Lateral Control of Autonomous Vehicle

*Abstract*—Autonomous vehicles and driver-assistance features have become increasingly more common for increased road safety over the past 5 years. In this paper, lateral control of vehicle has been modelled as a MIMO problem and PID, and controllers have been designed and tradeoffs between the controlled designs have been discussed.

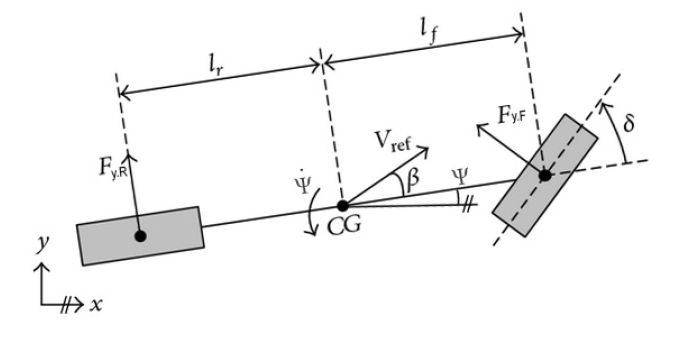
Keywords— MIMO, PID,,

# Introduction

In the recent times the automobile design in the industry is towards the direction of using more electronics, controllers for the purpose of improving the functionality and robustness of the system. The area in the vehicle design it is particularly affecting the most is active safety, which in vehicle design quite effectively complements the passive safety. Structural integrity of vehicle components is the purpose of passive safety. While on the other hand purpose of active safety is primarily to facilitate improved vehicle control to the driver and stability of the vehicle especially in emergency situations to prevent accidents for example which might be encountered while driving on the slippery parts of the road. In a study conducted from 2005 to 2007, which studied over 5000 crashes, concluded that 11% of the accidents happened because vehicles failed to stay in the proper lane and 22% of the crashes were a result of errors made in lane changing. Robust lateral control as an active safety for lane keeping and lane changing as level 1 autonomy in vehicles has a huge potential to reduce the road crashes for increase the road safety.

# Vehicle lateral dynamics model

In the analysis, kinematic model is used for lateral control of vehicle dynamics. For simplifying the analysis, the vehicle motion is assumed to be planar. For the purpose of neglecting the load transfer during the vehicle motion, the direction of center of gravity is taken as along the one projected into the paper. By the planar assumptions, we require one rotation and one translation for state estimation of the vehicle. The co-ordinate system used for the analysis is shown in figure 1. The x-axis of the vehicle points in the front of the vehicle, the y-axis towards the right hand side from the perspective of the driver and z-axis by right hand rule points towards the ground.



Longitudinal velocity of the vehicle is assumed to be constant for the analysis. The angle between the velocity vector and the front of the vehicle i.e. the x-axis is defined as the vehicle sideslip angle,

where is the velocity in x-direction and is the velocity in y-direction. The tire forces in longitudinal and lateral direction were computed independently for simpler implementation and representation of Pacejka’s ’Magic’ formula. The simplified form is,

where B,C,D and E are the parameters which give the shape of the characteristics of the tire, is lateral tire force and is sideslip angle of the tire. In the analysis, linear model of the tire is used,

where is tire force in the x-direction and is tire force in the x-direction, is tire slip, is tire slip coefficient

and is tire sideslip coefficient. When the sideslip angle is close to zero, the model gives accurate results. Non-linear behavior of the tire is not included in the model.

By balancing the forces in x and y direction and moments in the z direction at equilibrium we get the following equation,

In these equations, m is the mass of the vehicle, is the yaw rate of the vehicle, and are the distances between the center of gravity of the vehicle and the front and rear tire respectively, is moment of inertia of the vehicle about the z-axis, is the sideslip angle, is the steering angle of the front wheel, and are the forces in the x-direction i.e. longitudinal direction of front and rear tire respectively, similarly and are the forces in the y-direction i.e. lateral direction of front and rear tire respectively.

