Torque Vectoring

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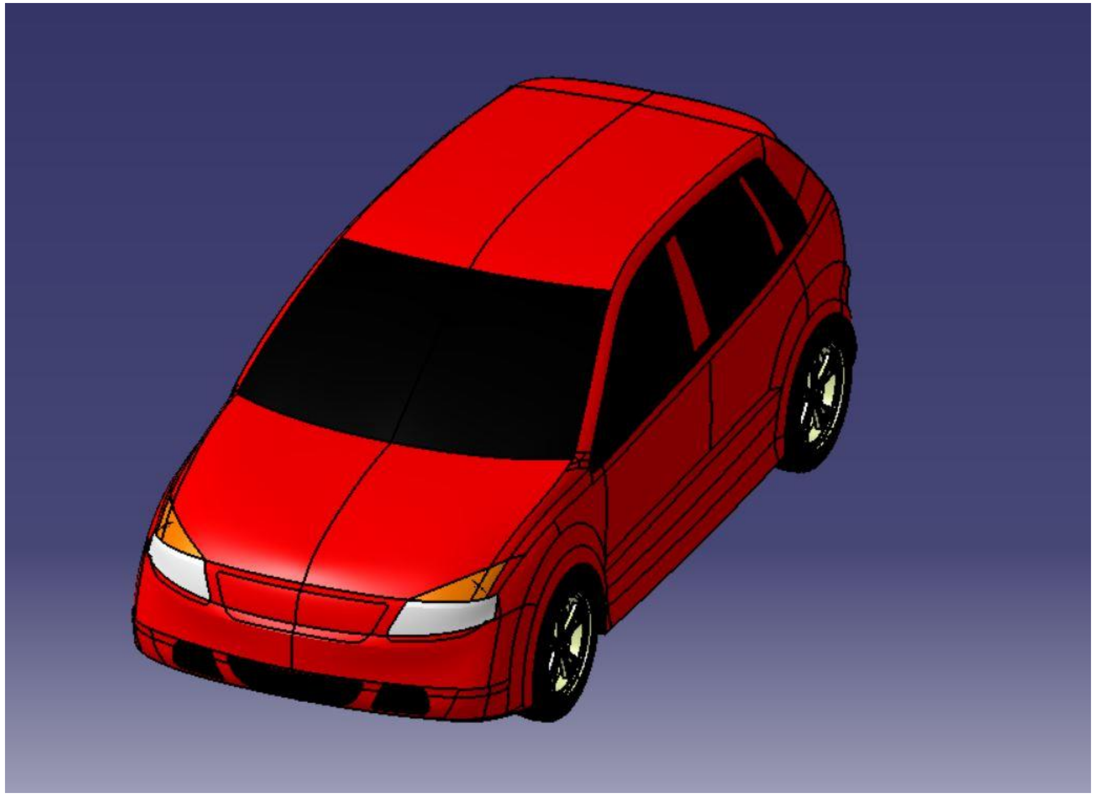
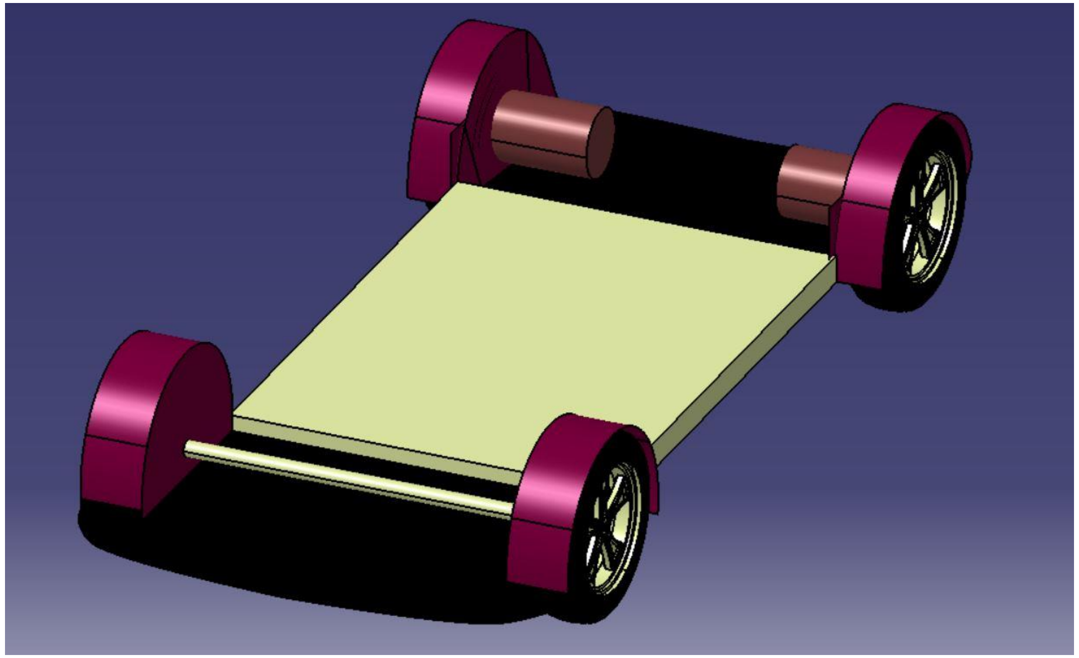
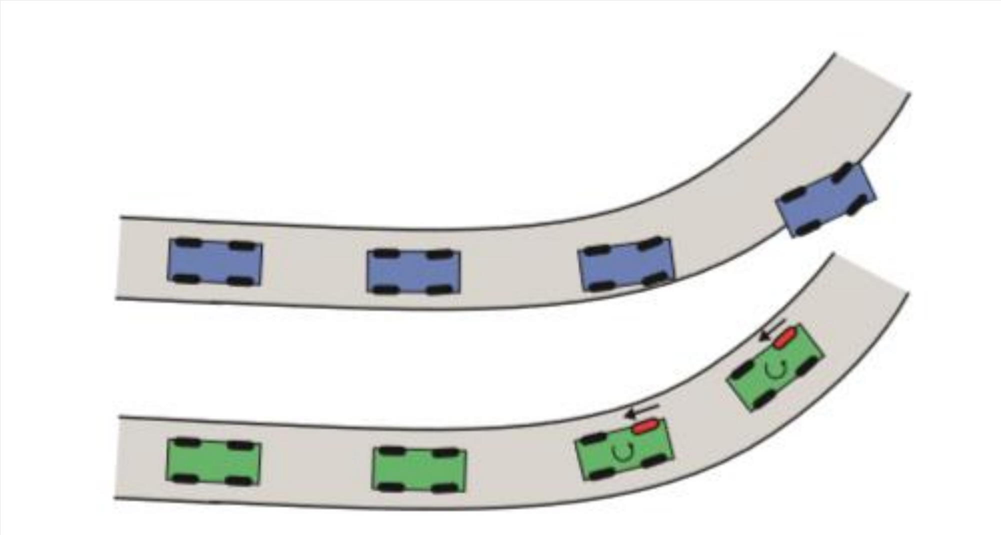
A Simple 3D Hatchback Model – CATIA V5



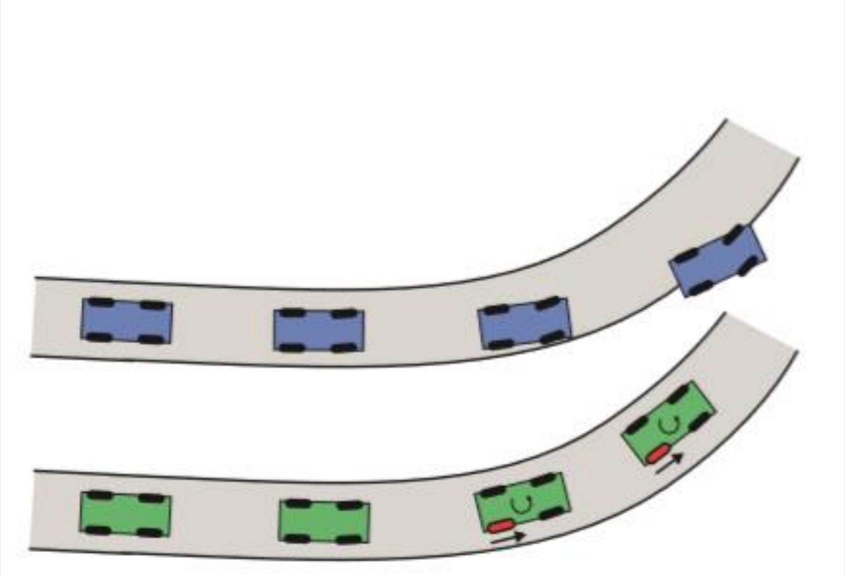


What is Torque Vectoring?

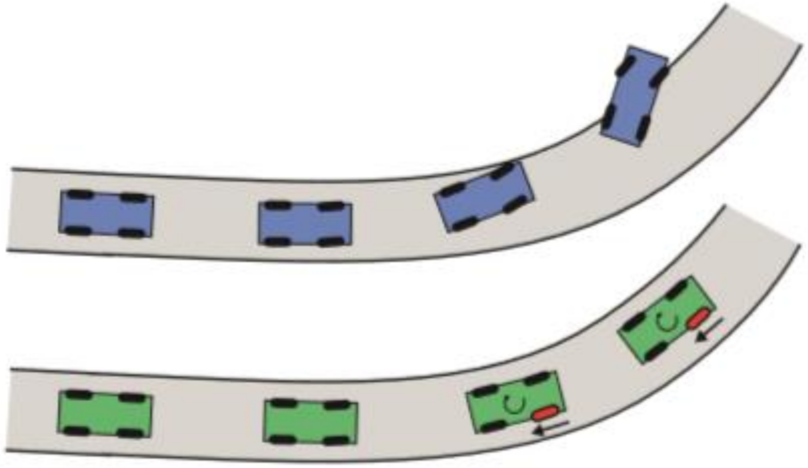
- The ESP is a prevalent stability system present in most of the automobiles.
- The ESP uses wheel braking to control the yaw moment given to the vehicle to stabilise the vehicle during navigation of turns. The difference in the braking forces in the wheels generates the yaw moment which is advantageous in critical situations to decelerate the vehicle.



