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NATIONAL INSTITUTE OF TECHNOLOGY CALICUT

NIT Campus P.O, Calicut, Kerala, India, 673601

DEPARTMENT OF MECHANICAL ENGINEERING

Simulation of

"Implementation of Torque Vectoring in Electric Cars for Enhanced Cornering Performance"

Submitted in partial fulfillment for the award of degree of

MASTER OF TECHNOLOGY

In

MACHINE DESIGN ENGINEERING

2024-2025

Submitted by

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Certificate

This is to certify that the project report entitled "**Implementation of Torque Vectoring in Electric Cars for Enhanced Cornering Performance**" submitted by Mr. SAREPALLI RAMHARAN (M240549ME), National Institute of Technology Calicut, towards the partial fulfilment of the requirements for the award of the Degree of Master of Technology in Machine Design Engineering, is a Bonafide record of the work carried out by him under my supervision and guidance.

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DECLARATION

I hereby declare that the entire work embodied in the project report entitled '**Implementation of Torque Vectoring in Electric Cars for Enhanced Cornering Performance**' has been independently carried out by me under the guidance of **Dr. Vineesh K P**, Assistant Professor, Department of Mechanical Engineering, National Institute of Technology, Calicut, in partial fulfilment of the requirements for the award of master of technology National Institute of Technology, Calicut.

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ABSTRACT

This project explores the implementation and evaluation of torque vectoring control for enhancing the handling ad stability of a vehicle. The project utilizes a vehicle dynamics model based on a bicycle representation to simulate lateral motion and yaw behaviour. Two simulation scenarios are investigated: a baseline scenario without torque vectoring and a controlled scenario employing a PID-based torque vectoring strategy.

The baseline simulation considers a constant steering input, analysing the vehicle's lateral velocity, yaw rate, and trajectory. The torque vectoring model introduces a PID controller to minimize yaw rate error by distributing torque between the wheels, enhancing vehicle stability and responsiveness. Results compare the trajectory and yaw rate of the baseline and controlled systems, highlighting the effectiveness of torque vectoring in achieving improved handling characteristics under dynamic conditions.

The torque vectoring system leverages the principles of active yaw control to counteract understeer and improve cornering performance. By adjusting the torque distribution between the left and right wheels based on the desired and actual yaw rates, the system ensures that the vehicle follows the intended path more accurately. The PID controller fine-tunes the torque vectoring input using proportional, integral, and derivative gains, effectively reducing yaw rate errors over time. The comparative analysis of the baseline and torque vectoring systems demonstrates significant improvements in vehicle stability, path accuracy, and overall handling, validating the effectiveness of the proposed control strategy in real-world applications.

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CHAPTER 1

INTRODUCTION

1.1 Overview

The latest spread of the electromobility in the vehicle industry promises to bring mostly the local improvement of the air pollution and lower operating costs of the vehicle. The electromobility brings not only these very often mentioned properties, but also other advantages such as an independent electric motor for each wheel. The different structure of the powertrain in electrical or hybrid vehicle offers new possibilities for independent control of each wheel. These opportunities require development of better control algorithms for vehicle stabilization or modification of the vehicle dynamics and behaviour. The modelling and simulation of the vehicle dynamics is very wide discipline. The high-fidelity dynamics models of the vehicle motion are often used for the simulations of the exact vehicle behaviour. However, simplified mathematical models are sufficient for fundamental study of the vehicle dynamics.

1.2 Electric Vehicles

Electric vehicles (EVs) have emerged as a pivotal innovation in the automotive industry, representing a significant shift from traditional internal combustion engine (ICE) vehicles towards more sustainable and environmentally friendly transportation solutions. Unlike ICE vehicles that rely on fossil fuels, EVs are powered by electric motors and energy stored in rechargeable batteries, offering a cleaner alternative that reduces greenhouse gas emissions and dependence on non-renewable energy sources. This transition is driven by increasing environmental concerns, advancements in battery technology, and supportive government policies aimed at mitigating climate change and promoting sustainable development.

The rapid advancement in EV technology has been fuelled by significant improvements in battery efficiency, energy density, and cost reduction, making electric mobility more accessible and practical for a broader range of consumers. Modern EVs benefit from innovations such as lithium-ion batteries, which provide longer driving ranges and shorter charging times, addressing some of the primary concerns associated with earlier electric models. Additionally, the integration of regenerative braking systems enhances energy efficiency by recovering and storing energy that would otherwise be lost during braking. These technological strides have not only improved the performance and reliability of EVs but have also made them increasingly competitive with their ICE counterparts in terms of cost and functionality.

Beyond environmental benefits, EVs offer numerous advantages that contribute to their growing popularity. They provide a quieter and smoother driving experience due to the absence of engine noise and vibrations, enhancing passenger comfort. The lower maintenance requirements of electric motors, which have fewer moving parts compared to traditional engines, result in reduced upkeep costs and increased vehicle longevity. Furthermore, the rise of smart grid technology and the expansion of charging infrastructure have made EVs more convenient and user-friendly, encouraging wider adoption among consumers and businesses alike.

Despite the promising advancements, the widespread adoption of electric vehicles faces several challenges that need to be addressed to realize their full potential. Key obstacles include the need for extensive charging infrastructure, the environmental impact of battery production and disposal, and the current limitations in battery energy storage capacity. Additionally, ensuring the sustainability and ethical sourcing of materials used in batteries, such as lithium and cobalt, remains a critical concern. Ongoing research and development efforts are focused on overcoming these challenges through innovations in battery chemistry, recycling technologies, and the integration of renewable energy sources to power EVs. As the industry continues to evolve, electric vehicles are poised to play a crucial role in shaping the future of transportation, contributing to a more sustainable and resilient global economy.

.Electronic stability program

(ESP) is a control system, which uses the vehicle braking system as the control actuator for vehicle stabilization.This system usually breaks one wheel to achieve higher or lower vehicle yaw rate depending on the vehicle understeer or oversteer behaviour.This control action should provide the vehicle stability during all critical situations. An example of the ESP control of the understeer and oversteer vehicles are presented in figure 1.1 and 1.2.

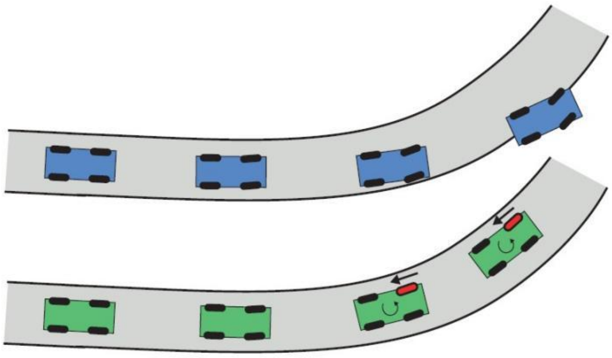

Figure 1.1: The comparison of the understeer vehicle manoeuvre without and with ESP control systems. The red wheel is slowed down by the control system and creates additional vehicle yaw moment, which stabilizes the vehicle. The term understeer means the tendency of the vehicle to steer less than the driver wants to

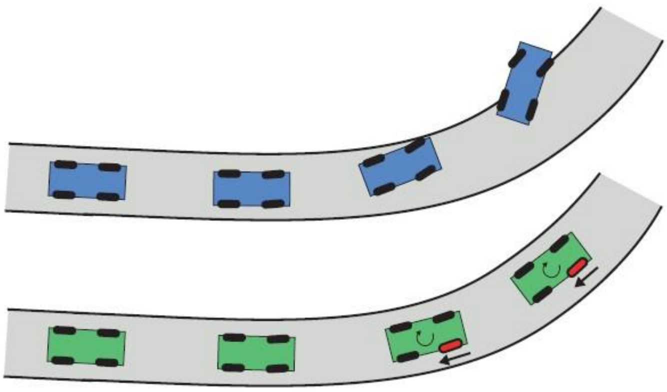

Figure 1.2: The comparison of the oversteer vehicle manoeuvre without and with ESP control systems. The control system slows down the red wheel and creates additional vehicle yaw moment, which stabilizes the vehicle. The term oversteer means the tendency of the vehicle to steer more than the driver wants to.

1.3 Torque vectoring

The primary motivation for developing the torque vectoring system is to control the vehicle stability and lateral dynamics, and improve the vehicle manoeuvrability to lower the driver's effort. All these goals are also goals of the electronic stability system. ESP control system uses breaking of selected
wheels to control the vehicle stability, but the torque vectoring system uses the difference of torques for the same purposes.

However, the torque vectoring system is used in different situations than the electronic stability program. The torque vectoring system is used to modify the lateral vehicle dynamics, improve the stability and manoeuvrability of the vehicle. These situations are usually not critical. On the other hand, the electronic stability system controls the vehicle stability in dangerous situations, where some severe damage might be caused. If the ESP system starts to control the vehicle, the torque vectoring system should be turned off.

The vehicle yaw moment is stabilized by additional yaw moment created by the difference in the vehicle torque distribution. The figure 1.4 displays the difference between rear wheel torque and the generated yaw moment. This yaw moment then forces the vehicle dynamics to turn more, which results to increment of the vehicle yaw rate.

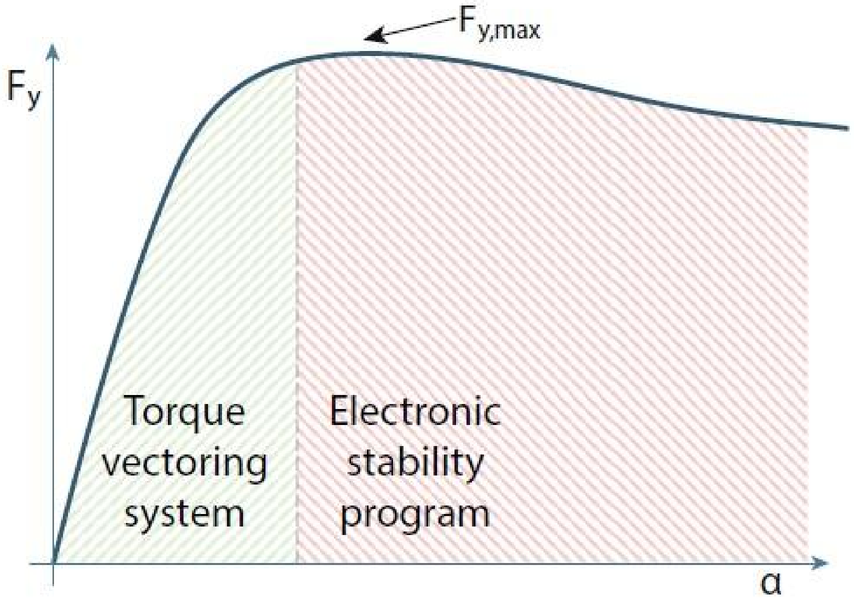

Figure 1.3: Area of lateral side-slip characteristics, where torque vectoring is used to control and modify the vehicle dynamics.

