

VDCA *Swedish Vehicle Dynamics Competence Area*

Proceedings from
2024 Vehicle Dynamics seminar
**Virtual verification for Vehicle
Dynamics**

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editors:

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The contents of these proceedings include both **presentations and poster material** and are published at <https://www.sveafordon.com/>, <https://research.chalmers.se/en/publication/541312>, and <https://kth.diva-portal.org/>.

The seminar was arranged by the workgroup Vehicle Dynamics Competence Area in Swedish Vehicular Engineering Association (SVEA, <https://www.sveafordon.com/>).

VDCA *Swedish Vehicle Dynamics Competence Area*

a workgroup in



member of

FISITA 

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Mattias Hjort, VT

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Presentation 6: *Motion cueing for driving simulators* Henrik Hvitfeldt, KTH

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Presentation 7c: *Vehicle engineering courses at LiU*, Jan Åslund, LiU

Poster 1: *Virtual Verification Framework for Vehicle Motion Systems* Albijon Blakqori and Mille Kotur, ZEEKR and Chalmers

Poster 2: *Advancing Road Vehicle Path Length Estimation via Geospatial Analytics*
Yogeswaran Amsavalli, Volvo Trucks, Chalmers and Università di Trento

Poster 3: *Vehicle motion control interface for L4 automated driving* Rishabh Parmar and Bala Chinni, Volvo Trucks and Chalmers

Poster 4: *Modelling of truck tyre wear* Nathan Hedouin, Scania and ESTACA

Poster 5: *Trailer backup assist using steer by wire* Jakob Roempke, Chang Liu, KTH

Poster 6: *Pendulum turn, the optimal racing line through a hairpin curve?* Lukas Jonsson, LiU

Note that the pdf file is generated with these “headings as pdf bookmarks”, so you can also navigate via the “bookmark pane” in your pdf reader.

Announcement of the Seminar

Fordonsdynamiksseminarium 2024

Virtuell verifiering för fordonsdynamik

Vehicle Dynamics seminar 2024

Virtual verification for Vehicle Dynamics



About the seminar

Virtual verification becomes increasingly important and useful in nearly all areas within vehicle development, both for late verification of requirement fulfilment and early elaborative development testing. The development loops can be closed more frequently and on different levels during the vehicle programmes, which cuts calendar time and improves the design. For vehicle dynamics, the methods span from pure off-line simulations, via HIL and SIL, to driving simulators.

Wednesday 29 May

The seminar will be in “hybrid format”, meaning that participation is possible both in-real-life and on-line.

In-real-life:

Hosted by VTI in:

Ada Lovelace room, Entrance 27, B-building, Campus Valla,
Linköping University, Linköping, Sweden (map in the end of this document)

On-line:

Link to on-line meeting will be sent to those who register as on-line.

Registration

Registration is made via https://www.sveafordon.com/en/?post_type=event_listing&p=4644&preview=true. If you are member, it is easier to register if you log in before registration.

You will get a confirmation that you are registered.

Seminar costs

The seminar is free for SVEA members. SVEA will sponsor food (fika and light lunch) for members (incl. pending membership applicants) who attend in-real-life.

For non-members attending in-real-life there will be a fee of 210 SEK (>membership fee).

So, we encourage to apply for SVEA membership (<https://www.sveafordon.com/en/for-members/register/>). SVEA membership fee is 200 SEK/year (junior 0 SEK, senior 100 SEK).

Presenters (incl. poster presenters) attend for free.

Purpose with the seminar

- Present and discuss interesting issues within and challenges for Virtual verification for Vehicle Dynamics
- Create understanding and interest for vehicle dynamics
- Develop, increase, and spread competence
- Networking between engineers, organisations, and students

SVEAs objectives

- To make vehicular technology's voice heard in an increasingly more challenging debate among different vehicle types and transport modes both domestic and globally
- To build a network for efficient distribution of technological information
- To attract the next generation of Swedish vehicular engineers

Poster exhibition

There will be an exhibition of posters. It can be, e.g., master theses or PhD theses, both concluded and almost concluded. Please contact Lars Drugge larsd@kth.se or Bengt Jacobson bengt.jacobson@chalmers.se if you would like to propose a poster.

Each poster presenter should do a poster and a “micro presentation” with a few slides. Then the presenter should also be available for questions at the poster stands.

Proceedings

There will be proceedings from the seminar. This means that the presenters, including poster presenters, are welcome with a paper, or at least a public version of their presentation material. The proceedings will be published on the SVEA web. It will include a list of seminar participants, unless you ask us to not list your name.

The seminar is arranged by the Swedish Vehicle Dynamics Competence Area (VDCA) and Swedish Vehicular Engineering Association (SVEA). The seminar is arranged with VDCA representatives from:

*AFRY Automotive
AstaZero
Chalmers
KTH
LiU
Polestar
Scania
Volvo Cars
Volvo Trucks
VTI
Zeekr Technology Europe*

VDCA Swedish Vehicle Dynamics Competence Area



SVEA is the society member for Sweden in FISITA:

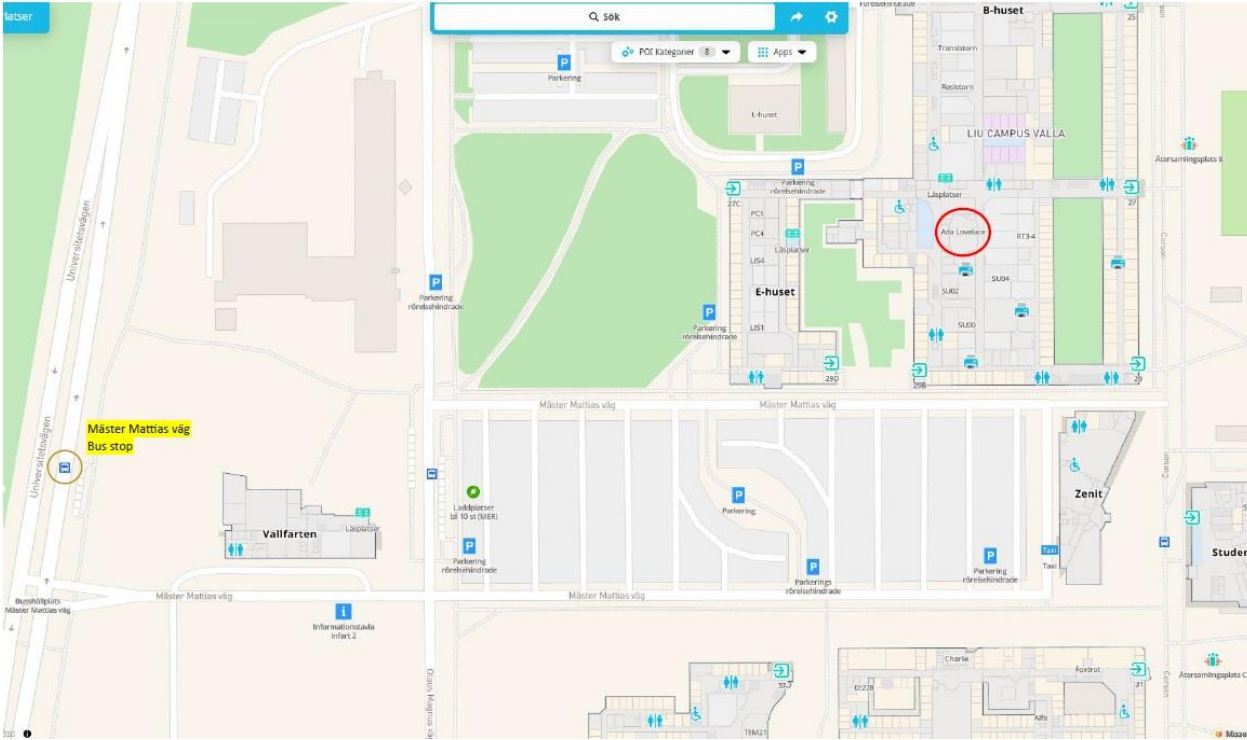


Agenda

Moderator: Lars Drugge and Bengt Jacobson, VDCA and SVEA

09:00-09:30	Coffee and registration	
09:30-09:40	Welcome	Moderator TBD, Bengt Jacobson (VDCA), Mattias Hjort/Sogol Kharrazi (VTI)
09:40-11:00	Session 1: Presentations (20+10 min each): 1. <i>EU-project Roadview</i> 2. <i>Steer-by-wire</i> Micro presentations of posters (3-5 min each): 1. <i>Virtual Verification Framework for Vehicle Motion Systems</i> 2. <i>Advancing Road Vehicle Path Length Estimation via Geospatial Analytics</i> 3. <i>Vehicle motion control interface for L4 automated driving</i> 4. <i>Modelling of truck tyre wear</i> 5. <i>Trailer backup assist using steer by wire</i> 6. <i>Pendulum turn, the optimal racing line through a hairpin curve?</i>	Sogol Kharrazi and Mattias Hjort, VTI Matthijs Klomp, Volvo Cars Albijon Blakqori and Mille Kotur, ZEEKR and Chalmers Yogeswaran Amsavalli, Volvo Trucks, Chalmers and Università di Trento Rishabh Parmar and Bala Chinni, Volvo Trucks and Chalmers Nathan Hedouin, Scania and ESTACA Jakob Roempke, Chang Liu, KTH Lukas Jonsson, LiU
11:00-12:30	Demo tour 3. <i>VTIs simulators & tyre test facility</i> Small presentation and a visit at VTI	Mattias Hjort, VTI
12:30-13:30	Light lunch with networking and manned posters in seminar room.	
13:30-14:30	Session 2: Presentations (20+10 min each): 4. <i>Energy consumption comparison of actuator coordination methods on heavy vehicles in realistic transport applications</i> 5. <i>Tyre models for virtual assessment of heavy vehicles,</i>	Sachin Janardhanan, Volvo Trucks and Chalmers Niklas Fröjd, Volvo Trucks, Jolle Ijkema, Scania, and Sogol Kharrazi, VTI
14:30-15:00	Coffee with networking and manned posters	
15:00-16:00	Session 3: Presentations (20+10 min each): 6. <i>Motion cueing for driving simulators</i> 7. <i>Swedish vehicle engineering education</i> <ul style="list-style-type: none"> ○ <i>KTH, Vehicle engineering</i> ○ <i>Chalmers, Mobility engineering</i> ○ <i>LiU, Vehicle engineering courses</i> ○ Discussion 	Henrik Hvitfeldt, KTH Mikael Nybacka, KTH Dag Henrik Bergsjö, Chalmers Jan Åslund, LiU All
16:00-16:15	Wrap-up <ul style="list-style-type: none"> • Feedback on present years seminar. Proposals for next year's seminar. • Discussion on other 	

Map to seminar venue



Registered participations

63 registered

Name	Affiliation	e-mail
Abhijeet Behera	VTI	abhijeet.behera@vti.se
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Zhaohui Ge	Chalmers	ZHAOHUI.GE@CHALMERS.SE

Introduction to the seminar



VDCA *Swedish Vehicle Dynamics Competence Area*

VEA SWEDISH
VEHICULAR
ENGINEERING
ASSOCIATION
SVERIGES FORDONSTEKNISKA FÖRENING

Introduction to

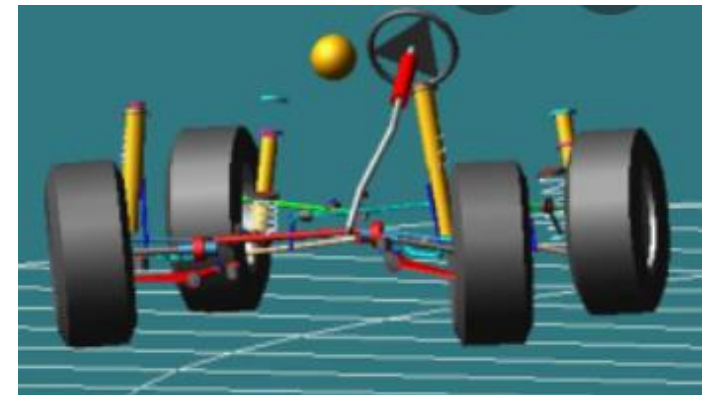
Vehicle Dynamics Seminar 2024

Virtual verification for Vehicle Dynamics

Wednesday 2024-05-29 hosted by VTI at LiU in Linköping, Sweden

About the seminar topic

- Virtual verification
- Useful in vehicle development
 - Both late verification of requirement fulfilment and early elaborative development testing
 - Quicker development loops
 - Cuts calendar time, and
 - Improves the design (through “understanding”)
- For vehicle dynamics, the methods span
 - from pure off-line simulations,
 - via HIL and SIL,
 - to driving simulators.



Purpose with the seminar

- **Present and discuss** interesting issues within and challenges for Vehicle Dynamics within a topic which is selected each year
- Create **understanding and interest** for vehicle dynamics
- Develop, increase, and spread **competence**
- **Networking** between engineers, organisations, and students

Arrangement of the seminar

- Annual since 2013
- Arranged by **VDCA**,
 - which is (now) a workgroup in **SVEA**
 - which is Sweden's "society member" in **FISITA**



SVEAs objectives

- To make **vehicular technology's voice heard** in an increasingly more challenging debate among different vehicle types and transport modes both domestic and globally
- To build a **network** for efficient distribution of technological information
- To attract the **next generation of Swedish vehicular engineers**

VDCA “core group” has representatives and contributions from vehicle manufacturers, university groups in the subject Vehicle Dynamics, and VTI:

AFRY Automotive

AstaZero

Chalmers

KTH

LiU

Polestar

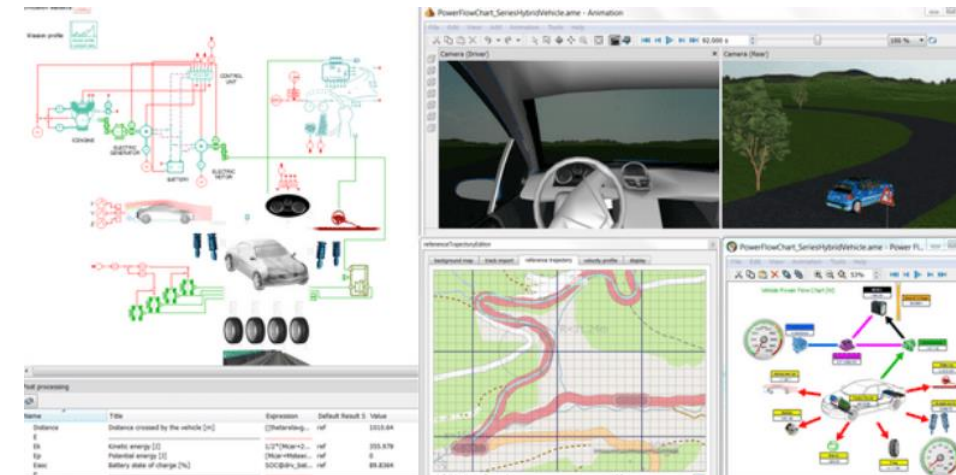
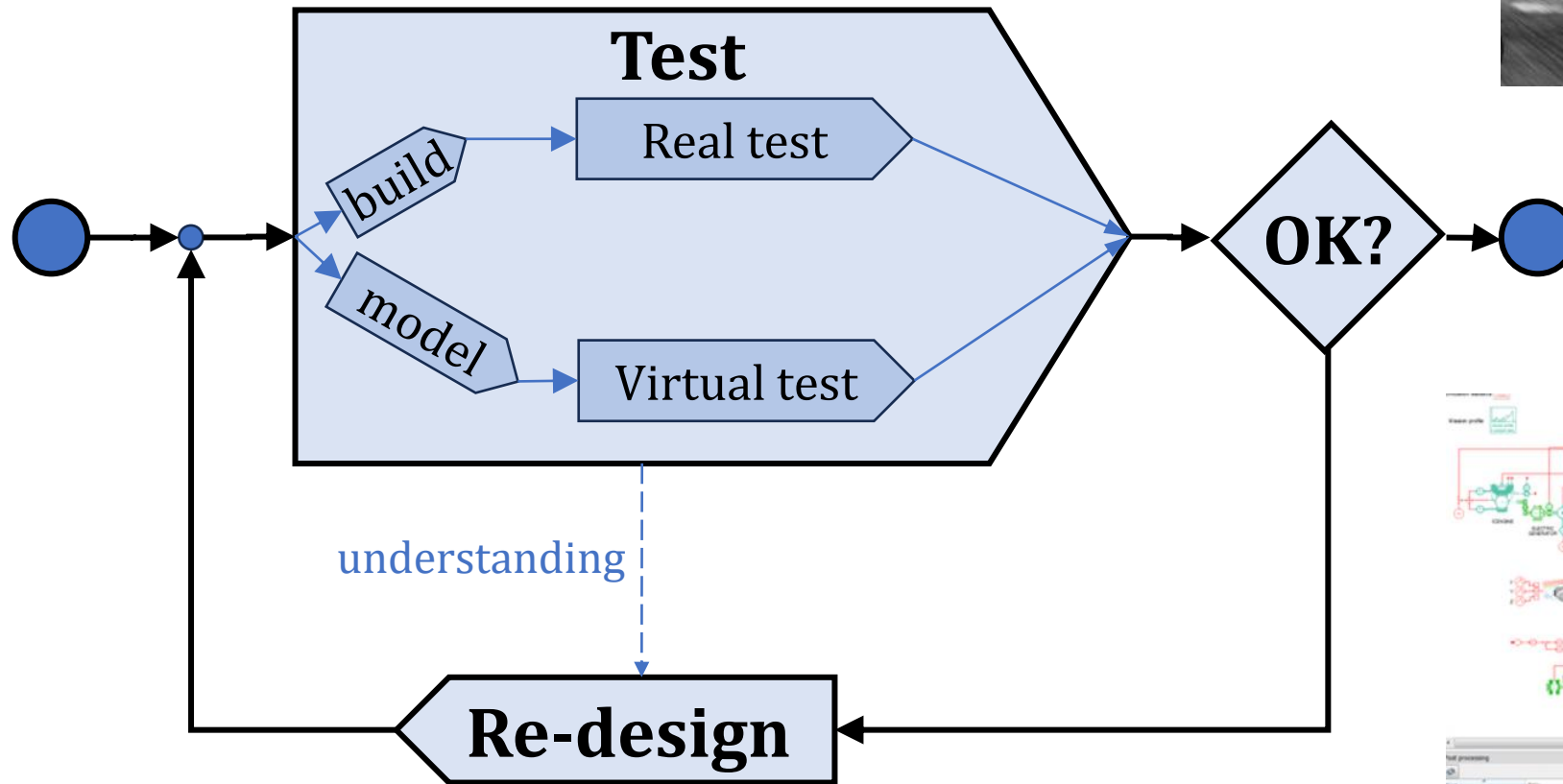
Scania

Volvo Cars

Volvo Trucks

VTI

Zeekr Technology Europe



Agenda to lunch

Agenda

Moderator: Lars Drugge and Bengt Jacobson, VDCA	
09:00-09:30	Coffee and registration
09:30-09:40	Welcome
09:40-11:00	Session 1: Presentations (20+10 min each): 1. <i>EU-project Roadview</i> 2. <i>Steer-by-wire</i> Micro presentations of posters (3-5 min each): 1. <i>Virtual Verification Framework Vehicle Motion Systems</i> 2. <i>Advancing Road Vehicle Estimation via Geospatial</i> 3. <i>Vehicle motion control of L4 automated driving</i> 4. <i>Modelling of truck tyre wear</i> 5. <i>Trailer backup assist using</i> 6. <i>Pendulum turn, the optimal racing line through a hairpin curve?</i>
11:00-12:30	Demo tour 3. <i>VTIs simulators & tyre test facility</i> Small presentation and a visit at VTI
12:30-13:30	Light lunch with networking and manned posters in seminar room.
13:30-14:30	Session 2: Presentations (20+10 min each): 4. <i>Energy consumption comparison decrease coordination methods vehicles in road traffic transport</i> 5. <i>Tyre models for virtual assisted vehicles.</i>
14:30-15:00	Coffee with networking and manned posters
15:00-16:00	Session 3: Presentations (20+10 min each): 6. <i>Motion cueing for driving simulators</i> 7. <i>Swedish vehicle engineering education</i>
16:00-16:15	Wrap-up • Feedback on present years seminar. Proposals for next year's seminar. • Discussion on other

Moderator: Lars Drugge and Bengt Jacobson, VDCA and SVEA

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11:00-12:30	Demo tour	Mattias Hjort, VTI
12:30-13:30	Light lunch with networking and manned posters in seminar room.	

Micro presentations of posters (3-5 min each):

- Virtual Verification Framework for Vehicle Motion Systems* Albijon Blakqori and Mille Kotur, ZEEKR and Chalmers
- Advancing Road Vehicle Path Length Estimation via Geospatial Analytics* Yogeswaran Amsavalli, Volvo Trucks, Chalmers and Università di Trento
- Vehicle motion control interface for L4 automated driving* Rishabh Parmar and Bala Chinni, Volvo Trucks and Chalmers
- Modelling of truck tyre wear* Nathan Hedouin, Scania and ESTACA
- Trailer backup assist using steer by wire* Jakob Roempke, Chang Liu, KTH
- Pendulum turn, the optimal racing line through a hairpin curve?* Lukas Jonsson, LiU

Agenda from lunch

Agenda

Moderator: Lars Drugge and Bengt Jacobson, YDCA and S	
09:00-09:30	Coffee and registration
09:30-09:40	Welcome
09:40-11:00	<p>Session 1: Presentations (20+10 min each)</p> <ol style="list-style-type: none"> 1. <i>EV Project Roadview</i> 2. <i>Steer-by-wire</i> <p>Micro presentations of posters (3-5 min)</p> <ol style="list-style-type: none"> 1. <i>Virtual Verification Framework for Vehicle Motion Systems</i> 2. <i>Advancing Road Vehicle Path Length Estimation via Geospatial Analytics</i> 3. <i>Vehicle motion control interface for automated driving</i> 4. <i>Modelling of truck tyre wear</i> 5. <i>Trailer backup assist using steer by wire</i> 6. <i>Pendulum turn, the optimal racing line through a hairpin curve?</i>
11:00-12:30	<p>Demo tour</p> <ol style="list-style-type: none"> 1. <i>VTI's simulators & tyre test facility</i> <p>Small presentation and a visit at VTI</p>
12:30-13:30	Light lunch with networking and manned posters
13:30-14:30	<p>Session 2: Presentations (20+10 min each)</p> <ol style="list-style-type: none"> 4. <i>Energy consumption comparison of actuator coordination methods on heavy vehicles in realistic transport applications</i> 5. <i>Tyre models for virtual assessment of vehicles</i>
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16:00-16:15	<p>Wrap-up</p> <ul style="list-style-type: none"> • Feedback on present years seminar. Proposals • Discussion on other

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16:00-16:15	<p>Wrap-up</p> <ul style="list-style-type: none"> • Feedback on present years seminar. Proposals • Discussion on other 	

Posters still manned:

- | | |
|--|---|
| 1. <i>Virtual Verification Framework for Vehicle Motion Systems</i> | Albijon Blakqori and Mille Kotur, ZEEKR and Chalmers |
| 2. <i>Advancing Road Vehicle Path Length Estimation via Geospatial Analytics</i> | Yogeswaran Amsavalli, Volvo Trucks, Chalmers and Università di Trento |
| 3. <i>Vehicle motion control interface for L4 automated driving</i> | Rishabh Parmar and Bala Chinni, Volvo Trucks and Chalmers |
| 4. <i>Modelling of truck tyre wear</i> | Nathan Hedouin, Scania and ESTACA |
| 5. <i>Trailer backup assist using steer by wire</i> | Jakob Roempke, Chang Liu, KTH |
| 6. <i>Pendulum turn, the optimal racing line through a hairpin curve?</i> | Lukas Jonsson, LiU |

Presentation 1:

EU-project Roadview

Sogol Kharrazi and Mattias Hjort, VTI



Robust Automated Driving in Extreme Weather

Vehicle and Tire Models for Harsh Weather Conditions

SVEA vehicle dynamics seminar 2024

Sogol Kharrazi and Mattias Hjort, VTI



Co-funded by
the European Union



Co-funded by the European Union (grant no. 101069576) and supported by Innovate UK (contract no. 10045139) and the Swiss State Secretariat for Education, Research and Innovation (contract no. 22.00123). Views and opinions expressed are however those of the author(s) only and do not necessarily reflect those of the European Union or the European Climate, Infrastructure and Environment Executive Agency (CINEA). Neither the European Union nor the granting authority can be held responsible for them.

Roadview project objective



ROADVIEW is an EU-funded Horizon Europe Innovation Action aiming to develop robust and cost-efficient in-vehicle perception and decision-making systems for connected and automated vehicles with enhanced performance under harsh weather conditions and different traffic scenarios.



Impact of harsh weather conditions

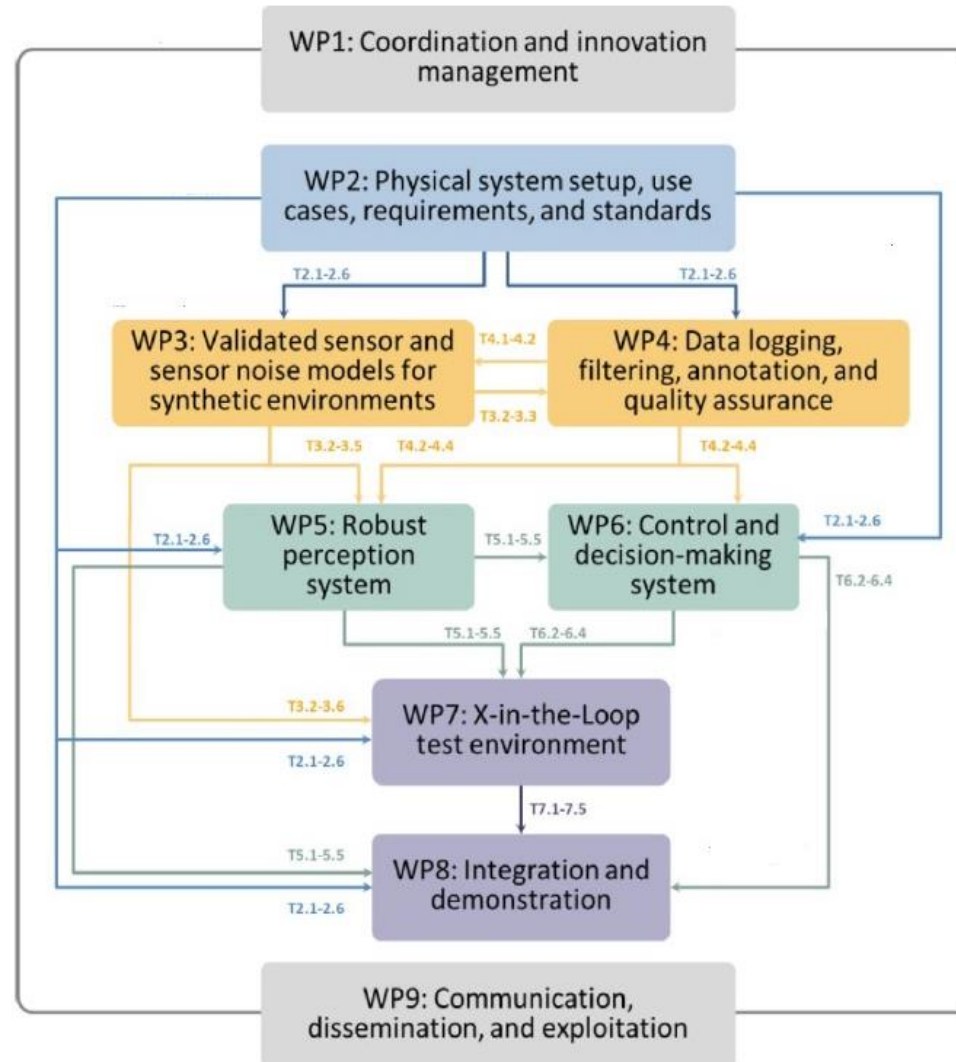


The challenges for automated driving systems caused by harsh weather conditions, such as fog, rain, and snow, are substantial, as these affect the functioning of their key technologies and their development:

