

DRIVER MODEL

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INTRODUCTION

- ▶ Recent developments in the automotive industry have led to increasingly complex vehicle subsystems incorporating a large number of electronic control systems.
- ▶ In particular the electronic safety systems, such as the antilock braking system, electronic stability program or active front steering (AFS), influence the vehicle's behaviour by actively intervening in the driver's input .
- ▶ The aim of these systems is to counteract unstable situations such as excessive side slip, yaw motion or wheel spin
- ▶ This is achieved by controlling engine torque, independently controlling the brake pressure at each wheel, or applying a steering angle which is in addition to the driver input.
- ▶ In this presentation we are going to deal with closed-loop driver model with steering angle and accelerator/braking as driver inputs
- ▶ The anti-lock braking system (ABS), traction control system (TCS), four wheel steering system (4WS) and active yaw moment control system (AYC) are the components of the active chassis system.

CAD model





DRIVER MODEL INTRODUCTION

- ▶ Driver model may be considered as a controller which receives input from the vehicle and environment and passes a few control signals to the vehicle
- ▶ There are two important inputs from the driver
 - ▶ Steering angle, δ
 - ▶ Accelerator/brake
- ▶ In literature the driver model employs various techniques such as reference vector fields, PID controllers for modelling the human behaviour.
- ▶ In this presentation we use reference vector fields for modelling.
- ▶ A controller has been designed by linear matrix inequalities (LMIs) based on a three-degree-of freedom (3-DOF) reduced vehicle model.

DRIVER MODEL CONTD.....

- ▶ The closed loop driver model can be divided into 3 parts
 - ▶ Vehicle model
 - ▶ Tire model
 - ▶ Driver model
- ▶ In vehicle model a 8-DOF vehicle model is considered for both lateral and longitudinal dynamics.
- ▶ Modelling of the tire force plays an important role in determining the vehicle dynamic behaviour.
- ▶ As was told above the driver model implemented here is based on the reference vector field model

Vehicle Model

- ▶ The vehicle model used in this study is shown in Fig. 1. It is an 8-DOF vehicle model for both lateral and longitudinal dynamics, which also takes into account the nonlinearities between vehicle body dynamics and tire forces
- ▶ Degrees of freedom associated with the model are the longitudinal and lateral speeds, yaw rate, roll rate and four wheel rotational speeds.
- ▶ The dynamics equations governing the motion are derived as:

Longitudinal motion:

$$m(\dot{u} - \gamma v) + m_s h_s \dot{\gamma} \varphi = F_{xfl} + F_{xrl} + F_{xrr}$$

Lateral motion:

$$m(\dot{v} + \gamma u) - m_s h_s \dot{\gamma} \varphi = F_{yfl} + F_{yrl} + F_{yrr}$$

Yaw motion:

$$I_{zz}\dot{\gamma} - I_{xz}\ddot{\phi} = L_f(F_{yfl} + F_{yfr}) - L_r(F_{yrl} + F_{yrr}) - \frac{T_f}{2}(F_{xfl} - F_{xfr}) - \frac{T_r}{2}(F_{xrl} - F_{xrr})$$

Roll motion:

$$I_{xx}\ddot{\phi} - I_{xz}\dot{\gamma} = m_s h_s (\dot{v} + u\gamma) + m_s h_s g\varphi - (K_{\phi f} + K_{\phi r})\phi - (D_{\phi f} + D_{\phi r})\dot{\phi}$$