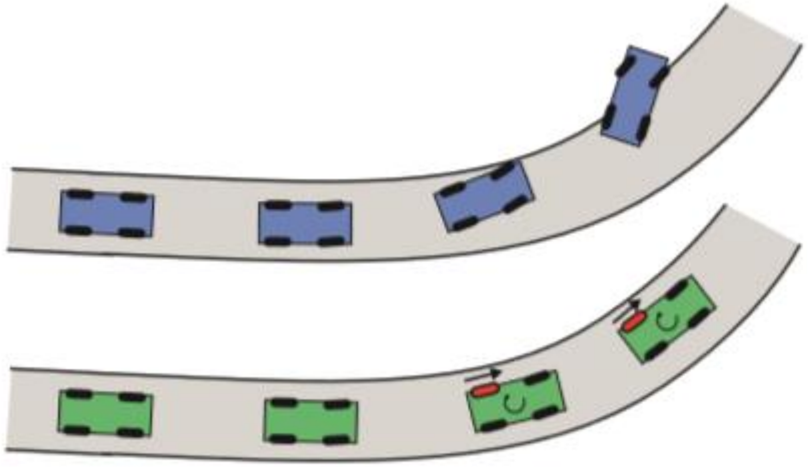
ESP in an understeer vehicle



TV in an understeer vehicle

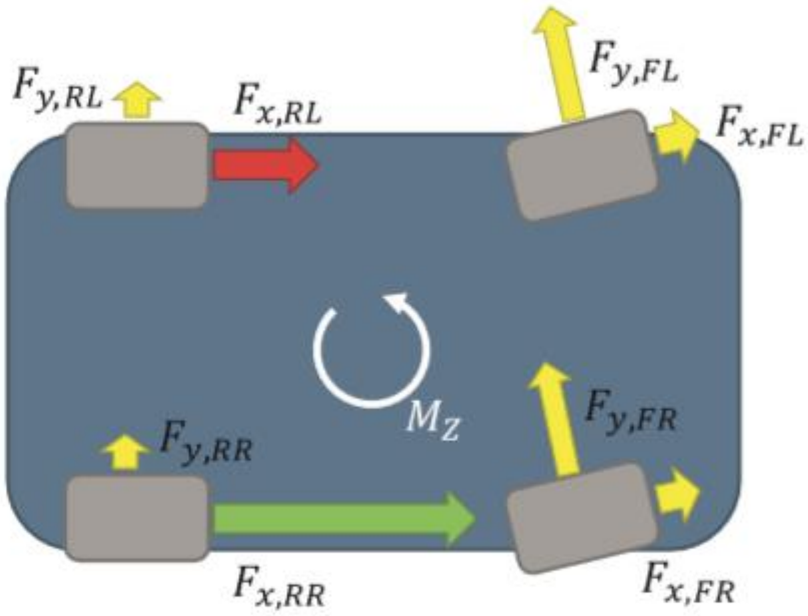


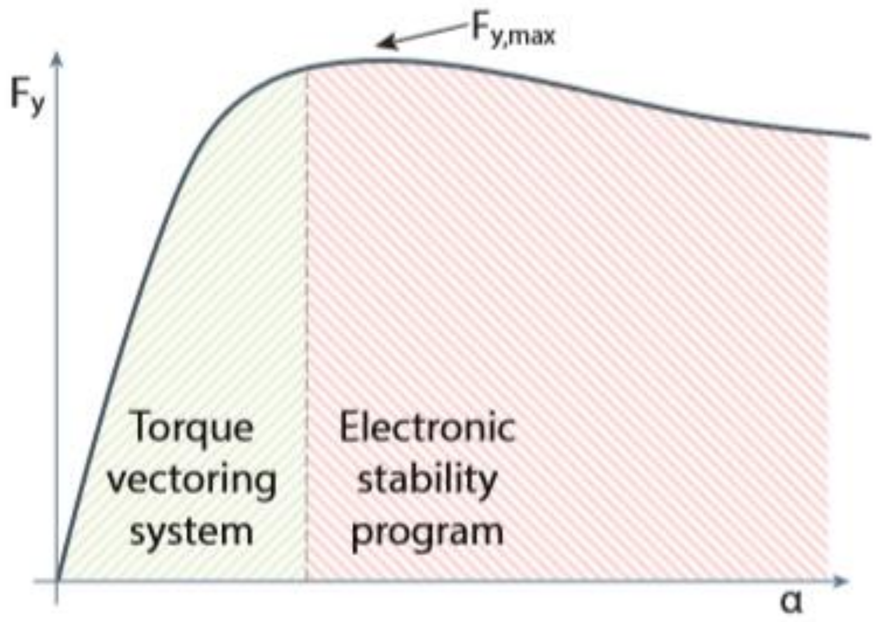
ESP in an oversteer vehicle



TV in an oversteer vehicle

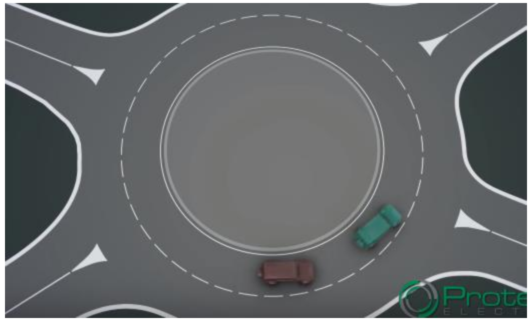
- ESP stabilises the vehicle by slowing down the vehicle which is undesirable in cases where the driver intends to accelerate the vehicle.
- Torque Vectoring systems do not slow down the vehicle as there is **no braking** rather only a redistribution of torques between the two wheels of the driven axle.
- Torque Vectoring Systems help redistribute torques to wheels under different traction conditions. An ECU combined with sensors deliver inputs like tractions, yaw rates, lateral accelerations to the ECU which then decides to distribute the power accordingly as Front-Rear, Left-Right or All Wheels.





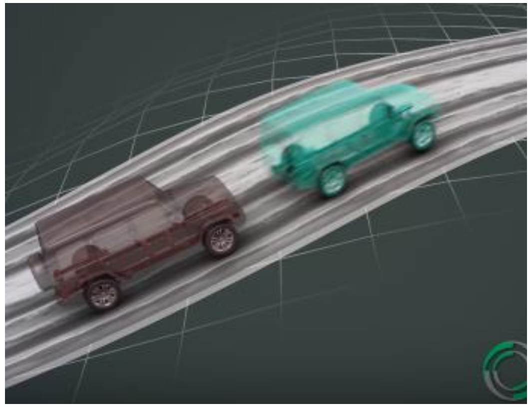
- As there is no change in the longitudinal forces, there is no slowing down of the vehicle.
- As the torque is redistributed, a yaw moment is given to the vehicle which corrects the path of the vehicle

Torque Vectoring Application scenarios

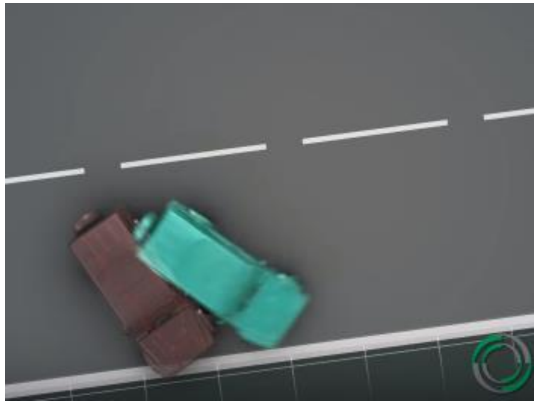


Permits high cornering speeds

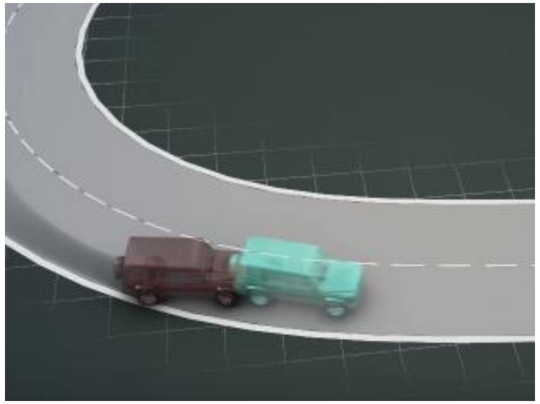
Sudden lane change stability



Traction Control



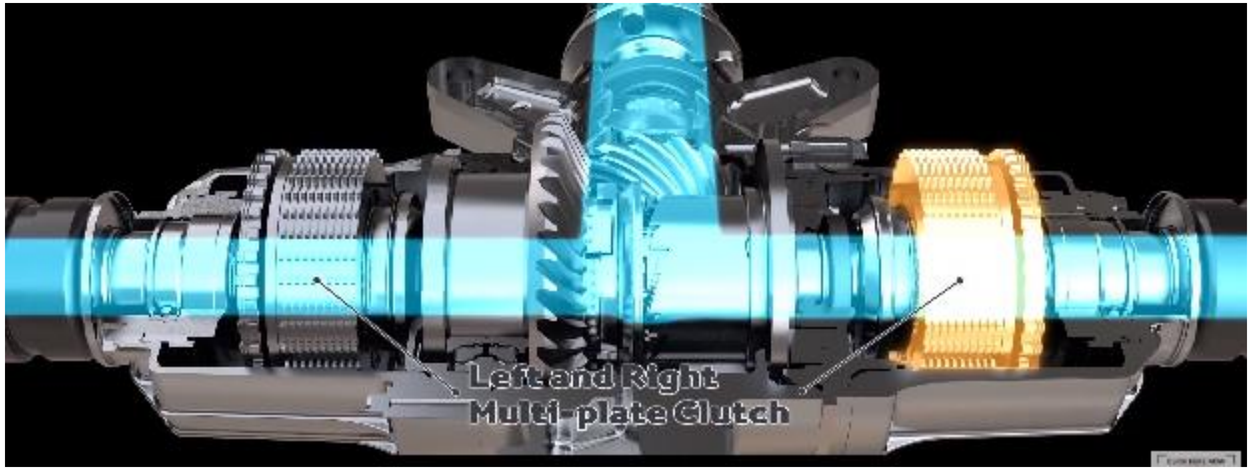
Low Speed Driving
Manoeuvrability



Active Safety

- Torque Vectoring with electronic control systems detect the traction at the different wheels and redistribute the power to the wheel with better traction.

- The distribution can be front-rear or left-right depending on the differential present. Electric systems respond much faster and the response is less complex as the torque/power is variable at each of the driving motors.



Application of clutch plates with hydraulic actuators to Distribute power between the wheels

Governing Equations

- Considering only cornering forces at steady state conditions.

- $For\ small\ \beta\ and\ \delta_i, the\ terms\ containing\ the\ longitundinal\ forces\ of\ the\ tires\ can\ be\ neglected\ and\ the\ equilibrium\ equations\ reduce\ to$

$$\Sigma F_{y_i} = \frac{mV^2}{R}$$

$$\Sigma F_{y_i} x_i = 0$$

Governing Equations

- Governing equation for handling for high speed cornering

$$mV(\dot{\beta} + r) + mV\beta = Y_\beta \beta + \dot{Y}_r r + Y_\delta + F_{ye}$$

$$J_z \dot{r} = N_\beta \beta + N_r r + N_\delta + M_{ze}$$

$m -$ *mass of the vehicle*

$V -$ *net speed of the vehicle*

$\beta -$ *side slip angle of the vehicle*

$r -$ *yaw rate or velocity*

$Y_\beta, Y_r, Y_\delta, N_\beta, N_r, N_\delta$ *are the stability derivatives*

F_{ye} *and* M_{ze} *are the external lateral force and external moment respectively*

- We obtain two first order differential equations which when in written in the state space form

$$\dot{z} = Az + B_c u_c + B_e u_e$$

where the state and input vectors $z, u_c \ and \ u_e \ are$

$$z = \begin{Bmatrix} \beta \\ r \end{Bmatrix}, u_c = \delta \ and \ u_e = \begin{Bmatrix} F_{ye} \\ M_{ze} \end{Bmatrix}$$

- $A = \begin{bmatrix} \frac{Y_\beta}{mV} - \frac{\dot{V}}{V} & \frac{Y_r}{mV} - 1 \\ \frac{N_\beta}{J_z} & \frac{N_r}{J_z} \end{bmatrix} \ is \ the \ dynamic \ matrix$

- $And\ the\ input\ gain\ matrices\ are$

$$B_c = \begin{bmatrix} \frac{Y_\delta}{mV} \\ \frac{N_\delta}{J_z} \end{bmatrix}\ and\ B_e = \begin{bmatrix} \frac{1}{mV} & 0 \\ 0 & \frac{1}{J_z} \end{bmatrix}$$

- $Stability\ is\ decided\ from\ the\ eigen\ values\ of\ the\ dynamic\ matrix\ A.$

$$|A - \lambda I| = 0$$

Vehicle Parameters and Model

VEHICLE PARAMETERS USED :

m = 1500; L = 2160 mm; a = 870 mm; b = 1290 mm
w = 1560 mm; Rw = 300 mm;