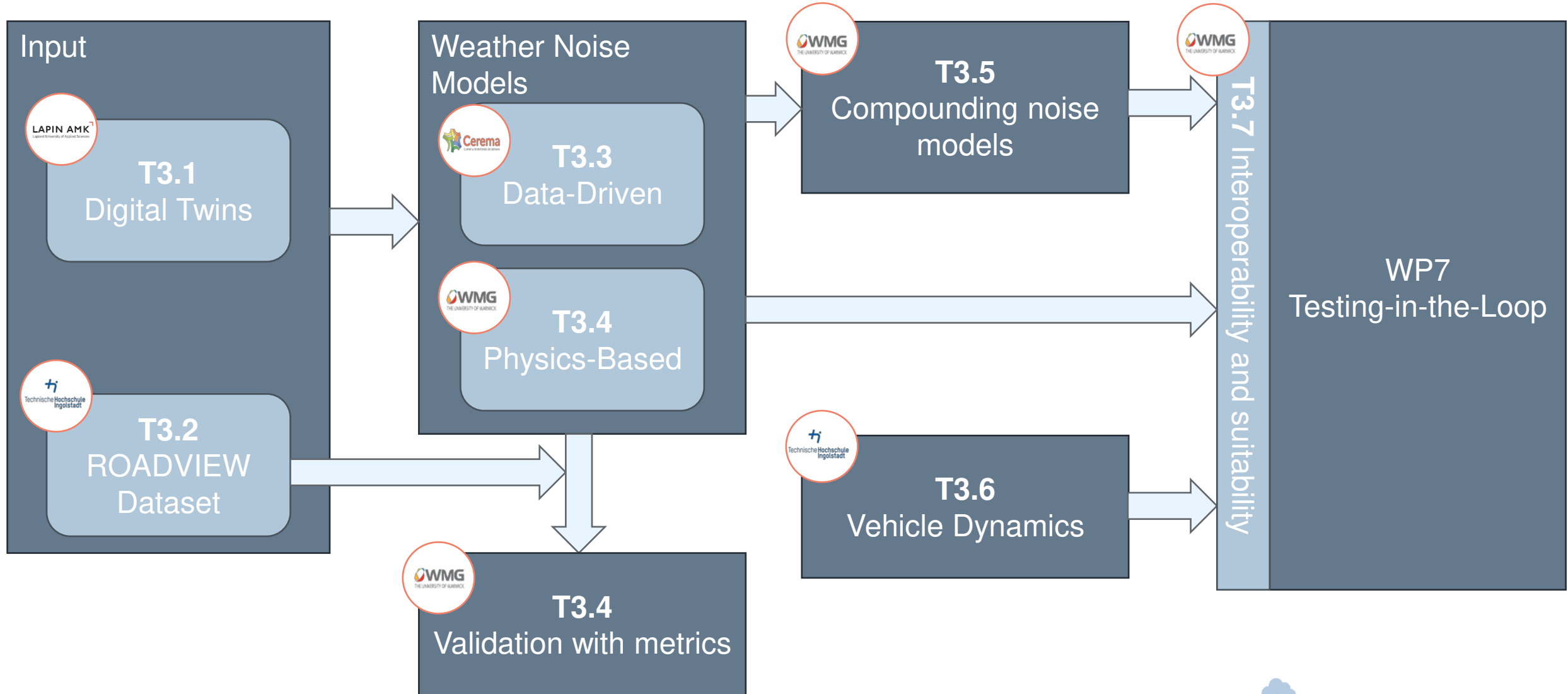
- Sensors
- Detection
- Control
- System testing



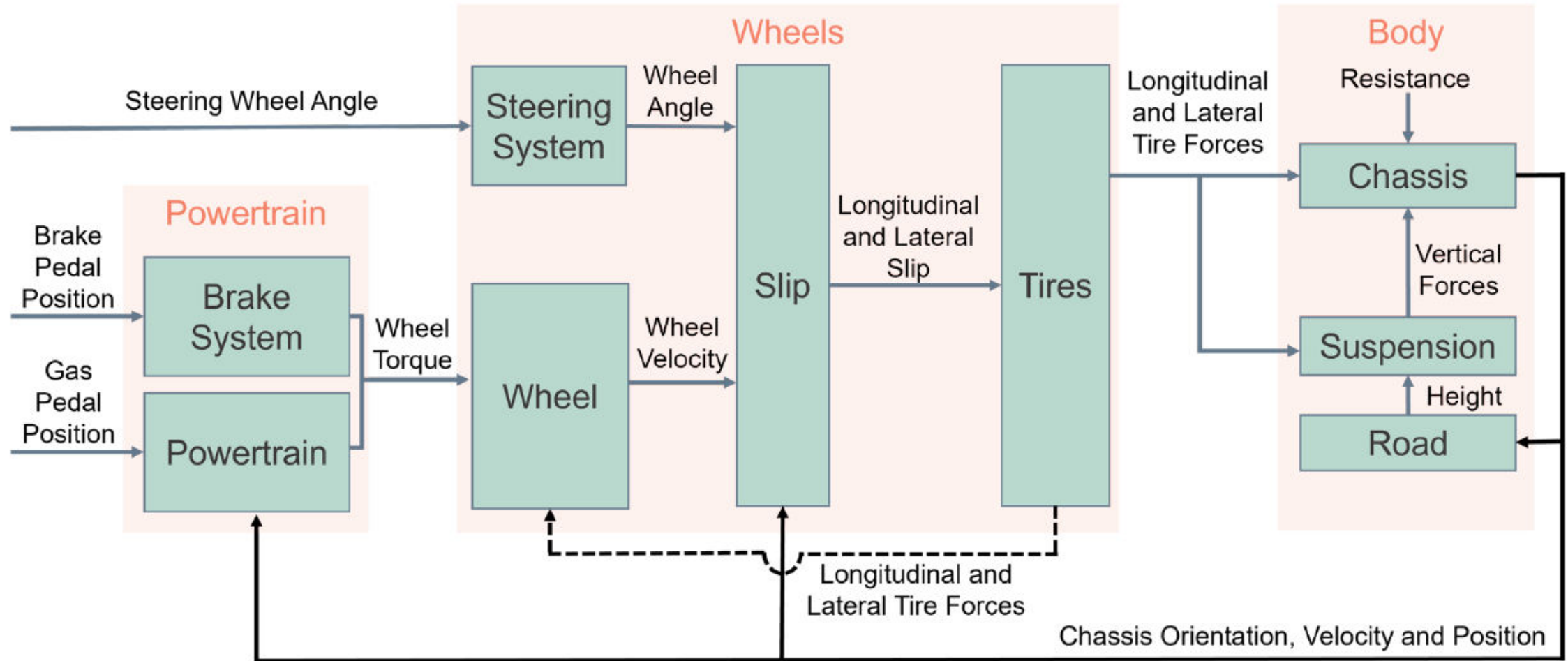
Roadview methodology



WP3 – Sensor and vehicle models



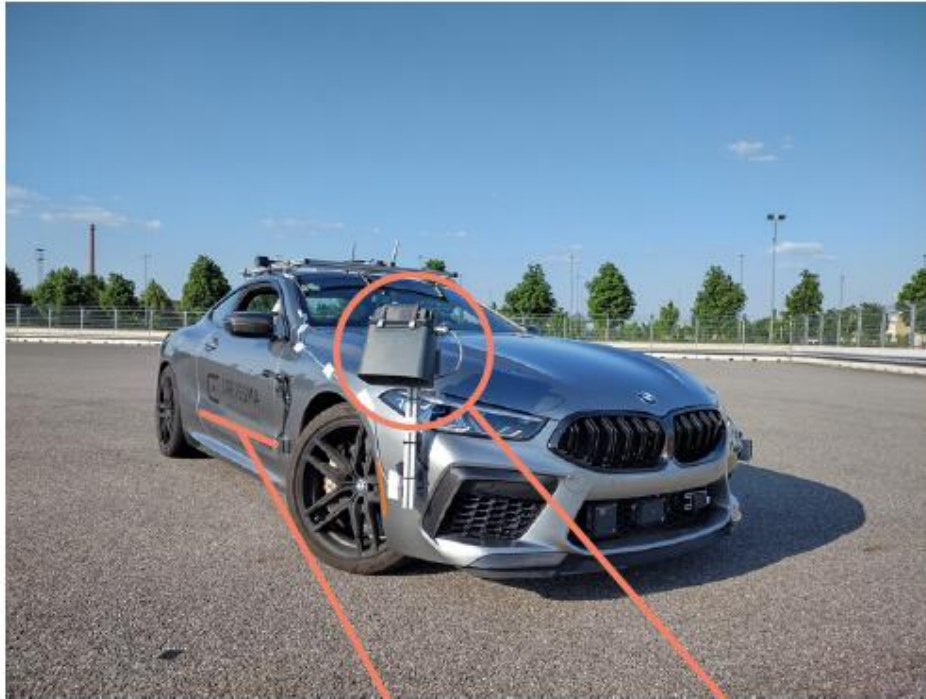
Vehicle model



Data collection for vehicle parameterization

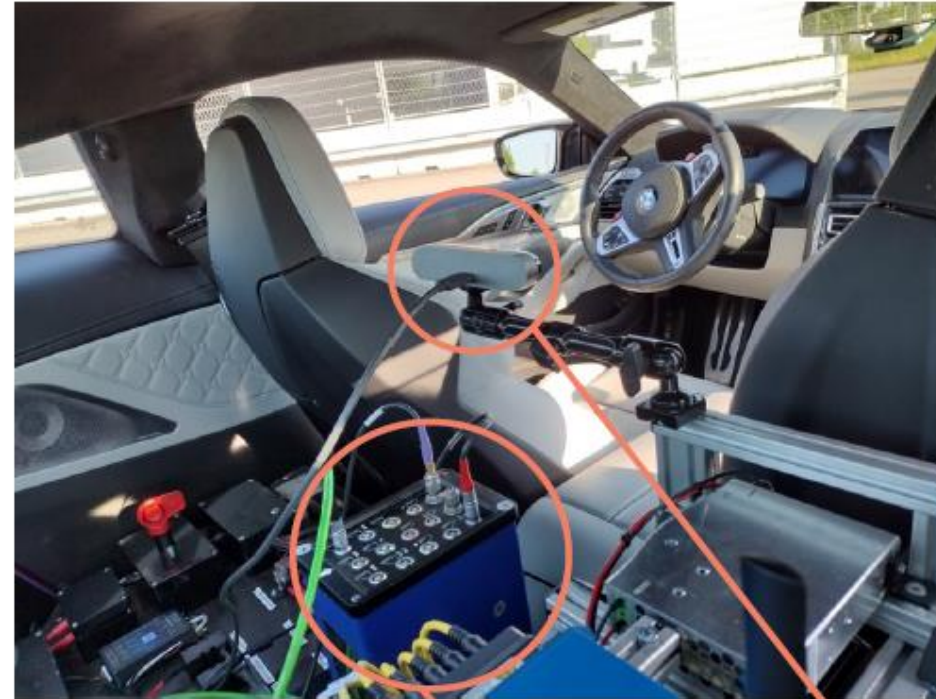


Test vehicle, BMW M8



Vehicle CAN

Marwis



GNSS + IMU

Onboard camera

Data collection, static measurements



Vehicle Mass



Position of the centre of Gravity



Data collection, dynamic tests



VIDEO



Robust Automated Driving in Extreme Weather



01/09/2022 - 31/08/2026



Funded by
the European Union



Funded by the European Union (grant no. 101069576). Views and opinions expressed are however those of the author(s) only and do not necessarily reflect those of the European Union or European Climate, Infrastructure and Environment Executive Agency (CINEA). Neither the European Union nor the granting authority can be held responsible for them.

UK and Swiss participants in this project are supported by Innovate UK (contract no. 10045139) and the Swiss State Secretariat for Education, Research and Innovation (contract no. 22.00123) respectively.



@roadview_eu



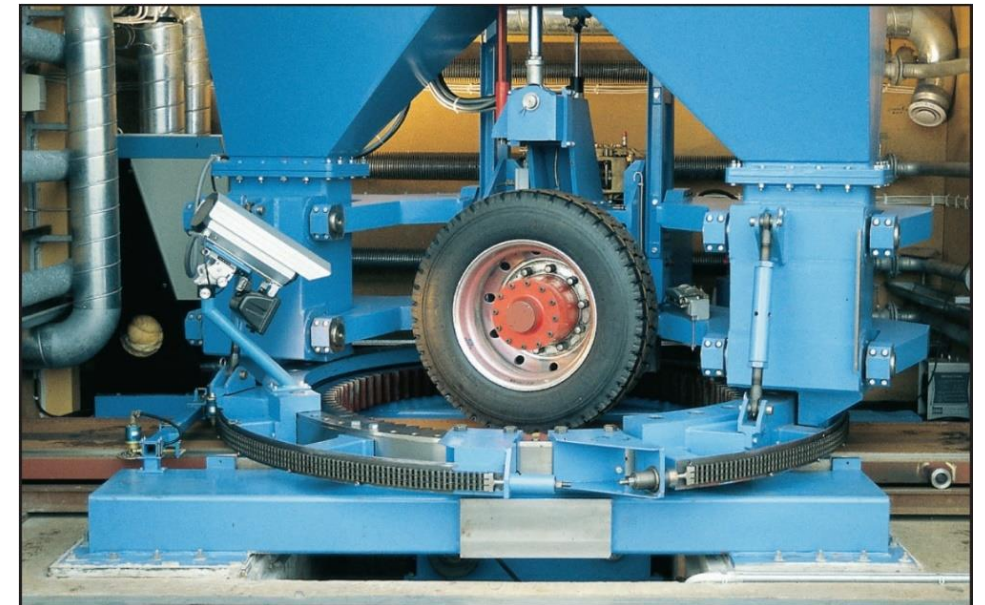
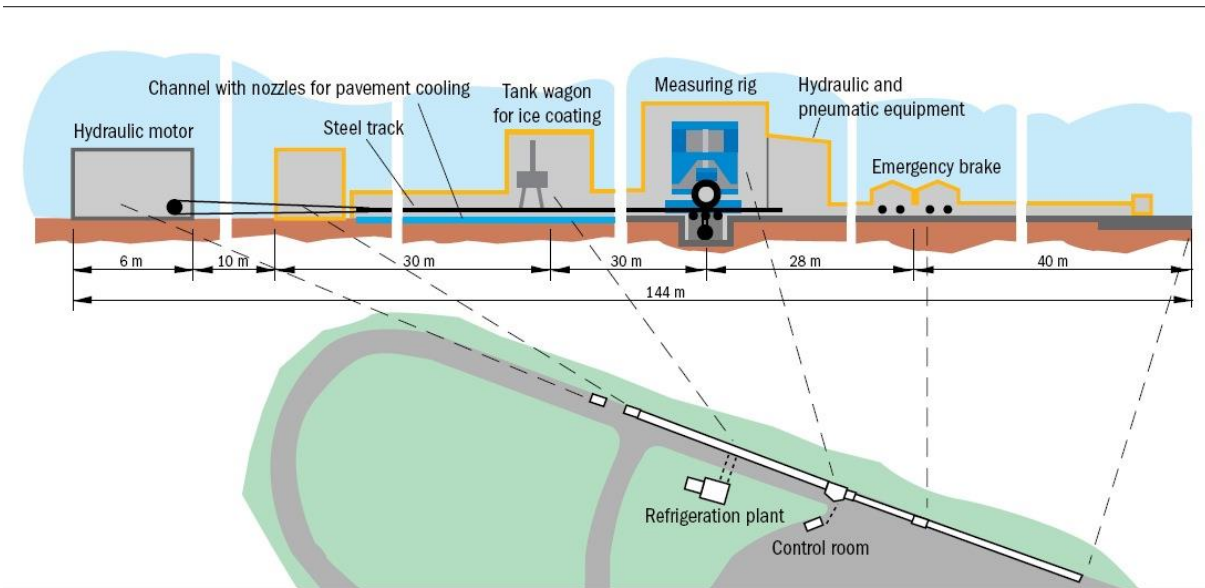
www.roadview-project.eu



Data collection, tire measurements



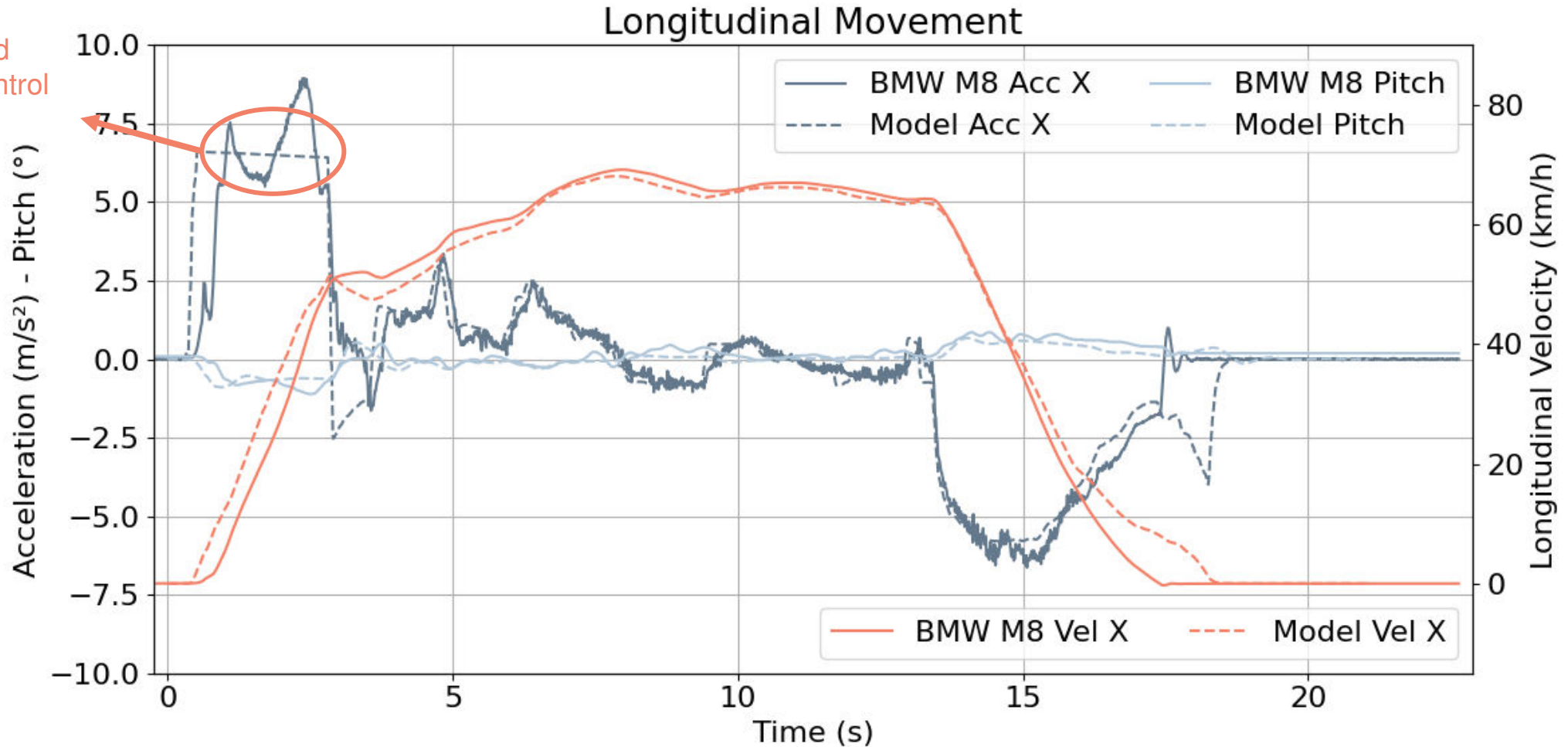
VTI tire testing facility



Vehicle model vs test data, sample results



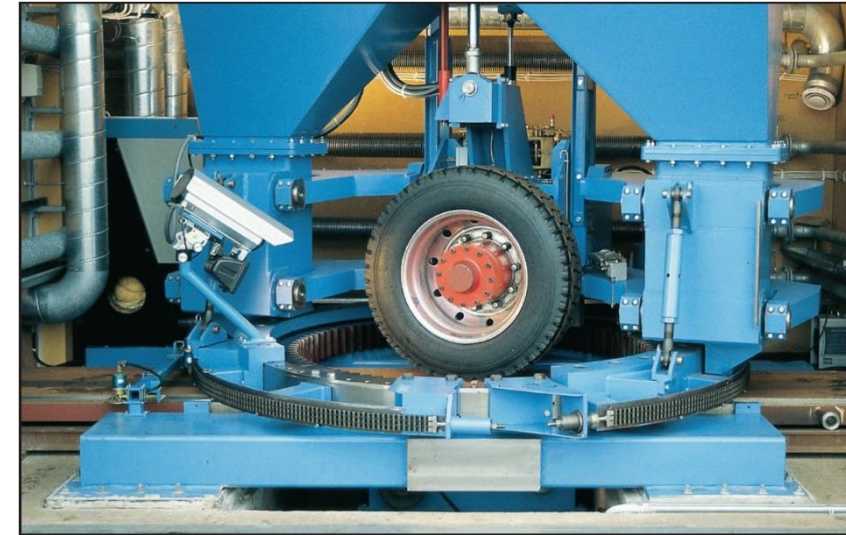
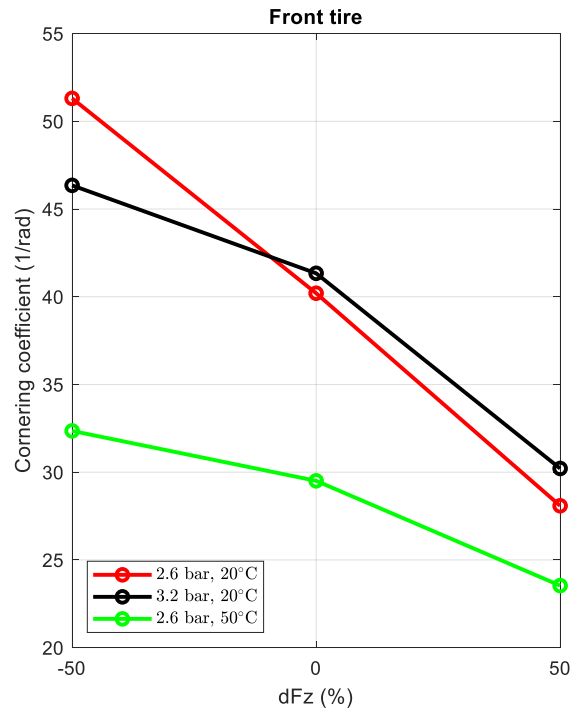
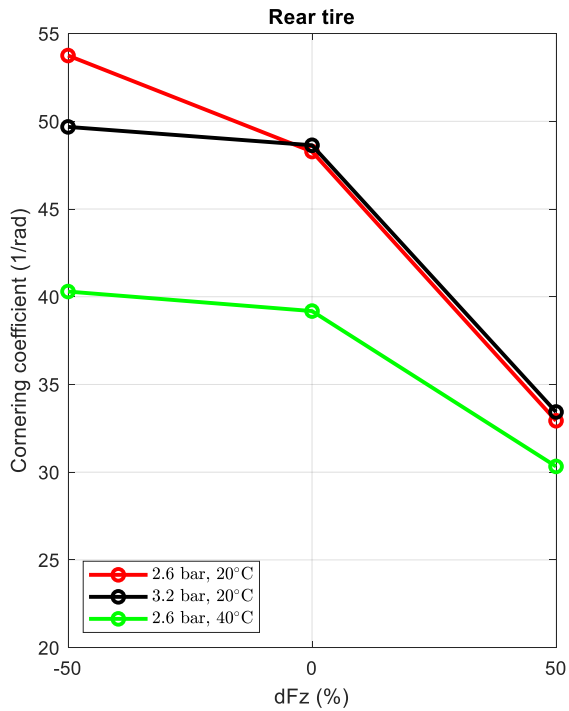
Simplified
Traction Control



Tyre model for high friction



Cornering coefficient:

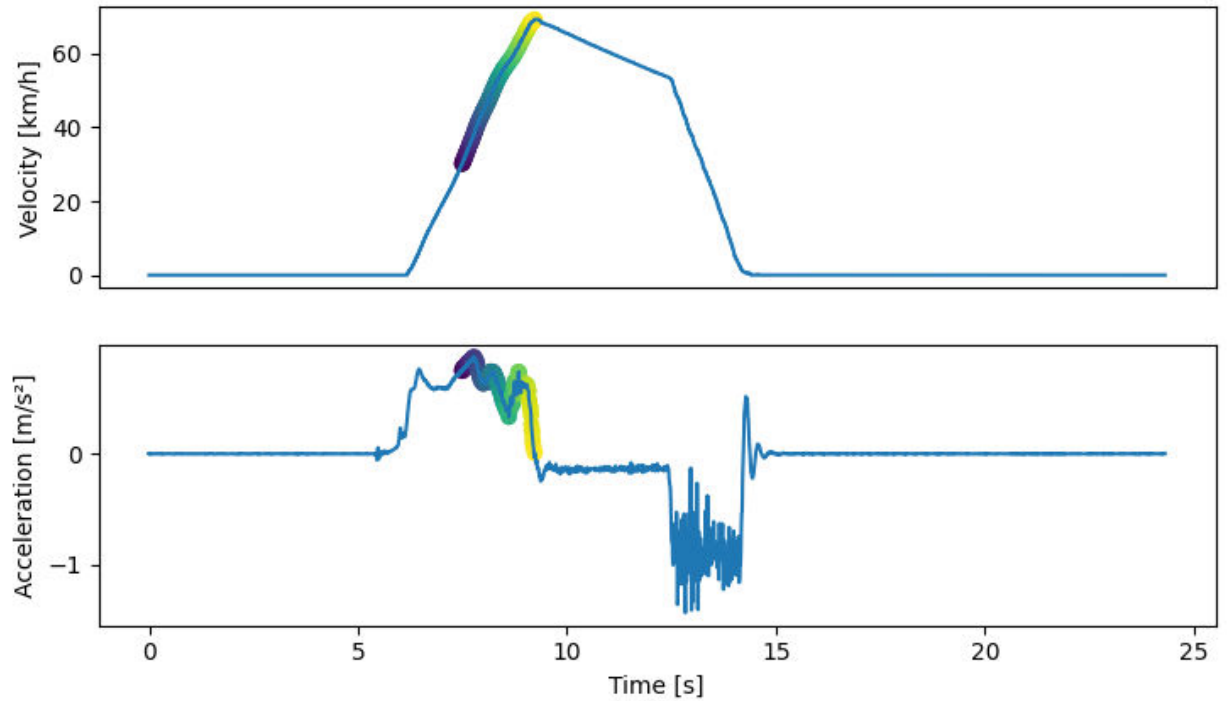
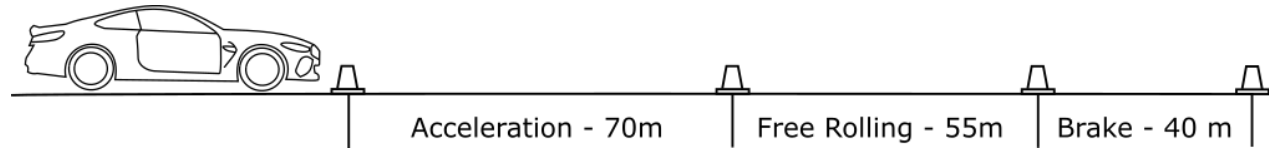


Tire	Inflation pressure (bar)	Tire temp	CC (1/rad) 50% n.l.	CC (1/rad) 100% n.l.	CC (1/rad) 150% n.l.
Front	2.6	Cold (20°C)	51.3	40.2	28.1
Front	3.2	Cold (20°C)	46.3	41.3	30.2
Front	2.6	Heated: 50°C	32.4	29.5	23.5
Rear	2.6	Cold (20°C)	53.7	48.3	32.9
Rear	3.2	Cold (20°C)	49.7	48.6	33.4
Rear	2.6	Heated: 40°C	40.3	39.2	30.3

Tyre model for high friction



Longitudinal slip stiffness:



Tyre model for ice and snow



Use data from VTI tyre measurements carried out 2015:

- 78 different winter tyres were measured on ice and snow
- 28 new tyres + 50 worn tyres
- 78 winter tyres + 8 summer tyres measured on salted cold wet asphalt (around 0 °C)