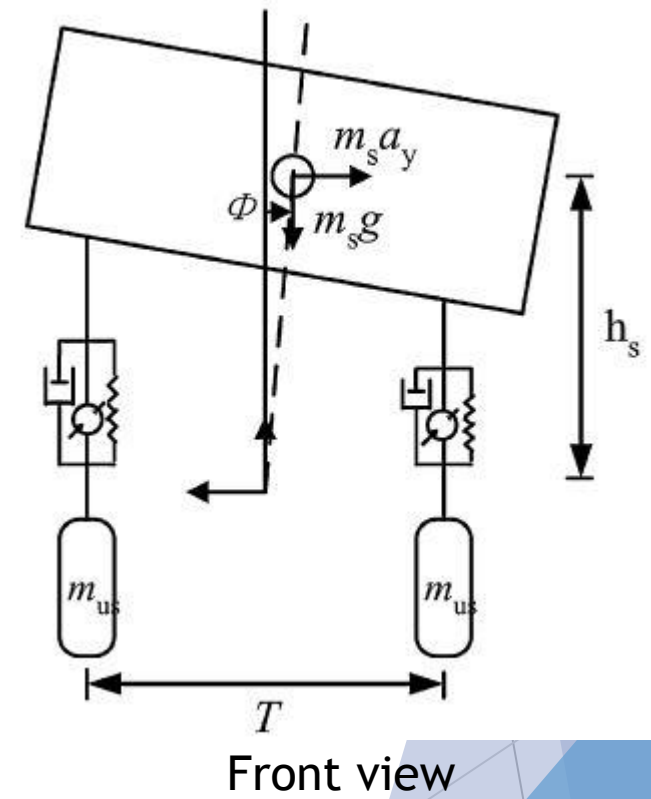
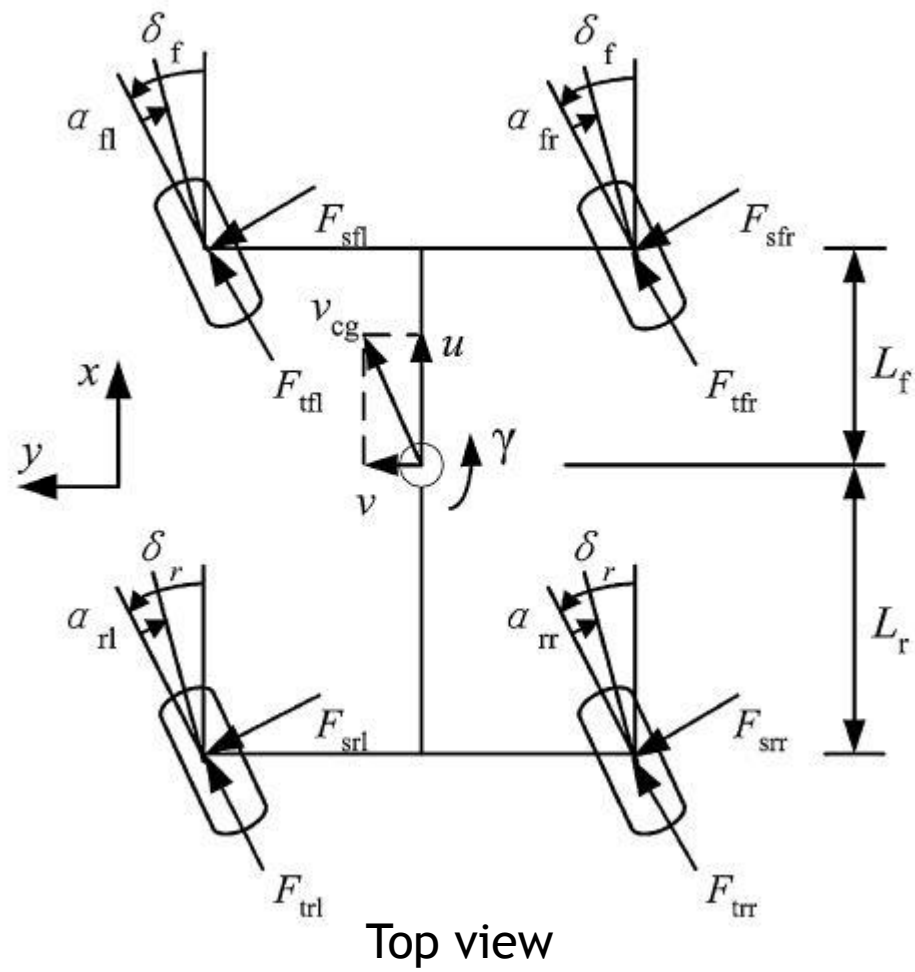


Fig. 1. Parameter definitions for the vehicle model.