Cornering Stiffnesses :

$$C_f = C_1 = 67369 \frac{N}{rad}$$

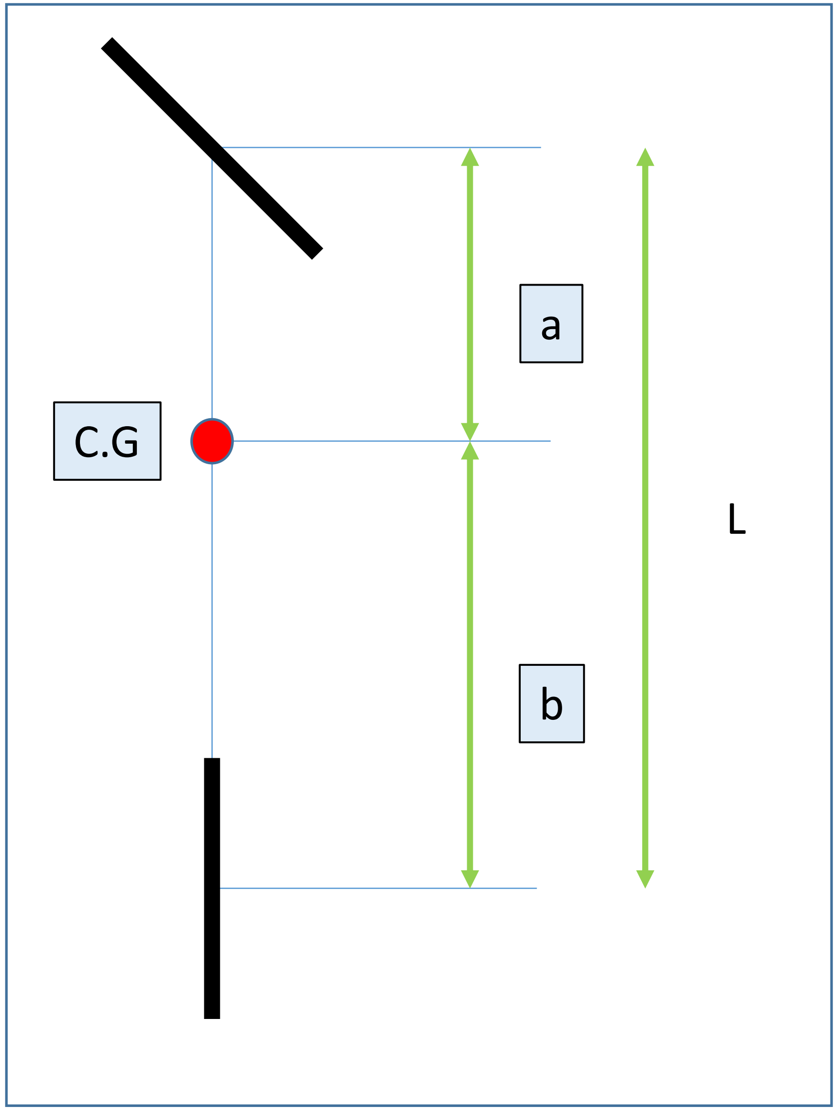
$$C_r = C_2 = 63411 \frac{N}{rad}$$

$$S = 1.7\ m^2; C_{y,\beta=-2.2}; C_{mz,\beta} = 0.6;$$

$$mz1_{,\alpha} = 2010; mz2_{,\alpha} = 1366;$$

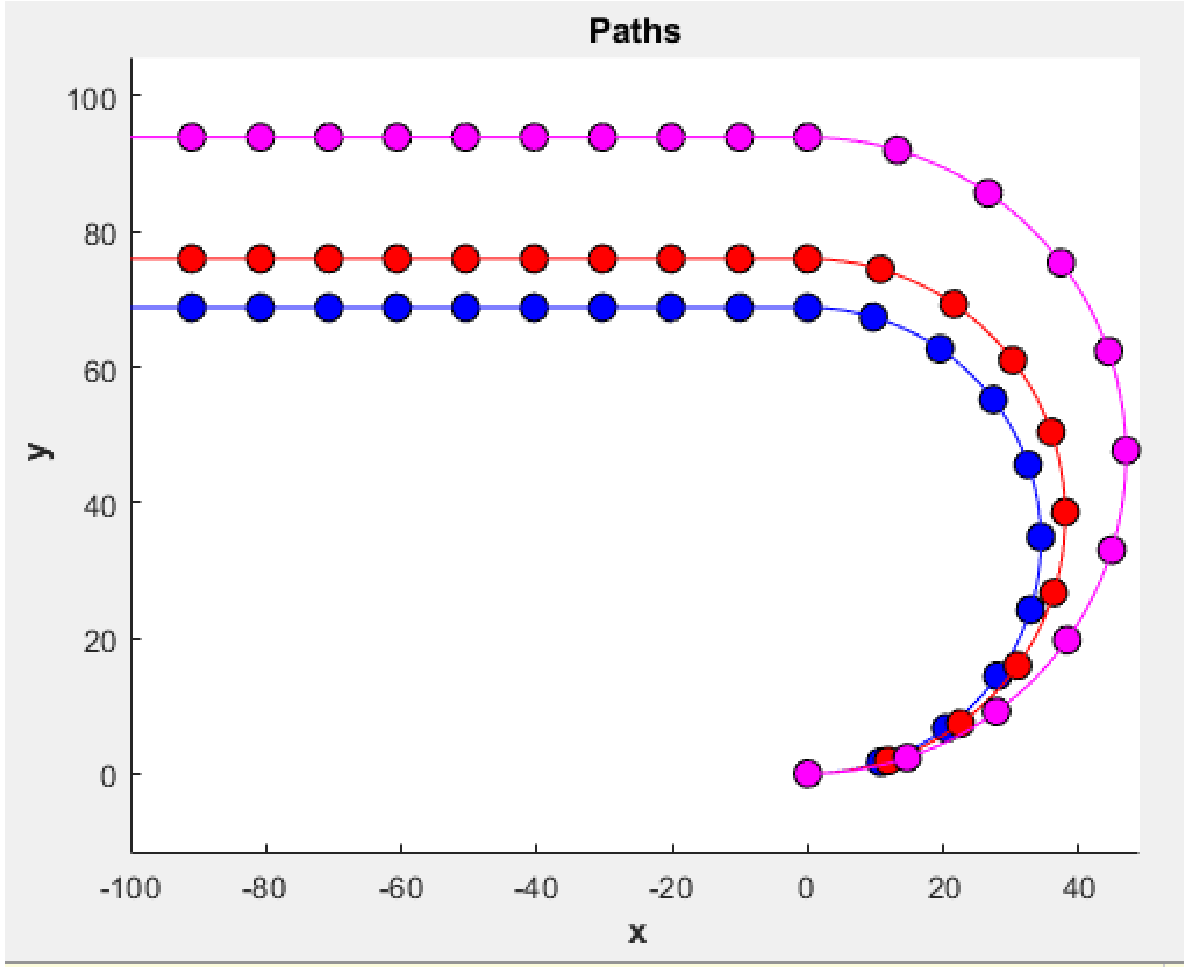
$$\rho = 1.225 \frac{kg}{m^3}; J_z = I = 2000\ kgm^2$$

(Obtained from *The Automotive Chassis Volume2 : System Design , Giancarlo Genta and Lorenzo Morello*



Single Track / Bicycle Model

Paths Followed by Vehicles with understeers for different considerations

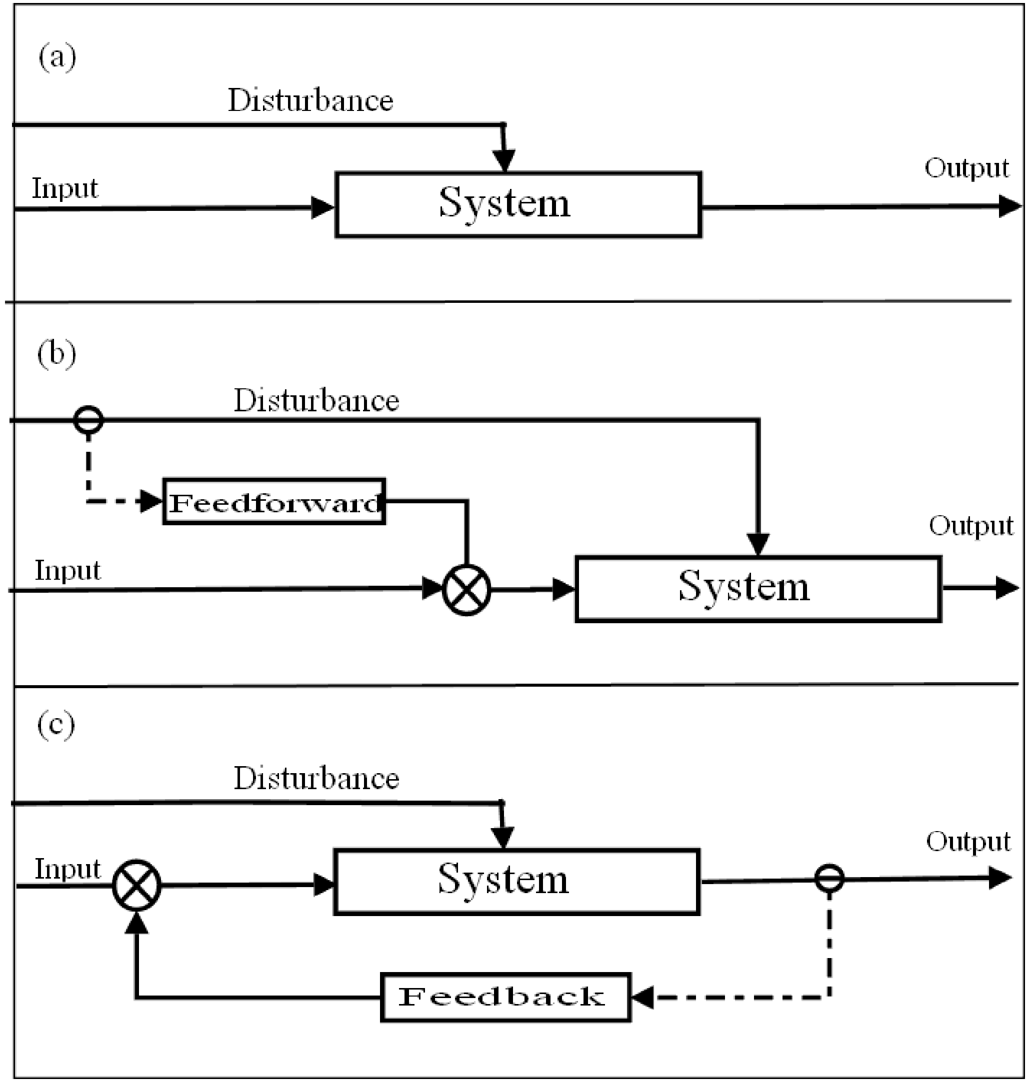


Under Kinematic Steering

Under Simplified Conditions With Only Cornering Forces

By considering cornering forces, aerodynamic forces, aligning torques.

