Performance Characteristics for Automated Driving of Long Heavy Vehicle Combinations Evaluated in Motion Simulator

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Abstract—This paper presents a driving simulator experiment in which manual and automated driving of a prospective long vehicle combination has been studied. Based on post analysis of manual and automated driving trajectories, characteristic measures reflecting the manual drivers behavior have been proposed. It was observed that the drivers had a round shape of the utilized accelerations while negotiating the curves. A similar shape was found when using an objective function which included minimizing the resultant jerk vector.

I. INTRODUCTION

To meet future environmental goals and upcoming legislation on transported goods [1], studies on the use of longer vehicle combinations than those agreed upon in EU directive 96/53 have been conducted e.g. in [2], [3]. The prospective Long Vehicle Combinations (LVCs) described in [4] are based on a modular concept and typically range between 25-35[m] in length and have at least two articulated joints. The productivity of these combinations is seen to improve by approximately 20 percent [5]. An important aspect in a potential introduction of the prospective LVCs is how these vehicle combinations should be managed safely in traffic.

A hypothesis is that future automated driving functionality designed for LVC are to some extent different from solo vehicles. This is because consideration must be taken to the length of the vehicle combinations as well as a more extensive longitudinal and lateral vehicle dynamics [6] and [7]. In a conducted pilot project for LVC utilization in Sweden [8], the drivers orally states that the increased vehicle length and weight needs to be compensated by the driver using extended planning including increased planning horizon. Typical statements are: "earlier braking due to increased weight", "avoid stopping in uphills in winter conditions" and "important to maximize the radius during low speed cornering" [8]. This type of behavior needs to be included in automated functionalities of LVC. In this paper, a driving simulator study is presented. The objective is to understand how experienced truck drivers drive a LVC effectively and safely during a transport mission. In order to capture the drivers utilization of the vehicle performance characteristics e.g. rearward amplification (RWA) and high speed steady-state off-tracking (HSSO), a Swedish country road has been selected as driving route. The route has a relatively high curve density and was intended to reflect a demanding LVC transport mission but without lane changes and safety critical maneuvers. In the driving scenario additional roadusers has been excluded.

The studied LVC was an A-double combination illustrated in Fig. 1, which consisted of a 6x4 tractor unit followed by a tree-axle semi-trailer, two-axle dolly and a tree-axle semi-trailer unit. The total vehicle length was 32[m] and the total weight was set to 80[t]. The driver’s manual lane keeping and speed profile was stored for post analysis and in addition, driver subjective acceptance for automated driving trajectories was also conducted in the large movement simulator. During post analysis of the trajectories from the manual driving, new optimal control problem formulations were compared with the manual driving trajectories. Proposal of characteristic measures which are significant for the driving of the experienced truck drivers are given. The proposed driving characteristics can be utilized in the framework of extended automated driving functionality for LVCs.

![Fig. 1: Illustration of an A-double vehicle combination](image)

The remainder of the text is organized as follows: In section II the vehicle modeling and the important vehicle characteristics are presented. In section III, the control design of the automated driving modes are given. In Section IV the driving simulator experiment is presented. In section V the results from the simulator experiment is given and in section VI conclusions are drawn.

II. VEHICLE MODELING

In this section three different mathematical vehicle models are presented. The vehicle models have been used in different context to describe the dynamics of the A-double combination. In common, all three vehicle models have been used for the lateral dynamics in high speed. First, a kinematic one-track model is described. This model has been used...
in the predictive control design of the automated driving. Secondly, a one-track model with linear tire slip is presented. The model has been used in the post analysis for generating optimal driving trajectories for a selected curve which was compared with the simulator experiment results. Finally, a high fidelity two-track model is described. This model was used as a vehicle plant in the simulator experiment. The lateral performance measures connected to high speed maneuvering [6], are explained, evaluated, and compared for all models.