Wheel rotation motion:

$$I_{\omega} \dot{\omega}_{ij} = -R_{\omega} F_{xij} + T_{eij} - T_{bij}$$

- ▶ where u , v , γ and φ are the longitudinal speed, lateral speed, yaw rate and vehicle roll angle. $\dot{\omega}_{ij}$, T_{eij} and T_{bij} represent the wheel angular acceleration, driving and braking torques.
- ▶ The subscripts $i = f, r$ and $j = l, r$ represent the corresponding tire. For example, $\dot{\omega}_{fl}$ stands for wheel angular acceleration on the front left wheel.

- F_{xij} and F_{yij} denote the tire forces in the x and y direction, which can be related to the longitudinal tractive force F_{tij} and the lateral tire force F_{sij} :

$$F_{xij} = F_{tij} \cos \delta_{ij} - F_{sij} \sin \delta_{ij}$$

$$F_{yij} = F_{tij} \sin \delta_{ij} + F_{sij} \cos \delta_{ij}$$

Where δ_f and δ_r are the front and rear wheel steering angles, and $\delta f_l = \delta f_r = \delta f$, $\delta r_l = \delta r_r = \delta r$.

- Defining the earth coordinates for longitudinal displacement X_G and lateral deviation Y_G , the vehicle velocity in earth coordinates is given as:

$$\dot{X}_G = u \cos \psi - v \sin \psi$$

$$\dot{Y}_G = u \sin \psi + v \cos \psi$$

Where ψ is the vehicle yaw angle.

Tire Model

- ▶ We use semi empirical model to study tires.
- ▶ The general form of the formula reads:

$$F_t = D_t \sin\{C_t \arctan\{B_t\lambda - E_t[B_t\lambda - \arctan(B_t\lambda)]\}\}$$

$$F_s = D_s \sin\{C_s \arctan\{B_s\alpha - E_s[B_s\alpha - \arctan(B_s\alpha)]\}\}$$

- ▶ F_t and F_s are tire tractive and lateral forces, λ and α are tire longitudinal wheel slip ratio and lateral wheel slip **angle**
- ▶ Self aligning torque, $M_z = F_s \sigma_a$, Where σ_a is pneumatic trail and is given by,

$$\sigma_\alpha = D_\alpha \cos\{C_\alpha \arctan\{B_\alpha\alpha - E_\alpha[B_\alpha\alpha - \arctan(B_\alpha\alpha)]\}\}$$

- ▶ The coefficients in the formula depend on the type of tire and road conditions.
- ▶ Transient behavior is taken into account in the formula.
- ▶ Vehicles net load on each tire is:

$$F_{zfl} = \frac{mgL_r}{2L} - ma_x \frac{h}{2L} - \frac{ma_y L_r h}{LT_f} - \frac{(K_{\phi f} \phi + D_{\phi f} \dot{\phi})}{T_f}$$

$$F_{zfr} = \frac{mgL_r}{2L} - ma_x \frac{h}{2L} + \frac{ma_y L_r h}{LT_f} + \frac{(K_{\phi f} \phi + D_{\phi f} \dot{\phi})}{T_f}$$

$$F_{zrl} = \frac{mgL_f}{2L} + ma_x \frac{h}{2L} - \frac{ma_y L_f h}{LT_r} - \frac{(K_{\phi r} \phi + D_{\phi r} \dot{\phi})}{T_r}$$

$$F_{zrr} = \frac{mgL_f}{2L} + ma_x \frac{h}{2L} + \frac{ma_y L_f h}{LT_r} + \frac{(K_{\phi r} \phi + D_{\phi r} \dot{\phi})}{T_r}$$

- ▶ F_{zij} is a function of both the vehicle's static and dynamic load transfer and where a_x and a_y are vehicle longitudinal and lateral accelerations