Use ISO model and/or MF model to create models at different friction levels



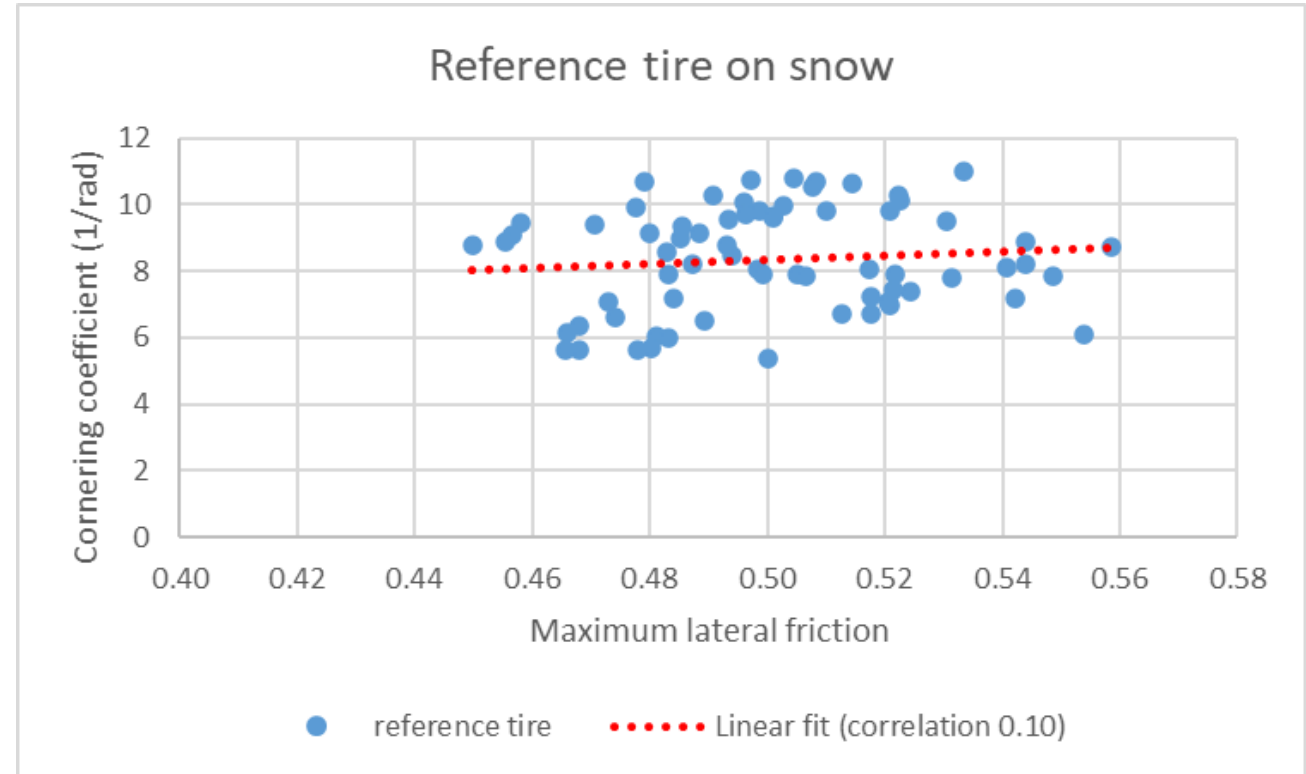
Snow



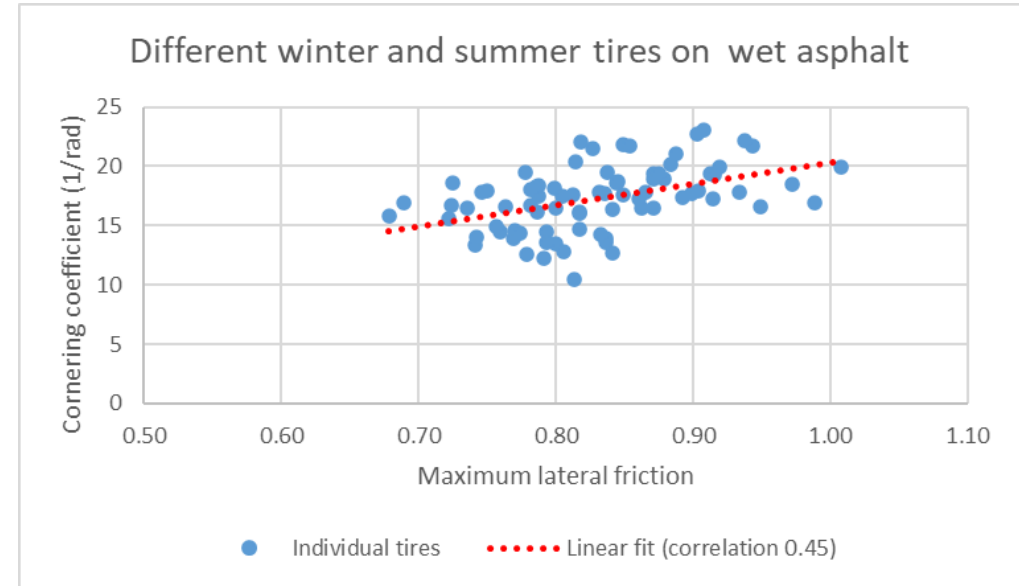
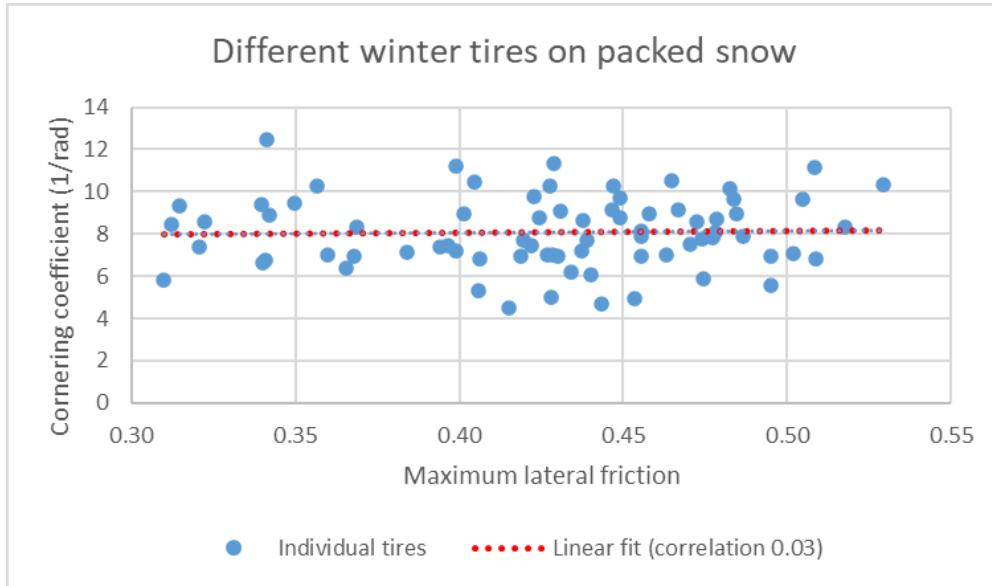
70 separate measurements with the **reference tyre**:

- Peak friction: 0.45 - 0.55
- Cornering Coefficient (CC): 5 – 10 rad^{-1}

Essentially no correlation between peak friction and CC on snow for this type of snow surface



Snow



Essentially no correlation between peak friction and CC on snow for this type of snow surface

Clear correlation on wet asphalt

Snow



Tire type	Number of tires	Wet asphalt		Snow	
		Cornering coefficient	Peak friction	Cornering coefficient	Peak friction
New studded	9	12.9	0.79	7.4	0.47
New Nordic unstudded	9	14.5	0.81	7.6	0.49
New European unstudded	9	16.7	0.87	7.6	0.46
New summer	4	17.6	0.93		
Used studded	18	17.0	0.80	8.3	0.42
Used Nordic unstudded	16	17.5	0.79	8.5	0.43
Used European unstudded	16	20.2	0.87	8.3	0.35
Used summer	4	21.5	0.89		

New tyres:

CC snow : 45-55% of CC asphalt

Worn tyres:

CC snow : 40-45% of CC asphalt

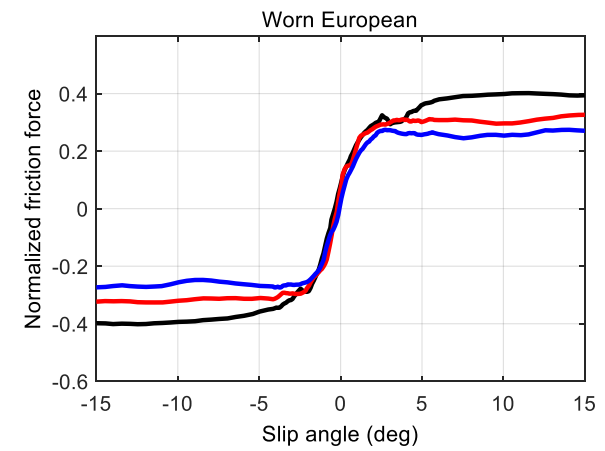
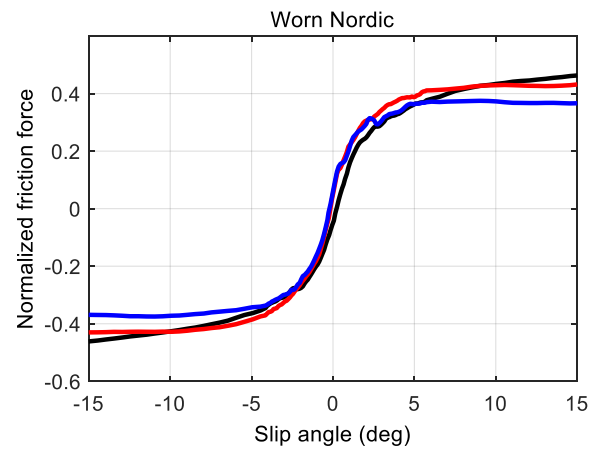
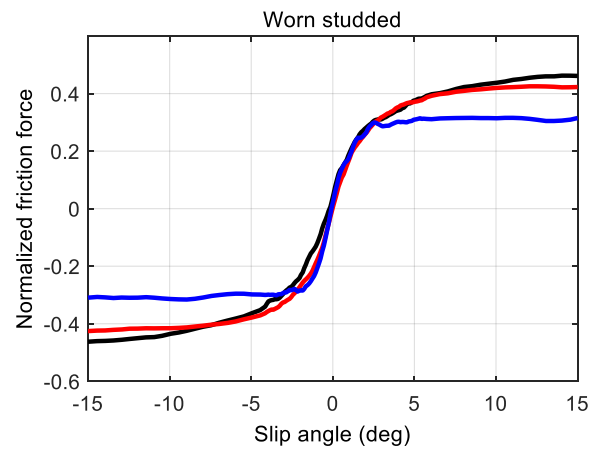
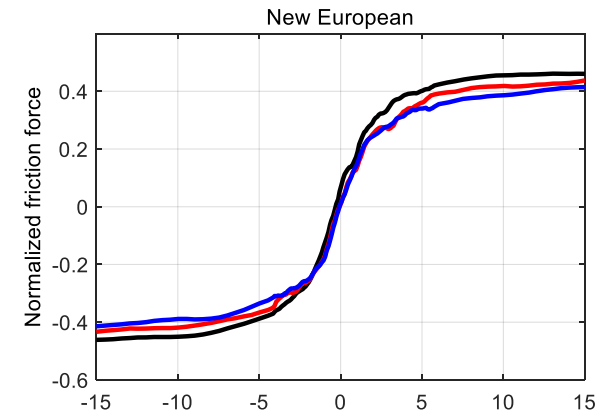
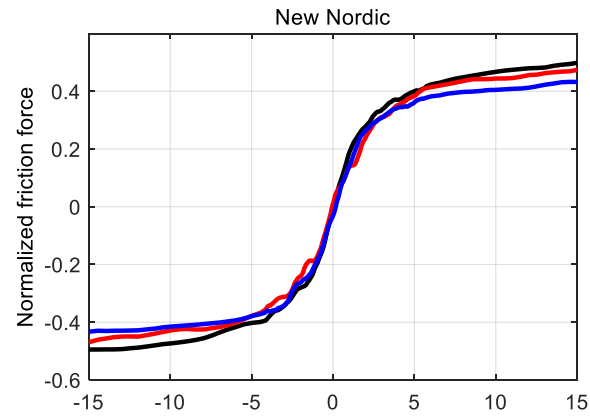
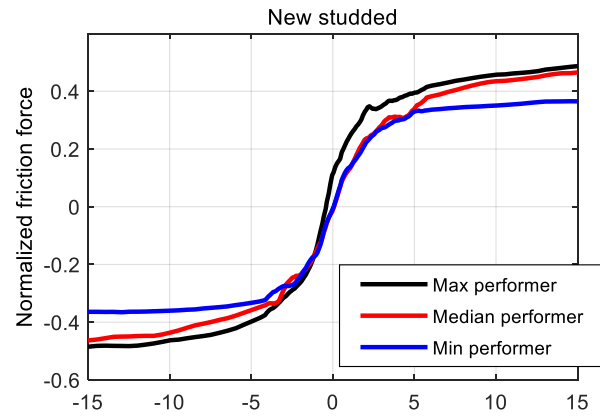
Simple assumption:

- Peak friction 0.3 – 0.5
- CC on snow 50% of CC on asphalt (dry or wet)

Snow



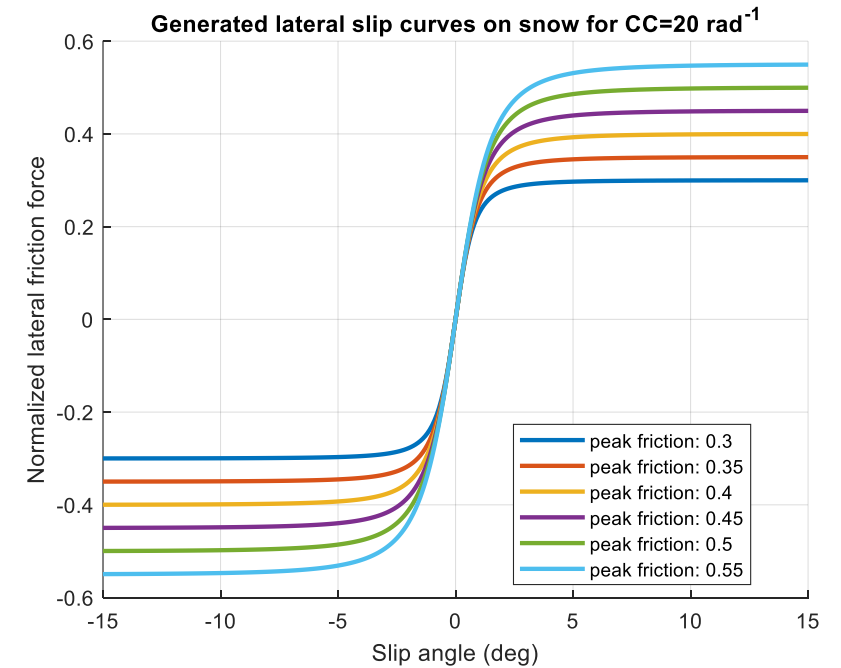
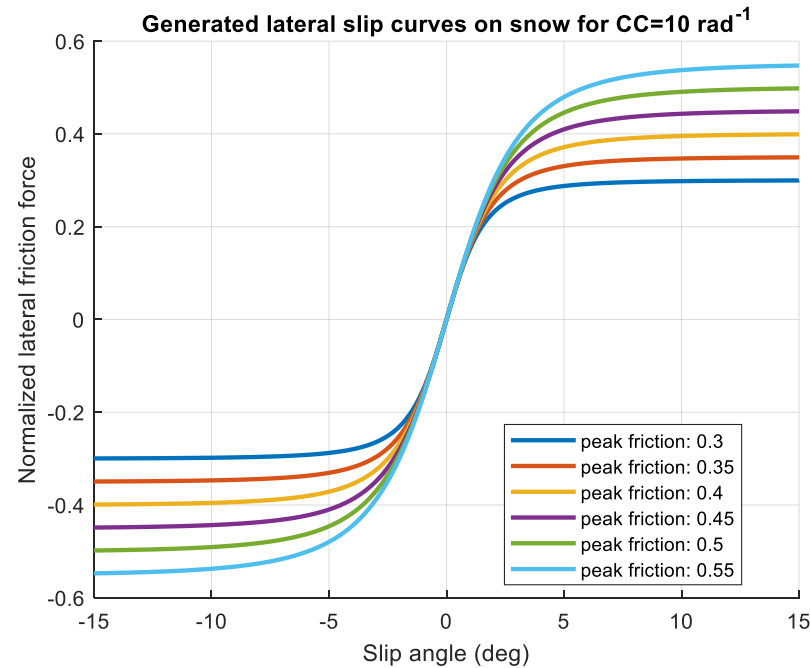
Steer tests on snow



Snow



- **Peak position:**
25 deg seems to generate similarly shaped slip curves
- **Load influence:**
Recent study on brake friction suggests a minimum influence from $\pm 30\%$ change of wheel load



Ice – lateral force



- **Peak position:**

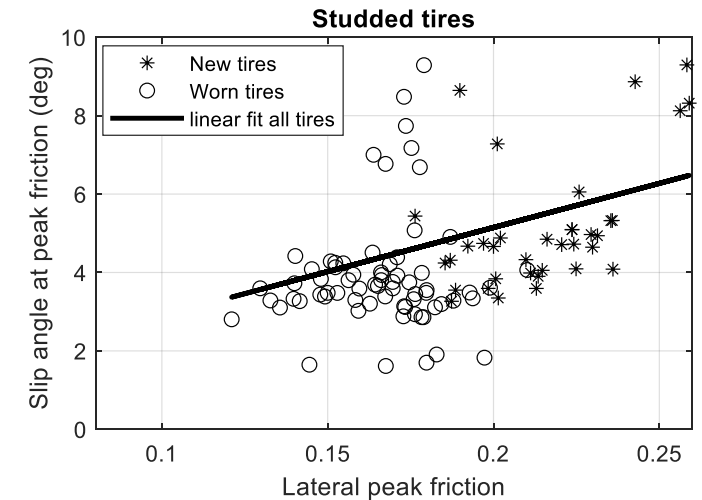
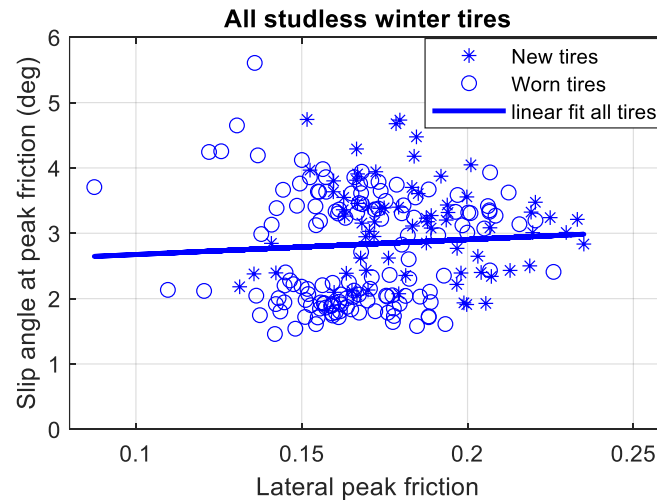
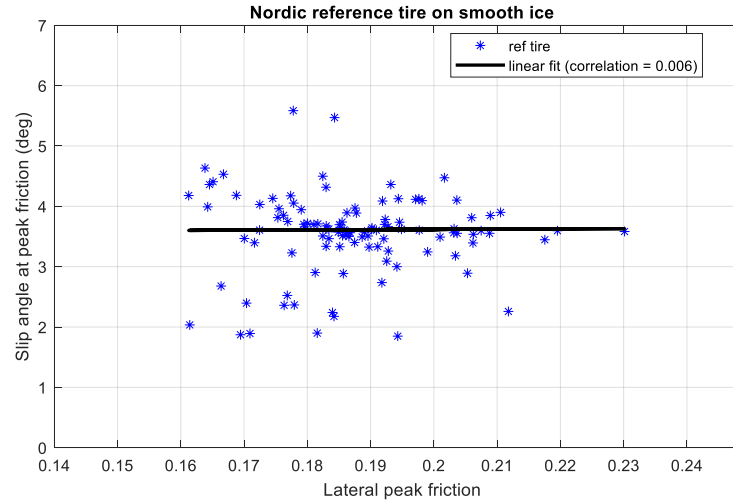
For ref tyre no correlation between peak friction and peak position

For studless tyres, no clear correlation seen.

Peak friction: 0.1 – 0.25

Peak pos: 2 – 4 deg

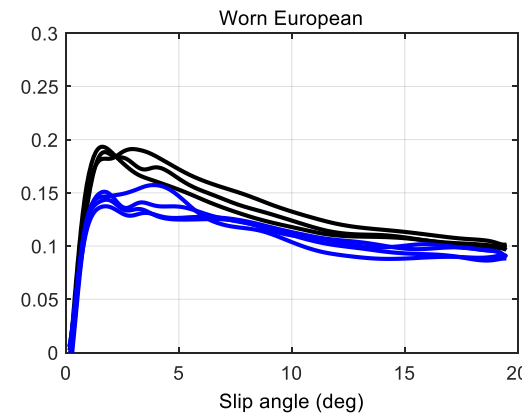
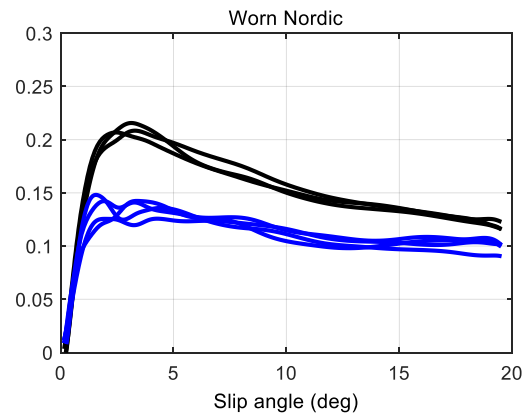
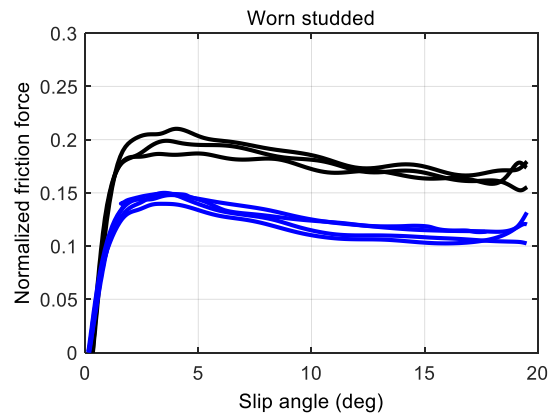
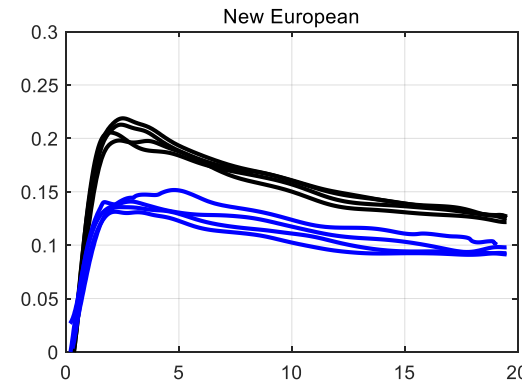
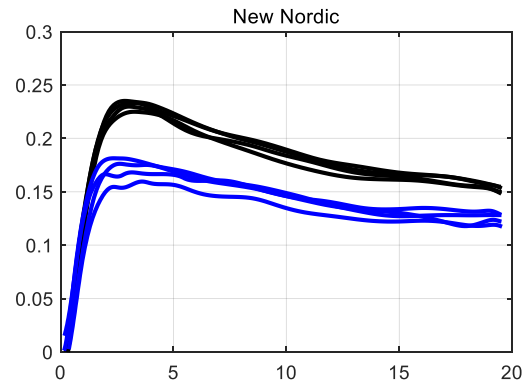
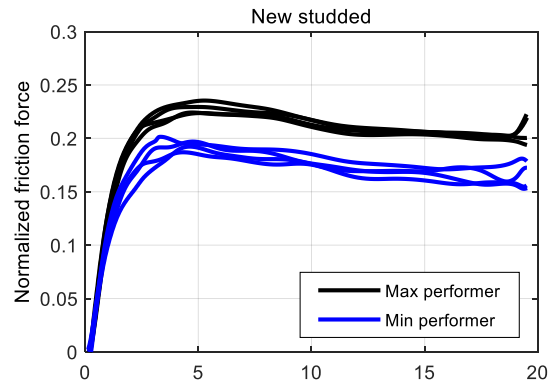
3 deg on average



Ice – lateral force



Steer tests on ice



Typical lateral slip curves

Blue: tyre with worst performance

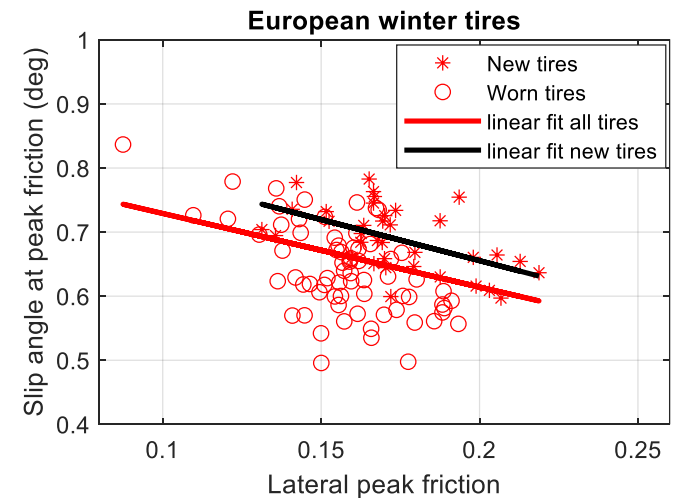
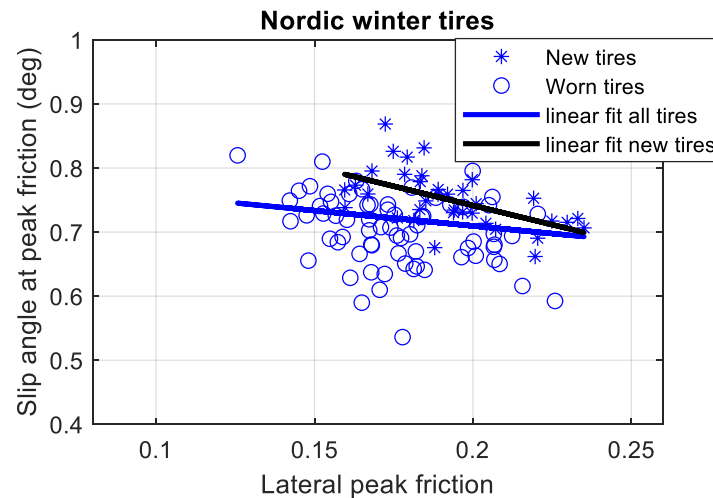
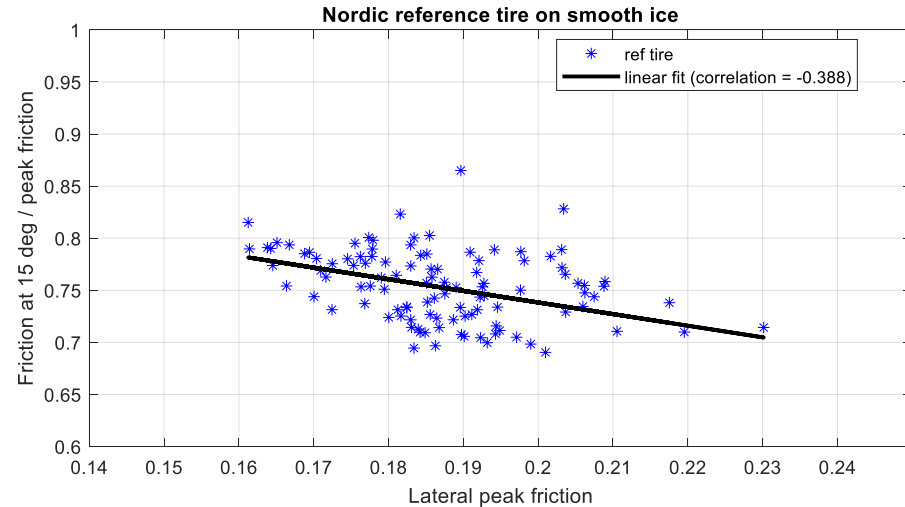
Black: tyre with best performance

Ice – lateral force



Friction ratio peak vs 15 deg slip angle

- For ref tyre the ratio becomes smaller with increasing peak friction
- For studless tyres, a similar trend.
- Worn tyres generally have a large friction drop

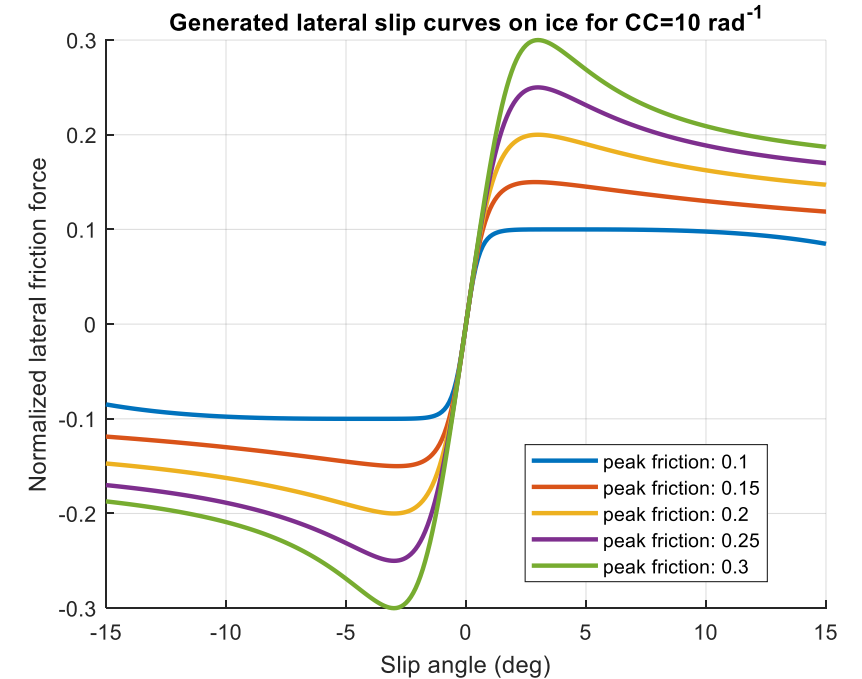
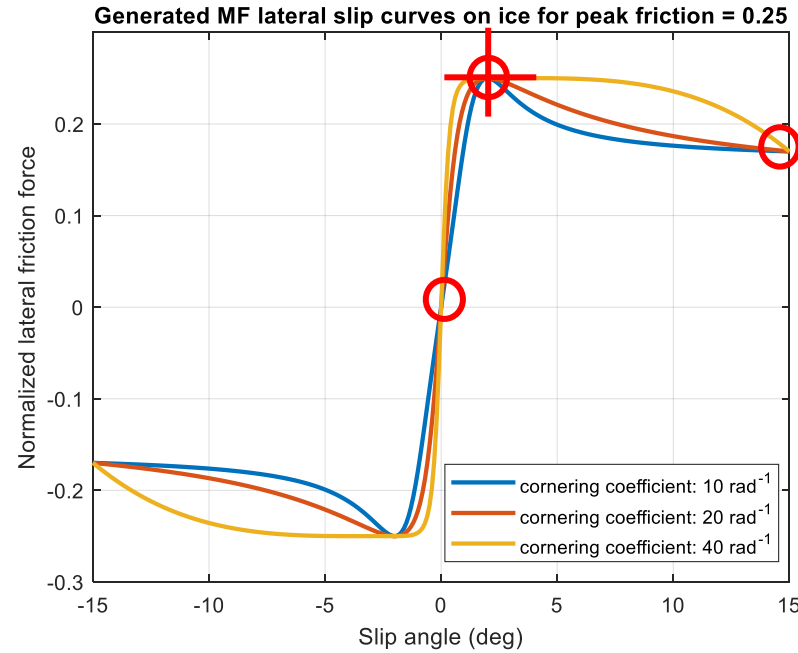


Ice – lateral force



Need the extra degree of freedom from the MF E-parameter

- Too high CC may pose a problem
- In practice, lower CC than for asphalt can be used
- Work in progress



Conclusion



- **Vehicle dynamics model can be parameterized for high friction conditions from some basic tyre measurements + measurements with selected vehicle manoeuvres**
- **Winter road conditions are introduced through tyre parameterizations (ISO or MF model) based on general trends seen for tyre measurements on new and worn tyres**

Presentation 2:
Steer-by-wire
Matthijs Klomp, Volvo Cars

Presentation 3:

VTIs simulators & tyre test facility

(Small presentation and a visit at VTI)

Mattias Hjort, VT

An aerial photograph of a city street, likely in a European city, showing a mix of old and new buildings, a river, and a boat. The street is paved and has some greenery. The river is dark and has a white boat on it. The buildings are mostly multi-story and have various roof colors and styles. The overall scene is a dense urban environment.