A. Kinematic one-track model

A main model assumption of the kinematic one-track model is that all wheels follow Ackerman steering geometry and they are ideally tracking their own direction (zero side slip). In Fig. 2 the vehicle parameters are shown. The vehicle motion in the road plane is described by following set of differential equations:

\[
\begin{align*}
\dot{X}_1 &= v_1 \cdot \cos \theta_1 \\
\dot{Y}_1 &= v_1 \cdot \sin \theta_1 \\
\dot{\theta}_1 &= \frac{v_1}{l_1} \cdot \tan \delta \\
\phi_1 &= -\dot{\theta}_1 - \frac{v_1}{l_2} \cdot \sin \phi_1 - \dot{\theta}_1 \cdot \frac{c_1}{l_2} \cdot \cos \phi_1 \\
\phi_2 &= -\frac{1}{l_3} \left[ c_1 \cdot \dot{\theta}_1 \cdot \cos (\phi_1 + \phi_2) \\
&\quad + v_1 \cdot \sin (\phi_1 + \phi_2) \\
&\quad + (l_2 + c_2) \cdot (\dot{\theta}_1 + \dot{\phi}_1) \cdot \cos (\phi_2) \\
&\quad + l_3 \cdot (\dot{\theta}_1 + \dot{\phi}_1) \right] \\
\phi_3 &= -\frac{1}{l_4} \left[ c_1 \cdot \dot{\theta}_1 \cdot \cos (\phi_1 + \phi_2 + \phi_3) \\
&\quad + v_1 \cdot \sin (\phi_1 + \phi_2 + \phi_3) \\
&\quad + (l_3 + c_3) \cdot (\dot{\theta}_1 + \dot{\phi}_1 + \dot{\phi}_2) \cdot \cos (\phi_3) \\
&\quad + (l_2 + c_2) \cdot (\dot{\theta}_1 + \dot{\phi}_1) \cdot \cos (\phi_2 + \phi_3) \\
&\quad + l_4 \cdot (\dot{\theta}_1 + \dot{\phi}_1 + \dot{\phi}_2) \right]
\end{align*}
\]

where \( v_1 \) is the velocity of the first unit rear axle position, \( l_i \) are the wheel base distance of the different units, \( c_i \) are distance from the coupling points to the rear axle positions and \( \delta \) is the front wheel steering angle. The used model parameters are provided in [9].

B. One-track model with linear tire slip

The one-track model was derived using Lagrangian formulation of the vehicle’s motion [10]. The benefit with this starting point is that coupling forces within the vehicle are excluded. The vehicle parameters are illustrated in Fig. 3. The model has been used for generating optimal driving trajectories in cornering situations. The vehicle motion can be described by:

\[
\dot{x} = M^{-1} \cdot (A \cdot x + B \cdot u) - E
\]

where \( x = [y_1, \phi_1, \dot{\theta}_1, \dot{\phi}_2, \theta_2, \theta_3, \phi_3, x_1]^T \) and \( u = [\delta, F_{x,1}]^T \). The model matrices and the used parameters are provided in [10]. The kinematics, steering, and the used tire model have been linearized using small angle assumption. However, the model is still considered to be non-linear due to variable longitudinal velocity and air resistance drag force. The linear cornering stiffness values have been tuned using results from the high fidelity two-track model.

C. High fidelity two-track model

A Volvo in-house developed high fidelity two-track model library was used in the simulator experiments to emulate the detailed vehicle dynamics while driving the A-double combination on uneven road. The high fidelity model is valid in the frequency range 0 to 5[Hz]. The model comprised detailed sub-models of the vehicle chassis, cab suspensions, steering system, powertrain, and brakes. The frame torsion flexibility of the tractor has been considered by using multiple frame bodies connected through springs. Similar chassis set up was used for the dolly and the semitrailers. The units were connected by articulation joints. The Magic Formula tire model [11] with combined slip, dynamic relaxation, and rolling resistance, was used for all tires.

D. Performance based characteristics

As stated, rearward amplification (RWA), high speed steady-state off-tracking (HSSO), high-speed transient off-tracking (HSTO) and yaw damping coefficient (YDC) are important characteristics [6] used in this study. Rearward amplification is the relationship between the maximum movement of the first and last vehicle unit during a specified steering maneuver [12] and vehicle speed. It is usually given in the metrics lateral acceleration gain, or as
here, in yaw velocity gain. It expresses the increased risk for a last unit rollover or swing-out which can occur if a sudden steering maneuver is performed. The RWA has been calculated for the three vehicle models described Sec. II-A, II-B and II-C, using a pseudo-random steering input [12] and constant vehicle speed in the range 50-90[km/h]. In Fig. 4 the one-track model with linear tire slip and the high fidelity two-track model shows similar results with increasing RWA for increasing vehicle speed. The kinematic model cannot correctly capture the dynamics of RWA which is dependent on the generation of lateral tyre forces.