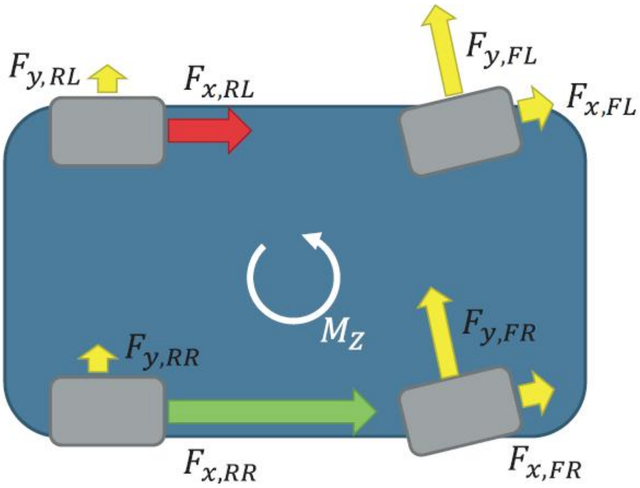

Figure 1.4: The difference in the rear wheel torque creates additional vehicle yaw moment, which can be used for the control of the vehicle stability.

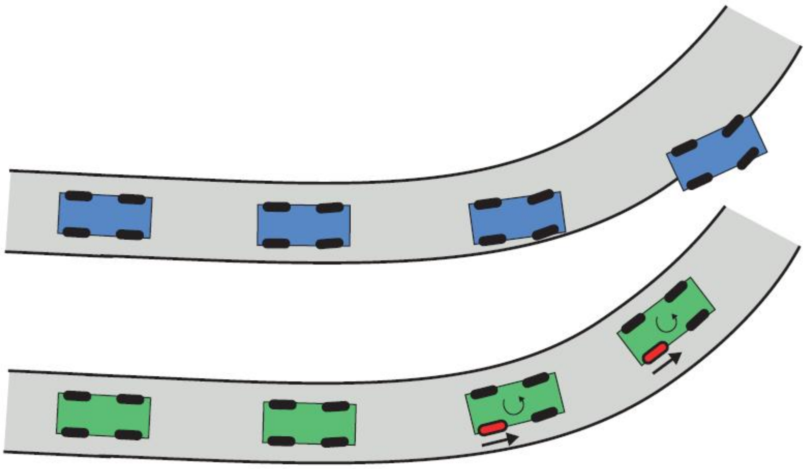

Figure 1.5: The comparison of the understeer vehicle manoeuvre without and with torque vectoring control system. The control system increases the torque on the wheel with the red colour. This action creates additional vehicle yaw moment, which stabilizes the vehicle.

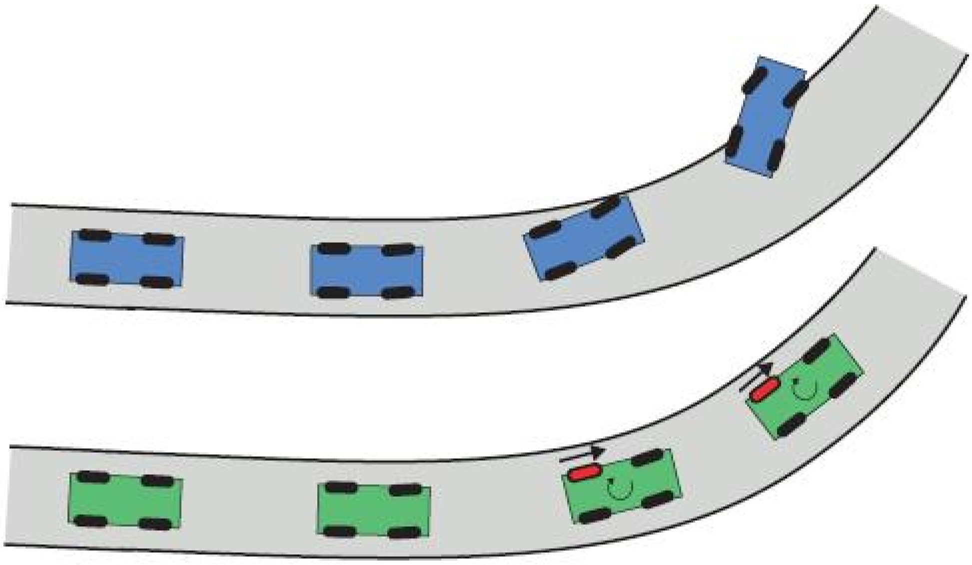

Figure 1.6: The comparison of the oversteer vehicle manoeuvre without and with torque vectoring control system. The control system increases the torque on the wheel with the red colour. This action creates additional vehicle yaw moment, which stabilizes the vehicle.

1.4 Torque vectoring Application scenarios

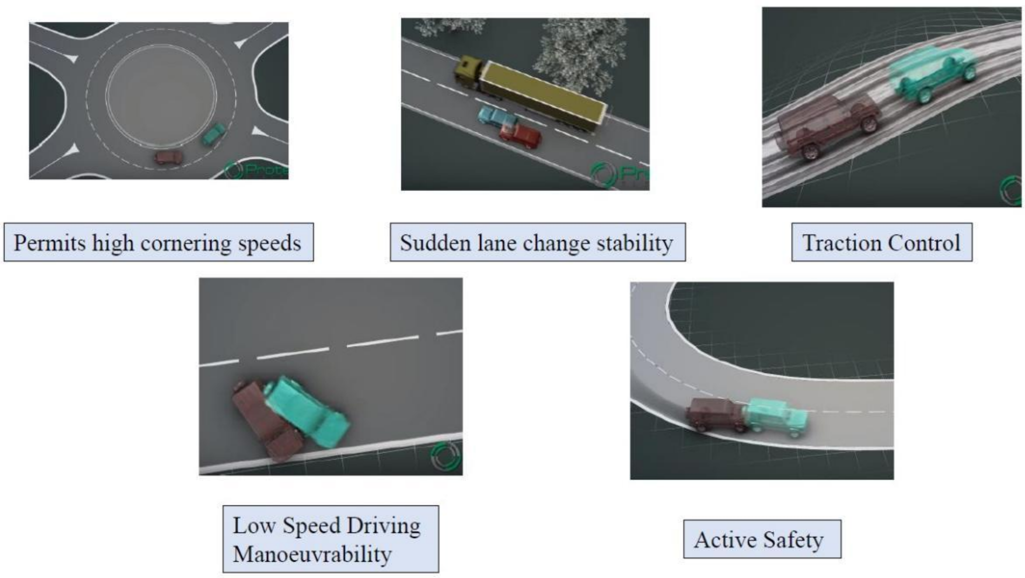

Figure 1.7: Torque vectoring application scenarios

CHAPTER 2

OBJECTIVES

Objectives of the Project: The primary objectives of this project, which combines the baseline simulation and the implementation of a torque vectoring system with PID control, are as follows:

1. Simulate Vehicle Dynamics:

1. Develop a vehicle dynamics model using the **bicycle model** to understand lateral and yaw behaviors under a constant steering input

2.Analyze how key parameters such as cornering stiffness, vehicle mass, and axle distances influence the trajectory and stability.

2. Evaluate Baseline Behavior:

1. Simulate the vehicle's natural path and yaw rate without any active control mechanisms.

2. Observe the handling characteristics, including understeer or oversteer tendencies, and use these results as a baseline for comparison.

3. Implement Torque Vectoring:

1. Introduce a **torque vectoring control system** to actively influence yaw dynamics and improve vehicle stability.

2. Use a PID controller to calculate corrective torque inputs based on yaw rate errors, aligning the vehicle's yaw rate with the desired value.

4.Improve Stability and Handling:

1. Assess the effectiveness of torque vectoring in reducing yaw rate error and stabilizing the vehicle during cornering.
2. Evaluate how the controlled dynamics enhance the vehicle's trajectory compared to the baseline.

5. Performance comparison

1. Compare the results of the baseline and controlled systems to demonstrate the impact of torque vectoring on path accuracy, yaw rate behavior, and overall handling stability.
2. Plot and analyze key performance metrics such as trajectory and yaw rate over time.

6.Analyze Controller Effectiveness:

1. Evaluate the performance of the PID controller by examining its proportional, integral, and derivative components.
2. Analyze how the controller responds to dynamic changes in yaw rate error and whether it effectively achieves the desired stability.

CHAPTER 3

SPECIFICATIONS

3.1 Specifications of the prototype vehicle

Term	symbol	Value	Units
Mass of the vehicle	m	356	kg
Moment of inertia	I_{zz}	120	m^4
Front wheel base	l_f	0.873	m
Rear wheel base	l_r	0.717	m
Track width	tf	1.3	m
Steering angle	δ	0.8726	rad
Cornering stiffness at the rear wheel	Cy,r	21429	N/rad
Cornering stiffness at front wheel	Cy,f	15714	N/rad
Gear ratio	G_r	4.4	---
Radius of the wheel	Rw	0.265	m
Yaw rate	ψ	---	rad/s
Longitudinal velocity	vx_o	---	m/s
Lateral velocity	vy	---	m/s
Vehicle velocity	VCG	---	m/s
Torque difference between rear wheel	ΔT	---	Nm

3.2 Linearized Model:

Assumptions:

1. Velocity of the vehicle's center of gravity is considered constant along the longitude of its trajectory.
2. All lifting, rolling and pitching motion will be neglected.
3. The mass of the vehicle is assumed to be at the center of gravity
4. Front and rear tires will be represented as one single tire, one each axle.
5. Aligning torque resulting from the side slip angle will be neglected.
6. The wheel-load distribution between front and rear axles is assumed to be constant.
7. The longitudinal forces on the tires, resulting from the assumption of a constant longitudinal velocity, will be neglected.

3.3 Mathematical model

The vehicle's lateral dynamics and yaw motion are modelled using a **bicycle model**. Below are the equations that describe the system dynamics.

1. **Lateral Velocity ($\dot{v_y}$)**

The lateral velocity is governed by the following equation:

$$\dot{v_y} = \frac{F_{yf} + F_{yr}}{m} - \dot{\psi} \cdot v_x$$

Where:

- F_{yf} and F_{yr} are the lateral forces at the front and rear tires

$$F_{yf} = C_{y,f} \cdot \alpha_f \quad , \; F_{yr} = C_{y,r} \cdot \alpha_r$$

2. **Slip angles**

- α_f is the front slip angle
- α_r is the rear slip angle

$$\alpha_f = \delta - arctan\left(\frac{v_y + l_f \cdot \dot{\psi}}{v_x}\right), \;\; \alpha_r = -\,arctan\left(\frac{v_y - l_r \,.\, \dot{\psi}}{v_x}\right)$$

3. **Yaw Rate ($\ddot{\psi}$)**
   The yaw rate dynamics are expressed as:

$$\ddot{\psi} = \frac{l_f \cdot F_{yf} - l_r \cdot F_{yr}}{Izz}$$

In the case of **torque vectoring**, an additional torque (ΔT) introduces a yaw moment:

$$\ddot{\psi} = \frac{l_f \cdot F_{yf} - l_r \cdot F_{yr}}{Izz} + \frac{\Delta T}{0.05 \cdot Izz}$$

4. **Position Updates (x, y)**

The global position of the vehicle (x, y) is updated using the following kinematic equations:

$$\dot{x} = v_x \cdot cos(\psi) - v_y \cdot sin(\psi)$$

$$\dot{y} = v_x \cdot sin(\psi) + v_y \cdot cos(\psi)$$

These are integrated over time to determine the path of the vehicle.