Types of Control Systems



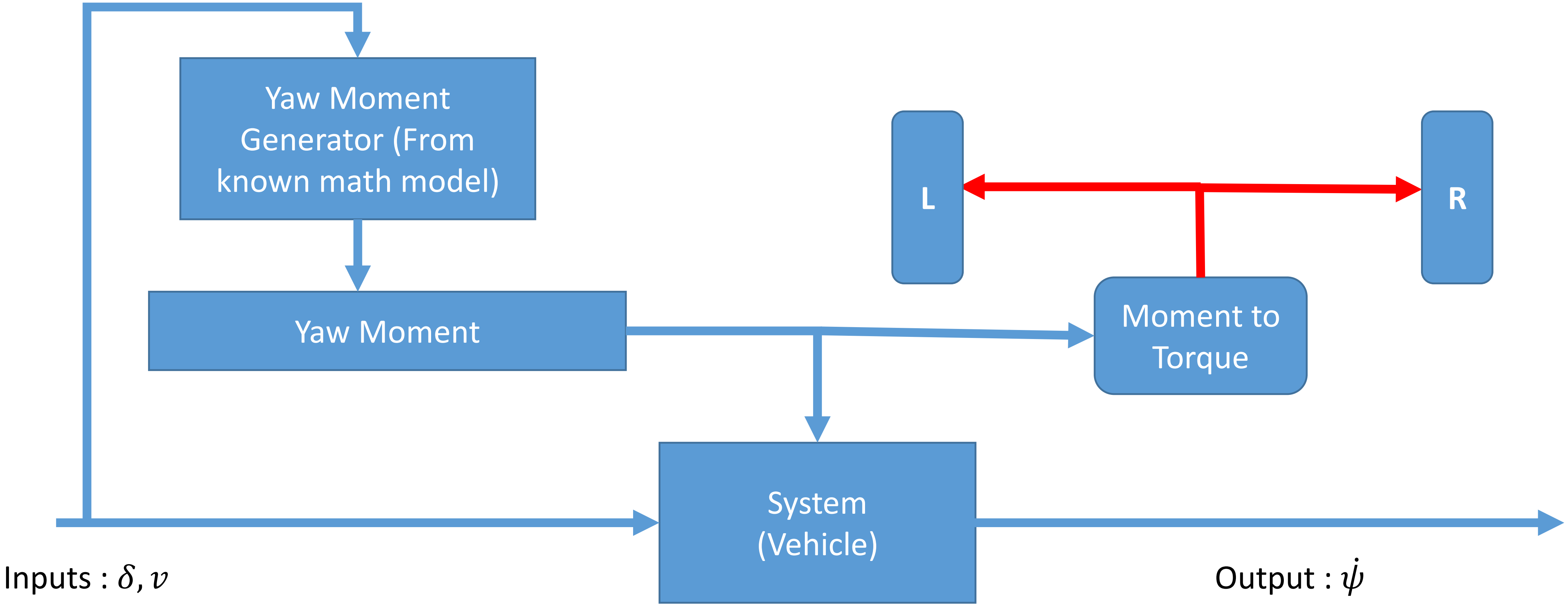
Source: Wiki

Open Loop Control System

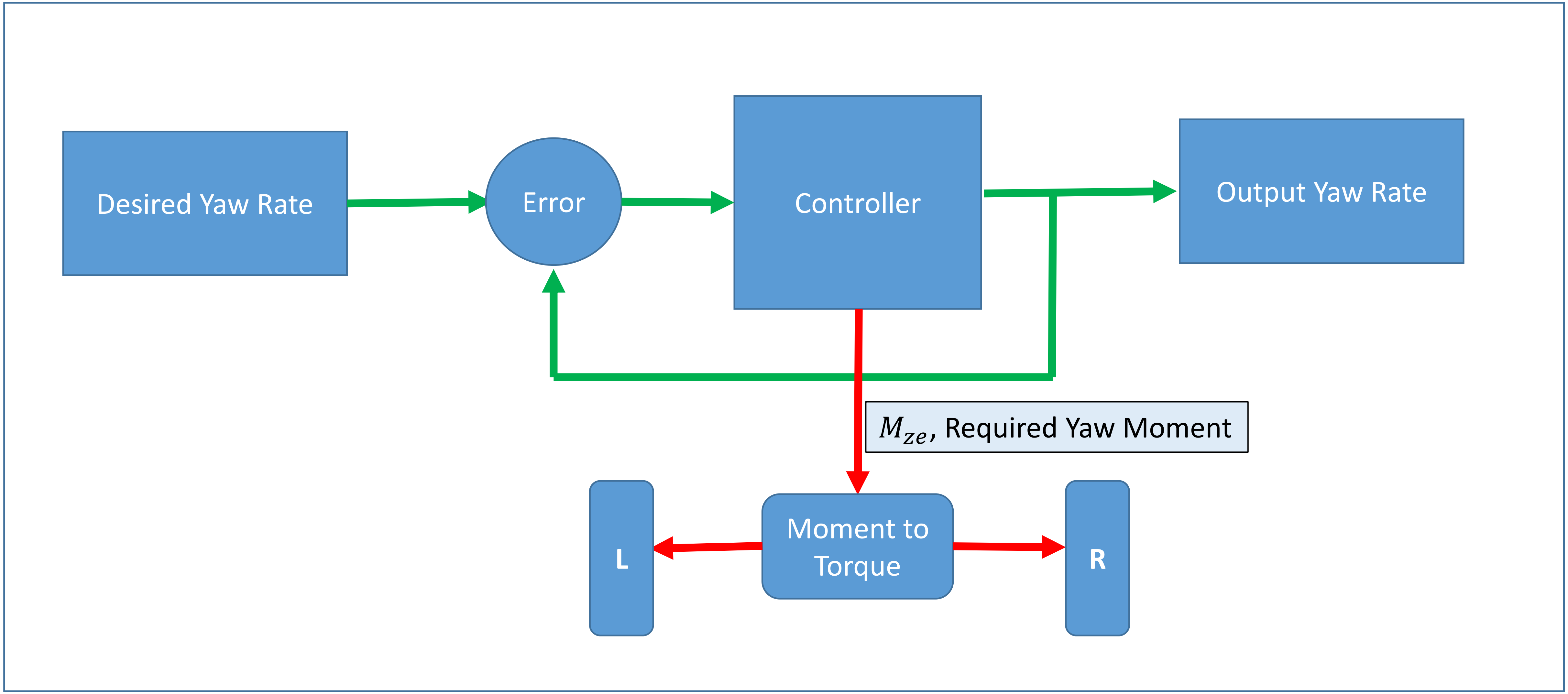
Feedforward Control System

Feedback Control System (Closed Loop)

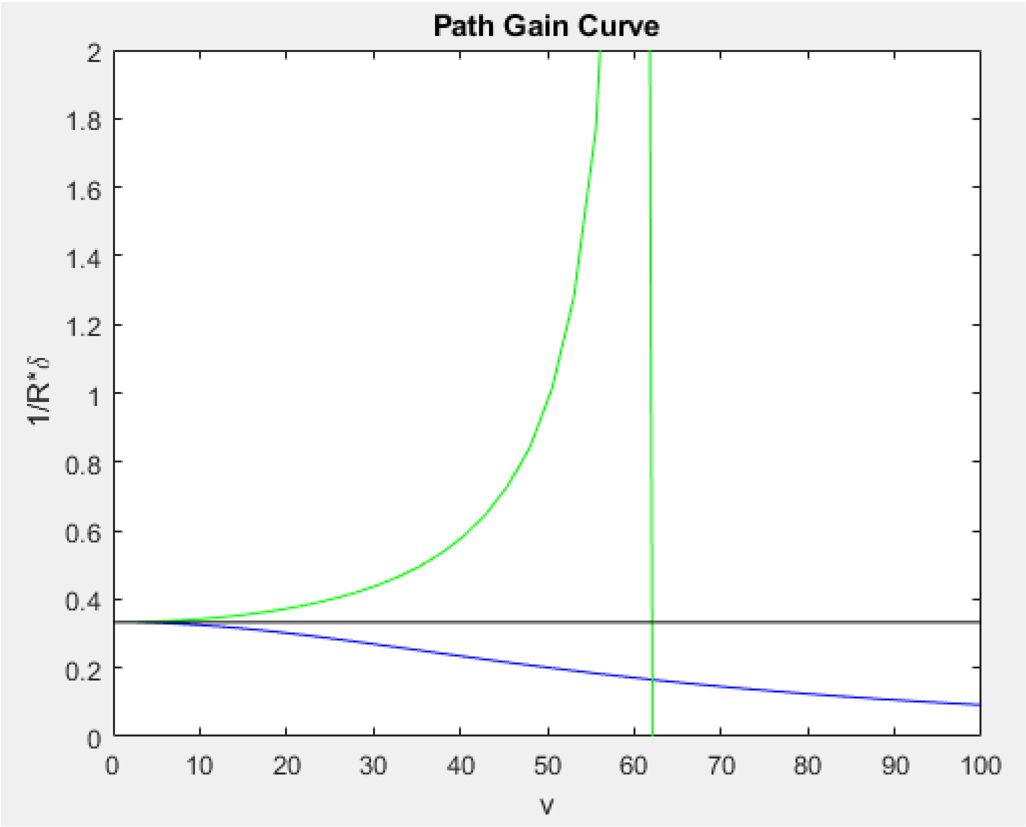
Simple Feedforward Control With Yaw Moment as Input



Feedback Control Model

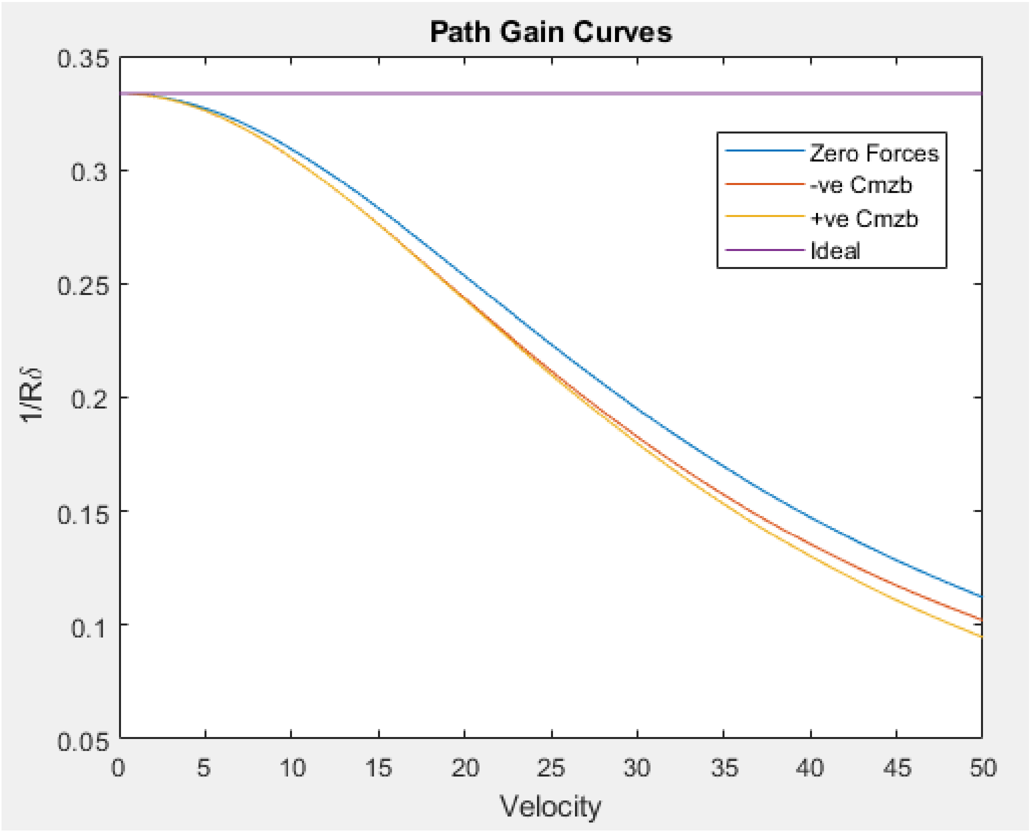


Path Gain Curves



$$\frac{1}{R\delta} = \frac{1}{l}\frac{1}{1 + \frac{K_{us}V^2}{gl}}$$

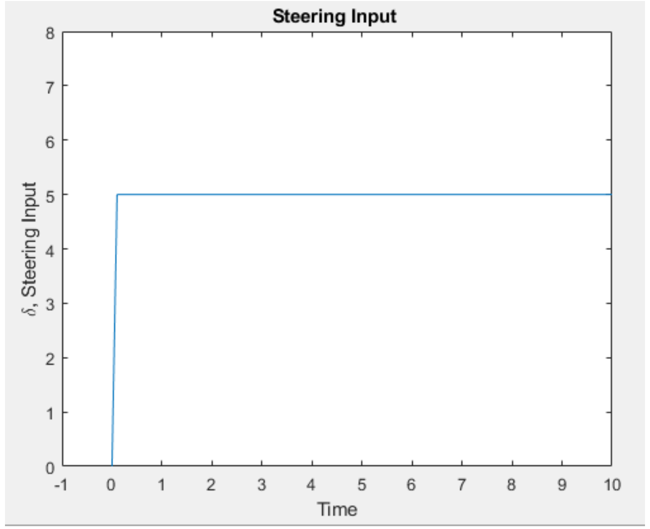
Neglecting the aerodynamic forces and aligning torques.