The offtracking characteristics, HSSO and HSTO, both describes the lateral deviation between the path of the front axle and the path of the most severely offtracking axle of the last unit. These measures express the additional space needed for the last unit in a specific steering maneuver and vehicle speed. A positive value of the HSSO and HSTO means that the last unit is tracking inward of the first unit. In Fig. 5 and Fig. 6 HSSO and HSTO values calculated for constant vehicle speed in the range 50-90[km/h] are shown. The HSSO values have been calculated for three levels of constant lateral acceleration: 0.5[m/s²] (top), 1.5[m/s²] (middle) and 3[m/s²] (bottom). Fig. 5 shows that the HSSO values are similar for the one-track model with linear tire slip and the high fidelity two-track model. For a constant lateral acceleration, the values decrease with increasing vehicle speed meaning that for low vehicle speed the last unit is tracking inward of the first unit. For high vehicle speed the last unit is instead tracking outward of the first unit. The HSSO values of the kinematic model is always positive. The model is relatively correct for low vehicle speed but cannot describe HSSO at high vehicle speed. Fig. 6 shows the HSTO values, calculated using a single sine-wave lateral acceleration [12] with the frequency 0.4[Hz] and the lateral acceleration 2[m/s²]. The absolute values of the HSTO for the one-track model and the high fidelity two-track model increase with increasing vehicle speed. This effect is not captured by the kinematic model and the HSTO is instead constant for all vehicle speeds.

The yaw damping coefficient is the damping ratio of the least damped articulation joint’s angle during free yaw oscillations of the vehicle combination after a specific steering maneuver and vehicle speed. A longer decay time might result in higher driver workload and increased safety risk for other roadusers. In Fig. 7 the YDC values have been calculated using a single sinusoidal steering input with the frequency 0.4[Hz] for constant vehicle speed in the range 50-90[km/h]. The YDC for the one-track model with linear tire slip and the high fidelity two-track model decrease with increasing vehicle speed. The kinematic model cannot represent the dynamics of YDC correctly which is in accordance with the results of the RWA and HSTO.

Finally, step response analysis has been carried out for the vehicle speed 70[km/h] in order to compare the dynamic response times of the vehicle models. When applying a steering step input, the first unit of the kinematic one-track model has an instantaneous response and the last vehicle unit responds after approximately 1.7[s]. For the one-track model with linear tire slip the first unit respond after approximately 0.1[s] and the last vehicle unit responds 2.4[s] later. The results for the high fidelity two-track model is similar to the one-track model with linear tire slip. The difference in response times influence the performance of the kinematic one-track model when used as prediction model for path planning, see Sec III-B.2.

The specific values of the different characteristics reflects the actual vehicle configuration, such as number of units, equivalent wheel base, towing bar length and position, and tire characteristics, during a dynamic maneuver. However, while driving a realistic transport mission it is possible and sometimes necessary for the driver to consider the vehicle characteristics and compensate for these to ensure safe maneuvering. Typical examples are: maximize cornering radius while negotiating a curve, lowering the vehicle speed, and/or avoid sudden steering. The evaluation of the performance characteristics during a specific driving route is here defined as utilized performance characteristics and aims to reflect the combination of vehicle performance, road characteristics and driver preference.
with different geometric reference paths were used. In the driving simulator experiment the same longitudinal control system was used for both automated modes.