5. **Desired Yaw Rate for Torque Vectoring**

The desired yaw rate is derived using a steady-state steering model:

$$\dot{\psi}_{desired} = \frac{v_x}{R}$$

6. **Turning radius: (R)**

$$R = \frac{1}{\frac{\delta}{(l_r + l_f)\ + K_u\, .VCG^2}}$$

7. **Understeer gradient:**

$$Ku = \frac{lr.m}{C_{y,f}(l_r + l_f)} - \frac{l_f.m}{C_{y,r}(l_r + l_f)}$$

3.4 State-Space Representation

Using the above equations, the state-space representation of the system can be expressed as follows:

Without Torque Vectoring:

$$\begin{bmatrix} \dot{v}_y \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} -\dfrac{C_{y,f} + C_{y,r}}{mv_{x0}} & \dfrac{-l_f C_{y,f} + l_r C_{y,r}}{mv_{x0}} - v_{x0} \\ \dfrac{-l_f C_{y,f} + l_r C_{y,r}}{I_{zz} v_{x0}} & \dfrac{l_f^2 C_{y,f} + l_r^2 C_{y,r}}{I_{zz} v_{x0}} \end{bmatrix} \begin{bmatrix} v_y \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} \dfrac{C_{y,f}}{mv_{x0}} \\ \dfrac{l_f C_{y,f}}{I_{zz}} \end{bmatrix} \delta$$

3.4.1 Linear Model equations with additional yaw moment

The way the model is presented, the equations only generate lateral force by the steering angle input. The goal with the torque vectoring is to generate yaw moment based on controlling the torque longitudinal force) at the driven wheels. For this it will be necessary to introduce a new term Mz that will represent the additional yaw moment generated by the torque distribution.

$$\dot{\boldsymbol{x}} = \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}\boldsymbol{u}\mathbf{1} + \boldsymbol{C}\,\boldsymbol{u}\mathbf{2}$$

$\dot{\boldsymbol{x}} = \frac{\dot{v_y}}{\ddot{\psi}}$; **u_1 = Mz ;** **$u_2 = \boldsymbol{\delta}$**

$$\begin{bmatrix} \dot{v_y} \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} -\dfrac{C_{y,f}+C_{y,r}}{mv_{x0}} & \dfrac{-l_fC_{y,f}+l_rC_{y,r}}{mv_{x0}} - v_{x0} \\ \dfrac{-l_fC_{y,f}+l_rC_{y,r}}{I_{zz}v_{x0}} & -\dfrac{l_f^2C_{y,f}+l_r^2C_{y,r}}{I_{zz}v_{x0}} \end{bmatrix} \begin{bmatrix} v_y \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} 0 \\ \dfrac{1}{I_{zz}} \end{bmatrix} M_z + \begin{bmatrix} \dfrac{C_{y,f}}{mv_{x0}} \\ \dfrac{l_fC_{y,f}}{I_{zz}} \end{bmatrix} \delta$$

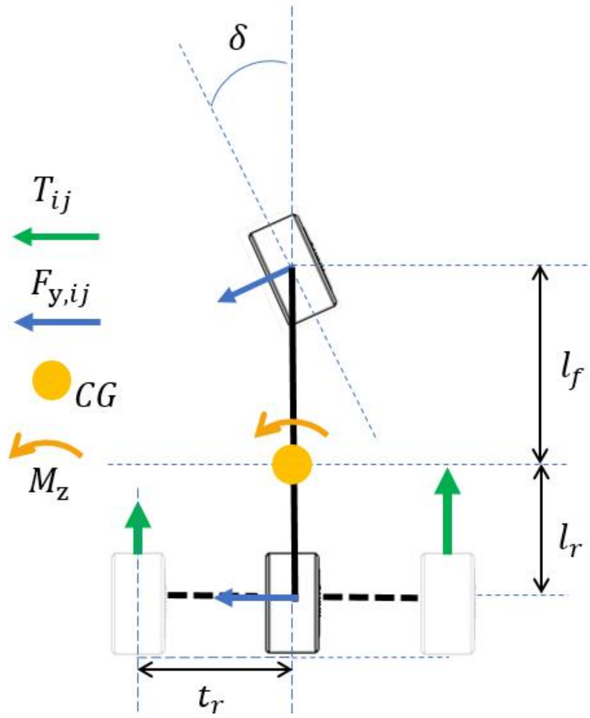

Fig 3.1: Vehicle's linear model with extension of the rear wheels in order to generate yaw moment based on the longitudinal force

The added momentum results from the difference between the left wheel torque Trl and the right wheel torque Trr. This difference multiplied by the half-track of the car constitutes the additional yaw momentum in the model. If the right wheel as more torque than the left wheel, the car will have a positive yaw momentum, thus turning to the left. If the opposite happens the car while have a negative momentum and will turn right.

$$Mz = \Delta T * tf = (Trr - Trl)tf$$

The torque at the wheel is not the same as the torque at the motor. Between the motor and the wheel there is a planetary gear set. This gear set amplifies the torque from the motor by a gear ratio, i.e. Tweel = Gr * Motor. Then the torque at the wheel has to be divided by the wheel radius Rw to obtain the force at the ground.

Table 3.1: Table with values and terms for the torque conversion

Term	symbol	Value	Units
Gear ratio	G_r	4.4	
Half the track of the car	t_r	0.65	m
Radius of the wheel	R_w	0.265	m

Putting together all this information in the above the yaw moment from the difference in torque is given by:

$$T = \frac{R}{2 * t_r * G_r} * M_z = \frac{0.265}{2 * 0.65 * 4.4} * M_z = 0.05 * M_z$$

$$\begin{bmatrix} \dot{v}_y \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} -\frac{C_{y,f} + C_{y,r}}{m v_{x0}} & \frac{-l_f C_{y,f} + l_r C_{y,r}}{m v_{x0}} - v_{x0} \\ \frac{-l_f C_{y,f} + l_r C_{y,r}}{I_{zz} v_{x0}} & \frac{l_f^2 C_{y,f} + l_r^2 C_{y,r}}{I_{zz} v_{x0}} \end{bmatrix} \begin{bmatrix} v_y \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} \frac{C_{y,f}}{m v_{x0}} \\ \frac{l_f C_{y,f}}{I_{zz}} \end{bmatrix} \delta + \begin{bmatrix} 0 \\ \frac{1}{0.05 * I_{zz}} \end{bmatrix} \Delta T$$

CHAPTER 4: METHODOLOGY

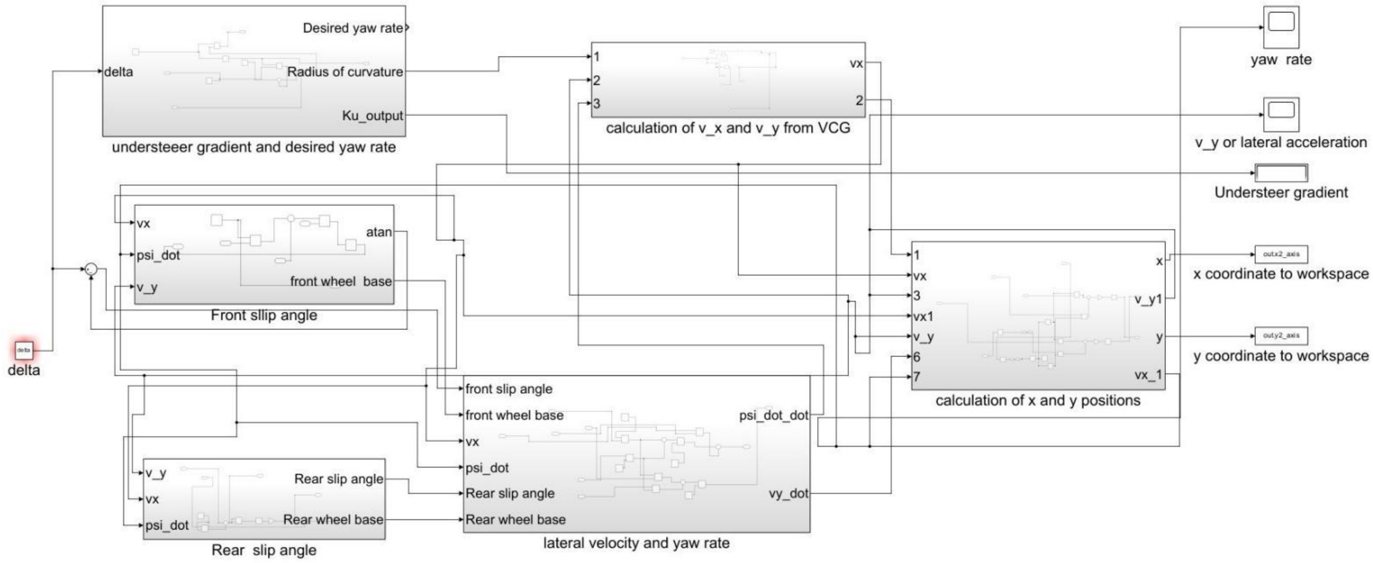

Fig 4.1 Simulink model without torque vectoring

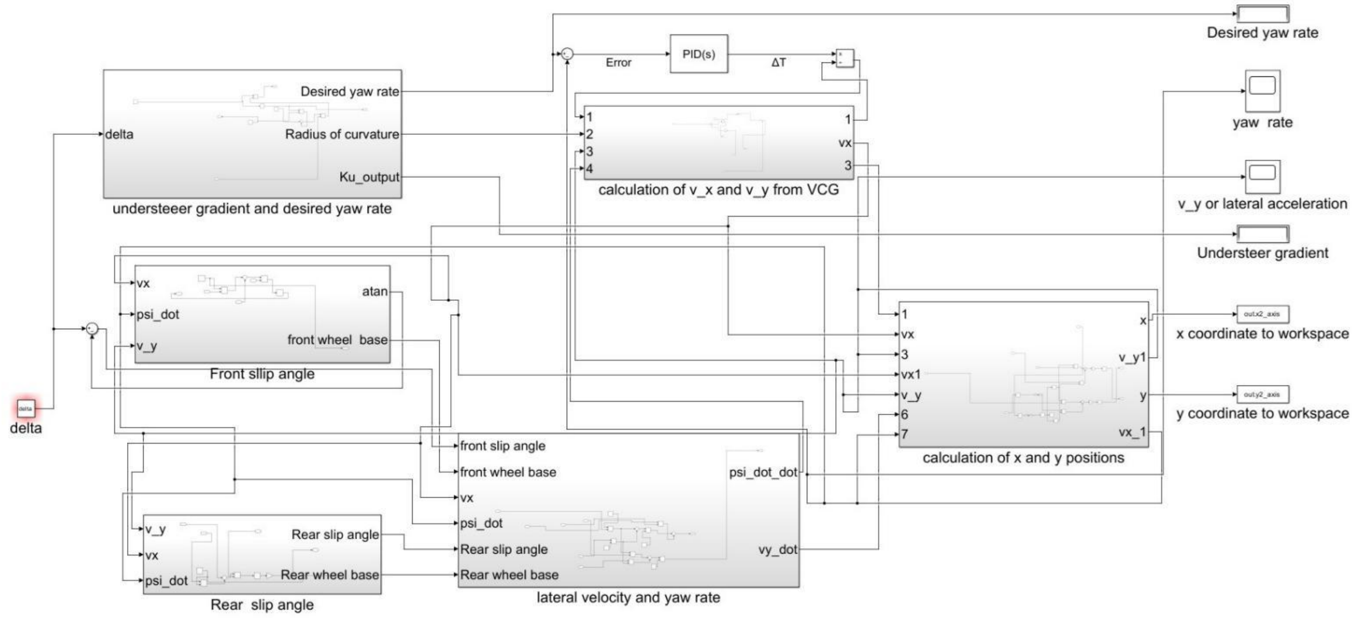

Fig 4.2 Simulink model with torque vectoring

Methodology for the Simulink Model (Simplified)