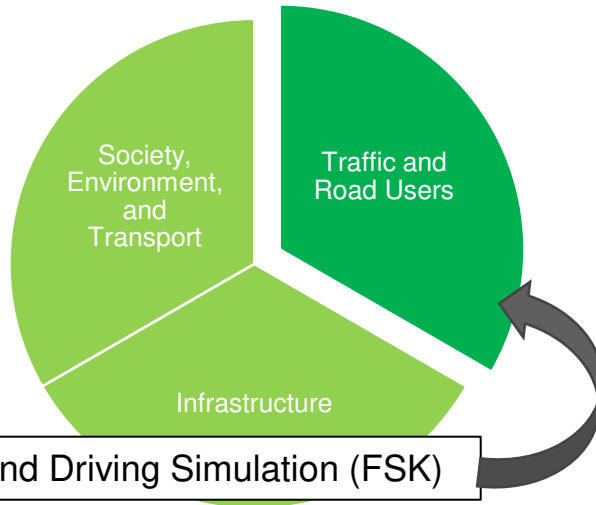
Vehicle and VRU simulators at VTI

Anders Andersson, 2024-05-17

vti

VTI

- The Swedish National Road and Transport Research Institute
- Approximately 250 employees in four office locations
- 3 research departments



Simulator benefits

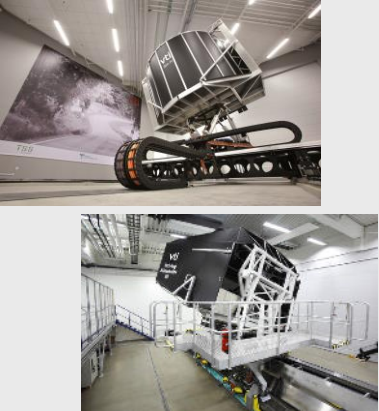

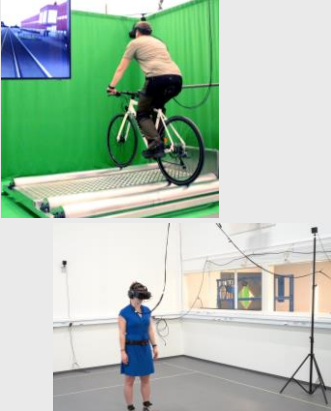

- **Repeatable**
Every test participant experience the same situation and the simulator is not affected by e.g., unexpected weather.
- **Controlled and safe**
Simulators can investigate otherwise dangerous situation that cannot be performed on test tracks or in real traffic.
- **High degree of freedom**
In “our world” there are no limits. We can create any vehicle or any environment. We are not bound by natural laws. 😊

A simulator challenge

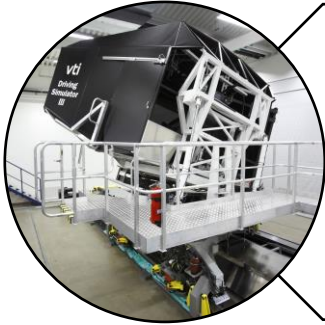
Higher degree of realism gives a higher quality of results.



Overview – Simulators at VTI

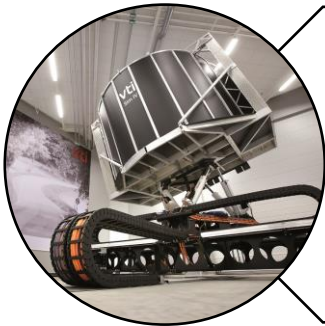
Moving base simulators	Static simulators	VRU simulators	Train simulators
<ul style="list-style-type: none">• High driving realism• Safe and repeatable• Car / truck / bus / ambulance	<ul style="list-style-type: none">• Mobile (on wheels)• Cost efficient (cheaper and more than one can be run in parallel)	<ul style="list-style-type: none">• Traffic is not only vehicles• Shared spaces	<ul style="list-style-type: none">• Passenger trains• Freight trains• Traffic management
			

The VTI moving base simulators in short



SimIII in Linköping

- Large linear motion in one degree (± 3.75 m, 8.0 m/s²)
- Pitch: -9 to 14 degrees, Roll: ± 24 degrees
- Cabin on a vibration table (e.g. road irregularities)
- Car, truck and ambulance cabins



SimIV in Göteborg

- Large XY-linear motion (± 2.5 m, 5.0 m/s²)
- Pitch and roll: ± 16.5 degrees
- Car, truck and bike cabins

The VTI Minisims in short



With screens

- One or more screens for the environment
- Sensodrive steering wheel motor and pedals
- Cabin on small wheels for transport
- Screens can be folded in during transport



With XR

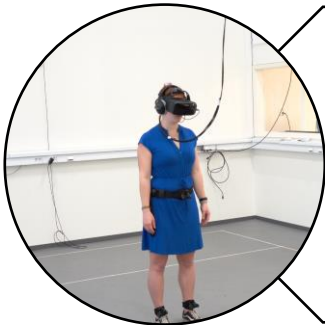
- Varjo XR 3
- Sensodrive steering wheel motor and pedals
- Cabin on small wheels for transport
- Green screen can be moved with a small truck

The VTI Bike and pedestrian simulators in short



Bike simulator

- Varjo XR 3
- Three 2 m long rollers connected by chain
- Green screens can be easily transported
- Can be used with any bike



Pedestrian simulator

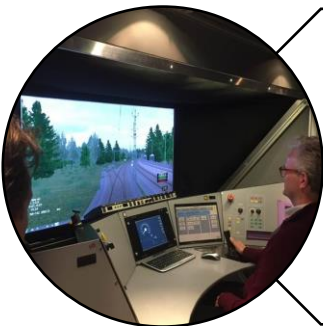
- Varjo VR 3
- About 4x7m of walking space
- Extra sensors for body positions (motion capture)

The VTI train simulators in short



Portable train simulator

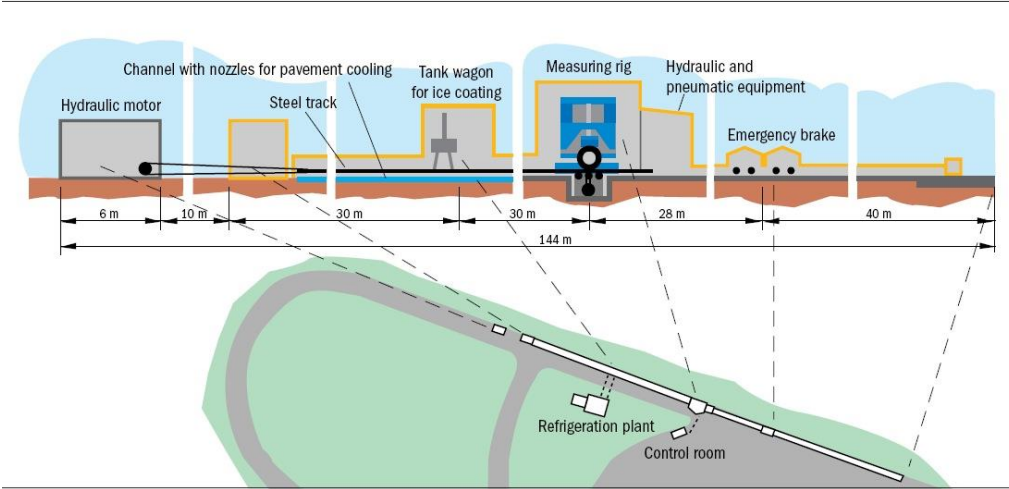
- Train driver training
- Portable levers and one computer solution
- Used by many operators



Train simulator

- Can be mounted on moving platform
- Regina interface

Tyre measurement equipment



Presentation 4:

***Energy consumption comparison of actuator
coordination methods on heavy vehicles in
realistic transport applications***

Sachin Janardhanan, Volvo Trucks and
Chalmers

Presentation 5:

***Tyre models for virtual assessment of heavy
vehicles,***

Niklas Fröjd, Volvo Trucks, Jolle Ijkema, Scania,
and Sogol Kharrazi, VTI

Tire Models for Virtual Assessment of Heavy Vehicles

Niklas Fröjd, Volvo Trucks

Jolle Ijkema, Scania

Sogol Kharrazi, VTI



The not so magic formula

an outcome from PBS2 project



ISO 23373:2024

Heavy commercial vehicles and buses —
Vehicle dynamics simulation and
validation — Tyre model for lateral
estimation of heavy vehicle combinations
operated at dry paved road surface

Published (Edition 1, 2024)



Scope

- ✓ Commercial vehicle combinations
- ✓ Two-track models primarily, low and high centre of gravity
- ✓ Lateral force modelling for lateral stability performance (yaw plane, roll plane)
- ✓ Insignificant longitudinal forces assumed
- ✓ Dry paved surface
- ✓ Moderate friction utilization (typically below peak friction)
- ✓ Covered tyre load (*vertical tyre force*) from wheel-lift to double load
- ✓ Relaxation and non-linearities included to not underestimate safety risks
- ✓ Magic Formula relation
- ✓ Camber, asymmetry, and combined slip neglected
- ✓ Nominal* tyre load main scaling parameter
- ✓ All parameters are understandable characteristics, all specified at nominal load
- ✓ 5 parameters characterize steady-state properties
- ✓ 2 parameters characterize transient properties (relaxation)
- ✓ Parameters possible to estimate from limited test efforts at around nominal tyre load
- ✓ Standard ISO parameters available for missing test data

** Rated tyre load often used as the nominal tyre load of the model*

Steady-state model

Main scaling parameters

- F_{Z0} , Nominal tyre load [N]

Steady-state characterization, at nominal load

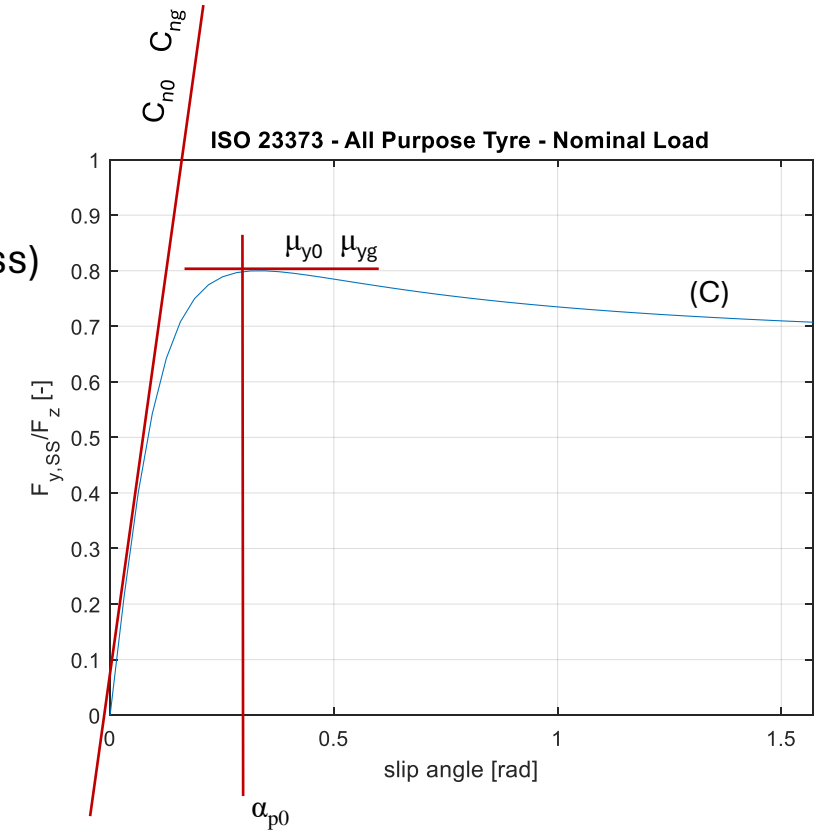
- C_{n0} , Cornering coefficient [1/rad] (normalized cornering stiffness)
- C_{ng} , Cornering coefficient gradient [-]
- μ_{y0} , Peak friction [-]
- μ_{yg} , Peak friction gradient [-]
- α_p , Slip angle at peak lateral friction [-]

$$F_{Y,SS} = F_Z \mu_y \sin \left[C \operatorname{atan} \left(\frac{C_n}{C \mu_y} \alpha \right) \right]$$

SS for steady-state

$$\text{where } C \text{ from } \alpha_{p0} = \left(\frac{C \mu_{y0}}{C_{n0}} \right) \tan \left(\frac{\pi}{2C} \right)$$

$$C_n = C_{n0} \left(1 + C_{ng} \frac{F_Z - F_{Z0}}{F_{Z0}} \right) \quad \mu_y = \mu_{y0} \left(1 + \mu_{yg} \frac{F_Z - F_{Z0}}{F_{Z0}} \right)$$



Transient model

Transient characteristics, at nominal load

- σ_{y0} , Relaxation length [m]
- σ_{yg} , Relaxation gradient [-]

$$\frac{\sigma_y}{v_{XT}} \dot{F}_Y + F_Y = F_{YT,SS}$$

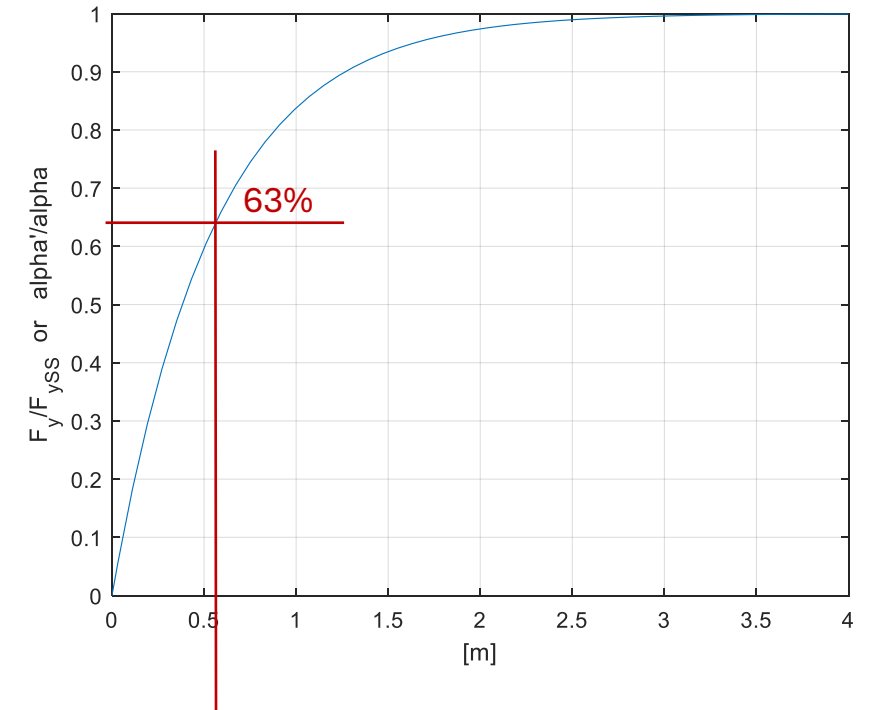
$$\sigma_y = \sigma_{y0} \left(1 + \sigma_{yg} \frac{F_Z - F_{Z0}}{F_{ZT0}} \right)$$

option - relaxing slip angle

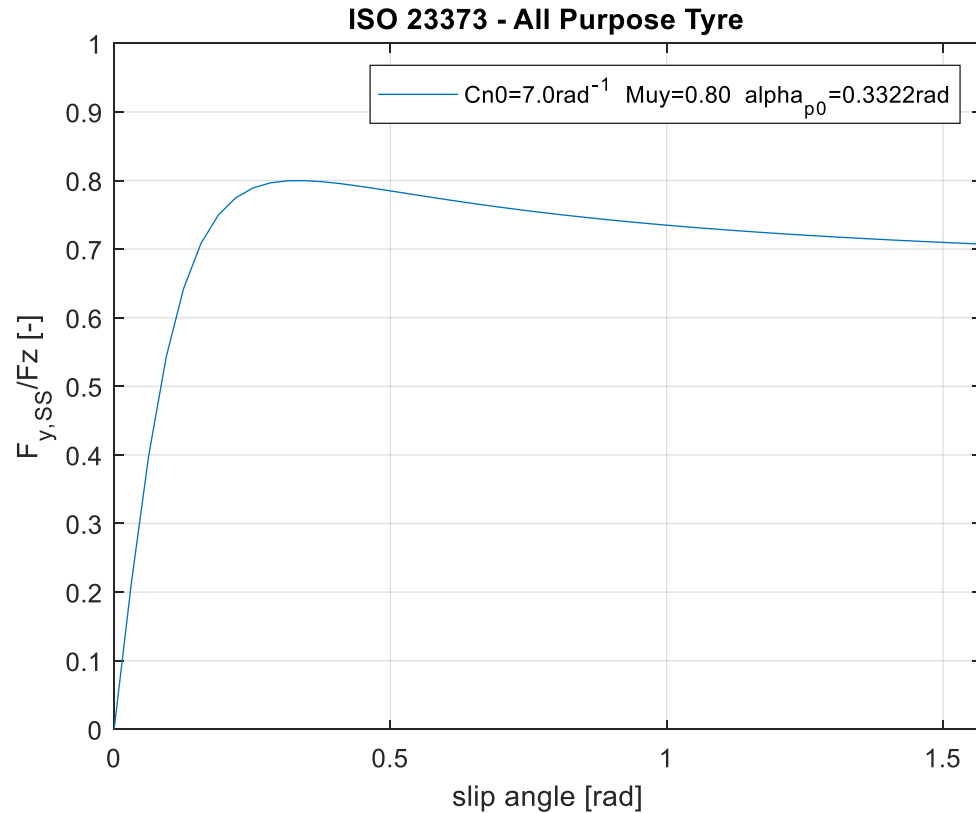
$$\frac{\sigma_y}{v_{XT}} \alpha' + \alpha' = \alpha$$

$$F_Y = F_Z \mu_y \sin \left[C \operatorname{atan} \left(\frac{C_n}{C \mu_y} \alpha' \right) \right]$$

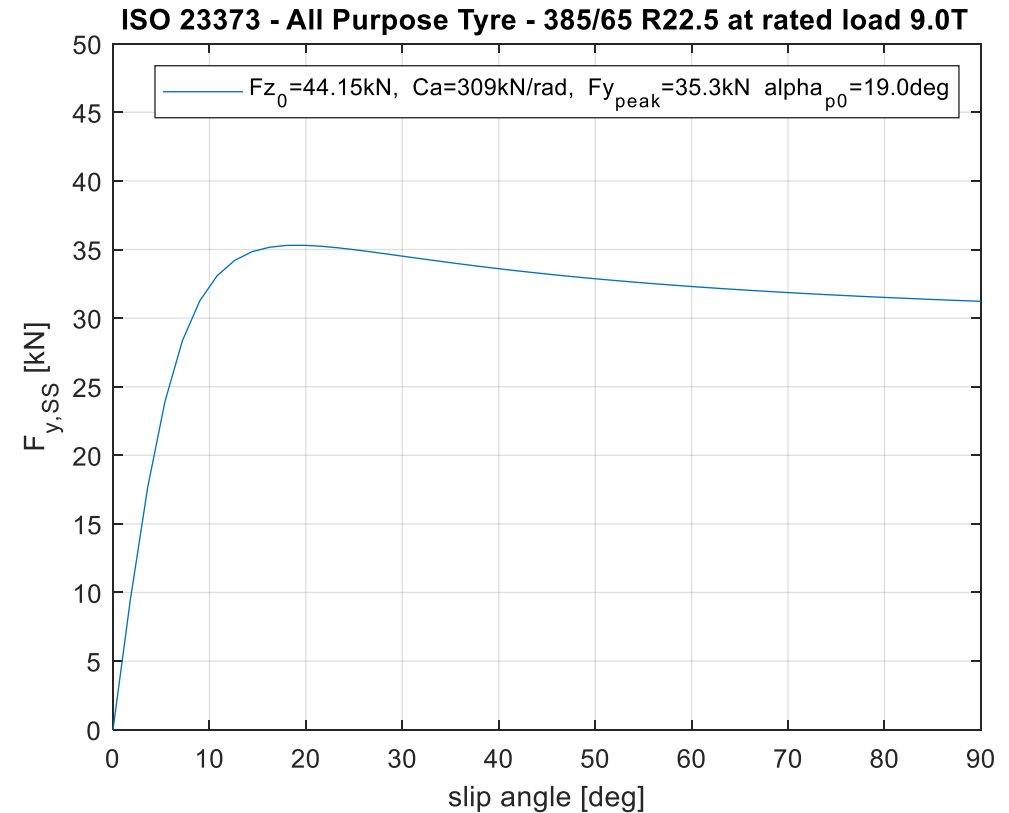
Relaxation length – distance the tyre must roll to build up 63% of the final value



All-purpose tyre



C_{n0} , Nominal cornering coefficient	7,0 rad^{-1}
C_{ng} , Cornering coefficient gradient	-0,2
μ_{y0} , Nominal peak friction	0,8
μ_{yg} , Peak friction gradient	-0,15
α_p , Slip angle at peak lateral friction	0,3322 rad
(C, shape factor	1.4)



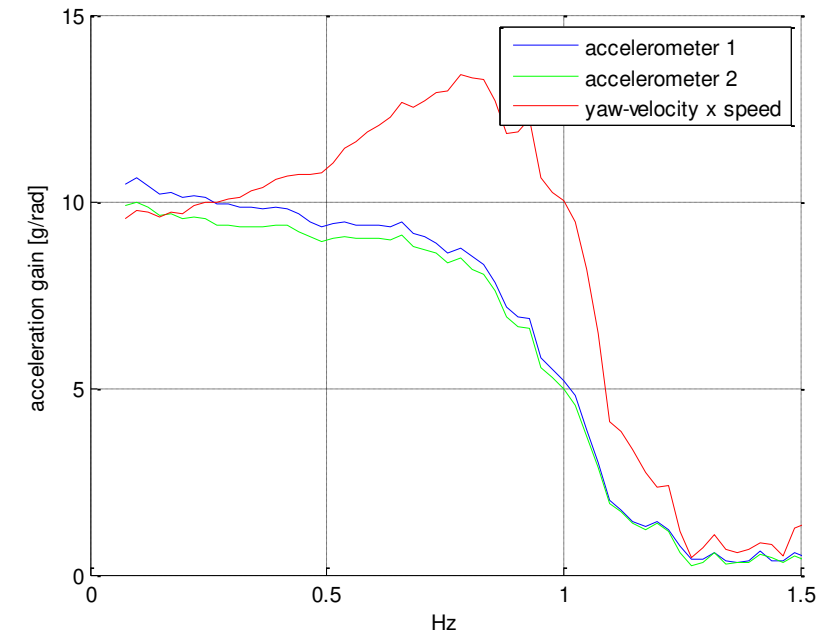
Cornering stiffness	309 kN/rad
Peak lateral force [-]	35,3 kN
Slip angle at peak lateral friction	19,0 deg

Example, parameter estimation with limited effort

- by pseudo random steering testing -



- Accelerometer on trailer axle
- Optical slip sensor on trailer axle
- Axle weight measured on weigh scales
- Option yaw-rate sensor on trailer axle



Frequency response $a_y/\alpha \rightarrow$

- Direct cornering coefficient $\approx 9.5 \pm 0.5$ g/rad (at ~ 0 Hz)
- Relaxation length from phase angle (at ~ 0 Hz)



Test vehicle





Tire – varying thread depth

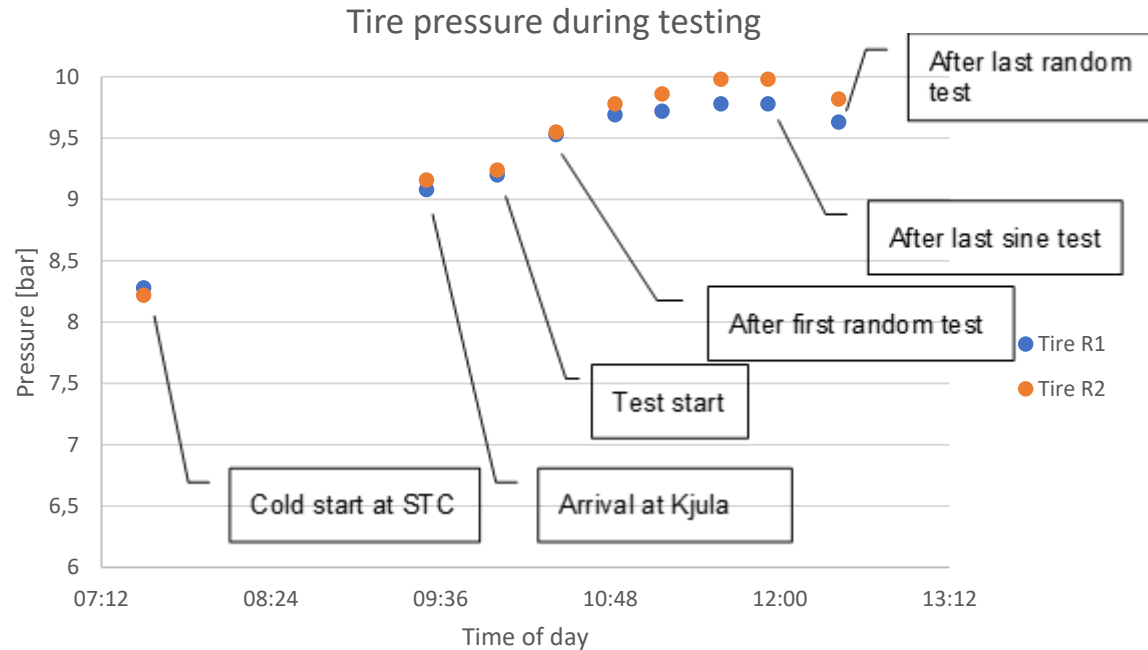
Michelin XTE3 385/65R22.5



		NewNew		WornNew		WornWorn	
		Left	Right	Left	Right	Left	Right
Dolly	Axle 1	17 mm	17 mm	7 mm	6 mm	7 mm	6 mm
Dolly	Axle 2	17 mm	17 mm	11 mm	10 mm	11 mm	10 mm
Trailer	Axle 3	17 mm	17 mm	17 mm	17 mm	5 mm	4 mm
Trailer	Axle 4	17 mm	17 mm	17 mm	17 mm	5 mm	5 mm
Trailer	Axle 5	17 mm	17 mm	17 mm	17 mm	4 mm	8 mm










Tire pressure – "9 bar"



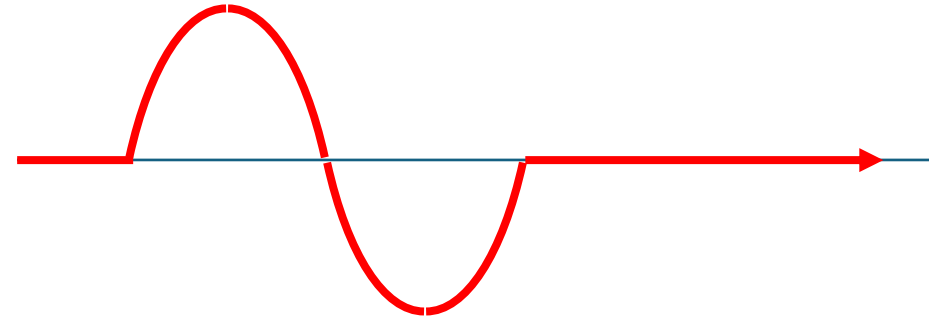
Ideal gas law:

$$\frac{pV}{T} = nR$$

	215/50R18 92V			
				
				
kPa	250	250	260	350
(bar)	(2.5)	(2.5)	(2.6)	(3.5)
<psi>	<36>	<36>	<38>	<51>
	 = 75 kg			(DB3CA)