A. Longitudinal control design

The longitudinal motion control of the vehicle was performed by a velocity interface for coordinating engine propulsion and braking of each wheel in the combination. The coordination of these actuators was performed by Control Allocation formulation according to [13] and [14]. The closed-loop control of the longitudinal velocity was defined as a linear feedback proportional tracking controller using a pre-calculated reference velocity profile \( v_{ref}(s) \) along the distance \( s \). The total longitudinal force for the vehicle combination was defined as

\[
F_i = K_{F_i} \cdot \left( v_{ref} - \frac{v}{\Delta t} \cdot m_{tot} \right) \tag{8}
\]

where \( F_i \) is the input for the control allocation, \( K_{F_i} \) is the proportional gain factor, \( m_{tot} \) is the total vehicle mass, \( v \) is the actual longitudinal velocity, and \( \Delta t \) is a tuning variable. The reference velocity was initially set according to

\[
v_{ref}(s) = \min\left(v_{max}, \sqrt{a_{max}^2/c(s)}\right), s \in [0,s_{end}] \tag{9}
\]

where \( v_{max} \) denotes the speed limit, \( c(s) \) denoted the road curvature and \( a_{max} \) denotes the maximum allowed lateral acceleration which was set to 3 [m/s^2]. Certain distance prior to a curvature the speed was reduced to the maximum velocity allowed for the specific road segment (9). The algorithm for calculating the speed profile for the complete road is shown in [15].

B. Lateral control design

This section describes the two different closed-loop control systems used to handle the vehicle steering in the automated driving modes. Both controls systems were defined as linear feedback tracking controllers using a pre-calculated geometric reference path.

1) Auto1: In the first case, the reference path was defined as the lane center and the requested road wheel angle \( \delta(t) \) was calculated according to

\[
\delta(t) = K_e \cdot (e(t) + \dot{e}(t) \cdot t_p) + K_{ff} \cdot \delta_{ff} \tag{10}
\]

where \( K_e \) is a gain factor, \( t_p \) is a tuning variable, \( e \) and \( \dot{e} \) are the perpendicular distance offset to the centerline and its time derivative. The used values of the gain factor \( K_e = 0.3 [\text{rad/m}] \) and the tuning variable \( t_p = 1 [\text{s}] \), were determined in subjective evaluation to accomplish smooth and comfortable driving.

2) Auto2: In the second case, the geometric reference path was calculated using a path planning algorithm based on a receding horizon optimization approach which was formulated as a Nonlinear Programming (NLP) problem with the open source Interior Point OPTimizer (IPOPT) as a solver in Simulink [9]. The prediction horizon for path planner the was set to 10 [s]. The locally optimized reference path were calculated using the following cost function and constraints

\[
\min_{\bar{C}_0} \sum_{i=1}^{N} \left[ K_j \cdot \dot{r}_j^2 + K_{d_i} \cdot \dot{d}_{i,j}^2 + K_{d_i} \cdot d_{i,j}^2 \right] \tag{11}
\]

subject to

\[
\begin{align*}
-\bar{a}_{1,B} \leq \bar{a}_i & \leq \bar{a}_{1,B} \\
-\bar{J}_i \leq \bar{J}_i & \leq \bar{J}_B \\
-\bar{d}_{1,B} \leq \bar{d}_i & \leq \bar{d}_{1,B} \\
-\bar{d}_{1,B} \leq \bar{d}_i & \leq \bar{d}_{1,B}
\end{align*}
\tag{12}
\]

\[
\begin{bmatrix}
\bar{\theta}_i \\
\bar{\dot{\theta}}_i \\
\bar{\ddot{\theta}}_i
\end{bmatrix} = \bar{C}_0 \cdot \bar{M}_eq \begin{bmatrix}
\bar{C}_{0,1} \\
\bar{C}_{0,2} \\
\bar{C}_{0,3}
\end{bmatrix} \tag{13}
\]

where the optimization variable \( \bar{C}_0 \) is the B-spline control points and \( j = 1, ..., N \) are the number of collocation points. Further, \( K_j, K_{d_i}, \) and \( K_{d_i} \) are gain factors which values were set to \( K_j = 50 \), \( K_{d_i} = 100 \) and \( K_{d_i} = 25 \). The lateral acceleration \( \bar{a}_i \) and the lateral jerk \( \bar{J}_i \) for unit \( i = 1, ..., 4 \) were constrained by lower and upper limits which were set to \( \bar{a}_{1,B} = 1.6 [\text{m/s}^2] \) and \( \bar{J}_{1,B} = 3 [\text{m/s}^3] \). The perpendicular distance offset of unit 1 and 4 with respect to the preferred path, \( \bar{a}_1 \) and \( \bar{d}_4 \), were constrained by the lower and upper limits set to \( \bar{d}_{1,B} = 0.7 [\text{m}] \) and \( \bar{a}_{UB} = 4.2 [\text{m}] \). The equality constraints (13) are needed for the initial state at the start of the prediction. To achieve realtime performance of the receding horizon optimization approach, the kinematic vehicle model, see Sec. II-A was used as a prediction model for path planning. One benefit with the kinematic model is that part of the model exploits differential flatness which enables realtime performance. The trajectory planner had poor closed loop performance when used togather with the vehicle motion control and the high fidelity plant model. The main cause for this is the lack of transient dynamics in the kinematic model, e.g. response time, see Sec.II-D. Instead the optimized reference path and feedforward road wheel angle used in the simulator experiments were pre-calculated with the kinematic model also as a vehicle plant.

