Driver assistance system design A

Vehicle Stability Control

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Outline

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2. DST model with yaw moment
3. Vehicle Stability Control
   - Vehicle Stability Control basic
   - Vehicle Stability Control with reference
   - Vehicle Stability Control with observer
4. Discussion

1 Introduction

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Introduction

- The goal of stability control is to prevent vehicles from spinning, drifting and running off the road.
- Motivations:
  - High tire-road friction coefficient: the driver's steering action provides the lateral force required by the vehicle to track a curved road.
  - Small friction coefficient: the vehicle does not follow the nominal motion expected by the driver – it instead travels on a trajectory of larger radius and the vehicle may spin.
  - The common driver is not able to recognize a friction coefficient change and may react in a wrong way, causing an accident.
  - Stability control aims to bring the yaw rate and slip angle of the vehicle close to the nominal one expected by the driver.
- A variety of names are used by carmakers: ESC (electronic stability control), YC (yaw control), DYC (direct yaw control), VSA (vehicle stability assist), VDC (vehicle dynamics control), VSC (vehicle stability control), ESP (electronic stability program).

Introduction

- Three main classes of stability control systems have been proposed:
  - **Differential Braking systems**. They use the ABS brake system to apply differential braking between the right and left wheels to control the yaw motion.
  - **Steer-by-Wire systems**. They correct the driver's steering angle by adding a suitable steering angle, increasing the yaw stability properties.
  - **Active Torque Distribution systems**. They use active differentials and all wheel drive technologies to control the torque distributed to each wheel, thus providing a suitable control action for both traction and yaw motion.
- In the following, we will focus on differential braking systems since they have been implemented on several production vehicles.
- A modified DST model is now introduced, accounting for the moment given by the braking system. The model is also written in a form suitable for controller design.

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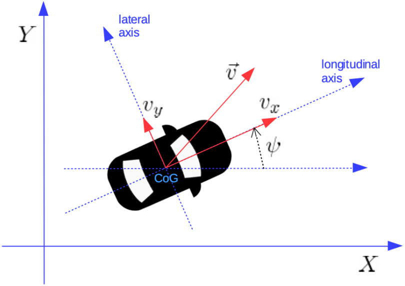
DST model with yaw moment

- We recall the dynamic single-track (DST) model.



- Vehicle variables:

  X, Y: coordinates of the vehicle CoG in an inertial reference frame

  ψ: yaw angle

  $\omega_\psi \doteq \dot{\psi}$: yaw rate

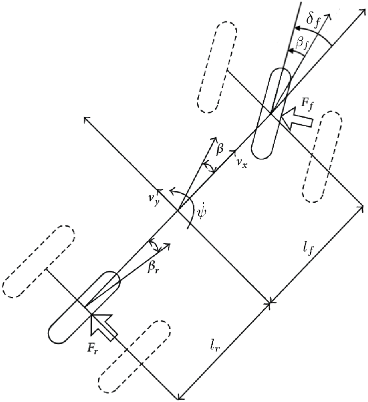
  $\vec{v} \equiv V$: velocity vector in the inertial frame

  v_x: longitudinal speed $=$ $\vec{v}$ component along the longitudinal axis

  v_y: lateral speed $=$ $\vec{v}$ component along the transverse axis

  a_x: longitudinal acceleration.

DST model with yaw moment



- Vehicle variables:
  δ_f: steering angle
  β: vehicle slip angle = angle between the vehicle longitudinal axis and velocity vector.
  β_f, β_r: tire slip angles = angles between the tire longitudinal axis and velocity.
- Vehicle parameters:
  CoG: center of gravity
  m, J: mass and moment of inertia
  l_f: distance CoG - front wheel center
  l_r: distance CoG - rear wheel center
  c_f, c_r: front/rear cornering stiffnesses (multiplied by load factors).