1. Vehicle Dynamics Subsystem

- Simulates the vehicle’s motion using a bicycle model.
- Takes the steering angle and torque vectoring input to calculate lateral velocity, yaw rate, and yaw angle.
- Outputs these parameters for position updates.

2. Tire Force Computation

- Calculates the slip angles and lateral forces acting on the tires based on the steering angle and vehicle states.
- Feeds the forces back to the dynamics subsystem to determine the vehicle’s behavior.

3. Torque Vectoring Control (PID Subsystem)

- Compares the actual yaw rate to the desired yaw rate to compute the control error.
- Uses a PID controller to calculate a corrective torque vectoring input.
- Use the inbuilt matlab PID tuning module to tune the Ki, Kp, Kd values of the PID controller.
- The torque vectoring input adjusts the yaw dynamics to enhance the vehicle's stability and control.

4. Kinematics Subsystem

- Computes the vehicle's trajectory by integrating lateral velocity and yaw angle over time.
- Outputs the x, y positions for trajectory plotting.

5. Simulation Framework

- The entire system is simulated over a defined time frame with inputs like constant vehicle speed and steering angle.
- The results include the vehicle’s path, yaw rate, and stability behavior.

7. Visualization

- Plots the vehicle's path and yaw rate over time to show improvements in stability and control when torque vectoring is applied.

CHAPTER 5 : RESULTS AND DISCUSSIONS

The following are the results obtained in the study correcting the path of the vehicle at different speeds and different steering angle

In MATLAB a two Simulink models are created with the derived mathematical model one without torque vectoring and with torque vectoring.

Path of the vehicle with steering angle 50 degrees and velocity 20 m/s :

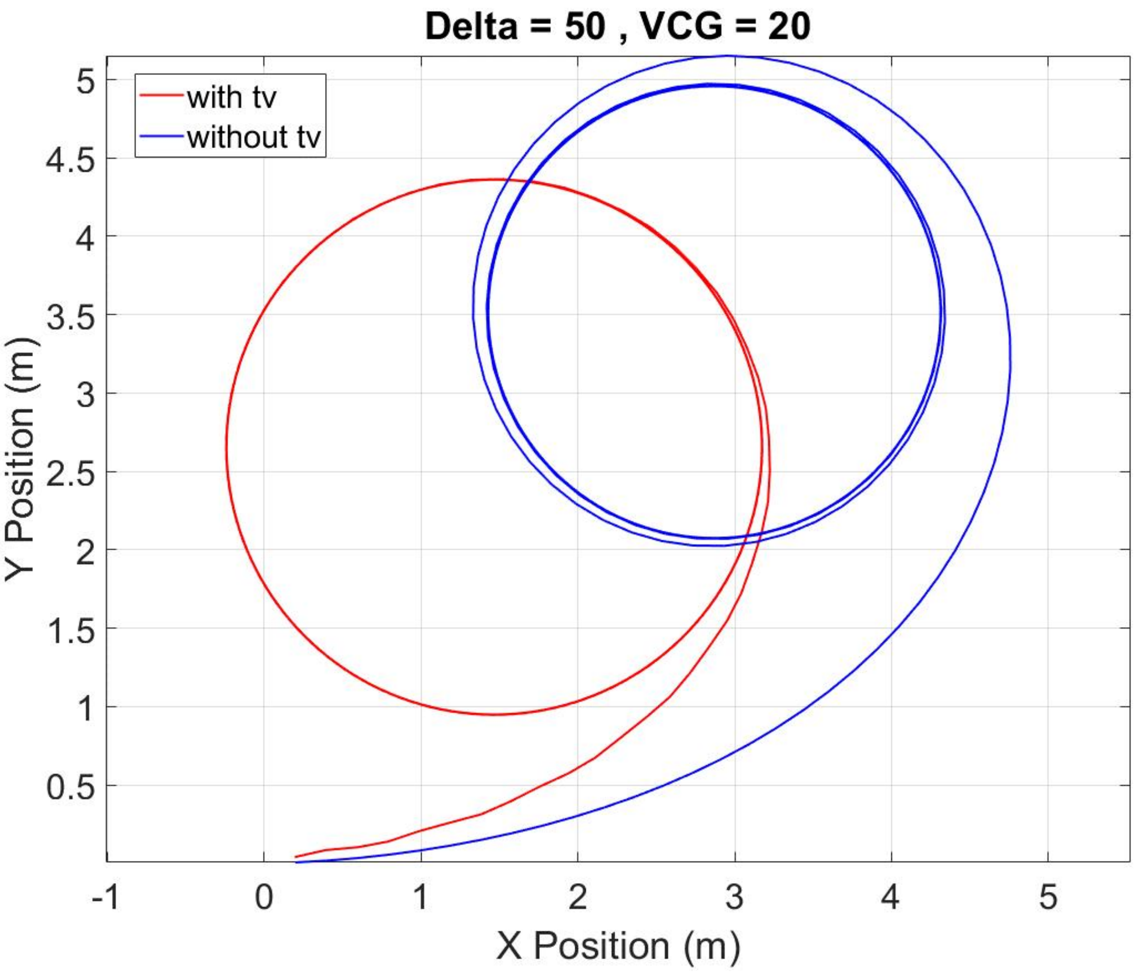

Fig. 5.1 path of the vehicle with and without torque vectoring for 20 m/s

The above figure shows the path of the vehicle at cornering one without torque vectoring and other with torque vectoring. The understeer gradient is positive in this case and the vehicle is understeering so in order to correct the path a PID control is used in the Simulink model to add the additional yaw moment.

Yaw rate vs time of the vehicle with steering angle 50 degrees and velocity 20 m/s :

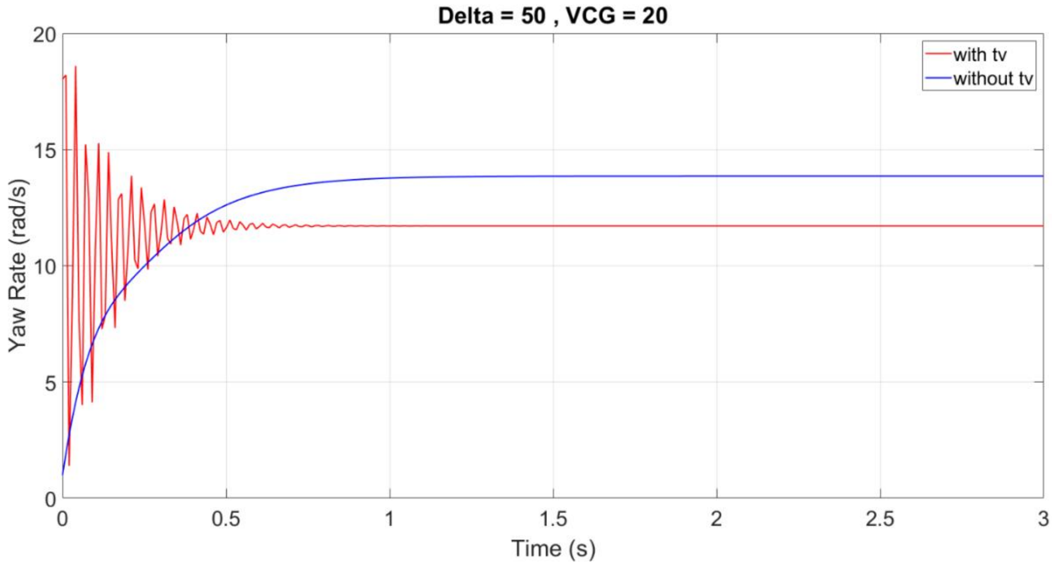

Fig 5.2 yaw rate vs time for with and without torque vectoring for 20 m/s

The above figure shows the yaw rate with and without the torque vectoring. In the figure we see the initial yaw rate is 0 and it increases gradually up to 14 rad/s. but in order to correct the path PID controller added a torque in order to obtain the initial yaw rate of 18 rad/s.

Tuning of the PID controller with steering angle 50 degrees and velocity 20 m/s :

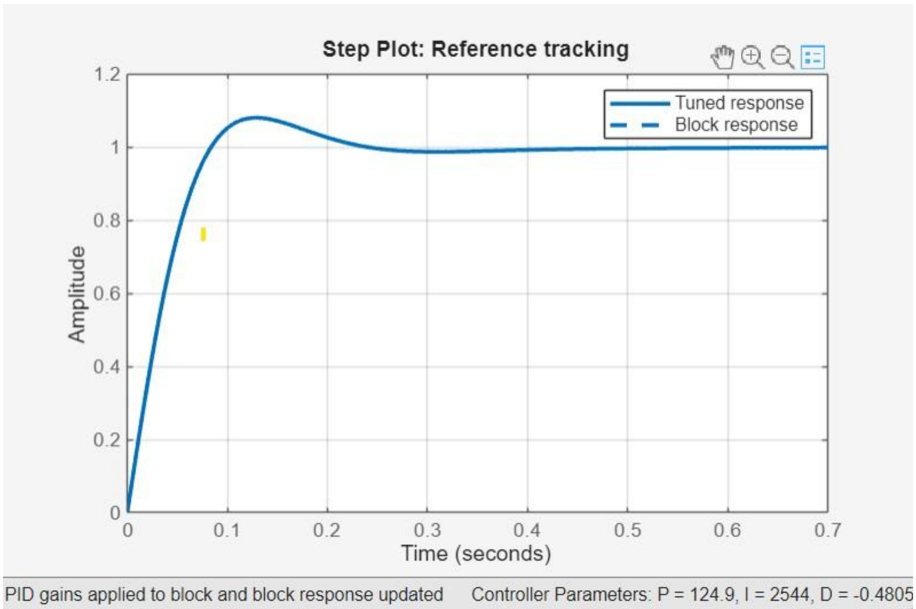

Fig 5.3 Step response after PID tuning for 20 m/s

The step response graph illustrates the performance of the PID controller for reference tracking. The tuned response shows a small overshoot around 1.2 at approximately 0.1 seconds and quickly settles to the reference value of 1.0, demonstrating minimal steady-state error. The PID gains used for tuning were P=124.9, I=2544, and D=−0.4805, ensuring effective tracking and system stability.