The requested road wheel angle \( \delta(t) \) was calculated according to

\[
\delta(t) = K_e \cdot (e(t) + \dot{e}(t) \cdot t_p) + K_{ff} \cdot \delta_{ff} \tag{14}
\]
where $K_e$ and $K_{ff}$ are gain factors, $t_p$ is a tuning variable, $e$ and $\dot{e}$ are the perpendicular distance offset to the centerline and its time derivative and $\delta_{ff}$ refers to a feedforward road wheel angle calculated in the path planning algorithm. The used values of the gain factors $K_e = 0.2$[rad/m], $K_{ff} = 0.1$[-] and the tuning variable $t_p = 1$[s], were determined in subjective evaluation to accomplish smooth and comfortable driving.

IV. SIMULATOR EXPERIMENT

The description of the driving simulator experiment has been divided into four sub-sections: Driving simulator, Participants, Driving scenario and Experiment setup.

A. Driving simulator

The experiment was conducted in VTI driving simulator IV in Gothenburg, Sweden. The simulator is equipped with a motion system that allows for large movements in both longitudinal and lateral direction [16]. The high fidelity two-track model described in Sec. II-C was used to emulate the vehicle dynamics in the simulator. To accomplish high realism considering truck vehicle dynamics, extensive subjective testing was carried out prior to the experiment by experienced truck drivers. The final performance was judged as good with regard to the lateral dynamics and acceptable considering the longitudinal dynamics. One consequence of the chosen simulator setup was that it was not possible for the tractor unit exceeding 1.5[m/s$^2$] steady state lateral acceleration without reaching the physical endpoints of the motion platform. This meant that the route speed limit was reduced in one section including low radius cornering.

B. Participants

The participants consisted of 11 professional truck drivers from a haulage contractor and 7 drivers from Volvo Product Development. All participants had driver license for heavy truck and trailer except two, which had only driver license for heavy truck. Three of the professional drivers regularly drive longer vehicle combinations than normally allowed by legislation. This type of transports are done with special legislation. This type of transports are done with special vehicle characteristics. In order to evaluate the manual driving behavior a new optimal control problem has been formulated with different objective functions. The

TABLE I: Summary of data on participants in experiment. Group A refers to drivers from Volvo and group B refers to the professional truck drivers.

<table>
<thead>
<tr>
<th>Group</th>
<th>Age</th>
<th>Years w. license</th>
<th>km driven/year [10^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Av.</td>
<td>Range</td>
<td>Av.</td>
</tr>
<tr>
<td>A</td>
<td>40</td>
<td>27-64</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>0.1</td>
<td>0.01-0.5</td>
<td>100</td>
</tr>
</tbody>
</table>

C. Driving scenario

The experiment was conducted using a route based on measurement of eight kilometers of the Swedish country road RV180 between Borås and Alingsås. An overview of the main characteristics of the route can be seen in Table II. The road description implemented in the motion simulator included profiles of the road curvature, elevation, lateral inclination and the surface roughness. The traffic flow was right-hand traffic and the road lane width was set to 3.5[m]. The allowed speed limit was set to 70[km/h] at the major part of the road [15].

TABLE II: Characteristics of the route used in simulator experiment. The symbol $(\cdot)$ represent the different data. The total length of the route was 8.3[km].

