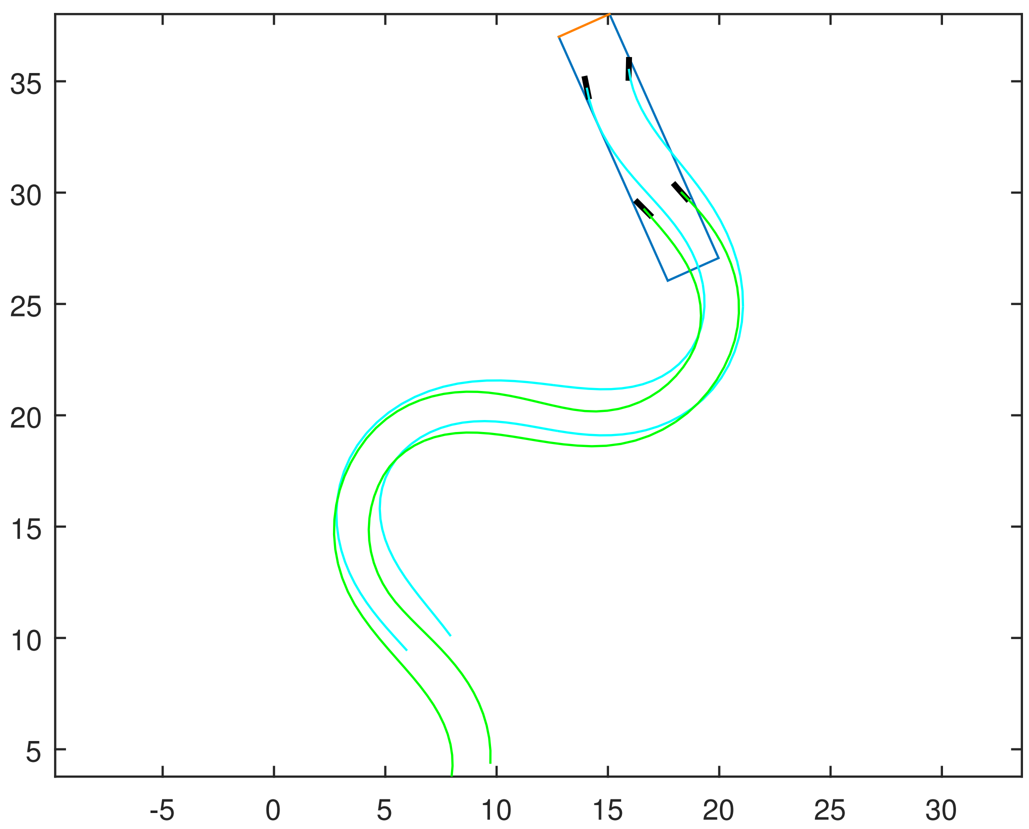

Rear axle steer-by-wire

System and Safety

Master's thesis in Systems, Control and Mechatronics

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Gothenburg, Sweden 2015

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Cover: Illustration of rear axle steering in a simulated heavy vehicle.

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Abstract

The automotive industry today is seeing significant advances in technology. One of the key segments within the automotive industry is development of heavy vehicles such as buses and trucks for passenger and freight transport respectively. It is, therefore, important to make heavy vehicles more comfortable and efficient, thereby impacting the growth of many other related industry segments. The standard design of a heavy vehicle is such that it has limited turning ability owing to its long vehicle body, making it hard for the vehicle to manoeuvre its way through narrow city lanes. By making the rear axle steerable, manoeuvrability can be greatly enhanced. The combination of front and rear steering gives rise to different steering modes, each with its own advantages.

In this thesis work, a model-based design approach is used to design and implement a rear axle steer-by-wire system in MATLAB and Simulink. This includes a state machine for switching between different steering modes, modelling the hydraulic system of rear axle actuator and designing a controller for the actuator. Both output feedback and state feedback controllers are evaluated and compared. P and PI controllers are designed as the output feedback controllers. While all designed controllers satisfy the required time domain performance characteristics, the state feedback controller has the best tracking performance for all test cases. The effect of rear axle steering on dynamics of the vehicle at low speeds is visualized by means of a simulator that is built based on a kinematic vehicle model. Simulator plots present a quantitative study of different steering modes that clearly reinforce their advantages. Before the designed system can be tested on a real vehicle, it is important to make sure that the system is safe. Concept phase functional safety analysis based on the ISO 26262 standard is therefore done to present a safety concept for the system.

Keywords: Controller; PID; State feedback; Safety concept; Functional safety; ISO 26262

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List of Abbreviations

ASIL Automotive Safety Integrity Level.

FMEA Failure Modes and Effects Analysis.

GPS Global Positioning System.

ISO International Organization for Standardization.

PID Proportional-Integral-Derivative.