Test method – single sine input

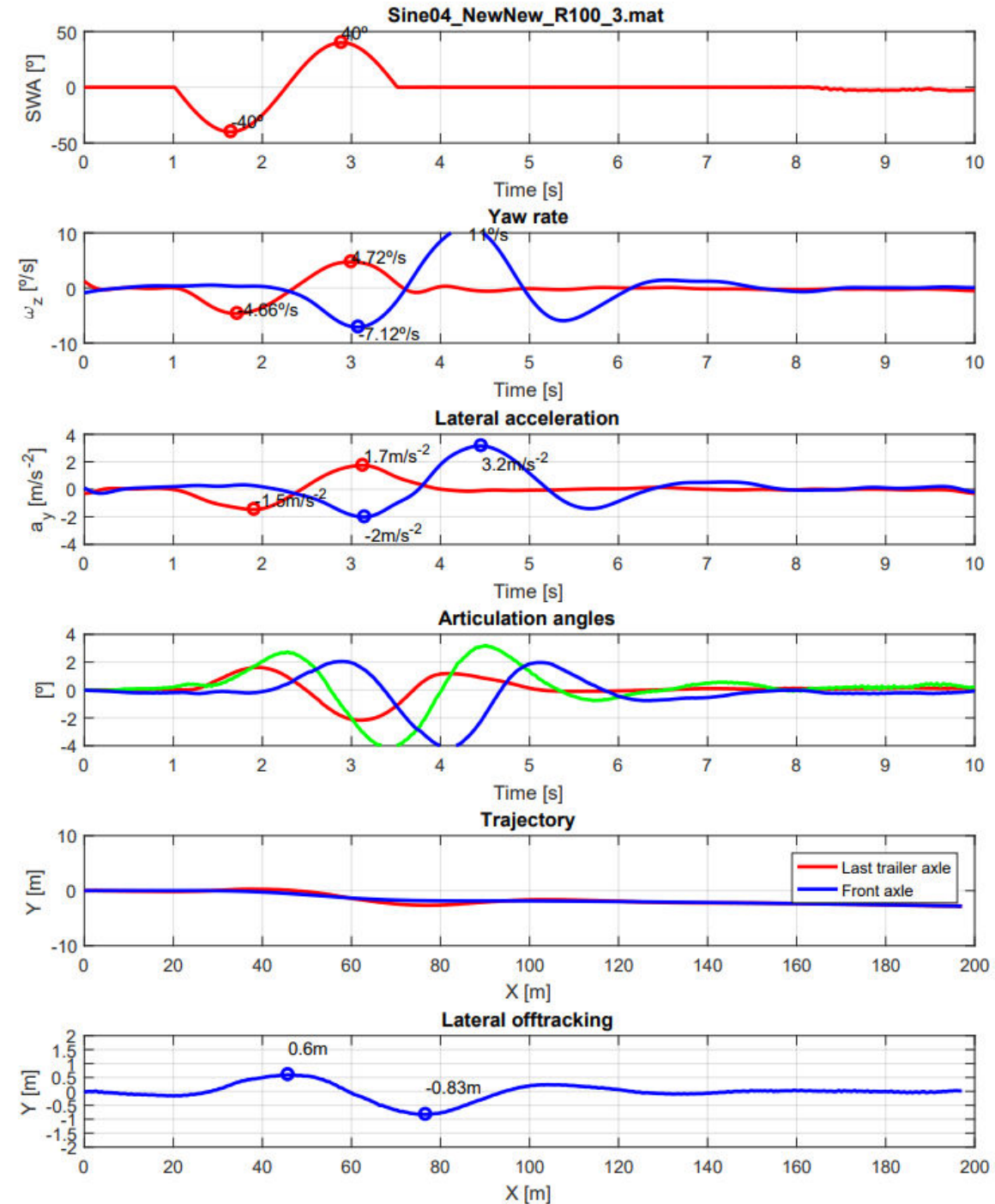
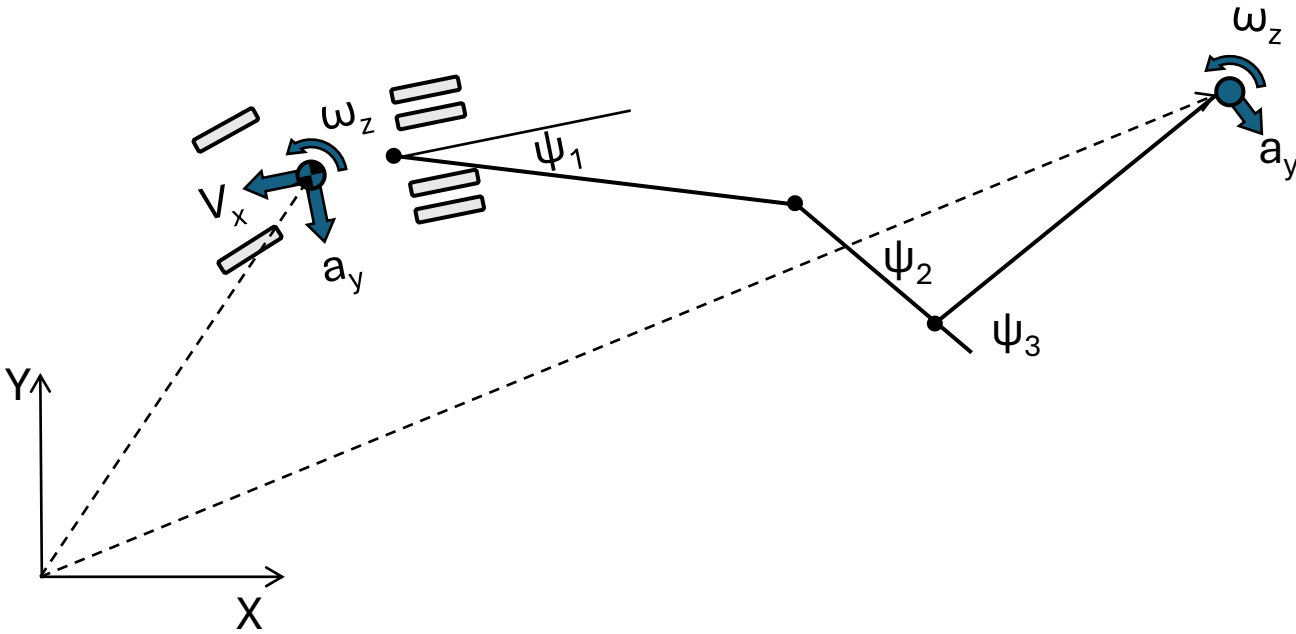


$$\delta = \hat{\delta} \sin 2\pi f t$$

$$\hat{\delta} \in [40\% \ 60\% \ 80\% \ 100\%] \propto 2^m / s^2$$

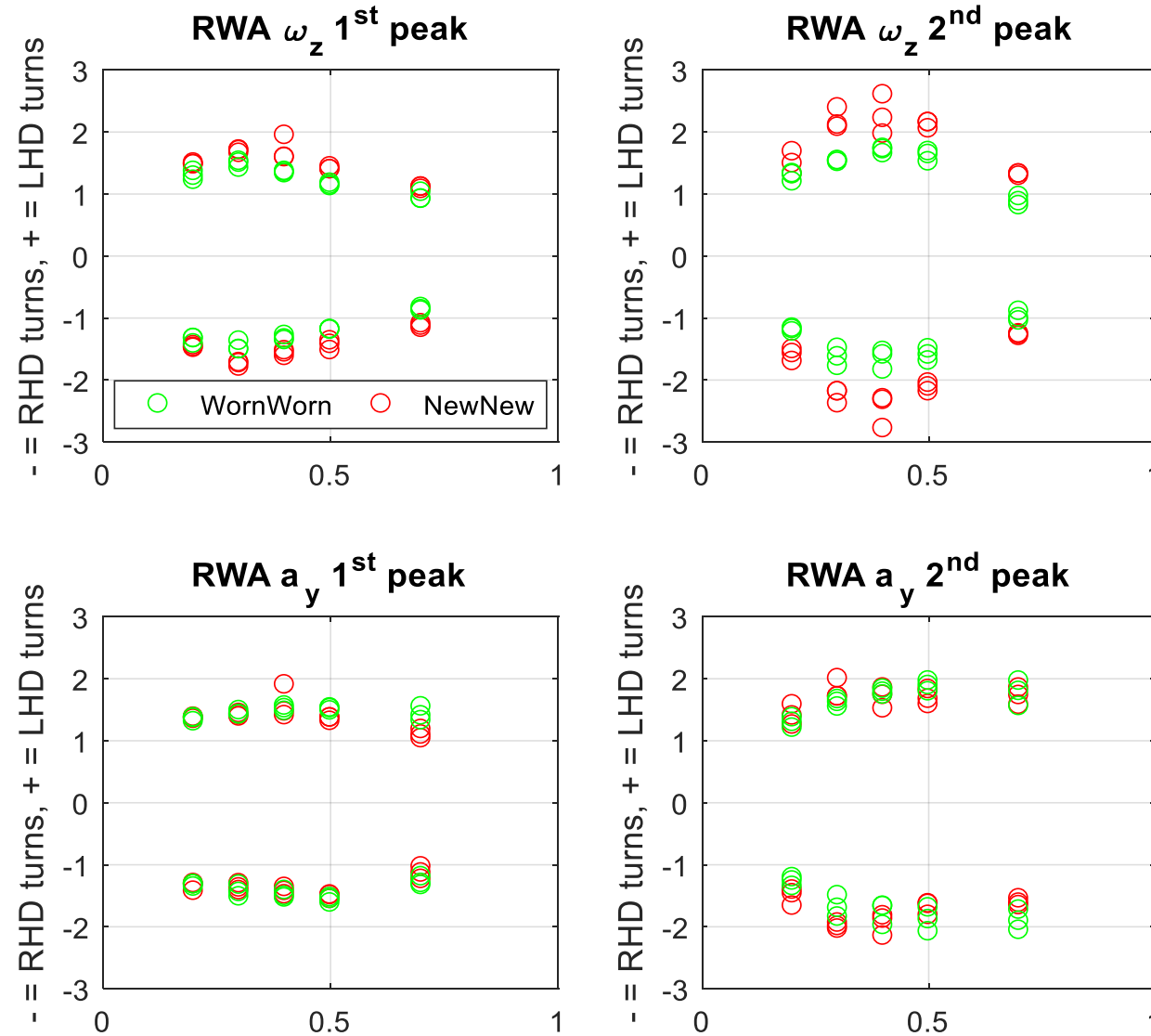
$$f \in [0.2 \ 0.3 \ 0.4 \ 0.5 \ 0.7]$$

Measured signals



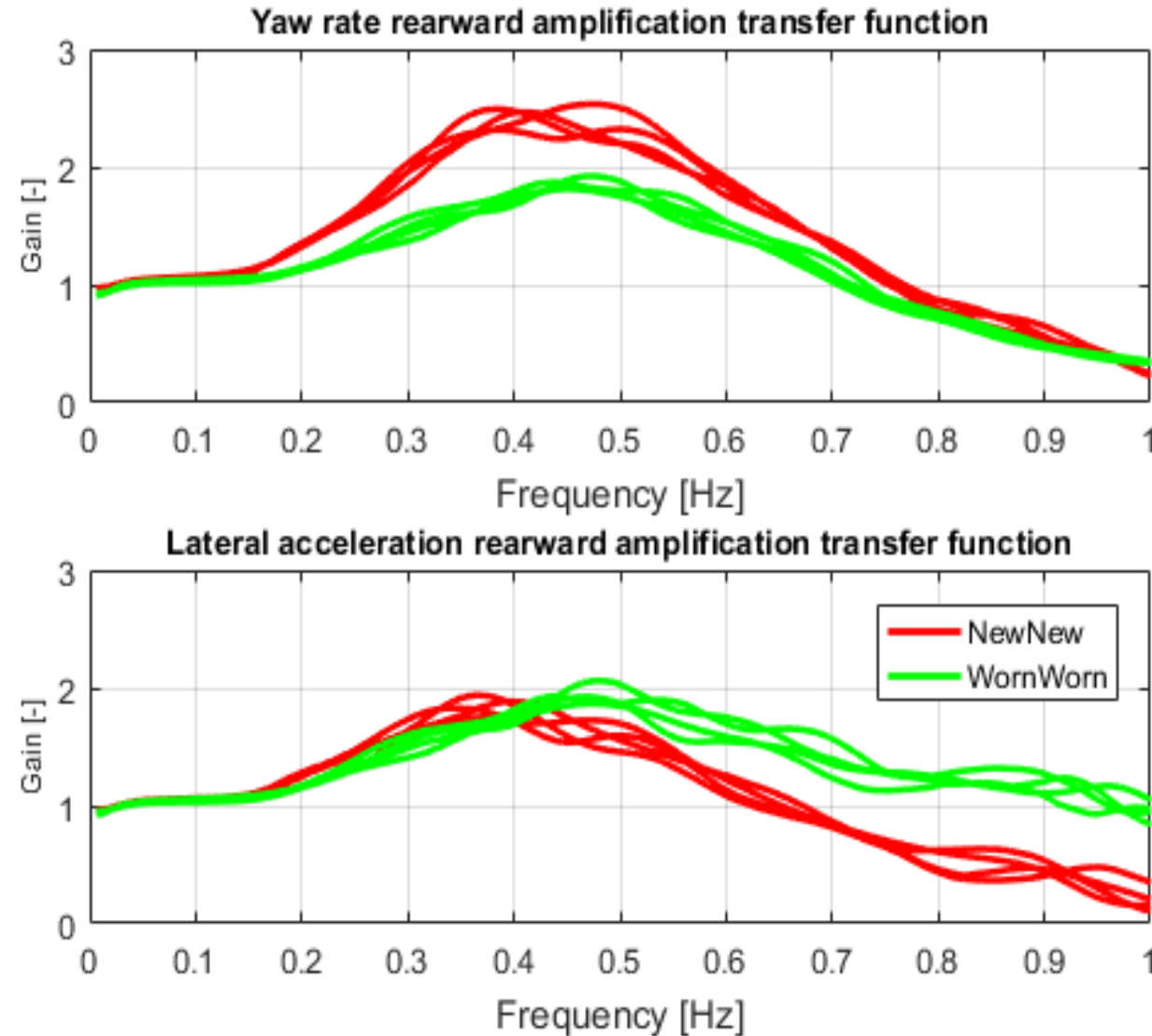


Rearward amplification – New vs Worn tires



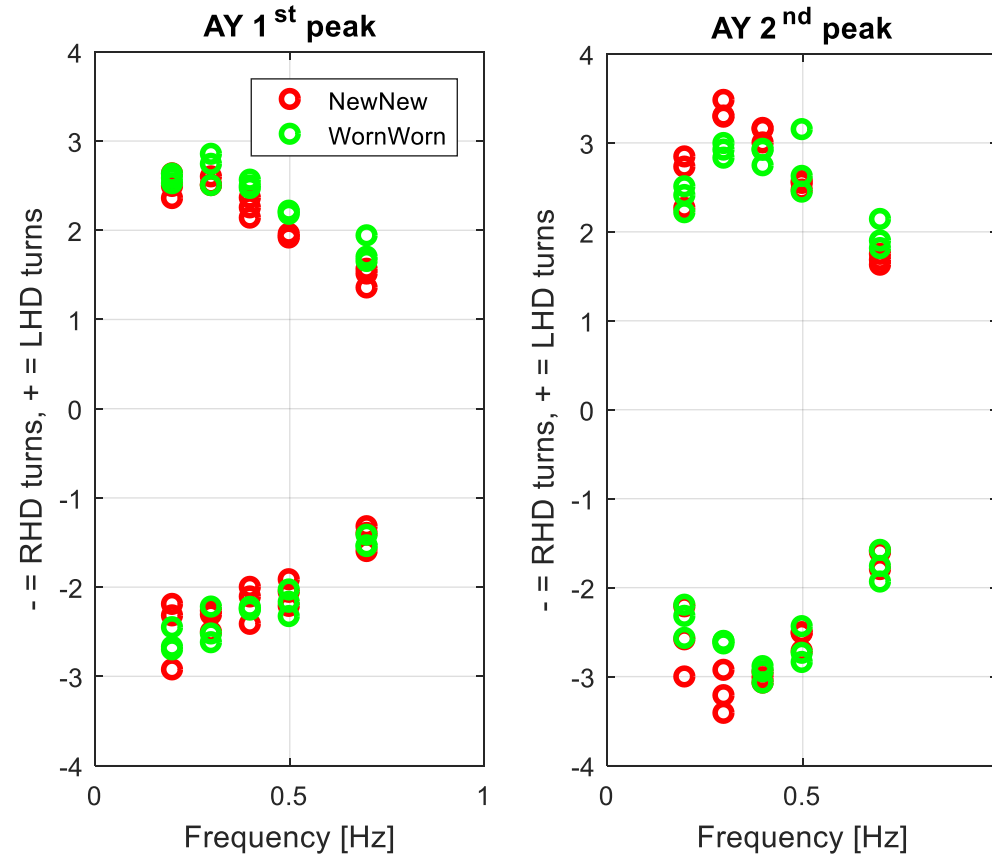


Rearward amplification – New vs Worn tires



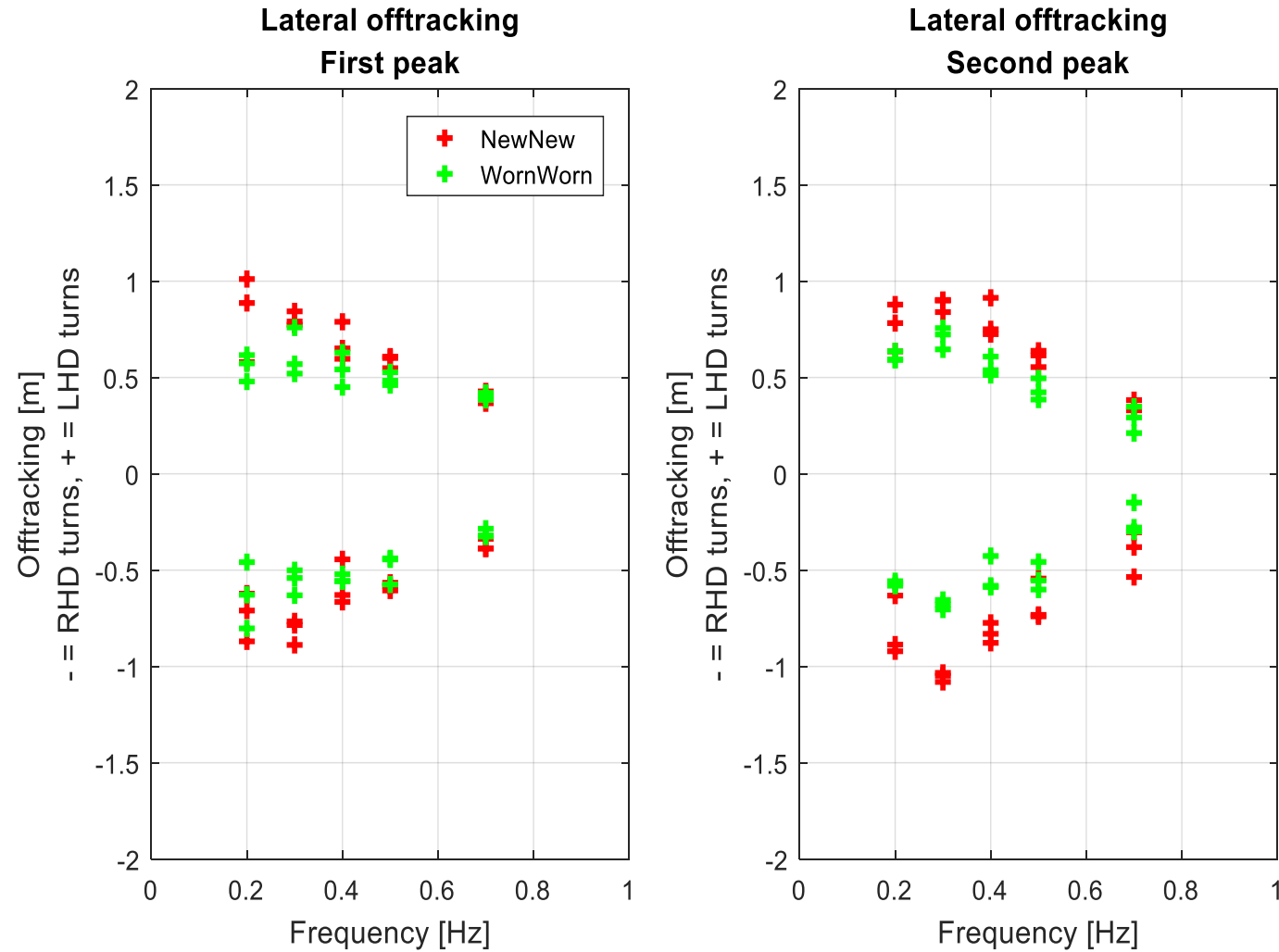


Acceleration peaks (= roll-over/skid risk)





Lateral off-tracking (= space on the road)



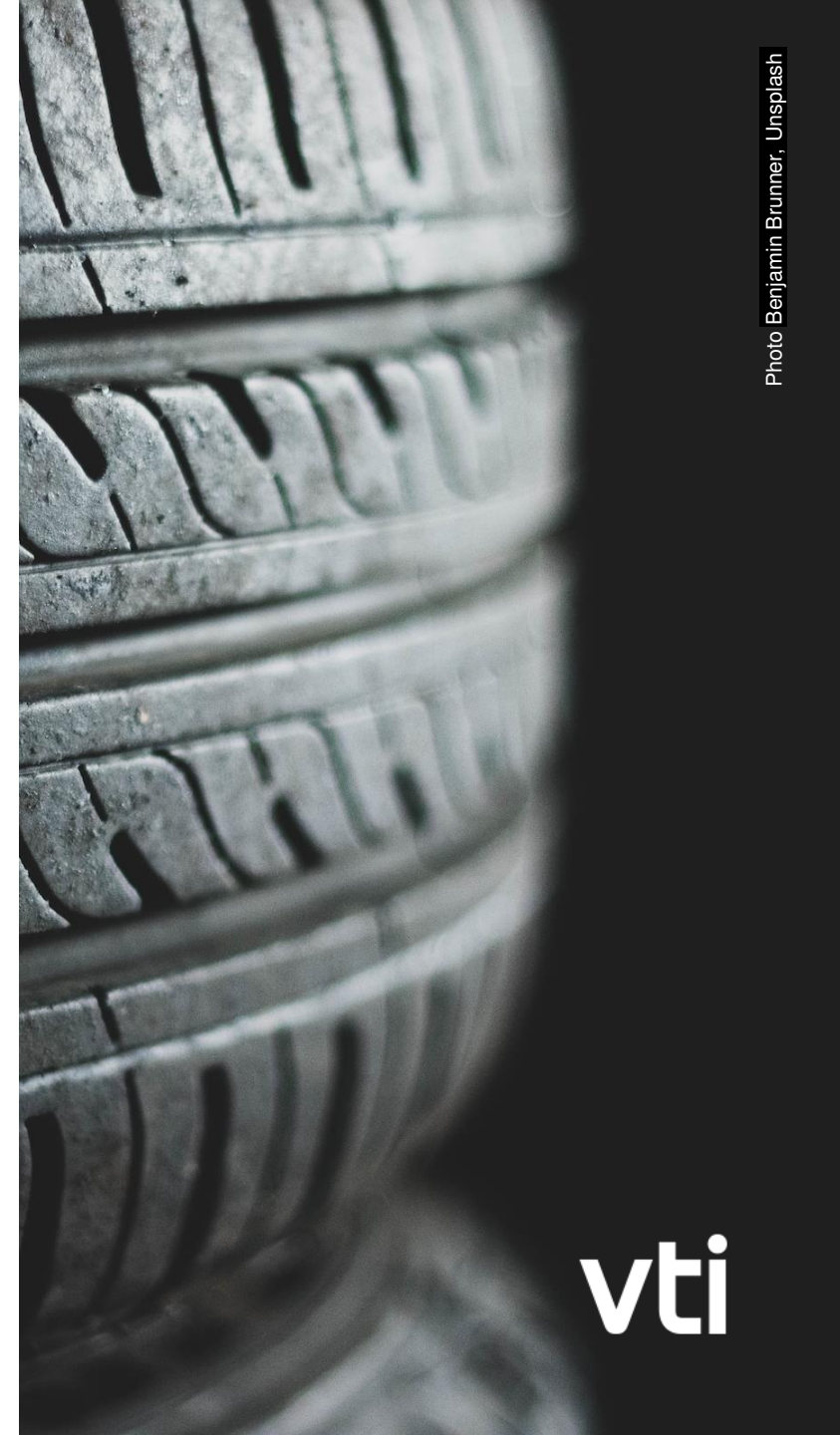


Conclusions testing

- Tire pressure is variable
 - 9.0 bar @ 20°C = 8.3 bar @ 4°C
 - good to get a hands-on experience
- New tires is the worst case
 - Higher rearward amplification of lateral acceleration at low frequencies
 - More off-tracking
- Data collected for model validation

Tested tires

- Eleven different truck tires were measured on the VTI tire testing machine, on dry asphalt and ice.
- A few tires were also measured on wet asphalt.
- 7 wide single steer/trailer tires (S), 2 dual mounted trailer tires (T), and 2 drive tires (D) were selected.
- New sample of each tire and worn specimens of seven of them were measured.
- Measurement data was used to parameterize the ISO tire model and create generic tire models for each tire category.