- The slip angle of each wheel is calculated as,

$$\alpha_{fl} = \delta_f - \arctan \left(\frac{v + L_f \gamma}{u - 0.5 T_f \gamma} \right)$$

$$\alpha_{fr} = \delta_f - \arctan \left(\frac{v + L_f \gamma}{u + 0.5 T_f \gamma} \right)$$

$$\alpha_{rl} = \delta_r - \arctan \left(\frac{v - L_r \gamma}{u - 0.5 T_r \gamma} \right)$$

$$\alpha_{rr} = \delta_r - \arctan \left(\frac{v - L_r \gamma}{u + 0.5 T_r \gamma} \right)$$

- These wheel longitudinal speeds are calculated

$$u_{fl} = \left(u - \frac{T_f}{2} \gamma \right) \cos \delta_f + (v + L_f \gamma) \sin \delta_f$$

$$u_{fr} = \left(u + \frac{T_f}{2} \gamma \right) \cos \delta_f + (v + L_f \gamma) \sin \delta_f$$

$$u_{rl} = \left(u - \frac{T_r}{2} \gamma \right) \cos \delta_r + (v - L_r \gamma) \sin \delta_r$$

$$u_{rr} = \left(u + \frac{T_r}{2} \gamma \right) \cos \delta_r + (v - L_r \gamma) \sin \delta_r$$

- ▶ Longitudinal wheel slip is defined as,

$$\lambda_{ij} = \frac{R_w \omega_{ij} - u_{ij}}{u_{ij}}, \quad R_w \omega_{ij} < u_{ij}$$

$$\lambda_{ij} = \frac{R_w \omega_{ij} - u_{ij}}{R_w \omega_{ij}}, \quad R_w \omega_{ij} \geq u_{ij}$$

- ▶ Where: R_w is the radius of the wheel.
 ω is the angular velocity of the wheel.

DRIVER MODEL MATHEMATICAL FORM(reference vector field)

- ▶ This driver model provides the control of longitudinal and lateral dynamics of a road vehicle
- ▶ The reference vector fields driver model represents the previewed kinematic control in form of a reference vector $\omega_{drv} = (\omega_x, \omega_y)$.
- ▶ The reference vector ω_{drv} , which targets on the compensative control of the vehicle, depends on the vehicle location in the path.
- ▶ The reference speed and the direction are given by $v_{cg}^d = |\omega_{drv}|$ and $\psi^d = \arctan\left(\frac{\omega_y}{\omega_x}\right)$ respectively.

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- ▶ The tangential and normal components of acceleration associated with the reference vector are

$$a_t^d = e_t^d \cdot (w_{drv} \cdot \nabla) w_{drv}$$

$$a_n^d = e_n^d \cdot (w_{drv} \cdot \nabla) w_{drv}$$

$$\text{where } \nabla \equiv (\partial / \partial w_y, \partial / \partial w_y)^T$$

Where e_t^d, e_n^d are unit vectors tangential and normal to the reference vector respectively.

- ▶ Vehicle longitudinal control is executed via an acceleration command a_x , which is converted to command traction force or braking torque.

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- ▶ Feedback of longitudinal acceleration is given by

$$\dot{a}_x = \frac{a_o - \dot{u}}{\tau_2}$$

where $a_o = \frac{|w_{drv}|^{-v_{cg}}}{\tau_1}$ and τ_1, τ_2 are design parameters

- ▶ The lateral speed is small compared to the longitudinal speed so,

$$u_d^d = v_{cg}^d$$

- ▶ The reference yaw rate is given by

$$\gamma_o = \frac{v_{cg} a_n^d}{|w_{drv}|^2} \text{ (for lateral control)}$$

LATERAL CONTROL

- ▶ Combined with the angular error, the reference yaw rate γ_o is modified as

$$\gamma_d^d = \gamma_o - \frac{\psi + \beta - \psi^d}{\tau_3}$$

where $\beta = \frac{v}{u}$ (vehicle side slip angle)

- ▶ For lateral control, the steering angle δ_f is computed using a similar integrator approach and is given by

$$\dot{\delta}_f = k_{lat} (\gamma_d^d - \gamma)$$

where τ_3 and k_{lat} are design control parameters.

