A path-following driver model with longitudinal and lateral control of vehicle's motion

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A path-following driver model with longitudinal and lateral control of vehicle's motion

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Abstract The use of closed-loop driver models is important for accurate vehicle simulations and in active safety systems evaluation. In this paper we present a combined longitudinal-lateral controller that is regulating the steering angle and throttle/brake levels by previewing the path ahead of the vehicle. The lateral steering controller is using, as input, the heading and position deviation between the vehicle and the road. The controller is using fixed gains with a simple gain scheduling based on the vehicle's speed. The longitudinal speed controller is using the curvature of the path ahead of the vehicle to determine the appropriate velocity of the vehicle. The longitudinal-lateral controller is tested by driving a double-lane change (ISO 3888-2) and a lap around a racing track.

Ein dem Weg folgendes Fahrermodell mit Längs- und Querregulierung der Fahrzeugbewegung

Zusammenfassung Der Gebrauch von Closed-Loop-Fahrermodellen ist für genaue Fahrzeugsimulationen wichtig. In diesem Beitrag stellen wir einen kombinierten Längs-Quer-Regler vor, der den Lenkungswinkel und die Gas-/Bremsstärke reguliert, indem er den Weg vor dem Fahrzeug voraussieht. Der Querlenkregler verwendet als Eingangswerte die Kurs- und Positionsabweichung zwischen dem Fahrzeug und der Straße. Der Regler arbeitet mit festen Steigerungswerten und mit einer einfachen Steigerungsplanung, die auf der Fahrzeuggeschwindigkeit basiert. Der Längsgeschwindigkeitsregler verwendet die Krümmung des Weges vor dem Fahrzeug, um die passende Geschwindigkeit des Fahrzeugs zu bestimmen. Der Längs-Quer-Regler wird geprüft, indem man einen Doppelspurwechsel (ISO 3888-2) und eine Runde um eine Rennstrecke fährt.

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List of symbols

s	[m]	distance covered along the desired path
k_t	[1/m]	desired path curvature
ψ_t	[° deg]	desired path tangent angle
x_t, y_t	[m]	desired path coordinates
μ_x, μ_y	[–]	longitudinal and lateral tyres-road friction coefficient
x, y	[m]	coordinates of the vehicle's centre of mass
ψ	[° deg]	heading angle of the vehicle
U, V	[m/s]	longitudinal and lateral speed of the vehicle
m	[kg]	mass of the vehicle
l	[m]	wheelbase length of the vehicle
a, b	[m]	distance of the front and rear axle from the centre of mass
C_1, C_2	[–]	cornering stiffness of the front and rear tyres of the vehicle
K	[–]	stability factor of the vehicle
T_p	[s]	preview time of the driver model
$a_{b,\max}$	[m/s^2]	braking deceleration
$a_{y,\max}$	[m/s^2]	maximum lateral acceleration
K_U	[–]	longitudinal speed control gain
$[W_\psi]_{1\times N_2}$	[–]	heading error weight matrix
$[W_d]_{1\times N_2}$	[–]	position error weight matrix
$K_{\psi,P}$	[–]	heading error proportional control gain
$K_{\psi,D}$	[–]	heading error derivative control gain

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$K_{\mathrm{d,P}}$	[–]	position error control gain
$s_{\mathrm{p,l}}$	[m]	preview distance for longitudinal speed control
$U_{\max}$	[m/s]	desired longitudinal speed of the vehicle
e_{U}	[–]	longitudinal speed control value
$s_{\mathrm{p,s}}$	[m]	preview distance for steering control
e_{ψ}	[–]	heading angle error control value
δ_{ψ}	[° deg]	steering angle due to the heading angle error
e_{d}	[–]	position deviation control value
δ_{d}	[° deg]	steering angle due to the position deviation
K_{rd}	[–]	gain scheduling parameter
δ	[° deg]	final output of the steering controller

1 Introduction

During the last years the automotive industry has extended the use of electronic control systems to improve the active safety of passenger and commercial vehicles. The research in the field of control systems includes active braking and/or torque distribution (ABS, ESP and AWD), active steering (2WS and 4WS), active suspensions, etc. The implementation of these systems in hardware is an expensive and time-consuming process. Therefore computer simulations can help to estimate the performance of these systems by simulating many different scenarios with minimal cost. An essential part of the simulation process is the use of a closed-loop driver model to control the vehicle. Detailed reviews of driver models that have been proposed in the literature are included in [2, 4].

Especially single or multi-point preview models are widely used in the literature, as well as in commercial vehicle simulation packages. These models preview the path ahead of the vehicle and calculate an error value (position, heading) that is used as input in the driver control system. To calculate the necessary control variable (steering angle – throttle/brake input) either a proportional controller with appropriate fixed gains [5] is used, or an optimal value is calculated by using an adaptive control model [1, 6, 8]. Adaptive control models use a simplified internal representation of the vehicle to predict the future trajectory of the vehicle and adjust the controller's response in order to achieve optimal driver control. However the performance of a controller with fixed gains or simple gain scheduling is more suitable for applications that need to simulate an average driver and not a professional driver. Also the use of this type of controller provides a driver behaviour that is consistent and can be tuned by changing a small set of parameters. This is more suitable for evaluation of different vehicle configurations, because in a comparison process, we need to keep as much of the system unchanged, in order to be able to correctly attribute the variation of the results.

So in this paper a modification and expansion of the model described in [5] is proposed, as well as a method to select the parameters needed for practical applications. We will use a simple PD controller with gain scheduling depending on a single state variable (longitudinal speed) to control the steering angle and we will also extend the model to control the longitudinal speed dynamically instead of using a pre-determined speed profile. Finally we evaluate the performance of the proposed driver model by performing a simulated double-lane change (ISO 3888-2) and simulating a lap around a racing track, using a full 14-DOF-vehicle model.

2 Driver model

The task of a human driver is to collect, through his senses, information about the speed, heading and position of the vehicle and the road ahead and adjust accordingly the steering wheel angle and the throttle/brake values, in order to match the heading of the vehicle with the heading of the road.