<table>
<thead>
<tr>
<th>Data</th>
<th>Min</th>
<th>Max</th>
<th>Mean[\cdot]</th>
<th>Std</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curvature [10^{-2}] [1/m]</td>
<td>-6.7</td>
<td>6.8</td>
<td>1</td>
<td>1.5</td>
</tr>
<tr>
<td>Elevation [m]</td>
<td>147</td>
<td>231</td>
<td>190</td>
<td>26</td>
</tr>
<tr>
<td>Lateral inclination [rad]</td>
<td>-0.09</td>
<td>0.07</td>
<td>0.03</td>
<td>0.02</td>
</tr>
</tbody>
</table>

D. Experiment setup

Before starting the experiment the participants received both written and oral information about the simulator and the experiment. The experiment was divided in three modes referred to as 'manual driving', 'auto 1' and, 'auto 2'. The two modes with automated driving differs from each other by the usage of two different setup of the lateral control, see Sec. III-B. All three modes where driven on the same eight kilometer route which took approximately ten minutes each. All participants started the experiment with the manual driving and then the order of the two different automated driving modes was shifted randomly. An overview of the experimental setup is provided in Fig. 8, see further [15].

![Fig. 8: An overview of the driving simulator experiment procedure.](image)

V. RESULTS

In this section, post-analysis of data from the driving simulator experiment are presented. The analysis was divided in two groups; the complete route and selected curve scenario. In the analysis of the complete route, the manual driving was compared with the two automated driving modes from the simulator experiments. Utilized performance characteristics are defined and shown. Accumulated performance characteristics for the complete route was also analyzed.

The analysis of the selected curve scenario allows detailed studies of utilized vehicle characteristics. In order to evaluate the manual driving behavior a new optimal control problem has been formulated with different objective functions. The
optimal control problem simulations included the vehicle model described in Sec. II-B.

A. Complete route

Important vehicle performance characteristics for LVC high speed maneuvering has been introduced in Sec II-D. The utilized performance characteristics are defined by vehicle performance, road characteristics, and specific driver/auto preference. One of these is the utilized rearward amplification (RWₐ) which is defined as

\[
RWₐ_{u,i} = \frac{r_{f,i}}{r_{1,i}}, \quad i = 1, \ldots, N
\]  \hspace{1cm} (15)

where \(i = 1, \ldots, N\) are the synchronized extreme values of the yaw rate of the first unit \(r_{f,1}\) and the yaw rate of the last unit \(r_{f,N}\). The threshold value \(r_{min}\) has been used to remove small extreme values not connected to lateral maneuvering of the vehicle. Fig. 9 shows the utilized RWₐ for the manual, auto 1, and auto 2 modes and the vehicle performance RWₐ. The top panel in Fig. 9 shows that utilized RWₐ in the manual driving up to 60[km/h] can be higher than vehicle performance RWₐ. One reason for this is that road input such as lateral inclination and the surface roughness influence the utilized RWₐ. For speeds higher than 70[km/h] the manual driving has lower utilized RWₐ than vehicle performance RWₐ. For auto 1 and auto 2 modes the utilized RWₐ around 70[km/h] is quite close to the vehicle performance RWₐ. The auto modes were limited in maximum speed of 70[km/h].

Utilized high speed offtracking (HSOₐ) has been defined as

\[
HSOₐ_{i} = (d_{i,11} - d_{i,1}) \cdot \text{sgn} \left( \frac{dθ_{i,1}(s)}{ds} \right), \quad i = 1, \ldots, N
\]  \hspace{1cm} (16)

where \(i = 1, \ldots, N\) are the synchronized extreme values of \(d_{i,1}\) and \(d_{i,11}\) which are the perpendicular centerline distance offset for axle 1 and 11, \(θ_{i,1}\) is the heading of the first unit. To account for left or right curve the sign of \(\frac{dθ_{i,1}(s)}{ds}\) has been used. The threshold value \(HSOₐ_{min}\) has been used to remove small extreme values not connected to lateral maneuvering of the vehicle. The HSOₐ aims to capture effects from both HSSO and HSTO. In addition, the HSOₐ is also influenced by road input such as lateral inclination and the surface roughness. Fig. 10 shows the utilized HSOₐ for the different driving modes, manual driving (top), auto 1 (middle) and auto 2 (bottom). For low speeds the HSOₐ shows inward offtracking and for higher speeds the inward offtracking is reduced.