LINEAR VEHICLE MODEL (3-DOF)

- To design the integrated active chassis controller, a 3-DOF yaw plane model is described in vehicle coordinates. Its state space equation is represented as:

$$\dot{x} = Ax + EW + BU$$

$$\text{Where } x = [u \ v \ \gamma]^T, \ w = [\delta_f \ \gamma_d^d \ u_d^d], \ U = [\Delta F_x \ \delta_r \ M]$$

$$A = \begin{bmatrix} \gamma & 0 & 0 \\ 0 & -\frac{2(C_f + C_r)}{mu} & -u - \frac{2(C_f L_f - C_r L_r)}{mu} \\ 0 & -\frac{2(C_f L_f - C_r L_r)}{I_{zz}u} & -\frac{2(C_f L_f^2 - C_r L_r^2)}{I_{zz}u} \end{bmatrix}, \ E = \begin{bmatrix} 0 & 0 & 0 \\ \frac{2C_f}{m} & 0 & 0 \\ \frac{2C_f L_f}{I_{zz}} & 0 & 0 \end{bmatrix},$$

$$U = \begin{bmatrix} \frac{1}{m} & 0 & 0 \\ 0 & \frac{2C_r}{m} & 0 \\ 0 & \frac{-2C_r L_r}{I_{zz}} & \frac{1}{I_{zz}} \end{bmatrix}$$

BLOCK DIAGRAM

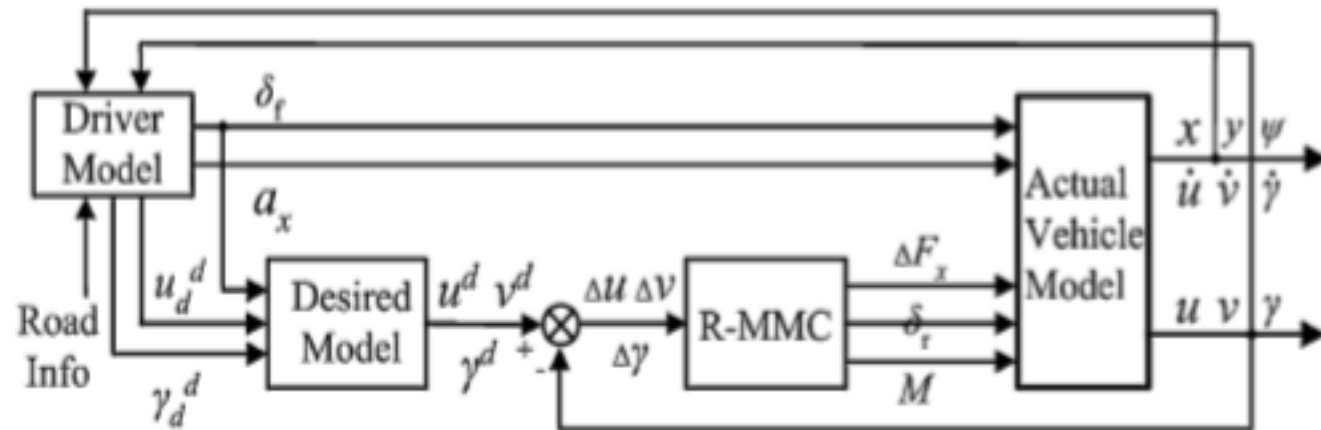


Fig. 2. Block diagram of the closed-loop driver-vehicle system.

- ▶ Where C_f and C_r indicate the cornering stiffness of the front and rear tires. To match the desired model, a control input vector U is added to the yaw plane model.
- ▶ $M = C_\lambda \lambda_{ij} F_{zij}(T_i/2)$.
- ▶ C_λ is the slope in the linear region of the μ - λ curve, and T_i is the track width of the vehicle.