A typical driver when approaching a corner, first of all tries before actually reaching the corner to match the vehicle's longitudinal speed to the maximum longitudinal speed in which he believes the vehicle will safely negotiate the corner and for this purpose our driver model includes a longitudinal speed controller that is using a kinematics relation to calculate this speed, whereas the model in [5] uses a pre-determined speed profile.

Then, as he reaches the corner, he adjusts the steering wheel angle in order to match the vehicle's heading with the road heading, without deviating much from the road's centre line. The model described in [5] uses preview information of the position deviation of the vehicle's future path and the desired path and only compares the present heading angle of the vehicle to the heading angle of the road. The preview of the heading deviation is more related to the actual way a human is driving and it contains more information for the status of the vehicle compared to the position deviation. So, in our model we preview information for the deviation of the current heading angle of the vehicle compared to the tangent angle of the desired path, in addition to the deviation of the current and future position of the vehicle compared to the desired path, and use them as the input variables of our control algorithm. By using both the heading and the position deviation as control inputs we can select the optimum combination of control variables. It should be noted that during our trials the optimisation algorithm consistently selected values for the controller parameters that resulted to the use of the heading deviation as the primary control input (see Sect. 4.3).

2.1 Desired path description

The first step in implementing a driver model is to create a method for describing the desired path that the vehicle should follow. For this purpose a five-column array of information is used with data points spread in regular intervals along the path. The interval step is chosen to be as small as it's needed for the accurate description of the path. The first column is the distance covered along the path from what is considered the starting point and can also be used as a unique index for look-up purposes:

$$[s\,(\mathrm{m}) \quad k_\mathrm{t}\,(1/\mathrm{m}) \quad \psi_\mathrm{t}\,(\mathrm{rad}) \quad x_\mathrm{t}\,(\mathrm{m}) \quad y_\mathrm{t}\,(\mathrm{m})]\ .$$

The path curvature k_t, the tangent angle and the position coordinates x_t and y_t are the other four parameters of the array which provide to the driver model the necessary information about the path it should follow. It should be noted, that the array values can be easily calculated if the position coordinates $(x_\mathrm{t}, y_\mathrm{t})$ of the path are available, so it's easy to describe a track or a road from topographic plans or GPS data.

2.2 Path preview information

From our vehicle model it is easy to calculate the position of the vehicle on the x–y plane and its speed (longitudinal, lateral, yaw). In order to determine the relevant position on the desired path array the vehicle should have been, we use the current vehicle attributes, longitudinal and lateral speed (U, V), heading angle (ψ) and position (x, y) in Eq. 1 and Eq. 2, taken from [5], to calculate the distance covered along the desired path

$$s = \int \frac{\mathrm{d}s}{\mathrm{d}t} \tag{1}$$

$$\frac{\mathrm{d}s}{\mathrm{d}t} = \frac{U\cos(\psi-\psi_\mathrm{t}) - V\sin(\psi-\psi_\mathrm{t})}{1-k_\mathrm{t}\left[(y-y_\mathrm{t})\cos(\psi_\mathrm{t})-(x-x_\mathrm{t})\sin(\psi_\mathrm{t})\right]}\ . \tag{2}$$

In order to calculate the desired position $(x_\mathrm{t}, y_\mathrm{t})$, path curvature k_t and the tangent angle ψ_t of the path, we perform a linear interpolated table lookup on the data array of the desired path.

Because our driver model needs also preview information on the path ahead, we will calculate the desired set of variables at N intervals placed evenly along a preview distance s_p that is added to s. The matrix of the preview distances that are used as input in the lookup is described by

$$[s_\mathrm{r}]_{N\times 1} = s + \begin{bmatrix} 0 \\ \vdots \\ \left(\frac{n-1}{N-1}\right) s_\mathrm{p} \\ \vdots \\ s_\mathrm{p} \end{bmatrix}_{N\times 1}\ . \tag{3}$$

2.3 Longitudinal speed control

The first task of the driver model is to ensure that the vehicle's longitudinal speed is appropriate when entering a corner. To calculate this speed we use the relation between curvature, speed and lateral acceleration, given in Eq. 4, and we solve for speed, with a known lateral acceleration, as it is shown in Eq. 5

$$k_\mathrm{t} = \frac{a_y}{U^2} \tag{4}$$

$$U_\mathrm{max} = \left(\frac{a_{y,\mathrm{max}}\mu_y}{|k_\mathrm{t}|}\right)^{1/2}\ . \tag{5}$$

The maximum lateral acceleration $a_{y,\mathrm{max}}$ is a characteristic parameter of the vehicle and it shows how well it can utilize the maximum lateral forces acting on the tyres and μ_y is the lateral tyres-road friction coefficient. We have to note that the road is considered flat and aerodynamic lift or down-force is neglected.

In order for the driver model to have enough time to reduce the speed of the vehicle before reaching the corner, a suitable preview distance is selected

$$s_\mathrm{p,l} = \frac{U^2}{2a_{b,\mathrm{max}}\mu_x}\ . \tag{6}$$

The maximum braking deceleration $a_{b,\mathrm{max}}$ is another characteristic parameter of the vehicle and it shows its ability to utilize the maximum longitudinal forces acting on the tyres and μ_x is the longitudinal tyres-road friction coefficient. Using Eq. 3 we create the preview matrix $[s_\mathrm{r,l}]_{N\times 1}$, which we use to perform a lookup on the path table to retrieve the curvature values of the path along a distance $s_\mathrm{p,l}$ ahead of the vehicle and the maximum absolute value of path curvature ahead of the vehicle is calculated

$$k_\mathrm{t} = \max \begin{bmatrix} |k_{\mathrm{t},1}| \\ \vdots \\ |k_{\mathrm{t},N}| \end{bmatrix}_{N\times 1}\ . \tag{7}$$

This value is used in Eq. 5 to determine at each moment the desired longitudinal speed, which is compared with the longitudinal speed of the vehicle and multiplied with the longitudinal speed control gain K_U, to calculate the longitudinal speed control value e_U

$$e_\mathrm{U} = K_\mathrm{U}\,(U_\mathrm{max} - U)\,, \quad 1 \le e_\mathrm{U} \le 1\ . \tag{8}$$

If the value is positive, it corresponds to the position of the acceleration pedal and if it is negative to the position of the brake pedal, where a value of 1 or -1 corresponds to full throttle or brakes, respectively. The torque applied to the wheels is calculated by using a simplified representation of the engine/drivetrain and braking system.