Comparison of the path of the vehicle for 50 degrees steering angle at 23 m/s and 25 m/s

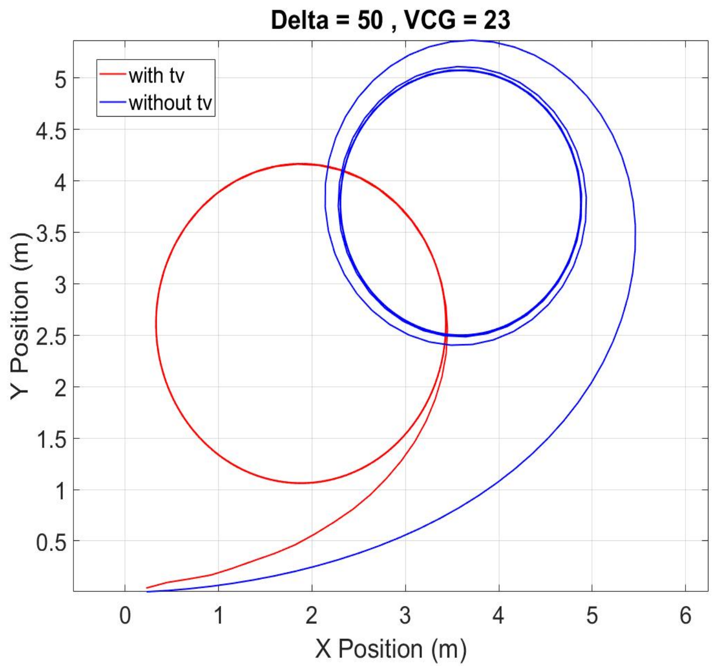

Fig 5.4 Path of the vehicle with and without torque vectoring for 23 m/s

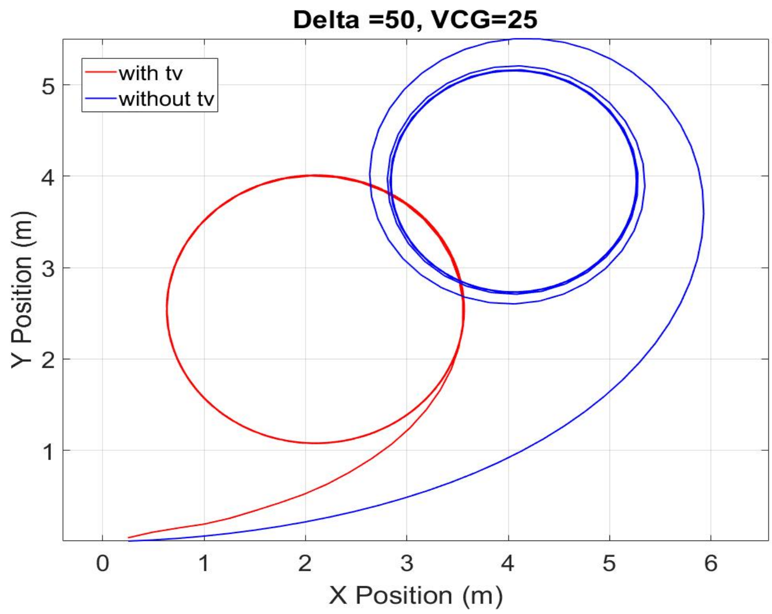

Fig 5.5 Path of the vehicle with and without torque vectoring for 25 m/s

Comparison of the yaw rate of the vehicle for 50 degrees steering angle at 23 m/s and 25 m/s

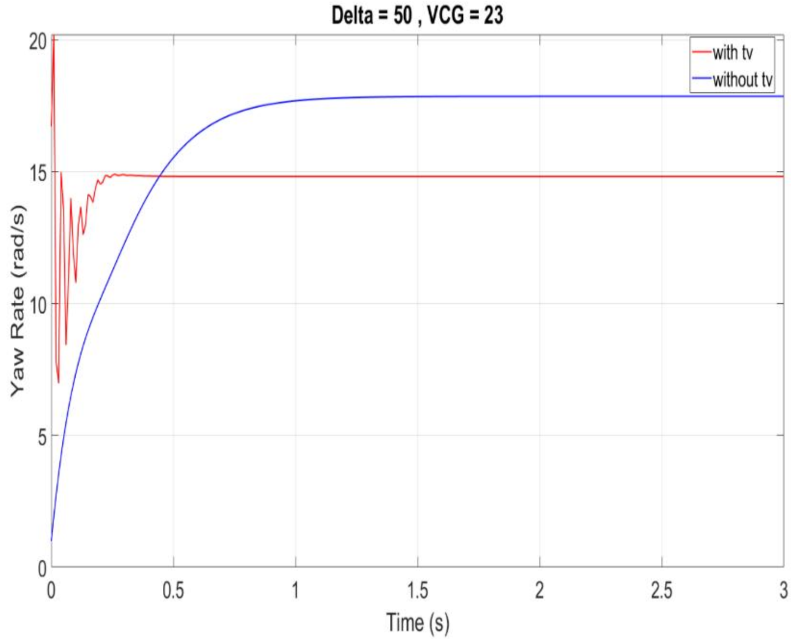

Fig 5.6 yaw rate vs time with and without torque vectoring for 23 m/s

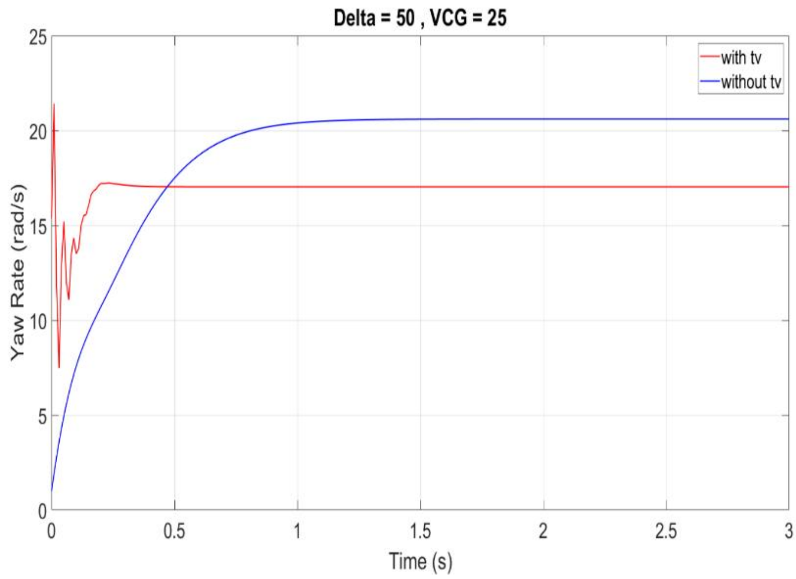

Fig 5.7 yaw rate vs time with and without torque vectoring for 25 m/s

Comparison of the step response tuning of PID for 50 degrees steering angle at 23 m/s and 25 m/s