Chapter 1

Introduction

Technological advancement is the core of modern day automotive industry. One of the key segments within the automotive industry is development of heavy vehicles which serve a wide range of purposes from inter- and intra-city transport of passengers by buses to shipment of cargo by trucks. Therefore, improvements made to heavy vehicles will indirectly benefit many other industry segments.

The manoeuvrability of heavy vehicles within cities can be difficult since it is not unusual for cities to have narrow lanes and junctions. One of the main reasons for limited manoeuvrability of heavy vehicles is their long vehicle bodies which results in having a long wheelbase. Wheelbase is the distance between the centres of the front and rear wheels [1]. The wheelbase of a typical $12m$ long heavy vehicle is usually around $6m$, nearly twice the wheelbase of a car. As a result, such vehicles have limited turning ability compared to lighter vehicles like cars. The turning ability of vehicles is quantified in terms of turning radius which is a measure of a vehicle's turning circle when the steering wheel is turned to its limit [2]. The wheelbase of heavy vehicles are optimally designed to achieve the best turning radius while supporting the weight of the vehicle body. Therefore, it is usually not possible to shorten the wheelbase of these vehicles any further. But if both front and rear wheels can be steered, it is possible to achieve a much smaller turning radius, thereby ameliorating the manoeuvrability of heavy vehicles greatly.

A smaller turning radius is not the only advantage that comes with rear axle steering. Rear axle steering, in general, improves the steering response. For example, with rear axle steering, it is possible to achieve stable manoeuvrability even at high speeds. This type of steering creates a flexible steering system, which optimizes the vehicle's manoeuvrability depending on the drive situation. This thesis attempts to explore, in depth, the possibilities that arise with rear axle steering and present a qualitative and quantitative analysis.

1.1 Background

The history of 4-wheel steering dates back to 1987 when Honda launched Prelude – the world's first steering angle sensing 4-wheel steering vehicle [3]. Rear axle steering is more often used in race cars or off-road vehicles like forklift trucks and construction vehicles. But implementation of active rear axle steering in street-legal heavy vehicles is a new concept.

Rear axle steering in vehicles can be passive or steer-by-wire. Steer-by-wire technology relies mainly on electronics to control steering. Over the years, a lot of research has gone into integrating computers and electronics into modern road vehicles. Steer-by-wire systems have the potential to increase comfort, functionality, fuel consumption and safety during the drive. Infiniti Q50 [4] was the first production vehicle to implement steer-by-wire and was launched in 2013.

Lam et al. [5] present a steer-by-wire interface for a four wheel independent steering vehicle with a feedback controller to synchronize the steering interface and the orientations of wheels. Some of the scientific challenges in such a system include understanding the steering actuator dynamics and vehicle body dynamics in order to achieve the required vehicle motion.

A steer-by-wire solution is a safety-critical system. Therefore, an evaluation in regard to safety needs to be made. This is done based on the functional safety standard ISO 26262 [6]. ISO 26262 consists of ten parts and provides safety guidelines for electrical and/or electronic systems within road vehicles.

The ISO 26262 is a relatively new standard that was released in December 2011 and hence there are few references available in literature on how to best make use of this standard. Ward and Cozier [7] outline the ISO 26262 approach to ASIL (Automotive Safety Integrity Level) decomposition. In an example of ASIL decomposition of 4-wheel steer-by-wire system, they have illustrated how the method can be misinterpreted resulting in incorrect ASIL determination. In general, Ward and Cozier [7] present some insight to approach the concept of functional safety. However, Ward and Cozier [7] focus on part 9 (ASIL-oriented and safety-oriented analyses) of the ISO 26262 whereas in this thesis, the focus has been on part 3 (Concept phase).

1.2 Problem statement

This thesis work was carried out at CPAC Systems AB and aimed at system level work that involved implementing a software application for the overall system architecture of a rear axle steer-by-wire system for a heavy vehicle and developing safety strategies.

While there can be many different technical solutions for a given task, looking at the same concepts from a safety perspective may warrant design changes. The real challenge lies in designing and implementing a system that satisfies all the functional requirements without compromising safety. Concept phase safety analysis in accordance with the ISO 26262 standard was an important part of the thesis work to ensure that the demonstration of a working system can be carried out safely with minimal technical risk.

The contributions of this thesis work are listed below:

- identification of the different possible ways in which a combination of front and rear axle steering can be used in a vehicle, and defining a state machine for switching between different steering modes based on vehicle speed and driver input;

- creation of a plant model for the rear axle actuator and designing a controller for it;

- development of a simulator to demonstrate the lateral vehicle dynamics of a vehicle enabled with rear axle steer-by-wire in the different steering modes and draw comparisons with its vehicle dynamics without any rear steering;

- preliminary concept phase functional safety analysis consisting of item definition, hazard analysis and risk assessment, and a functional safety concept.