2.4 Steering control

In order to implement the steering control, we will use two control variables, the heading angle error (the deviation between the heading of the car and the heading of the road) and the positional error (the deviation of the car's trajectory from the desired path trajectory).

2.4.1 Heading angle control

The heading angle error will be calculated by comparing the current vehicle-heading angle ψ with the tangent angle of the desired path ahead of the vehicle. The preview distance $s_{\mathrm{p,s}}$, will be calculated by the relation $s_{\mathrm{p,s}} = UT_{\mathrm{p}}$, where the preview time T_{p} is one of the control variables selected by the user.

As before, using Eq. 3, we create the preview matrix $[s_{\mathrm{r,s}}]_{N\times 1}$, which we use to perform a lookup on the path table and retrieve the heading angles of the path $[\psi_{\mathrm{t,i}}]_{n\times 1}$ at N points evenly distributed along a distance $s_{\mathrm{p,s}}$ ahead of the vehicle, as it's illustrated in Fig. 1.

Then we calculate the error values $[e_{psi,\mathrm{i}}]_{N\times 1} = [\psi_{\mathrm{t,i}}]_{N\times 1} - [\psi]_{N\times 1}$. This $N \times 1$ matrix is averaged per N/N_2 elements (N should be an integer multiple of N_2) to reduce its size and multiplied with a weighting matrix $[W_\psi]_{1\times N_2}$ to conclude to a single value that is representative of the path ahead of the vehicle and will be used as an input to our controller

$$\begin{bmatrix} e_{\psi,1} \\ \vdots \\ e_{\psi,N} \end{bmatrix}_{N\times 1} \rightarrow$$

$$\left[W_\psi\right]_{1\times N_2} \begin{bmatrix} \operatorname{avg}\left(e_{\psi,1}\dots e_{\psi,N/N_2}\right) \\ \vdots \\ \operatorname{avg}\left(e_{\psi,1+(n-1)N/N_2}\dots e_{\psi,n+N/N_2}\right) \end{bmatrix}_{N_2\times 1} = e_\psi . \tag{9}$$

After a series of trials we concluded that it is suitable to use a Proportional-Derivative controller for the heading error-steering control. The derivative term adds damping to the system and so it improves the stability of the driver model and reduces oscillatory behaviour, especially in emergency manoeuvres like the lane change test. This effect is described in Sect. 4.2 and is shown in Fig. 5. However the differentiation also amplifies noise in the error signal and can produce "jagged" steering behaviour, if large values of the derivative gain ($K_{\psi,\mathrm{D}}$) are used and so the selection of this parameter should be done carefully, depending on the application of the driver model. The use of an integral term was also considered, but it was found to have no considerable effect on the performance of the model, since the existence of a small steady-state error is not significant. The final expression of the steering angle due to the heading angle error is given by

$$\delta_\psi = K_{\psi,\mathrm{P}} e_\psi + K_{\psi,\mathrm{D}} \frac{\mathrm{d}e_\psi}{\mathrm{d}t} . \tag{10}$$

2.4.2 Position deviation control

The position error will be calculated by comparing the current and the future vehicle positions with the positions of the desired path ahead of the vehicle.

The preview distance is also in this case $s_{\mathrm{p,s}} = UT_{\mathrm{p}}$.

Similar, as in heading error control, we use Eq. 3 to create the preview matrix $[s_{\mathrm{r,s}}]_{N\times 1}$, which we then use to perform a lookup on the path table to retrieve the coordinates of the path $[x_{\mathrm{t,i}}]_{N\times 1}$ and $[y_{\mathrm{t,i}}]_{N\times 1}$ at N points evenly distributed along a distance $s_{\mathrm{p,s}}$ ahead of the vehicle.

Next, we evaluate the future position of the vehicle if it continues moving in its current heading with the same lon-