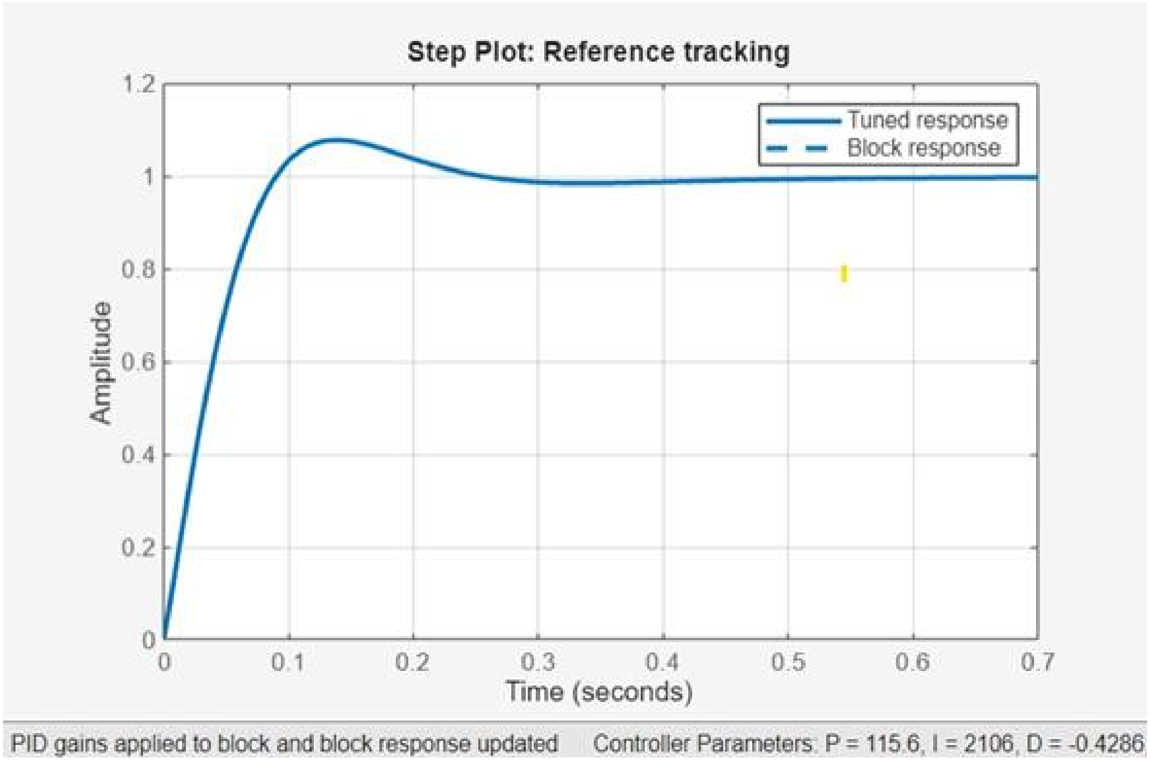

Fig 5.8 step response tuning of PID controller for 23 m/s

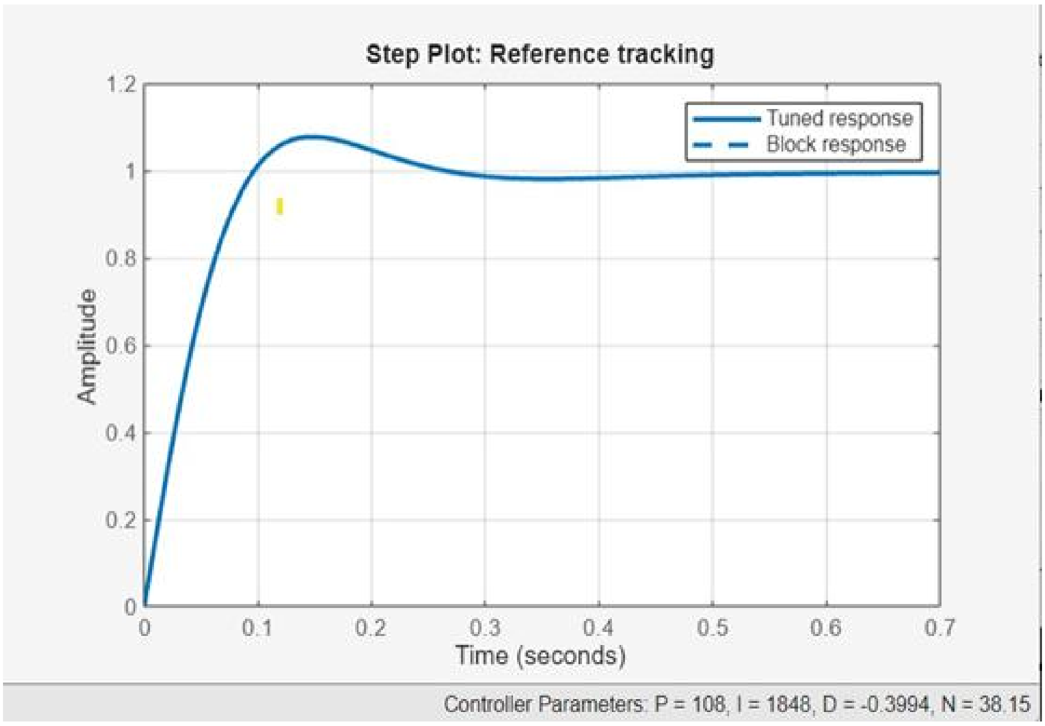

Fig 5.9 step response tuning of PID controller for 25 m/s

<u>Comparison of the path of the vehicle for 60 degrees steering angle at 20 m/s, 23 m/s and 25 m/s</u>

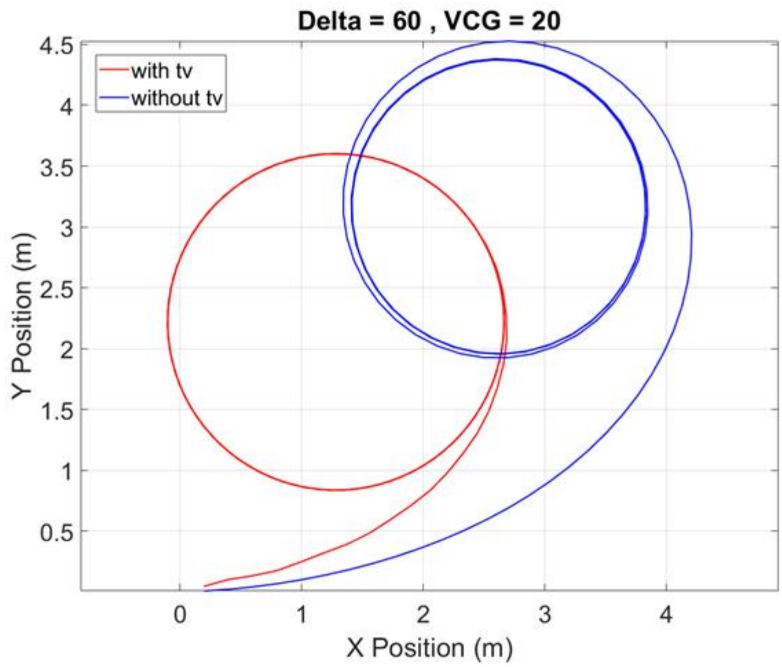

Fig 5.10 Path of the vehicle with and without torque vectoring for 20 m/s

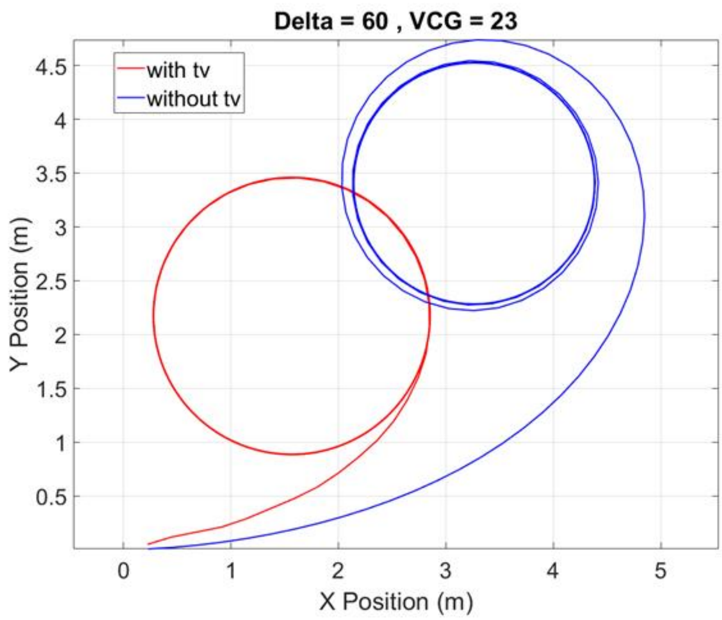

Fig 5.11 Path of the vehicle with and without torque vectoring for 23 m/s

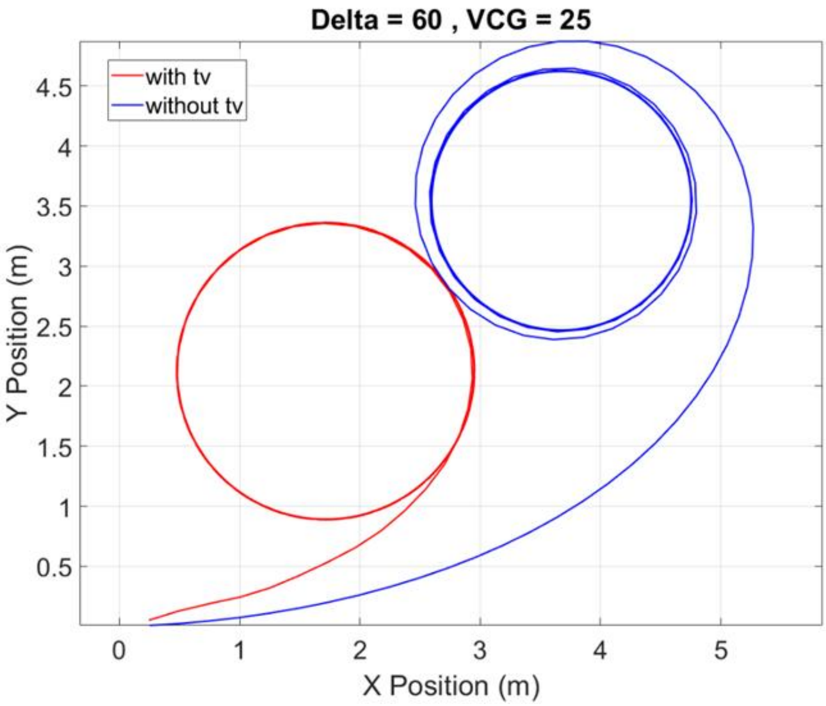

Fig 5.12 Path of the vehicle with and without torque vectoring for 25 m/s

Comparison of the yaw rate of the vehicle for 60 degrees steering angle at 20 m/s , 23 m/s and 25 m/s