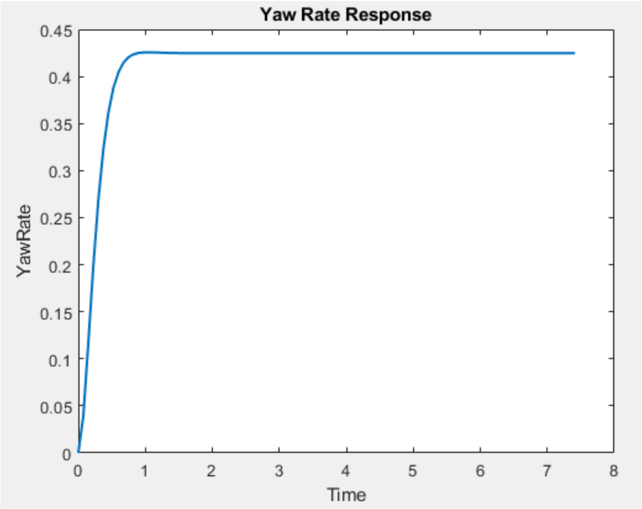
$$\frac{1}{R\delta} = \frac{y_\delta n_\beta - n_\delta y_\beta}{n_\beta(V^2 m - V y_r) + V n_r y_\beta}$$

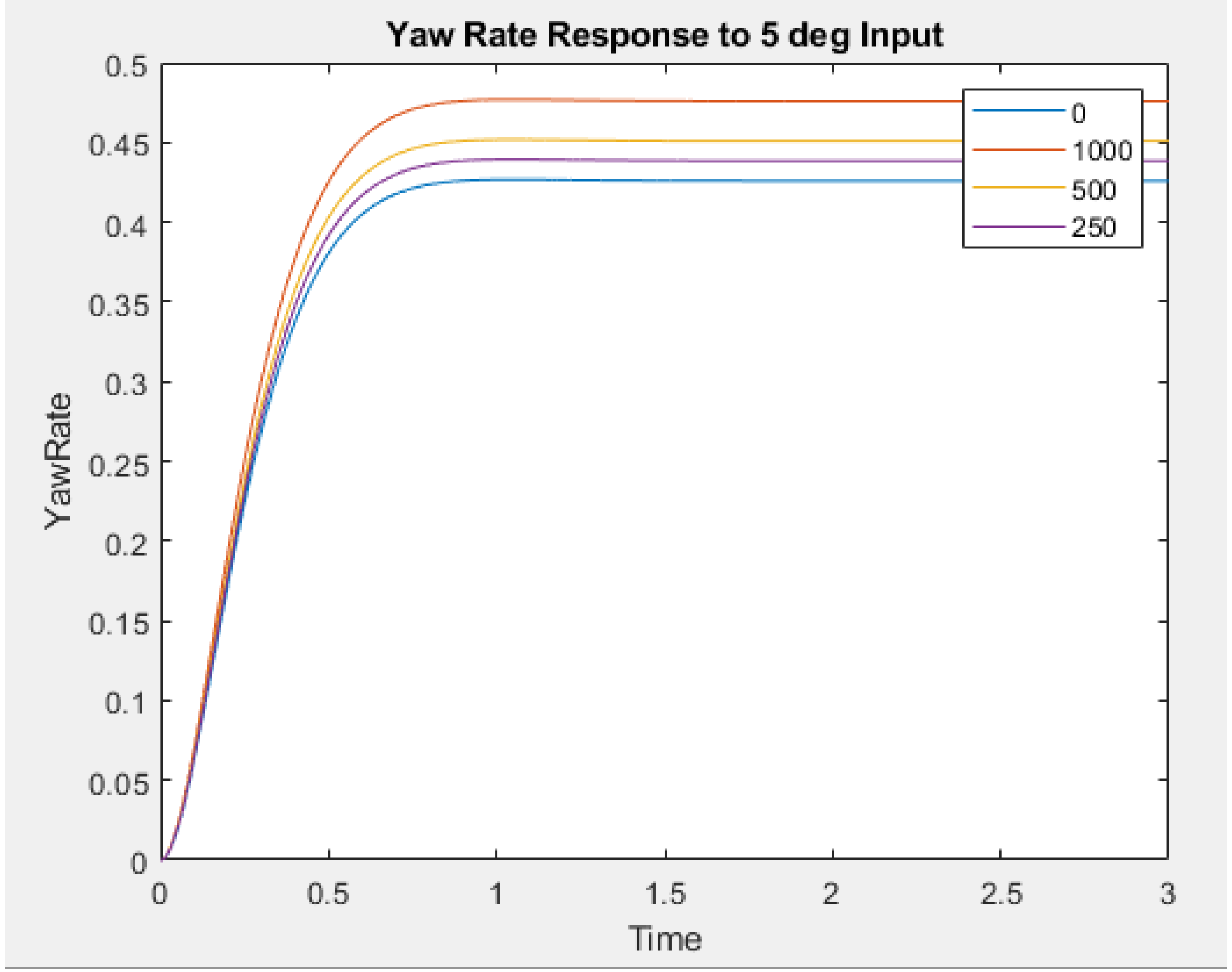
Yaw Rate Responses at **Various Yaw Moments** to 5 degree step steering Input

Steering Input

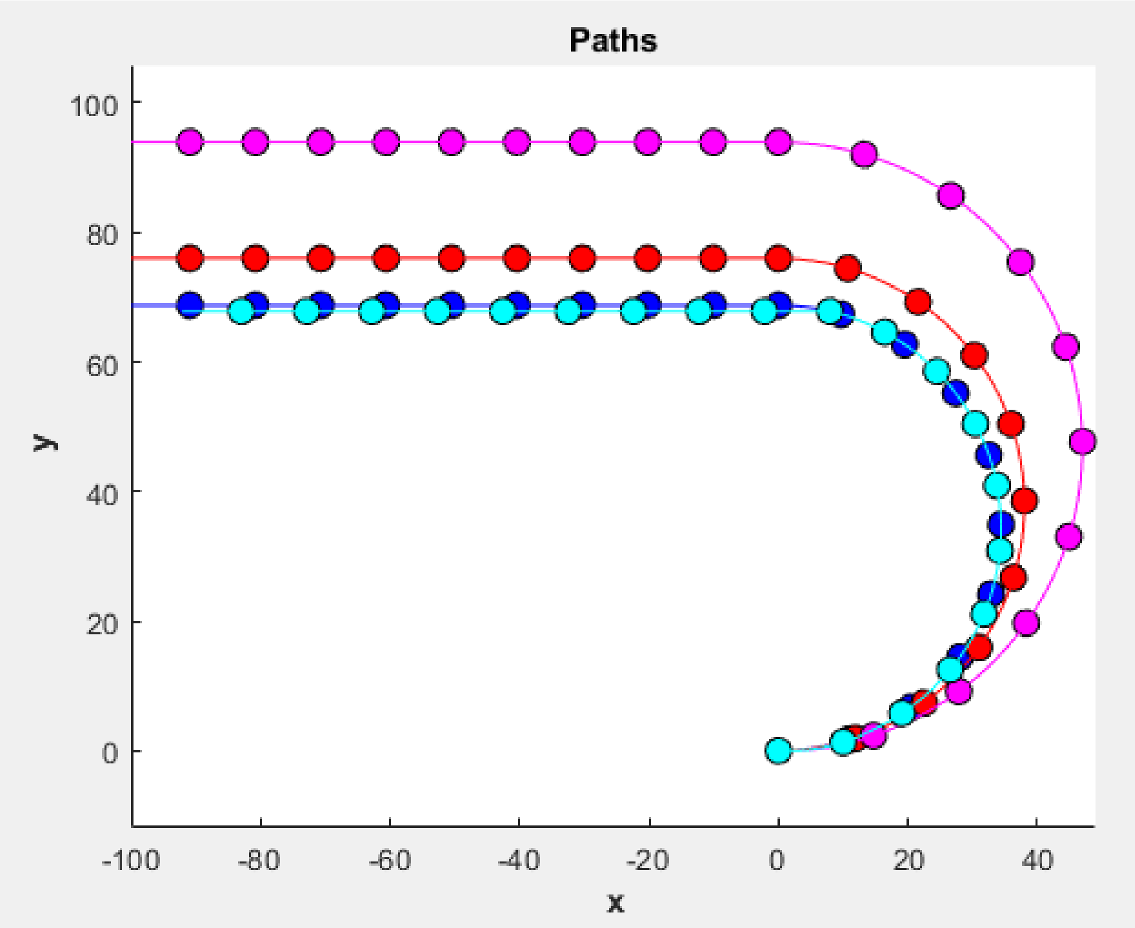


Yaw Rate (from ode45)