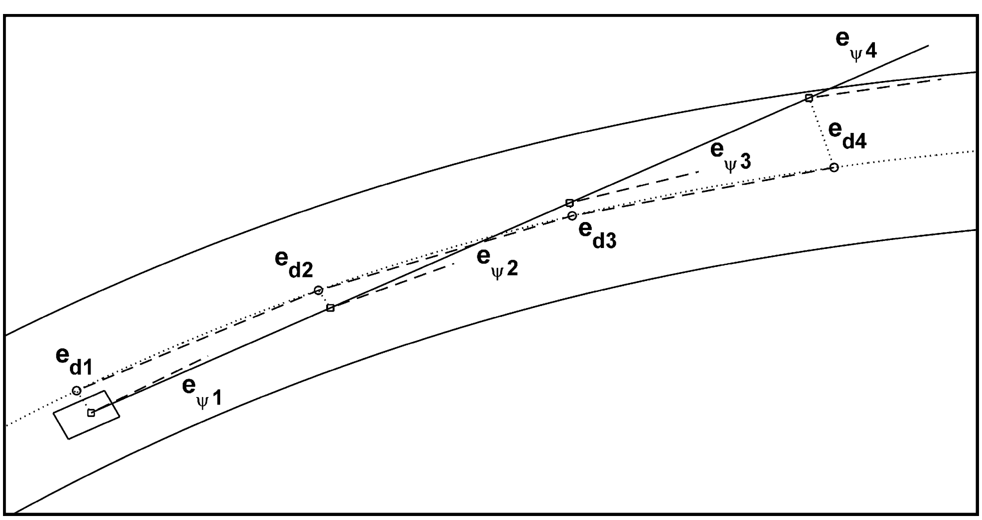

Fig. 1 Path and heading error preview

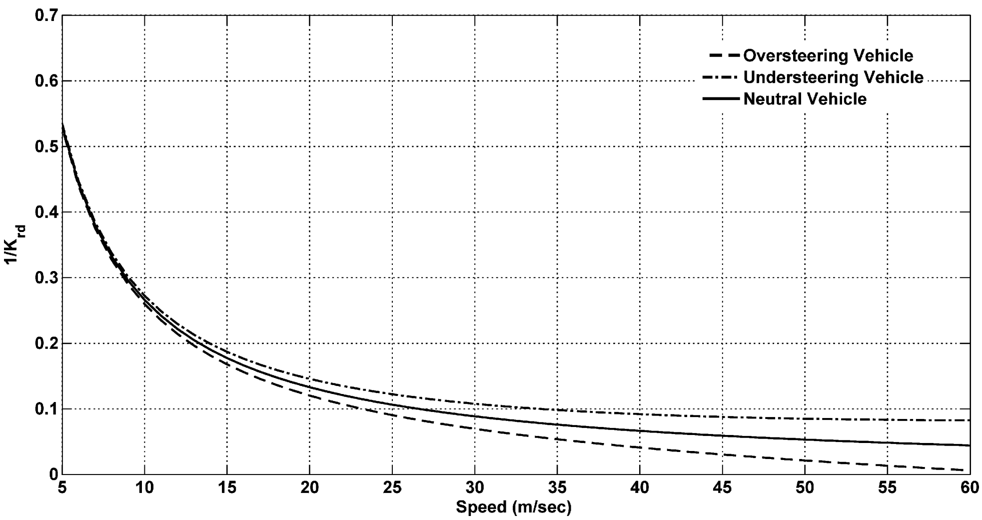

Fig. 2 Steering-velocity gain scheduling

gitudinal speed (see Fig. 1)

$$[x_\mathrm{i}]_{N\times 1} = \cos(\psi)[s_\mathrm{r,s}]_{N\times 1} + x \tag{11}$$

$$[y_\mathrm{i}]_{N\times 1} = \sin(\psi)[s_\mathrm{r,s}]_{N\times 1} + y\,. \tag{12}$$

Finally we calculate the position error $[e_\mathrm{d,i}]_{N\times 1}$.

$$[e_\mathrm{d,i}]_{N\times 1} = \left([y_\mathrm{t,i}]_{N\times 1} - [y_\mathrm{i}]_{N\times 1}\right)\cos(\psi) - \left([x_\mathrm{t,i}]_{N\times 1} - [x_\mathrm{i}]_{N\times 1}\right)\sin(\psi)\,. \tag{13}$$

Using the same procedure that was described in Eq. 9 the error matrix $[e_\mathrm{d,i}]_{N\times 1}$ is averaged and multiplied with a weighting matrix $[W_\mathrm{d}]_{1\times N_2}$ to be reduced to a single error e_d value that will be used in our controller. In this case a simple proportional controller was found to be adequate. The final expression of the steering angle due to the position error is given by

$$\delta_\mathrm{d} = K_\mathrm{d,P} e_\mathrm{d}\,. \tag{14}$$

2.4.3 Steering output – gain scheduling

Finally we need to introduce a form of gain scheduling to compensate for the vehicle's longitudinal speed. In general the average driver reduces the amount of steering as the speed of the vehicle increases. The gain scheduling should also take into account the understeer or oversteer characteristics of the vehicle. For this purpose we use the yaw rate/steering angle response, as it is calculated for a simplified linear 2-DOF vehicle model and is given in Eqs. 15 and 16

$$K_\mathrm{rd} = \frac{r}{\delta} = \frac{U}{l(1+KU^2)} \tag{15}$$

$$K = \frac{m}{l^2}\left(\frac{b}{C_1} - \frac{a}{C_2}\right)\,. \tag{16}$$

K is the stability factor, m the vehicle's mass, l the wheelbase length, a and b the distance of the front and rear axle from the centre of gravity and C_1 and C_2 the cornering stiffness of the front and rear tyres. The stability factor is calculated for the vehicle in a static state and we neglect the dynamic changes of the cornering stiffness of the tyres, caused by load transfer.

The final output of the steering controller is the sum of the two partial control outputs δ_ψ and δ_d, divided by K_rd.

$$\delta = \frac{1}{K_\mathrm{rd}}\left(\delta_\psi + \delta_\mathrm{d}\right)\,, \quad |\delta| \le 40\ \mathrm{deg}^\circ, \quad \left|\frac{\mathrm{d}\delta}{\mathrm{d}t}\right| \le 50\ \mathrm{deg}/\mathrm{s} \tag{17}$$

The influence of vehicle's longitudinal speed to the gain scheduling parameter is shown in Fig. 2 for a neutral, an understeering and an oversteering vehicle. The limitations of the steering angle and the steering angle rate of change are not a part of the driver model but a result of physical constraints of the simulated driver–vehicle system (maximum steering angle and speed of the steering system) and can be adjusted for different drivers and vehicles.

2.5 Parameter selection

The driver model is using the following three parameters for the longitudinal speed control and 6 parameters for the steering control.

Longitudinal speed control: $a_{b,\max}$, $a_{y,\max}$, K_U,
Steering control:
T_p, $[W_\psi]_{1\times N_2}$, $[W_\mathrm{d}]_{1\times N_2}$, $K_{\psi,\mathrm{P}}$, $K_{\psi,\mathrm{D}}$, $K_\mathrm{d,P}$.

These parameters determine the behaviour of our virtual driver and its performance. Depending on the vehicle and on the desired path, different values of the parameters will be needed to simulate the scenario we want to recreate. The analytical evaluation of the parameters, especially those affecting the steering control, is not possible due to the non-linearity of the vehicle model, especially at high lateral accelerations.

The parameters can be selected either intuitively or by using an optimisation algorithm to minimize a penalty function. Especially we can use a genetic stochastic algorithm, which has proven to be capable of optimising non-linear models without being influenced by local minima. The penalty function is selected depending on the desired behaviour of our driver model. For example, we can choose to minimise the time that is needed for the vehicle to complete a given path or the mean deviation from the desired path or a combination of these.

