{V2}(t)\ \dots\ x_{Vm}(t)\ \dots\ x_{VM}(t)]^T , \tag{8.1} $$

where m, $1 \leq m \leq M$, is an index of M measured signals $x_{Vm}(t)$ of the vehicle on-board measurement system. In the "Interpret. vehicle data, ISV-V" block, this continuous dynamic state ${}_e\mathbf{x}_V(t)$ is interpreted as an discrete $6 \times K$ state matrix ${}_g\mathbf{X}_{e,k}$,

$$ {}_g\mathbf{X}_{e,k} = [\mathbf{x}_{e1}\ \mathbf{x}_{e2}\ \dots\ \mathbf{x}_{ek}\ \dots\ \mathbf{x}_{eK}] , \tag{8.2} $$

where $\mathbf{x}_{ek}$ is the dynamic vehicle state at a certain discrete time denoted by index k, $1 \leq k \leq K$. Symbol K is the number of the stored states. Note that matrix ${}_g\mathbf{X}_{e,k}$ consists of dynamic states vectors ${}_g\mathbf{x}_{ek}$ with respect to the global coordinate system g which read:

$$ {}_g\mathbf{x}_{ek} = [x_{ek}\ y_{ek}\ \psi_{ek}\ v_{x,ek}\ v_{y,ek}\ \omega_{z,ek}]^T , \tag{8.3} $$

[4]Examples are adaptive belt load limiters, adaptive airbag inflators or ventholes.

where x_{ek}, y_{ek}, ψ_{ek} are position and $v_{x,ek}$, $v_{y,ek}$, $\omega_{z,ek}$ velocity data of the ego-vehicle. The duration of the recorded (past) time $T_{p,e}$ for the ego-vehicle reads:

$$T_{p,e} = \sum_{k=1}^{K} \Delta T_k \ , \tag{8.4}$$

where ΔT_k is the length of the considered time step. In many cases of measurement systems, ΔT_k is constant, $\Delta T_k = \Delta T$.

Next, the result of the vehicle on-board sensing system ${}_e\mathbf{x}_V(t)$[5] is fed back to the ADAS and inner vehicle dynamic control loop (e.g. ESC and ABS). This inner control loop ("Vehicle dynamics control" block) is dashed, since the vehicle can be equipped either with or without such systems.

The environment provides input to the "ERS (Env. recognising syst.)" block, in which involved objects are measured with appropriate ERS sensors, ${}_e\mathbf{x}_{E,i}(t)$. It includes the received signals of I objects, indexed by $i = 1, 2, ..., I$ and measured in the vehicle coordinate system e. Vectors ${}_e\mathbf{x}_{E,i}(t)$ read:

$${}_e\mathbf{x}_{E,i}(t) = \left[x_{i1}(t) \ x_{i2}(t) \ \ldots \ x_{ir}(t) \ \ldots \ x_{iR}(t)\right]^T \ , \tag{8.5}$$

where $x_{ir}(t)$ is one of R measurements, indexed by r, $1 \leq r \leq R$.

In the "Interpret. ERS data" block, the objects and their dynamic states are calculated. Objects are interpreted as single rigid bodies with a specific reference point. Similar to the ego-vehicle, the past states are stored in the $6 \times L$ matrix ${}_e\mathbf{X}_{i,l}$,

$${}_e\mathbf{X}_{i,l} = \left[\mathbf{x}_{i1} \ \mathbf{x}_{i2} \ \ldots \ \mathbf{x}_{il} \ \ldots \ \mathbf{x}_{iL}\right] \ , \tag{8.6}$$

where L denotes the number of stored states for a discrete time interval $l = 1, 2, ..., L$. The dynamic states ${}_e\mathbf{x}_{il}$ read

$${}_e\mathbf{x}_{il} = \left[x_{il} \ y_{il} \ \psi_{il} \ v_{x,il} \ v_{y,il} \ \omega_{z,il}\right]^T \ , \tag{8.7}$$

where x_{il}, y_{il}, ψ_{il} are position and $v_{x,il}$, $v_{y,il}$, $\omega_{z,il}$ velocity data of obstacle i. The duration of the stored states is $T_{p,i}$,

$$T_{p,i} = \sum_{l=1}^{L} \Delta T_l \ , \tag{8.8}$$

where, similar to the ego-vehicle, ΔT_l are discrete time-steps which are usually constant, $\Delta T_l = \Delta T$.

[5]Theoretically the already processed data ${}_e\mathbf{X}_{e,k}$ could be used also.

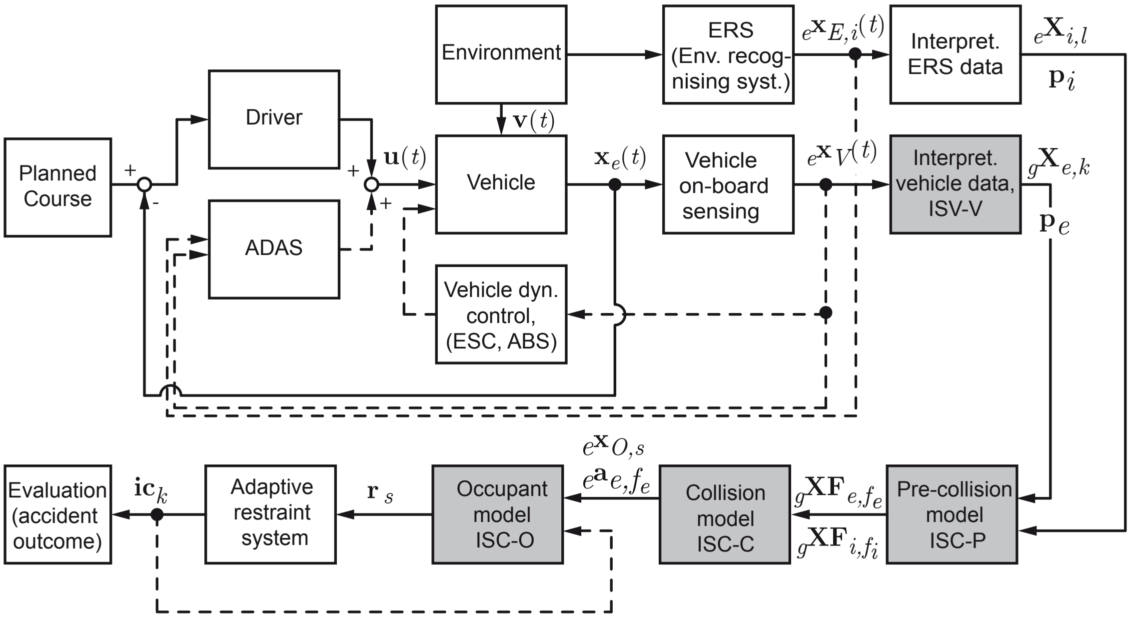


Figure 8.3.: Control block chart of the traffic system, including the ISC approach

The dynamic states of ego-vehicle and obstacles are complemented with the parameter vectors $\mathbf{p}_e$ for the ego-vehicle and $\mathbf{p}_i$ for the i-th obstacle. These vectors read:

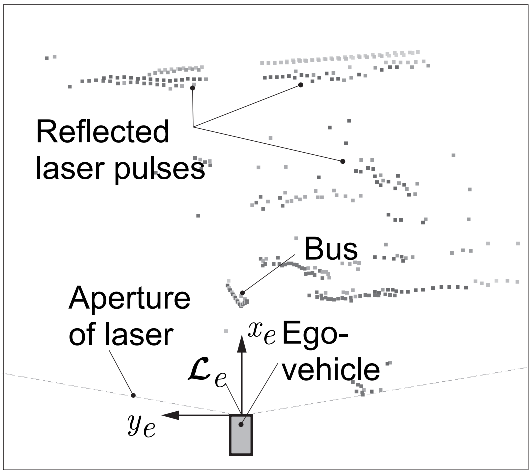
$$\mathbf{p}_{e(i)} = [l_{e(i)}\ w_{e(i)}\ m_{e(i)}\ c_{e(i)}]^T \ , \tag{8.9}$$

where parameter $l_{e(i)}$ is the vehicle length, $w_{e(i)}$ the vehicle width, $m_{e(i)}$ the vehicle mass and $c_{e(i)}$ an obstacle stiffness identifier of the objects. This data is needed for the pre-collision model ISC-P and collision model ISC-C. Parameter $c_{e(i)}$ describes the force deflection characteristics of the crush zone. The parameter vectors p_i are derived from ERS measurements[6].

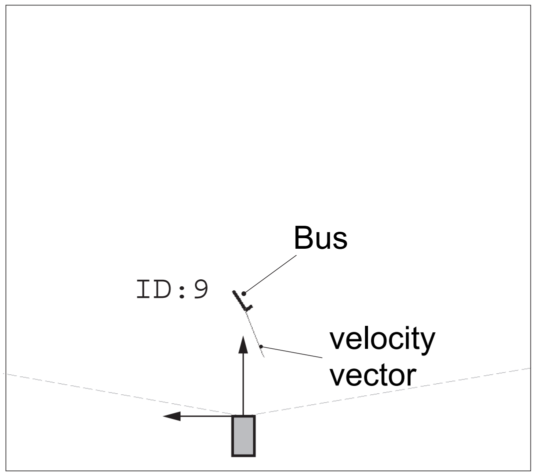
As an illustrative example for an ERS measurement, Fig. 8.4 shows the results produced by a laser-scanner ERS. The ego-vehicle passes by a stopped bus on the left side, Fig. 8.4(a). Fig. 8.4(b) depicts the reflected laser pulses in top view as dots indicating possible obstacles. Fig. 8.4(c) shows the result of the interpretation: the bus is classified as a *passenger vehicle* type of obstacle with the object identification number $i = 9$. The outline represents length l_9 and width w_9, as well as orientation and magnitude of the reference point velocity vector, $_e v_{x9}$, $_e v_{y9}$. The velocity is still defined in the ego-vehicle reference coordinate system e.

[6]Video-based determination of mass and stiffness has not yet been developed and is part of future work.

(a) Traffic situation



(b) Result of ERS scan (${}_e\mathbf{x}_{E,i}$)



(c) Interpretation of ERS data

Figure 8.4.: Work flow of the Environment Recognising System

(a) Traffic situation: a stopped bus is passed on the left side. (b) The laser scanner delivers reflected laser pulses. (c) Interpretation: the bus is recognised as passenger vehicle with ID $i = 9$; a relative velocity of the object is depicted in the vehicle fixed coordinate system $\{\mathcal{L}_e, x_e, y_e, z_e\}$ with $\mathcal{L}_e$ the laser scanner position.

In the "Pre-collision model ISC-P" the stored discrete dynamic states ${}_gX_{e,k}$ and ${}_eX_{i,l}$ are predicted into the future, ${}_gXF_{e,f_e}$ and ${}_eXF_{i,f_i}$, where $f_{e(i)}$, $0 \leq f_{e(i)} \leq F_{e(i)}$, is the index of the predicted states and $F_{e(i)}$ the number of predicted states. The prediction is carried out for certain time intervals $T_{f_{e(i)}}$, which read:

$$\begin{aligned} T_{f_e} &= \sum_{f_e=1}^{F_e} \Delta T_{f_e} \ , \\ T_{f_i} &= \sum_{f_i=1}^{F_i} \Delta T_{f_i} \ , \end{aligned} \tag{8.10}$$

where $\Delta T_{f_{e(i)}}$ are time steps for the collision prediction. First of all, matrices ${}_e\mathbf{X}_{i,l}$ of the I obstacles are transformed into the global coordinate system g, ${}_g\mathbf{X}_{i,l}$. Therefore, at the instant moment of time, $t = t_0$, the global coordinate system g is set to the ego-vehicle coordinate system e,

$$
\begin{aligned}
{}_g\mathbf{x}_{e,K} &= {}_g\ \mathbf{xf}_{e,0} \ , \\
{}_g\mathbf{x}_{i,L} &= {}_g\ \mathbf{xf}_{i,0} \ .
\end{aligned}
\tag{8.11}
$$

The prediction results in future state matrices ${}_g\mathbf{XF}_{e,f_e}$ and ${}_g\mathbf{XF}_{i,f_i}$. Next, a collision check between the ego-vehicle and the obstacles is performed. This check has to take into account that with increasing look-ahead time $T_{f_{e(i)}}$ the accuracy of collision detection will decrease.

In case of an unavoidable collision at the discrete time step $f = f_{coll}$, the predicted state matrices ${}_g\mathbf{xf}_{e,f_{coll}}$ and ${}_g\mathbf{xf}_{i,f_{coll}}$ are passed to the "Collision model ISC-C" block. This allows for the calculation of the main collision parameters: impact velocity, angle of impact and offset of vehicles. The "ISC-C" calculates the deceleration of the passenger compartment ${}_e\mathbf{a}_{e,f_e}$, see section 8.5, which is input to the "Occupant model ISC-O" block.

Vector ${}_e\mathbf{a}_{e,f_e}$, is complemented with occupant relevant information of S monitored occupants, ${}_e\mathbf{x}_{O,s}$, indexed s, which holds

$$
{}_e\mathbf{x}_{O,s} = [\ m_{O,s}\ p_{O,s}]^T \ . \tag{8.12}
$$

It consists of the mass of the s-th occupant $m_{O,s}$ and the relative position of an occupant reference point on the thorax with respect to the dashboard $p_{O,s}$, measured at the instant moment of time $t = t_0$. Vector ${}_e\mathbf{x}_{O,s}$ has to be assessed by appropriate onboard sensing systems, such as weight sensitive seat mats, video-based occupant monitoring and the amount of seat belt spooled out from the belt retractor.

The occupant model derives optimised input for the restraint system, $\mathbf{r}_s$,

$$
\mathbf{r}_s = [TTF_{SB,s}\ RF_{SB,s}\ TTF_{AB,s}\ RF_{AB,s}]^T \ , \tag{8.13}
$$

which consists of trigger time $TTF_{SB,AB}$ and restraint force $RF_{SB,AB}$ of seat belt SB and frontal airbag AB for S occupants. By using the collision prediction approach, values for TTF can be negative, which means pre-firing before first contact of ego-vehicle and obstacle, see section 8.6.

The final output is the occupant loading $\mathbf{ic}_s$,

$$
\mathbf{ic}_s = [a_{s,mean}\ a_{s,max}]^T \ , \tag{8.14}
$$

as a vector of injury criteria of the s-th occupant, which is a measure of the final evaluation - the accident outcome. In the current application, $\mathbf{ic}_s$ includes mean $a_{s,mean}$ and maximum acceleration $a_{s,max}$ of a single mass point occupant model. Finally, another feedback loop from the adaptive restraint block to the observer in the ISC-O is feasible. Hence, the occupant loading can be fed back to the occupant model to derive new values for the restraint force $RF_{SB,AB}$ in order to receive further reduction of the injury criteria. Since this is not implemented, the feedback loop is dashed.

8.3. Pre-collision model ISC-P

As mentioned above, the function of ISC-P is the calculation of predicted collision parameters. It receives the following information from the ISC-I:

- **Ego-vehicle**
  State matrix ${}_g\mathbf{X}_{e,k}$ of the ego-vehicle for a certain time interval $T_{p,e}$ as determined by ISC-V and parameter vector $\mathbf{p}_e$ consisting of length l_e, vehicle width w_e, mass m_e and stiffness c_e;

- **Environment information**[7]
  Examples are: road-tyre grip potential μ, longitudinal road inclination γ, wind forces ${}_eW_x$, ${}_eW_y$;

- **Obstacles**
  State matrix ${}_e\mathbf{X}_{i,l}$ of i obstacles for a certain time span $T_{p,i}$ as determined by the ERS.

ISC-P calculates the collision probability by predicting the ego-vehicle and obstacle vehicle trajectories. In the case of an unavoidable accident, the state vectors and additional information of the involved vehicles/obstacles are returned to the ISC-I.

The state-of-the-art in collision prediction and the known methodologies are discussed in the next section. Next, one of the collision prediction models used in the ISC-P are presented, and verification results from simulation data is given.

8.3.1. Collision prediction

Prediction of collisions is a major topic in ADAS, since intervention strategies are often based on forecasts of traffic situations. Collision prediction models range from the simple anticipation of stationary objects in front of the ego-vehicle up to complex models that incorporate driver reactions and environment models derived from environment recognising systems and digital maps. This section introduces theoretical approaches and models from the literature in order of increasing complexity.

[7] Not yet implemented

8.3.2. Collision prediction for FCWS/PBA (one DoF)

As described in chapter 7, Frontal Collision Warning Systems (FCWS) and Predictive Brake Assist (PBA) are among the most effective primary safety systems. In [Win09] three different strategies for the prevention of accidents related to longitudinal traffic flow are mentioned:

- **Preventive assisting systems**
  Latent dangerous situations can be handled by ADAS which increase the scope of possible driver action or improve the driver's abilities. In particular, ACC was mentioned, since field studies have shown that drivers using ACC use safer distances than in manual driving [WDS09]. These drivers are released from longitudinal vehicle control and have, theoretically, more available mental workload, if not engaged in secondary tasks.

- **Driver reaction support**
  Driver reaction support includes attracting the driver's attention, clarifying the traffic situation and supporting driver intervention. Hence, systems are addressed which provide the driver with suitable warnings, determine a desirable driver reaction, and support this reaction, such as by applying full brake pressure even if the brake pedal is not fully engaged.

- **Emergency manoeuvre**
  Whenever preventive systems or support systems are not successful, the last alternative is an autonomous manoeuvre carried out by the vehicle itself. In straight ahead traffic, this could be either evasion or braking. While autonomous and semi-autonomous braking have been on the market for years, autonomous evasive manoeuvres are mainly limited to research due to the limitations of environment recognising systems and reasons of product liability.

Because of the focus of FCWS and PBA on longitudinal vehicle dynamics, which are less critical to product liability than autonomous steering, the first collision prediction models were developed for this application. The following sections present some theoretical approaches and models from the relevant literature.

8.3.2.1. Theoretical considerations

The published FCWS and PBA systems are based on simple mechanical models using Newton's equation of motion for a single mass point and a single degree of freedom in longitudinal direction. Often, they assume constant decelerations of the involved vehicles. The problem which remains to be solved is to find an acceptable compromise between missed detections of dangerous situations and false alarms based on the driver's preferences. As the following theoretical approach shows, the main reasons for this difficulty are the unknown driver reaction to a dangerous situation and grip potential between road and tyre.

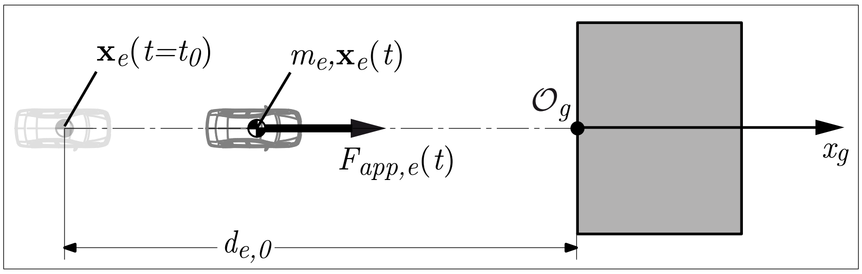


Figure 8.5.: Single mass point collision prediction model for stationary objects
The global coordinate system $\{\mathcal{O}_g, x_g\}$ is fixed to the stationary object. In $F_{app,e}$ all external forces (driving or braking forces and driving resistances) are put together.

8.3.2.1.1. Stationary preceding objects The simplest collision prediction model deals with stationary objects in front of the ego-vehicle. The ego-vehicle is reduced to a mass point in the centre of gravity with a single degree of freedom in longitudinal direction, Fig. 8.5. The equation of motion is given by

$$m_e \cdot a_e(t) = F_{app,e}(t) = \mu_e(t) \cdot m_e \cdot g \cdot p_e(t) \ , \tag{8.15}$$

with the initial state of the vehicle

$$\mathbf{x}_e(t_0) = \begin{bmatrix} d_{e,0} \\ v_{e,0} \end{bmatrix} \ . \tag{8.16}$$

In (8.15), m_e denotes the vehicle mass, and $F_{app,e}(t)$ represents external forces applied on the vehicle centre. This force $F_{app,e}(t)$ consists of the driving resistances and the driving or braking force. In the following discussion, the influences of dynamic wheel loads, inertia of the drive train and non-linear tyre forces are neglected. In addition, the reaction time of the driver and the response time of the braking system are not taken into account. Symbol $\mu_e(t)$ is the grip potential between the road and the tyres, and p_e[8] is a coefficient for acceleration or deceleration of the vehicle, which represents the driver control input. With constant vehicle deceleration by constant values for μ_e and p_e, (8.15) is analytically solved in

$$\mathbf{x}_e(t) = \begin{bmatrix} x \\ v \end{bmatrix} = \begin{bmatrix} t \cdot (v_{e,0} + \frac{\mu_e g p_e}{2} \cdot t) - d_{e,0} \\ v_{e,0} + \mu_e g p_e t \end{bmatrix} \ , \tag{8.17}$$

with $\mathbf{x}_e(t)$ the state vector of the mass point.

The two main shortcomings of this approach are exemplified in Fig. 8.6. An initial velocity $v_{e,0}$ of 100 kph and an initial distance $d_{e,0}$ of 50 m are applied at t_0. Without information about the grip potential μ_e and the acceleration or deceleration input p_e of

[8] The "pedal" position p_e is defined as negative for braking and as positive for driving, where $-1 \leq p \leq +1$.

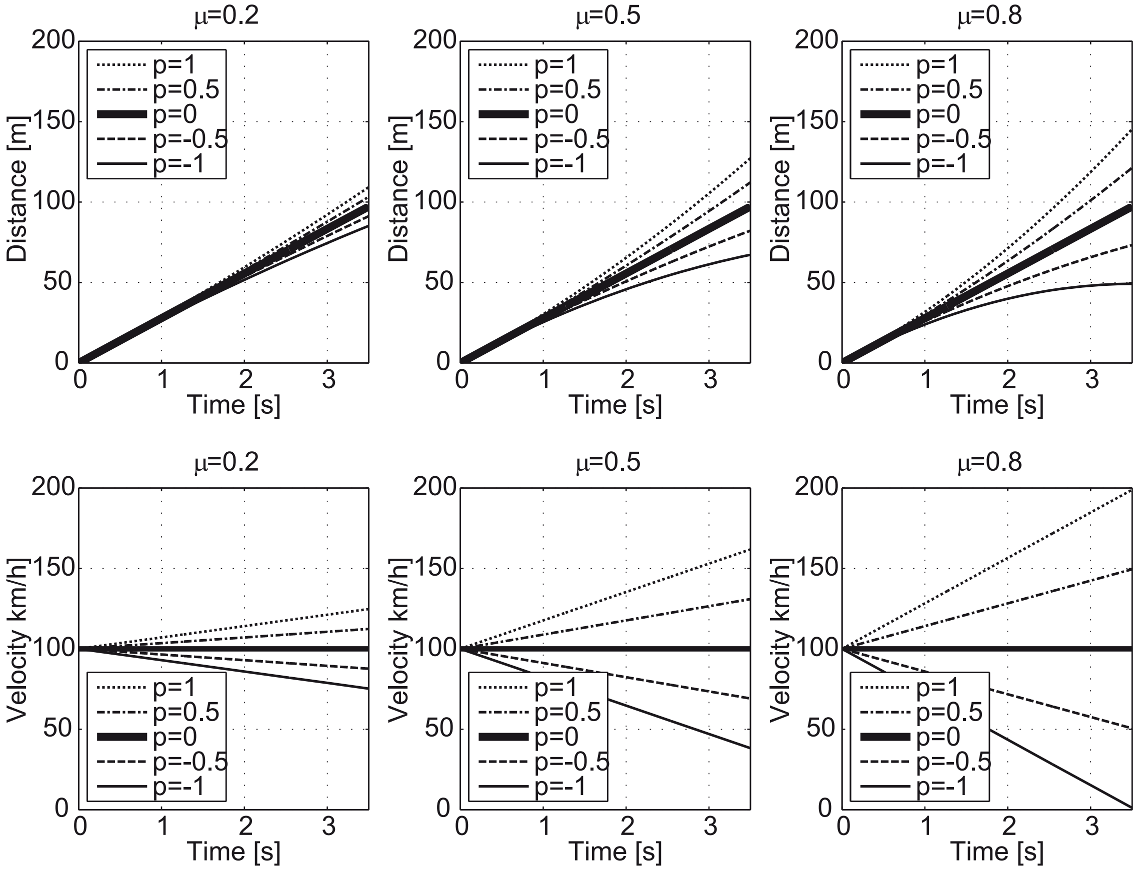


Figure 8.6.: Variation in stopping distance and collision velocity

Results for the single mass point collision prediction model. An initial velocity $v_{e,0}$ of 100 kph and distance to the preceding vehicle $d_{e,0}$ of 50m is applied. The prediction of the vehicle trajectory shows large differences due to brake/throttle pedal position p_e and grip potential μ_e.

the driver, the possible difference in stopping distance and collision velocity increases in time. After a travel of 3.5 seconds, the collision speed could be either zero or 200 kph on dry road conditions ($\mu_e = 0.8$). This results in issues with the warning or intervention strategy ranging from too early (frequent and annoying) warnings to too late warnings (which would result in collisions) even when the ERS is working perfectly. Another shortcoming is the missing consideration of the motion of the preceding vehicle.

This theoretical single mass point model qualitatively shows the theoretical range of the pre-fire times of secondary safety systems (airbag, belt, adaptive crush zones). Prefiring offers an activation of these systems with reduced and therefore less aggressive deployment and an early coupling of the occupant to the decelerating vehicle. In [GSE$^+$09] a significant reduction of the injury risk for the occupant's head by pre-firing a frontal

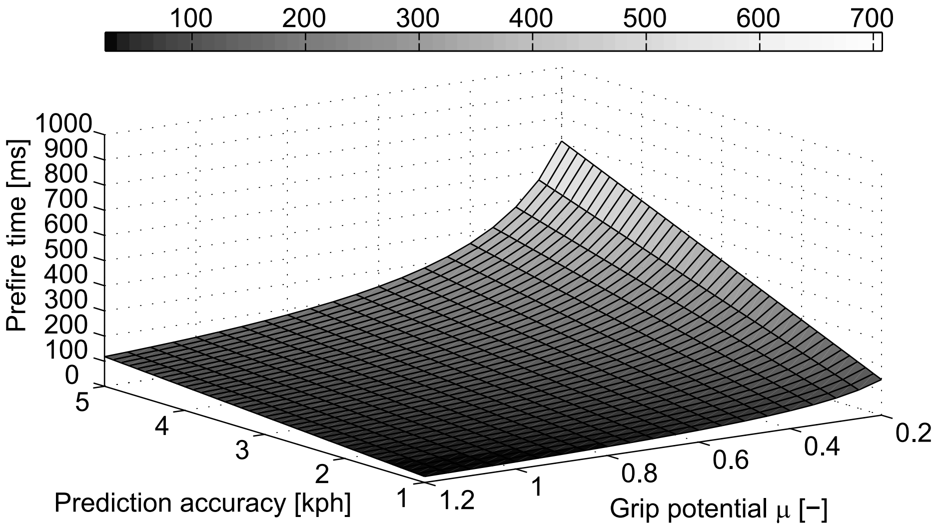


Figure 8.7.: Pre-fire times calculated by the single mass point collision prediction model
The theoretical pre-fire time as calculated by the single mass point collision prediction model. Depending on the accuracy of the prediction of the collision velocity and the adhesion potential between road and tyre, the values vary from 25 to about 1000ms, Tab. 8.1

airbag in FMVSS 208 frontal crash tests[9] is found with a pre-fire time of 80 ms. Significant reductions were also found for the FMVSS 208 Out-of-Position[10] requirements.

Depending on the required accuracy of the collision velocity prediction and the grip potential between road and tyre, a wide range of applicable pre-fire times can be observed, see Fig. 8.7 and Tab. 8.1. As an example, regardless of the driver input, at $\mu_e = 0.8$ a pre-fire time of 35 ms is still available for an accuracy of the predicted collision velocity of ± 1 kph, which corresponds to the deployment time of a standard frontal driver airbag. A pre-fire time of 80 ms as suggested by [GSE+09], would result in a theoretical accuracy of ± 3.4 kph at an extreme $\mu_e = 1.2$, and ± 2.3 kph for $\mu_e = 0.8$. These values are considered acceptable and also meet the accuracy of the environment recognising systems.

8.3.2.1.2. Moving preceding objects For theoretical considerations, the simplified equations of motion read:

$$\begin{aligned} \begin{bmatrix} m_e & 0 \\ 0 & m_o \end{bmatrix} \cdot \begin{bmatrix} a_e \\ a_o \end{bmatrix} &= \begin{bmatrix} F_{app,e} \\ F_{app,o} \end{bmatrix} , \\ \mathbf{M} \quad \cdot \quad \mathbf{a} &= \mathbf{F} . \end{aligned} \tag{8.18}$$

[9] Legal requirement in the US of the National Highway Traffic Safety Administration on advanced airbags; it includes frontal crashes against a rigid barrier.

[10] Out-of-Position means that occupants in a collision are not located in a standard position at time zero; examples are stationary airbag deployments using child dummies with the head positioned on the airbag cover, see also section 6.1.2.