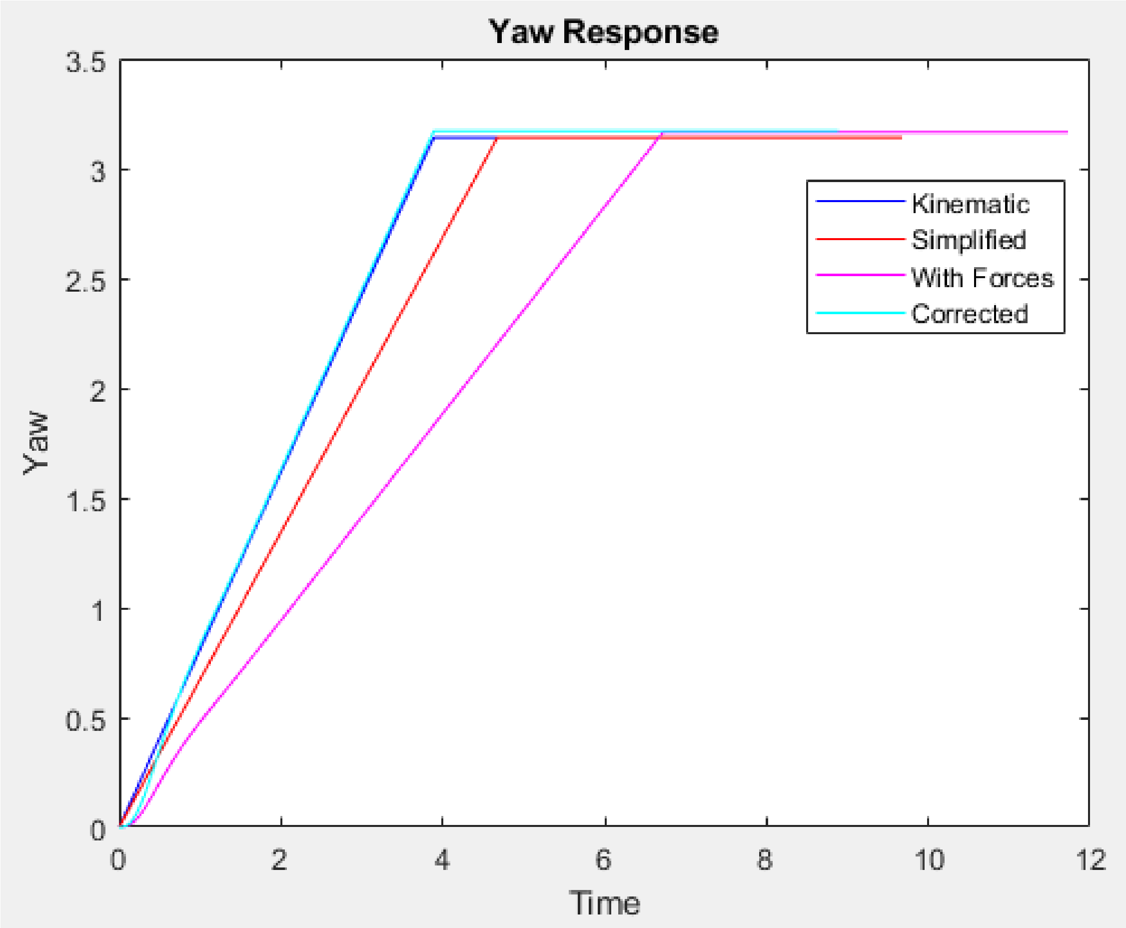




Corrected Path of Vehicle at 72 kmph

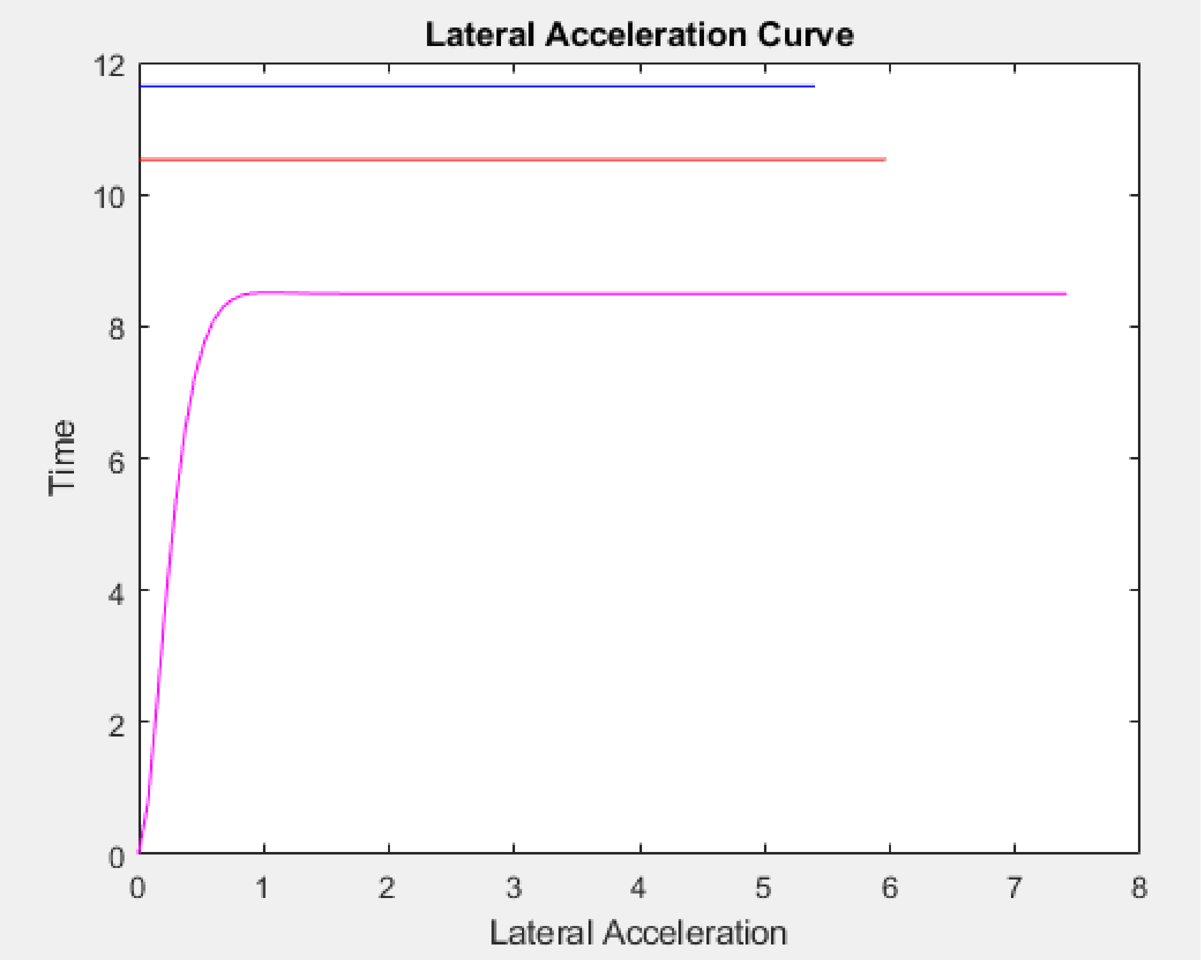


Path Followed by the Vehicle



Yaw Response of the Vehicle

Lateral Acceleration Curves



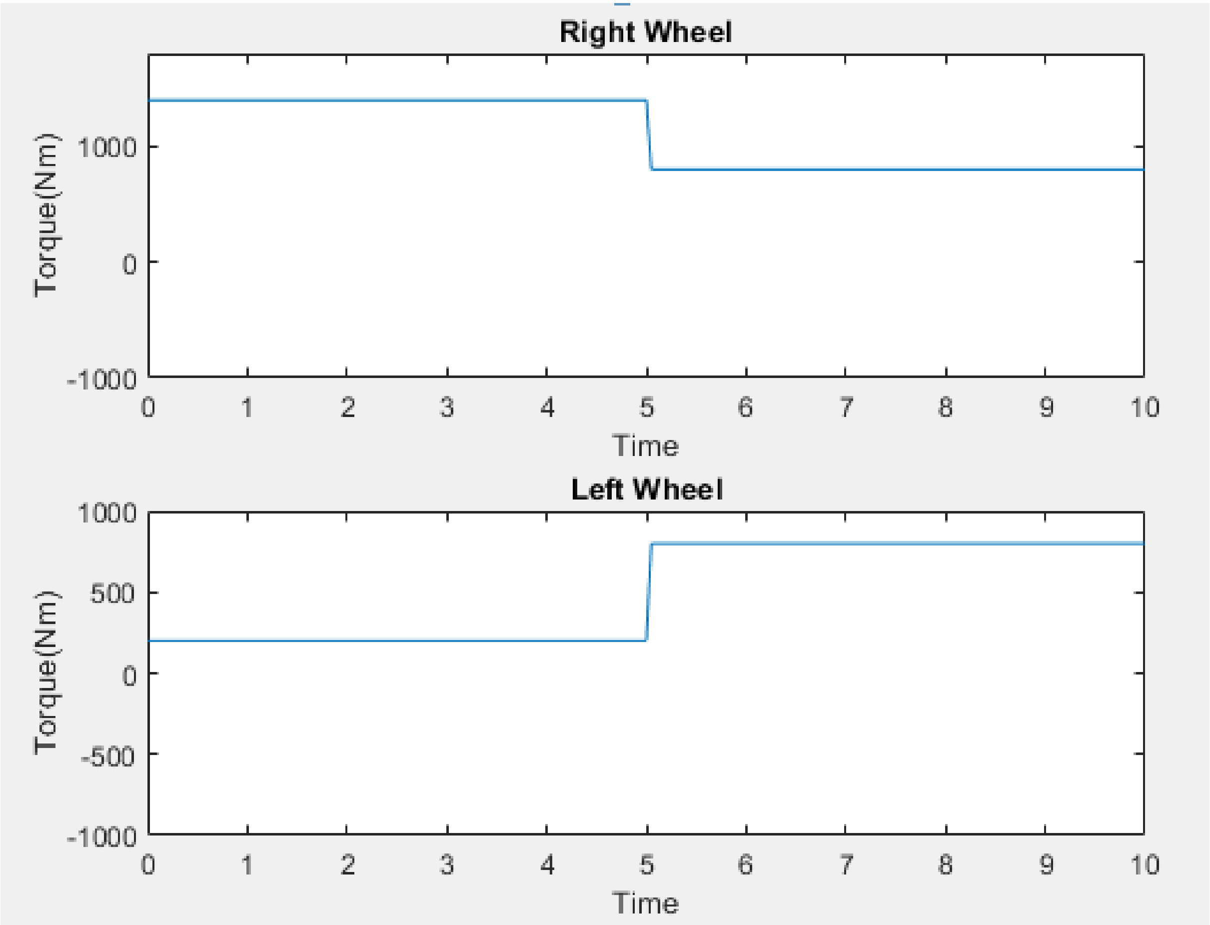
- The lateral accelerations are determined for the different scenarios by the formula

$$a_g = \frac{v^2}{r}$$

- For the kinematic and the simplified cases the movement was considered to be steady and the lateral accelerations were determined as constants and for the case with aerodynamic forces and the aligning torques also considered, the transient state also was considered due to which the lateral acceleration follows a similar curve.

- A feedback control can be used to find the error between the desired ay value and the real time value and the required correction can be done using a controller.

Distributed Torque to Left and Right Wheels at 72 kmph



Required Yaw moment converted to Torque :

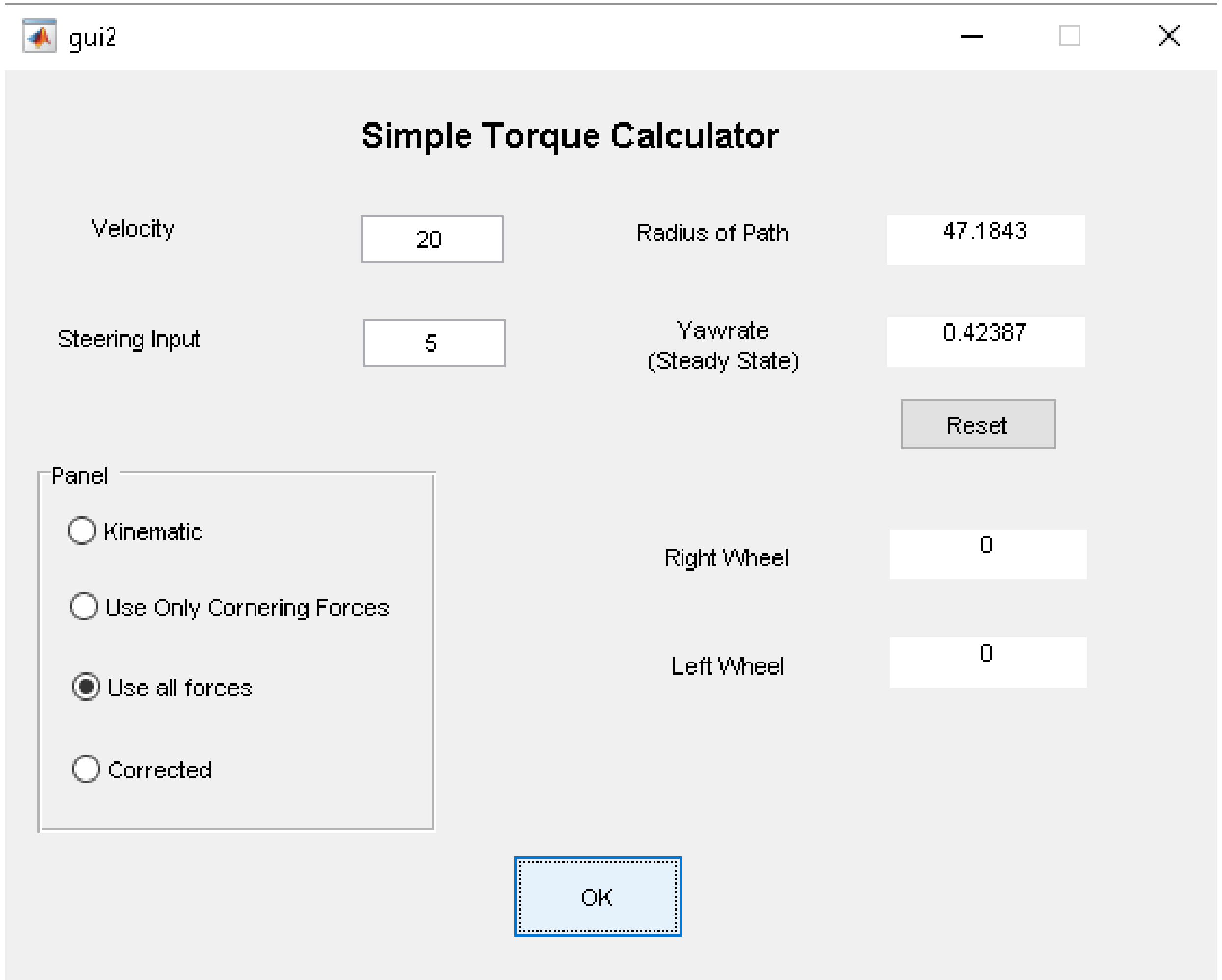
$$\Delta T_{TV} = \frac{M_{ze}}{w} r_w$$

$$T_{RW} = T_{req} + \Delta T_{TV}$$
$$T_{LW} = T_{req} - \Delta T_{TV}$$

Special Implementations

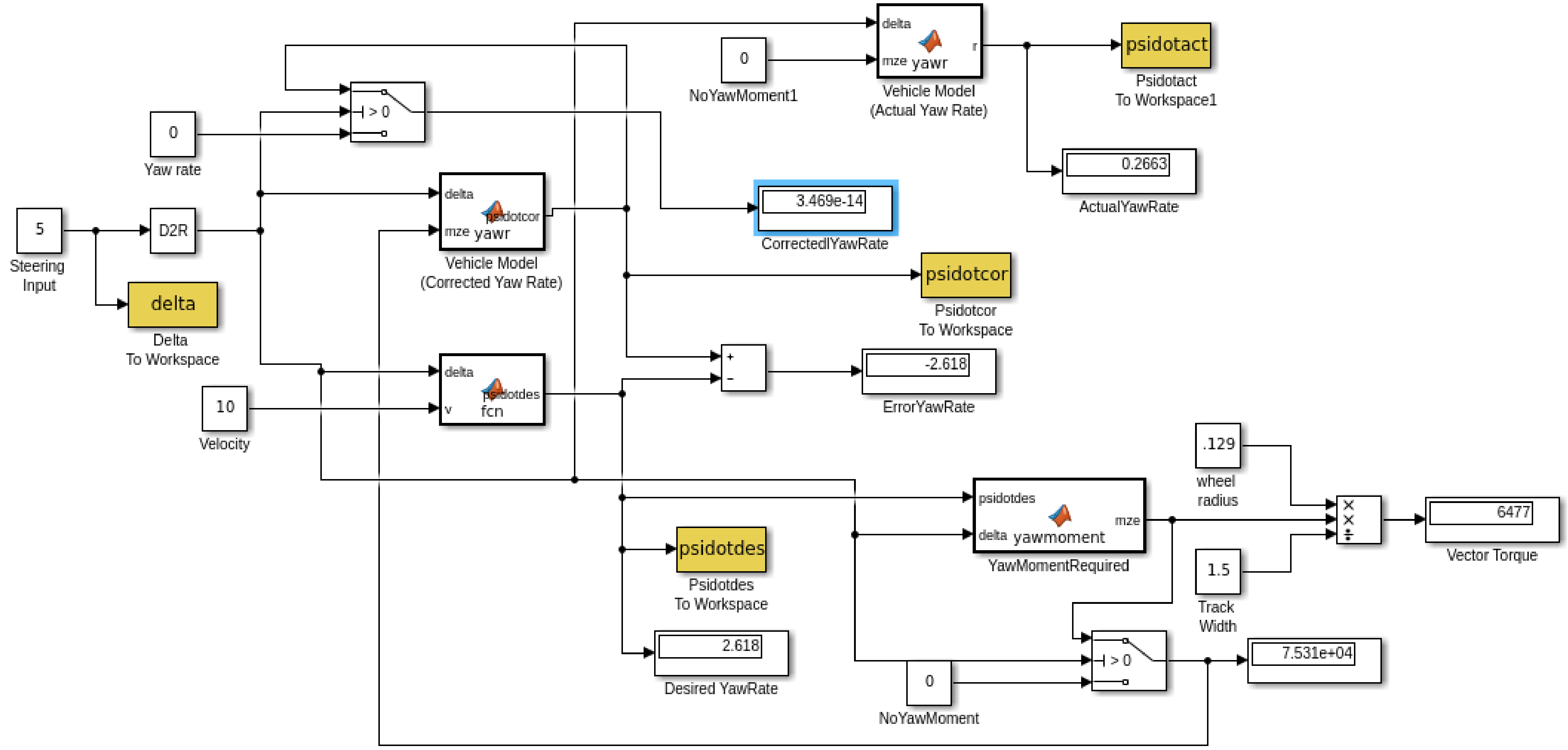
- Active Yaw Control by Mitsubishi
- Torque Vectoring Differential by Lexus
- Dynamic Performance by BMW
- Quattro with Torque Vectoring by Audi
- Torque Vectoring by Mercedes Benz (4Matic)