3 Vehicle model

3.1 Chassis

The vehicle chassis is represented by a rigid body with six degrees of freedom. The non-sprung masses are considered rigidly attached to the vehicle body, but the induced forces by the vertical displacement of the suspensions are calculated and taken into account. The geometry of the suspensions is not taken into consideration and all the torques and forces are considered to be applied along the vehicle-bound system of axes. The model is presented in detail in [7].

3.2 Wheels – tyres

The wheels are considered to have one degree of freedom (rolling around their lateral axis). The camber angle of the wheels is considered constant. The vertical force that is applied to each wheel is calculated by applying the vertical displacement of the point, where each wheel is attached to the vehicle body to a simple linear spring–dampener suspension. The steering angle, the driving and the braking torque of each wheel are considered a system input chosen by the driver or the control system and applied directly to the wheel. The dynamics of the steering system, the engine–transmission system and the brake system are not taken into consideration. The tyre lateral and longitudinal forces are calculated by using Pacejka's Magic Formula Model [3], for combined slip conditions, with longitudinal and lateral slip quantities evaluated for each wheel.

4 Model evaluation and numerical results

4.1 Test parameters

The vehicle parameters that will be used in the tests will simulate a medium-sized front-wheel-drive passenger vehicle (mass 1150 kg – wheelbase 2.56 m – 58%/42% front/rear weight ratio). The tyre parameters correspond to a 205/60 R15 tyre and can be found in [3]. The road–tyre friction coefficient will be 0.85, which is typical for dry asphalt. The available driving torque at the front wheels is 750 Nm and the available braking torque is 2400 Nm at the front wheels and 1600 Nm at the rear wheels.

We will test the driver model during a double-lane change (ISO 3888-2) and a lap around a racing track (Imola F1 track), as they are shown in Figs. 3 and 4.

4.2 Double-lane change (ISO 3888-2)

The ISO double lane change is a demanding manoeuvre, which tests the lateral acceleration limit of the vehicle under severe lateral weight transfer. The driver is required to utilize the full potential of the vehicle's friction limits to achieve the maximum entrance speed. Braking is not allowed during the manoeuvre, so the longitudinal speed controller is disabled.