Table 8.1.: Estimation of pre-fire times in [ms]

μ_e	Accuracy ± 1 kph	± 2.5 kph	± 5 kph
0.2	142	354	704
0.5	57	142	283
0.8	35	88	177
1.1	26	64	129

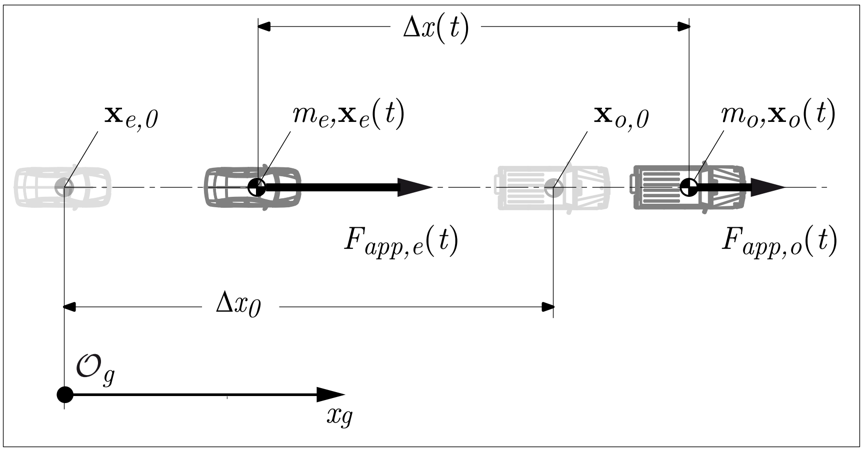


Figure 8.8.: Single mass point collision prediction model, moving preceding vehicle

These equations are similar to the single mass collision prediction model with stationary obstacles, Fig. 8.8. In (8.18) the applied forces are

$$F_{app,i} = \mu_i m_i g p_i, \quad i = e, o \tag{8.19}$$

and the initial conditions are

$$\mathbf{x}_{i,0} = \begin{bmatrix} d_{i,0} \\ v_{i,0} \end{bmatrix} \ . \tag{8.20}$$

The analytical solution is

$$\mathbf{x}_i(t) = \begin{bmatrix} x_i \\ v_i \end{bmatrix} = \begin{bmatrix} t \cdot (v_{i,0} + \frac{\mu_i g p_i}{2} \cdot t) + d_{i,0} \\ v_{i,0} + \mu_i g p_i t \end{bmatrix} \ . \tag{8.21}$$

Introducing relative distance and velocity,

$$\begin{aligned} \Delta x &= x_e - x_o \ , \\ \Delta v &= v_e - v_o \ , \\ \Delta x_0 &= d_{e,0} - d_{o,0} \ , \\ \Delta v_0 &= v_{e,0} - v_{o,0} \ , \end{aligned} \tag{8.22}$$

yields

$$\mathbf{\Delta x}(t) = \begin{bmatrix} \Delta x \\ \Delta v \end{bmatrix} = \begin{bmatrix} t \cdot [\Delta v_0 + \frac{gt}{2} \cdot (\mu_e p_e - \mu_o p_o)] - \Delta x_0 \\ \Delta v_0 + gt(\mu_e p_e - \mu_o p_o) \end{bmatrix} . \tag{8.23}$$

The prediction of the collision velocity with a moving preceding vehicle is clearly less accurate. In an extreme case (100% acceleration of the ego-vehicle and 100% braking of the preceding vehicle), the quality of the prediction is halved, see Fig. 8.9.

With $p_o = 0$, which means no external forces acting on the preceding vehicle (i.e. constant velocity), this is equal to the equations for stationary objects (8.17) when fixing the previously inertial coordinate system to the moving preceding vehicle (i.e. the absolute velocity of the ego-vehicle is replaced by the relative velocity of the two vehicles), the state vector $\mathbf{\Delta x}_{rel}(t)$ for the ego-vehicle in this case reads:

$$\mathbf{\Delta x}_{rel}(t) = \begin{bmatrix} \Delta x \\ \Delta v \end{bmatrix} = \begin{bmatrix} t \cdot [\Delta v_0 + \frac{gt}{2} \cdot \mu_e p_e] - \Delta x_0 \\ \Delta v_0 + gt\mu_e p_e \end{bmatrix} . \tag{8.24}$$

This theoretical single mass point approach assumes constant acceleration or deceleration of the involved vehicles, no latencies or response times of the drivetrain/braking system and immediate driver control. For more precise warning or intervention strategies of FCWS and PBA systems, additional considerations with respect to the driver and the braking system are necessary.

For the driver, the definitions of reaction time are not consistent in the literature. Fig. 8.10 depicts a qualitative description of the driver reaction time which in principle follows the definition in [HG09].

The driver reaction time τ_R is composed of:

- τ_{al}, time for visual acquisition of objects (allocation)
  Time between a specific warning at t_{wa} and the visual acquisition of an object at t_{al}.

- τ_{de}, time for decision making
  Time between t_{al} and the decision upon the preferred reaction at t_{de}.

- τ_{rc}, time to reach the vehicle controls
  Time between t_{de} and the moment t_{rc} when the driver reaches the corresponding vehicle control system (e.g. moving the foot from gas to brake pedal).

- τ_{lc}, latency time of vehicle control system
  Time between t_{rc} and the moment $t_{o,s}$ when the activated vehicle control system actually starts to operate (first increase in braking force). τ_{lc} is added to τ_R , since it belongs to terms without vehicle acceleration in (8.25). For the braking system this value is about 0.05 s.

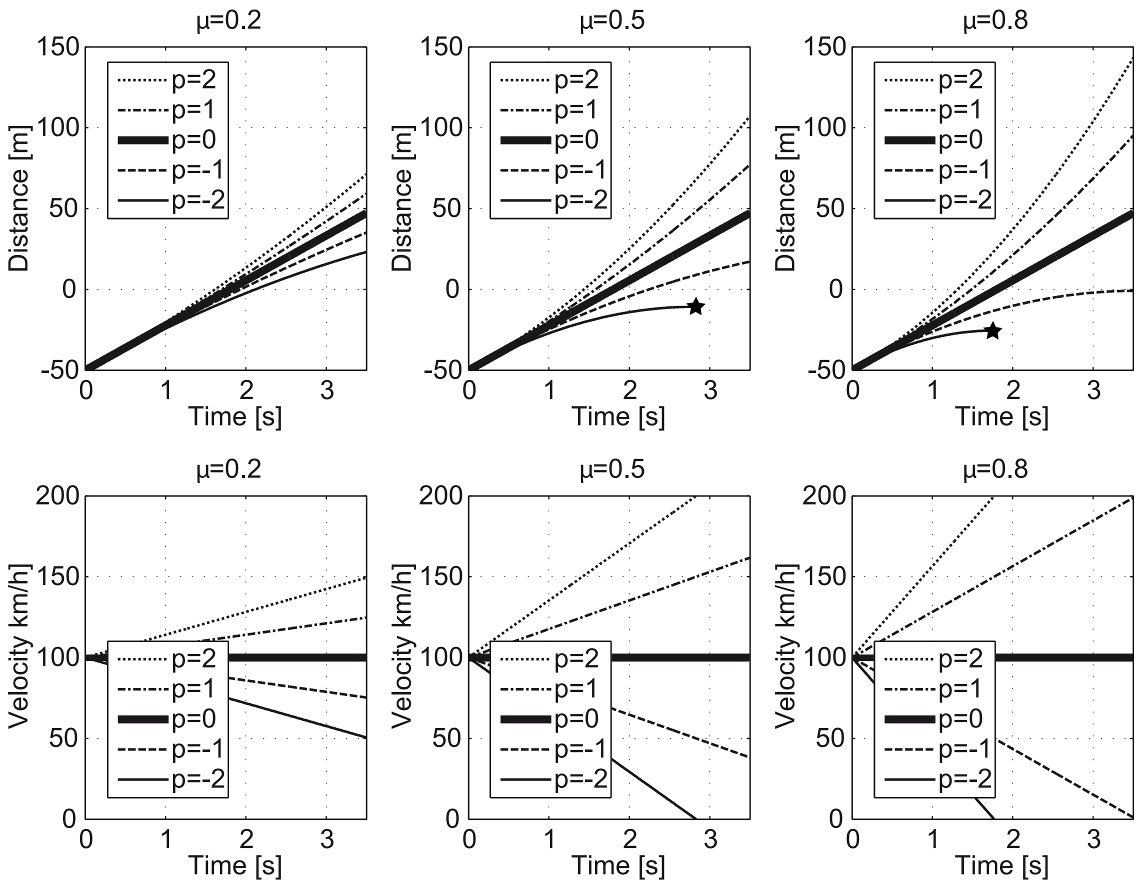


Figure 8.9.: Variations in relative distance and velocity between ego and moving preceding vehicle

Results for the single mass point collision prediction model. An initial relative velocity Δv_0 of 100 kph and distance to the preceding vehicle Δx_0 of 50 m is applied. The prediction of relative distance and velocity is less accurate than with stationary obstacles, leading to decreased pre-fire times. Symbol ★ denotes a collision between the vehicles.

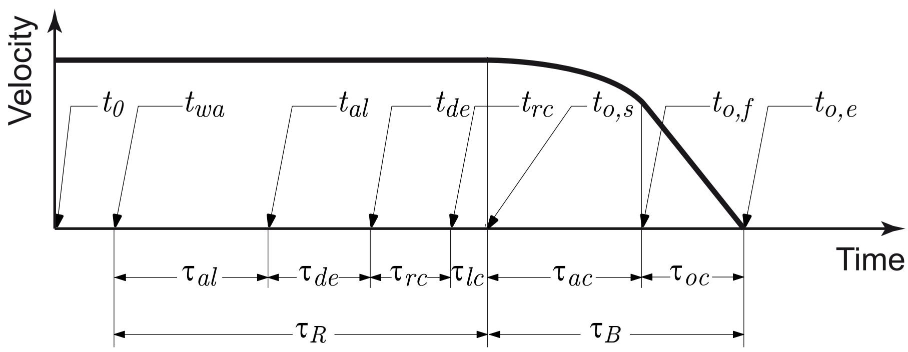


Figure 8.10.: Qualitative description of driver reaction

τ_C (time during control of vehicle) is composed of:

- τ_{ac}, time for activation of vehicle controls
  Time between $t_{o,s}$ and the full operation of the vehicle controls at $t_{o,f}$.
- τ_{oc}, time of operation of vehicle controls
  Time between $t_{o,f}$ and the end of the operation sequence at $t_{o,e}$.

In [Win09] the equations for warning distances which are necessary to avoid a collision are given with respect to non-decelerated obstacles, decelerated obstacles and obstacles decelerating until stopped.

- **Non-accelerated/decelerated obstacles**
  The warning distance Δx_w for non-accelerated/decelerated obstacles are given as a first approximation by

$$\Delta x_w = \Delta v \cdot (\tau_R + \tau_B) + \frac{\Delta v^2}{2a_{max}} \ , \tag{8.25}$$

  where a_{max} is the average deceleration during the braking sequence, τ_B the effective loss of time until the braking deceleration a_{max} is achieved and τ_R the reaction time of the driver. Time span τ_B,

$$\tau_B = \frac{\tau_{ac}}{2} \ , \tag{8.26}$$

  is half of τ_{ac} when assuming a linear increase in the braking deceleration.

- **Decelerated obstacles**

$$\Delta x_w = (\Delta v + a_{rel}\frac{\tau_R + \tau_B}{2})(\tau_R + \tau_B) + \frac{[\Delta v + a_{rel}(\tau_R + \tau_B)]^2}{2a_{max,rel}} \tag{8.27}$$

  where a_{rel} is the relative acceleration between obstacle a_o and ego-vehicle a_e,

$$a_{rel} = a_o - a_e \tag{8.28}$$

  and $a_{max,rel}$ the maximum relative acceleration.

- **Obstacles decelerated until stopped**

$$\Delta x_w = \frac{v_e^2}{2a_{max,e}} - \frac{v_o^2}{2a_{max,o}} + v_e \cdot \tau_B + (\Delta v + a_{rel}\frac{\tau_R}{2}) + \tau_R \ , \tag{8.29}$$

  where v_e and v_o are velocities of ego- and obstacle vehicle, $a_{max,e}$ and $a_{max,o}$ the maximum decelerations.

The Time-to-Collision (TTC) is then calculated by

$$TTC = \frac{\Delta x_w}{\Delta v}, \ \Delta v > 0 \ . \tag{8.30}$$

8.3.2.2. State-of-the-art in FCWS

While several models related to FCWS and PBA systems have been introduced to the market, most of them are not published in enough detail to gain precise insight into their functions. The following section reviews some of the published literature. FCWS and PBA systems are logical enhancements of Automatic Cruise Control (ACC), since the hardware (ERS and actuating system) is then available. Nevertheless, FCWS and PBA have to operate in well-balanced cooperation with the human driver, whereas ACC systems can run autonomously within defined safe operation boundaries.

Early investigations in longitudinal control and driver braking behaviour were conducted in [Lee76]. A mathematical theory was presented that describes the control of the braking sequence by a human driver and his visual sense. The theory is based on the changing optic array at the driver's eyes during the braking manoeuvre. It was concluded that the most relevant information is Time-to-Collision (TTC), rather than mere information about the distance, velocity or acceleration/deceleration of the preceding vehicle.

A literature review on published FCWS and PBA systems was carried out in [LP05]. It was found that the two main challenges are firstly the development of reliable ERS which run errorless in any environmental condition and secondly finding a compromise between false (annoying) and missed (no driver support) alarms/interventions. It was argued that warning or intervention strategies of FCWS and PBA are a matter of signal detection and decision making theory. In practice, due to uncertainties in ERS measurement and interpretation as well as driver experiences or expectations, a 100% perfect separation of warning/intervention and non-warning/intervention is not achievable. In [LP05] the following five different algorithms described in literature were evaluated with data from driving tests:

- TTC=10 s
- FCWS/PBA algorithm by Doi et al.
- FCWS/PBA algorithm by Fujita et al.
- FCWS/PBA algorithm by Barber et al.
- FCWS/PBA algorithm by Brunson et al.

The input data was derived from a database with records of the real driving of 15 human drivers, [FES$^+$98]. In [LP05] the driving data was processed and data sets with safe and threatening driving situations with respect to rear-end collisions were separated. The data was applied to the FCWS/PBA algorithms and their performance with respect to the cases of Tab. 8.2 was analysed. It was concluded that

- the simple $TTC = 10\ s$ algorithm has the best rate for identifying threatening situations (True positive),

Table 8.2.: Distinction of cases

Abb.	Description	Driving sit.	Prediction	Result
TP	True Positive	Threatening	Threatening	Correct alarm/intervention
FN	False Negative	Threatening	Safe	Missed alarm/intervention
TN	True Negative	Safe	Safe	Correct no alarm/intervention
FP	False Positive	Safe	Threatening	False alarm/intervention

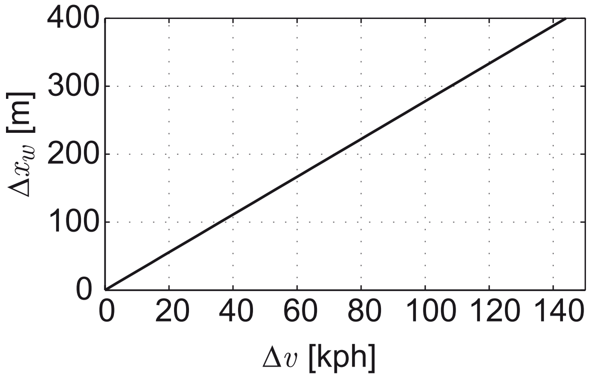


Figure 8.11.: Collision warning distance Δx_w of the TTC_{10} FCWS model

- all algorithms fail to perfectly separate safe and threatening driving situations,
- all algorithms have higher rates of detecting safe data than threatening situations,
- FCWS/PBA algorithms are currently calibrated to reduce false alarms/interventions, at the cost of the detection of emergency situations.

These FCWS/PBA models are briefly discussed in the following sections.

8.3.2.2.1. $\mathbf{TTC_{10}}$ model In the early literature, [Lee76], a Time-to-Collision of 10 s is proposed for FCWS systems, which was derived from empirical data of driver preferences. The equation for the collision warning distance is a simple linear relationship between the relative distance Δx_w and the relative velocity Δv,

$$\Delta x_w = TTC_{10} \cdot \Delta v \ , \tag{8.31}$$

with the graphical interpretation in Fig. 8.11. As mentioned above, this algorithm warns of hazardous situations related to rear-end collisions at high detection rates for correct alarms/interventions ("True Positive"). Its drawback is its failure to consider the human control input and the grip potential, which results in unacceptably high rates of false alarms/interventions ("False Positive").

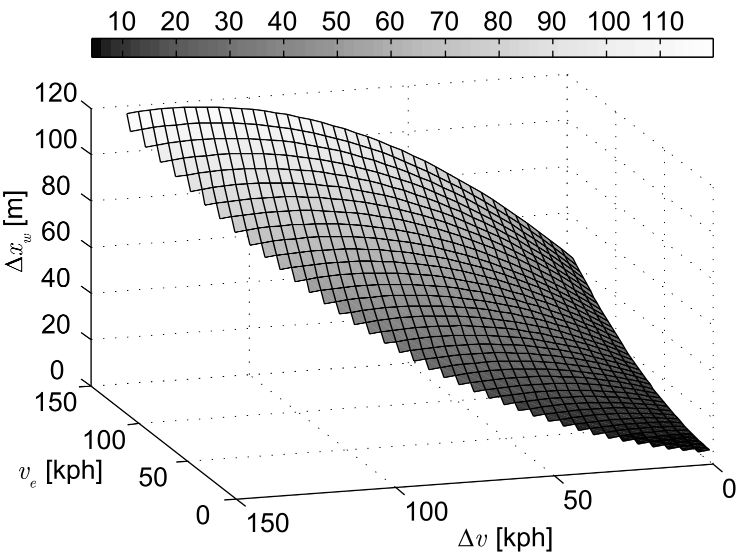


Figure 8.12.: Warning/Braking Δx_w distance of algorithm by Doi et al.
The relative distance $\Delta x_w = f(\Delta v, v_e)$ where collision warning/autonomous braking is applied on the ego-vehicle. Results for negative values for v_o (the preceding vehicle is approaching) are removed in the figure.

8.3.2.2.2. FCWS/PBA algorithm by Doi et al. One of the first algorithms for full-vehicle application was developed by [DBN+94]. The algorithm is described as follows

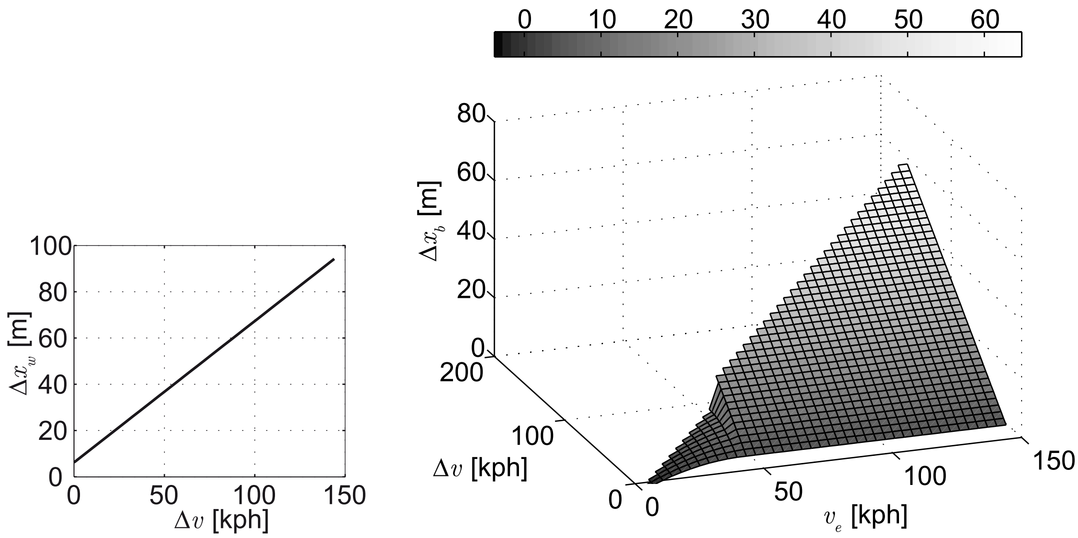
$$\Delta x_w = f(\Delta v, v_e) = \frac{1}{2}\left(\frac{v_e^2}{a_e} - \frac{v_o^2}{a_o}\right) + v_e \tau_e - \Delta v \tau_o + \Delta x_{min} \ , \tag{8.32}$$

with v_e, v_o the observed velocity of ego-vehicle e and obstacle o. The proposed parameters used in the model are:
$a_e = 6$ [m/s^2], applied deceleration on ego-vehicle
$a_o = 8$ [m/s^2], supposed braking of preceding vehicle
$\tau_o = 0.6$ [s], time when supposed heavy braking of preceding vehicle is applied
$\tau_e = 0.1$ [s], time when ego-vehicle starts to brake[11]
$x_{min} = 5$ [m], minimum distance where braking is applied
It assumes that the preceding vehicle is engaging full braking which results in a collision warning/emergeny braking of the ego-vehicle at a certain relative distance Δx_w.

In Fig. 8.12, $\Delta x_w = f(\Delta v, v_e)$ is depicted. The warning distance Δx_w increases progressively with the ego-vehicle velocity v_e. At higher relative velocities Δv, the influence becomes less important. In [LP05] a poor performance of the algorithm in the simulated driving data was reported.

8.3.2.2.3. FCWS/PBA algorithm by Fujita et al. Published in [FAS95], it consists of a warning and an autonomous braking element. A warning is emitted at a relative

[11] After start of braking of preceding vehicle



(a) Δx_w

(b) Δx_b

Figure 8.13.: Warning and braking Δx_w, Δx_b distance of algorithm by Fujita et al. The left figure depicts $\Delta x_w = f(\Delta v)$; note the simple linear relationship. The right fugure depicts the autonomous braking intervention. Braking is applied at $\Delta x_b = f(\Delta v, v_e)$

distance of Δx_w,

$$\Delta x_w = f(\Delta v) = 2.2\Delta v + 6.2 \ , \tag{8.33}$$

and the autonomous braking is initiated at Δx_b,

$$\begin{aligned} \text{if } x_e \ &\geq 42 \ kph: \ \Delta x_b = f(\Delta v, v_e) = \tau_o \Delta v + \tau_e \tau_o a_e - 0.5 a_e \tau_e^2 \ , \\ \text{if } v_e \ &< 42 \ kph: \ \Delta x_b = f(\Delta v, v_e) = \tau_o v_e - 0.5 a_e (\tau_o - \tau_e)^2 - \frac{v_o^2}{2a_o} \ . \end{aligned} \tag{8.34}$$

Fig. 8.13 illustrates the dependance of Δx_w and Δx_b on Δv and v_e. Compared to the algorithm of Doi et al. it relies more on linear relationships. The braking is applied at smaller distances. The performance in the simulated driving tests was poor, [LP05]. The braking intervention algorithm Δx_b produces only a rate of 3% of correct brake interventions (True Positive).

8.3.2.2.4. FCWS/PBA algorithm by Barber et al. The model described in [BC98] again consists of a warning and a braking element. Depending on whether the preceding obstacle vehicle is moving or not, the warning distance Δx_w is calculated by:

$$\begin{aligned} \text{if } v_o = 0: \ & \Delta x_{w,stat} = f(\Delta v) = TTC_4 \cdot \Delta v \ , \\ \text{if } v_o > 0: \ & \Delta x_{w,mov} = f(\Delta x, \Delta v, \Delta a) = 4\Delta v \frac{2\Delta x}{\Delta v \pm \sqrt{\Delta v^2 - 2\Delta x \Delta a}} \ . \end{aligned} \tag{8.35}$$

The braking distance Δx_b is calculated by

$$\Delta x_b = f(\Delta v) = \frac{1}{2} a \Delta v^2 \ . \tag{8.36}$$

In the case of a stationary preceding object ($v_o = 0$), the warning part simply warns at 4 s to collision (TTC_4) with respect to the relative velocity Δv. In the case of moving objects ($v_o > 0$) constant decelerations $a_e, \ a_o$ of both vehicles are assumed and predicted into the future. The warning distance $\Delta x_{w,mov}$ is derived from:

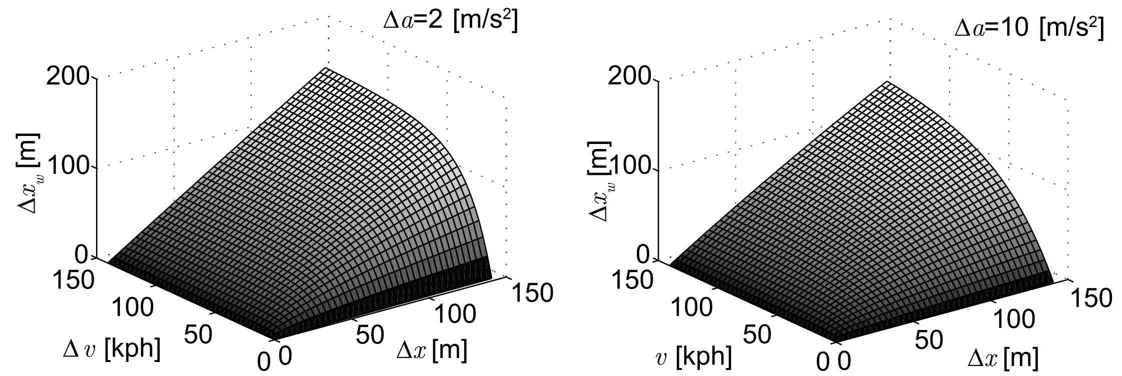
$$\begin{aligned}
&a_e = const., \ a_o = const. \text{ yields :} \\
&x_e(t) = a_e t^2/2 + v_{e,0} t + x_{e,0} \ , \\
&x_o(t) = a_o t^2/2 + v_{o,0} t + x_{o,0} \ , \\
&\text{where } x_{e(o),0}, \ v_{e(o),0} \text{ are initial conditions.} \\
&\text{Introducing } \Delta a = a_e - a_o, \ \Delta v = v_e - v_o, \ \Delta x = x_e - x_o \ , \\
&\text{with } x_e = x_o \rightarrow \ t = TTC \ , \text{ in case of a collision} \\
&\text{yields} \\
&TTC = \frac{-\Delta v \pm \sqrt{\Delta v^2 - 2 \Delta x \Delta a}}{\Delta a} \ .
\end{aligned} \tag{8.37}$$

Now, $\Delta x_{w,mov}$ can be derived by

$$\begin{aligned}
&\Delta x_{w,mov} = TTC_4 \Delta v = 4 \ \Delta v \ TTC = \\
&= 4 \ \Delta v \ \frac{-\Delta v \pm \sqrt{\Delta v^2 - 2 \Delta x \Delta a}}{\Delta a} \ = \\
&= 4 \ \Delta v \ \frac{-\Delta v \pm \sqrt{\Delta v^2 - 2 \Delta x \Delta a}}{\Delta a} \ \frac{\Delta v \pm \sqrt{\Delta v^2 - 2 \Delta x \Delta a}}{\Delta v \pm \sqrt{\Delta v^2 - 2 \Delta x \Delta a}} \ = \\
&= 4 \Delta v \frac{2 \Delta x}{\Delta v \pm \sqrt{\Delta v^2 - 2 \Delta x \Delta a}} \quad .
\end{aligned} \tag{8.38}$$

Complex solutions of TTC mean "no collision". If both solutions of TTC are positive, the smaller value has to be taken. For calculation of the warning distance $\Delta x_{w,mov}$ from the predicted TTC, four times the calculated TTC is used. Fig. 8.14 illustrates $\Delta x_{w,mov}$ for moving preceding objects with different relative accelerations Δa. For an automotive application, the relative deceleration Δa has to be measured which requires derivations of measured distances and velocities and results in inaccuracies. Fig. 8.15 compares the warning and braking strategy in the case of stationary preceding objects. The simple linear relationship of $\Delta x_{w,stat}$ results in an earlier warning at lower relative velocities.

8.3.2.2.5. FCWS/PBA algorithm by Brunson et al. A more complex model was developed in [BKPP02], which is described in detail in [LP05]. The algorithm is based on the Time-to-Stop of the preceding vehicle. In principle, it divides the overall incident into three segments: Firstly from the time of beginning of the observation of the



(a) Relative deceleration between both vehicles at $\Delta a = 2$ m/s^2 (b) Relative deceleration between both vehicles at $\Delta a = 10$ m/s^2

Figure 8.14.: Warning distance in the case of a moving preceding vehicle, algorithm by Barber et al.

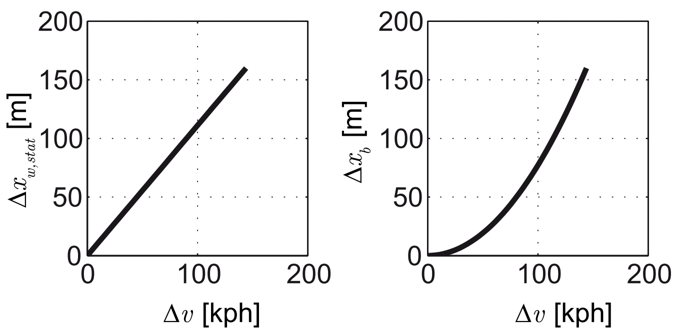


Figure 8.15.: Warning/Braking distance in the case of a stationary preceding vehicle, algorithm by Barber et al.

oncoming collision until the assumed driver reaction; secondly from the driver reaction until stop of the preceding vehicle; and thirdly from stop of the preceding vehicle until ego-vehicle stop.

8.3.2.2.6. FCWS/PBA model of Wada et al. A more recent algorithm was presented in [WDI$^+$07]. It was hypothesised that drivers sense the risk for collision with a preceding vehicle by assessing the area change of the moving obstacle projected on the retina[12]. The theory is based on the fact that the area of the obstacle projected on the retina is proportional to the distance $1/\Delta x^2$, Fig. 8.16, derived by:

[12]The retina is a light sensitive tissue located on the inner surface of the eye.

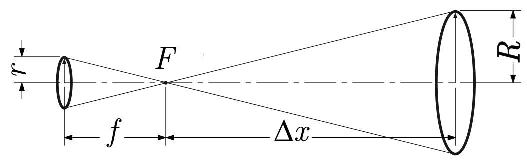


Figure 8.16.: Scheme of an object projected on the human retina
A circular obstacle (radius R) is projected on the human retina (radius r). Δx is the distance, and f the focus length of the human eye.