The tire forces are defined using the tire model. Sideslip angles of tires are significantly impacted by tire position and steering angle,

Where and are the front and rear tire sideslip angles respectively. The equation for the tire slip angles can be written as,

With the assumption of small steering angle and sideslip angles the force and moment equilibrium vehicle differential equations can be written as,

These equations can be put in state space form with lateral velocity and vehicle yaw rate as state variables,

= +

Where the brake steer force and brake steer moment is,

,

T is the vehicle track. The state space model has two inputs and two outputs.

|  |  |
| --- | --- |
| Weight | 1000kg |
| Vehicle speed | 20 m/s |
| Moment of inertia | 1500 kg |
| Vehicle track | 1.5m |
| Distance of front wheel and CG | 1m |
| Distance of rear wheel and CG | 1.5m |
| Nominal cornering stiffness of front tire | 55000 N/rad |
| Nominal cornering stiffness of rear tire | 45000 N/rad |

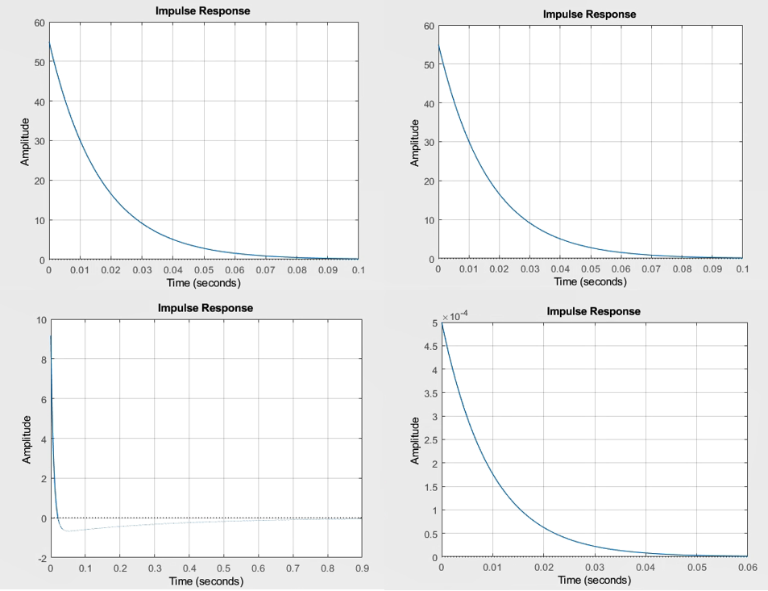
For the controller design in the state space form, the C matrix is taken as identity and D matrix are the disturbances.

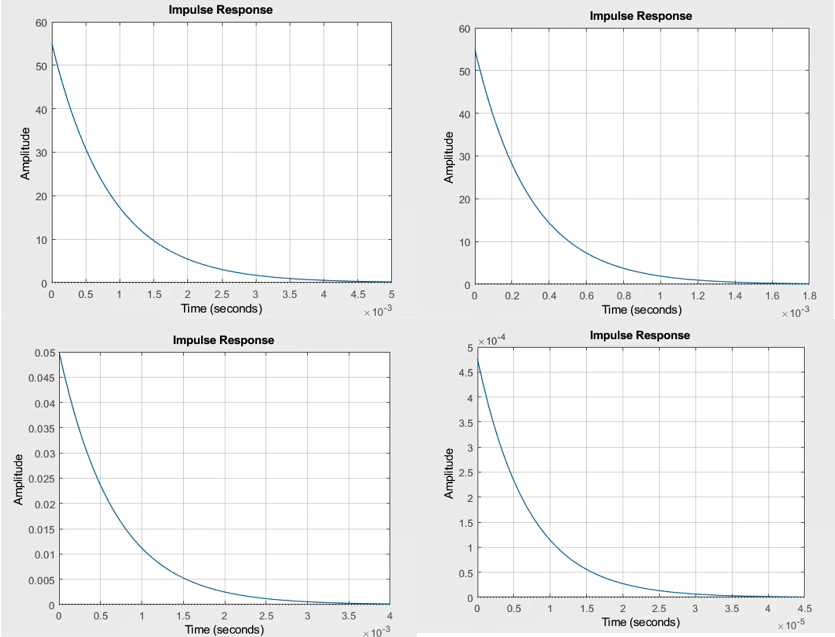
# PID Controller design

The controller design using PID is a relatively extreme form of a phase lead-lag compensator design having one of the poles at origin and the other pole tending to infinity. The PI and the PD controller designs similar to PID are extreme cases of phase-lag and phase-lead compensators, respectively. A PID control has the following standard form of transfer function,

Where, and are proportional, integral and derivative gains respectively.