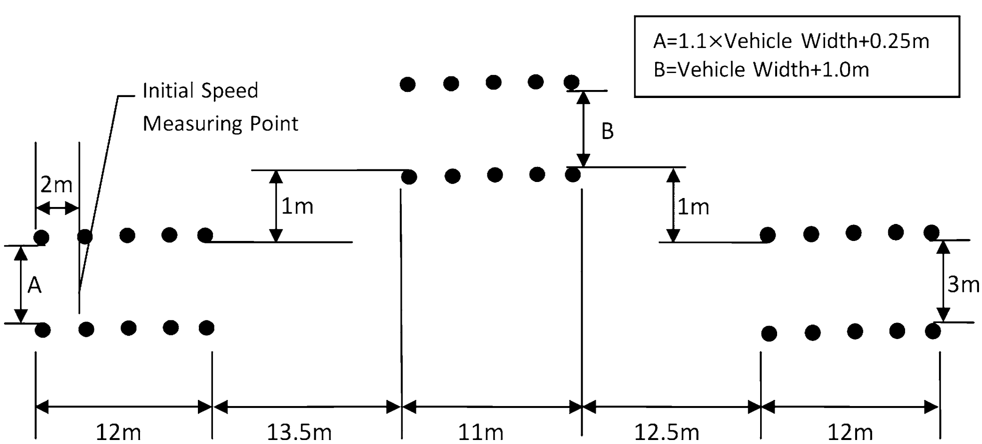

Fig. 3 ISO 3888-2 double lane change

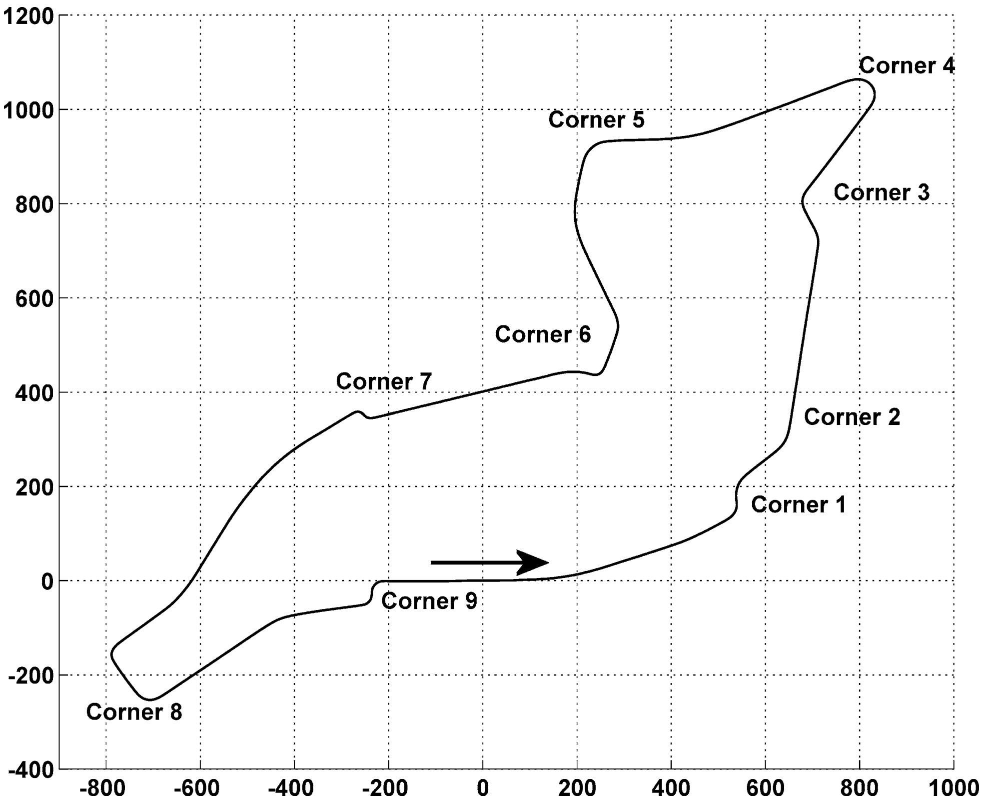

Fig. 4 Imola F1 track

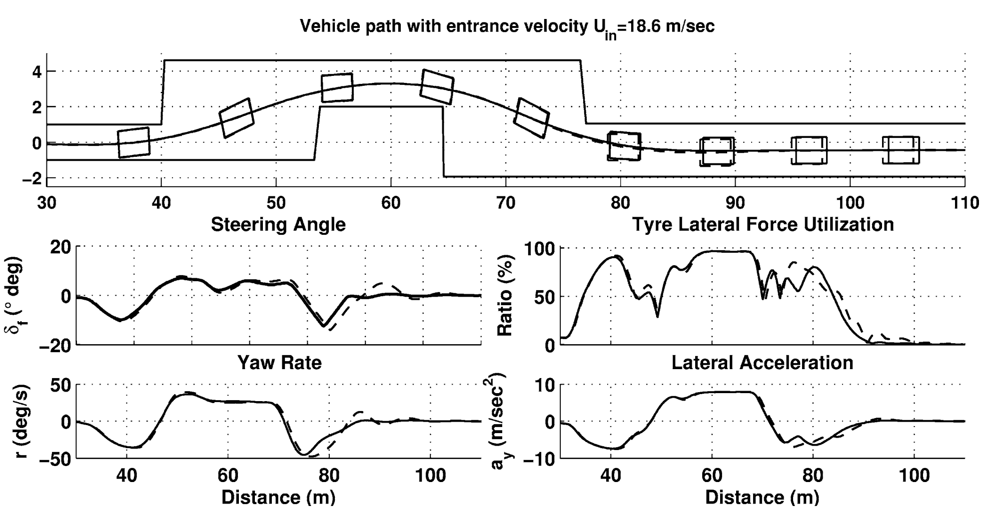

Fig. 5 ISO double lane change test

We optimised the driver parameters to achieve the maximum entrance speed for the lane change manoeuvre and the results are:

$$T_{\mathrm{p}} = 0.7\,\mathrm{s}$$
$$[W_{\psi}]_{1\times N_2} = [0.25 \quad 0.25 \quad 0.25 \quad 0.25]$$
$$[W_{\mathrm{d}}]_{1\times N_2} = [0.25 \quad 0.25 \quad 0.25 \quad 0.25]$$
$$K_{\psi,\mathrm{P}} = 4.92\,\mathrm{deg/deg}$$
$$K_{\psi,\mathrm{D}} = 0.087\,\mathrm{deg/(deg/s)} \quad \left(\text{and } K_{\psi,\mathrm{D}} = 0\,\mathrm{deg/(deg/s)}\right)$$
$$K_{\mathrm{d,P}} = 98\,\mathrm{deg/m}\,.$$

As we can see from the results in Fig. 5 the virtual driver successfully negotiates the course at an entrance speed of 18.6 m/s (67 km/h). The driver during the manoeuvre utilizes nearly the full potential of the vehicle's lateral acceleration limits, reaching a maximum lateral acceleration of 0.81 g.

A critical point of the lane change manoeuvre is the exit, where the rapid lateral load transfer destabilizes the vehicle and the driver should apply suitable steering corrections to avoid oscillatory motion or even a spin of the vehicle. To demonstrate the contribution of the derivative term of the controller, we simulated the manoeuvre also with the derivative gain ($K_{\psi,\mathrm{D}}$) set at 0 with the rest of the parameters unchanged, and the results are included in Fig. 5 (dashed line). We can see that the derivative gain is clearly reducing the oscillatory movement of the steering wheel at the exit of the lane change manoeuvre and stabilizes the behaviour of the vehicle.

4.3 Imola track

The racing track tests the driver's model ability in a variety of slow and fast bends. In this test, both the longitudinal and the lateral controller are used. In Fig. 6 we can see the effect