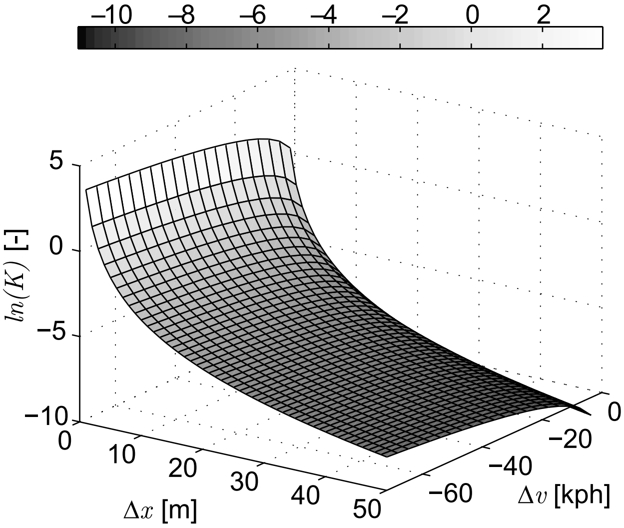


Figure 8.17.: Index K denoting collision risk according to [WDT+08]
Relationship of lnK (collision risk) in (8.40). The dominant influence is the relative distance Δx.

$$\begin{aligned} &r/f = R/\Delta x \ , \\ &a_1 = r^2\pi, a_2 = R^2\pi \ , \\ &\frac{a_1}{a_2} \sim \frac{1}{\Delta x^2} \ . \end{aligned} \tag{8.39}$$

Assume K as an index denoting the collision risk :

$$K(\Delta x, \Delta v) = \frac{d}{dt}\left(\frac{1}{\Delta x^2}\right) = -\frac{2}{\Delta x^3}\Delta v \ , \tag{8.40}$$

with Δx and Δv the relative distance and velocity between ego- and preceding vehicle, Fig. 8.17 depicts this relationship. The model was extended and verified with experiments using a driver simulator and expert drivers in [WDT+08]. It was concluded that by measuring the relative distance and velocity, a “smooth” deceleration profile can be determined for different situations with this approach. Application of this model with respect to the warning and intervention strategy in FCWS and PBA systems are described in [IT07].

8.3.2.2.7. Influence of Environment Recognising Systems (ERS) The influence of errors of ERS was investigated by [ZM03]. The investigated algorithms were:

- Doi et al.
- Fujita et al.
- Brunson et al.

It was concluded that sensor uncertainties are less important when measurements are taken directly from ERS or when only first derivatives are used. For more complex algorithms and higher order derivatives, the sensor accuracy could be of significant influence on the collision warning.

8.3.3. Planar collision prediction, 2 and 3 degrees of freedom

FCWS/PBA systems focus on traffic ahead of the ego-vehicle and rear-end impacts. Steering input of the driver in hazardous situations is not considered. Mathematical modelling of evasive manoeuvres requires additional degrees of freedom. This section investigates the planar driving behaviour of vehicles. For collision prediction, the translational degrees of freedom in longitudinal and lateral direction of the vehicle are used, as well as the rotational degree of freedom around the vertical axis. In principle, collision prediction can be considered as an extrapolation of vehicle trajectories in time (Fig. 8.18). Starting from time zero, the ego-vehicle (indexed e) with the current state vector $\mathbf{x}_e(t)$ moves along trajectory tr_e. At the same time, another vehicle (indexed o) with the current state vector $\mathbf{x}_o(t)$ moves along trajectory tr_o. Based on these trajectories and other inputs[13], future trajectories $tr_{e,p}$ and $tr_{o,p}$ are predicted with different methods, which are investigated in the next sections. At time $t+t_f$, the prediction algorithm has calculated a collision with collision parameters derived from $\mathbf{x}_e(t+t_f)$ and $\mathbf{x}_o(t+t_f)$.

However, in the example depicted in Fig. 8.18, both drivers have executed an evasive manoeuvre with the real trajectories $tr_{e,r}$ and $tr_{o,r}$ at time $t+t_f$. This illustrates that collision prediction is a matter of probability, which has to be taken into account by the algorithms. The target is an acceptable compromise with a trade-off between false alarms or interventions and missed detections. Nevertheless, the algorithms have to be designed for real-time performance.

The next section discusses state-of-the-art in planar collision prediction. Because of the large amount of relevant literature dealing with this topic, this cannot be complete.

8.3.3.1. State-of-the-art in planar collision prediction

A comprehensive literature survey on the field of target tracking was done in [LJ00]. In [Jan05], approaches focused on automotive collision avoidance were investigated. It was

[13] E.g. information of the dynamical state, driver input, grip potential

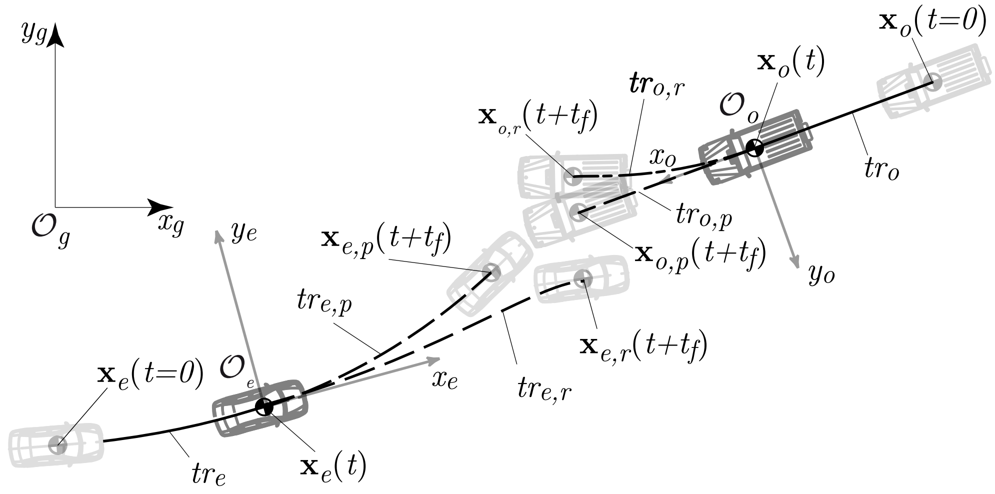


Figure 8.18.: Illustration of collision prediction

pointed out that the most common methods for the model-based estimation of vehicle states are Kalman Filter (KF), Extended Kalman Filter (EKF) and Particle Filter (PF). In [PTAA07], an approach was proposed for the prediction of vehicle trajectories. The future states are defined in state space. Vector $\mathbf{x}_k$ of size $n \times 1$ is the dynamic state vector of the vehicle, the future state $\mathbf{x}_{k+1}$ can be predicted by

$$\mathbf{x}_{k+1} = \mathbf{A}\mathbf{x}_k + \mathbf{w}_k \ , \tag{8.41}$$

where $\mathbf{A}$ is the $n \times n$ transition matrix for a specific dynamic model and $\mathbf{w}_k$ is the process noise with a covariance matrix $\mathbf{Q}$. The literature mentions the following dynamic models:

- **Constant velocity model**
  Constant velocity is assumed piecewise between two time steps $\Delta T = t_{k+1} - t_k$ and motion in longitudinal and lateral direction is assumed to be independent. The dynamic state $\mathbf{x}_k$ reads:
  $$\mathbf{x}_k = \begin{bmatrix} x & y & v_x & v_y \end{bmatrix}^T , \tag{8.42}$$
  and the estimated dynamical state $\mathbf{x}_{k+1}$ is calculated by
  $$\mathbf{x}_{k+1} = \begin{bmatrix} 1 & 0 & \Delta T & 0 \\ 0 & 1 & 0 & \Delta T \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ v_x \\ v_y \end{bmatrix} + \begin{bmatrix} \frac{\Delta T^2}{2} & 0 \\ 0 & \frac{\Delta T^2}{2} \\ \Delta T & 0 \\ 0 & \Delta T \end{bmatrix} \mathbf{v}_k \ . \tag{8.43}$$
  The transition is modelled with Gaussian noise by the covariance matrix $\mathbf{Q}_k$, it includes the standard deviations:
  $$\mathbf{Q}_k = \begin{bmatrix} \sigma_{v_x}^2 & 0 \\ 0 & \sigma_{v_y}^2 \end{bmatrix} \ . \tag{8.44}$$

- **Constant acceleration model**
  Constant acceleration is assumed piecewise between two timesteps $\Delta T = t_{k+1} - t_k$. The dynamic state $\mathbf{x}_k$ reads:

$$\mathbf{x}_k = \begin{bmatrix} x & y & v_x & v_y & a_x & a_y \end{bmatrix}^T , \tag{8.45}$$

  and the estimated dynamical state $\mathbf{x}_{k+1}$ is calculated by

$$\mathbf{x}_{k+1} = \begin{bmatrix} 1 & 0 & T & 0 & \frac{\Delta T^2}{2} & 0 \\ 0 & 1 & 0 & T & 0 & \frac{\Delta T^2}{2} \\ 0 & 0 & 1 & 0 & T & 0 \\ 0 & 0 & 0 & 1 & 0 & T \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ v_x \\ v_y \\ a_x \\ a_y \end{bmatrix} + \begin{bmatrix} \frac{\Delta T^3}{6} & 0 \\ 0 & \frac{\Delta T^3}{6} \\ \frac{\Delta T^2}{2} & 0 \\ 0 & \frac{\Delta T^2}{2} \\ \Delta T & 0 \\ 0 & \Delta T \end{bmatrix} \mathbf{v}_k \ . \tag{8.46}$$

  The transition is again modelled with Gaussian noise by the convariance matrix:

$$\mathbf{Q}_k = \begin{bmatrix} \sigma_{a_x}^2 & 0 \\ 0 & \sigma_{a_y}^2 \end{bmatrix} \ . \tag{8.47}$$

- **Coordinated turn model**
  Piecewise constant velocity in longitudinal direction between two timesteps $\Delta T = t_{k+1} - t_k$ and a constant lateral force in lateral direction resulting in constant yaw rate ω_z is assumed, [MWF82]. The dynamic state $\mathbf{x}_k$ reads:

$$\mathbf{x}_k = \begin{bmatrix} x & y & v_x & v_y \end{bmatrix}^T , \tag{8.48}$$

  and the estimated state vector $\mathbf{x}_{k+1}$ reads:

$$\mathbf{x}_{k+1} = \begin{bmatrix} 1 & 0 & \frac{\sin(\omega_z \Delta T)}{\omega_z} & \frac{1-\cos(\omega_z \Delta T)}{\omega_z} \\ 0 & 1 & \frac{1-\cos(\omega_z \Delta T)}{\omega_z} & \frac{\sin(\omega_z \Delta T)}{\omega_z} \\ 0 & 0 & \cos(\omega_z \Delta T) & \sin(\omega_z \Delta T) \\ 0 & 0 & -\cos(\omega_z \Delta T) & \sin(\omega_z \Delta T) \end{bmatrix} \begin{bmatrix} x \\ y \\ v_x \\ v_y \end{bmatrix} + \begin{bmatrix} \Delta T^2 & 0 \\ 0 & \Delta T^2 \\ \Delta T & 0 \\ 0 & \Delta T \end{bmatrix} \mathbf{v}_k \ . \tag{8.49}$$

  The yaw rate ω_z is assumed to be known. The covariance matrix $\mathbf{Q}_k$ is defined in the same way as for the constant velocity model,

$$\mathbf{Q}_k = \begin{bmatrix} \sigma_{v_x}^2 & 0 \\ 0 & \sigma_{v_y}^2 \end{bmatrix} \ . \tag{8.50}$$

- **Nearly Coordinated turn model**
  The coordinated turn model is enhanced by an additional estimated state ω_z, the dynamic state $\mathbf{x}_k$ then reads:

$$\mathbf{x}_k = \begin{bmatrix} x & y & v_x & v_y & \omega_z \end{bmatrix}^T , \tag{8.51}$$

the estimated state vector $\mathbf{x}_{k+1}$ is given by:

$$\mathbf{x}_{k+1} = \begin{bmatrix} 1 & 0 & \frac{\sin(\omega_z \Delta T)}{\omega_z} & \frac{1-\cos(\omega_z \Delta T)}{\omega_z} & 0 & \\ 0 & 1 & \frac{1-\cos(\omega_z \Delta T)}{\omega_z} & \frac{\sin(\omega_z \Delta T)}{\omega_z} & 0 & \\ 0 & 0 & \cos(\omega_z \Delta T) & \sin(\omega_z \Delta T) & 0 & \\ 0 & 0 & -\cos(\omega_z \Delta T) & \sin(\omega_z \Delta T) & 0 & \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ v_x \\ v_y \\ \omega_z \end{bmatrix} + \begin{bmatrix} \Delta T^2 & 0 & 0 \\ 0 & \Delta T^2 & 0 \\ \Delta T & 0 & 0 \\ 0 & \Delta T & 0 \\ 0 & 0 & 0 \end{bmatrix} \mathbf{v}_k \ . \tag{8.52}$$

The covarinace matrix $\mathbf{Q}_k$ reads:

$$\mathbf{Q}_k = \begin{bmatrix} \sigma_{v_x}^2 & 0 & 0 \\ 0 & \sigma_{v_y}^2 & 0 \\ 0 & 0 & \sigma_{\omega_z}^2 \end{bmatrix} \ . \tag{8.53}$$

Further dynamic models, such as the *Constant Acceleration and Yaw Rate Derivatives Model* or the *Constant Turn Rate and Constant Tangential Acceleration Model* can be found in [PTAA07, TPA05]. The choice of the dynamic models depends on the application and the dynamic state, since the models described above aim at certain driving states of the vehicle. In [PTAA07] it is even proposed to choose the model based on the driving state by use of heuristic rules, Adaptive Dynamic (AD) model.

- **Elastic band method**
  In [SB08, BSW07, GS07], the "elastic band" path planning method was used in automated driving application. The method was derived from applications in robotics [Kha86, Kro83]. In principle, nodes connected by spring-type elements ("elastic bands") form a trajectory, which is followed by the vehicle. The location of the elastic bands is determined by artificial external forces. The computed equilibrium of the elastic bands reacting to the forces forms the planned path. Artificial hazard potential fields generate these forces and represent virtual models of the environment. For example, the boundaries of the road, stationary and moving obstacles serve as sources for hazard fields. In the case of moving obstacles, their motion has to be anticipated in the future. In [SB08, BSW07] an extrapolation strategy to these moving objects was applied, see Fig. 8.19. The extrapolation is based on the assumption of small side slip angles $\beta \to 0$ of the obstacle's motion. It differentiates between the cases "extrapolation in the lane" and "extrapolation with lane departure". The decision between the two cases is made by evaluating the difference between the orientation of vehicle x_i and the related road centreline $x_{r,i}$.

- **Dynamic vehicle modelling**
  Apart from the methods described above, state estimation of vehicles can be carried out with more detailed vehicle dynamics modelling. The most common approaches are the single-track and the double-track vehicle models. For the ISC, a single-track model will be used, which is described in section 8.4.1. The application for

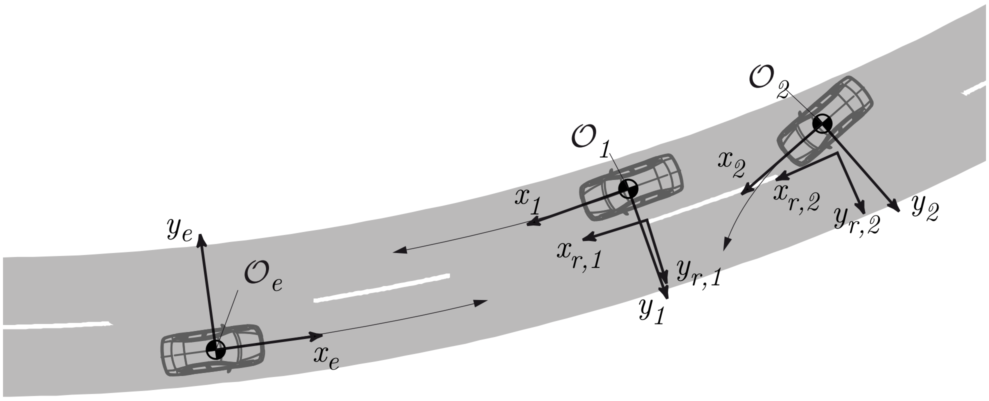


Figure 8.19.: Estimation of vehicle trajectories, according to [BSW07]
Vehicles indexed 1 and 2 are moving obstacles. At $t = 0$, the kinematic course is extrapolated. For vehicle 1, the case "extrapolation in the lane", for vehicle 2 "extrapolation with lane departure" is applied, which leads to different predicted vehicle paths.

state estimation and trajectory prediction for ego- vehicle and obstacle vehicle is investigated below.

The following chapter briefly describes one selected collision prediction model used in the ISC approach. Currently, several different approaches are being investigated, [Pij08, Pue09, BH97]. Since collision avoidance is not the target but rather a reliable prediction of collision parameters approximately 100 ms before first contact, the focus is on simple and fast algorithms.

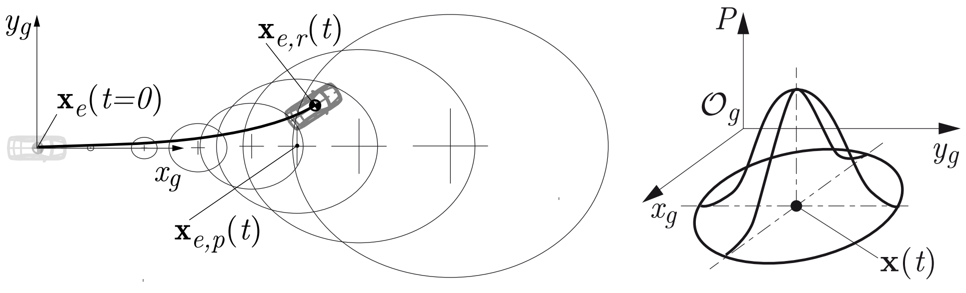
8.3.3.2. Collision prediction with Kalman filter

In 1960 the Kalman Filter (KF) method was introduced for control of linear dynamic systems, [Kal60]. Since then, the method - a special case of the Bayesian filter [Wat08] - has been used in many applications and scientific disciplines. Subsequently, it has received extensions for non-linear processes, the Extended Kalman Filter (EKF), Central-Difference Kalman Filter, Unscented Kalman Filter (UKF), and recently the Cubature Kalman Filter (CKF), [AH09]. Several points make the KF advantageous for automotive collision prediction:

- It is based on physical models of the dynamic system.
  The model-based approach allows for the simulation of the actual process with the desired accuracy and keeps the results within specific limits with physical meaning.

- It considers inaccuracies of modelling and measurement.
  A white and Gaussian distributed noise is applied to the process and the measurements to take this into account.

- It allows real-time performance by efficient algorithms.
  It is designed as a recursive predictor-corrector algorithm, with a small number



(a) Areas with probable locations of a vehicle

(b) Uncertainty ellipsoid concept

Figure 8.20.: Uncertainty of trajectory prediction, adapted from [Win09]
The left figure illustrates probable locations of a passenger vehicle. The vehicle starts with $v_x(0)$=10 m/s. Every 0.5 s an ellipsoid is depicted which is the boundary of the probable locations of the vehicle's CoG. Here, the maximal longitudinal and lateral accelerations are limited to 1.2 and 1.0 g respectively. The right figure depicts the uncertainty ellipsoids concept.

of equations using methods of linear algebra. CPU intensive time integration of dynamic processes is not required.

The mathematical background of the Kalman Filter method is not treated in this thesis, further details can be found in literature, e.g. [GA01, Wel09, Kal60, BH97]. The main idea of KF is the estimation of a discrete $n \times 1$ state vector $\hat{\mathbf{x}}_k$ of a dynamic process, and the correction of the estimated state with measurement data $\mathbf{z}_k$. The initial KF method was limited to linear discrete time-controlled processes.

As mentioned in section 8.3.2.1 predictions of vehicle trajectories are a matter of probability, depending on driver input and grip potential. Theoretically, external forces that can be applied in non-collision driving situations are limited to the forces between road and tyre. Fig. 8.20(a) depicts possible locations of a vehicle during a prediction of its motion. The locations are constrained by ellipsoids, which grow in the time of the prediction. The sizes of the ellipsoids increase with a quadratic relationship of time and are related to the maximal tyre forces. These maximal forces are constrained by Kamm's circle, [Rei09, Hir09a].

In the KF approach, the uncertainties in the prediction can be visualised with the uncertainty ellipsoid concept [MT08]. The used covariance matrix, [Pij08, Pue09], can be depicted as an ellipsoid, which encloses the area where the predicted object is located, Fig. 8.20(b). The ellipsoid's centre is the estimated state, and the size of the ellipsoid is determined by the standard deviation of the estimated state in longitudinal and lateral direction. The probability P of the vehicle's location follows the Gaussian distribution around the estimated state. The application of the discrete KF for automotive collision prediction, ADAS and estimation of vehicle dynamics is mentioned several times in literature, e.g. [MPT07, LUL00, Cav07, CZZ$^+$07, WH97, VN99]. In [Pij08, Pue09] these approaches were slightly modified for the present pre-crash application. The discrete dynamical state vector $\mathbf{x}_k$ of a moving obstacle with two DoF is described by

$$\mathbf{x}_k = [x_k \; y_k \; v_{x,k} \; v_{y,k}]^T \;, \tag{8.54}$$

with $x_k,\ y_k$ the position of the vehicle's reference point[14] at time step k and $v_{x,k},\ v_{y,k}$ the velocities respectively. Quantities $x_k,\ y_k$ are obtained by discrete ERS measurement (Radar, Lidar, Video, GPS, C2C communication, ...) and $v_{x,k},\ v_{y,k}$ are measured[15] or calculated by the differential quotient $\mathbf{d}_k$,

$$\mathbf{d}_k = \begin{bmatrix} v_x \\ v_y \end{bmatrix}_k = \begin{bmatrix} (x_k - x_{k-1})/\Delta T_k \\ (y_k - y_{k-1})/\Delta T_k \end{bmatrix} \;, \tag{8.55}$$

with

$$\Delta T_k = (t_k - t_{k-1}) \;. \tag{8.56}$$

The index k denotes the discrete time-step which may be variable in time. The physical model is based on constant accelerations $a_{x,k}$ and $a_{y,k}$ between two time-steps which are again obtained by direct measurement or calculated by the differential quotient $\mathbf{u}_k$,

$$\mathbf{u}_k = \begin{bmatrix} a_x \\ a_y \end{bmatrix}_k = \begin{bmatrix} (v_{x,k} - v_{x,k-1})/\Delta T_k \\ (v_{y,k} - v_{y,k-1})/\Delta T_k \end{bmatrix} \;, \tag{8.57}$$

where $\mathbf{u}_k$ represents the control input in the Kalman filter. The constant velocity model as described in (8.43), [Jan05, MPT07], was used. For verification of the model, numerical simulation with PC-Crash software was used to generate position data of the vehicle CoG. The selected vehicle was a BMW X3 SUV, [Pij08]. The TM-Easy tyre model option was chosen, and the tyre model parameters were derived from test data. Different driving manoeuvres were simulated in order to obtain results for low, medium and high tyre forces. A grip potential between tyre and road of $\mu = 1$ was chosen to represent the worst case with the highest tyre forces and therefore the most inaccurate vehicle motion prediction.

Fig. 8.21 depicts an example of the results. The obtained vehicle CoG position is represented by the thick black line. The CoG data is sampled at 40 Hz, which was taken from technical specifications from ERS[16]. This data is fed into the algorithm, and a prediction is obtained for each new time-step. For reasons of legibility, only predictions at every 1.5 s are depicted. The predictions start at the black dots and progress with the grey line. The uncertainty ellipsoids which grow in time are also depicted. The driver input (steering, throttle and brake) is taken into account by the control term $\mathbf{u}_k$ in the Kalman filter. Regarding the measurement noise, inaccuracies of position data of ± 30 mm were chosen, again taken from the ERS specifications. After 1.5 s, the prediction is stopped, which is the considered look-ahead time. For a perfect prediction, the

[14] E.g. Centre of Gravity (CoG)
[15] Radar ERS obtain the velocity directly by application of the Doppler effect.
[16] IBEO Alasca XT laser-scanner

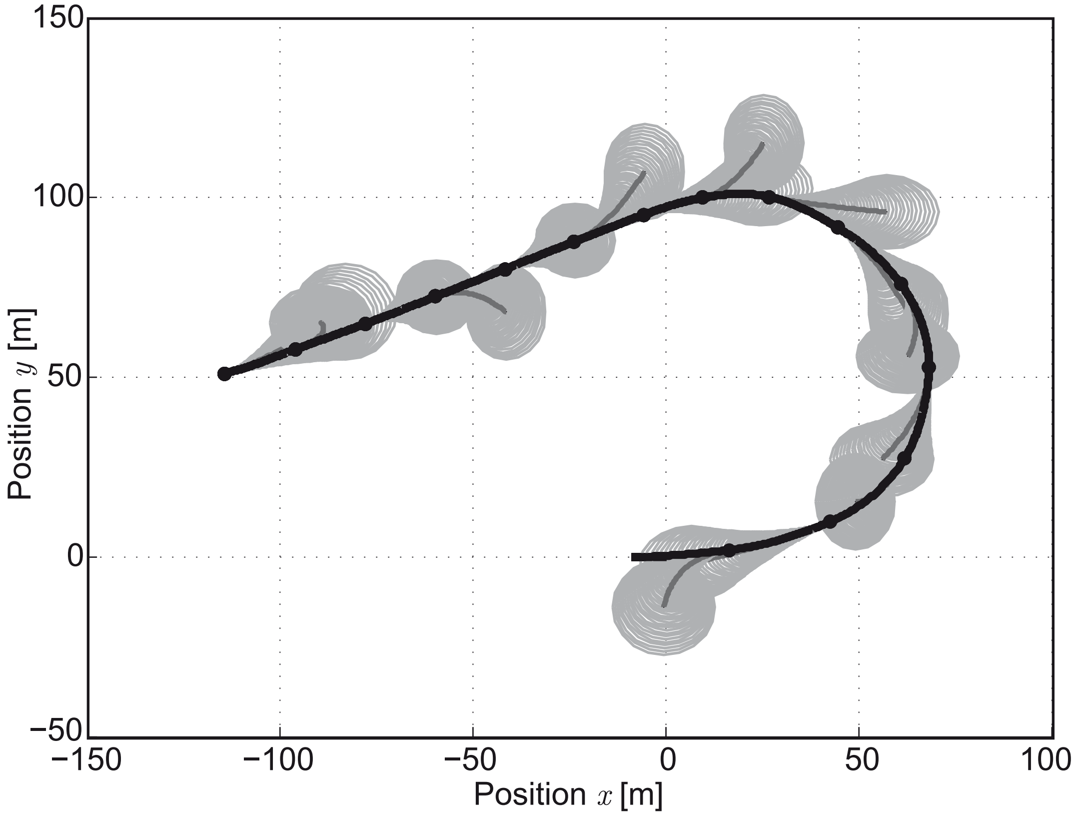


Figure 8.21.: Trajectory prediction with linear KF model, adapted from [Pij08, Pue09]
The thick black line represents the vehicle trajectory obtained by simulation. The grey lines are predictions by the linear KF model. Each prediction starts at a time step represented by the black dot. The ellipsoids depict possible locations of the vehicle, depending on driver control input.

grey and the black line would be coincident. As long as the actual trajectory stays in the area of the ellipsoids, the prediction is valid.

Fig. 8.22 is a cutout from Fig. 8.21. The actual driven trajectory does not stay within the uncertainty ellipsoids after about 1 s. The reason for this behaviour is explained by the calculation of CoG accelerations $a_{x,k},\ a_{y,k}$ as the second derivative of the discrete position data $x_k,\ y_k$ which is the measurement input in the KF. The velocities and accelerations derived from the position data according to (8.55), (8.57) are depicted in Fig. 8.23. Discrete time derivations result in noisy signals. To a certain extent, the Kalman filter "forgives" these inaccuracies. Nevertheless, the linear KF collision prediction model fails in driving situations with high vehicle accelerations at lower look-ahead times.

Fig. 8.23 depicts also the limitation of "measured" accelerations to multiples of $\pm$ 16 m/s^2. This can be explained by the fact that the position data output from

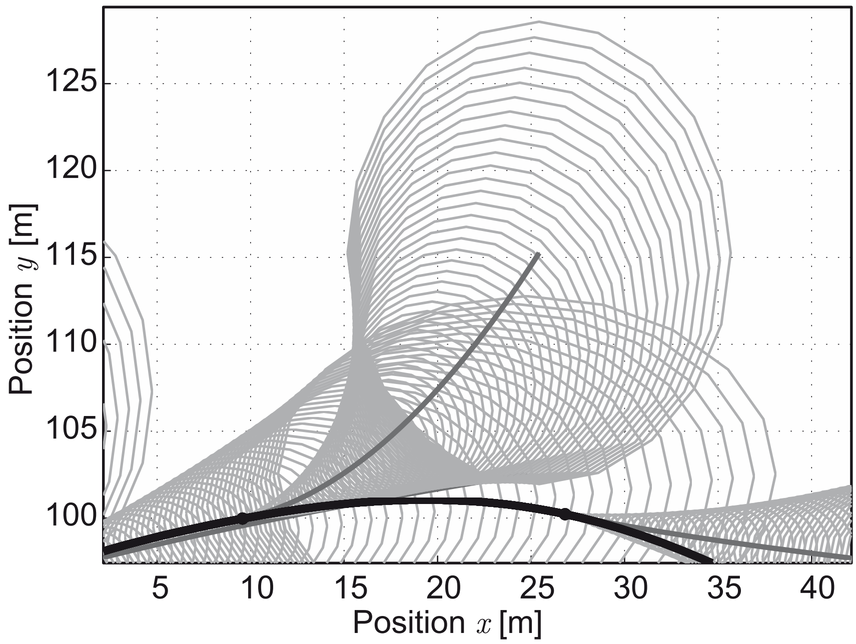


Figure 8.22.: Failure of prediction with linear KF model model
The figure is a detail from 8.21, adapted from [Pij08, Pue09]. It shows that in certain cases, the actual trajectory does not stay within the uncertainty ellipsoids.

the simulation is restricted to an accuracy of 0.01 m in the used software, and the time step for position data sampling was set to 25 ms, which equals a sampling frequency of 40 Hz. Therefore, the minimum differences in velocities and accelerations between two time steps are $0.01 \cdot 40 = 0.4$ m/s and $0.4 \cdot 40 = 16$ m/s^2, respectively.