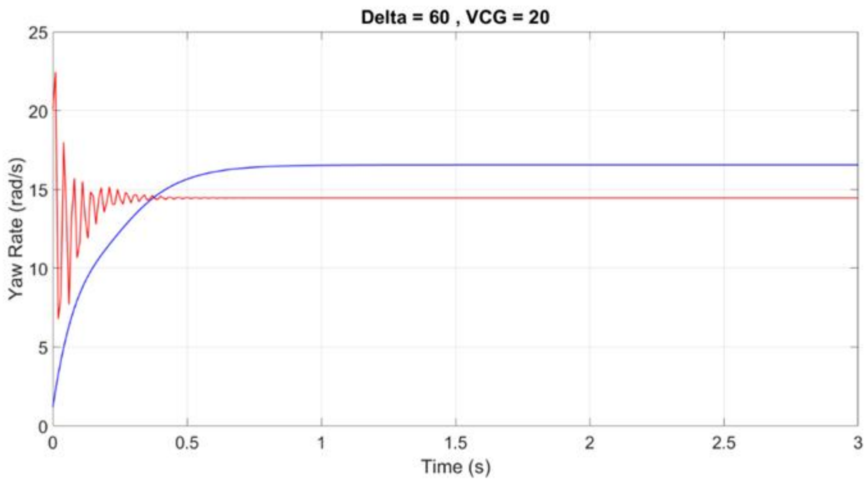

Fig 5.13 yaw rate vs time with and without torque vectoring for 20 m/s

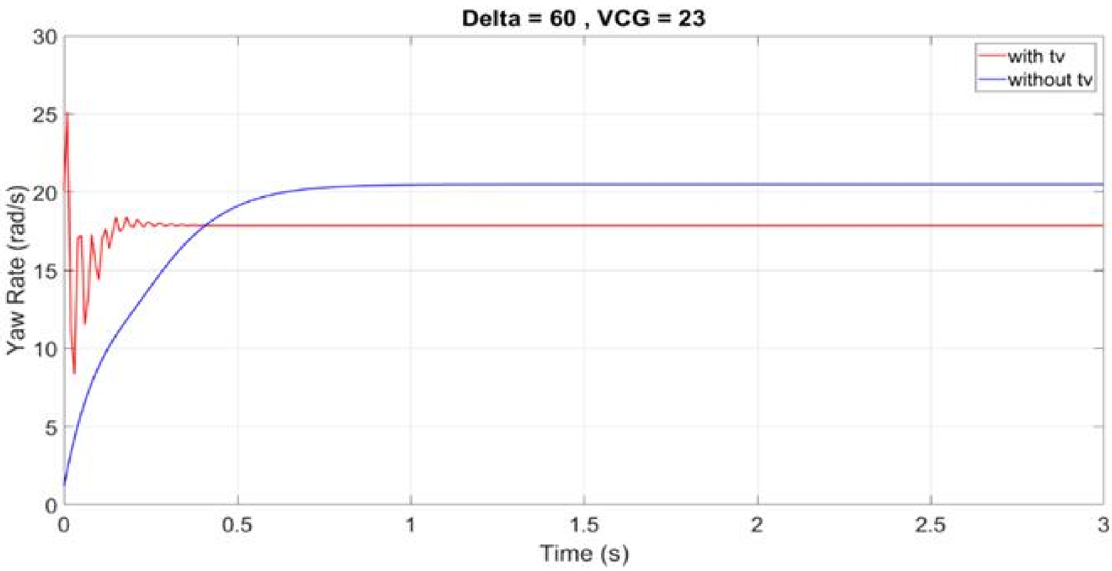

Fig 5.14 yaw rate vs time with and without torque vectoring for 23 m/s

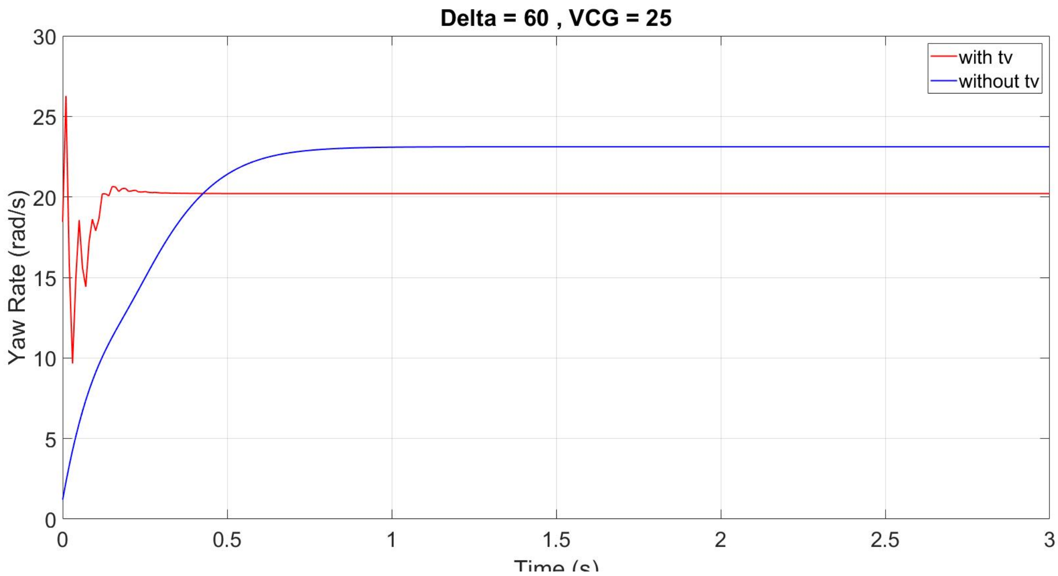

Fig 5.15 yaw rate vs time with and without torque vectoring for 25 m/s

Comparison of the step response tuning of PID for 60 degrees steering angle at 20 m/s , 23 m/s and 25 m/s

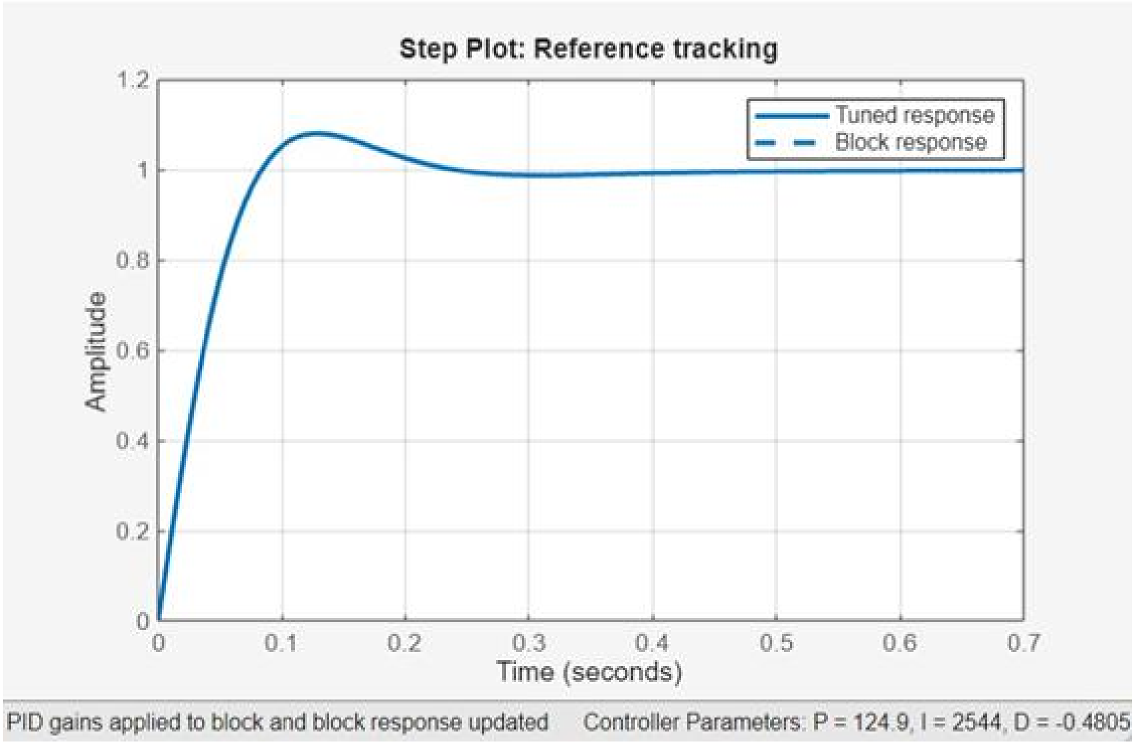

Fig 5.16 step response tuning of PID controller for 20 m/s

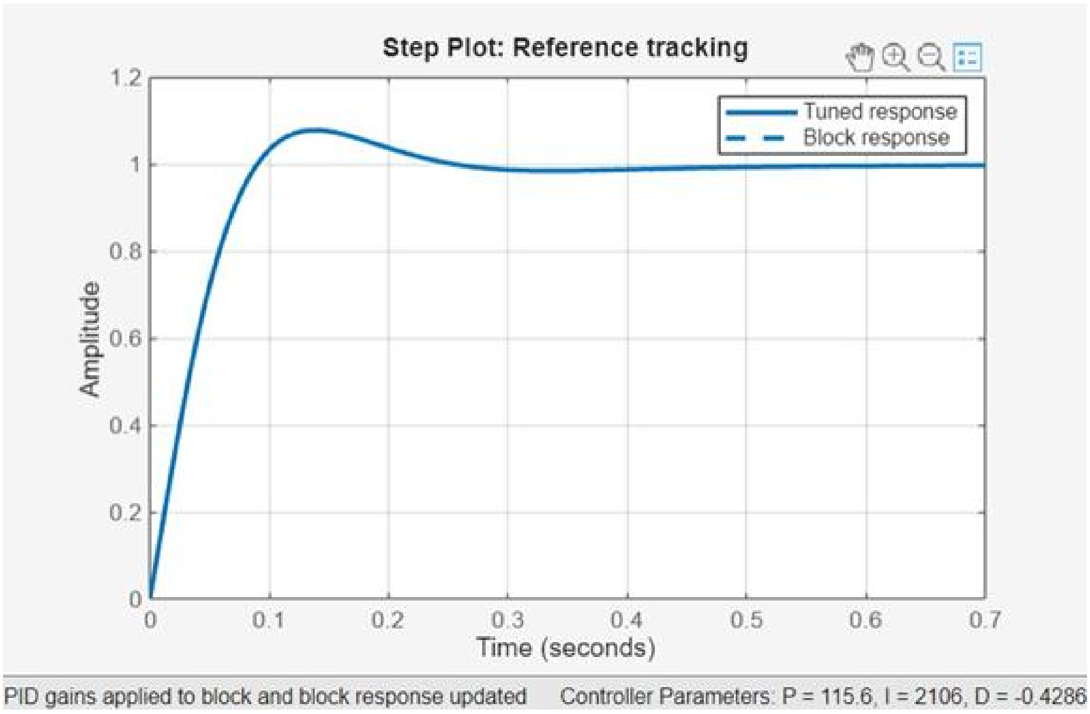

Fig 5.17 step response tuning of PID controller for 23 m/s

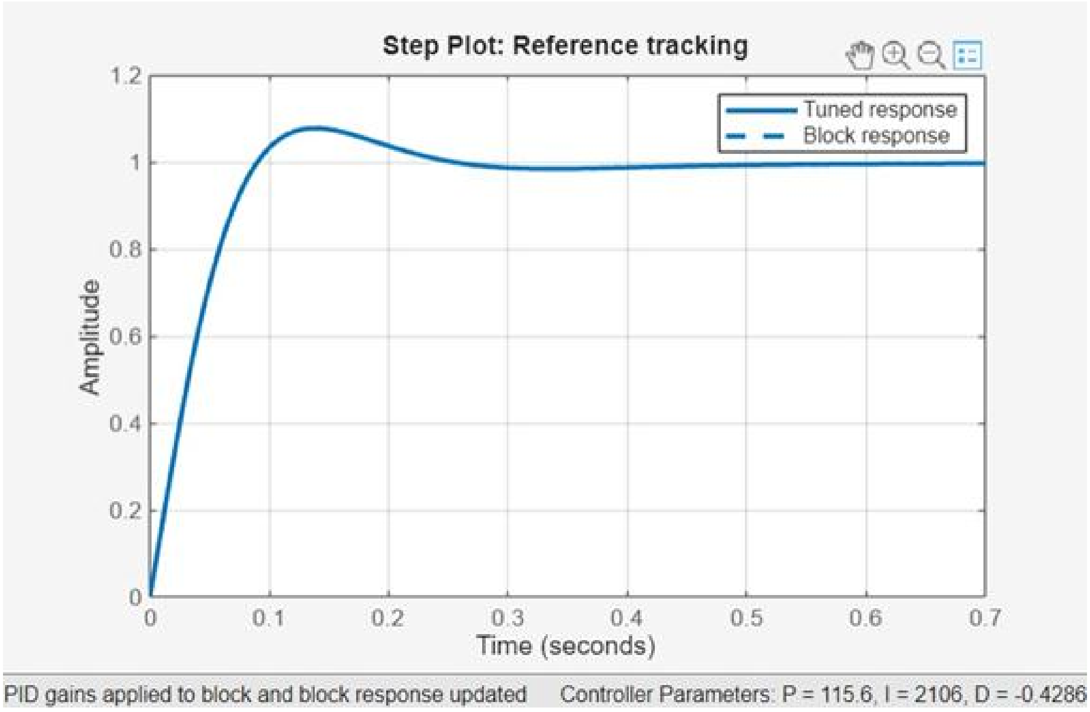

Fig 5.18 step response tuning of PID controller for 25 m/s

5.2 Comparative analysis for proportional gain(kp), Integral gain (ki) and Derivative gain(kd) for 50 degrees steering angle