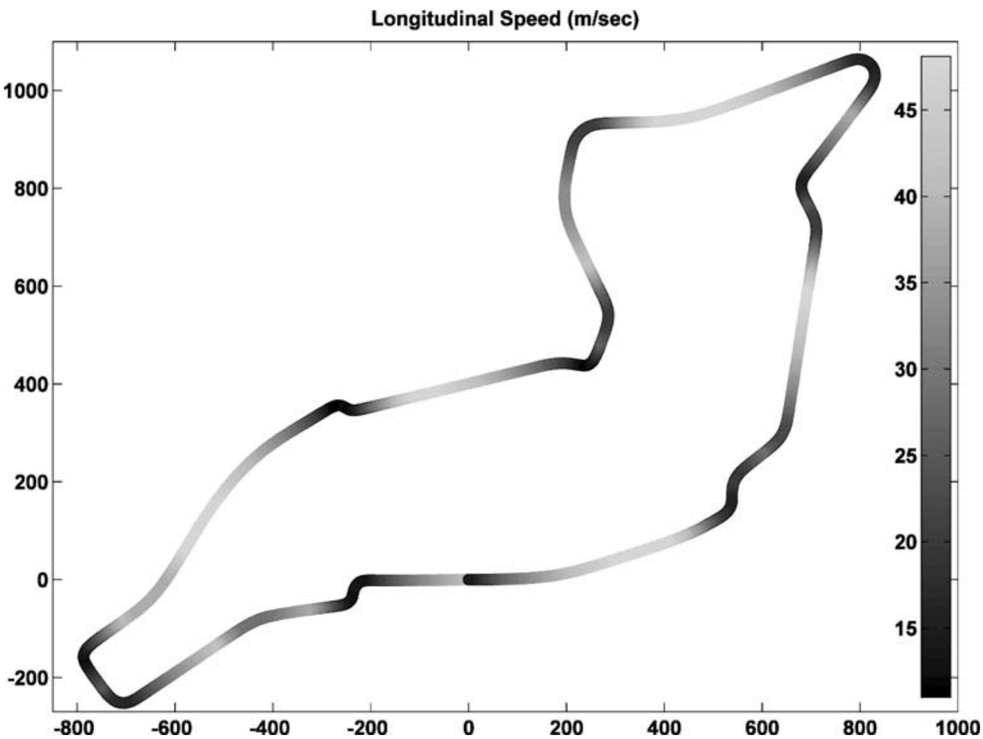

Fig. 6 Longitudinal speed

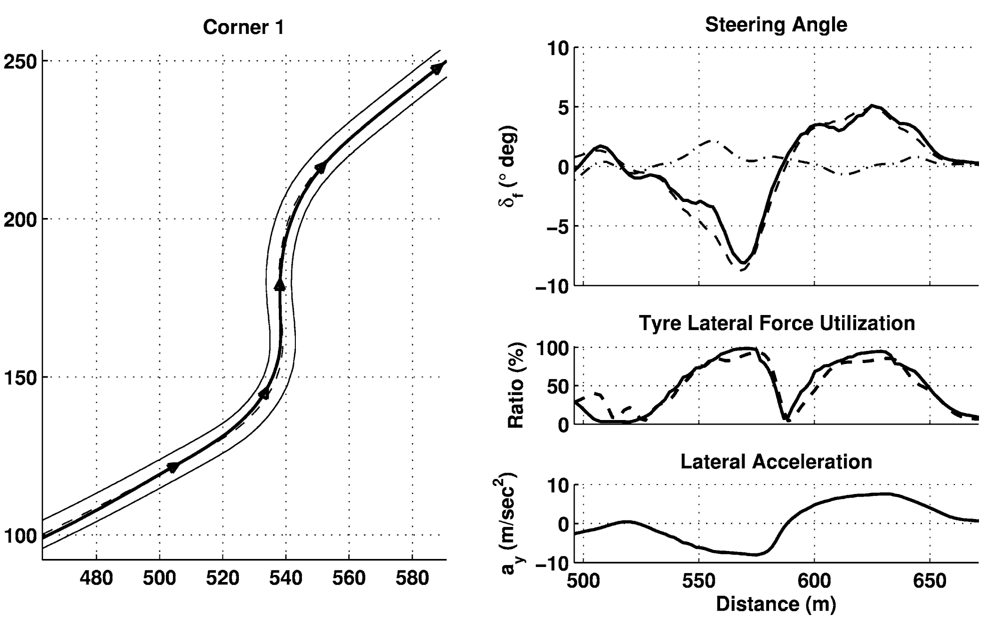

Fig. 7 Corner 1

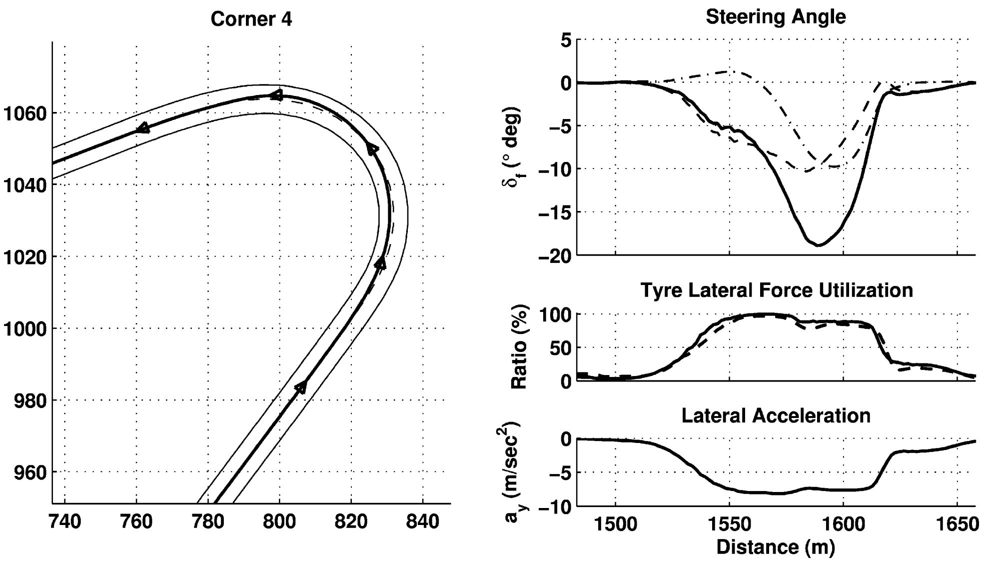

Fig. 8 Corner 4

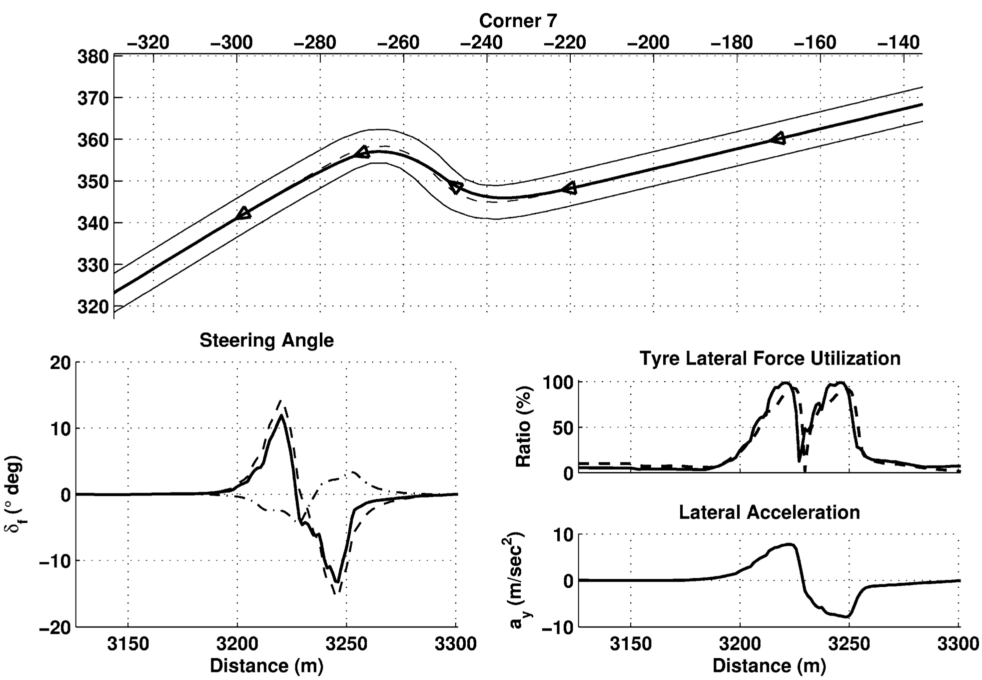

Fig. 9 Corner 7

of the longitudinal speed controller at the speed of the vehicle, according to the curvature of each corner. In Figs. 7–10 we can see the path and important state variables of the vehicle and tyres. In the steering angle graphs, in addition to the steering angle δ (solid line) we have included the two components of the steering control, δ_ψ (dashed line) and δ_d (dot-dashed line).

The parameters for the longitudinal speed controller are:

$$a_{b,\max} = a_{y,\max} = 11\ \mathrm{m/s^2}$$
$$K_\mathrm{U} = 0.3\,.$$

The steering control parameters were optimised to minimize the time needed to complete the track and to minimize the maximum and mean deviation from the desired path. The results are:

$$T_\mathrm{p} = 1.25\ \mathrm{s}$$
$$[W_\psi]_{1\times N_2} = [0.20 \quad 0.60 \quad 0.17 \quad 0.03]$$
$$[W_\mathrm{D}]_{1\times N_2} = [0.47 \quad 0.19 \quad 0.30 \quad 0.04]$$
$$K_{\psi,\mathrm{P}} = 3.97\ \mathrm{deg/deg}$$
$$K_{\psi,\mathrm{D}} = 0\ \mathrm{deg/(deg/s)}$$
$$K_{\mathrm{d,P}} = 27.36\ \mathrm{deg/m}\,.$$

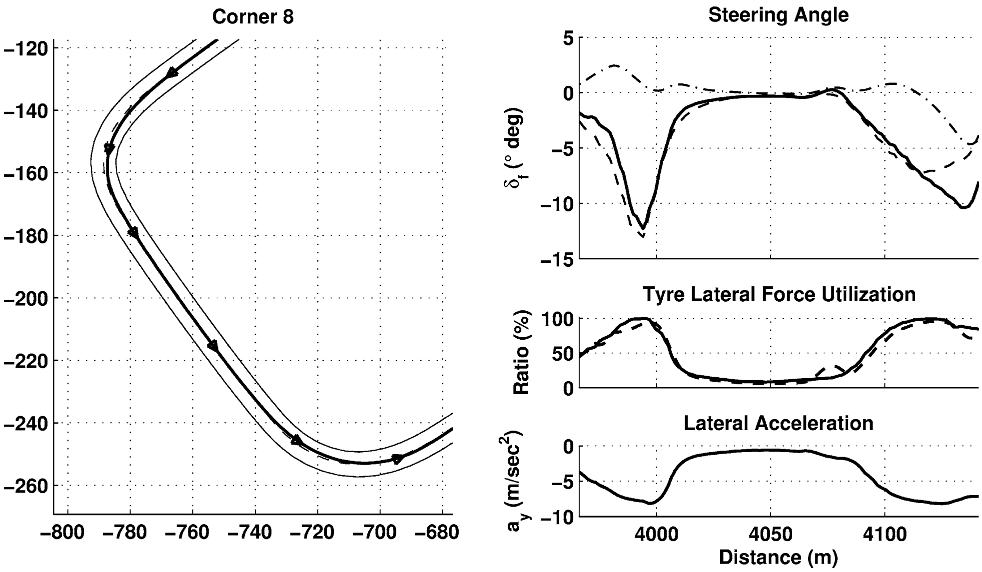

Fig. 10 Corner 8

From the steering angle graphs we can see that, using these optimised parameters for the steering controller, the δ_ψ (heading deviation control) is the primary value that controls the lateral motion of the vehicle with δ_{d} (position deviation control) only contributing when the vehicle is deviating from the desired path. This is an indication that for the optimisation goals, that we have selected, the heading deviation control offers improved performance compared to the position deviation control used in [5].