This drawback is only relevant to a certain extent for the present application. Without further enhancements, the linear discrete KF cannot be used for reliable collision warnings and automatised avoidance manoeuvres, which would require to look ahead for some seconds. However, the results are sufficiently accurate for the reliable anticipation of collision parameters used for the prediction of the vehicle acceleration during collision, see section 8.5, since the look ahead time for this application is about 30 to 100 ms.

The quality estimation index J, see [Pij08, Pue09], is a measure of the reliability of the state estimation and, in the application for trajectory prediction, an indicator for the pre-firing of restraint systems. This index equals 1 in case of a 100% reliable prediction, which is only true for the start of the prediction at $t = 0$. The quality of the estimation using this index is illustrated by Fig. 8.24. For this example J is about 0.97 for 100 ms and still 0.90 for 200 ms, which shows the suitability of this model for the present application with a required look-ahead time of 100 ms. Further research will focus on alternatives with respect to accuracy and computing time.

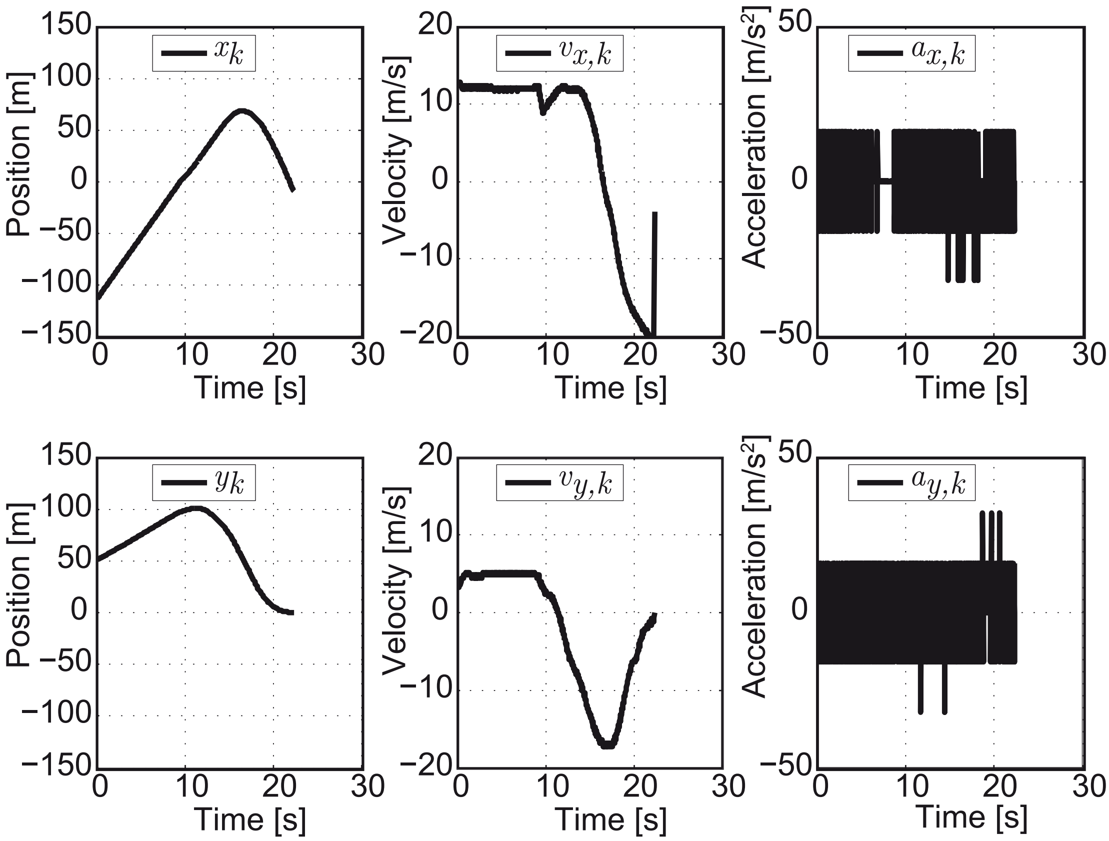


Figure 8.23.: Derivatives of position data model

Velocities and accelerations (here expressed in the global coordinate system g) are derived numerically from discrete data. The output accuracy of position data of the used software is $\pm$ 1 cm.

Currently, the suitability of further methods of collision prediction using state estimation methods and their verification with driving tests is ongoing; their description is omitted in the thesis. Another alternative, aside from Kalman filter, is the time integration of the equations of motion of a simplified model of the vehicle, which is suitable for real-time application. This is especially effective, since additional measurements are available for the ego-vehicle from the on-board measurement system of the ESC system. The model is discussed in the next section.

8.4. Vehicle model ISC-V

The vehicle model ISC-V has two functions. On the one hand, it represents the vehicle which is necessary for off-line development of the algorithms. On the other hand, it can be used for trajectory prediction, especially of the ego-vehicle. Intervening ADAS (Anti-Lock Braking System, Electronic Stability Control and Predictive Brake Assist) can be modelled as an add-on module to the ISC-V, which improves the prediction. The

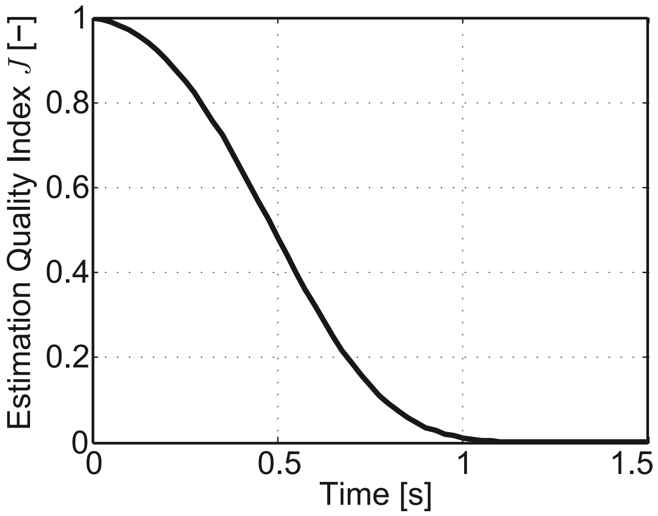


Figure 8.24.: Result for the estimation quality index J of the linear KF model, adapted from [Pij08, Pue09]

modelling of ABS, ESC and PBA is not described here and can be found in the literature, e.g. [Wal09, Web04, ETRH09]. The vehicle model receives information about the environment from the integration module at defined time-steps. As discussed in the previous sections, the most important parameter is the grip potential μ between road and tyre. Currently, $\mu = 1$ is assumed, future work will implement grip potential estimators. Since collision avoidance is not the target of ISC, a complex description of the environment with driving lanes, obstacles and others is not needed. The estimation of collision parameters in the future with an observation time less than 1 s is sufficient[17].

Additional inputs are the initial conditions for the time integration of the model, ${}_e\mathbf{x}_e(t = 0)$, (8.73). At time zero (start of a new prediction), the system time is set to zero, and the earth-fixed global coordinate system $\{\mathcal{O}_g, x_g, y_g, z_g\}$ coincides with the moving coordinate system $\{\mathcal{O}_e, x_e, y_e, z_e\}$ fixed to the ego-vehicle, $x_e(0) = 0,\ y_e(0) = 0$. Wheel speeds $\omega_{e,f(r)}$ of front (rear) axle and yaw rate $\omega_{e,z}$ are taken from the vehicle on-board sensing system, as well as the current driver control steering angle $\delta_e(0)$ and the position of brake/throttle pedal $p_e(0)$. The velocity in longitudinal direction and lateral direction can be derived from vehicle state estimation [VN99, WBBW06, CH08]. The next section describes briefly the vehicle model using a single-track model approach.

8.4.1. Single-track model

To achieve a later real-time application on the full-vehicle level, a single-track model [RS40, MW04, Hir09a] was chosen, with the following assumptions and simplifications:

[17]For pre-firing of the restraint system the look ahead time is about 50 to 100 ms

- Roll and pitch motion is omitted.
  Nevertheless, the dynamic wheel load change due to vehicle acceleration is added; therefore the height of gravity h of the vehicle has to be known.
- The grip potential between road and tyre is known, which is usually not feasible in normal traffic situations.
  For the present application, it was set to $\mu = 1.0$. For application to standard traffic situations, this represents the worst case in terms of the accuracy of the prediction. In a later state of the project a grip potential estimator will be added.
- Basic vehicle parameters are known.
  These are vehicle mass, wheelbase, location of the centre of gravity in longitudinal direction, moment of inertia around vertical axis and main tyre parameters.

The model includes 3 degrees of freedom for the position vector ${}_g\mathbf{y}$,

$$ {}_g\mathbf{y}(t) = \begin{bmatrix} x_g \\ y_g \\ \psi \end{bmatrix} , \tag{8.58} $$

in the global coordinate system $\{\mathcal{O}_g, x_g, y_g, z_g\}$ and 5 degrees of freedom for the generalised velocity vector ${}_e\mathbf{z}(t)$,

$$ {}_e\mathbf{z}(t) = {}_e\begin{bmatrix} v_x \\ v_y \\ \omega_z \\ \omega_f \\ \omega_r \end{bmatrix} , \tag{8.59} $$

in the vehicle fixed reference coordinate system $\{\mathcal{O}_e, x_e, y_e, z_e\}$. The used coordinate systems are depicted in Fig. 8.25. Coordinates x_g and y_g describe the position with respect to the global coordinate system $\mathcal{O}_g$ and ψ is the yaw angle.Velocities ${}_ev_x$, ${}_ev_y$ are the longitudinal and lateral vehicle velocities, ω_z the yaw rate. The equations of motion are formed with respect to $\mathcal{O}_e$, which is a vehicle fixed coordinate system according to ISO 8855, located in the vehicle's centre of gravity $\mathcal{CG}$. Coordinate systems $\{\mathcal{O}_f, x_f, y_f, z_f\}$ and $\{\mathcal{O}_r, x_r, y_r, z_r\}$ are vehicle fixed coordinate systems located in the front and rear axle centre. The distances between axles and $\mathcal{CG}$ are l_f and l_r respectively. Additionally $\{\mathcal{O}_f, x_f, y_f, z_f\}$ is rotated around the vertical axis by the steering angle δ_e. Air drag and wind forces are taken into account by R_D, W_x and W_y respectively; point $\mathcal{A}$ depicts the position where these forces are applied. Distances between $\mathcal{A}$ and $\mathcal{CG}$ are denoted with a_x and a_z. A climbing angle γ is introduced to consider climbing resistance. Symbols m_e and J_e are the inertial properties of the vehicle. The kinematic quantities are depicted in Fig. 8.26, and the relationship between ${}_g\dot{\mathbf{y}}(t)$ and ${}_e\mathbf{z}(t)$ is denoted in

$$ \begin{aligned} {}_g\dot{\mathbf{y}} &= \mathbf{T}_{ge}(\mathbf{y}) \cdot {}_e\mathbf{z} , \\ \text{with } \mathbf{T}_{ge} &= \begin{bmatrix} \cos\psi & -\sin\psi & 0 & 0 & 0 \\ \sin\psi & \cos\psi & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix} . \end{aligned} \tag{8.60} $$

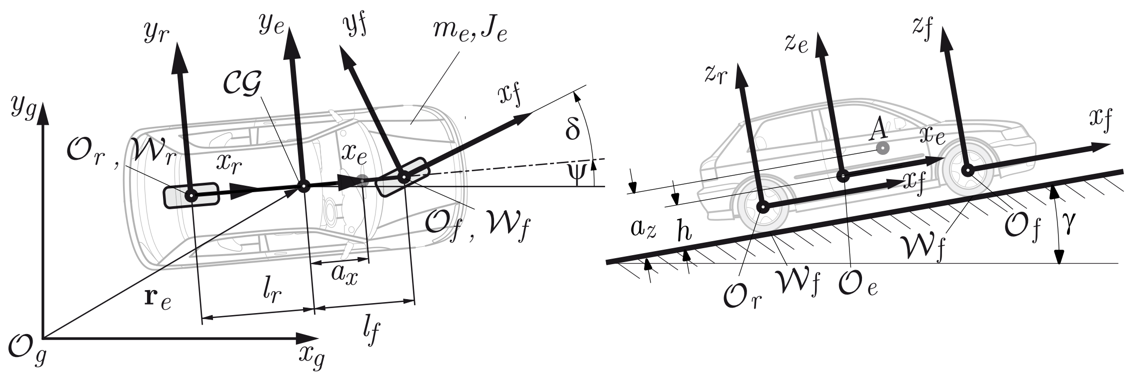


Figure 8.25.: Coordinate systems for single-track model

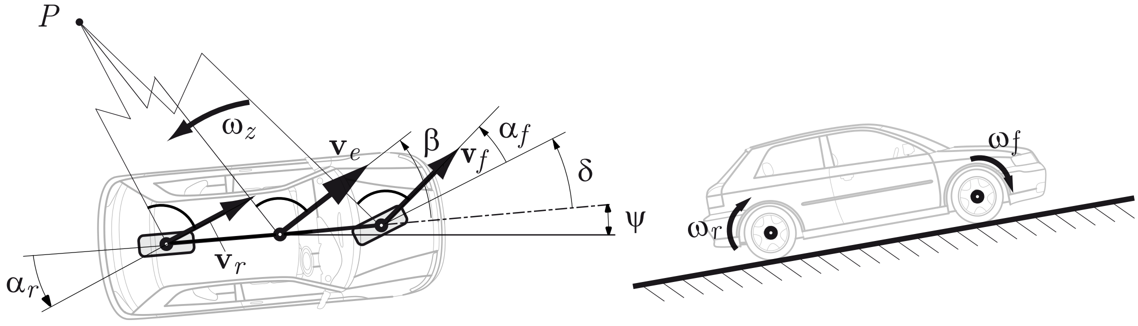


Figure 8.26.: Kinematic quantities for single-track model

Velocities $\mathbf{v}_f$ and $\mathbf{v}_r$ (consisting of $v_{x,f}$, $v_{y,f}$, $v_{x,r}$ and $v_{x,r}$) are the translational wheel velocities at the wheel points $\mathcal{W}_{f(r)}$ with respect to the coordinate systems $\{\mathcal{O}_f, x_f, y_f, z_f\}$ and $\{\mathcal{O}_r, x_r, y_r, z_r\}$, see Fig. 8.26. With the tyre longitudinal axes, $\mathbf{v}_f$ and $\mathbf{v}_r$ form the tyre slip angles α_f, α_r, and $\mathbf{v}_e$ forms the side slip angle β with the vehicle longitudinal axis x_e. Note that the normals on the velocity vectors intersect in the instantaneous centre of rotation $\mathcal{P}$. The equations of motions with respect to the vehicle reference coordinate system $\mathcal{O}_e$ read:

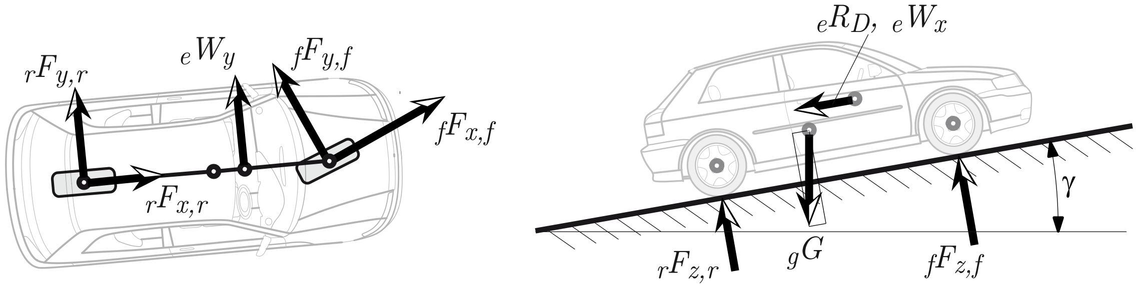


Figure 8.27.: Kinetic quantities for single-track model

$$
\begin{aligned}
&\dot{\mathbf{z}} = \mathbf{M}^{-1} \cdot (\mathbf{k} + \mathbf{q}) \\
&\text{with the} \\
&\text{mass matrix} \quad \mathbf{M} = \begin{bmatrix} m_e & 0 & 0 & 0 & 0 \\ 0 & m_e & 0 & 0 & 0 \\ 0 & 0 & J_e & 0 & 0 \\ 0 & 0 & 0 & J_f & 0 \\ 0 & 0 & 0 & 0 & J_r \end{bmatrix} , \\
&\text{the gyroscopic and centrifugal forces} \quad \mathbf{k} = \begin{bmatrix} -\omega_z\ _ev_y \\ \omega_z\ _ev_x \\ 0 \\ 0 \\ 0 \end{bmatrix} , \\
&\text{and the applied forces} \quad \mathbf{q} = \begin{bmatrix} _e\sum \mathfrak{F}_x \\ _e\sum \mathfrak{F}_y \\ _{\mathcal{O}_e}\sum \mathfrak{M}_z \\ _{\mathcal{O}_f}\sum \mathfrak{M}_{c,f} \\ _{\mathcal{O}_r}\sum \mathfrak{M}_{c,r} \end{bmatrix} .
\end{aligned}
\tag{8.61}
$$

Fig. 8.28 depicts the kinetic quantities for the front and rear axle, all with respect to the front (rear) axle centre $\mathcal{O}_{f(r)}$. $F_{x,f(r)}$ and $F_{y,f(r)}$ are wheel forces, and $V_{x,f(r)}$ and $V_{y,f(r)}$ are intersecting forces applied at the vehicle body. Moment M_D is the driving or braking torque from the engine with positive and negative values, respectively. Moments $M_{R,f(r)}$ are the rolling resistance moments of the tyres. Symbol $J_{f(r)}$ is the moment of inertia around the front and rear axle. The wheel spins with a rotational velocity of $\omega_{f(r)}$, and the dynamic tyre radius is $r_{dyn,f(r)}$. Radius $r_{dyn,f(r)}$ is defined by the circumference of the deflected tyre $U_{dyn} = 2\ r_{dyn}\ \pi$ and can be approximated by

$$
r_{dyn} = \frac{2}{3} r_0 + \frac{1}{3} r_s(t) \ , \tag{8.62}
$$

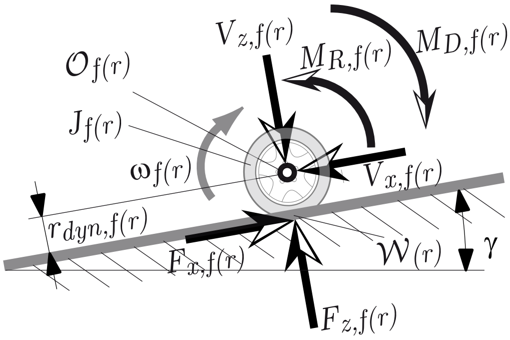


Figure 8.28.: Kinetic quantities for front (f) and rear tyre (r)

where r_0 is the unloaded tyre radius and $r_s = f(F_z, p)$ the loaded tyre, [Hir09a]. Here, F_z is the vertical tyre force, and p is the pressure in the tyre. The applied forces $\mathbf{q}$ read:

$$
\begin{aligned}
{}_e\sum \mathfrak{F}_x &= F_{x,f} + F_{x,r} - W_x - R_D - G \sin\gamma \ , \\
{}_e\sum \mathfrak{F}_y &= F_{y,f} + F_{y,r} + W_y \ , \\
{}_{\mathcal{O}_e}\sum \mathfrak{M}_z &= (F_{y,f} \cdot l_f + W_y \cdot a_x - F_{y,r} \cdot l_r)/J_e \ , \\
{}_{\mathcal{O}_f}\sum \mathfrak{M}_y &= (M_{D,f} - F_{x,f} \cdot r_{dyn,f} - M_{R,f})/J_f \ , \\
{}_{\mathcal{O}_r}\sum \mathfrak{M}_y &= (M_{D,r} - F_{x,r} \cdot r_{dyn,r} - M_{R,r})/J_r \ ,
\end{aligned}
\tag{8.63}
$$

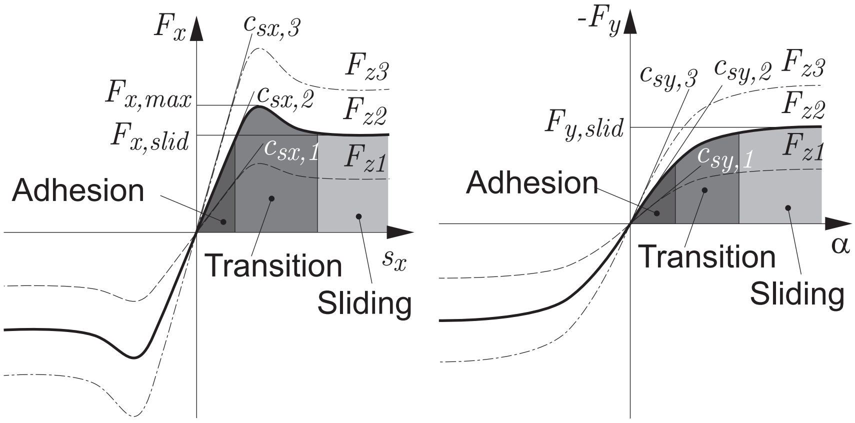
where G denotes the vehicle weight,

$$G = m_e \cdot g \ , \tag{8.64}$$

R_D is the air drag,

$$R_D = c_D A_e \rho \frac{{}^e v_x^2}{2} \ , \tag{8.65}$$

and $W_{x(y)}$ denotes an additional wind force, which can be added if measured, see Fig. 8.27 and 8.28. In (8.65) c_D is the air drag coefficient, A_e the projected frontal area of vehicle e, $\rho = f(T, p)$ the air density as a function of ambient temperature and pressure. The remaining task is the calculation of the tyre forces. For the present application two different approaches are used. The first approach is called “linear tyre model”. A linear relationship between the longitudinal tyre slip $s_{f(r)}$ and the tyre force $F_{x,f(r)}$ is assumed, in an analogous manner a linear relationship between the tyre slip angle $\alpha_{f(r)}$ and the tyre lateral force $F_{y,f(r)}$. The linear relationship is characterised by the longitudinal and lateral tyre stiffness $c_{sx,f(r)}$ and $c_{sy,f(r)}$. No combined tyre forces are taken into account.



(a) Longitudinal tyre forces (b) Lateral tyre forces

Figure 8.29.: Qualitative description of tyre forces
Longitudinal and lateral tyre forces are depicted qualitatively. Note the difference in tyre stiffness c_s with varying tyre vertical forces F_z.

The related equations read

$$\begin{aligned} F_{x,f(r)} &= c_{sx,f(r)} \cdot s_{x,f(r)} \ , \\ F_{y,f(r)} &= -c_{sy,f(r)} \cdot \alpha_{f(r)} \ , \\ s_{x,f(r)} &= \frac{r_{dyn,f(r)} \cdot \omega_{f(r)} - v_x}{r_{dyn,f(r)} \cdot \omega_{f(r)}} \ , \\ \alpha_{f(r)} &= \arctan(\frac{v_{y,f(r)}}{v_{x,f(r)}}) \ . \end{aligned} \tag{8.66}$$

This approach has the advantage that tyre model parametrisation is limited to the tyre stiffness and the dynamic tyre radius. However, it is restricted to driving manoeuvres of approximately 0.4 g lateral or longitudinal vehicle acceleration (standard traffic situation, no emergency driver reaction). Fig. 8.29 depicts qualitatively longitudinal and lateral tyre forces characteristic of a standard passenger car tyre. In the area of adhesion, a linear relation between tyre forces and the slip parameters s_x and α can be observed. The relationship is characterised by the tyre stiffness c_s, which depends on the magnitude of the tyre vertical force F_z.

In situations where the tyre forces approach the borders of Kamm's circle, these assumptions are not valid. Therefore, the second approach is to use a more complex tyre model. For the present application, which on the one hand is limited by the unknown grip potential and driver reaction, and on the other hand requires fast algorithms, the TM_simple tyre model, [Hir09b] was chosen. TM_simple is a semi-physical tyre model consisting of 17 parameters that have to be identified prior to simulation. This can be done by accommodating the parameters to standard tyre test data, which is comparatively simple because of their physical interpretation. Therefore, the non-linear tyre characteristics depicted in Fig. 8.29 are respected. In addition, combined tyre forces are taken into account through superposition of longitudinal and lateral tyre forces. The input variables for calculating the tyre forces and moments in a single contact point $\mathcal{W}_{f(r)}$ are: tyre vertical force, tyre longitudinal, lateral velocities and wheel speed, Fig. 7.4.

The tyre vertical loads $F_{z,f(r)}$ are calculated by:

$$\begin{aligned} F_{z,f} &= \frac{G(\cos\gamma \cdot l_f - \sin\gamma \cdot h) - m_e \cdot \dot{v_x} \cdot h - (W_x + R_D)a_z}{l_f + l_r} \;, \\ F_{z,r} &= \frac{G(\cos\gamma \cdot l_r + \sin\gamma \cdot h) + m_e \cdot \dot{v_x} \cdot h + (W_x + R_D)a_z}{l_f + l_r} \;. \end{aligned} \tag{8.67}$$

The rotational wheel speed is calculated by integration of the respective equation of motion, see (8.61), so only the tyre velocities $\mathbf{v}_f$ and $\mathbf{v}_r$ are remaining. Because of the rigid link between $\mathcal{O}_e$, $\mathcal{O}_f$, and $\mathcal{O}_r$, this can be achieved by transformation of the components of the CG velocity. The transformation is achieved by:

$$\begin{aligned} {}_e\mathbf{v}_f &= {}_e\,\mathbf{v}_e + \boldsymbol{\omega} \times \mathbf{r}_f = {}_e\!\begin{bmatrix} v_x \\ v_y \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \omega_z \end{bmatrix} \times \begin{bmatrix} l_f \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} v_x \\ v_y + \omega_z l_f \\ 0 \end{bmatrix} \;, \\ {}_e\mathbf{v}_r &= {}_e\,\mathbf{v}_e + \boldsymbol{\omega} \times \mathbf{r}_h = {}_e\!\begin{bmatrix} v_x \\ v_y \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \omega_z \end{bmatrix} \times \begin{bmatrix} -l_r \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} v_x \\ v_y - \omega_z l_r \\ 0 \end{bmatrix} \;. \end{aligned} \tag{8.68}$$

For the front wheel, its rotation around the steering axis with steering angle δ has to be considered, the transformation is achieved by

$${}_f\mathbf{v}_f = \mathbf{T_{ef}} \cdot {}_e\,\mathbf{v}_f = \begin{bmatrix} \cos\delta & \sin\delta & 0 \\ -\sin\delta & \cos\delta & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} v_x \\ v_y + \omega_z l_f \\ 0 \end{bmatrix} = \begin{bmatrix} \cos\delta \cdot v_x + \sin\delta(v_y + \omega_z l_f) \\ -\sin\delta \cdot v_x + \cos\delta(v_y + \omega_z l_f) \\ 0 \end{bmatrix} \;. \tag{8.69}$$

During driving, the steering angle at the wheels δ is small, therefore linearisation of the trigonometric functions with $\sin\delta \approx \delta$ and $\cos\delta \approx 1$ yields

$${}_f\mathbf{v}_f = \begin{bmatrix} v_x + \delta(v_y + \omega_z l_f) \\ -\delta \cdot v_x + (v_y + \omega_z l_f) \\ 0 \end{bmatrix} \;. \tag{8.70}$$

After calculation of tyre forces, they are scaled with the grip potential μ between road and tyre:

$$F_{\mu,x(y),f(r)} = \mu \cdot F_{x(y),f(r)} \;. \tag{8.71}$$

As explained above, $\mu = 1.0$ was chosen. Finally, the ordinary differential equations (ODE) in (8.59) and (8.61) are solved. Because of the non-linearities in the model, numerical integration is necessary, the result is the vehicle state $\mathbf{x}(t)$:

$$\mathbf{x}(t) = \begin{bmatrix} x_g & y_g & \psi_g & v_x & v_y & \omega_z & \omega_f & \omega_r \end{bmatrix}^T \;, \tag{8.72}$$

with the initial conditions $\mathbf{x}(0)$[18]

$$\mathbf{x}(0) = \begin{bmatrix} 0 & 0 & 0 & v_{x,0} & v_{y,0} & \omega_{z,0} & \omega_{f,0} & \omega_{r,0} \end{bmatrix}^T . \tag{8.73}$$

For use of the model in vehicle trajectory prediction, the values for $v_{x,0}$ and $v_{y,0}$ are derived from vehicle state estimators, [VN99, WBBW06, CH08]. Quantities $\omega_{z,0}$ and $\omega_{f(r),0}$ can be derived directly from on-board measurements.

8.5. Collision model ISC-C

Injuries in traffic accidents are caused by the following main mechanisms [Kra08]:

- Direct loading
  The injury is caused by direct contact of the injured body region to the vehicle interior, the restraint system or intruding objects (e.g. bone fracture).

- Indirect loading
  An outer body region is loaded, leading to injuries in another (inner) body region (e.g. pulmonary laceration).

- Inertial forces
  Injuries are caused by displacements of inner organs due to inertial forces (e.g. aorta rupture).

- Hyperextension and -flexion
  A motion beyond the anatomical range leads to the injury (e.g. ligament rupture).

The risk for injury is related to the severity of the loading, which is caused by:

- Deceleration of the passenger compartment[19],

- Intrusions to the passenger compartment,

- Intruding objects.