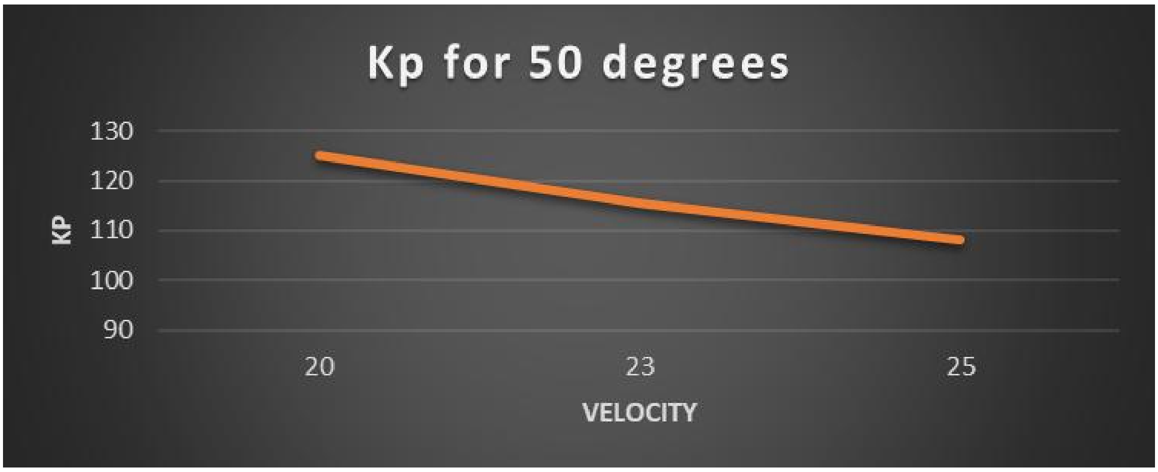

Fig 5.19 Kp value variation with respect to veloity at 50 degrees

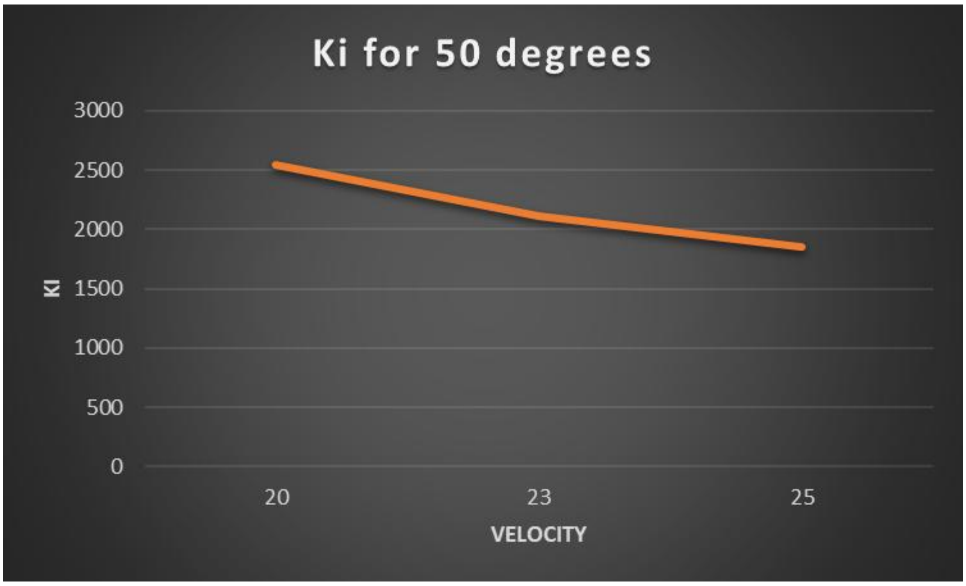

Fig 5.20 Ki value variation with respect to veloity at 50 degrees

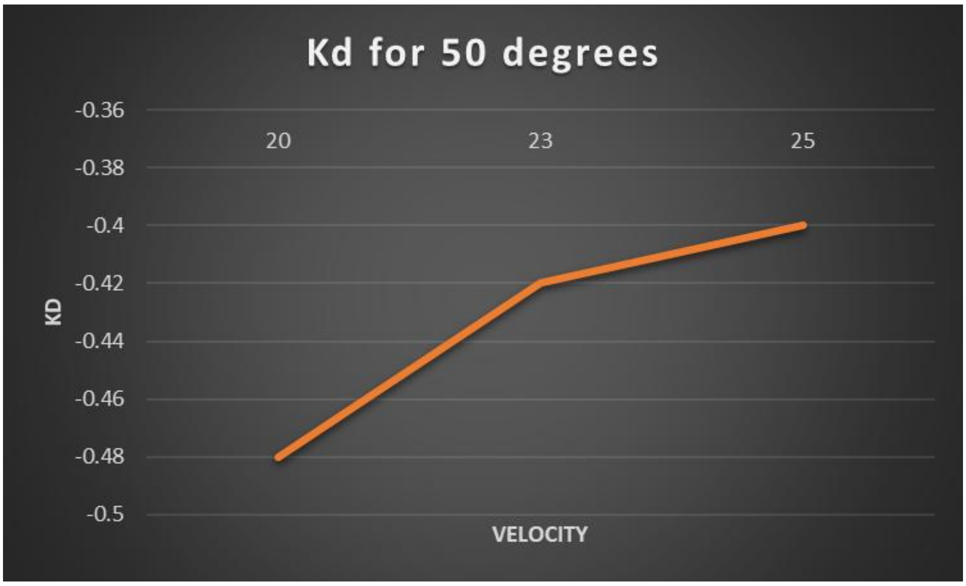

Fig 5.21 Kd value variation with respect to veloity at 50 degrees

5.2.1 Comparative analysis for proportional gain(kp), Integral gain (ki) and Derivative gain(kd) for 60 degrees steering angle

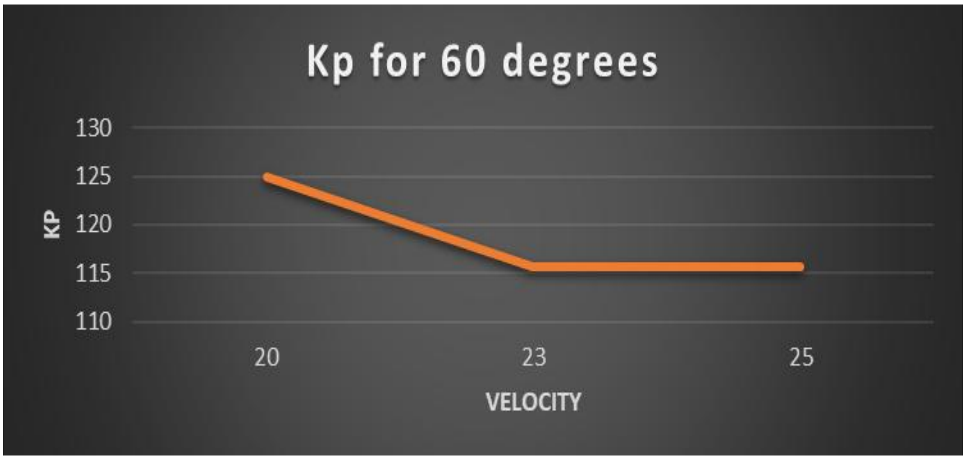

Fig 5.22 Kp value variation with respect to veloity at 60 degrees

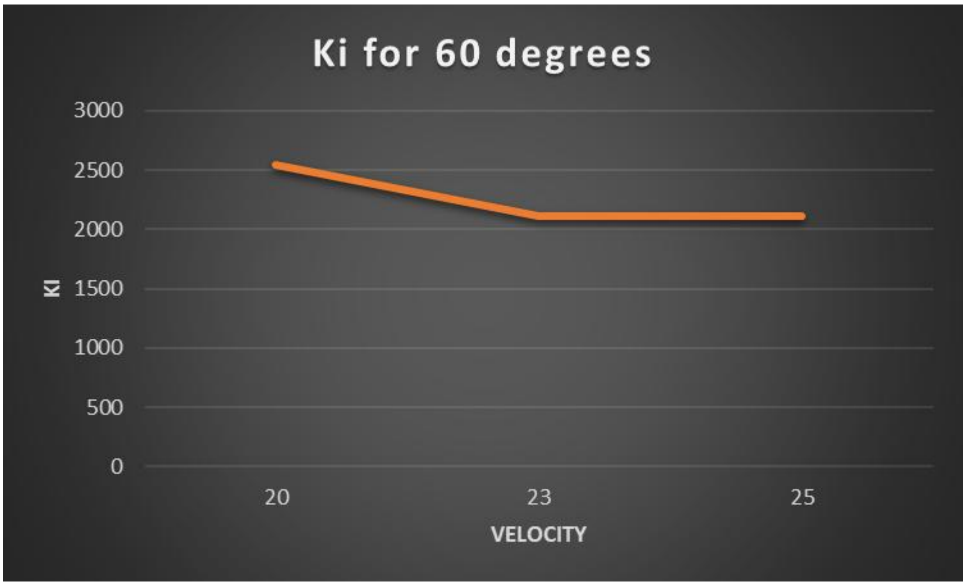

Fig 5.23 Ki value variation with respect to veloity at 60 degrees

Fig 5.24 Kd value variation with respect to veloity at 60 degrees

CHAPTER 7 : SUMMARY

The analysis indicates that at high vehicle speeds and during sharp cornering with a large steering angle, the vehicle exhibits significant under-steer, requiring an increased yaw rate to maintain stability and control.

As the under-steer gradient(K_u) for all the cases is positive the vehicle is actually under-steering in all the cases while cornering. So the turning radius is increasing and the additional yaw moment correcting the path i.e., the turning radius is decreased and maintain constant.

Table 6.1 comparison of ki, kp, kd values for 2 steering angles

steering angle	50 degrees			60 degrees		
velocity	**Kp**	**Ki**	**Kd**	**Kp**	**Ki**	**Kd**
20	125	2544	-0.48	125	2544	-0.48
23	115.6	2106	-0.42	115.6	2106	-0.42
25	115.6	2106	-0.42	108	1848	-0.42

CHAPTER 8

CONCLUSION

This project successfully demonstrates the application of a PID-controlled torque vectoring system in vehicle dynamics simulations. The integration of the PID controller effectively modulates the torque distribution to improve the vehicle's yaw rate control, which directly enhances stability and maneuverability. By continuously adjusting the vehicle's torque output based on yaw rate errors, the system ensures that the vehicle follows the desired path more accurately, reducing unwanted oscillations in yaw and improving overall control, particularly in high-speed or sharp-turn scenarios.

The performance of the vehicle dynamics, as shown in the simulation, indicates that the PID controller plays a vital role in achieving better path-following and smoother handling. The tuning of the PID parameters—proportional (Kp), integral (Ki), and derivative (Kd)—is crucial for fine-tuning the vehicle's response to changes in yaw rate, ensuring optimal performance without overshoot or instability. The results highlight the importance of carefully selecting these parameters to achieve the desired balance between responsiveness and stability.

This approach can be applied to real-world vehicle systems, particularly in the context of advanced driver-assistance systems (ADAS) or autonomous vehicles, where precise control of yaw rate is necessary for improving vehicle safety and handling. Overall, the project demonstrates the potential of PID-controlled torque vectoring systems to enhance vehicle performance in dynamic driving conditions, offering a promising solution for improving vehicle stability and safety.

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