The virtual driver completes the track in 170.8 s, with an average speed of 28.73 m/s (103.4 km/h). The maximum longitudinal acceleration is measured at 1.04 g (during braking before entering Corner 3) and the maximum lateral acceleration at 0.84 g (in Corner 4). The driver model is capable to follow the desired path at high speeds by utilizing nearly the full potential of the tyres, showing the characteristics that we expected from a driver model. It reduces the vehicle's speed before entering the corner and accelerates at the exit of the corner. Also at corners with large curvature it doesn't follow exactly the path but is "cutting" the corner in order to minimise the curvature of the vehicle's path, as we can see in Fig. 9. The amount of "corner cutting" is determined by the combination of the preview time and the gains of the steering controller.

5 Conclusion

In Vehicle Dynamics simulations and in active safety systems evaluations, we need a realistic driver's model. The model proposed in this paper can follow a given path at high accelerations, near the vehicle's limits, by adjusting the longitudinal speed through accelerator/brakes and the heading of the vehicle through the steering wheel. Compared to the model described in [5] we added a dynamic preview longitudinal speed controller instead of a predetermined speed profile and we modified the lateral part of the driver model to include preview information about the heading of the vehicle compared to the road and included a derivative part in the controller, that helps to stabilize the behaviour of the driver model in specific manoeuvres like the lane change test. This driver model is easy to program and include in computer simulations and it is not computing intensive. Also it can be tuned to simulate different driver behaviours, so it can be used to test different aspects of vehicle's performance. It was used successfully in different test cases (ISO lane change, Imola F1 track) and the results are satisfactory.

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