In the 3-DOF model, the front and rear wheel slip angles α_f and α_r are approximated as:

$$\alpha_f = \delta_f - \beta - \frac{L_f \gamma}{u}$$

$$\alpha_r = \delta_r - \beta - \frac{L_r \gamma}{u}$$

The lateral forces generated by the front and rear tires can be calculated as:

$$F_{yfl} + F_{yfr} \approx -2C_f \alpha_f$$

$$F_{yrl} + F_{yrr} \approx -2C_r \alpha_r$$

The desired longitudinal velocity, lateral velocity and yaw rate are three independent first-order systems according to the driver's inputs. They can be represented as:

$$\dot{x}_d = A_d x_d + E_d W = \begin{bmatrix} -\frac{1}{\tau_u} & 0 & 0 \\ 0 & -\frac{1}{\tau_v} & 0 \\ 0 & 0 & -\frac{1}{\tau_\gamma} \end{bmatrix} \begin{bmatrix} u^d \\ v^d \\ \gamma^d \end{bmatrix} + \begin{bmatrix} 0 & 0 & \frac{k_u}{\tau_u} \\ \frac{k_v}{\tau_v} & 0 & 0 \\ 0 & \frac{k_\gamma}{\tau_\gamma} & 0 \end{bmatrix} \begin{bmatrix} \delta_f \\ \gamma_d^d \\ u_d^d \end{bmatrix}$$

- ▶ where τ_u , τ_v and τ_y are design time constants. k_u , k_v and k_y are design gains.
- ▶ The desired lateral speed is always zero.
- ▶ In this study, a desired oversteer yaw moment control can be obtained by adjusting the slip ratio of the outer front wheel.
- ▶ Similarly, a desired understeer yaw moment control is obtained by adjusting the slip ratio of the inner rear wheel.

MODEL MATCHING CONTROLLER (ERROR VECTOR CALCULATION)

- ▶ The results from mathematical model which we obtained in above slides are not the desired results and these are called the actual results, so the error vector is used to calculate the difference between the desired and the actual values.
- ▶ In order to make the actual vehicle longitudinal speed, lateral speed and yaw rate follow their desired values, the error variables are defined as:

$$e = x_d - x = [\Delta u \ \Delta v \ \Delta \gamma]^T$$

- ▶ Accordingly, the error state space equation is given by

$$\dot{e} = \dot{x}_d - \dot{x} = A_d(x_d - x) + (A_d - A)x + (E_d - E)W - BU$$

Contd.....

- So the augmented error state space equation can be written as

$$\begin{bmatrix} \dot{e} \\ \dot{x} \end{bmatrix} = \begin{bmatrix} A_d & A_d - A \\ 0 & A \end{bmatrix} \begin{bmatrix} x_d - x \\ x \end{bmatrix} + \begin{bmatrix} E_d - E \\ E \end{bmatrix} W + \begin{bmatrix} -B \\ B \end{bmatrix} U$$

the above equation can be written in the form of

$$\dot{X} = A_1 X + B_1 W + B_2 U \text{ -----(1)}$$

where A_1 , B_1 , B_2 are the state space matrices

and the state vector is given by

$$X = [\Delta u \ \Delta v \ \Delta \gamma \ u \ v \ \gamma]^T$$

where Δu , Δv , $\Delta \gamma$ are the errors in longitudinal, lateral velocities and yaw rate

Contd.....

- ▶ The state matrices are given by

$$A_1 = \begin{bmatrix} A_d & A_d - A \\ 0 & A \end{bmatrix}$$

$$B_1 = \begin{bmatrix} E_d - E \\ E \end{bmatrix}$$

$$B_2 = \begin{bmatrix} -B \\ B \end{bmatrix}$$

- ▶ In this the simulation is done by H_∞ optimal controller in terms of LMI's
- ▶ The performance output vector Z is defined as follows

$$Z = [\Delta u \ \Delta v \ \Delta \gamma \ \Delta F_x \ \delta_r \ M]^T = C_1 X + D_{11} W + D_{12} U$$

where C_1, D_{11}, D_{12} are state matrices derived from equation (1)

CONTD.....

- ▶ The output feedback variables are Y are defined as:

$$Y = [\Delta u \ \Delta v \ \Delta \gamma]^T = C_2 X + D_{21} W + D_{22} U Z$$

where C_2 , D_{21} , D_{22} are also state matrices derived from equation (1)

- ▶ The state space error model is given by

$$G_p = \begin{bmatrix} A_1 & B_1 & B_2 \\ C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{bmatrix} \text{ (approx.)}$$

Thank You 😊