A Simple GUI for Torque Calculation



- **Uses a set of the vehicle parameters in this study to obtain the yaw rate and the path radius.**
- **For a step input of steering which can be entered to the application along with the velocity of the vehicle which formed the inputs to the system.**
- **Scope for further additions like a variable steering input and an all wheel torque distribution for stable vehicle dynamics, the purpose served by the torque vectoring system.**

A Simple Simulink Feedforward Control



Control Systems Integration

- The required torques are calculated from the inputs from the throttle.
- The completed control system software developed using the Matlab-Simulink is compiled into C code. The code also contains the instructions necessary to communicate the information with the module on the vehicle.

Module Used for Communication with the vehicle. Src: (1)

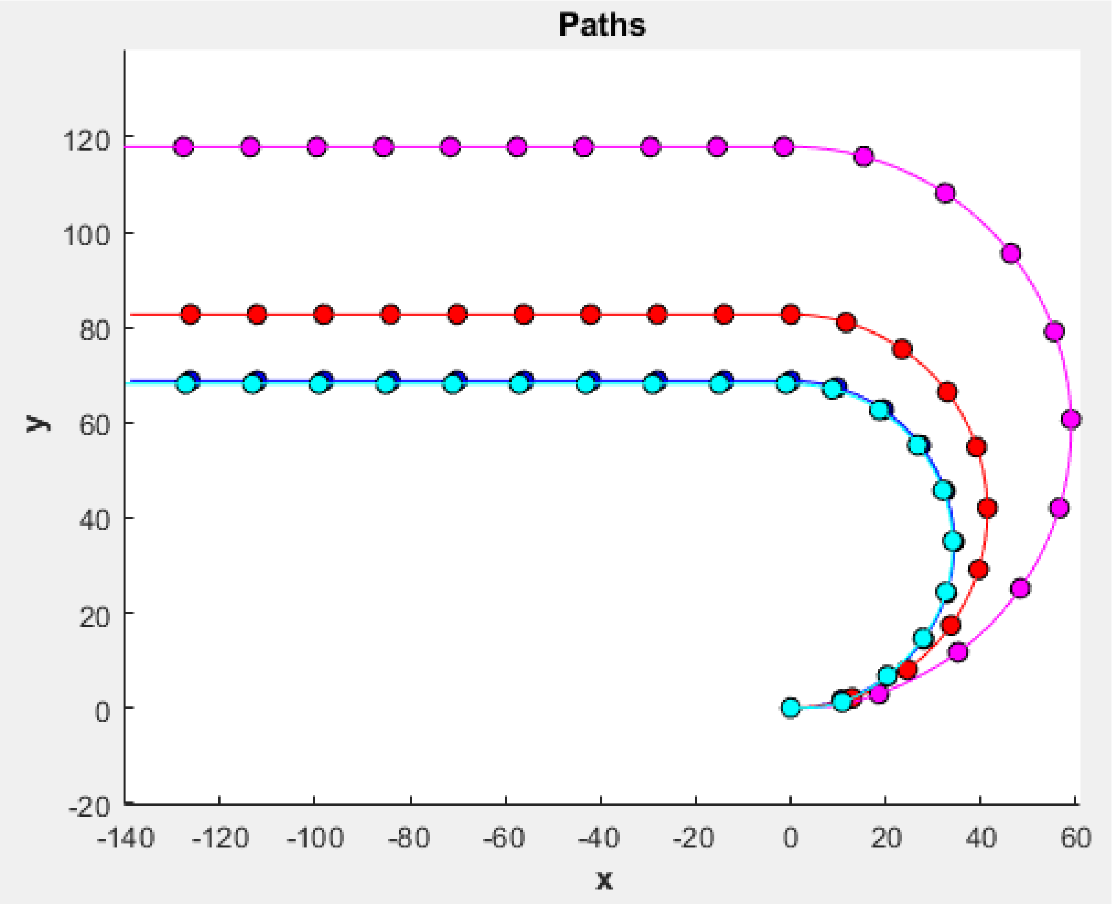
References

1. Active Torque Vectoring Systems for Electric Drive Vehicles, Main Thesis, Martin Mondek, Czech Technical University in Prague.
2. A Torque Vectoring Strategy for Improving the Performance of a Rear Wheel Drive Electric Vehicle, Jyotishman Ghosh et al, Turin, Italy, 2015.
3. The Automotive Chassis Volume 2 : System Design, Giancarlo Genta and Lorenzo Morello.
4. Vehicle Stability by Dean Karnopp.
5. Video Ref : https://www.youtube.com/watch?v=AgqWcsivlAA by Protean Electric In_Wheel Drive Motors