The presented approach focuses on the passenger compartment deceleration, because on the one hand intruding objects cannot be handled with state-of-the-art restraint systems, and on the other hand intrusion to the passenger compartment is only observed at high-impact energy levels in modern passenger cars. High severity accidents are not the main objective of adaptive restraint systems. The function of the collision model ISC-C is the calculation of the ego-vehicle's passenger compartment deceleration. When an unavoidable collision at time $t = t_{coll}$ is predicted by the pre-collision model, ISC-C receives the following input data:

[18] At $t = 0$, the global is set to the vehicle fixed coordinate system
[19] Passenger compartment acceleration is often denoted as vehicle "pulse".

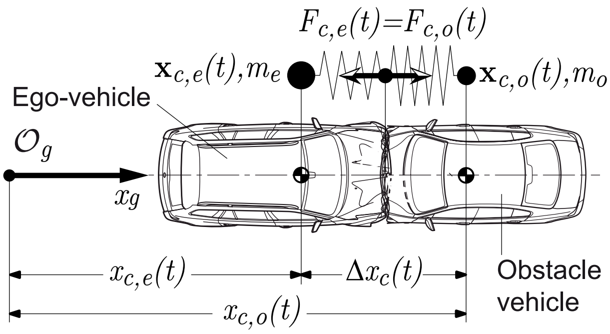


Figure 8.30.: Sketch of combined virtual force concept
The deformation characteristics are described by a virtual spring with non-linear characteristics and hysteresis. The pre-collision model provides the necessary initial conditions at collision start $\mathbf{x_e}(t_c)$ and $\mathbf{x_e}(t_c)$.

- The predicted state vector ${}_g\mathbf{xf}_{e,fcoll}$ of the ego-vehicle at collision start $t = t_{coll}$, complemented with the parameter vector $\mathbf{p}_e$ containing width length l_e, w_e, mass m_e and stiffness identifier c_e,
- The predicted state vector ${}_g\mathbf{xf}_{i,fcoll}$ of the colliding obstacle i, complemented with the parameter vector $\mathbf{p}_i$ containing length l_i, width w_i, mass m_i and stiffness c_i.

8.5.1. Calculation of the deceleration pulse of ego-vehicle

The following section describes the method for calculating the deceleration pulse of the deforming ego-vehicle in a frontal collision, [WEH09, Wal09]. In this thesis, only full overlapped frontal collisions are treated. The extension of the model for angled and partially overlapped collisions is part of future work. The collision model consists of two rigid bodes with a single degree of freedom for each in longitudinal direction ($x_{c,e}$, $x_{c,o}$), [Wal09], Fig. 8.30. They represent the ego-vehicle (mass m_e) and the obstacle (mass m_o). The bodies are linked together by a non-linear spring with hysteresis, ($F_{c,e} = F_{c,o} = F_c$). The equations of motion of this model read:

$$\begin{aligned} m_e \dot{v}_e + F_c &= 0 \; , \\ m_o \dot{v}_o + F_c &= 0 \; , \end{aligned} \tag{8.74}$$

with the spring force

$$\begin{aligned} F_c &= f(\Delta x_c) \; , \\ \Delta x_c &= x_{c,o} - x_{c,e} \; . \end{aligned} \tag{8.75}$$

The dynamic states during collision are $\mathbf{x}_{c,e} = [x_{c,e} \; v_{c,e}]^T$ and $\mathbf{x}_{c,o} = [x_{c,o} \; v_{c,o}]^T$ for ego- and obstacle vehicle. Coordinates $x_{c,e}$ and $x_{c,o}$ represent the displacements of the vehicle's CoG during the collision, $v_{c,e}$ and $v_{c,o}$ the corresponding velocities. The initial conditions $\mathbf{x}_{c,e}(0)$, and $\mathbf{x}_{c,o}(0)$ are derived from the pre-collision module evaluating the dynamic state vectors ${}_g\mathbf{xf}_{e,fcoll}$ and ${}_g\mathbf{xf}_{i,fcoll}$ at detected collision at $t = t_{coll}$.

8.5.2. Calculation of combined virtual deformation spring

Whereas equations (8.74) are comparatively simple, the calculation of the characteristics of the nonlinear spring F_c including hysteresis in the unloading phase is the key point of the collision model. The presented approach derives the deformation characteristics of obstacles from crash tests in a full overlapped collision scenario. For these purpose, data from crash tests published by NHTSA[20] [Nat10] were investigated. In [Wal09] a total of 53 different vehicles were analysed. First, the complete raw measurement data of a crash test for a specific vehicle was downloaded from the internet [Nat10]. The focus was on the US-NCAP[21] load case, which is a full frontal impact against a rigid barrier with an impact velocity of 56 kph. At least for modern cars, this load case has the advantage that it is more or less the limit where intrusions to the passenger compartment are initiated. This is also the limitation for application of the ISC-C.

8.5.2.1. Test data preparation

The test data contained several acceleration signals measured at different locations ranging from the vehicle's front to the rear. Locations in the front were excluded, since they are in zones of high deformation. These positions are exposed to high frequency oscillations, for example due to contact of the engine to other components. A measurement location relevant to the restraint system is preferably near the occupant, in a non- or low-deformation zone. These signals have to be prepared for further calculations.

In order to ensure comparable measurement results in different laboratories, data processing for crash tests has been standardised, [Tec02, Soc07]. To remove high frequency oscillations, a class of so-called SAE CFC[22] low-pass filters are prescribed. They represent Butterworth zero-phase filters and can be derived by forward/backward filtering using a second order Butterworth filter to comply with the request of zero phase shifting. The filtered acceleration $a_{CFC}(t)$ is computed by

$$a_{CFC}(t) = A_0 \cdot a_{raw}(t) + A_1 \cdot a_{raw}(t-1) + A_2 \cdot a_{raw}(t-2) + B_1 \cdot a_{CFC}(t-1) + B_2 \cdot a_{CFC}(t-2) \ , \tag{8.76}$$

where $a_{raw}(t)$ is the unfiltered time-discrete acceleration data and A_0, A_1, A_2, B_1, B_2 are filter parameters which read:

$$\begin{aligned} A_0 &= \frac{\omega_a^2}{1+\sqrt{2}\omega_a+\omega_a^2} \ , \qquad A_1 = 2a_0 \ , \qquad A_2 = a_0 \ , \\ B_1 &= \frac{-2(\omega_a^2-1)}{1+\sqrt{2}\omega_a+\omega_a^2} \ , \qquad B_2 = \frac{-1+\sqrt{2}\omega_a-\omega_a^2}{1+\sqrt{2}\omega_a+\omega_a^2} \ . \end{aligned} \tag{8.77}$$

Variable ω_a is computed by

$$\omega_a = \frac{sin\omega_d \cdot \frac{\Delta T}{2}}{cos\omega_d \cdot \frac{\Delta T}{2}} \ , \tag{8.78}$$

[20] National Highway Traffic Safety Administration
[21] United States-New Car Assessment Program, crash test program for consumer information
[22] Society of Automotive Engineers Channel Filter Class

Filter type	**Filter parameter**	
CFC 60	3 dB- Limit frequency	100 Hz
	Stop damping	-30 dB
	Sample rate	at least 600 Hz
CFC 180	3 dB- Limit frequency	300 Hz
	Stop damping	-30 dB
	Sample rate	at least 1800 Hz
CFC 600	3 dB- Limit frequency	1000 Hz
	Stop damping	-40 dB
	Sample rate	at least $60,000$ Hz
CFC 1000	3 dB- Limit frequency	1650 Hz
	Stop damping	-40 dB
	Sample rate	at least $10,000$ Hz

Table 8.3.: CFC filter parameters [CdVO+08]

where ΔT is the sampling rate, which was used during the measurement of the crash test. The limit frequency $f_d = \frac{\omega_d}{2\pi}$ is defined slightly differently in SAE [Soc07] and ISO [Tec02] standards and reads:

$$\begin{aligned} &\omega_{d,SAE} = 2\pi \cdot CFC \cdot 2.0775 \\ &\text{or} \\ &\omega_{d,ISO} = 2\pi \cdot CFC \cdot 2.0833\ . \end{aligned} \tag{8.79}$$

The filter parameters A_0, A_1, A_2, B_1, B_2 depend on the chosen CFC class and ΔT, see Tab. 8.3. For vehicle acceleration data, the mentioned standards prescribe filter classes CFC 180 (preferably) or CFC 60.

An example of data preparation is depicted in Fig. 8.31, [Wal09]. The raw data, acceleration signals from a single measurement position usually sampled at 10 kHz to 20 kHz, are depicted pointwise as small grey stars. The filtered signals according to CFC 60 and CFC 180 are depicted as thin lines. The mean values (thick lines) represent the averaged curves of several different acceleration signals. In test data of [Nat10], they are usually located at the vehicle sill, floor-pan or rear-cross-member.

This mean acceleration data $a_{c,mean}(t)$ reads

$$a_{c,mean}(t) = \frac{\sum\limits_{l=1}^{L} a_{c,l}(t)}{L}\ , \tag{8.80}$$

where $a_{c,l}(t)$ are L measurements from those different locations described above. The reason for calculating the mean value is to improve the overall reliability of the acceleration signals and to avoid numerical artefacts from single peaks. The selected acceleration-time

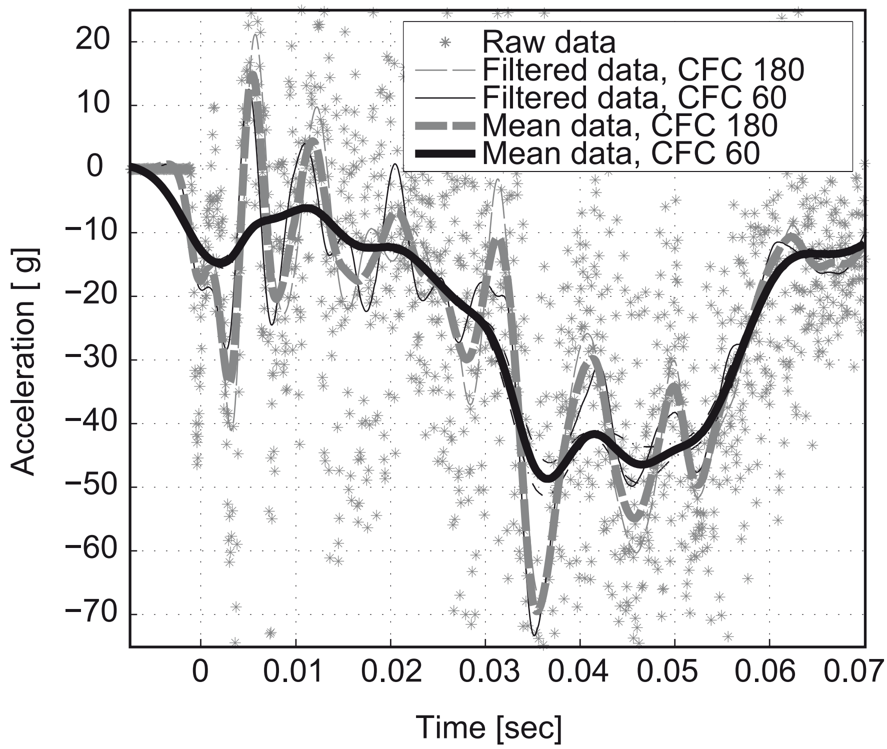


Figure 8.31.: Example of acceleration data processing, [Wal09]
Raw and filtered signals according to CFC 60 and CFC 180 from a single measurement point are depicted. The mean values correspond to the average of several different acceleration signals, usually taken from the positions "sill", "floorpan", "rear cross member", [Nat10].

signals $a_{c,l}(t)$ were checked for plausibility. This was done by comparing the time-velocity histories derived from acceleration signals of several measurement locations. The filtered data forms the basis for further analysis. For assessing the displacements $s_c(t)$ due to deformation, the acceleration time signal $a_{c,mean}(t)$ was integrated twice:

$$\begin{aligned} v_c(t) &= \int\limits_{t_{coll}}^{t_e} a_{c,mean}(t)dt \ , \\ s_c(t) &= \int\limits_{t_{coll}}^{t_e} v_c(t)dt \ , \end{aligned} \tag{8.81}$$

with the following initial conditions:

$t_{coll} = 0$ $[s]$	Collision start
$a_c(t_0) = 0$ $[m/s^2]$	Correction of filter inaccuracy
$v_c(t_0) = v_0$ $[m/s]$	Collision velocity from test report
$s_c(t_0) = 0$ $[m]$	No initial deformation

Hence, for a specific vehicle i, the acceleration $a_{c,i}(t)$ and deformation time histories $s_{c,i}(t)$ are derived. The next step is to calculate the deformation force $F_{c,i}(t)$,

$$F_{c,i}(t) = m_i \cdot a_{c,i}(t) \ , \tag{8.82}$$

and finally combine $F_{c,i}(t)$ and $s_i(t)$ to the desired deformation characteristics $F_{c,i}(s_i)$.

8.5.2.2. Combination of deformation springs

Assuming straight full overlapped impacts and knowledge about mass and deformation characteristics of ego-vehicle and obstacle by environment recognising systems, the deceleration pulse of the ego-vehicle can be derived according to (8.74), if the combined deformation characteristic F_c is determined. A simple method for this would be a connection in series of linear springs with the spring rates c_e and c_o. The resulting combined linear spring rate c then reads:

$$\frac{1}{c} = \frac{1}{c_e} + \frac{1}{c_o} \ . \tag{8.83}$$

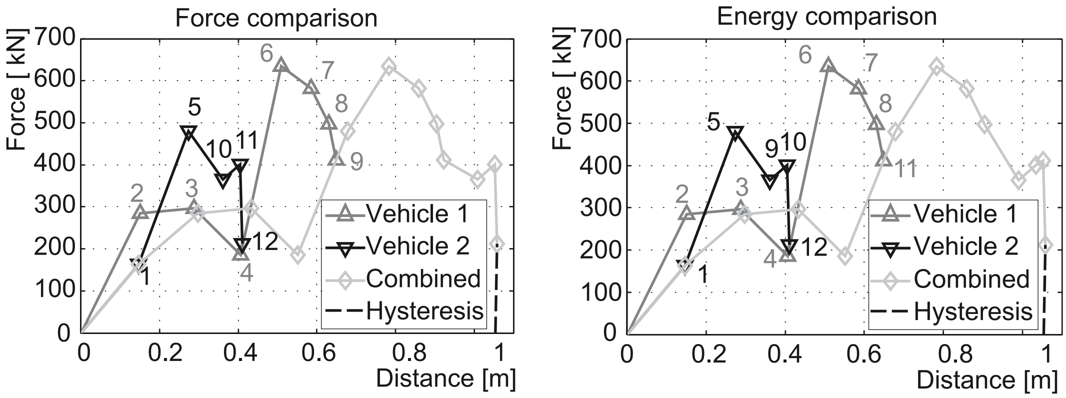
Because of the nonlinear characteristics of $c_e, c_o \neq const.$, this linear approach is not applicable. Therefore, the individual stiffnesses of the colliding vehicles were combined pointwise to a virtual deformation spring F_c. A detailed description of the pointwise numerical calculation of F_c with two different approaches can be found in [Wal09]. Fig. A.5 in the appendix depicts the flow chart of these approaches. The first calculation method, Fig. A.5(a), assumes that the vehicle with the lower force level at a specific deformation deforms until the force level of the other vehicle is exceeded. Next, the second vehicle deforms until the level of the first is reached, and so on. This continues until the end of the force deflection curves at $s = s_f$ is reached. The further deformation behaviour is characterised by deformation of the passenger compartment. At this level, ISC-C and its designation for adaption of restraint systems has reached its range of application. In reality, the restraint system would be fired at its full capacity.

The second method is analogous to the first, but instead of the force levels, the deformation energies $E_{def,e}$, $E_{def,o}$ of ego- and obstacle vehicle defined by

$$\begin{aligned} E_{def,e} &= F_{c,e} \cdot \Delta s_e \ , \\ E_{def,o} &= F_{c,o} \cdot \Delta s_o \ , \end{aligned} \tag{8.84}$$

are compared. The background is that with different discrete deformation steps Δs_e and Δs_o, the deformation energy taken by the first vehicle could be greater ($E_{def,e} > E_{def,o}$) even when the force level is lower ($F_{c,e} < F_{c,o}$). Fig. A.5(b) shows the flow chart of the second F_c calculation method.

Fig. 8.32 depicts a qualitative result of the combined spring generation. Although both methods show approximately the same result, the more numerically stable solution is the force-based method on the left side of the figure [Wal09].



(a) Comparison of force levels

(b) Comparison of energy levels

Figure 8.32.: Qualitative result of generation of combined spring F_c

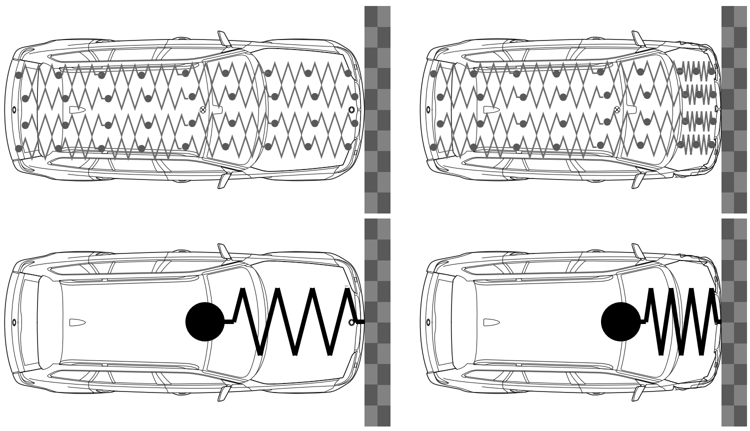


Figure 8.33.: Comparison of single mass and multi mass model
The continuous vehicle is understood as a multi mass model having many degrees of freedom. A simple single mass model serves for idealiation which can reveal the first order deformation dynamics.

A shortcoming of the presented method is the concentration of the mass into the CoG of the vehicle. In [Wal09] it was pointed out that, in reality, the vehicle is a continuum with non-discrete distributed mass. Mass which has been decelerated in the front of the vehicle will contribute less to the total internal deformation energy compared to the single mass point approach, see Fig. 8.33.

This drawback was investigated by comparing the computed time histories of force $F_{c,v_i}(t)$ of 21 vehicles ($v_i,\ i = 1\ to\ 21$) with corresponding measurements of rigid barrier loads $F_{c,b_i}(t)$. For these vehicles, plausible measurements of rigid crash barriers instrumented with load cells were available at [Nat10]. The deformation energies of both measurements, vehicle and barrier, were calculated by

$$E_{v_i} = \int\limits_{s_0}^{s_f} F_{c,v_i}(s_{v_i}) ds_{v_i} \ , \qquad E_{b_i} = \int\limits_{s_0}^{s_f} F_{c,b_i}(s_{v_i}) ds_{v_i}, \tag{8.85}$$

where $s_0 = 0$ and s_f denote initial and final vehicle deformation. Next the impulses,

$$I_{v_i} = \int\limits_{t_0}^{t_f} F_{c,v_i}(t) dt \ , \qquad I_{b_i} = \int\limits_{t_0}^{t_f} F_{c,b_i}(t) dt, \tag{8.86}$$

for both measurements were calculated analogously, where $t_0 = 0$ and t_f are the moments of time at the beginning and the end of the collision. Finally, for all 21 vehicles i, E_{v_i} was compared to E_{b_i} as well as I_{v_i} to I_{b_i} by

$$E_{v_i,rel} = \frac{E_{v_i}}{E_{b_i}} \cdot 100[\%] \ , \qquad I_{v_i,rel} = \frac{I_{v_i}}{I_{b_i}} \cdot 100[\%] \ . \tag{8.87}$$

The result was,

- The error is comparatively low:
  the average value $\bar{m}$ of $E_{v_i,rel}$ was 94.2% (7.3% standard deviation σ). For $I_{v_i,rel}$ the result was $\bar{m}$=100.6% (5.6% σ), Fig. 8.34.
- There was no real global trend,
  for some vehicles, the vehicle measurement was higher than the barrier measurement and vice versa.

In the following, the influence of the decelerated mass in the vehicle front is neglected. Nevertheless, if measurements of vehicle and barrier are available and plausible, it is recommended to scale the vehicle force deflection curve $F_{c,v}(s)$ to the barrier measurement $F_{c,b}(s)$ by using the correction factor *corr* calculated with the total deformation energies:

$$F_{c,corr}(s) = corr \cdot F_{c,v}(s) \ , \qquad corr = \frac{\int_{s_0}^{s_f} F_{c,v}(s) ds}{\int_{s_0}^{s_f} F_{c,b}(s) ds} \ . \tag{8.88}$$

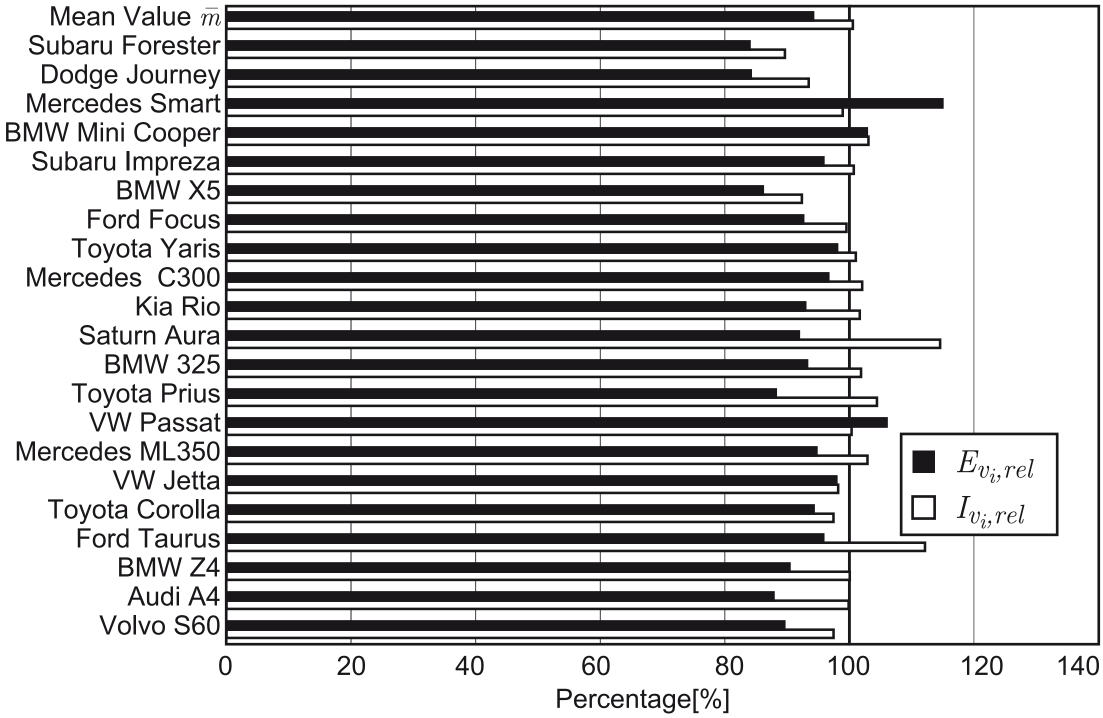


Figure 8.34.: Comparison of impulse and energy between vehicle and rigid barrier measurements

The figure compares the relative energy $E_{v_i,rel}$ and the relative impulse $I_{v_i,rel}$ between rigid barrier and vehicle measurements. Basis of the comparison (100%) is E_{b_i} and I_{b_i}.

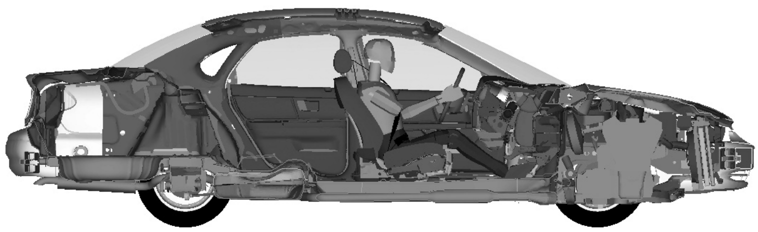


Figure 8.35.: Numerical FEM Model of Ford Taurus, [Nat06]

8.5.3. Verification

The verification of the presented model was carried out by a detailed off-line simulation, [WEHC09]. This off-line simulation was based on a Finite-Element-Method (FEM) model of a full frontal car to car crash. A FEM model of the Ford Taurus [Nat06] was used for simulation of the crush behaviour of ego- and opponent vehicle, Fig. 8.35. The vehicle structure was reinforced in order to withstand the US-NCAP frontal impact with minimal intrusions to the passenger compartment.

The procedure for deriving the combined virtual deformation spring was the same as described above for experimental crash tests. First, an US-NCAP crash test was simulated and the force deflection curves $F_{c,e}(s_e)$ and $F_{c,o}(s_o)$[23] was processed based on the vehicle pulse measured at the location of the Airbag ECU. The same data preparation with respect to data filtering as in the experimental tests was used. The combined virtual deformation spring $F_c(s)$ was processed and the ego-vehicle's deceleration of the passenger compartment was calculated with the ISC-C model. Finally the result was compared to the deceleration measured in several FEM car-to-car frontal impact simulations. Some sample results are depicted in Fig. 8.36.

Fig. 8.36(a) compares the predicted accelerations in the US-NCAP load case. The acceleration of the passenger compartment predicted by the collision model (solid line) follows the acceleration calculated by the FEM model (dashed line) with deviations at peaks with higher frequencies. The time velocity history depicted in Fig. 8.36(b), which is relevant for the design of the restraint system, is accurately predicted with minor deviations in the rebound phase of the collision due to the chosen hysteresis model. The same holds true for the car-to-car collision using two modified Ford Taurus FEM models in full frontal impact. Figs. 8.36(c) and (d) illustrate the results with a collision velocity of 31 kph for each vehicle. Once again, the prediction of the ego-vehicle is sufficiently accurate for control of an adaptive restraint system.

8.5.4. Outlook for collision model

The ISC-C model was verified only with the modified Ford Taurus FEM model at different impact severities. Though the principal validity of the approach was shown, future work will deal with a verification matrix including different vehicles, impact severities, variations in impact angle and collision overlap. In additional improvements in the prediction of higher frequency peaks will be investigated.

8.6. Occupant Model ISC-O

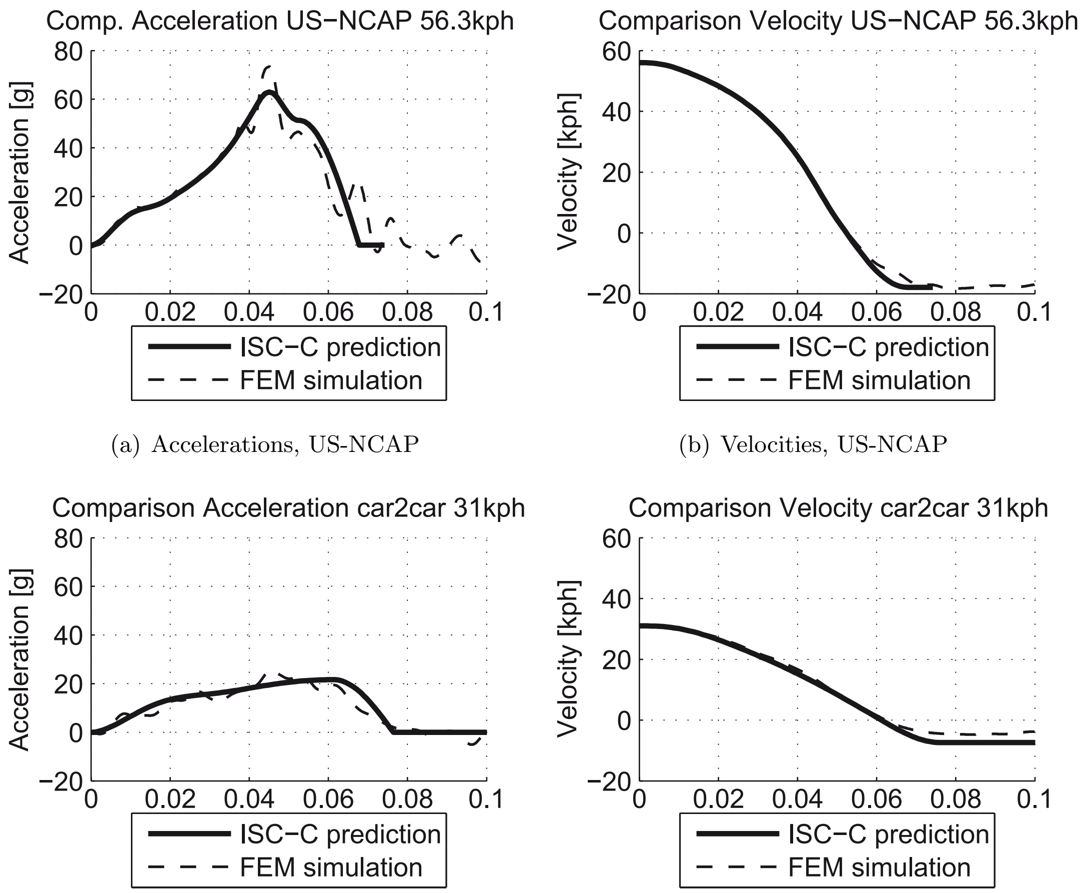
The function of the occupant model is the calculation of the injury risk $\mathbf{ic}_s$ of the s-th considered occupant and the appropriate setting $\mathbf{r}_s$ of the restraint system, consisting of Time-to-Fire (TTF) and force levels of frontal airbag and seatbelt. It receives the following input data:

- The passenger compartment deceleration pulse ${}_e\mathbf{a}_e$
- Occupant parameters ${}_e\mathbf{x}_{O,s}$(mass and position of the sth occupant)

The returned output $\mathbf{r}_s$ to the integration module is:

- Optimised (pre-)fire times of the restraint system,
- Optimised force levels of an assumed adaptive restraint system.