1.3 Delimitations

In this particular thesis work, the focus has only been on rear axle actuation; the steering of front wheels is taken care of by an existing system, the Volvo Dynamic Steering [8]. Additionally, motor control is beyond the scope of this thesis work. Therefore, the rear axle actuator modelled in this thesis does not include a motor controller.

The rear axle steering concept investigated in this thesis is 2-axle steering. Hence it has limited steering modes compared to a 4-wheel independent steering system. Being a proof-of-concept thesis project, the goal is to test the different steering modes in low speed manoeuvres on a heavy vehicle. Therefore, it is reasonable to assume that switching modes at low speed will not compromise safety. While this thesis work acknowledges that a safe mode switching mechanism needs to be in place, particularly for high speeds, it does not investigate this any further.

1.4 Thesis organization

The report is organized into eight chapters. Chapter 2 discusses the different steering modes and gives an overview of the system architecture. Chapter 3 gives a background on the lateral dynamics of a 4-wheel vehicle. In chapter 4, the modelling of the rear axle actuator is described. Chapter 5 details the design of a controller for the actuator and discusses the simulation results. Chapter 6 goes through the implementation of the simulator. Chapter 7 describes the entire process of concept phase functional safety analysis. Finally, chapter 8 summarizes and concludes the thesis work.

Chapter 2

System overview

This chapter starts with section 2.1 defining steering modes in a vehicle with rear axle steering. Once steering modes have been well understood, section 2.2 on model-based design outlines a sketch of how the entire rear axle steer-by-wire system is implemented in order to bring to effect the different steering modes in a vehicle.

2.1 Steering modes

When a vehicle's front as well as rear wheels can be steered, a number of different steering modes can be defined based on the relative angular positions between the front and rear wheels. Described below are the different theoretically/conceptually possible steering modes:

Front steering

In this steering mode, only the front wheels of the vehicle steer. This is the same as conventional steering in a typical vehicle.

Rear steering

In this mode, only the rear wheels are steered. The rear steering mode is not intended to be driver controlled. Using data gathered from various sensors, the rear wheels are steered to correct unwanted drift in the vehicle caused due to side winds and road conditions, e.g. while driving on a banked road. This mode is only activated when there is no steer signal from the driver, i.e. the driver intends to drive straight. The functional advantage provided by this mode is in theory very similar to Volvo Dynamic Steering which is designed to compensate for kicks from road ruts, pot holes and road markings; correct course after braking on a mixed-friction surface; and correct disturbances due to side winds [8]. The Volvo Dynamic Steering, however, automatically corrects course by turning the front wheels. Sudden steering of front wheels may disturb the driver. Using rear wheels instead of front wheels for course correction will not divert the driver's attention.

Clamp steering

In the clamp mode, the rear wheels steer in a direction opposite to that of the front wheels allowing the vehicle to attain a smaller turning radius. Typical scenarios when this mode may be helpful are while parking or taking turns at junctions. This mode should be used only when the vehicle speed is low. As pointed out by Shimomura and Morita [9], even a small change in the steering angle at high speed in clamp steering mode significantly affects the direction of the vehicle body and hence its safety.

Crab steering

In the crab mode, the rear wheels steer in the same direction as that of the front wheels. Unlike the clamp mode, the crab mode is more versatile and has advantages in both low and high speed vehicle manoeuvres. Firstly, the crab mode makes lateral movement easy at low speeds, e.g. it can be used for docking and undocking a bus at stops where passengers alight and board. Secondly, at high speeds, crab mode provides better yaw stability, e.g. while changing lanes and/or overtaking in highways.

Table 2.1 summarizes all the steering modes including their advantages and possible sccenarios where they could be used. Theoretically, it must be possible to just reverse the functionality of the front and rear wheels when driving in reverse direction. However, there might be practical difficulties in doing so. One of the hindrances could be that the maximum possible steer angle of the front and rear wheels may not be the same and hence swapping the functionality of front and rear wheels will not result in the same vehicle motion in reverse as in the forward direction. Additionally, to be able to swap the functionality of front and rear wheels, they must be coordinated by the same electronic system.