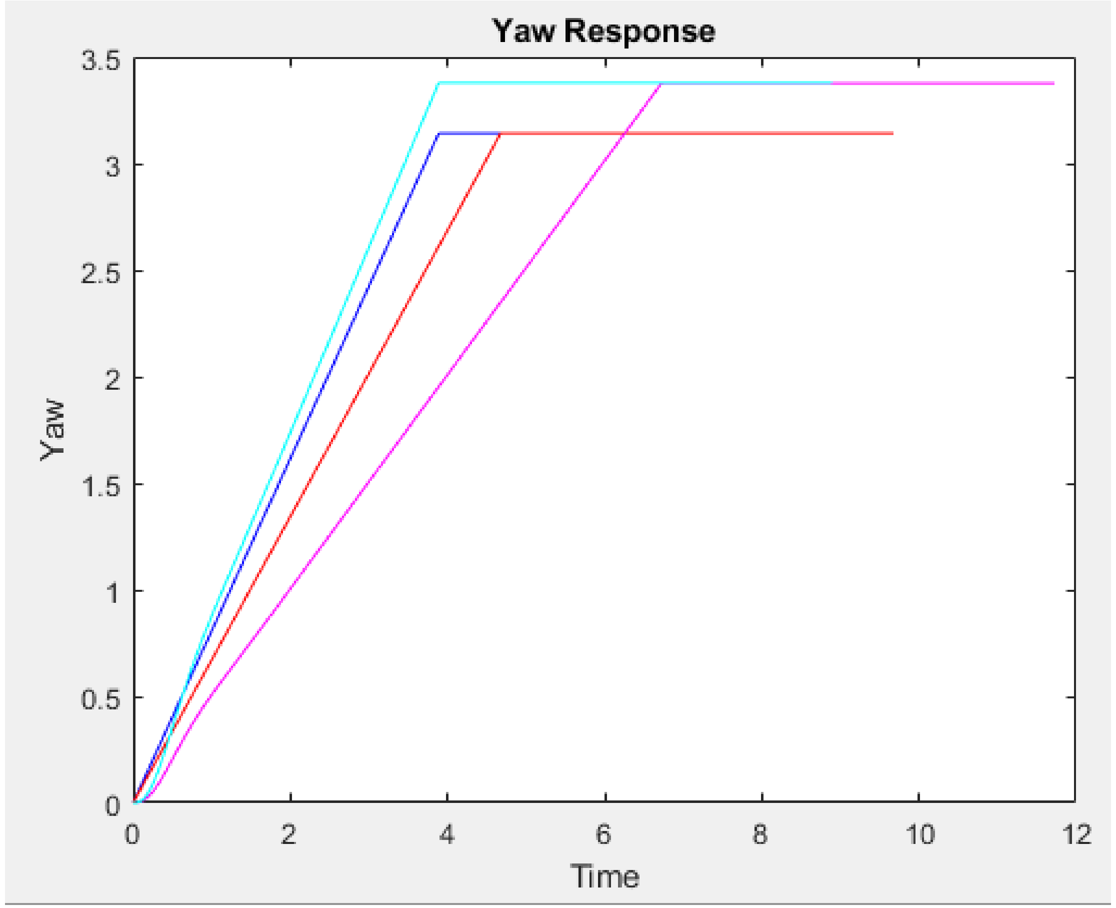
Dankeschön

Secondary Slides

Corrected Path of Vehicle – at 100 kmph

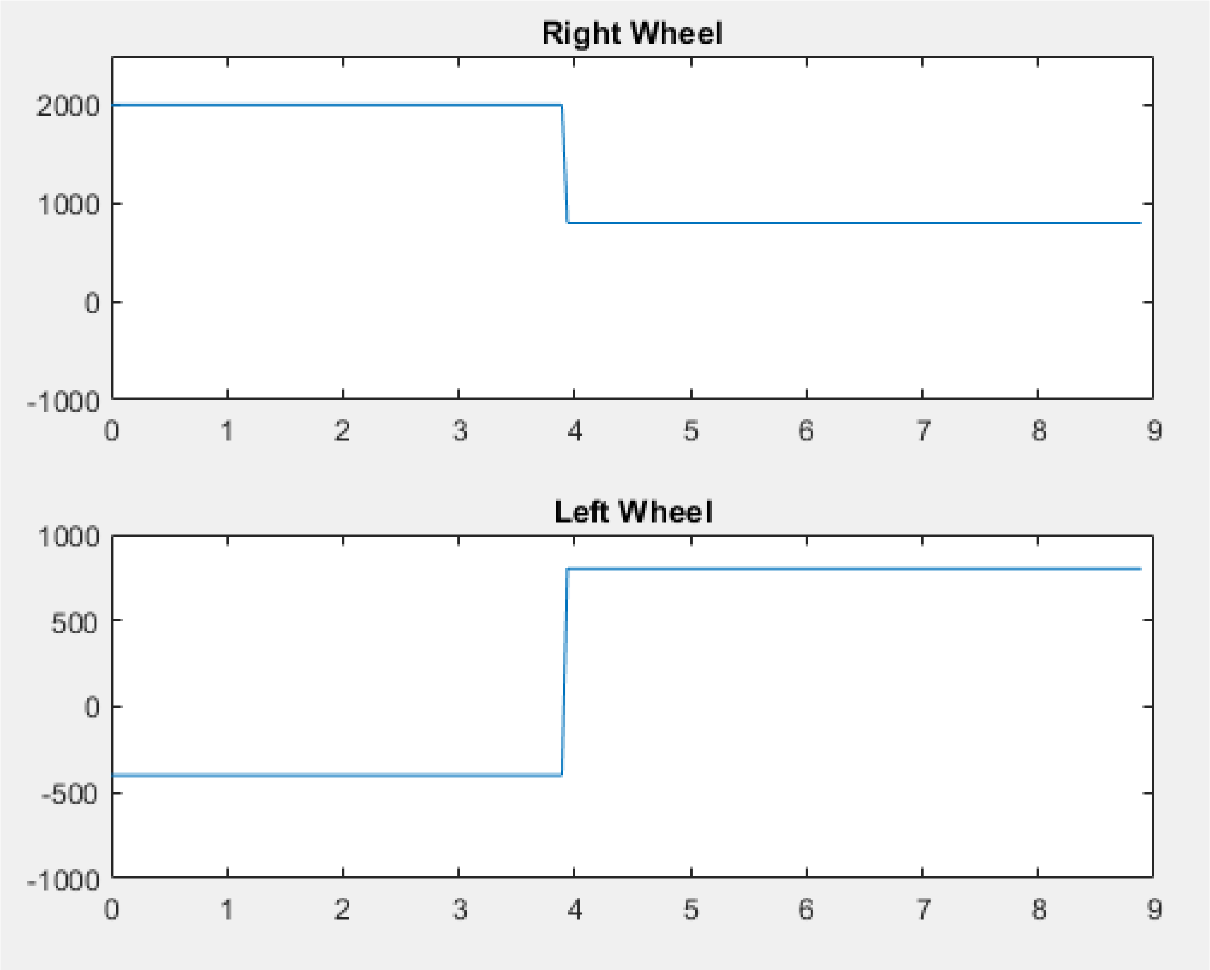


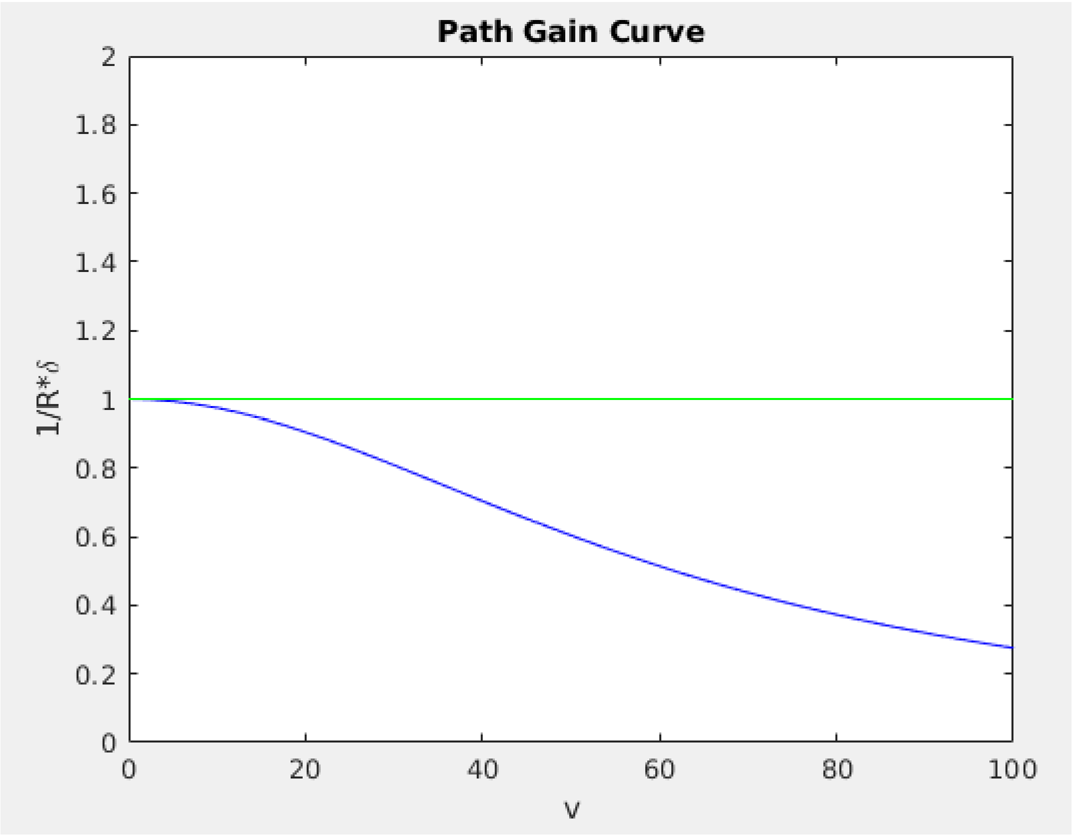
Path Followed by the Vehicle



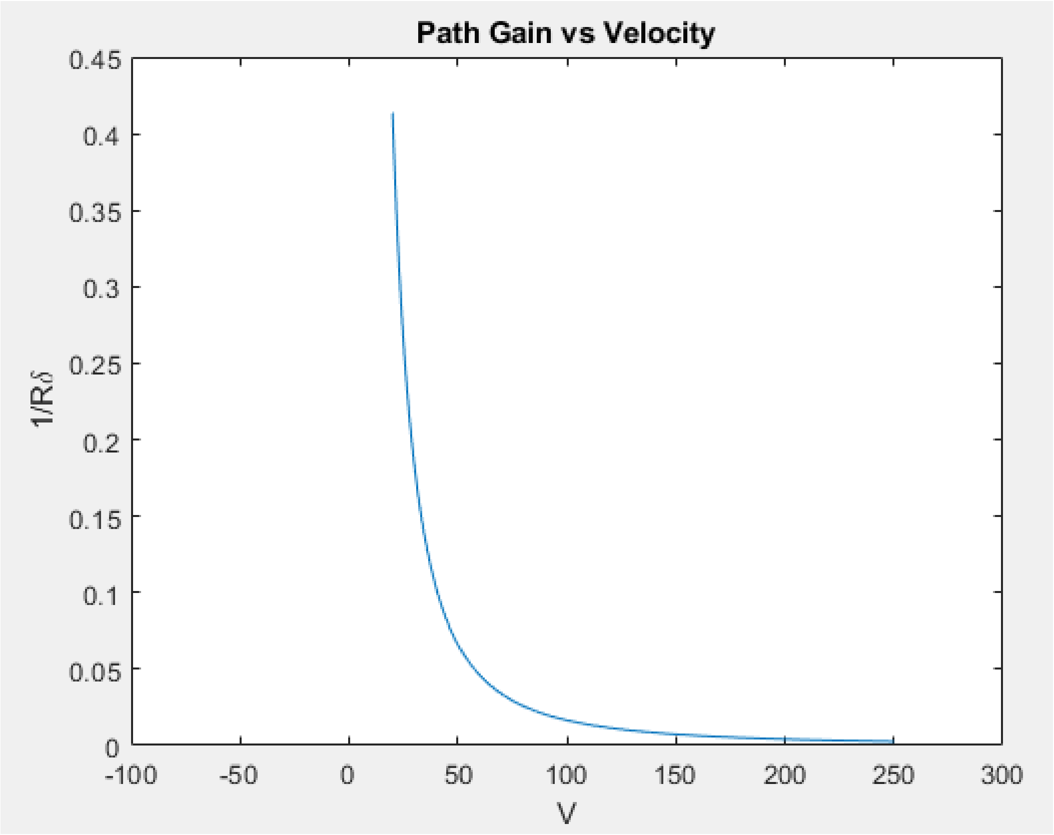
Yaw Response of the Vehicle

Distributed Torque to Left and Right Wheels at 100 kmph

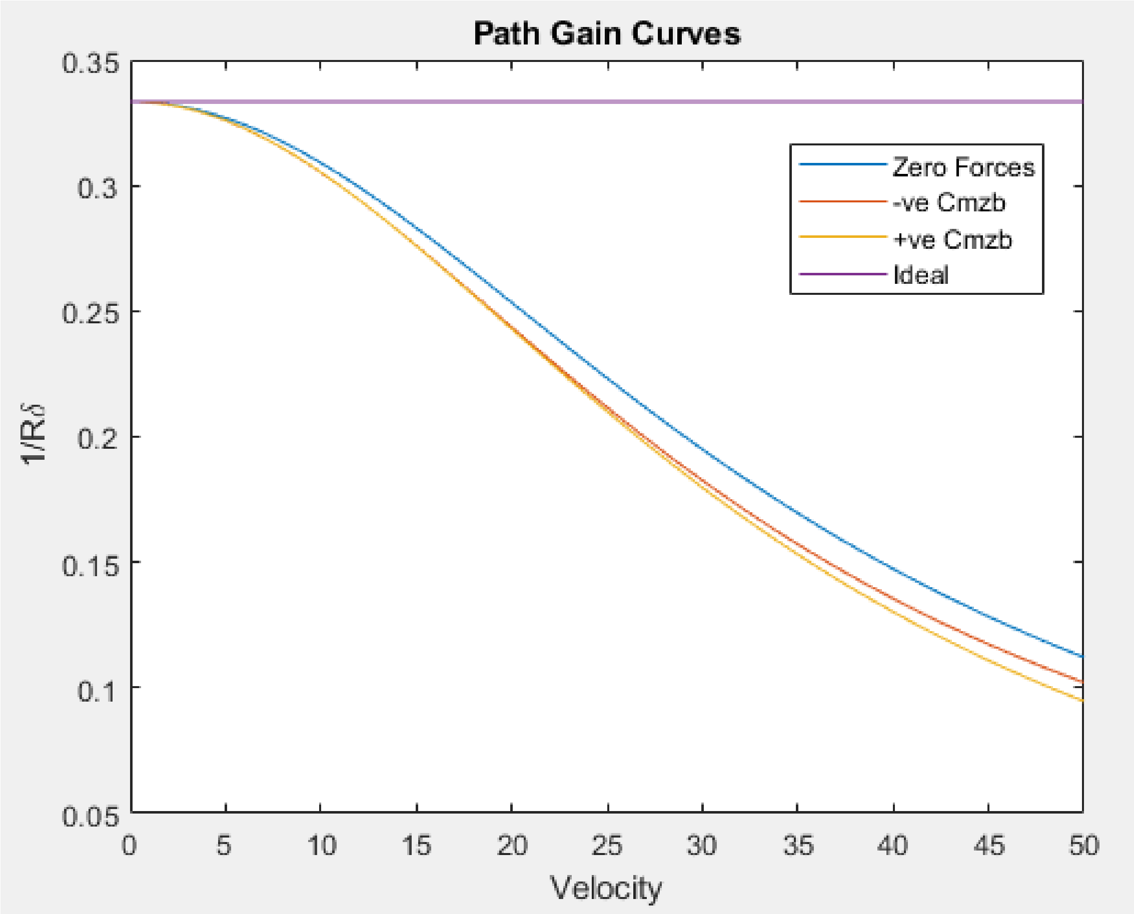




Path Gain Curve
2
1.8
1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
1/R*δ
0
20
40
60
80
100
v



Path Gain vs Velocity
0.45
0.4
0.35
0.3
0.25
0.2
0.15
0.1
0.05
0
1/Rδ
-100
-50
0
50
100
150
200
250
300
V



The path gains were plotted for the condition with no aerodynamic forces and aligning torques, Condition with positive $C_{mz,\beta}$ and also for the Condition with negative $C_{mz,\beta}$ and finally the ideal kinematic path gain has also been plotted for a Car track length of 3 m.

Why can a monotrack model be used?

- $\alpha_i = \beta + \frac{x_i}{V} r - \delta_i$
- There is no y value dependence of the side slip angle of the wheels which implies that the side slips are the same for the two wheels of the same axle, neglecting the difference in the steering angles of the two wheels.
- Side slips are small, so constant cornering stiffness is assumed C0, linear region in the slip-mu curve.

Toe in and load transfer

- Nor is allowance made for toe in and transversal load transfer. If the dependence of the cornering stiffness were linear with the load Fz, this would be correct since the increase of cornering stiffness of the more loaded wheel would exactly compensate for the decrease of the other wheel. As this is not exactly the case, the load transfer causes a decrease of the cornering stiffness of each axle, but this effect is usually considered negligible, at least for lateral accelerations lower than 0.5g.
- Toe in causes in increase of the cornering stiffness of the axle if it is positive, a decrease if it is negative.

Aerodynamic Yawing Moment

- The aerodynamic yaw moment produces a strong effect. If the derivative of Cz wrt beta is negative, the effect is increasing oversteer or decreasing understeer, at increasing speed. If a critical speed exists, such an aerodynamic effect lowers it and has an overall destabilising effect, increasing with the absolute value of Cmz,beta. Opposite occurs if Cmz,beta is positive.

- Considering only cornering forces at steady state conditions

$$\frac{mV^2}{R}\cos(\beta) = \Sigma F_{x_i}\sin(\delta_i) + \Sigma F_{y_i}\cos(\delta_i)$$

$$\Sigma \mathrm{F}_{\mathrm{x_i}}\sin(\delta_i)x_i + \Sigma F_{y_i}\cos(\delta_i)\,x_i = 0$$

$$u = V cos(\beta) \approx V; v = V sin(\beta) \approx V\beta$$

Feedforward Control System

- In this a control the output is known and assumed to be unchanging. A feedforward signal is input to the system and no output is fed back to any controller. Therefore it does not consider the response of the system which is a sort of a passive system. The error between the desired parameter and the response is not considered in such systems to modify the system response.