Selected wide single steer/trailer tires

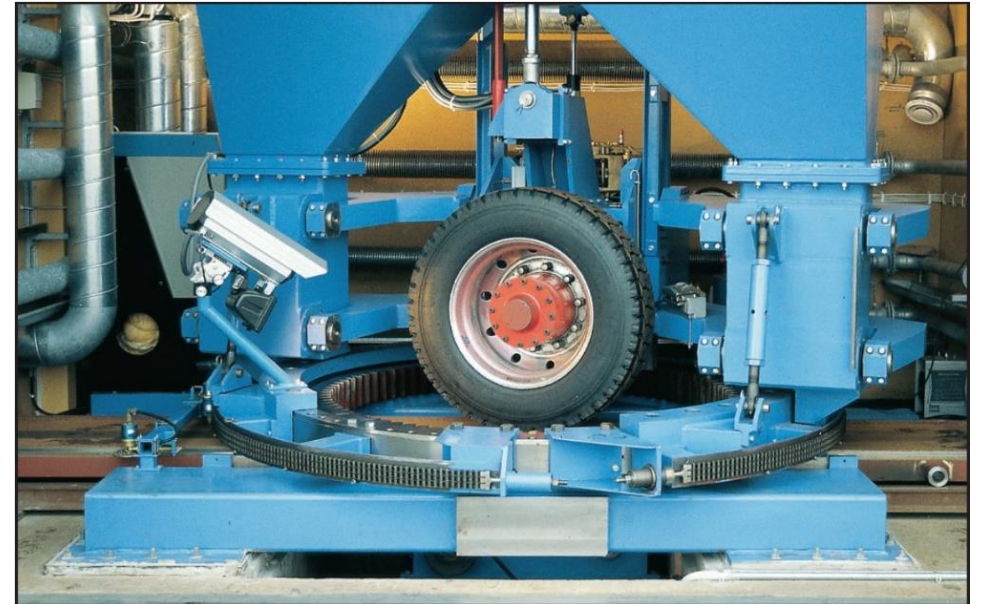
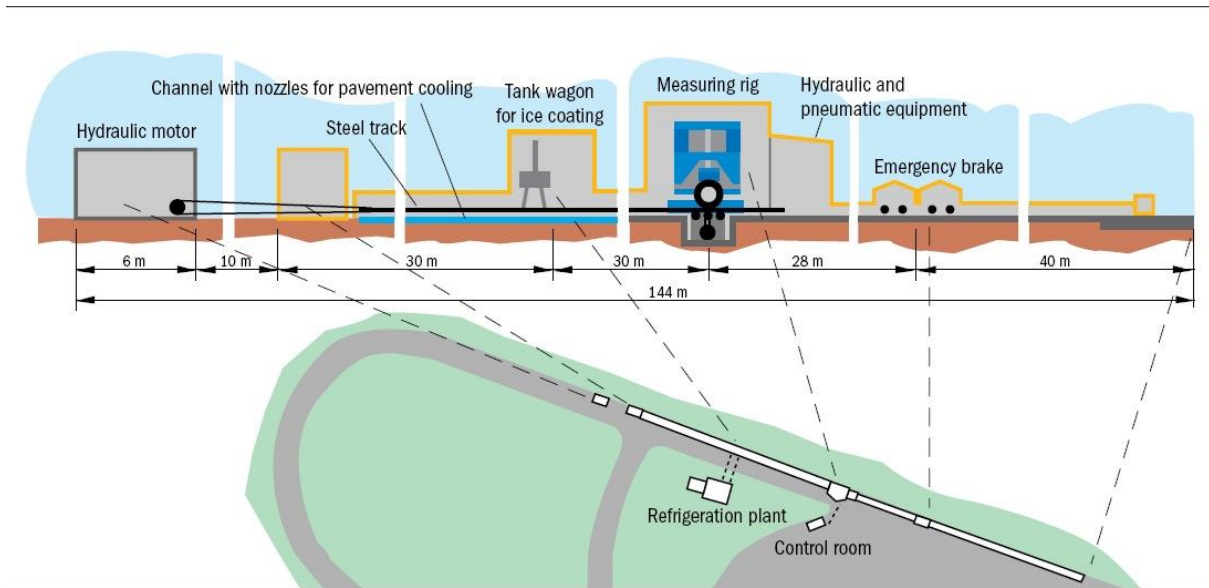
No	Code	Tire size	Tire type	Rated load, kg	Condition	Tread depth, mm	Hardness
1	S1	385/65 R22.5	Multi	4500	new	17	63
2	S1W1				worn	10	64
3	S1W2				worn	6	67
4	S1W3				worn	3	64
5	S2	385/65 R22.5	Winter	4500	new	16	64
6	S2W1				worn	8	64
7	S2W2				worn	5.5	68
8	S3	385/65 R22.5	Multi	4500	new	15	64.5
9	S3W				worn	7	67
10	S4	385/65 R22.5	Fuel	5000	new	13	70.5
11	S5	385/55 R22.5	Winter	4500	new	15	74.5
12	S6	385/65 R22.5	Winter	4500	new	16	66
13	S7	385/65 R22.5	Multi ¹	4500	new	12.5	65

Selected dual mounted drive and trailer tires

No	Code	Tire size	Tire type	Rated load, kg	Condition	Tread depth, mm	Hardness
14	D1	315/70 R22.5	Multi	3750/3350	new	18	68
15	D1W			3750/3350	worn	3.5	72
16	D2	315/70 R22.5	Winter	3550/3150	new	19	65.5
17	D2W			3550/3150	worn	6	65

No	Code	Tire size	Tire type	Rated load, kg	Condition	Tread depth, mm	Hardness
18	T1	265/70 R19.5	Multi	2725/2650	new	13.5	65
19	T1W			2725/2650	worn	3	64
20	T2	265/70 R19.5	Winter	2725/2650	new	14	65.5
21	T2W			2725/2650	worn	2	67

VTI flat track machine



Test conditions

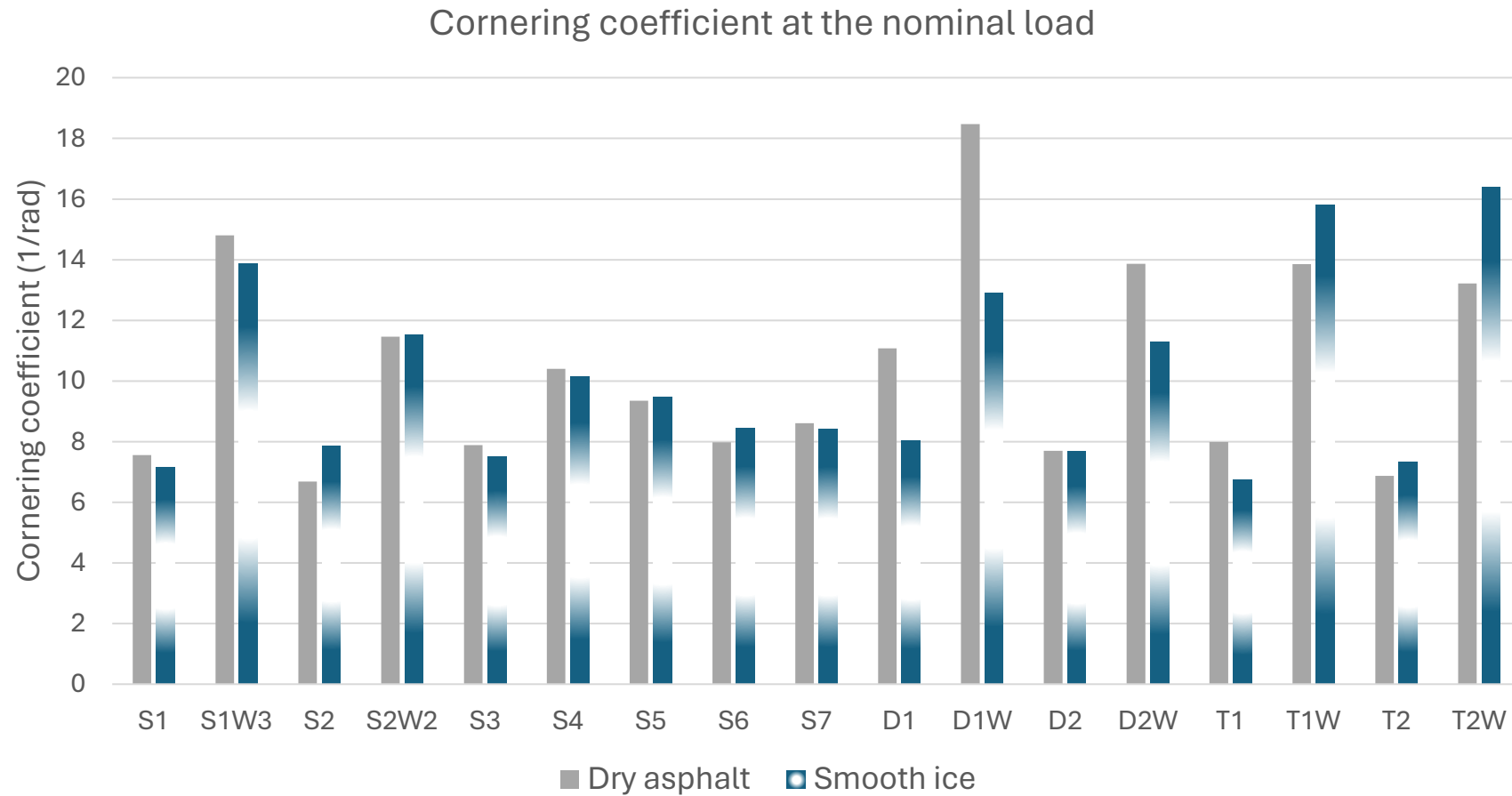
- Nominal load and inflation pressure were set based on common working conditions.
- Tires were measured at 50%, 100%, and 150% of the nominal load.
- Steady state force-slip curves were constructed from two measurements where tire slip angle was swept to -15° and $+15^\circ$.

	Single steer/trailer tires	Dual drive tires	Dual trailer tires
Nominal axle load (ton)	8.0	9.0	8.0
Wheel load (kN)	20, 40, 60	11.25, 22.5, 33.75	10, 20, 30
Inflation pressure (bar) Cold tire (-2°C) on ice	9.0	7.0	8.5
Inflation pressure (bar) Cold tire ($+20^\circ\text{C}$) on dry asphalt	9.0	7.5	8.5
Inflation pressure (bar) Heated tire on dry asphalt	10.0	8.5	9.5



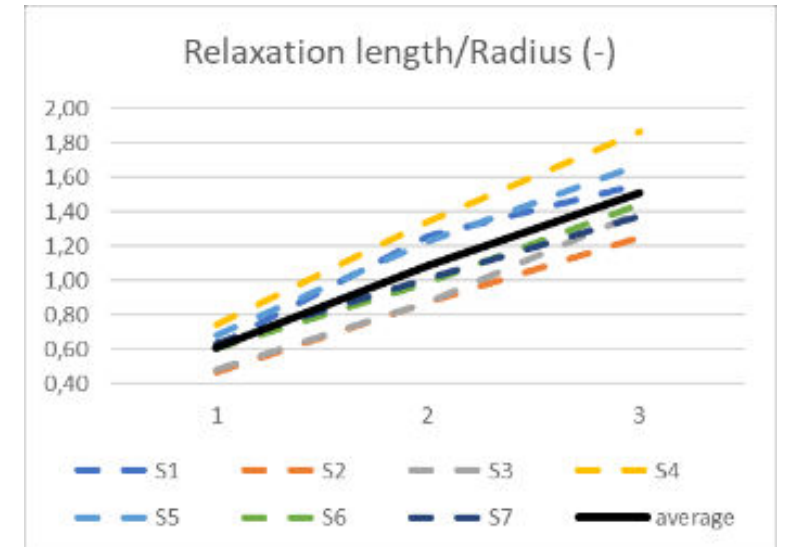
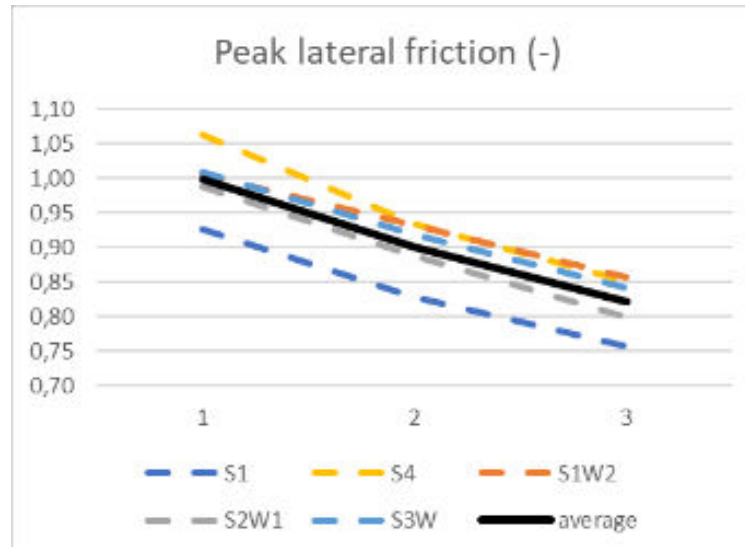
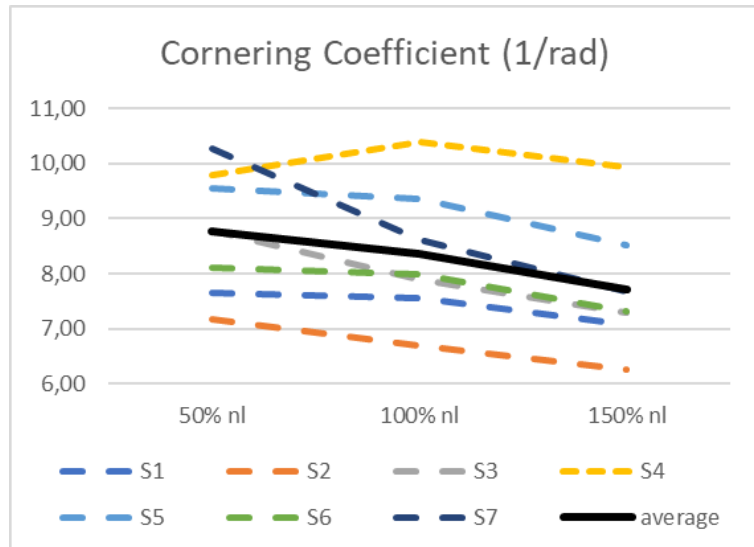
vti

Sample measurement results

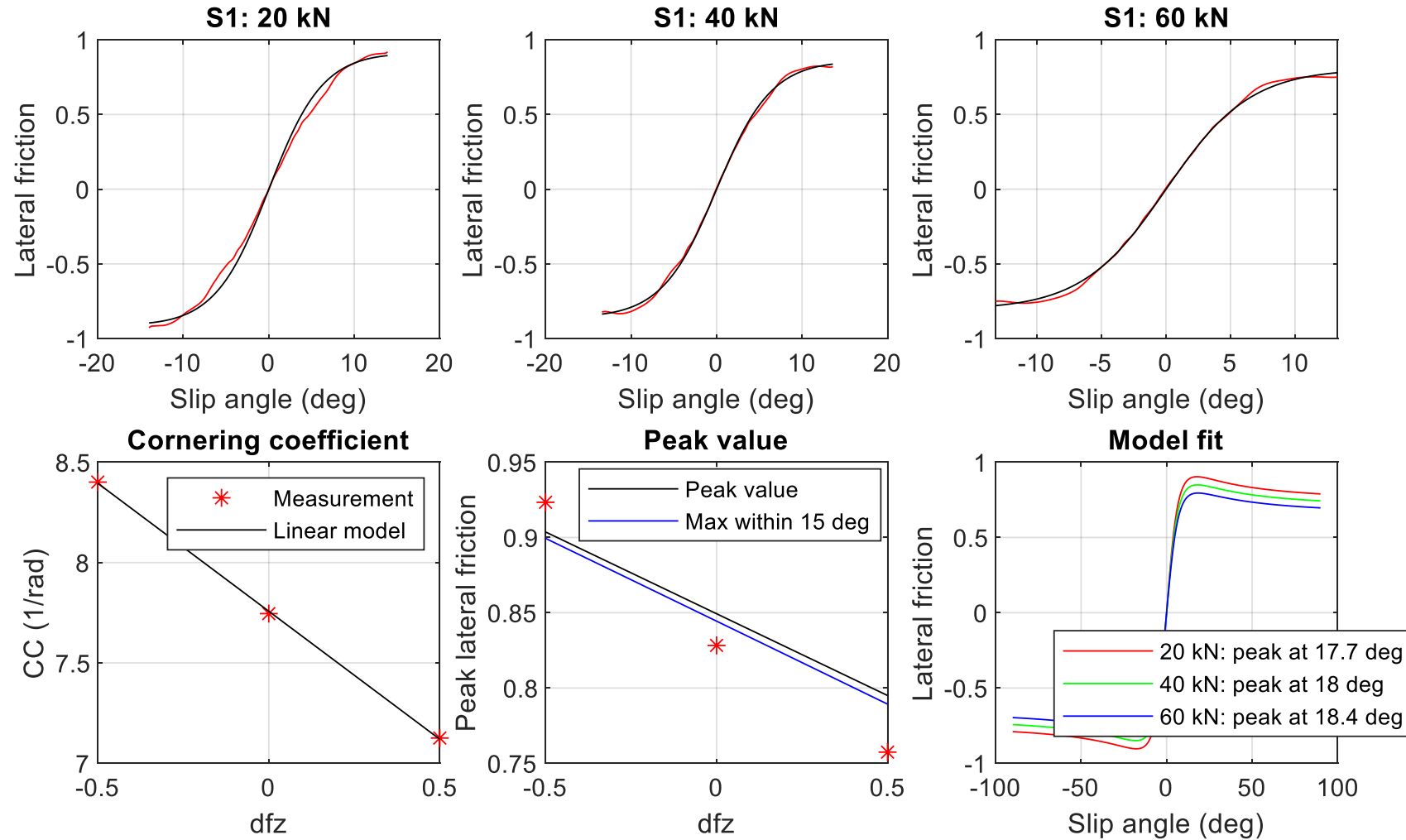


Tire model assumptions vs test data

Linear dependencies on load are assumed for all the three properties of cornering coefficient, peak lateral friction, and relaxation length



Generic tire model for new wide steer/trailer tires

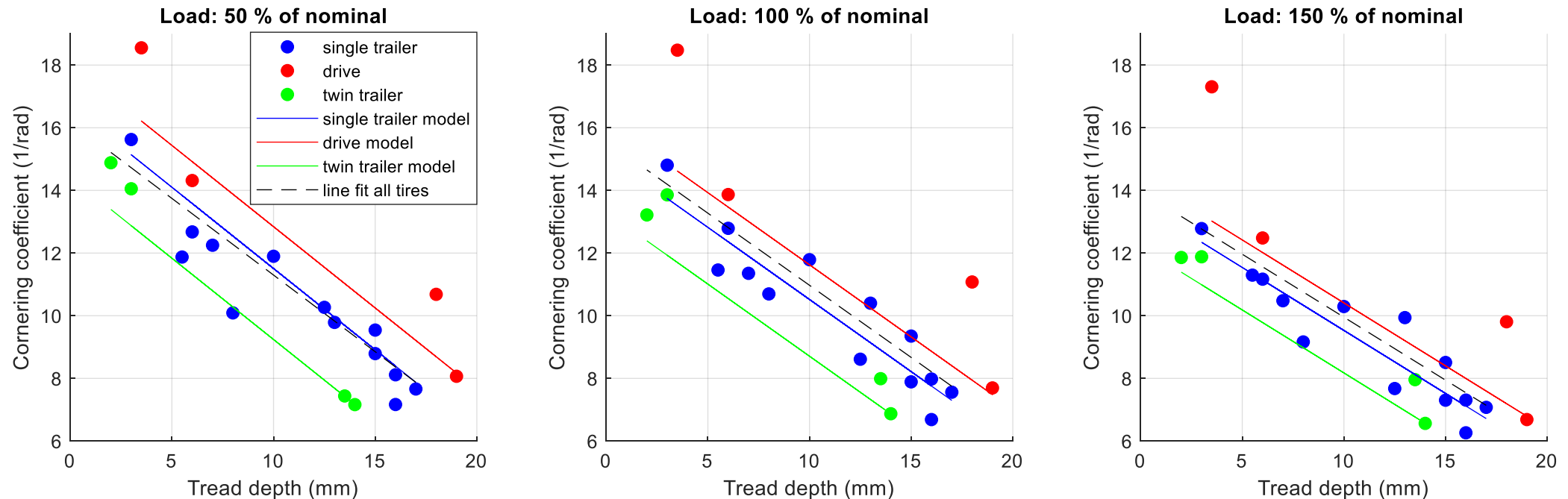


Standard values suggested for the ISO tire model

	Single mounted steer/trailer tyre	Twin mounted drive tyre	Twin mounted trailer tyre	All-purpose tyre
Nominal normal force (kN)	45	35,5	27,25	rated load
Nominal cornering coefficient (1/rad)	7,60	6,69	6,65	7,0
Cornering coefficient gradient (-)	-0,19	-0,33	-0,12	-0,2
Nominal peak lateral friction (-)	0,84	0,77	0,78	0,8
Peak lateral friction gradient (-)	-0,15	-0,12	-0,30	-0,15
Shape factor (-)	1,41	1,31	1,42	1,4
Nominal lateral relaxation length (m)	0,59	0,52	0,50	0,55
Lateral relaxation length gradient (-)	0,87	0,67	0,91	0,8

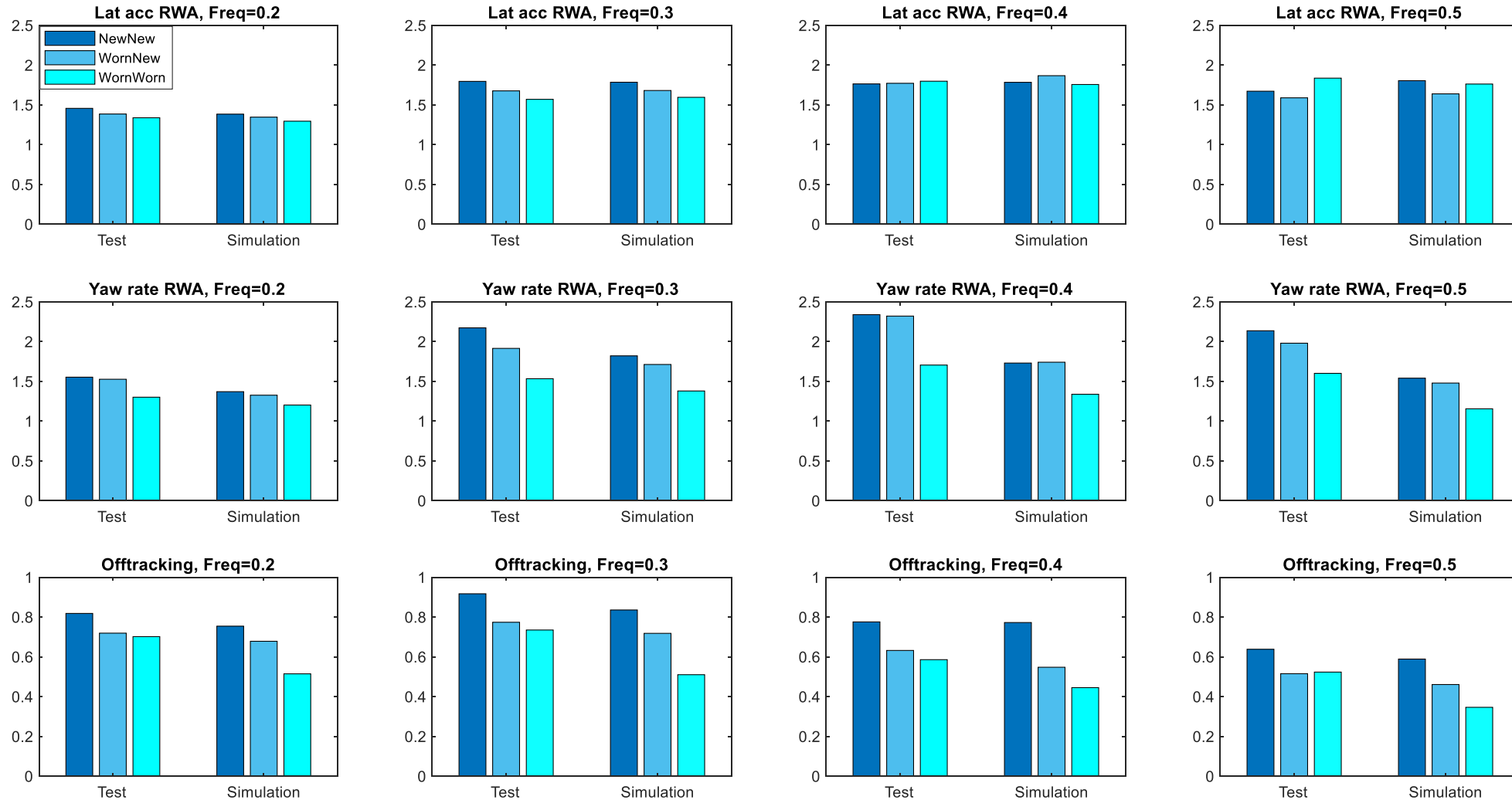
Including tread depth effect, a brief note

Effect of wear on cornering coefficient is similar for all tire types. Reducing the tread depth by one millimeter increases cornering coefficient by about 0.5 rad^{-1}



For more information refer to: S. Kharrazi and M. Hjort, "Effect of tread depth on the tire cornering performance," in the 28th International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD), Ottawa, Canada, 2023.

Simulation with developed models vs Scania tests



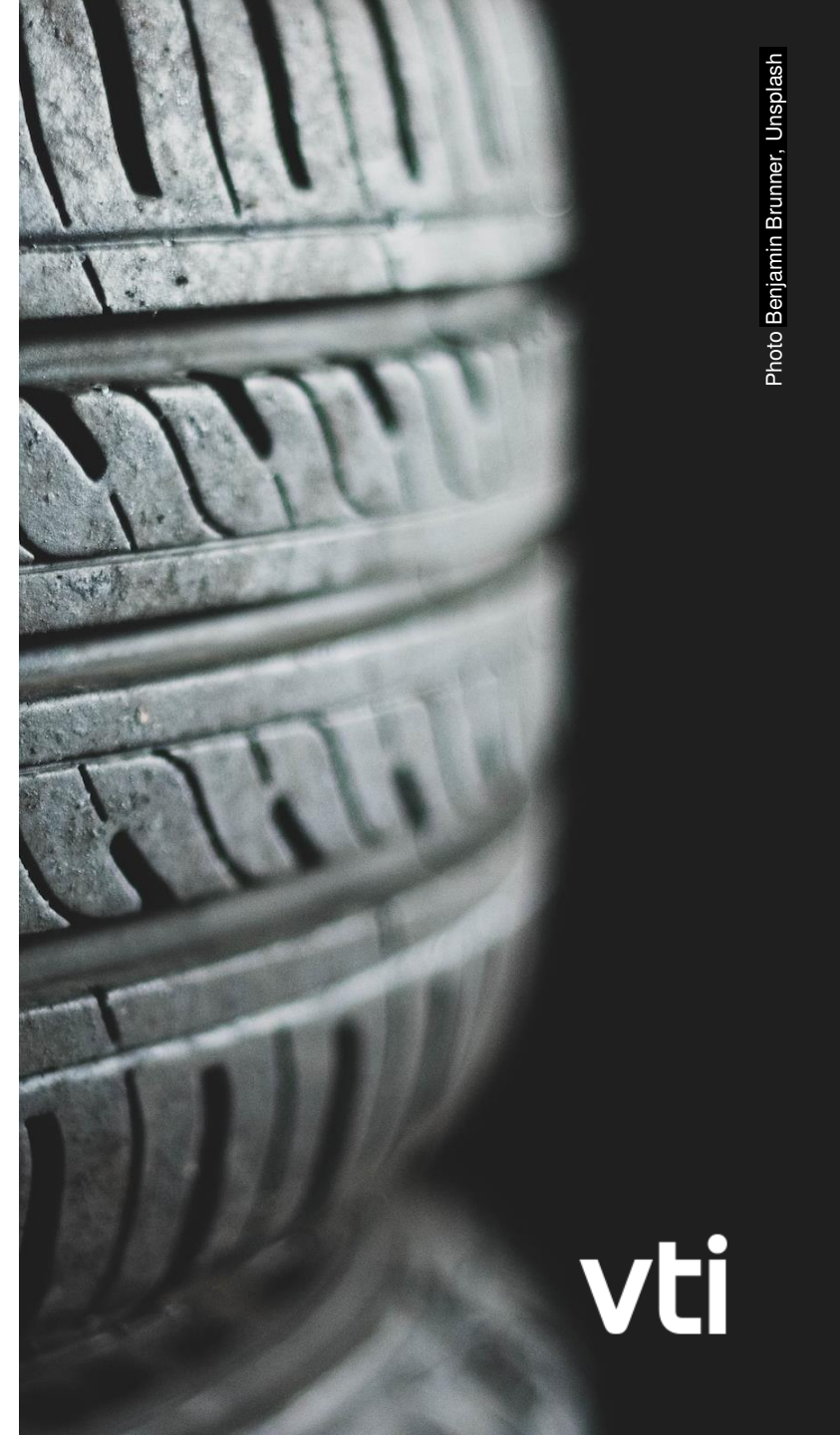
Summary

A simplified tire model, based on Magic Formula, is developed and published as an ISO model.

Several truck tires were measured and used to parameterize generic tire models for different tire categories.

Experiments on test track were performed to study effects of new/worn tires on the performance of a heavy vehicle combination.

Results of simulations with the developed tire models were compared with the test track data, showing similar performance can be captured by the models.



Presentation 6:
Motion cueing for driving simulators
Henrik Hvitfeldt, KTH

Presentation 7a:
Vehicle Engineering MSc programme at KTH,
Mikael Nybacka, KTH

Presentation 7b:
*Mobility engineering MSc programme at
Chalmers,*
Dag Bergsjö, Chalmers

Presentation 7c:
Vehicle engineering courses at LiU,
Jan Åslund, LiU

Poster 1:

**Virtual Verification Framework for Vehicle
Motion Systems**

Albijon Blakqori and Mille Kotur, ZEEKR and
Chalmers

**Poster 2: Advancing Road Vehicle Path
Length Estimation via Geospatial Analytics**
Yogeswaran Amsavalli, Volvo Trucks, Chalmers
and Università di Trento

Estimation of Path length using Geospatial Data Analysis

STUDYING VARIOUS ESTIMATION METHODS TO COMPARE AND
IDENTIFY THE MOST RELIABLE METHOD

Thesis Research by Yogeswaran Amsavalli, Dual MSc. in
Autonomous Systems



About me



Yogeswaran Amsavalli

Msc. Autonomous Systems

2018 - 2022

Bachelor of Engineering

Kumaraguru College of Technology
affiliated to ANNA University

Graduated in Electronics and
Instrumentation

2022 - Present

Master of Science

Dual Master's Degree in Autonomous
Systems

Budapest University of Technology
(Software Engineering Program)

University of Trento
(Mechatronics Engineering Program)

🔍 distance

Introduction

Objective:

- To compare and develop distance estimation methods
- Create a deterministic operating cycle (dOC) model where distance is the independent variable.
- The model includes comprehensive operating conditions data and is adjustable for different vehicle types.

Research Overview - What?

- **What:** Studied 11 estimation methods, focusing on traditional to advanced geospatial distance Estimation methods.

Methods:

Traditional: Haversine and Vincenty, used this as a base and added Elevation, slope, Curvature etc. to the same and did the estimation.

Sphere:

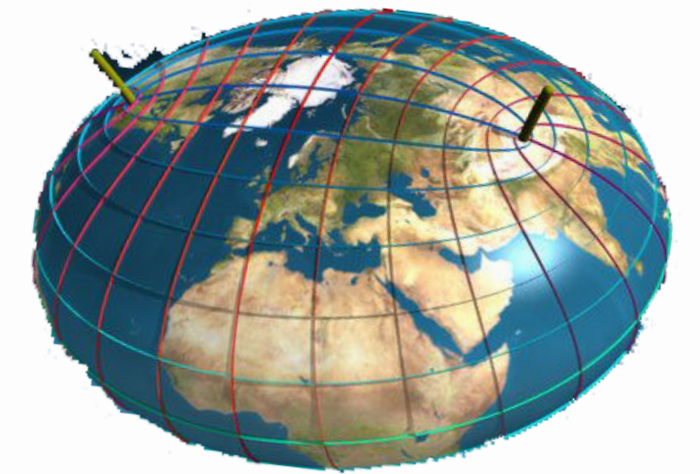
- Haversine
- Spherical Law of Cosines

Ellipsoidal:

Geodesic
Vincenty

other:

3D distance in Cartesian
Space



Research Overview - Methods



Sphere Model :

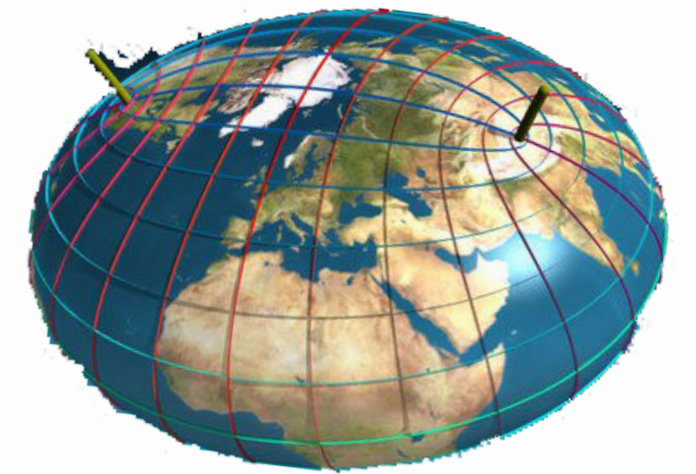
- **Purpose:** Calculate the **great-circle distance between two points on a sphere.**
- **Advantages:**
 - a. Simplicity and ease of implementation.
 - b. Sufficient accuracy for short to medium distances.
- **Disadvantages:**

Assumes Earth as a perfect sphere, introducing errors over long distances or high-precision needs.