DST model with yaw moment

- The state equations of the DST model are

$$
\begin{aligned}
\dot{X} &= v_x \cos\psi - v_y \sin\psi \\
\dot{Y} &= v_x \sin\psi + v_y \cos\psi \\
\dot{\psi} &= \omega_\psi \\
\dot{v}_x &= v_y \omega_\psi + a_x \\
\dot{v}_y &= -v_x \omega_\psi + \frac{2}{m}\left(F_{yf} + F_{yr}\right) \\
\dot{\omega}_\psi &= \frac{2}{J}\left(l_f F_{yf} - l_r F_{yr}\right) + \frac{1}{J} M_b
\end{aligned}
$$

- M_b is a torque finalized at controlling the yaw motion. This torque can be actuated by the braking system.
- A linear model (for $v_x = const$) is considered for the tires:

$$F_{yf} = -c_f \beta_f, \quad F_{yr} = -c_r \beta_r, \quad \beta_f = \frac{v_y + l_f \omega_\psi}{v_x} - \delta_f, \quad \beta_r = \frac{v_y - l_r \omega_\psi}{v_x}.$$

DST model with yaw moment

- We consider the last two equations (using the linear tire model):

$$\dot{v}_y = -v_x\omega_\psi + \frac{2}{m}\left(-c_f\left(\frac{v_y + l_f\omega_\psi}{v_x} - \delta_f\right) - c_r\frac{v_y - l_r\omega_\psi}{v_x}\right)$$
$$\dot{\omega}_\psi = \frac{2}{J}\left(-l_f c_f\left(\frac{v_y + l_f\omega_\psi}{v_x} - \delta_f\right) + l_r c_r\frac{v_y - l_r\omega_\psi}{v_x}\right) + \frac{M_b}{J}.$$

- After simple manipulations:

$$\dot{v}_y = -\frac{2}{m}\frac{c_f + c_r}{v_x}v_y - \left(v_x + \frac{2}{m}\frac{l_f c_f - l_r c_r}{v_x}\right)\omega_\psi + \frac{2c_f}{m}\delta_f$$
$$\dot{\omega}_\psi = -\frac{2}{J}\frac{l_f c_f - l_r c_r}{v_x}v_y - \frac{2}{J}\frac{l_f^2 c_f + l_r^2 c_r}{v_x}\omega_\psi + \frac{2l_f c_f}{J}\delta_f + \frac{M_b}{J}.$$

DST model with yaw moment

- We consider the vehicle *slip (sideslip) angle*: $\beta \doteq \operatorname{atan}\frac{v_y}{v_x}$.
- β is the angle between the vehicle longitudinal axis and velocity vector. It is important for yaw/lateral stability:
  - small β $\rightarrow$ stable behavior
  - large β $\rightarrow$ unstable behavior.
- For small angles, $\beta \cong \frac{v_y}{v_x}$, and the DST equations can be written as

$$\dot{\beta} = -\frac{2(c_f + c_r)}{m v_x}\beta - \left(1 + \frac{2(l_f c_f - l_r c_r)}{m v_x^2}\right)\omega_\psi + \frac{2c_f}{m v_x}\delta_f$$

$$\dot{\omega}_\psi = -\frac{2(l_f c_f - l_r c_r)}{J}\beta - \frac{2(l_f^2 c_f + l_r^2 c_r)}{J v_x}\omega_\psi + \frac{2 l_f c_f}{J}\delta_f + \frac{M_b}{J}.$$

This model is called DSTβ.

DST model with yaw moment

- The DSTβ model can be written in matrix form as follows:

$$\dot{\eta} = A_\eta \eta + B_\alpha \alpha_b + B_s \delta_f$$

$$A_\eta \doteq \begin{bmatrix} -\frac{2(c_f+c_r)}{mv_x} & -1-\frac{2(l_f c_f - l_r c_r)}{mv_x^2} \\ -\frac{2(l_f c_f - l_r c_r)}{J} & -\frac{2(l_f^2 c_f + l_r^2 c_r)}{J v_x} \end{bmatrix}$$

$$B_s \doteq \begin{bmatrix} \frac{2c_f}{mv_x} \\ \frac{2l_f c_f}{J} \end{bmatrix}, \quad B_\alpha \doteq \begin{bmatrix} 0 \\ 1 \end{bmatrix}$$

  - state: $\eta \doteq (\beta, \omega_\psi)$
  - command input: $\alpha_b \doteq \frac{M_b}{J}$
  - disturbance: δ_f (steering angle).
- Using α_b instead of M_b is convenient for numerical conditioning reasons. The braking system provides a torque $M_b = J\alpha_b$.
- For a constant v_x, it can be easily verified that the model is asymptotically stable.