[23] In this case, they are equal.



(a) Accelerations, US-NCAP

(b) Velocities, US-NCAP

(c) Accelerations, car-to-car collision

(d) Velocities, car-to-car collision

Figure 8.36.: Verification of ISC-C acceleration prediction, [WEHC09]
The figure compares accelerations and velocities predicted by the collision model ISC-C and the actual behaviour derived by a FEM simulation (modified Ford Taurus). Two load cases, US-NCAP and car-to-car collision at 31 kph, are depicted. The prediction is sufficiently accurate for the control of an adaptive restraint system.

8.6.1. State-of-the-art in adaptive restraint systems

Adaptation of frontal airbag systems has been a field of research for many years, e.g. [CHKM01, PBR06, AWH97, GZS04, PBR05, Mac94, Wis03]. By statistical analysis of accidents, [CHKM01] presented a reduction of MAIS 2+[24] injuries in Europe by 33 to 41% and a 14 to 25% reduction for MAIS 3+. In a similar study, [GZBS05] showed the effectiveness of adaptive restraints of 25.7% for MAIS 3+ injuries for Germany.

Theoretically, a constant force restraint system (CFR) leads to the lowest injury risk for

[24]MAIS is the most severe injury of an occupant coded by AIS (Abbreviated Injury Scale), [GW08, GW06]. MAIS 2+ include all injures except minor ones.

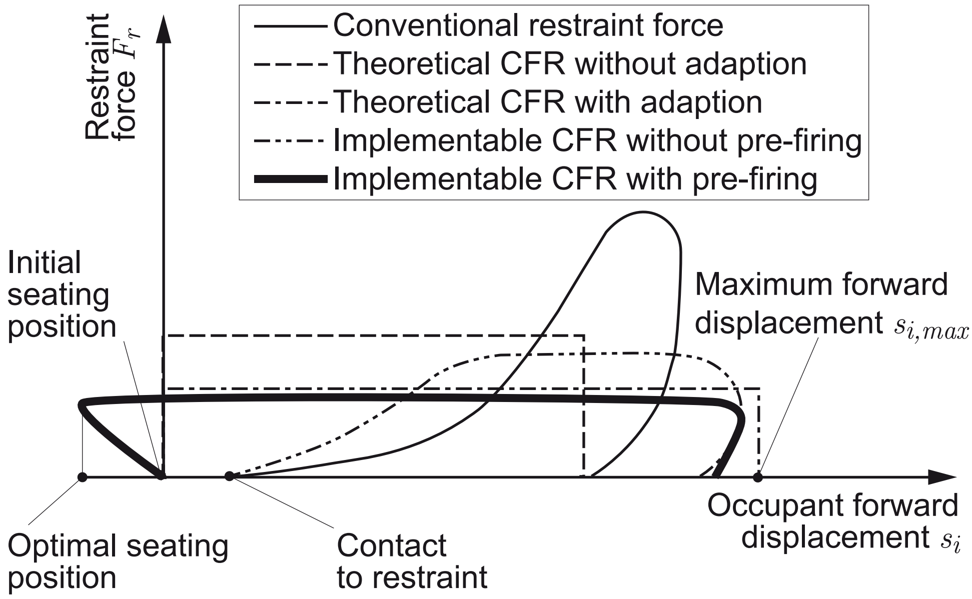


Figure 8.37.: Qualitative illustration of restraint forces, adapted from [Ste09b]

occupants if the force is set to a level such that the available survival space is fully utilised for deceleration of the occupant, [Mil96]. Fig. 8.37 shows a qualitative illustration of restraint forces in a frontal impact [Ste09b]. The solid thin line is a restraint force F_r of a conventional restraint system. Because of the required time for activation of the system[25], the gradient of F_r at contact between restraint system and occupant is comparatively low. Since state-of-the-art restraint systems do not consider the available survival space, the maximum forward displacement $s_{i,max}$ within the passenger compartment is not used. A theoretical CFR is depicted as a dashed line. The occupant is immediately restrained at a constant force level, therefore limiting the restraint forces. The dashed and dotted lines depict CFRs utilising $s_{i,max}$. The first one shows the theoretical best solution, and the second shows what is implementable in a practical application without pre-firing of the restraint system. The solid thick line represents a CFR with pre-firing, the occupant is positioned (backwards) into a optimal position w.r.t. to a longer displacement by the pre-fired restraint system. In Fig. 8.37 this backward positioning is depicted with negative values for s_i. At this restraint performance, the presented approach is aiming at. As described in section 8.2, in principle two main approaches for adapting occupant restraint systems are feasible:

- **Feed forward control (open-loop)**
  The accident scenario is assessed by an appropriate method[26] and the restraint system is activated in order to minimise the injury risk. The injury risk is not fed

[25] Examples are the time to detect the collision (TTF) and the airbag deployment time.
[26] In the present approach, the acceleration pulse of the passenger compartment is assessed by the ISC-C model.

back to the restraint system.

- **Feedback control (closed-loop)** Again, the accident scenario is assessed, but the injury risk is continuously fed back to the restraint system, which is controlled in a manner that continuously minimises the injury risk.

8.6.1.1. Feed forward control

Feed forward control of restraint systems has been state-of-the-art since the introduction of reliable airbag systems by Daimler in 1980. The airbag control unit determines whether the collision severity is high enough to activate the restraint system or not. Restraint forces are chosen with the intent of reducing the overall injury risk in a certain crash test configuration. Typical examples are the FMVSS 208 (belted and unbelted), US-NCAP and EURO-NCAP frontal crash tests. Dual stage airbags, introduced in 1998, have made the next step to adaptive restraints by providing different activation levels. Information on crash severity and the use of the seatbelt are input to determine the restraint force level. In the future, occupant mass and position are additional parameters to pre-set the restraint system, as well as pre-crash information to activate restraint systems before contact of the collision vehicles. Nevertheless, the issues to be solved are situation-based reliable algorithms to determine ideal fire-times and force levels.

In [PBR06] a simulation model of a 50-percentile male human consisting of three rigid bodies was presented. A reduction of pelvis, chest, and head accelerations by 56, 62, and 63% respectively in a US-NCAP frontal crash load case was reported using a constant force restraint system. In [SH09] benefits of adaptive restraints in the updated US-NCAP requirements were analysed. A significant benefit was found, particularly on the passenger side. In [GI01], the MADYMO[27] model of a 5th, 50th and 95th percentile Hybrid-III crash test dummy was enhanced with capabilities to output additional forces which make it possible to assess the effect of restraint forces on the dummy (injury criteria) and thereby optimise the level of the restraint force.

8.6.1.2. Feedback control

Feedback control of restraint systems was investigated by [Hes04]. Deflection and acceleration of the thorax, as well as head acceleration, were chosen as output variables. The actively controlled variables were airbag vent-hole size and belt-load-limiter force. Due to the lack of hardware for actively controlled restraint components, numerical simulation was used to design a suitable feedback control. The effectiveness of the feedback control was demonstrated in a US-NCAP load-case, which showed a reduction of the injury risk of the chest by 60% by active control of the belt force and of the head of 50% by control of the airbag vent-hole.

[27] Occupant simulation software package based on multi-body-dynamics

However, the presented approach is based on off-line simulation and not feasible for real-time application in a vehicle. In [vdL09] an approach for a Continuous Restraint Control (CRC) of a seat belt was presented. Sensors and actuators were added to the belt system to design a feedback control. The seat belt force was controlled by a semi-active actuator based on a pressure controlled valve. An injury estimator designed for real-time performance was presented. It was a two dimensional multi-body dynamical occupant model representing a Hybrid III dummy. A numerical controller was introduced to calculate the optimal belt force based on the estimated injury responses.

8.6.2. Single mass point approach

The following section presents an approach for a real-time feed forward restraint control by a simple single mass point [WEH09, WEHC09]. It assumes that the injury risk in a vehicle accident is reduced when mean a_{mean} and maximum accelerations a_{max} of the occupant are reduced. This is especially true in low to medium crash severities, where the integrity of the passenger compartment prevents intrusion-induced injuries. Modern cars are designed to withstand collision severities in frontal crash of up to 56 kph against a rigid barrier or 64 kph against a deformable honeycomb barrier with minimal intrusion into the foot-well.

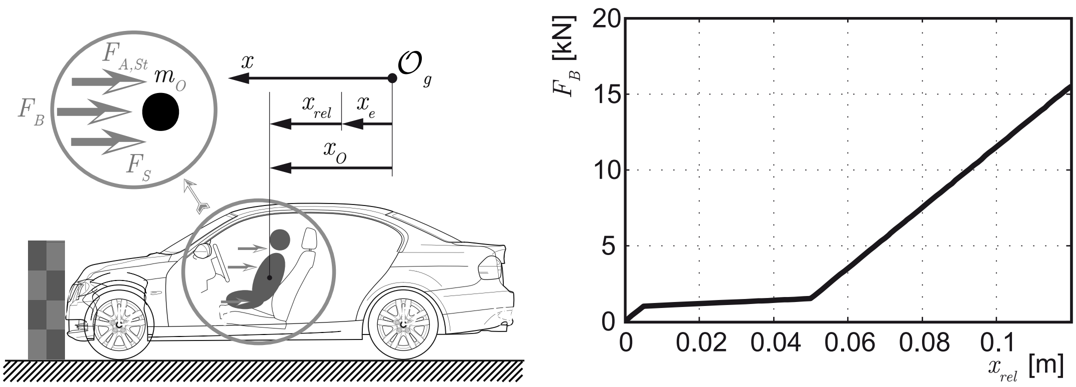
The single mass point model takes into account mass and seating position of the occupant. In a later vehicle application, these parameters have to be measured by occupant sensing systems. The simple equation of motions for the occupant model read:

$$\begin{aligned} m_O \cdot \ddot{x}_O(t) &= F_{A,St}(t) + F_B(t) + F_S(t) \ , \\ x_{rel}(t) &= x_O(t) - x_e(t) \ , \end{aligned} \tag{8.89}$$

where m_O denotes the occupant mass, $\ddot{x}_O(t)$ the absolute occupant acceleration, $F_{A,St}(t)$ the combined restraint force of steering column, steering wheel and airbag, $F_B(t)$ the belt force and $F_S(t)$ the restraint force of the seat, Fig. 8.38(a). The occupant position x_O is calculated using the relative displacement of the occupant inside the vehicle x_{rel} and is thereby connected to the ISC-C collision model and its dynamic state $x_e(t)$. The choice of this approach was driven by the real-time application of the complete ISC algorithm. Due to the modular system architecture, a more sophisticated occupant model similar to approaches described in [vdL09] can be implemented in a later stage.

The restraint forces are modelled with typical characteristics of the actual components, in order to consider time delays by triggering and activation. An example for the belt is depicted in Fig. 8.38(b). For the first 50 mm, the belt slack is considered. Thereafter, the force deflection characteristics of the belt band are considered. The belt pretensioner has separate characteristics.

Gradient based and genetic optimisation algorithms implemented in ISC-O [Wal09] determine suitable force levels and trigger times of the adaptive restraint system, based on the criterion of minimising the maximum a_{max} and mean acceleration a_{mean} of the



(a) Scheme of occupant model, [Wal09] (b) Belt characteristics, [Wal09]

Figure 8.38.: Occupant model ISC-O

occupant (represented by the single rigid body m_O). A secondary condition is the avoidance of vehicle interior contact by limiting the relative forward motion of the occupant rigid body inside the vehicle, $x_{rel} < x_{rel,max}$. The optimisation of the force levels and activation times of the restraint system will be the key issue for a real-time vehicle application, since the necessary processing time is far too high to be used in a real-time application. At the moment, putting the results of a several optimisation runs into a database (look-up table) is the most promising solution for real-time performance.

The potential of the single mass point open loop control was demonstrated by comparing the results of a standard restraint system optimised for FMVSS 208 requirements with a restraint system optimised according to the presented approach, [Wal09]. Three different frontal impact velocities (20, 40 and 54 kph) and five different occupant masses ranging from 30 to 125 kg were investigated. The occupant position parameter was not varied, but rather set to a standard seating position. The results are listed in Tab. 8.4. The result for the standard restraint system forms the basis for the comparison. It can be seen that a significant reduction is achieved in every load case. Obviously, the smallest improvements are in load cases near the US-NCAP with a 50th percentile occupant, see “54 kph”, “75 kg” in Tab. 8.4. An exception is the load case “125 kg” occupant in a 20 kph frontal crash, where the standard system is the optimum.

8.7. Outlook of the ISC-approach

Chapter 8 described an innovative approach for the integration of primary and secondary traffic safety systems. This approach aimed at decreasing injury risk in low to mid

[28] In Tab. 8.4, for results marked with *, no values for comparison for the standard restraint were available due to bottoming-out of the restraint system (head contact to vehicle interior). Therefore, the comparison was made with values for a 75 kg occupant at the same impact velocity.

Benefit of ISC				
	m_O	Increase of s_O[%]	Reduction of a_{max}[%]	Reduction of a_{mean}[%]
20 km/h	30 kg	1105%	−87%	−61%
	50 kg	718%	−79%	−61%
	75 kg	410%	−78%	−61%
	100 kg	268%	−71%	−55%
	125 kg	0%	0%	0%
40 km/h	30 kg	658%	−71%	−36%
	50 kg	380%	−61%	−38%
	75 kg	186%	−55%	−37%
	100 kg	50%	−45%	−27%
	125 kg	−1%	−32%	−14%
54 km/h	30 kg	451%	−63%	−27%
	50 kg	163%	−57%	−26%
	75 kg	0%	−33%	−15%
	100 kg	[28]*4%	*−37%	*−17%
	125 kg	*6%	*−38%	*−18%

Table 8.4.: Reduction of mean and maximum occupant acceleration, a_{max} and a_{max}, with an adaptive restraint system controlled by ISC-O

crash severities using a CFR restraint system, which adapted to the impact scenario and the occupants. Further research will deal with advancement of the algorithms to real-time performance and full-vehicle application. The pre-collision model ISC-P will be optimised for larger accurate look-ahead times. The vehicle model ISC-V will be prepared with intervening ADAS which further decrease collision severity. The collision model ISC-C will be enhanced for other load cases apart from the straight full overlapped frontal impact. Finally, the occupant model ISC-O will be detailed and feedback control of the restraint system will be investigated. A further topic of research is the benefit of the ISC approach using posteriori benefit evaluation. However, an accident database like GIDAS containing also injury accidents will be necessary. Results of the literature suggest a potential of 25% reduction in MAIS 3+ injuries, but reduction of collision severity by ADAS interventions are not covered by these studies.

9. Summary

This thesis summarised the author's contributions to scientific research related to the receiving of the venia docendi in *Automotive Engineering*.

Part I began with a description of the scientific subject of automotive engineering, which must continuously confront the high complexity of the *road vehicle* as an industrial product. This complexity was attributed to the diversity of the competing requirements, which have evolved due to demands from customers, legal and consumer test requirements, and in-house manufacturer requirements. To place this in context, the subsystems of road vehicles were described, and a summary of automotive history showed the evolution of road vehicles and the outlook for future trends. Future progress will depend on the ability to accomplish key objectives, such as increasing energy efficiency and decreasing emissions, as well as finding ways to meet the demands for greater transport capacity and enhanced traffic safety. These challenges must be met, since mobility is a basic human need, upon which future wealth and economic growth depend.

In answer to these demands, a multi-disciplinary practice has evolved, which depends on contributions from experts in many different scientific fields. The breadth of the scientific subject is evident from the numerous universities, colleges, universities of applied science, research departments of manufacturers and suppliers and non-academic research centres around the world that deal with automotive engineering. As an example of the scientific research and education being carried out today, Part I provided an overview of the related research, with a focus on German-speaking universities. At such institutions, research-based education is performed either by complete coverage of the subject, specialisation, or embedding in complementary courses. In this context, Part I also provided an overview of the author's contributions to this field, including teaching activities, research projects and scientific contributions.

Part II, the habilitation thesis, provided a deeper insight into the author's scientific focus, which is *traffic safety*. Chapter 5 divided the goals of traffic safety measures into three categories: avoiding collisions or mitigating collision severity (primary or active safety); mitigating the consequences of collisions during impact (secondary or passive safety); and post-crash treatment (tertiary safety). A holistic view of vehicle safety must deal with all of the elements involved: the human being, the vehicle and the environment. A current trend in traffic safety is to integrate these measures, which had traditionally been developed separately, in order to enhance their protective function. This thesis defined this trend as *Integrated Safety*.

Chapter 6 then detailed the author's contributions to secondary safety using the so-called *kind of impact* classification. The author's innovations in Computer-Aided (CAx) methods and traffic safety systems related to frontal impact, side impact, rear impact and rollover were summarised and compared to the current state-of-the-art. Firstly, the contributions to frontal impact included basic research on frontal crashes with narrow offset, which is not currently covered by legal or other requirements for secondary vehicle safety. Appropriate load cases for evaluating protection systems were shown, as well as a proven concept for such a system (Flexible Collision Deflector). Additional research in frontal impact focused on simulation methods for low-risk airbag deployment, which previously had only been developed through experimental methods. Secondly, with respect to side impact, a V-model type of a development process with an emphasis on virtual methods was presented. The main element was a sled test rig that was specially designed for the verification of simulation results. The development process was successfully used in a series development of a passenger car without a hardware prototype. Thirdly, for rear impact, the author conducted basic research dealing with the biomechanics of whiplash injuries. The application of the Neck Injury Criterion NIC for automotive development was investigated, and cadaver testing in experimental rear-impacts demonstrated the relevance for human beings of NIC, which had previously been developed by Swedish research on anesthetised pigs. Furthermore, evaluation methods with a generic sled test and the development of a neck protection system based on an Inflatable Head Restraint (IHR) was described. Finally, basic research in the rollover load case dealt with the evaluation of vehicle roof strength in simulation and experiment. In addition, a device called *Rolland* was proposed, which measures the head clearance after a so-called inverted drop test.

Chapter 7 described a study performed by the author which assessed the potential benefits of traffic safety systems and then compared the results of this study to state-of-the-art research. This generated a proposal for the prioritisation of future traffic safety systems based on an evaluation of fatal accidents in Austria. The innovative aspect of the described approach is a detailed reconstruction of the pre-collision phase by means of a computer simulation of the dynamics of the vehicles involved. The relatively high number of cases (217) enabled the preparation of an accident database that was detailed enough to provide insight into the individual case, while still providing sufficient quantity for the statistical evaluation of a representative database. In order to achieve statistical representativeness, the cases were weighted to the national statistics of the year considered (2003). This statistical weighting was necessitated by the underrepresentation of single-vehicle accidents. Although such cases are especially relevant for Electronic Stability Control (ESC) systems, the police do not report them in sufficient detail. The statistical weighting helped compensate for this fact and yielded a weighted database of 260 cases. Forty-three different traffic safety systems were then assessed using this database, thereby combining the advantages of statistical and in-depth accident databases. Wherever possible, the benefit of a safety system was directly assessed by simulating the system in the pre-collision phase. It was determined that Evasive

Manoeuvre Assistant (EMA), Lane Keeping Assist (LKA) and Predictive Emergency Braking (PBA) with autonomous braking of the vehicle are the most effective *intervening* systems. The high potential benefit of two non-intervening but *supporting* systems, Collision Warning Systems (CWS) and Driver Vigilance Monitoring (DVM), were also assessed.

In chapter 8, the author proposed a possible approach for integrating primary and secondary safety systems with the so-called *Integrated Safety Controller* (ISC). A control of an adaptive restraint system with the goal of constant force restraint (CFR) was discussed. It included pre-firing and pre-setting the restraint system with respect to the collision scenario and the vehicle occupant. The main innovation was the prediction of the acceleration behaviour (pulse) of the passenger compartment based on input data from the environment recognising system (ERS) and prior knowledge from crash tests or numerical simulation. The Time-to-Fire (TTF) of the restraint system and the level of the restraint force were optimised for minimised risk of injury.

The ISC approach consisted of the following models. The pre-collision model (ISC-P) predicted the trajectories of approaching vehicles and obstacles. The vehicle model (ISC-V) did the same for the ego-vehicle, but taking into account the interventions of Advanced Driver Assistance Systems (ADAS) and knowledge of the vehicle dynamics as measured by on-board sensors. In the case of an unavoidable collision, the anticipated collision parameters were forwarded to the collision model (ISC-C), which predicted the deceleration pulse of the ego-vehicle's passenger compartment. This pulse was then input to the occupant model (ISC-O), which calculated the optimal trigger time and force levels of an adaptive restraint system. Optimisation was carried out with respect to the injury risk of the relevant occupant. Thus, the mass and seating position of the occupant were taken into account. The ISC-I integration model handled the data management and enabled separate development of the different sub-models. In order to achieve real-time performance for full-vehicle application, the mechanics of all of the models described were kept simple. They were based on well-established methods, such as state estimation with Kalman filters and the use of a single-track vehicle dynamics model to assess the influence of primary safety systems. The presented approach was then verified in the straight frontal collision load case with full overlap. Future research will deal with full-vehicle application and real-time performance. The expansion to other load cases, such as oblique and offset frontal crashes or rear impact, is an additional topic. The final stage will incorporate intervening ADAS and collision avoidance.

To conclude, traffic safety is an important aspect of mobility which is of key public interest. The contributions presented in this thesis include several different topics that will support future progress in reducing the consequences of traffic accidents. In particular, the investigation of ADAS benefits serves as a useful support for the decision makers who are developing the traffic safety road map of the next decade. The contributions presented also included novel methods and processes for the virtual development of vehicles. Cost-efficient development and products, along with a fast time-to-market and a wide product portfolio, are the keys to success for vehicle manufacturers operating in today's global automotive market. The described ISC approach is a holistic approach to the integration of traffic safety systems. It offers a promising option that will help vehicle manufactures and system suppliers design and build the new and improved traffic safety systems of the future.

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A. Appendix

Table A.1.: Simulated vehicle based safety systems in RCS-TUG study

Abbr.	Vehicle Safety systems	Abbr. in [Roh09]	Vehicle Safety systems in [Roh09]
ABS	Anti-Lock Braking System	ABS	Antiblockiersystem
EMA	Evasive Manoeuvre Assistant	ES	Notfalllenkung
ESP	Electronic Stability Program	ESP	Elektronisches Stabilitätsprogramm
ESP cons.	ESP conservative	ESP Con	ESP Konservativ
ESP sport.	ESP sportive	ESP Sport	ESP Sportlich
PBA A a	Predictive Brake Assist, intervention strategy A, driver reaction a	BAS ES_A_FHa	BAS Eingriffstrategie A Fahrerhandlung a
PBA A b	Predictive Brake Assist, intervention strategy A, driver reaction b	BAS ES_A_FHb	BAS Eingriffstrategie A Fahrerhandlung b
PBA A c	Predictive Brake Assist, intervention strategy A, driver reaction c	BAS ES_A_FHc	BAS Eingriffstrategie A Fahrerhandlung c
PBA B b	Predictive Brake Assist, intervention strategy B, driver reaction b	BAS ES_B_FHb	BAS Eingriffstrategie B Fahrerhandlung b
PBA B c	Predictive Brake Assist, intervention strategy B, driver reaction c	BAS ES_B_FHc	BAS Eingriffstrategie B Fahrerhandlung c

Table A.2.: List and Classification of Vehicle Safety Systems

Abbr.	Safety Sys.	Safety	Vehicle.	IV Strat.	Exec. Lev.	Comf./Saf.	Complex.	Auton.	TRACE 1st	TRACE 2nd
ACC	Adaptive Cruise Control	P	C	C	G	C	2.7	3	Drive Safe	Cruise Control
A-ACC	Advanced Adaptive Cruise Control	P	C	C	G	C	3.1	3	Drive Safe	Cruise Control
CWS	Collision Warning	P	C	W	S	S	4.1	1	Drive Safe	Collision Protection
CWA	Collision Avoidance	P	C	IV	G	S	4.8	4	Drive Safe	Collision Protection
RI-C	Rear-Impact Countermeasures	P	C	W	G	S	2.6	5	Drive Safe	Collision Protection
VRU-P	Vulnerable Road Users Protection	P	C	W	G	S	2.3	5	Drive Safe	VRU Protection
LKA	Lane Keeping Assistant	P	C	C	G	C	3.5	3	Drive Safe	Lane Control
LDW	Lane Departure Warning and Control	P	C	W	G	S	3.5	1	Drive Safe	Lane Control
LCA	Lane Changing Assistance	P	C	IF	G	C	3.8	1	Drive Safe	Lane Control
ICA	Intersection Collision Avoidance	P	C	IV	G	S	4.3	3	Drive Safe	Lane Control
FDW	Following Distance Warning	P	C	W	G	S	2.4	1	Drive Safe	Speed Control
ISA	Intelligent Speed Adaptation	P	C	IF	G	S	3.5	1	Drive Safe	Speed Control
SAS	Speed Alerting System	P	C	W	G	S	1.6	1	Drive Safe	Speed Control
SLS	Speed Limiting System	P	C	C	G	S	1.2	4	Drive Safe	Speed Control
HDC	Hill Descent Control	P	C	C	S	C	1.9	3	Drive Safe	Speed Control
TSR	Traffic Sign Recognition and Alert	P	C	W	G	C	2.3	1	Drive Safe	Traffic / Signal Detection
BSM	Blind Spot Monitoring	P	C	W	G	S	3.3	1	Drive Safe	Traffic / Signal Detection
RV-D	Rear-View Displays	P	C	IF	G	C	3.8	1	Drive Safe	Traffic / Signal Detection
C2C	Inter-Vehicle Communication Systems	P	C	IF	N	S	4.8	1	Drive Safe	Traffic / Signal Detection
NAV	Navigation Systems	P	C	IF	N	C	1.9	1	Drive Safe	Traffic / Signal Detection
RPDA	Rear Parking Distance Aid	P	C	W	G	C	0.7	1	Drive Safe	Parking
PPA	Parallel Parking Assist	P	C	IV	G	C	0.7	2	Drive Safe	Parking
DVW	Driver Vigilance Monitoring	P	C	W	S	S	2.3	1	Drive Safe	Driver Aptness
AI	Alcohol Detection and Interlock	P	C	IV	N	S	2	1	Drive Safe	Driver Aptness
DDS	Deflation Detection System	P	C	W	G	S	1.3	1	Drive Safe	Tires
TPMS	Tyre Pressure Monitoring	P	C	W	G	S	1.3	1	Drive Safe	Tyres
RF-I	Runflat Indicator	P	C	W	G	S	1.3	1	Drive Safe	Tyres
ABS	Anti-Lock Braking System	P	C	IV	S	S	2.9	5	Braking Systems	Control
CBC	Cornering Brake Control	P	C	IV	S	C	2.9	5	Braking Systems	Control
SBC	Sensotronic Brake Control	P	C	IV	S	C	2.9	5	Braking Systems	Control
EBFD	Electronic Brake Force Distribution	P	C	IV	S	S	2.9	5	Braking Systems	Control
EMB	Electro Mechanical Brake	P	C	IV	S	S	2.9	5	Braking Systems	Cross by Wire Brakes
EHB	Electro Hydraulic Braking	P	C	IV	S	S	2.9	5	Braking Systems	Cross by Wire Brakes
EHPB	Electro-Hydraulic parking brake	P	C	IV	S	C	2.9	2	Braking Systems	Cross by Wire Brakes
EPB	Electronic Parking brake	P	C	IV	S	C	2.9	2	Braking Systems	Cross by Wire Brakes
BA	Brake Assist	P	C	IV	G	S	3.1	4	Braking Systems	Assisted
PAB	Predictive Assist Braking	P	C	IV	G	S	3.8	4	Braking Systems	Assisted
DBC	Dynamic Brake Control	P	C	IV	S	S	2.9	5	Braking Systems	Assisted
HBB	Hydraulic Brake Boost	P	C	IV	S	S	2.9	5	Braking Systems	Assisted
CC-HL	Cornering/Axis Controlled Headlights	P	C	IF	G	C	1.4	5	Visibility	Lights Activation
SP-HL	Speed Adapting Headlights	P	C	W	G	C	1.5	5	Visibility	Lights Activation
A-HL	Automated Headlights	P	C	IF	G	C	2.5	5	Visibility	Lights Activation
A-BL	Adaptive Brake Lights	P	C	W	G	S	2.3	5	Visibility	Lights Activation
NV	Night Vision	P	C	IF	G	S	3.2	5	Visibility	Vision Enhancement
HU-D	Heads-Up Display	P	C	IF	G	C	1.8	5	Visibility	Vision Enhancement
AWW	Automated Windscreen Wipers	P	C	IV	G	C	1.3	5	Visibility	Auto Rain Detection
MS	Moisture Sensing	P	C	IV	G	C	1.3	5	Visibility	Auto Rain Detection
HWS	Headlight Washing System	P	C	IV	G	S	1.3	5	Visibility	Other
M-ADF	Mirrors with Automatic Dip Function	P	C	IV	G	C	1.8	5	Visibility	Other
ESP	Electronic Stability Control	P	C	IV	S	S	3.4	5	Handling / Kinematics	Anti Rollover / Loss of Control
TC	Traction Control	P	C	IV	S	S	3.3	5	Handling / Kinematics	Anti Rollover / Loss of Control
ESP	Automatic Stability Control + Traction	P	C	IV	S	S	3.3	5	Handling / Kinematics	Anti Rollover / Loss of Control
ASR	Anti-Slip Regulation	P	C	IV	S	S	2.9	5	Handling / Kinematics	Anti slip
ASC	Acceleration Skid Control	P	C	IV	S	S	2.9	5	Handling / Kinematics	Anti slip

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Table A.2 – continued from previous page