Ellipsoidal Model:

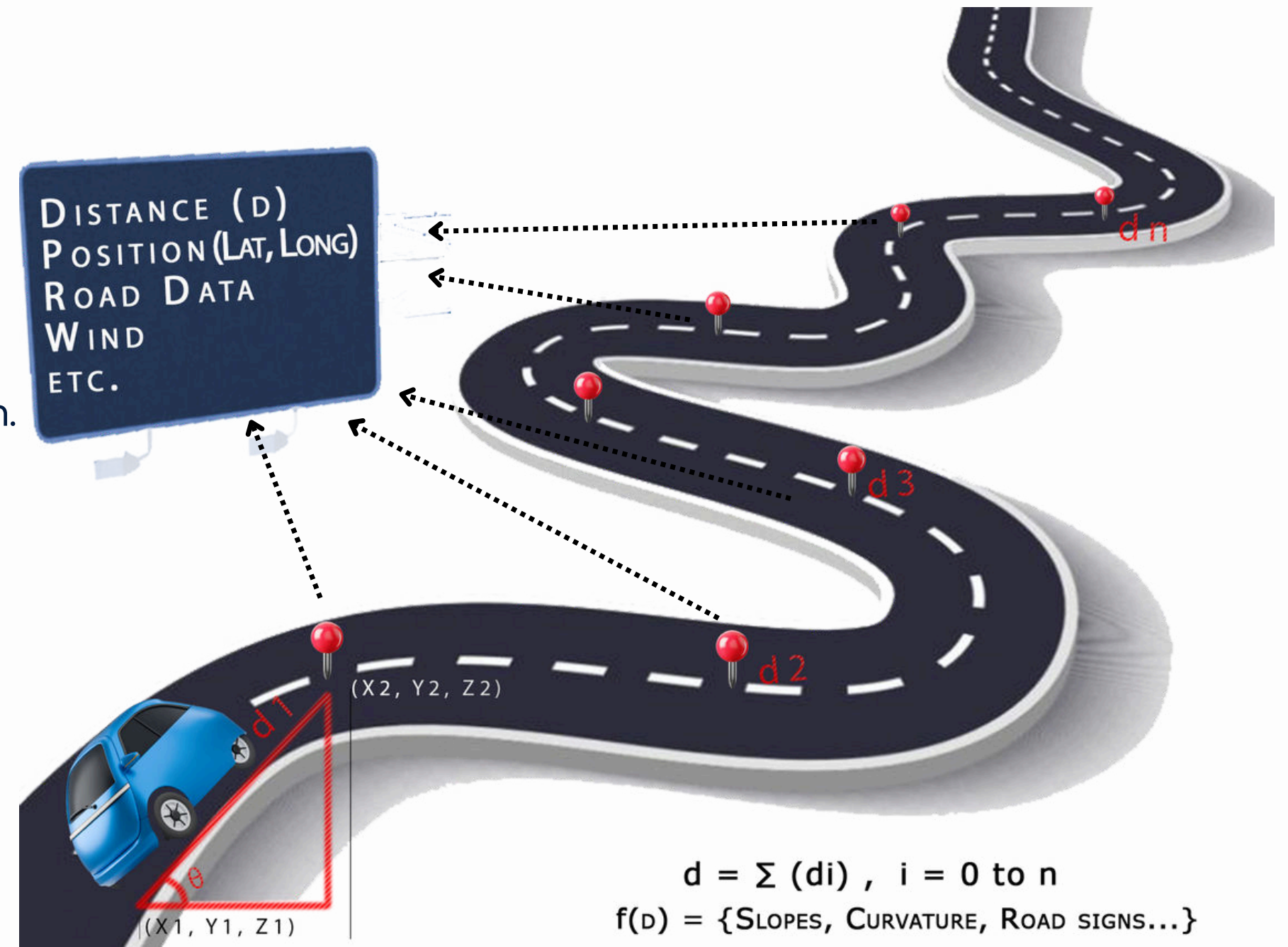
- **Purpose:** Calculate the **shortest path over the Earth's surface, considering its ellipsoidal shape.**
- **Advantages:**
 - a. Highest accuracy for distance calculations.
 - b. Essential for applications requiring high precision.
- **Disadvantages:**

Uses complex mathematical models to compute accurate distances. Hence, Computationally demanding.



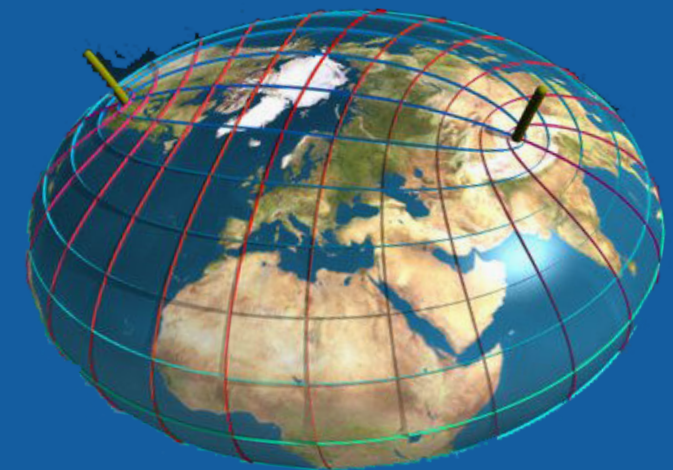
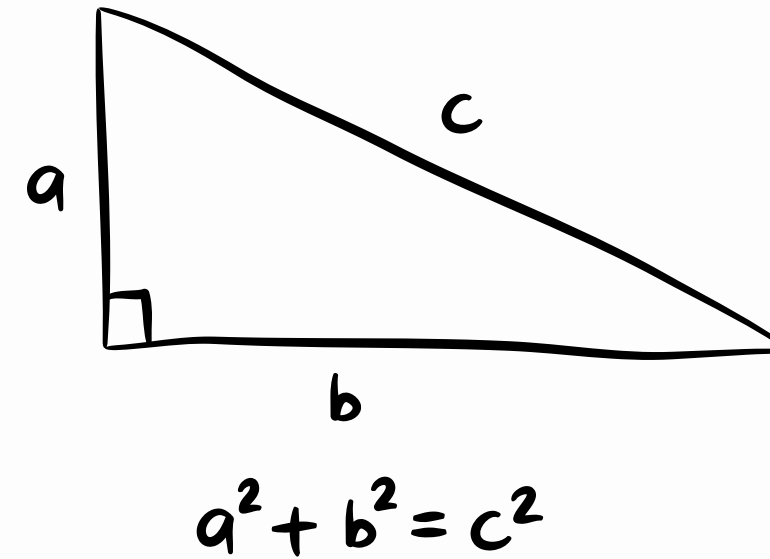
Research Overview - HOW?

- Data -> HERE Maps API
- Path -> Segments (Links)
- Links -> Points
- Intermediate Inclined distances in the vehicle path.
- Point -> Operating Conditions data



Research Overview - Methods

- ▶ Haversine Formula
- ▶ Vincenty Formula
- ▶ Geodesic - More accurate Distance
- ▶ Geodesic Inclined Distance
 - Elevation (Pythagoras)
 - Gradient (Slope)
- ▶ Haversine Inclined Distance
 - Elevation (Pythagoras)
 - Gradient (Slope)
- ▶ Spherical Law of Cosines
- ▶ 3D distance Formula



Deterministic Operating Cycle (dOC)

Data-driven approach used to simulate and predict the performance of vehicles based on specific and detailed road and driving condition data.

Usage and Advantages :

- Simulation of Real-Time Driving Conditions
- Deterministic Data(Specific and Measured) -> Accurate Simulations
- Integration Capability

Model Attributes

- Distance (m) -independent
- Latitude (°)
- Longitude (°)
- Elevation (m)
- Slope (°)
- Curvature (m)
- Headings (°)
- Speed Limit (m/s)
- Free Flow Speed (m/s)
- Stop Sign
- Traffic Signal
- Pedestrian Crossings
- Yield (giveway sign)
- Wind Direction (°)
- Wind Velocity (m/s)

Applications of dOC

01

Road data tool

02

Simulation Enhancement

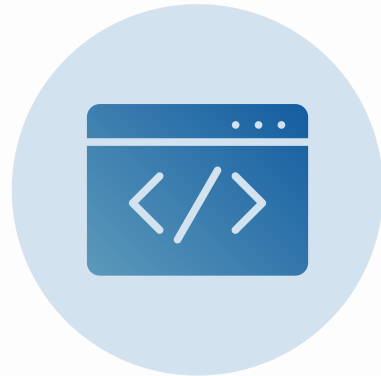
03

Predictive Accuracy

04

Safety Systems

Additional Considerations:



Integrability

The model encompasses all essential road attributes (latitude, longitude, elevation, slope, curvature, speed signs), making it a robust tool for various applications.

Additional Attributes can we integrated regardless of their position but with Distance



Energy Estimation

Precise distances enable better predictions for optimizing fuel and battery usage.

If the load varied at specific point in the path it can be precisely considered with distance and can modify the simulation accordingly



Know Better

Precisely Identify the uphill and downhill

Improves real-time responses to traffic lights, roundabouts, and other critical points.

Research Overview - WHY?

Precise distance calculations are crucial for

- Accurate vehicle simulation
- impacting energy Estimation
- safety systems.

As Distance being Independent Variable,

Identify

Identify the attribute with Distance rather than location in simulation

Maintenance

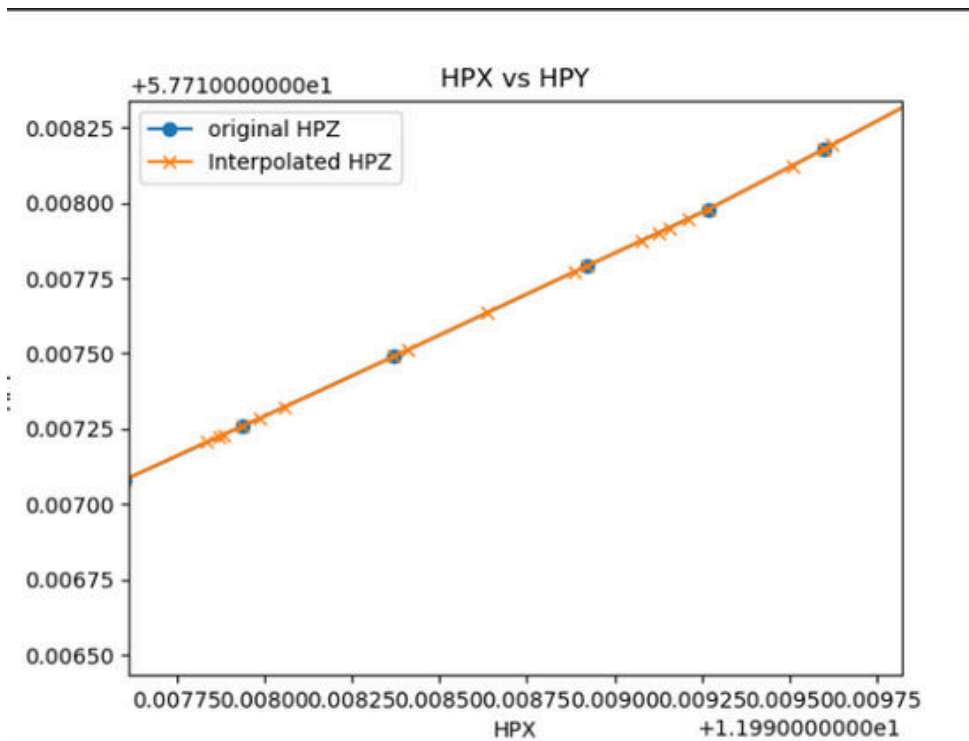
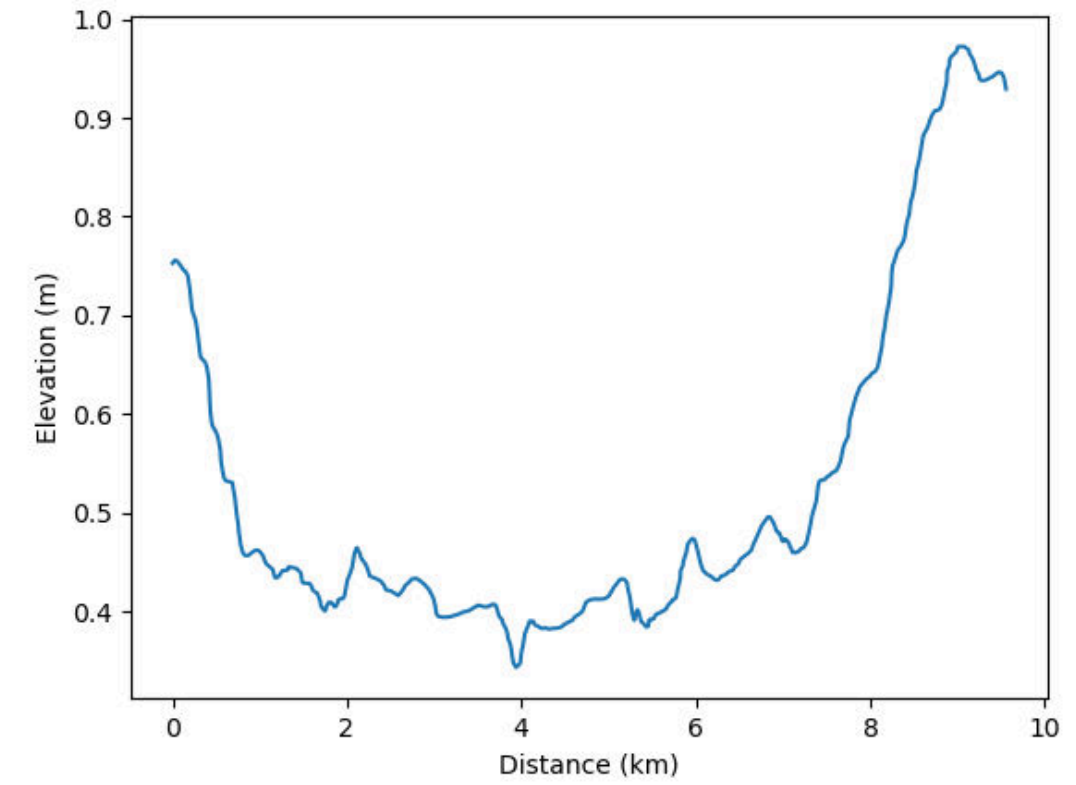
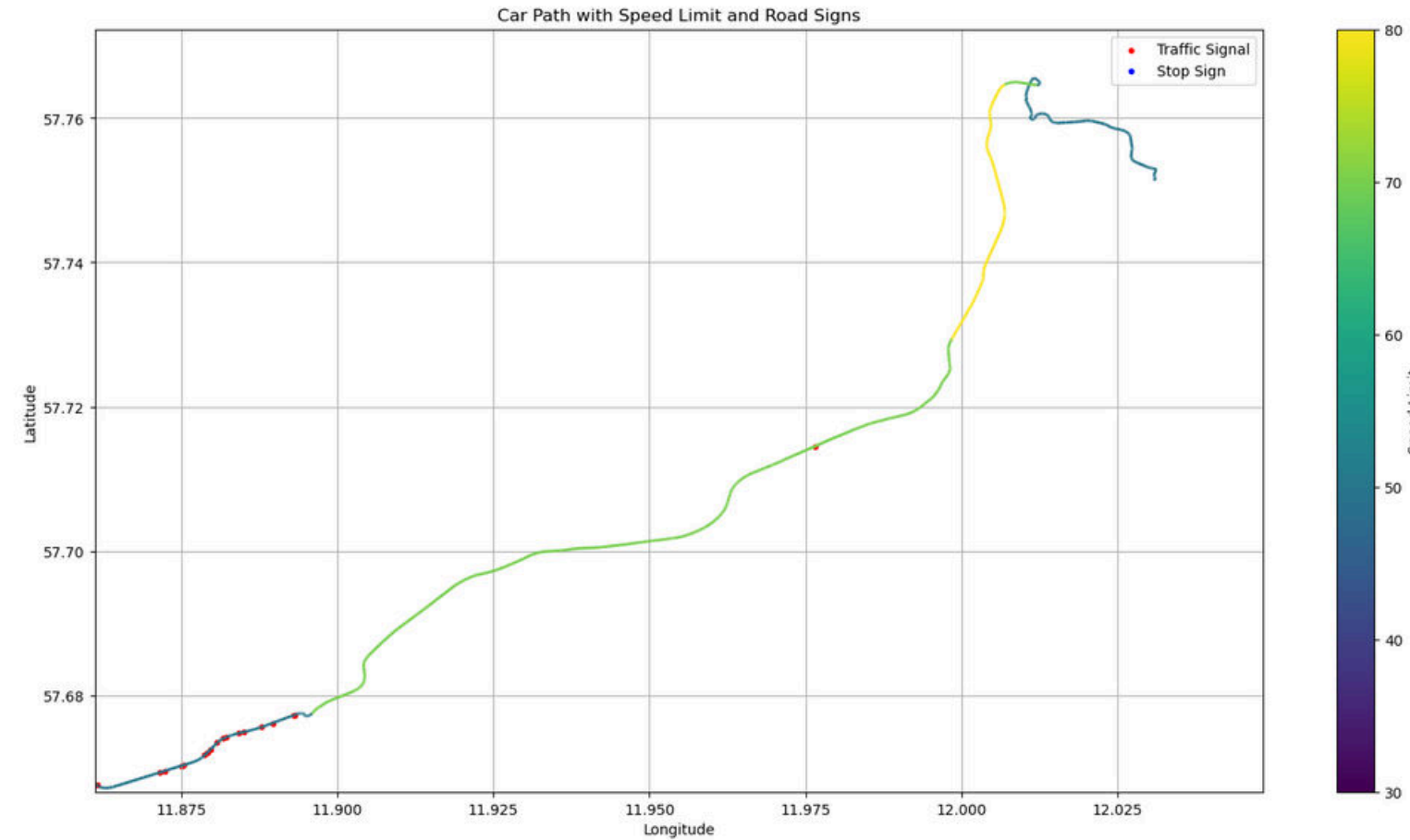
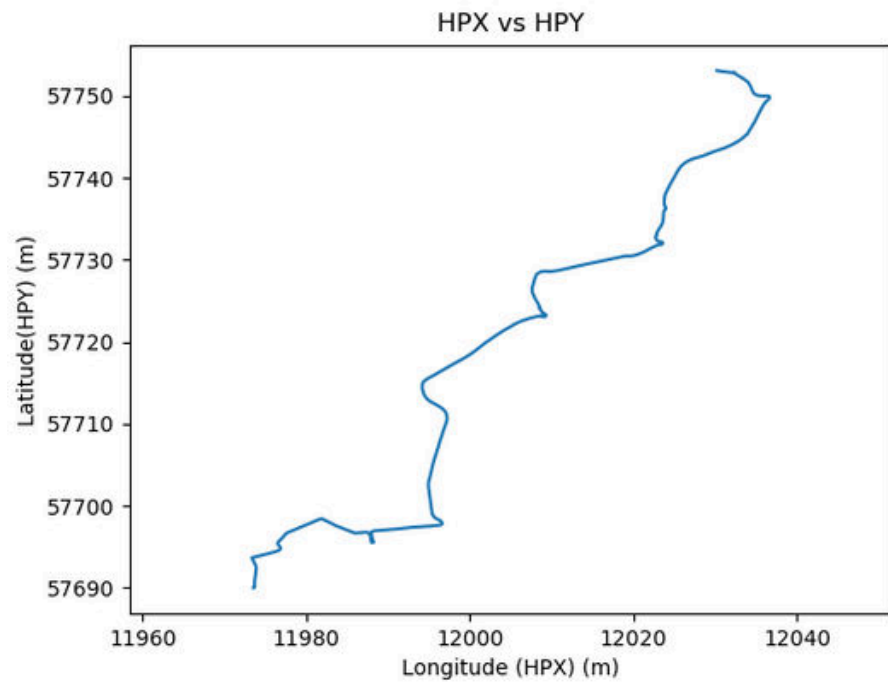
Distance-based data allows for precise monitoring of vehicle components, predicting maintenance needs based on the actual distance traveled under various conditions.

Versatility

Can be used for diverse purposes such as residual range prediction, gearshift strategy optimization, and real-time navigation assistance.