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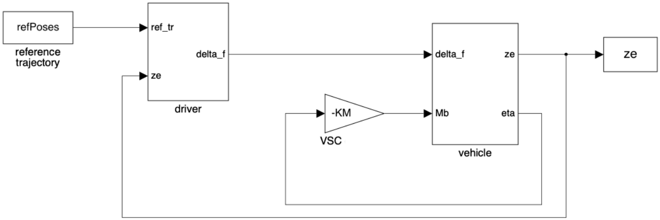
VSC basic

- The DSTβ model is suitable to design a controller, finalized at improving the yaw/lateral stability properties of the vehicle.
- The yaw acceleration α_b is actuated by the braking system, that provides a moment $M_b = J\alpha_b$.
- VSC systems include a dispatching algorithm (not studied here), that provides the braking actions on the four wheels needed to apply the control moment M_b.
- The desired yaw rate and slip angle cannot always be obtained. It may be not safe trying to impose given yaw rate and slip angle if the friction coefficient of the road is unable to provide the required tire forces.
  - To overcome this problem, the reference yaw rate can be bounded by a function of the tire-road friction coefficient (not studied here).

VSC basic

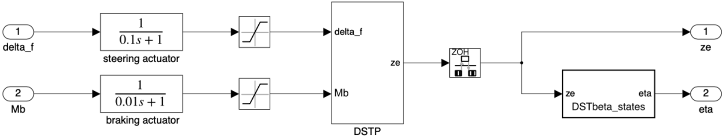
Closed-loop scheme



- Vehicle: DSTP model with input (δ_f, M_b) and actuators.
- Driver model.
- Reference trajectory: set of points giving the reference trajectory.
- VSC controller.

VSC basic

Vehicle model



- DSTP: dynamic single-track vehicle model with Pacejka's tire formula and input (δ_f, M_b).
  - Vehicle parameters: $l_f = 1.2\,\mathrm{m}$, $l_r = 1.6\,\mathrm{m}$, $m = 1575\,\mathrm{kg}$, $J = 4000\,\mathrm{kg\,m^2}$, $c_f = 27e3\,\mathrm{N/rad}$, $c_r = 20e3\,\mathrm{N/rad}$.
  - Tire parameters: $p_1 = \xi\, 3863\,\mathrm{N}$, $\xi \in [0,1]$, $p_2 = 1.5$, $p_4 = -0.5$, $p_3 = c_f/p_1/p_2$ (front tire) or $p_3 = c_r/p_1/p_2$ (rear tire).
  - A constant speed v_x is directly imposed in "DSTP vehicle model".
- Other blocks:
  - actuators
  - saturations ($\delta_f \in [-0.6, 0.6]\,\mathrm{rad}$, $M_b \in [-6.5, 6.5] \cdot 10^4\,\mathrm{Nm}$)
  - data sampling block (sampling time $0.05\,\mathrm{s}$)
  - block for computation of $\eta = (\beta, \omega_\psi)$.

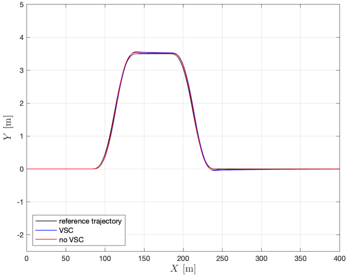
VSC basic

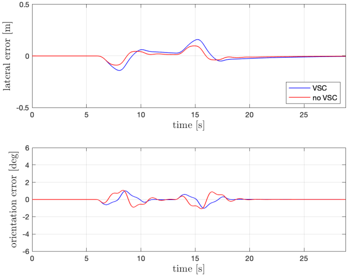
Controllers

- Driver model:
  - Any driver is a “biological” feedback controller.
  - A driver is similar to a PID controller:
    - integral $\leftrightarrow$ past
    - proportional $\leftrightarrow$ present
    - derivative $\leftrightarrow$ future.
  - Here, one of the PIDF controllers previously designed was used as the driver model (including calculation of ref. closest point and errors).
- Vehicle stability controller: $K_M = JK_\alpha$, where K_α is a LQR controller designed from the DSTβ model ($\xi = 1$), assuming $v_x = 110\,\mathrm{km/h}$, and $Q = \mathrm{diag}(100, 10)$ and $R = 1$.

VSC basic

Simulation results. Double lane change, $v_x = 50\,\mathrm{km/h}$, dry road ($\xi = 1$).

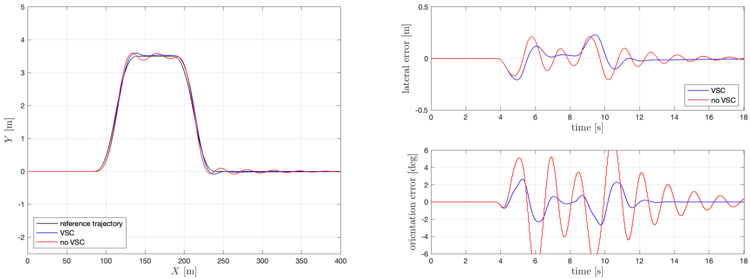




- Stable behavior with and without VSC.