Abbr.	Safety Sys.	Safety	Vehicle.	IV Strat.	Exec. Lev.	Comf./Saf.	Complex.	Auton.	TRACE 1st	TRACE 2nd
ABC	Active Body Control	P	C	IV	S	C	3.2	5	Handling / Kinematics	Differential suspension
Tor-D	Torsen Differential	P	C	IV	S	S	1.7	5	Handling / Kinematics	Differential suspension
VC-D	Viscous Coupling Differential Lock	P	C	IV	S	S	1.7	5	Handling / Kinematics	Differential suspension
A-D	Active Differential	P	C	IV	S	S	1.7	5	Handling / Kinematics	Differential suspension
Int-L	Interior Layout	S	C	P	n.a.	S	1.5	5	Structural	Ease of use
DF	Deformable Structure	S	C	P	n.a.	S	2	5	Structural	Robustness
SC	Safety Cell	S	C	P	n.a.	S	2	5	Structural	Robustness
SI-P	Side-Impact Protection	S	C	P	n.a.	S	2	5	Structural	Robustness
L-WS	Laminated Windscreen	S	C	P	n.a.	S	1.8	5	Structural	Glass
EPG	Enhanced Protective Glass	S	C	P	n.a.	S	1.5	5	Structural	Glass
Supp-S	Well Supported Seats	S	C	P	n.a.	S	1.5	5	Structural	Internal protective Structure
EA-SC	Energy Absorbing Steering Column	S	C	P	n.a.	S	2	5	Structural	Internal protective Structure
SPS	Safety Pedal System	S	C	P	n.a.	S	1.5	5	Structural	Internal protective Structure
F-AB	Frontal Airbags	S	C	P	n.a.	S	1.5	5	Airbags	General
S-AB	Side Airbags	S	C	P	n.a.	S	1.5	5	Airbags	General
K-AB	Knee / Leg Airbag	S	C	P	n.a.	S	1.5	5	Airbags	Area Specific
Cu-AB	Inflatable Curtain	S	C	P	n.a.	S	1.5	5	Airbags	Area Specific
Ca-AB	Inflatable Carpet	S	C	P	n.a.	S	1.5	5	Airbags	Area Specific
R-AB	Roofbags	S	C	P	n.a.	S	1.5	5	Airbags	Area Specific
AS-AB	Anti Sliding Airbags	S	C	P	n.a.	S	1.5	5	Airbags	Movement
Ext-AB	External Airbags	S	C	P	n.a.	S	2	5	Airbags	Pedestrian Protection
SP/WR-S	Seat Position and Weight Recognition Sensor	S	C	P	n.a.	S	1.5	5	Intelligent Systems	Pre-Crash
IPS	Intelligent Protection System	S	C	P	n.a.	S	2.6	5	Intelligent Systems	Pre-Crash
PC-S	Pre-Crash System	S	C	M	n.a.	S	4	5	Intelligent Systems	Pre-Crash
PreSafe	Pre-Safe system	S	C	M	n.a.	S	4.3	5	Intelligent Systems	Pre-Crash
AR	Automatic Rollbars	S	C	P	n.a.	S	2.1	5	Intelligent Systems	Rollover
RO-P	Rollover Protection	S	C	P	n.a.	S	1.8	5	Intelligent Systems	Rollover
PUB	Pop-up Bonnet Systems	S	C	P	n.a.	S	2.8	5	Intelligent Systems	Pedestrian Protection
ISCS	Impact-sensing Cut-off System	S	C	P	n.a.	S	1.5	5	Intelligent Systems	Post
SR	Seatbelt Reminder and Buckle Sensor	S	C	P	n.a.	S	1.5	5	Restraints	Reminder
SB	Seat Belts	S	C	P	n.a.	S	1.5	5	Restraints	Seat belts
ABTS	Belt-in Seat	S	C	P	n.a.	S	1.5	5	Restraints	Seat belts
SB-PT,LL	Seat Belt Pre-Tensioners, Load Limiters	S	C	P	n.a.	S	1.5	5	Restraints	Active Systems
CFR	Constant Force Retractors	S	C	P	n.a.	S	1.5	5	Restraints	Active Systems
CRS	Child Restraint System	S	C	P	n.a.	S	1.5	5	Restraints	Child Specific
CRSF	Child Restraint Seat Fixing	S	C	P	n.a.	S	1.5	5	Restraints	Child Specific
CRS-T	Lower Anchors and Tethers for Children	S	C	P	n.a.	S	1.5	5	Restraints	Child Specific
AHR	Active Head Restraint	S	C	P	n.a.	S	1.3	5	Seats	Driver and Passenger Prot. (Head)
SI-HR	Self-Inflating Head Restraints	S	C	P	n.a.	S	1.3	5	Seats	Driver and Passenger Prot. (Head)
AWS	Anti-Whiplash Seats	S	C	P	n.a.	S	1.1	5	Seats	Driver and Passenger Prot. (Head)
SS	Sliding Seats	S	C	P	n.a.	S	1.5	5	Seats	Driver and pass. prot. (movem.)
eCall	eCall	T	C	W	n.a.	S	0.3	1	Rescue	Accident aid systems
ACN	Automatic Crash Notification	T	C	W	n.a.	S	0.2	3	Rescue	Accident aid systems
ERS	Emergency Response System	T	C	W	n.a.	S	0.2	3	Rescue	Accident aid systems
CDR	Crash Data Recorder	T	C	IF	n.a.	S	1	5	Rescue	Accident aid systems
ABS	Anti-Lock Braking System	P	HGV	IV	S	S	3.1	5	Braking Systems	Brake Control
IBS	Intelligent Brake System	P	HGV	IV	S	S	3.1	5	Braking Systems	Brake Control
EBFD	Electronic Brake Distribution	P	HGV	IV	S	S	3.1	5	Braking Systems	Brake Control
BA	Brake Assist	P	HGV	IV	G	S	3.1	5	Braking Systems	Brake aid
Ret	Retarders	P	HGV	IV	G	S	3.1	5	Braking Systems	Brake aid
BrBl	Brake Blending	P	HGV	IV	S	S	3.1	5	Braking Systems	Brake aid
ACC	Adaptative Cruise Control	P	HGV	C	G	C	3.7	3	Drive Safe	Cruise Control
LDW	Lane Departure Warning and Control	P	HGV	W	G	C	3.6	1	Drive Safe	Detection and Warning System
ODS	Object Detection System	P	HGV	IF	G	S	4.5	1	Drive Safe	Detection and Warning System
PDS	Pedestrian Detection System	P	HGV	W	G	S	9	1	Drive Safe	Detection and Warning System
Parc	Parktronic	P	HGV	W	G	C	1.1	3	Drive Safe	Detection and Warning System

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Table A.2 – continued from previous page

Abbr.	Safety Sys.	Safety	Vehicle.	IV Strat.	Exec. Lev.	Comf./Saf.	Complex.	Auton.	TRACE 1st	TRACE 2nd
SLS	Speed Limiting System	P	HGV	C	G	S	1.4	1	Drive Safe	Speed
EEI	Extended Environmental Information	P	HGV	IF	G	C	4.3	1	Drive Safe	Surroundings
ESP	Electronic Stability Control	P	HGV	IV	S	S	2.9	5	Handling / Kinematics	Anti rollover / Loss of control
TC	Traction Control System	P	HGV	IV	S	S	2.7	5	Handling / Kinematics	Anti rollover / Loss of control
ARP	Active Rollover Protection	P	HGV	P	S	S	2.5	5	Handling / Kinematics	Anti rollover / Loss of control
AFS	Active Front Steering	P	HGV	IV	G	C	2.2	5	Handling / Kinematics	Steering
TL-HL	Twin Lens Headlamp	P	HGV	IF	G	C	1.5	5	Visibility	Lights
A-HL	Adaptive Head lights	P	HGV	IF	G	C	2.6	5	Visibility	Lights
HIDB	High Intensity Discharge Bulbs	P	HGV	IF	G	C	1.3	5	Visibility	Lights
NV	Night Vision	P	HGV	IF	G	S	3.5	5	Visibility	Night
ASC	Adaptive Steering Column	S	HGV	P	n.a.	S	1.5	5	Structural	Vehicle Prot. Structure (internal)
Int-L	Interior Layout	S	HGV	P	n.a.	S	1.5	5	Structural	Vehicle Prot. Structure (internal)
UPS	Underrun Protection Systems	S	HGV	P	n.a.	S	1.5	5	Structural	Vehicle Prot- Structure (external)
DCR	Dual Crash Resistant	S	HGV	P	n.a.	S	1.5	5	Structural	Vehicle Prot. Structure (external)
EmLS	Emergency Lighting Systems	S	HGV	W	n.a.	S	1.1	4	Structural	Vehicle Prot. Structure (external)
EmW	Emergency Windows	S	HGV	P	n.a.	S	1.4	5	Structural	Vehicle Prot. Structure (external)
F-AB	Frontal Airbags	S	HGV	P	n.a.	S	1.5	5	Airbags	General
K-AB	Knee / Leg Airbag	S	HGV	P	n.a.	S	1.5	5	Airbags	Specific Area
HRS	Head restraint system	S	HGV	P	n.a.	S	1.3	5	Seats	Restraints
ROP	Roll over protection system	S	HGV	P	n.a.	S	1.5	5	Intelligent Systems	Rollover
FSS	Fire suppression system	S	HGV	P	n.a.	S	1.5	5	Intelligent Systems	Fire protection systems
AFPS	Automatic fuel pump shut off	S	HGV	P	n.a.	S	1.5	5	Intelligent Systems	Fire protection systems
3-SB	3-points safety belt	S	HGV	P	n.a.	S	1.5	5	Restraints	Seat belts
2-SB	2-points safety belt	S	HGV	P	n.a.	S	1.5	5	Restraints	Seat belts
SB-PT,LL	Seat belt pre-tensioner	S	HGV	P	n.a.	S	1.5	5	Restraints	Systems
CRS-Boo	Child booster Cushion	S	HGV	P	n.a.	S	1.5	5	Restraints	Systems
EmC	Emergency Call	T	HGV	W	n.a.	S	0.8	3	Rescue	Emergency call
Helm	Helmet	S	MC	P	n.a.	S	1.5	5	Rider protection	Rider equipment
ProtC	Protective Clothing	S	MC	P	n.a.	S	1.5	5	Rider protection	Rider equipment
BackP	Back Protector	S	MC	P	n.a.	S	1.5	5	Rider protection	Rider equipment
Ri-AB	Rider Airbag	S	MC	P	n.a.	S	1.5	5	Rider protection	Rider equipment
ASS	Anti skid surface	P	R	IV	S	S	2.4	1	Road Layout	Road conditions / Adaptations
HeRo	Heated road	P	R	IV	S	S	2.3	1	Road Layout	Road conditions / Adaptations
Ro-M	Road markings	P	R	IF	G	S	2.8	1	Road Layout	Road signalling
Sign	Signing	P	R	IF	G	C	2.8	1	Road Layout	Road signalling
Sm-LI	Smart Lighting	P	R	IF	G	C	3.7	1	Road Layout	Road signalling
Round	Roundabout scheme	P	R	IF	G	S	1.8	1	Road Layout	Road signalling
STL	Shelter Tuning Lanes	P	R	IF	G	S	1.8	1	Road Layout	Road signalling
FCL	Flashing Crosswalk lights	P	R	IF	G	S	1.1	1	Road Layout	Road signalling
RLCam	Red light cameras	P	R	IF	G	S	0.8	1	Road Layout	Road cameras
Scam	Speed cameras	P	R	IF	G	S	1.3	1	Road Layout	Road cameras
SFBI	Speed Feedback Indicator	P	R	IF	G	s	0.9	1	Road Layout	Road cameras
posts	Lattix posts	S	R	P	n.a.	S	1.6	5	Energy Absorbing Structures	Road restraint systems
fence	Safety fences	S	R	P	n.a.	S	1.6	5	Energy Absorbing Structures	Road restraint systems
P-GR	Pedestrian guardrail	S	R	P	n.a.	S	1.6	5	Energy Absorbing Structures	Road restraint systems
ConM-B	Concrete Median Barrier	S	R	P	n.a.	S	1.6	5	Energy Absorbing Structures	Road restraint systems
Cr-B	Crash barriers	S	R	P	n.a.	S	1.7	5	Energy Absorbing Structures	Road restraint systems
WR-B	Wire ropes barriers	S	R	P	n.a.	S	1.7	5	Energy Absorbing Structures	Road restraint systems
DR-B	Double Road side barriers	S	R	P	n.a.	S	1.7	5	Energy Absorbing Structures	Road restraint systems
EmCB	Emergency Call Boxes	T	R	W	n.a.	S	0.1	1	Rescue	Road communication systems
DyLM	Dynamic Lane Merging	T	R	IF	n.a.	S	1.2	5	Rescue	Road communication systems

End of table

Table A.3.: Estimated safety systems in RCS-TUG study

Abbr.	Vehicle Safety systems	Abbr. in [Roh09]	Vehicle Safety systems in [Roh09]
ACC	Adaptive Crusie Control	ADR	Automatischer Abstand Regler
ACN	Automatic Crash Notification	Ecall	automatisches Notrufsystem
AFS	Active Front Steering	AFS	Aktivlenkung
AI	Alcohol Detection and Interlock	Alk Lock	Alkoholische Wegfahrsperre
ARP	Active Rollover Protection	ACS	Seitenneigungsstabilisator
ARS	Active Rear Steering	ARS	Mitlenkende Hinterachse
ASR	Anti-Slip Regulation	ASR	Antriebsschlupfregelung
AuDr	Autonomous Driving	Auto Dr	Autonomes Fahren
AuHi	Automated Highway	Auto Highway	Autonome Straße
AWD	All Wheel Drive	AWD	Allrad Antrieb
AYC	Active Yaw Control	AYV	Aktive Gierwinkelbeeinfussung
BSM	Blind Spot Monitoring	BSM	Todwinkelüberwachung
C2C	Inter-Vehicle Communication Systems	XFCD und C2C	Kommunikationssyteme
CC-HL	Cornering/Axis Controlled Headlights	Ad Headl	Adaptives Kurvenlicht
CWS	Collision Warning	CCS	Kollisionserkennung
DVM	Driver Vigilance Monitoring	DM Sleep	Fahrerüberwachung
ICA	Intersection colision Avoidance	X-Assistant	Kreuzungsassistent
IPS	Intelligent Crash Protection	Adap R	Unfallvorbereitungssysteme
LCA	Lane Changing Assistant	LCC	Spurwechselassistent
LDW	Local Danger Warning	LDW	Lokale Warnmeldung
LKA	Lane Keeping Assist	LA	Spurhalteassistent
NAV	Navigation Systems	GPS	GPS Navigation
NV	Night Vision	Night Vision	Nachtsichtsystem
Parc	Parctronic	PAS	Parkmanöverassistent
RO-P	Rollover Protection	RSS	Überschlagschutz
RTTI	Real Time Traffic Information	RTTI	Verkehrsinformation
SAS	Speed Alerting System	Speed Alert	Geschwindigkeitswarnung
SLS	Speed Limiting System	ISA	Höchstgeschwindigkeitsbegrenzung
Sp-R	Speed Recommendation	CSA	Sollgeschwindigkeitsassistent
SR	Seatbelt Reminder and Buckle Sensor	SBreminder	Gurtanlegeerinnerer
TP-C	Tyre Pressure Control	TPM	Reifendruckregler
TrMS	Traffic Management System	DTM (VMS)	Verkehrsleitsysteme
TSR	Traffic Sign Recognition and Alert	TSR	Verkehrszeichenerkennung

Table A.4.: Ranking of safety systems in weighted RCS-TUG database

Rank (avoid.)	**Abbr.**	**Safety System** Full description see [Roh09, BADS08]	**Avoid.**	**Potential**	**No potential**	**No evaluation**	**Rank** (potent.)
1	AuDr	Autonomous Driving	162	12	86	0	1
2	EMA	Evasive Manoeuvre Assistant	48	10	200	2	5
3	LKA	Lane Keeping Assist	43	33	182	2	4
4	PBA-A-a	Predictive Brake Assist, intervention strategy A, driver reaction a	40	16	203	1	6
5	AuHi	Automated Highway	39	3	218	0	12
6	ESC	Electronic Stability Program	27	15	216	2	14
7	PBA-A-b	Predictive Brake Assist, intervention strategy A, driver reaction b	26	30	203	1	7
8	ESC cons.	ESC conservative	25	17	216	2	13
9	SLS	Speed Limiting System	22	16	221	1	17
10	PBA-B-b	Predictive Brake Assist, intervention strategy B, driver reaction b	22	34	203	1	8
11	ESC sport.	ESC sportive	21	17	220	2	18
12	PBA-A-c	Predictive Brake Assist, intervention strategy A, driver reaction c	19	36	204	1	9
13	ICA	Intersection colision Avoidance	18	11	230	0	20
14	AI	Alcohol Detection and Interlock	16	5	216	23	23
15	CWS	Collision Warning	11	87	157	4	2
16	SR	Seatbelt Reminder and Buckle Sensor	10	43	199	8	10
17	ACC	Adaptive Crusie Control	8	8	243	1	27
18	BSM	Blind Spot Monitoring	8	4	248	0	32
19	LCA	Lane Changing Assistant	7	7	246	0	31
20	NV	Night Vision	6	39	216	0	11
21	ABS	Anti-Lock Braking System	4	8	247	1	33
22	PBA-B-c	Predictive Brake Assist, intervention strategy B, driver reaction c	4	30	225	1	19
23	RTTI	Real Time Traffic Information	2	2	256	0	37
24	DVM	Driver Vigilance Monitoring	1	82	169	8	3
25	SP-R	Speed Recommendation	1	39	219	1	16
26	C2C	Inter-Vehicle Communication Systems	1	14	245	0	28
27	CC-HL	Cornering/Axis Controlled Headlights	1	6	253	0	35
28	NAV	Navigation Systems	1	2	256	1	39
29	TSR	Traffic Sign Recognition and Alert	1	20	239	0	22
30	ARP	Active Rollover Protection	1	13	246	0	30
31	TP-C	Tyre Pressure Control	1	3	255	1	38
32	SAS	Speed Alerting System	0	41	218	1	15
33	AYC	Active Yaw Control	0	22	238	0	21
34	RO-P	Rollover Protection	0	17	243	0	24
35	AWD	All Wheel Drive	0	16	244	0	25
36	LDW	Local Danger Warning	0	16	244	0	26
37	ACN	Automatic Crash Notification	0	15	245	0	29
38	ARS	Active Rear Steering	0	11	249	0	34
39	TrMS	Traffic Management System	0	7	253	0	36
40	ASR	Anti-Slip Regulation	0	3	257	0	40
41	AFS	Active Front Steering	0	2	258	0	41
42	Parc	Parctronic	0	2	258	0	42
43	IPS	Intelligent Crash Protection	0	2	23	235	43

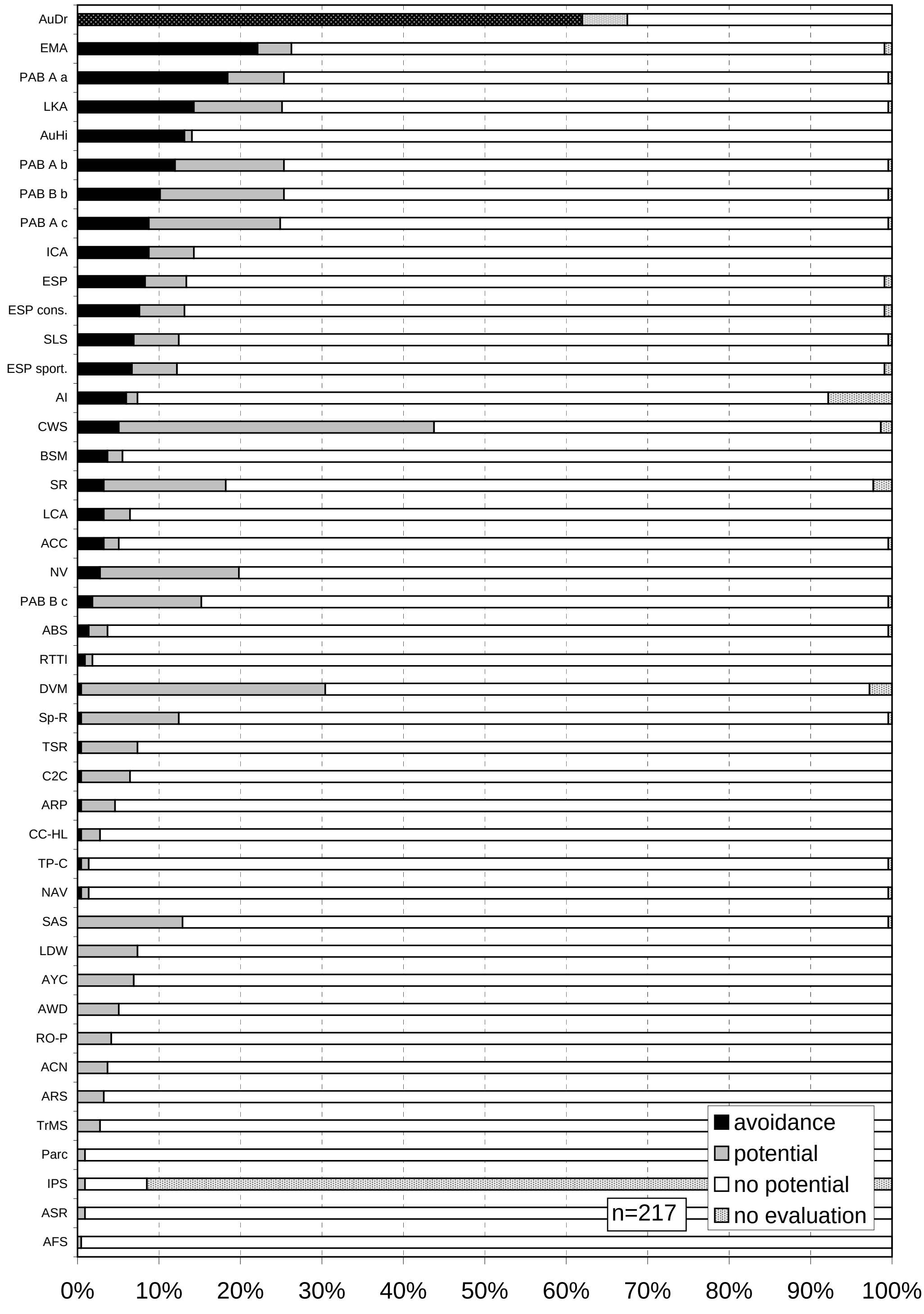


Figure A.1.: Safety potentials of all safety systems in RCS-TUG, ranked by avoidance

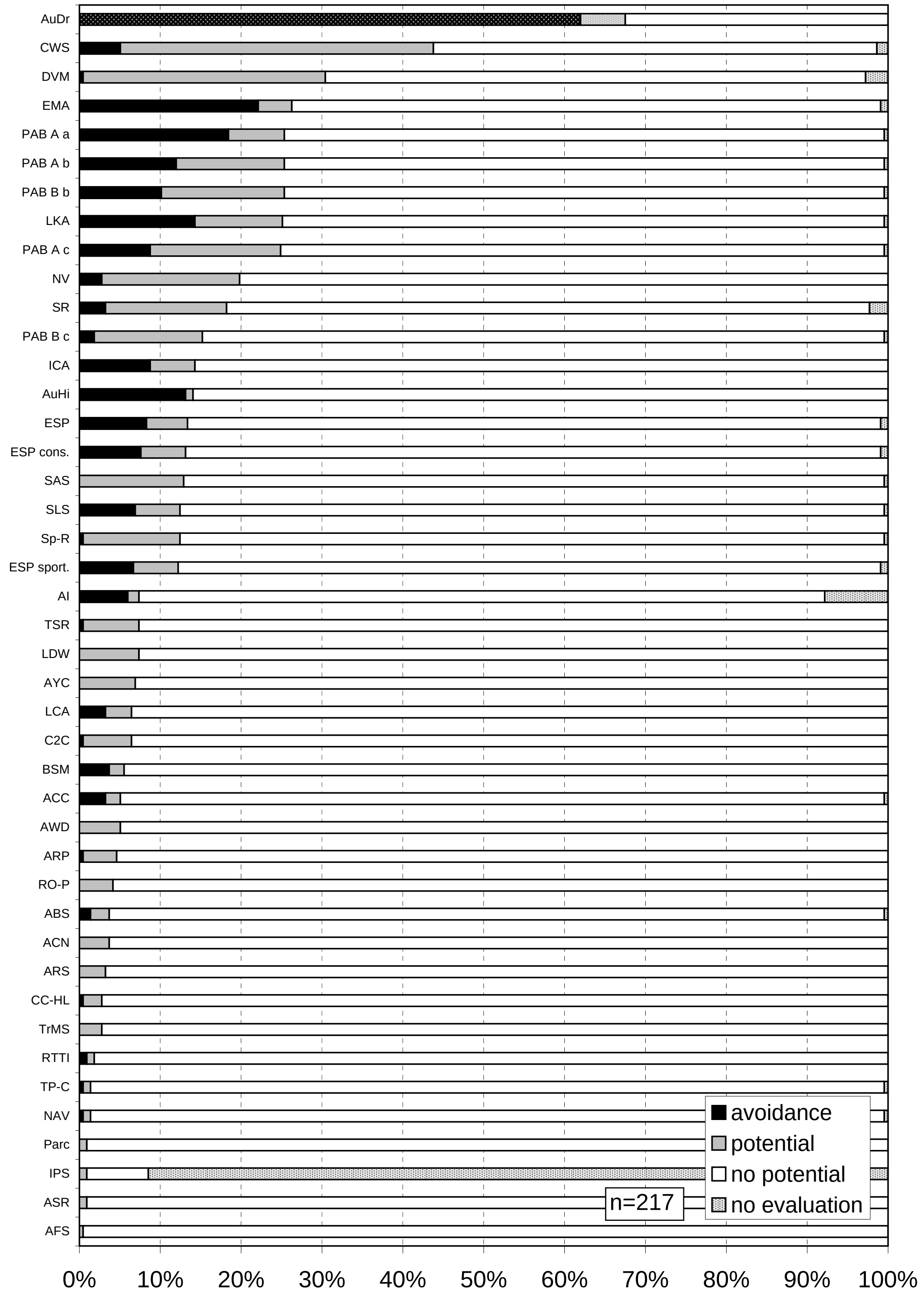


Figure A.2.: Safety potentials of all safety systems in RCS-TUG, ranked by potential

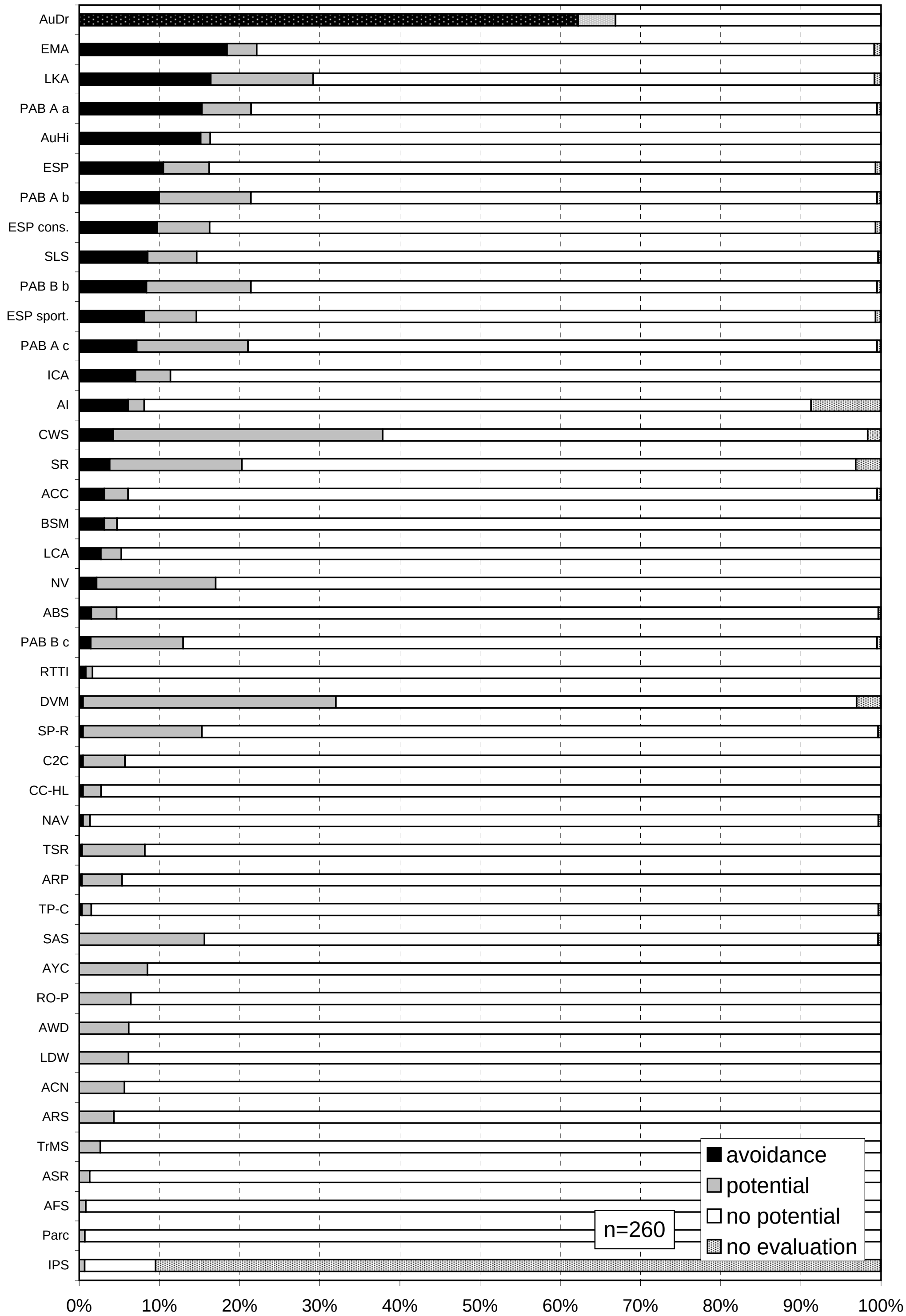


Figure A.3.: Safety potentials of all safety systems in Weighted RCS-TUG, ranked by avoidance