![Fig. 9: Utilized RWₐ vs vehicle speed. The manual driving (top), auto 1 (middle) and auto 2 (bottom) have been compared to vehicle performance RWₐ (solid line).](image1)

![Fig. 10: Utilized high speed offtracking vs vehicle speed for the modes manual driving (top), auto 1 (middle), and auto 2 (bottom). A positive value means that the last unit is tracking inward of the first unit](image2)

The utilized performance characteristics have been considered together with other driving characteristics as cost function components in an optimal control problem. The chosen characteristics aims to reflect the features traffic safety, ride comfort, transport efficiency, and general LVC traffic behavior. All different characteristics, apart from the final driving time \(t_f\), were squared and summed up over the complete route and normalized using the maximum value of the three compared driving modes. This was done in order to compare the manual driving cost function components with the automated modes. The cost function components were defined as

\[
C_{j,k} = \frac{\sum_{i=1}^{N} \left( \frac{(d_{i,j,k}^2)}{N_{j,k}} \right)}{\max_{j} \sum_{i=1}^{N} \left( \frac{(d_{i,j,k}^2)}{N_{j,k}} \right)} \cdot j = 1, \ldots, 12 \hspace{1cm} (17)
\]

\[
C_{j,k} = t_{f,k} \cdot j = 13 \hspace{1cm} (18)
\]

where \((\cdot)\) represents the characteristics defined in Table III, and \(k\) represents the driving modes manual, auto 1, and auto 2.

Fig. 11 shows that the cost function components RWₐ and HSOₐ were slightly smaller in the manual driving than in auto 1 and auto 2 mode. The centerline distance offset components \(d_{1}\) and \(d_{11}\) were significantly smaller for the auto 2 mode than the auto 1 mode. This can be due to the control system gain settings. The offset components for the manual driving mode was performing in between mode auto 1 and auto 2. Considering the lateral jerk and the longitudinal deceleration components, the levels were much lower, about 60-70 %, for the manual driving than both automated modes. This indicates that the manual driving was performed smoother and thus focused on the feature ride comfort. This is also supported by the negative longitudinal jerk in Fig. 11 which was much smaller, about 60 %, for the manual driving than than both automated modes. Fig. 11 also...
TABLE III: Characteristics used in the optimal control problem cost function.

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Symbol</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Utilized rearward amplification</td>
<td>RWA_\text{u}</td>
<td>[-]</td>
</tr>
<tr>
<td>Utilized offtracking</td>
<td>HSO_\text{u}</td>
<td>[m]</td>
</tr>
<tr>
<td>Centerline distance offset, axle 1</td>
<td>d_1</td>
<td>[m]</td>
</tr>
<tr>
<td>Centerline distance offset, axle 11</td>
<td>d_{11}</td>
<td>[m]</td>
</tr>
<tr>
<td>Lateral acceleration, unit 1</td>
<td>a_{y,1}</td>
<td>[m/s^2]</td>
</tr>
<tr>
<td>Lateral jerk, unit 1</td>
<td>\dot{a}_{y,1}</td>
<td>[m/s^3]</td>
</tr>
<tr>
<td>Positive longitudinal acceleration, unit 1</td>
<td>a_{i,1}</td>
<td>[m/s^2]</td>
</tr>
<tr>
<td>Negative longitudinal acceleration, unit 1</td>
<td>a_{i,-1}</td>
<td>[m/s^2]</td>
</tr>
<tr>
<td>Positive longitudinal jerk, unit 1</td>
<td>\dot{a}_{i,1}</td>
<td>[m/s^3]</td>
</tr>
<tr>
<td>Negative longitudinal jerk, unit 1</td>
<td>\dot{a}_{i,-1}</td>
<td>[m/s^3]</td>
</tr>
<tr>
<td>Road wheel angle</td>
<td>\delta</td>
<td>[rad]</td>
</tr>
<tr>
<td>Yaw rate, unit 4</td>
<td>\gamma_4</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>Final driving time</td>
<td>t_f</td>
<td>[s]</td>
</tr>
</tbody>
</table>

Additional optimal control simulations have been conducted in order to extend the analysis of the results from the experiment. The simulations includes the vehicle model described in Sec. II-B. The optimal control problem formulated as a NLP with the three different objective functions, was defined as