VSC basic

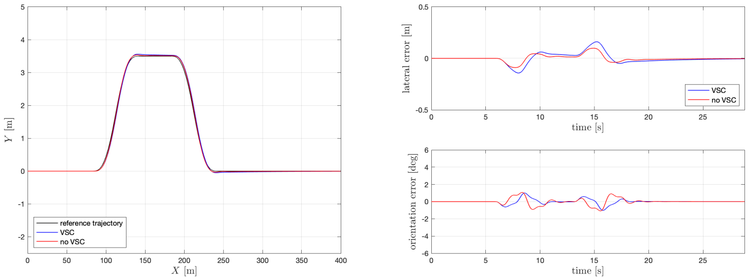
Simulation results. Double lane change, $v_x = 80\,\mathrm{km/h}$, dry road ($\xi = 1$).



- Stable behavior with and without VSC.
- Oscillations without VSC.

VSC basic

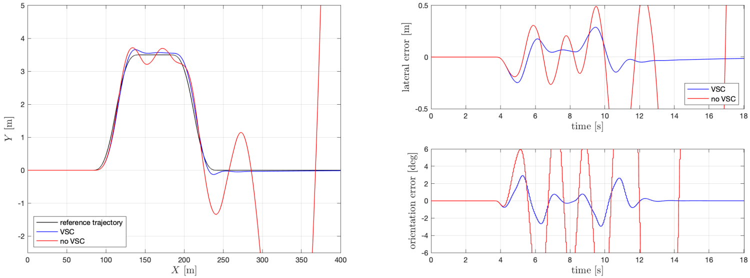
Simulation results. Double lane change, $v_x = 50\,\mathrm{km/h}$, wet road ($\xi = 0.6$).



- Stable behavior with and without VSC.

VSC basic

Simulation results. Double lane change, $v_x = 80\,\mathrm{km/h}$, wet road ($\xi = 0.6$).



- Stable behavior with VSC, unstable without VSC.

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VSC with reference

- The VSC developed above can be improved by using a suitable reference value for $\eta \doteq (\beta, \omega_\psi)$.
- Consider an ideal situation where the vehicle dynamics is LTI, there are no uncertainties, and VSC is not required.
- Suppose that a constant δ_f is imposed to obtain a desired curvature of the vehicle trajectory. In steady-state conditions:

$$0 = A_\eta \eta + B_s \delta_f.$$

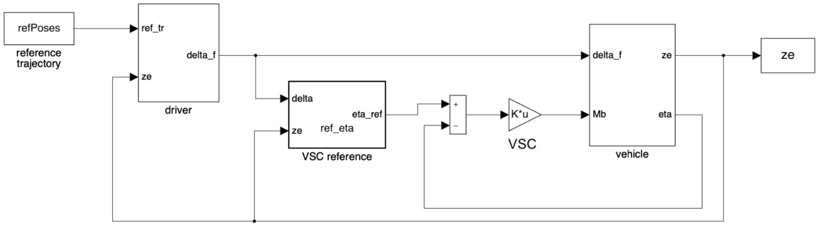
- It follows that

$$\eta_r = -A_\eta^{-1} B_s \delta_f$$

  is the value of η corresponding to the desired curvature in ideal conditions.
- Therefore, η_r can be used as a reference for VSC. It can be noted that it uses the information about the steering angle.

VSC with reference

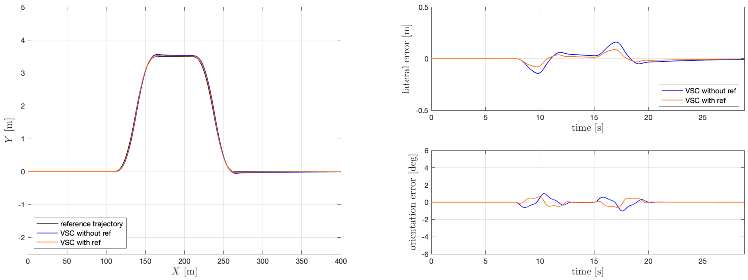
Closed-loop scheme



- vehicle: DSTP model with input (δ_f, M_b) and actuators.
- driver model.
- reference trajectory: set of points giving the reference trajectory.
- VSC reference η_r.
- VSC controller.