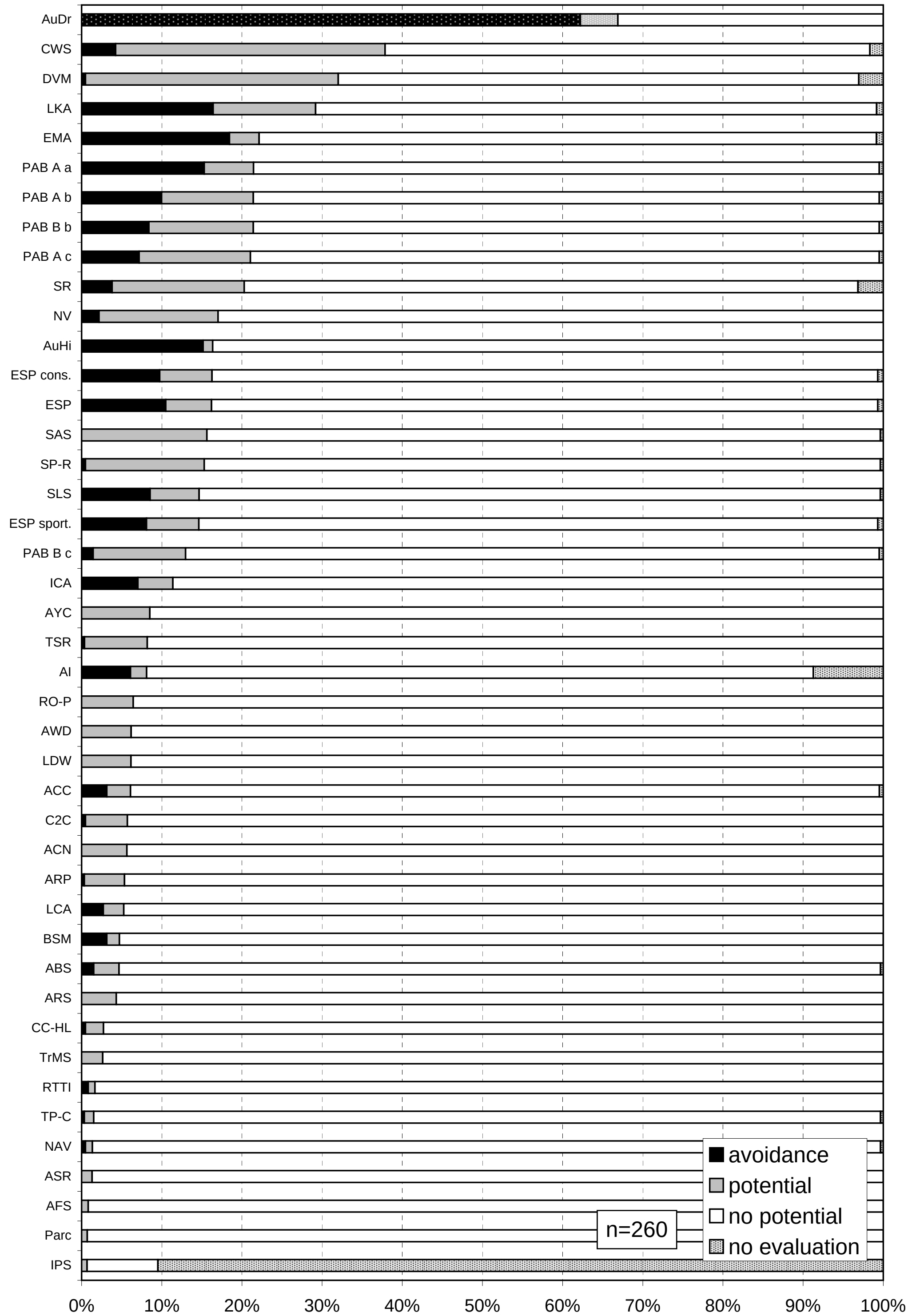
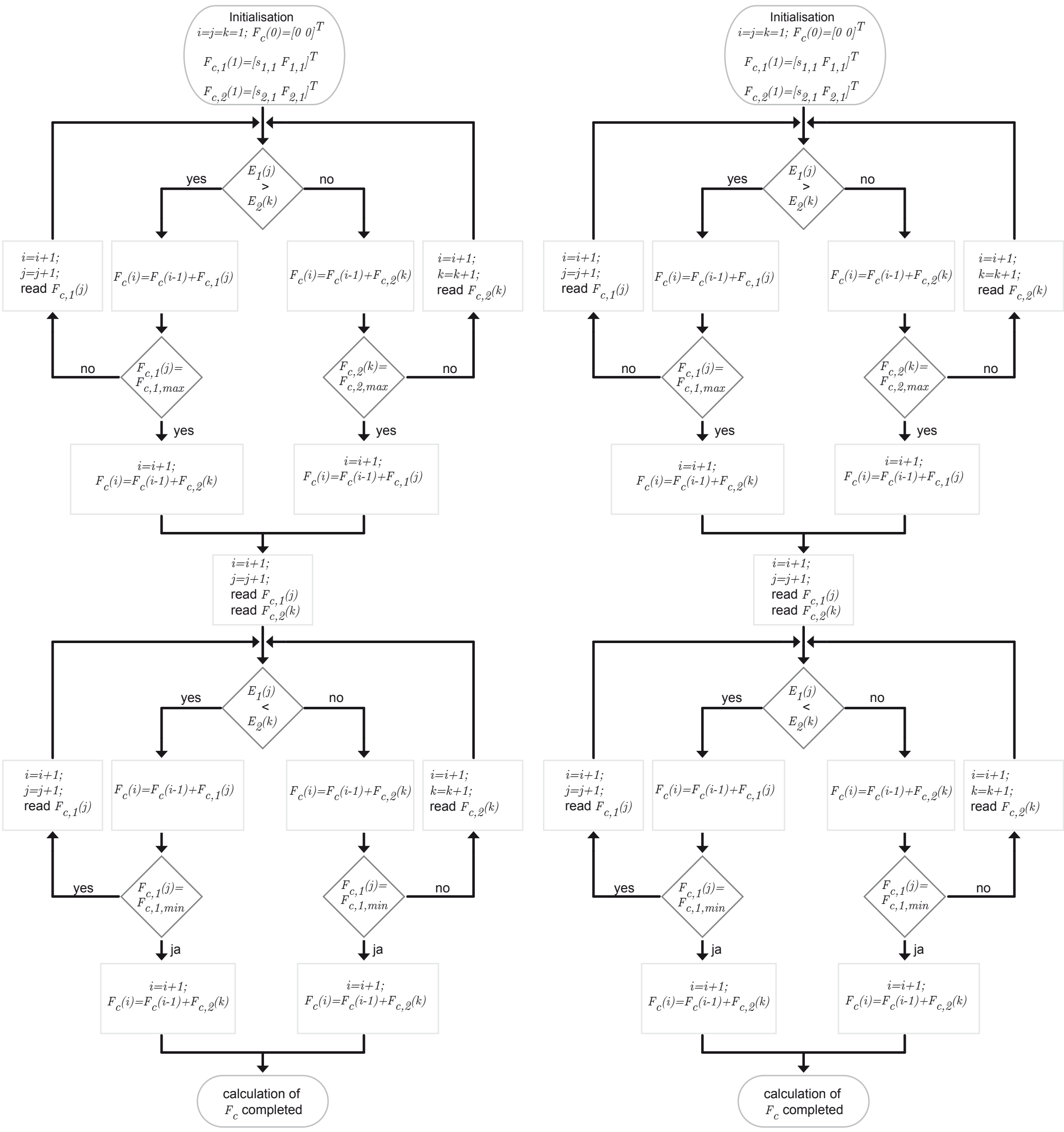


Figure A.4.: Safety potentials of all safety systems in Weighted RCS-TUG, ranked by potential



(a) Comparison of forces

(b) Comparison of deformation energies

Figure A.5.: Flow chart of pointwise numerical calculation of F_c, modified according to [Wal09]

The left figure depicts the work flow to process the deformation spring F_c based on comparison of the force level of the colliding vehicles. The right figure shows the alternative with comparison of deformation energies

Part III.

Author's details

B. Curriculum Vitae

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Personal Information

Born on 13 February 1969 in Leoben, Austria
Austrian citizen, married to Dr. med. univ. Ursula Eichberger
One child (Johanna Eichberger)
Mother tongue: German
Other languages: English (excellent in reading, writing and verbal skills), Spanish (Basic), Japanese (Basic)

Current position

- Assistant lecturer and Vice director at the Institute of Automotive Engineering, Graz University of Technology

- Scientific director of the area D (mechanics) of the Virtual Vehicle Research and Test Center

Education

05/1996 - 09/1998	Ph.D. (Dr. techn.) in Mechanical Engineering Graz University of Technology Graduated 11 December 1998 with distinction
10/1987 - 11/1995	M.Sc. in Mechanical Engineering Graz University of Technology Major: Verkehrstechnik (traffic technology) Graduated 30 November 1995
09/1979 - 06/1987	High school (BG/BRG Leoben, Austria and BG Bregenz, Austria) University entrance diploma 24 June 1987 with success
09/1975 - 06/1979	Primary school (VS Trofaiach, Austria)

Awards

07/2009	*IMETI 2009*, session's best paper award (presented by D. Wallner)
06/2009	*Virtual Vehicle*, special award
05/2009	*Universitätsforschungspreis der Industrie*, winner main category (with D. Wallner)
01/2005	*Forschungsfördergesellschaft FFG*, best project proposal in I^2 funding program
06/1996	*Jubiläumsstiftung Fahrzeugverband der Fahrzeugindustrie Österreichs*, 3rd price for master's thesis

Scientific functions

ongoing	Head of reviewing board of the *3rd Graz Symposium Virtual Vehicle*
ongoing	Reviewer of *ATZ (Automobiltechnische Zeitschrift)*
ongoing	Member of program committee of K2 center of excellence *Virtual Vehicle Research and Test Center*
2009	Reviewer of *IMETI 2010* (3rd International Multi-Conference on Engineering and Technological Innovation)
2009	Reviewer of *ICEME 2010* (International Conference on Engineering and Meta-Engineering)
2008	Reviewer of Elsevier Journal *SIMPRA Simulation Modelling Practice and Theory*
2008	Reviewer of *IMETI 2009* (2nd International Multi-Conference on Engineering and Technological Innovation)
2008	Member of professor search committee for "Technical Logistics" at Graz University of Technology
2008	Conference chair of the *1st Graz Symposium Virtual Vehicle*
2007	Vice scientific director of k-plus center of excellence *Virtual Vehicle*
2007	Head of special session "Advanced Predictability of Crash Models" at the *6th Congress on Modelling and Simulation (EUROSIM 2007)*
2007	Coordination of the technical description for the successful application of the Virtual Vehicle Research and Test Center for a COMET K2 center of excellence
2007	Member of the editorial board of the *1st Graz Symposium Virtual Vehicle*

Professional Experience

02/2007 - now

Institute of Automotive Engineeering
Graz University of Technology
Assistant lecturer

- Vice director
- Teaching in different subjects
- Supervision of diploma theses and students projects

01/2008 - now

Virtual Vehicle Research and Test Center
Graz University of Technology
Scientific head of working area D

- Strategic positioning of the area
- Project planning and controlling
- Responsibility for quantity and quality of publications and reports
- Supervision of diploma theses

10/1998 - 01/2007 **MAGNA STEYR Fahrzeugtechnik AG&CoKG**
Business unit *Engineering*
07/2006 - 01/2007 *Head of innovation field "Intelligent Safety"*

- Strategic road map
- Planning and coordination of advanced research projects
- Project controlling

07/2003 - 06/2006 *Group leader advanced research of vehicle safety department*

- Project management and lead in projects for advanced simulation methods and product innovation
- Planning and coordination of advanced research projects
- Project controlling
- Projects
  - Out of position simulation (k-plus research project A1-c3, FFG)
  - Pedestrian protection (k-plus research project A1-c4, FFG)
  - Sliding collision (k-plus research project A1-c5, FFG)
  - Forming to crash (k-plus research project A4-P1, FFG)
  - Advanced spot weld modelling for crash simulation (k-plus research project A1-c7, FFG)
  - FEM models of crash barriers (internal project)
  - Side impact test rig (internal project)
  - ROLLOVER (EC R&TD project contract nr. G3RD-CT-2002-00802)
  - Advanced protection vehicle (internal project)
  - Highway^3 (I^2 research project, FFG)

10/1998 - 06/2003 *Simulation and development engineer in vehicle safety department*

- Occupant simulation for frontal impact for a roadster and coupe
- Occupant simulation for frontal impact for an AWD estate car
- Head impact simulation for a SUV car (FMVSS201u)
- Engineering of a dual stage frontal airbag for a SUV car
- Occupant simulation for frontal and side impact for a convertible
- Engineering of a head-thorax side airbag for a convertible
- Engineering of an all-belt-to-seat for a convertible
- Engineering and responsibility of side impact protection on full-vehicle level for a SUV
- Engineering and responsibility of side impact protection on full-vehicle level for a compact car

05/1996 - 09/1998 **Institute of Mechanics**
Graz University of Technology
Junior researcher

- Research in biomechanics (whiplash injuries)
- EC-R&DT project "WHIPLASH" (Brite-Euram project nr. 1770)
- Development of whiplash protecting systems (topic of Ph.D. thesis))

11/1994 - 02/1995 **Red Cross Austria**
Civilian service
Paramedic

- Rescue and emergency medical services, patient transport

11/1994 - 02/1995 **Verband der Schadenversicherer (Munich, Germany)**
Accident research department
Junior researcher

- Research in incidence of rear impact (topic of master's thesis)

07/1994-08/1994 **Austria Metall AG (Ranshofen, Austria)**
Simulation department
Trainee

- FEM calculations using ANSYS

07/1993-08/1993 **EVG (Graz, Austria)**
Design department
Trainee

- Design of special machinery (welding station for wire fence)

06/1992 - 07/1992 **Toyo Tire (Osaka, Japan)**
Research department
Trainee

- Tyre research, measurement and analysis of tyre vibration tests

08/1991 - 09/1991 **Böhler Hochdrucktechnik (Kapfenberg, Austria)**
Design department
Trainee

- Design of high pressure valves

08/1990 - 09/1990 **Böhler Edelstahl (Kapfenberg, Austria)**
Computing department
Trainee

- Programming of software utilities (C+)

Training courses

- Several training courses dealing with social competence and personality development (e.g. didactics, rhetoric, communication and presentation skills)
- Several training courses dealing with technical topics (e.g. project management, training on specialised software packages, specialised training for automotive development, scientific paper writing)

C. Author's list of publications

Contributions to a book

- M. Darok, E. P. Leinzinger, A. Eichberger, and H. Steffan. *Neck Injury Criterion Validation Using Human Subjects and Dummies*, in *Frontiers in Whiplash Trauma: Clinical and Biomechanical*, pages 409–434. IOS Press, Amsterdam, Niederlande, 2000.

Peer-reviewed journal articles

- D. Wallner, A. Eichberger, and W. Hirschberg. A Novel Control Algorithm for Integration of Active and Passive Vehicle Safety Systems in Frontal Collisions. *Journal of Systemics, Cybernetics and Informatics*, 2010 (accepted).

- A. Eichberger and D. Wallner. Review of Recent Patents in Integrated Vehicle Safety, Advanced Driver Assistance Systems and Intelligent Transportation Systems. *Recent patents on mechanical engineering*, 3(1):1–13, 2010 (in press).

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- M. Mühlbauer, A. Eichberger, B. C. Geigl, and H. Steffan. Analysis of kinematics and acceleration behaviour of the head and neck in experimental rear-impact collisions. *Neuro-Orthopedics*, 25(1-2):1–17, 1999.

Peer-reviewed conference papers

- A. Eichberger. Introduction of a Peer-review Process to an Interdisciplinary Symposium on Virtual Vehicle Development. In *Proceedings of the 2nd International Symposium on Peer Reviewing, ISPR 2010*, (accepted), 29 June - 2 July 2010. Orlando, USA.

- A. Eichberger, E. Tomasch, R. Rohm, H. Steffan, and W. Hirschberg. Detailed analysis of the benefit of different traffic safety systems in fatal accidents. In *Proceedings of 19th Annual EVU Congress*, (in review), 14-16 October 2010. Prague, Czech Republic.

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- O. Boström, M. Krafft, B. Aldman, A. Eichberger, R. Fredriksson, Y. Haland, P. Lövsund, H. Steffan, M. Y. Svensson, and C. Tingvall. Prediction of Neck Injuries in Rear Impacts Based on Accident Data and Simulations. In *Proceedings of the 1997 International IRCOBI Conference on the Biomechanics of Impact*, September 1997. Hannover, Germany.

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- A. Eichberger and J. Bernasch. *Proceedings of the 1st Grazer Symposium Virtuelles Fahrzeug (GSVF 2008).* Virtual Vehicle Research and Test Center, Graz, Austria.
- B. Fachbach, J. Bernasch, and A. Eichberger. *Proceedings of the 3rd Grazer Symposium Virtuelles Fahrzeug (GSVF 2010).* Virtual Vehicle Research and Test Center, Graz, Austria.

Conference papers

- D. Wallner, A. Eichberger, W. Hirschberg, and R. Cresnik. A Situation Based Method to Adapt the Vehicle Restraint System in Frontal Crashes to the Accident Scenario. In *Proceedings of the 21st International Technical Conference on the Enhanced Safety of Vehicles Conference (ESV)*, 15–18 June 2009.
- A. Rieser, C. Nußbaumer, A. Eichberger, and H. Steffan. A Development Process for Creating Finite-Element Models of Crash Test Dummies Based on Investigations of the Hardware. In *Proceedings of the 21st International Technical Conference on the Enhanced Safety of Vehicles Conference (ESV)*, 15-18 June 2009. Stuttgart, Germany.
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- A. Eichberger, D. Michbronn, F. Wörgötter, and B. Fellner. A Novel Virtual Development Process for Side Impact at Magna Steyr Based on Numerical Simulations Verified by Component Testing. In *Proceedings of the 6th EUROSIM Congress on Modelling and Simulation*, 9-13 September 2007. Ljubljana, Slovenia.
- C. Ruff, A. Eichberger, and T. Jost. Simulation of an Airbag Deployment in Out-of-Position Situation. In *Proceedings of 5th LS-DYNA Forum*, 12-13 October 2006. Ulm, Germany.
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- W. Schimpl, G. Schönberger, and A. Eichberger. Entwicklung und Umsetzung eines Abgleitmechanismus zur Erhöhung der Fahrzeugsicherheit. In *Proceedings of 4th LS-DYNA Forum*, volume B-III, pages 11–24, 21 October 2005. Bamberg, Germany.
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Conference presentations

- A. Eichberger, H. Schluder, D. Watzenig, and D. Wallner. Absicherung der integrierten Fahrzeugsicherheit über eine unabhängige Co-Simulationsumgebung. In *Proceedings of the Vehicle Property Validation 2010*, Hannover, Germany, 4-5 May 2010. Automotive Circle. Bad Nauheim, Germany (conference cancelled).
- A. Rieser, H. Schluder, A. Eichberger, T. Heubrandtner, G. Trattnig, and A. Prüggler. CAE-Simulation und Absicherung. In *Proceedings of the 4th Grazer Safety Update 2009.* carhs GmbH, 30 September - 1 October 2009. Graz, Austria.
- F. Wörgötter, B. Fellner, D. Michbronn, and A. Eichberger. Neuer Entwicklungsprozess bei Magna Steyr zur Seitencrashauslegung basierend auf numerischer Simulation verifiziert durch Versuche auf Komponentenebene. In *Proceedings of the CTI Spezialtag Fahrzeugtüren: Seitenaufprall / Side Impact*, 7 July 2008. Nurnberg, Germany.
- H. Schluder, R. Cresnik, D. Wallner, and A. Eichberger. Kopplung von aktiven und passiven Sicherheitssystemen über eine unabhängige Co-Simulationsumgebung. In *Proceedings of the Integrated Safety Symposium 2008*, 2-3 July 2008. Hanau, Germany.
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- E. Dohr, A. Eichberger, J. Barrios, and T. Pyttel. Development of a FEM Simulation Procedure for an effective Design of the Vehicle Structure for a Rollover

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Patents

- D. Wallner and A. Eichberger. Rückhaltesystem-Steuerung basierend auf berechneten Beschleunigungsdaten, 10 March 2009. Application document number: 10 2009 012 407.1.

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- H. Steffan, B. C. Geigl, A. Moser, B. Fachbach, and A. Eichberger. Fahrzeugsitzanordnung mit einem Sitzteil und einer Lehne, 18 November 1996. Patent document number: DE 196 47 649 A1.

Scientific reports

- A. Rieser and A. Eichberger. Investigation of influence of impact angle and overlap on narrow offset collisions. Deliverable report for IIHS (assigned study), Virtual Vehicle Research and Test Center, 2009.

- A. Eichberger. Design Instruction; Final Version. Deliverable report T 6.2, European Community R&TD-Project Improvement of Rollover Safety of Passenger Vehicles, 2006.

- A. Eichberger. Performance Criteria - Final Version. Deliverable report T 6.1, European Community R&TD-Project Improvement of Rollover Safety of Passenger Vehicles, 2006.
- L. Lamy and A. Eichberger. Demonstration model; Verification of improvements; Hardware demonstrator ROLLAND. Deliverable report T6.3, T6.4, European Community R&TD-Project Improvement of Rollover Safety of Passenger Vehicles, 2006.
- A. Eichberger and E. Dohr. Report on Design Guidance for numerical models used to evaluate structures - Part 1: Rollover Simulations with PAM-CRASH. Deliverable report Deliverable D 4.1.2;, European Community R&TD-Project Improvement of Rollover Safety of Passenger Vehicles, 2006.
- A. Eichberger. PMHS-Tests. Activity report Activity 1.3 (1.3.2 Testing), European Community R&TD-Project Whiplash (Brite-EuRam project 3770), 1998.

D. Author's list of presentations and teaching activities

Oral presentations given by author[1]

- 19th Annual EVU Congress, Prague, Czech Republic, presentation: *Detailed analysis of the benefit of different traffic safety systems in fatal accidents*, 16 October 2010 (accepted).
- 21st International Technical Conference on the Enhanced Safety of Vehicles Conference (ESV), Stuttgart, Germany, presentation: *A Situation Based Method to Adapt the Vehicle Restraint System in Frontal Crashes to the Accident Scenario*. 17 June 2009.
- 17th Annual EVU Congress, Nice, France, presentation: *Development of a Crash Test Configuration for Car to Car Frontal Collisions with Small Lateral Overlap*, 7 November 2008.
- 24. VDI/VW-Gemeinschaftstagung Integrierte Sicherheit und Fahrerassistenzsysteme, Wolfsburg, Germany, short presentation: *Retrospektive Bewertung der Effektivität unterschiedlicher Fahrassistenzsysteme bei tödlichen Verkehrsunfällen*, 28 October 2008.
- Integrated Safety Symposium 2008, Hanau, Germany, presentation: *Kopplung von aktiven und passiven Sicherheitssystemen über eine unabhängige Co-Simulationsumgebung*, 3 July 2008.
- 1st Grazer Symposium Virtuelles Fahrzeug, Graz, Austria, presentation (together with G. Lang): *Kopplung von CAE-Disziplinen und Tools über eine unabhängige Co-Simulationsumgebung*, 23 April 2008.
- 6th EUROSIM Congress on Modelling and Simulation, Ljubljana, Slovenia, presentation: *A Novel Virtual Development Process for Side Impact at Magna Steyr Based on Numerical Simulations Verified by Component Testing*, 11 September 2007.
- 2006 International Crashworthiness Conference (icrash 2006), Athens, Greece, presentation: *A new crash test configuration for car to car frontal collisions with small lateral overlap*, 5 July 2006.

[1] Also mentioned in part C.

- Automotive Safety Summit 2006, Berlin, Germany, presentation: *Compatibility In Road Traffic: Innovative Ideas For Occupant Protection*, 16 February 2006.
- 1998 International Research Conference on the Biomechanics of Impact (IRCOBI), Gothenborg, Sweden, presentation: *Evaluation of the Applicability of the Neck Injury Criterion (NIC) in Rear End Impacts on the Basis of Human Subjects Tests*, 17 September 1998.
- 1996 International IRCOBI Conference on the Biomechanics of Impact, Dublin, Ireland, presentation: *Comparison of different car seats regarding head-neck kinematics of volunteers during rear-end impact*, 12 September 1996.

University courses

- *Practical course on Matlab/Simulink* (MATLAB Tutorium Fahrzeugdynamik, LV 331.100), Graz University of Technology, since 2007
- *Modelling and simulation in vehicle dynamics* (Modellbildung und Simulation in der Fahrzeugdynamik, LV 331.094), Graz University of Technology, since 2007
- *Student's project supervision*, Graz University of Technology, since 2007

  Design project in automotive engineering (LV 331.007)

  Work project in automotive engineering (LV 331.010)

  Bachelor project for Mechanical Engineering (LV 331.011)

  Bachelor project for Mechanical Engineering and Business Economics (LV 331.012)
- *Introduction to Automotive Engineering* (LV 331.088), Graz University of Technology, since 2007
- *Automotive Engineering and vehicle safety* (Fahrzeugtechnik und -sicherheit, LV 331.070), Graz University of Technology, since 2008
- *Integrated Vehicle Safety* (Integrierte Fahrzeugsicherheit, LV 333.052), Graz University of Technology, since 2008
- *CAx in Automotive and Engine Technology* (LV 313.066)

Invited guest lectures

- Guest lecture *Introduction to Automotive Engineering* for students of King Saud University, given at Graz University of Technology, August 2009

Supervised diploma theses

This section lists ongoing and completed supervision of master's thesis. The supervision was done either as an university lecturer or as an employee in automotive industry.

Ongoing

- G. Degefie. Vehicle Tracking and Collision Estimation. Institute of Automotive Engineering (Graz University of Technology).
- L. Denifl. Working title: Aktive mechatronische Stabilisatoren für Personenkraftfahrzeuge. Institute of Automotive Engineering (Graz University of Technology).
- P. Zotter. Working title: Simulation von Bremspedalkräften bei einem mechatronischen Bremssystem. Institute of Automotive Engineering (Graz University of Technology). **Scholarship: FSI-Leistungsipendium**.

Completed

- D. Puerta. Prediction of vehicle's motions by application of Extended Kalman Filter on non-linear dynamics models. Report on final project, 2009. Institute of Automotive Engineering (Graz University of Technology).
- R. C. Bognar. MKS Simulation eines Geländefahrzeuges zur Lastdatenermittlung am Beispiel der Mercedes G-Klasse. Master's thesis, 2009. Institute of Automotive Engineering (Graz University of Technology).
- T. Haberkorn. Wartungsarme LH2-Speichersysteme durch effiziente Vakuumtechnologien. Master's thesis, 2008. Institute of Automotive Engineering (Graz University of Technology).
- D. Wallner. Physikalische Modellbildung von integrierten Fahrzeugsicherheitssystemen. Master's thesis, 2008. Institute of Automotive Engineering (Graz University of Technology). **Award: Universitätsforschungspreis der Industrie, winner main category (with A. Eichberger)**. **Scholarship: FSI-Leistungsipendium**.
- I. Pijoan. Prediction of vehicle motions based on evaluation of data measured by object detection systems of advanced driver assistance systems. Report on final project, 2008. Institute of Automotive Engineering (Graz University of Technology).

- R. Cresnik. Pedestrian-Impaktoren in der Crashsimulation - Validierung und Robustheitsanalyse. Master's thesis, 2005. Vehicle Safety Institute (Graz University of Technology).
- C. Zarre. Virtuelles Crashsignal zur Auslegung von Airbagsteuergeräten auf rechnerischem Weg. FH Joanneum, Graz. Master's thesis, 2006. FH Joanneum Graz (University of applied sciences).
- M. Harzheim. Entwicklung eines Aktuators zur Verwirklichung eines vorübergehenden Radeinschlages im Falle eines Frontalcrashs mit geringer Überdeckung der Kollisionspartner. Master's thesis, 2005. FH Joanneum Graz (University of applied sciences).
- C. Werber. Konzeption einer Crashtestkonfiguration für Frontcrashs mit geringer Überdeckung auf Basis von Realunfällen. Master's thesis, 2005. FH Joanneum Graz (University of applied sciences).
- F. Wörgötter. Optimierung der Konstruktion und Untersuchung der Versuchseinflussparameter einer Seitenaufprallschlittenanlage. FH Joanneum, Graz. Master's thesis, 2005. FH Joanneum Graz (University of applied sciences).
- E. Dohr. Entwicklung eines FEM-Simulationsverfahrens zur effektiven Auslegung der Fahrzeugstruktur bei Fahrzeugüberschlag. FH Joanneum, Graz. Master's thesis, 2004. FH Joanneum Graz (University of applied sciences). **Award: 1st price Johann Puch Award 2004**.
- B. Hoislbauer. Inbetriebnahme und Kalibrierung einer Schlittenanlage für den Seitenaufprall. Master's thesis, 2004. Vehicle Safety Institute (Graz University of Technology).
- A. Rieser. Entwicklung eines Schlittens zur Simulation des PKW Seitenaufpralls. Master's thesis, 2004. Vehicle Safety Institute (Graz University of Technology).
- M. Santner. Rechnerische Untersuchung von Einflußgrößen für den ungegurteten Insassenschutz beim Frontalaufprall. Master's thesis, 2003. FH Joanneum Graz - University of Applied Sciences.
- M. Deutschbein. Konzeptentwurf und Prototypenentwicklung von aktiv wirkenden Kopfstützen. Master's thesis, 1997. Institute of Automotive Engineering (Graz University of Technology). **Award: Fahrzeugverband - Jubiläumsstiftung**.

Graz, 21 May 2010