Objective function 1, 2, and 3:

$$\min t_f, \min \sum_{i=1}^{N} \sqrt{a_{i,.1}^2 + a_{i,.2}^2 + \dot{a}_{y,1}^2 + \dot{a}_{y,2}^2} \min \sum_{i=1}^{N} \sqrt{a_{i,.1}^2 + a_{i,.2}^2 + \dot{a}_{y,1}^2 + \dot{a}_{y,2}^2}$$

subject to

$$0 \leq g(x(s_i), x(s_j))$$

$$u^{LB} \leq u(s_i) \leq u^{UB}$$

$$x(0) = x_0$$

where $a_{i,.1}$ and $a_{i,.2}$ are the longitudinal and lateral acceleration of the first unit front axle and $\dot{a}_{y,1}$ and $\dot{a}_{y,2}$ are the corresponding longitudinal and lateral jerk. The front axle position of the first unit corresponds well to the longitudinal position of the driver. The inequality constraint components $g$ represent the lateral acceleration of unit 1 and 4 which were constrained by an upper and lower limit of 1.5[m/s^2], the longitudinal velocity of unit 1, which was constrained in the interval 30-70[km/h] and the perpendicular centerline distance offset of axle 1 and 11 which were constrained by an upper and lower limit of 0.5[m]. Also, the final longitudinal velocity was constrained by a lower limit of 60[km/h]. The model input $u$ representing the front wheel steering angle $\delta$ and the longitudinal force $F_{x,1}$ was lower and upper bounded by the limits $\delta \in [-20,20][deg]$ and $F_{x,1} \in [-3 \cdot m_{tot}, 1 \cdot m_{tot}][N]$. The total mass $m_{tot}$ was set to 80[t]. The problem was solved by using the Matlab function fmincon [17].

The top panel in Fig. 13 shows the centerline distance offset for the front axle. The mean value of the manual driving was compared with the two automated modes and the results from new optimal control problems. Just before peak in maximum curvature, at distance 265[m], the main difference for the modes manual and auto 1 and 2 is seen. The manual driving is having a positive offset of approximately 0.25[m] in average while the auto modes have negative offset of approximately 0.25[m], see Fig. 13. When studying the simulations results from opt 1, opt 2 and opt 3 shows similar tendency as the manual driving mode, i.e. positive offset. The bottom panel in Fig. 13 shows the centerline distance offset for the last axle. The results for the last axle is similar to the front axle but with a time delay.

Fig. 14 shows the vehicle longitudinal velocity for the front axle in the curve scenario. The mean value of the manual driving is compared with the two automated modes and the results from new optimal control problems. Comparing the manual driving with the automated modes it can be noted that both the absolute level of the deceleration and the longitudinal jerk is higher for the automated modes. The manual driving is significantly smoother.
In Yamakado et al. [18], the coordination of the longitudinal and lateral acceleration for smooth and safe driving was investigated by using g-g diagrams [19] and relate lateral jerk to longitudinal acceleration. Here, similar approach of studying g-g diagrams was conducted for the first vehicle unit, see Fig. 15. Again, the mean value of the manual driving was compared with the two automated modes and the results from new optimal control problems. Similar round shaped g-g diagram was seen for the opt 3 solution when compared with the average of the manual driving mode for the curve scenario.

![G-g Diagram](image)

**Fig. 15:** G-g diagram for left curve. Manual driving (top left), auto 1 (top center), auto 2 (top right), opt 1 (bottom left) and opt2 2 (bottom right).

**VI. CONCLUSIONS**

A main hypothesis in this study has been that driver acceptance is important when introducing automated driving and safety functionality to long vehicle combinations. As a tool for studying manual/automated driving or designing automated driving, utilized characteristics have here been introduced. Typical performance characteristics are utilized rearward amplification, utilized high speed offtracking, and utilized accelerations. These characteristics can then either be found in an objective function and/or as constraints when defining an optimal control problem formulation.

An interesting observation was done while studying how the manual drivers coordinated the utilized longitudinal and lateral acceleration while negotiating a left curve, they had a rounder shape of the utilized accelerations, see g-g diagrams Fig.15. Similar shape was found when using an objective function which included minimizing the resultant jerk. Thus, this is believed to be important for the driver acceptance of automated driving of LVCs.

**REFERENCES**