VSC with reference

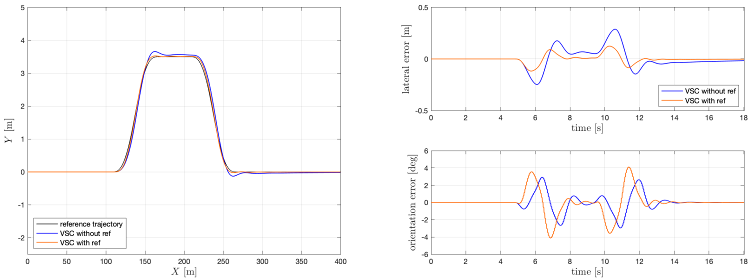
Simulation results. Double lane change, $v_x = 50\,\mathrm{km/h}$, wet road ($\xi = 0.6$).



- Satisfactory behavior in both cases.

VSC with reference

Simulation results. Double lane change, $v_x = 80\,\mathrm{km/h}$, wet road ($\xi = 0.6$).



- VSC with reference more precise in terms of lateral error.

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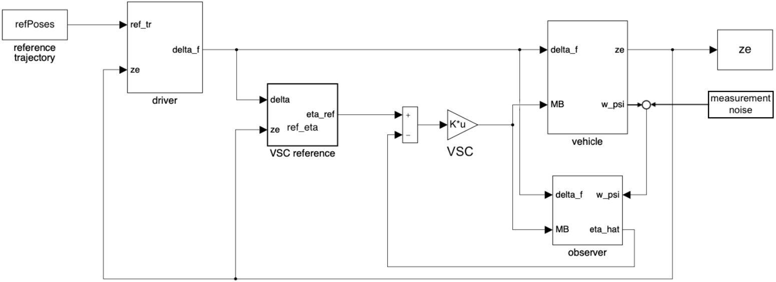
VSC with observer

- The presented VSC system requires the knowledge of the yaw rate $\dot{\psi} = \omega_\psi$ and the slip angle β.
  - The yaw rate can be measured by effective and cheap sensors.
  - Slip angle sensors are very expensive.
- The slip angle physical sensor can be replaced by a proper observer/filter which provides an estimate of this angle. The observer/filter is a so-called virtual sensor.
- Situations in which virtual sensors can be useful:
  - expensive physical sensor
  - redundancy
  - lack of space.
- An observer was designed from the DSTβ model. Observer eigenvalues: $\{1.5\sigma, 2\sigma\}$, where σ is the real part of the (complex conjugate) eigenvalues of $A_\eta - B_\alpha K_\alpha$.

VSC with observer

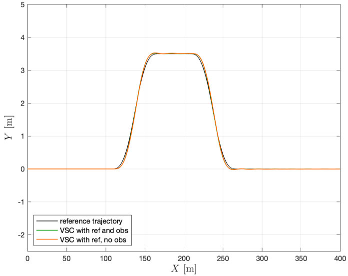
Closed-loop scheme

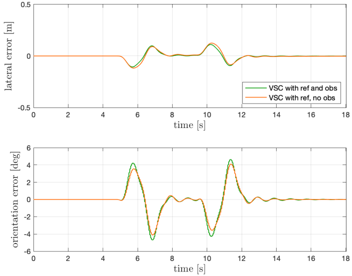


- Vehicle: DSTP model with input (δ_f, M_b) and actuators.
- Driver model.
- Reference trajectory: set of points giving the reference trajectory.
- VSC reference η_r.
- VSC controller.
- Observer (the M_b input must be divided by J)
- Measurement noise (Gaussian with mean=0, std=0.005).

VSC with observer

Simulation results. Double lane change, $v_x = 80\,\mathrm{km/h}$, wet road ($\xi = 0.6$).





- Similar behaviors without and with observer.

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4 Discussion

Discussion

- Problem 1: Linear controller and observer were used.
  - They work well for small values of the slip angle β.
  - When β becomes large, the vehicle dynamics becomes highly nonlinear and linear controllers/observers may not be adequate.
- Problem 2: It is in general difficult to estimate the parameters of the interaction force between tire and road (Pacejka's formula and/or c_f, c_r parameters).
  - These parameters are used to design the controller and observer.
  - If the parameter values are too wrong, the controllers/observers may not be adequate.
- These problems may be overcome by using machine learning techniques to design the controller and observer.
  - Possible to deal with nonlinear and uncertain dynamics.
  - Need of large amounts of experimental data.