The procedure used to design the PID control was, first the proportional gain was increased to an extent when he oscillations in the system increased to a considerable extent, then the derivative gain was increased to damp those oscillations and then the integral gain was increased to remove the steady state errors. The results obtained are as follows,

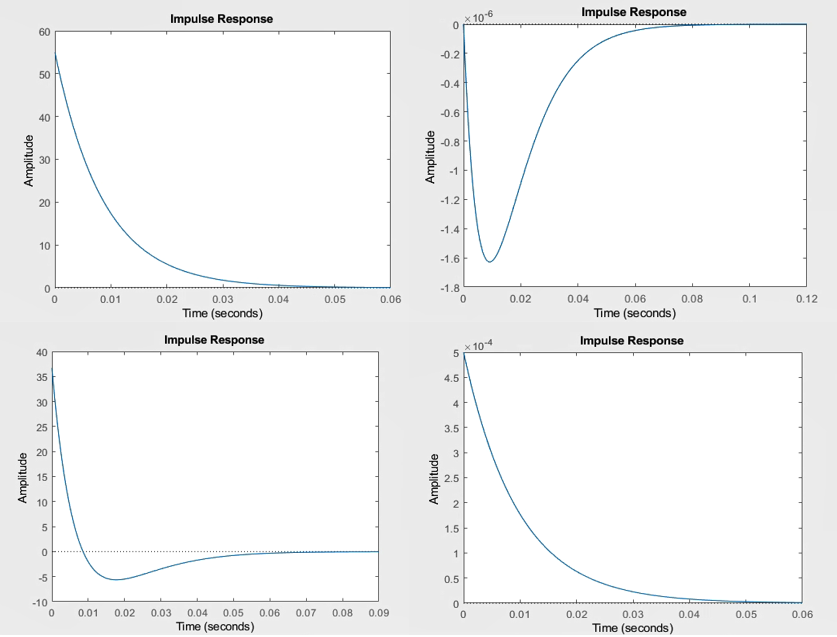




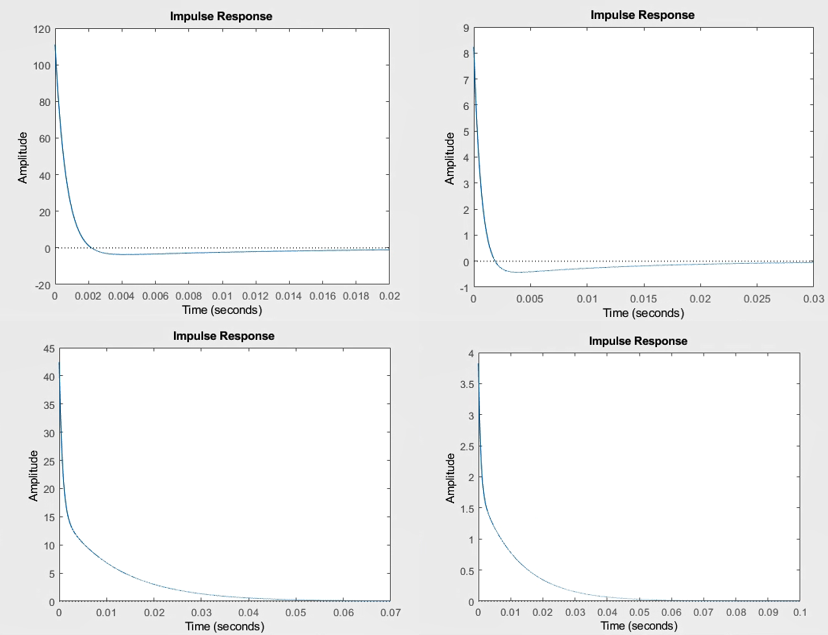
As we can note, PID controller works very well. The system settles down with the controller more than 20 times faster as compared to the uncontrolled system.

# controller design

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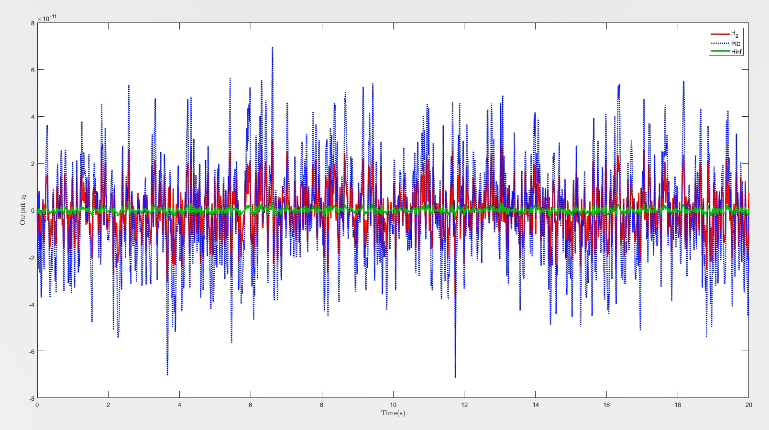
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