Plots and Images



```

yogr702 / mbcnemis_00c_generator_2024 / mbcnemis_00c_generator_2024 / check_model / CSV_files / input_csv / data
Latitude,Longitude,Elevation,Slopes,Curvatures,Headings,Distance,Speed Limits,free_flow_speed,Traffic Signals,Stop Sign,Yield Sign,Pedestrian Crossing,Wind Direction,Wind Velocity
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Summary

Abstract

Accurately predicting the **remaining driving range** of vehicles is essential for optimizing vehicle performance and reducing range anxiety among drivers, especially for electric vehicles (EVs). This study examines and measures the **accuracy of distance calculations** using various coordinate systems and estimation methods to develop a reliable methodology for estimating **residual range**.

Intro

- To improve and refine existing distance calculations.
- Overview of Model-based approaches.
- The model includes comprehensive road data and is adjustable for different vehicle types.

DOC for Simulation

The Deterministic Operating Cycle (DoC) model is a data-driven approach used to **simulate** and **predict** the performance of vehicles based on specific and detailed road and driving condition data.

Usages of DoC Model :

1. Simulation of Real- Time Driving Conditions
2. Energy Consumption Estimation
3. Residual Range Prediction

Advantages :

- Precision:** by using deterministic data (specific and measured) -> Accurate simulations
- Real-World applicability :** Predictions to actual road conditions rather than averaged scenario
- Integration Capability:** Integrated with various APIs to continually update and refine the model.

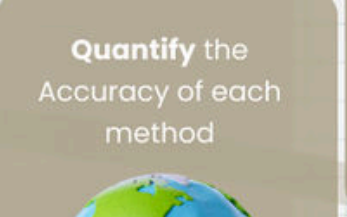
Evaluate various APIs for **Extracting Road data** (HERE Maps)



OBJECTIVES



Calculate path length with multiple estimation methods and compare



Quantify the Accuracy of each method

Develop a backend system to sort and process road data in a **Deterministic Operating Cycle (DOC)** format

METHOD

11

Estimation Methods

- Haversine, Vincenty's, Spherical Law of Cosines, 3D Distance
- Elevation
- Curvature embedded Calculation

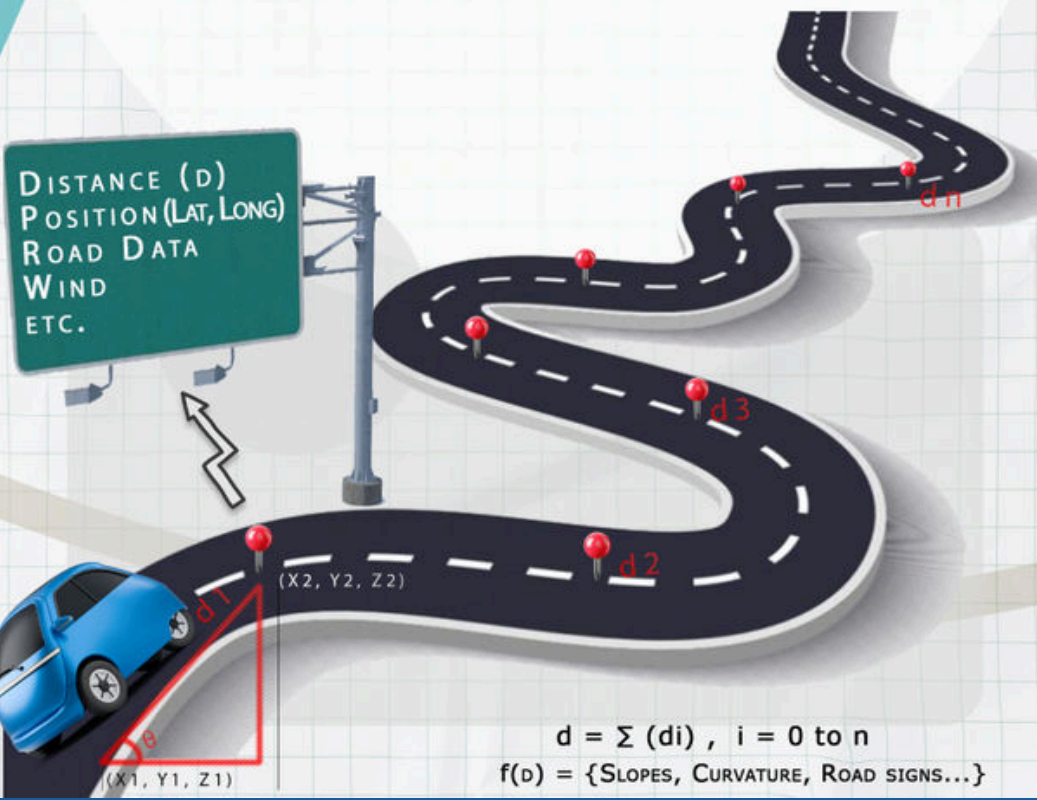
- Inclined distance using Elevation and Slope data
- Multiple Distance between two way-points

Advancing Path Length Estimation using Geospatial Data Analysis

Yogeswaran Amsavalli, MSc. Autonomous Systems, University of Trento, Italy

Affiliations: **Chalmers University of Technology**, Department of Vehicle Engineering and Autonomous Systems, Gothenburg, Sweden.
Volvo Group AB, Department of PSD, Gothenburg, Sweden.

DISTANCE (D)
POSITION (LAT, LONG)
ROAD DATA
WIND
ETC.



$$d = \sum (d_i) , i = 0 \text{ to } n$$

$$f(d) = \{SLOPES, CURVATURE, ROAD SIGNS...\}$$

RESULTS

Error Variation

ranging from 0.0006 to 21 % (350 kms)

Comparing and Increasing the efficiency in Computation and data processing

Identified

- Most Accurate
- Least Accurate
- General Trend

references:
 • Gy. Windarsari and A. Seliawan, "COMPARATIVE ANALYSIS OF VINCENTY AND GEODESIC METHOD APPROACHES IN MEASURING THE DISTANCE BETWEEN SUBDISTRICT OFFICES IN SIAATIGA CITY, BAREKENG- J. Math. & App. vol. 10, no. 4, pp. 1207-1220, Dec. 2022.
 • L. Romana, P. Johansson, E. Nordström, F. Brunzellus, R. Andersson and R. Jacobson, "A Classification Method of Road Transport Missions and Applications Using the Operating Cycle Format," in IEEE Access, vol. 10, pp. 73087-73121, 2022, doi:10.1109/ACCESS.2022.3988872.
 • S.-Q. Xin, Y. He and C.-W. Fu, "Efficiently Computing Exact Geodesic Loops within Finite Steps," in IEEE Transactions on Visualization and Computer Graphics, vol. 18, no. 6, pp. 879-889, June 2012, doi:10.1109/TVCG.2011.119



-Download Poster Here

THANK YOU!



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Abstract

Accurately predicting the **remaining driving range** of vehicles is essential for optimizing vehicle performance and reducing range anxiety among drivers, especially for electric vehicles (EVs). This study examines and measures the **accuracy of distance calculations** using various coordinate systems and estimation methods to develop a reliable methodology for estimating **residual range**.

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- The model includes comprehensive road data and is adjustable for different vehicle types.

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METHOD

11

Estimation Methods

- Haversine, Vincenty's,
- Spherical Law of Cosines,
- 3D Distance
- Elevation
- Curvature embedded Calculation

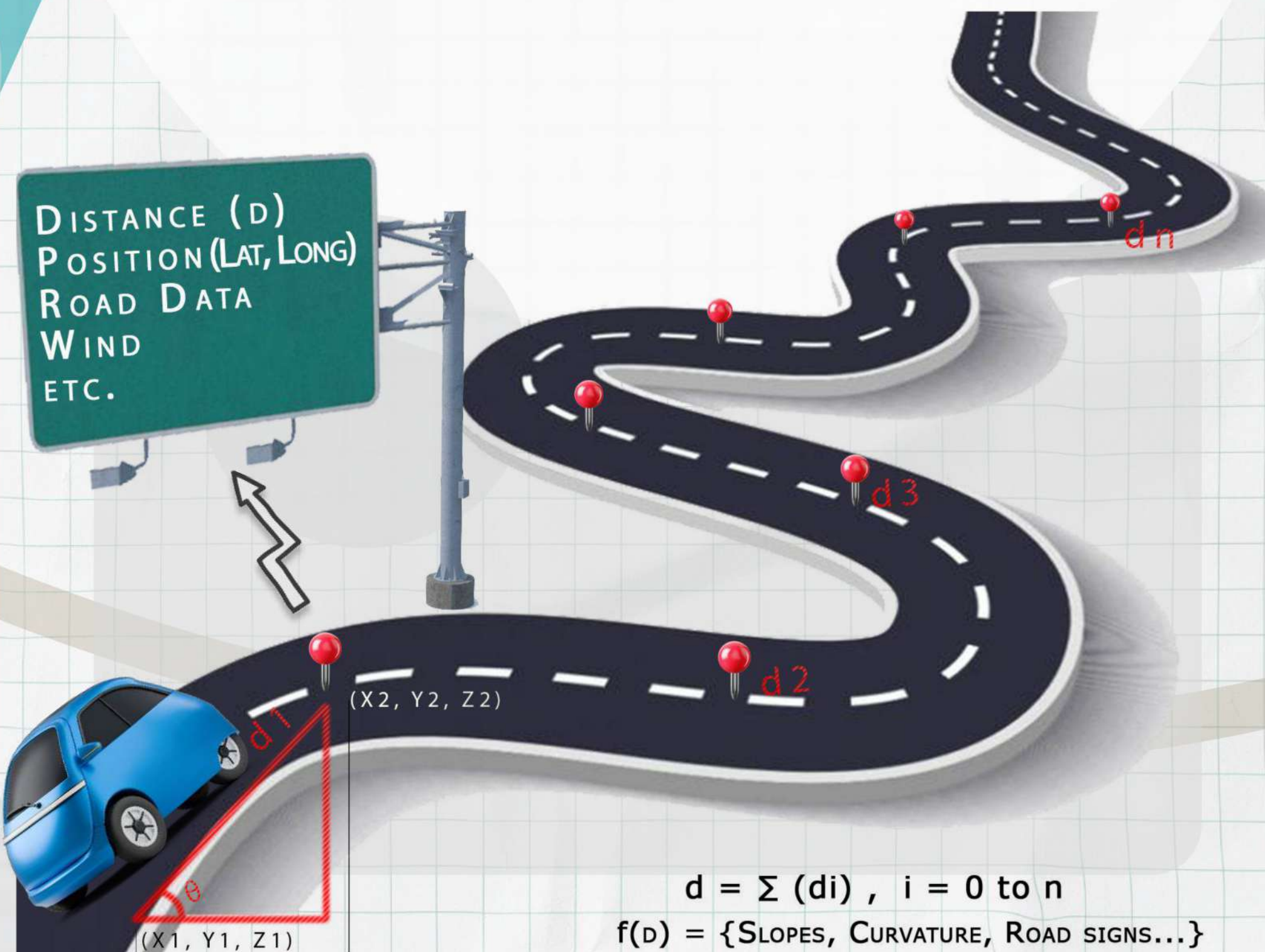
- Inclined distance using Elevation and Slope data
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Advancing Path Length Estimation using Geospatial Data Analysis

Yogeswaran Amsavalli, MSc. Autonomous Systems, University of Trento, Italy

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Volvo Group AB, Department of PSD, Gothenburg, Sweden.

DISTANCE (D)
POSITION (LAT, LONG)
ROAD DATA
WIND
ETC.



$$d = \sum (d_i), i = 0 \text{ to } n$$

$$f(D) = \{\text{SLOPES, CURVATURE, ROAD SIGNS...}\}$$

RESULTS

Error Variation

ranging from 0.0006 to 0.12 % (350 kms) and 0.002 to 0.25 % (1590 kms)

Comparing and Increasing the efficiency in Computation and data processing

Identified

- Most Accurate
- Least Accurate
- General Trend

references:

- G.V. Windarni and A. Setiawan, "COMPARATIVE ANALYSIS OF VINCENTY AND GEODESIC METHOD APPROACHES IN MEASURING THE DISTANCE BETWEEN SUBDISTRICT OFFICES IN SALATIGA CITY", BAREKENG: J. Math. & App., vol. 16, no. 4, pp. 1207-1220, Dec. 2022.
- L. Romano, P. Johannesson, E. Nordström, F. Bruzelius, R. Andersson and B. Jacobson, "A Classification Method of Road Transport Missions and Applications Using the Operating Cycle Format", in IEEE Access, vol. 10, pp. 73087-73121, 2022, doi: 10.1109/ACCESS.2022.3188872.
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Poster 3:

**Vehicle motion control interface for L4
automated driving**

Rishabh Parmar and Bala Chinni, Volvo Trucks
and Chalmers

V O L V O

VEHICLE MOTION CONTROL INTERFACE FOR L4 AUTOMATED DRIVING

FOCUS ON – MAXIMUM SWEEP PATH WIDTH

Master thesis | Rishabh Parmar , Bala Naga Bharath Chinni – Chalmers University of Technology

Supervisors : Esteban Gelso and Maliheh Kati Sadeghi - Volvo Trucks

Examiner: Nikolce Murgovski – Chalmers University of Technology

Volvo Trucks

Background and Aim

- For heavy combination vehicles, navigating roundabouts is critical due to the vehicle's swept path, which are vital for the safe operation of these vehicles.
- Providing the Automated Driving System (ADS) with a parameterized vehicle model to support its planning.

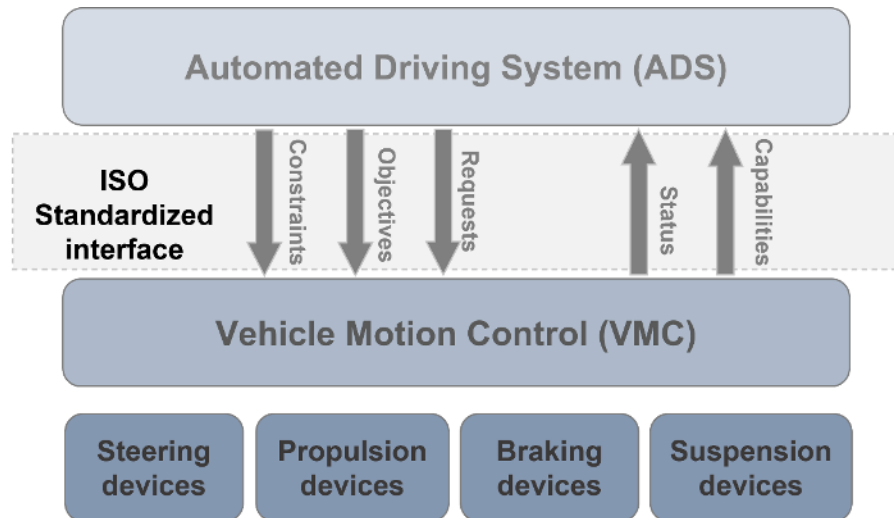


Fig. 1: ISO standardized interface connecting Automated Driving System (ADS) and Vehicle Motion Control (VMC).

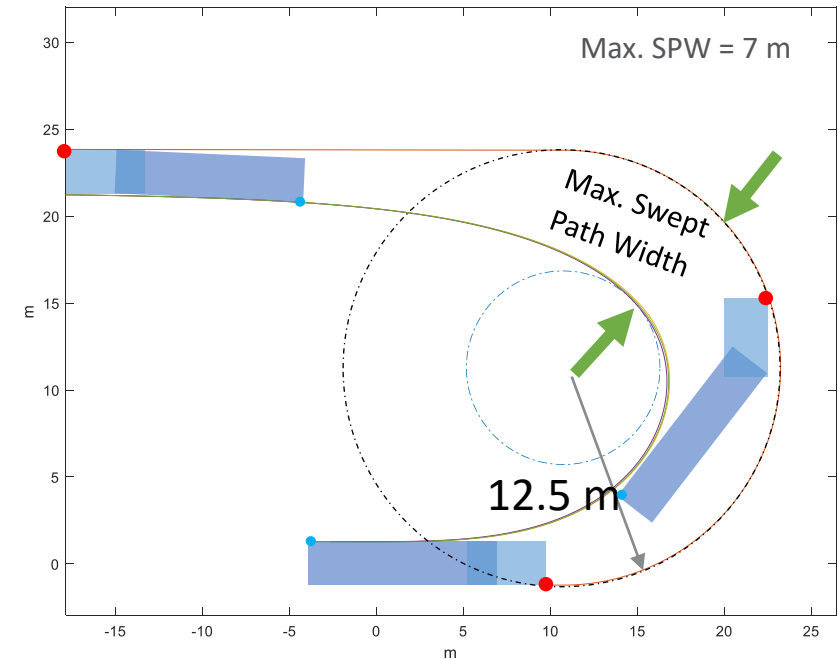


Fig. 2: Maximum swept path width illustration

Results

- Maximum swept path width varies based on turn ,speed and road friction.
- Active steering can reduce the swept path width for a combination vehicle.
- Fig. 3 Illustrates the interaction between the **Automated Driving System (ADS)** and the **Vehicle Motion Control (VMC)** where VMC parameterizes and give $x_0(t)$ vector which then can be used by ADS .

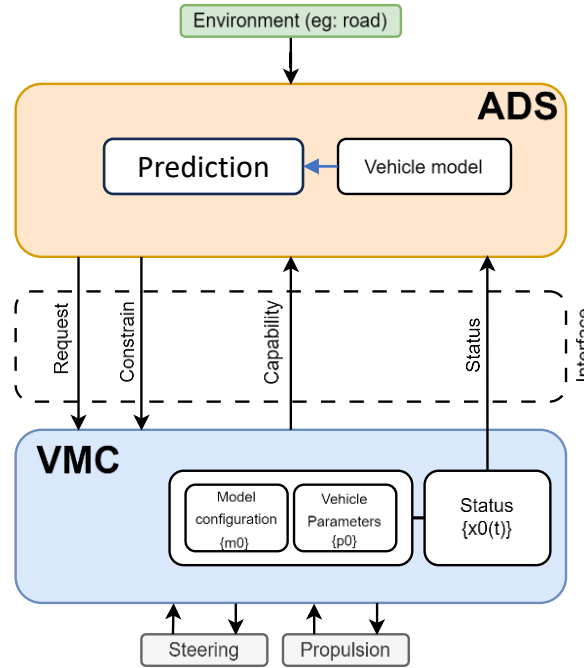


Fig. 3 Interface

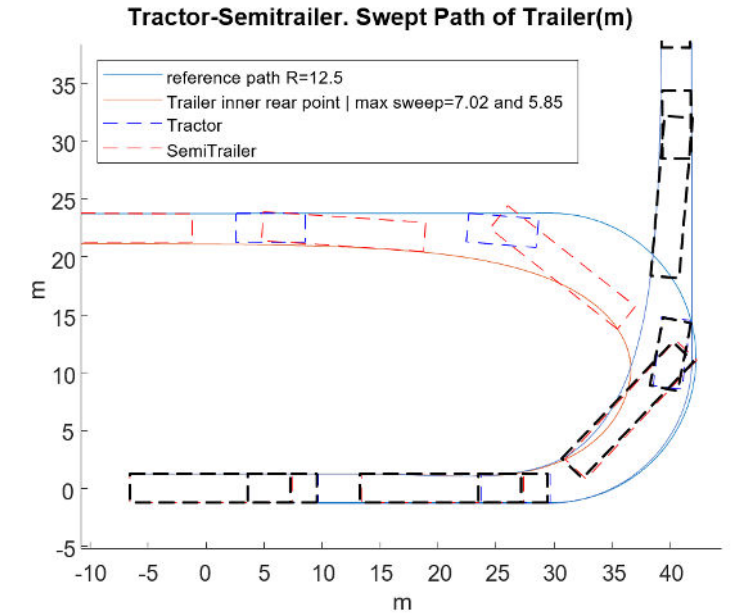


Fig. 5: Swept path width for 90 ° and 180° turn

Model Combination	Maximum Swept Path Width 12.5 m (U-Turn) $v_x = 1$ m/s
Rigid Truck (7.5 m)	2.51 m
Tractor Semi-Trailer (13.6 m)	7.9 m
A-double (Tractor SemiTrailer (13.6 m) + Dolly + SemiTrailer (13.6 m))	10.05 m

Table. 1: Swept path width for various Heavy vehicle combinations

Off-tracking of wheels of last unit and SweptPath(m)

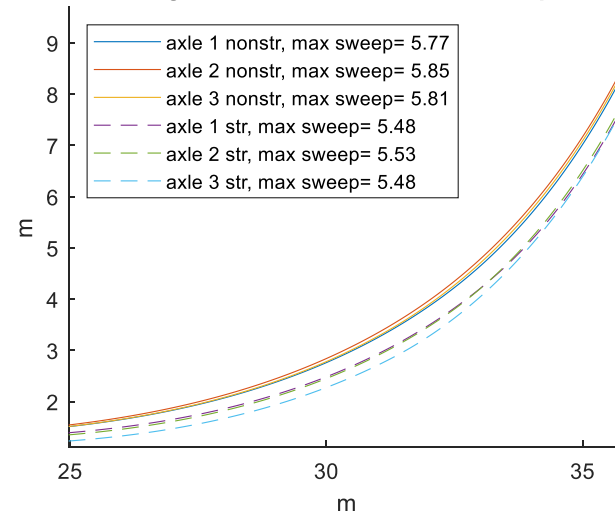


Fig. 4: Steering on Trailer affect on Swept path width

Tractor/Semi(13.6 m)/Dolly/Semi(13.6 m). Path followed by last unit(m)

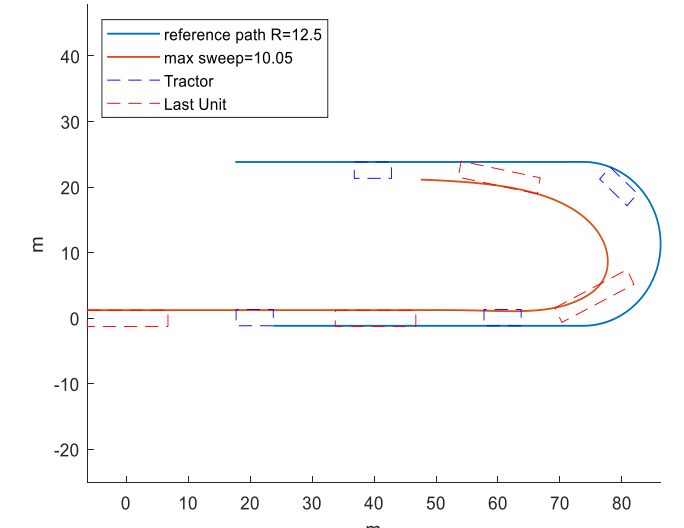


Fig. 6: Swept path width for A-double combination (U turn R=12.5)

V O L V O

THANK YOU

Poster 4:

Modelling of truck tyre wear

Nathan Hedouin, Scania and ESTACA

Poster 5:

Trailer backup assist using steer by wire

Jakob Roempke, Chang Liu, KTH

Trailer backup assist using Steer by Wire*

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1 Abstract

Backing up a trailer can be a daunting task, even for experienced drivers. The main challenge being the non-minimum-phase property of car-trailer kinematics resulting from off-axle interconnections. With steer-by-wire systems, the mechanical connection between the steering wheel and the road wheels is replaced by an electrical connection. The aim of this paper is to help the driver to steer the trailer directly and not have to worry about the non-minimum phase property of the car-trailer kinematics causing unstable motion during reversal. This is achieved with a steer-by-wire system coupled with a closed-loop trailer reversal control. This paper investigates which sensor information is necessary and develop ways to obtain this information. Furthermore, the paper presents how to use the steering wheel input and torque feedback to interact with the backup assist function. The developed control and user interaction is subjectively and objectively evaluated using computer simulation and a physical prototype of a vehicle equipped with steer-by-wire.

The results from the simulation experiments shows that drivers with and without experience of driving a trailer can do the wanted manoeuvres with higher accuracy as well as within a short time span with the controller that is developed in this thesis.

2 Literature review

2.1 Bicycle model for the car based on geometry

The way that the kinematic model of the car trailer system will be derived for this thesis will be based on the method that is used in [1]. This model which will be used in the car to make it possible to control the angle or angular rate between the trailer and car is based on a geometric analysis related to the method that is used in [1]. The model is based on a bicycle model of a car connected to a trailer.

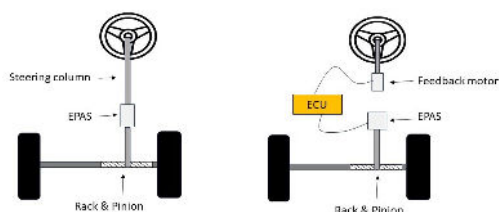
There are both fixed variables such as the wheelbase, length from the tow hook to the rear axle of the car as well as the length from tow hook to the axle on the trailer, and also values that will change with time such as, actual and demanded car-trailer angle ϕ , the actual and demanded angular rate $\dot{\phi}$ as well as the speed of the car.

The angle between the car and trailer ϕ is defined in this thesis as seen in Equation 1 where θ_1 and θ_2 is the car and trailer global angle respectively. This can be seen in Figure 2:

$$\phi = \theta_1 - \theta_2 \quad (1)$$

The angular rate of the angle ϕ is defined in this thesis as:

$$\dot{\phi} = \dot{\theta}_1 - \dot{\theta}_2 \quad (2)$$



(a) Conventional steering

(b) Steer-by-Wire

Fig. 1: The difference between conventional steering and Steer-by-Wire.

The points IC_T and IC_V are the trailers and cars rotational centres respectively. These are defined as the points around which the bodies rotate.

* Supported by Volvo Cars

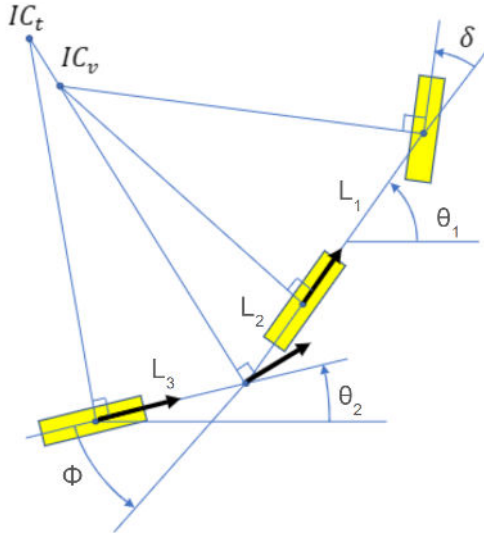


Fig. 2: An overview of the kinematic model.

Based on the bicycle model in Figure 2 equations can be derived on how the angular rate of the angle between the car and trailer ($\dot{\phi}$) depends on the angle between the car and trailer (ϕ) and the steering angle (δ) [1], which can be seen in Equation 3.

$$\dot{\phi} = \frac{-v \cdot \sin(\phi)}{L_3} + \frac{-v}{L_1} \cdot \left(1 + \frac{L_2 \cdot \cos(\phi)}{L_3}\right) \cdot \tan(\delta) \quad (3)$$

As stated in [1], this equation is non-linear. The non-linearity is a problem for stability and robustness if it is to be controlled [2]. This means that a linearisation of the equation has to be done. There are multiple ways to linearise this system but a simple and effective method of doing it is to use a first-order Taylor expansion, which is also used in [1]. This is a local linearisation and the linearisation will be done around 0 for ϕ .

The first-order Taylor expansion gives the following equation:

$$\dot{\phi} = -\frac{v}{L_3} \cdot \phi - \frac{v}{L_1} \cdot \left(1 + \frac{L_2}{L_3}\right) \cdot \delta \quad (4)$$

If the equation is derived for δ , it gives:

$$\delta = \frac{\dot{\phi} + \frac{v}{L_3} \cdot \phi}{\frac{v}{L_1} \cdot \left(1 + \frac{L_2}{L_3}\right)} \quad (5)$$

2.2 Stability analysis

To analyze the stability of the system, pole position analysis of the system is required. The stability with and without the assist system are examined by checking the eigenvalues of

the state-space equations. For the car-trailer system without any backup assist, the main equation is Equation 4. The eigenvalue of the system is in the form of $-\frac{v}{L_3}$, which lies in the left-hand side of the complex plane if and only if v takes on a positive value, indicating forward driving. In this sense, the system is unstable when reversing. The analysis result matches what happens in reality since driving forward with a trailer is controllable and stable while reversing with a trailer is not. The instability of reversing raises the need for an assist system to ensure that the eigenvalue is in the left-hand side of the complex plane.

2.3 Controller

Based on Equation 4 and Equation 5, whether to control ϕ or $\dot{\phi}$ can be chosen and corresponding control loops can be designed and tuned. Equation 5 serves as the steering angle block that connects the control loop with the steering reg on the car. In this section, the control loop designs are shown and explained based on the value to control. Details regarding the functions and instructions for different assist modes are shown in Chapter 6. No matter what driving modes are going to be implemented on the car, it is either controlling ϕ or $\dot{\phi}$ of the system.

Controlling ϕ with $\dot{\phi}$ as the control value For controlling ϕ , the control value should be given to $\dot{\phi}$ since this is the first-order derivative with time of the former. The control value is based on the difference between the real-time ϕ and the target ϕ , which is the error. The value is fed to a P controller with the output as the $\dot{\phi}$ input to the steering angle block. The output of this block is directly the steering angle that should be on the front wheels of the car. The flow chart is shown in Figure 3.

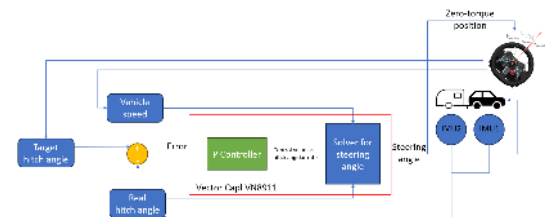


Fig. 3: The flow chart for controlling ϕ .

To do the stability analysis, the control value in Equation 6 needs to replace $\dot{\phi}$ in Equation 5 as shown in Equation 7 where k is the k_p for the P controller.

$$\dot{\phi} = k \cdot (\phi - \phi_{target}) \quad (6)$$

$$\delta = \frac{(k \cdot (\phi - \phi_{target})) + \frac{v}{L_3} \cdot \phi}{\frac{v}{L_1} \cdot (1 + \frac{L_2}{L_3})} \quad (7)$$

By applying Equation 7 to Equation 4, the eigenvalue of the system is changed to $(-k - \frac{2v}{L_3})$. When the car-trailer system is reversing, v is a negative value. To ensure the stability of reversing, k needs to be larger than $-\frac{2v}{L_3}$ so that the eigenvalue lies in the left-hand side of the complex plane.

3 Length estimation of trailer

For the model that is developed in this project to work, there is a need for an input of the length of the trailer to its wheel axle. This is a cumbersome process if it is supposed to be done by the driver in many ways. The length of the trailer is not always known and it can also be difficult to accurately measure the length. Therefore it would be beneficial if there was a way for the car to determine the length of the trailer on its own accurately. This could be done using a error prediction method, using the parameter value that minimizes the quadratic sum of the rediction errors [2]. But to make the system compact and more efficient, another simple yet effective way of determining the length of the trailer is created and utilized. This is by using Equation 3 to derive the length of the trailer L_3 . The equation for L_3 can be seen in Equation 8.

$$L_3 = \frac{-v * (\sin(\phi) * L_1 + L_2 * \cos(\phi) * \tan(\delta))}{\dot{\phi} * L_1 + v * \tan(\delta)} \quad (8)$$

This equation will give the length of the trailer whenever all of the variables in the equation is stable. The system automatically starts the length estimation the moment the driver drives the car forward after the trailer is connected to the car for the first time. The car-hitch angle as well as its angular rate, steering angle, and car speed will be recorded during a short period when there is a stable set of inputs from the driver and the distance between the trailer axle and the tow hook can be calculated using Equation 8. The easiest way to get stable inputs is to do the start the length estimation while going forward in a corner with a fixed steering angle. This means that $\dot{\phi}$ will move towards zero and the speed and ϕ will move towards a steady value. This will make the equation easier to solve as well as make the result more stable. If the trailer gets replaced by a different one, the system will work again to estimate the new geometry. In theory, the estimation also works during reversing if the trailer assist system is not active. After the automatic length estimation, the result will be sent to the controller as input of trailer length. The driver can also choose to manually input the length.

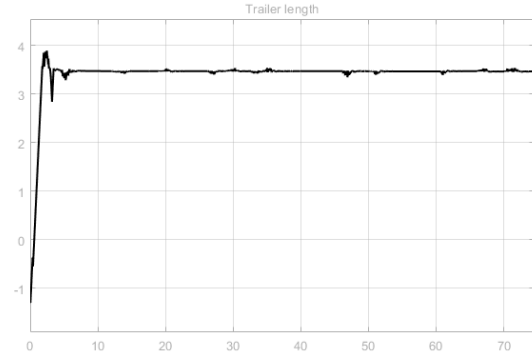


Fig. 4: Simulation results of the length estimation of a 3.5 meter long trailer.

As can be seen from Figure 4, the trailer length estimation gives reasonable values after about 5 seconds. This delay is due to the road being straight in the beginning and leads to that there is no steering angle to start with.

3.1 Determining the angle using yaw sensors

The angle can also be determined using yaw sensors. This works by having one yaw sensor on the car and the other on the trailer. The angle between the car and trailer is then determined by integrating the difference between the car and trailer yaw angular rate ($\dot{\theta}_1$ and $\dot{\theta}_2$) over time.

One problem with this way of determining the trailer-car angle is that there is no way for the car and trailer to know their global angle relationship, therefore a calibration of what is zero degree angle between the car and trailer has to be done. This could be done by letting the car go straight on a plane surface for a short period to ensure the angle between them is zero.

Another problem with this setup is that it is very common to have integration drift when IMUs are used. This leads to that the heading of the IMU will drift towards one direction over time. There are several more or less complicated ways to reduce the influence of the IMU drift. In this thesis we will observe the drift over time and when the car is stand still we will remove the observed drift rate from the yaw rate and hence reduce the influence of the integration drift.

4 How to avoid jack-knifing

To find the critical angle between the trailer and the car that was mentioned in the literature review, calculations can be done to find when the car and trailer share the same instant centre of rotation since this is the turning point where the car stops to have a higher angular speed than the trailer, under the assumption that the maximum geometrical angle on the front wheels is reached. Since the instant centre of rotation is shared by both the car and the trailer, it is defined

as O in Figure 5, With this theory, the critical angle can be calculated based on equation 9. [2] [3]

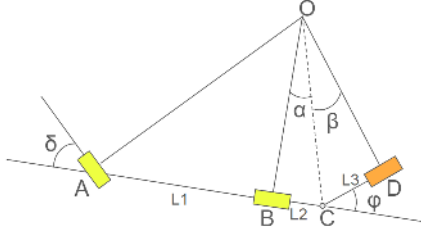


Fig. 5: How the critical angle is determined.

$$\phi_{critical} = \arctan\left(\frac{L_2}{OB}\right) + \arcsin\left(\frac{L_3}{OD}\right) \quad (9)$$

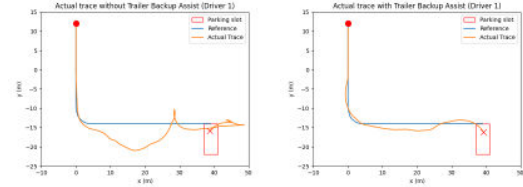
$$OB = \frac{L_1}{\tan(\delta)}, \quad OD = \sqrt{OB^2 + L_2^2 - L_3^2} \quad (10)$$

where OB and OD are the instant radius of rotation of the rear axle on the car and the trailer axle respectively, and L_2 and L_3 are the distance between the car back-end to the tow hook and the distance to the trailer's axle respectively.

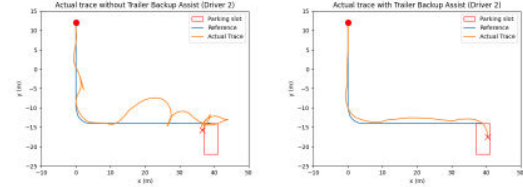
This means that to prevent jack-knifing from occurring, ϕ should not be allowed to pass the critical angle. The width of the car and trailer is not considered in this calculation which might lead to that the critical angle might not be reached before the car and trailer hit each other if the car or the trailer is very wide.

Parking experiment

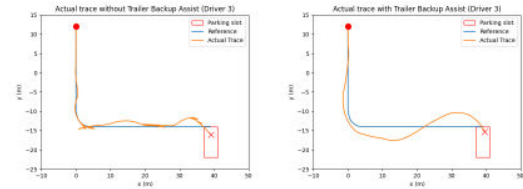
The second experiment is set under a different scenario. This time, the reference lane is no longer a straight road, but a path to park the trailer into a parking lot.



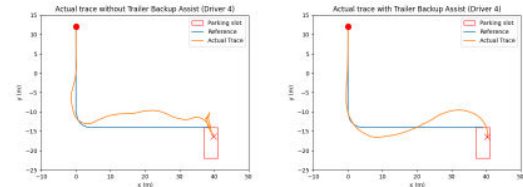
(a) Driver 1 without assist (b) Driver 1 with "Hybrid mode"



(c) Driver 2 without assist (d) Driver 2 with "Hybrid mode"



(e) Driver 3 without assist (f) Driver 3 with "Hybrid mode"



(g) Driver 4 without assist (h) Driver 4 with "Hybrid mode"

In this experiment, time-to-reach is the main output for evaluation. Similarly, 5 test drivers are put into exactly the same driving scenario depicted as above. Their driving traces with and without the assist are recorded and shown in ???. Qualitatively speaking, driving without the assist normally requires more correction movement and less possibility in keeping to the reference lane due to greater challenge in hitch angle control. With these disadvantages, time-to-reach shows a great difference between assisted and non-assisted driving. For driver 1,3 and 5 who do not have much trailer parking experience, time-to-reach has been significantly reduced by 4 to 5 times. For driver 2 and 4 who are experienced, it has also been reduced by 3 times, either with the "Hybrid mode" or "joystick mode". The "joystick mode", even though considered less intuitive than the "Hybrid mode", shows great potentials since it also reduces the time-to-reach for the drivers.

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Poster 6:
**Pendulum turn, the optimal racing line
through a hairpin curve?**

Lukas Jonsson, LiU