2.2 Model-based design

Model-based design is widely used in all industries that work with embedded control systems development. It typically supports automatic code generation from models. Extensive Verification and Validation at an early stage ensure fewer bugs

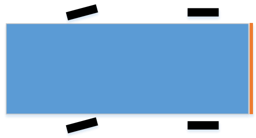
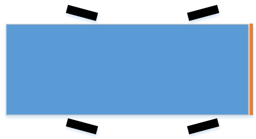
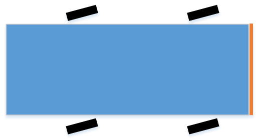
Mode	Action	Advantage	Application
Front steering	Only front wheels steer 	Conventional Driving	Driving straight on a flat road
Rear steering	Only rear wheels steer 	Corrects unwanted drift due to side wind and road conditions	Driving on a banked road
Clamp steering	Rear wheels steer in a direction opposite to that of front wheels 	Smaller turning radius	Parking, sharp turns at junctions
Crab steering	Rear wheels steer in the same direction as the front wheels 	Easy lateral movement, stability at high speeds	Docking and undocking at stops, changing lanes on a highway

Table 2.1: Steering modes. Red line indicates front end of the vehicle.

and less rework during final system integration and test since finding and fixing bugs is much easier on a desktop rather than encountering it while testing the system on a vehicle [10].

Implementation of a rear axle steer-by-wire system consists of different blocks as seen in figure 2.1. The first step is to determine which steering mode the vehicle should be in based on steering mode request from the driver and current vehicle speed. Once the steering mode is assigned, the required rear axle angle needs to be determined based on the assigned steering mode and the steering input. The desired rear axle angle then becomes the set point for a controller which outputs the torque required to actuate a hydraulic system and achieve the required rear axle angle. MATLAB with Simulink was used to design and create the entire control system including building a plant model for the rear axle actuator, analyzing and synthesizing a controller for the plant, and simulating the plant and controller.

It is interesting to take rear axle steer-by-wire solution a step further and analyze the effect of rear steering on the dynamics of the vehicle. Therefore, a vehicle simulator was built such that a virtual vehicle could be driven by interfacing external hardware (e.g. a joystick that could provide the different driver inputs like vehicle speed, steering and mode selection) with the Simulink model. In figure 2.1, the vehicle dynamics block generates the position coordinates and heading of the vehicle based on steering inputs, followed by plotting of those coordinates for visualization.

In the following chapters, each of the blocks in figure 2.1 is developed one by one - from right to left - and then integrated together. The theory of lateral vehicle dynamics used for the simulator is explained in the next chapter.

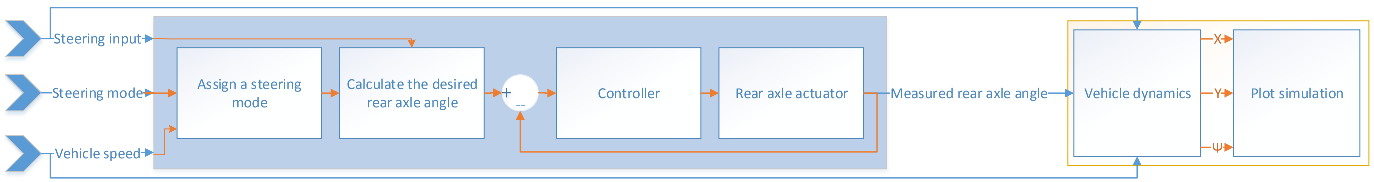


Figure 2.1: Model-based design outline of a rear axle steer-by-wire system.

Chapter 3

Lateral vehicle dynamics

To be able to simulate and plot the movement of a vehicle, the position and heading of the vehicle should be known. This is done by modelling the vehicle and using its equations of motion. A 4-wheel vehicle model used in this thesis is based on the work presented by Wang and Qi [11]. As pointed out by Wang and Qi [11], a large 4-wheel vehicle can be modelled approximately as a bicycle if the assumptions of planar motion, rigid body and non-slippage of tyres are made. In the bicycle model, the two front wheels of a 4-wheel vehicle are considered as one single wheel and the rear wheels are considered as the other, as depicted in figure 3.1.

A kinematic model gives a mathematical description of a vehicle's motion without considering the external forces affecting it. It is reasonable to model the lateral motion of a vehicle with a kinematic model for low speeds ($< 5m/s$) because then the lateral forces are small [12]. Since the initial testing of rear axle steering is planned for low speeds, it is sufficient to use a kinematic model for simulation.

Planar motion is described using X and Y position coordinates of the center of gravity of the vehicle, and orientation of the longitudinal axis of the vehicle body with respect to global X-axis, ψ, as seen in figure 3.1. V is the velocity of the vehicle at centre of gravity and β, the slip angle of the vehicle, is the angle between the vehicle's velocity vector and longitudinal axis of the vehicle.

The instantaneous rolling center for the vehicle is denoted by O and is defined as the intersection of lines drawn perpendicular to the orientation of the two rolling wheels of the bicycle. R is radius of the vehicle's path and defined as the distance between point O and centre of gravity. Direction of velocity V is perpendicular to radius of the vehicle's path.

By using geometrical relationships in figure 3.1, equations of motion are derived [11]. These equations are as follows:

$$\dot{X} = V cos(\psi + \beta) \tag{3.1}$$

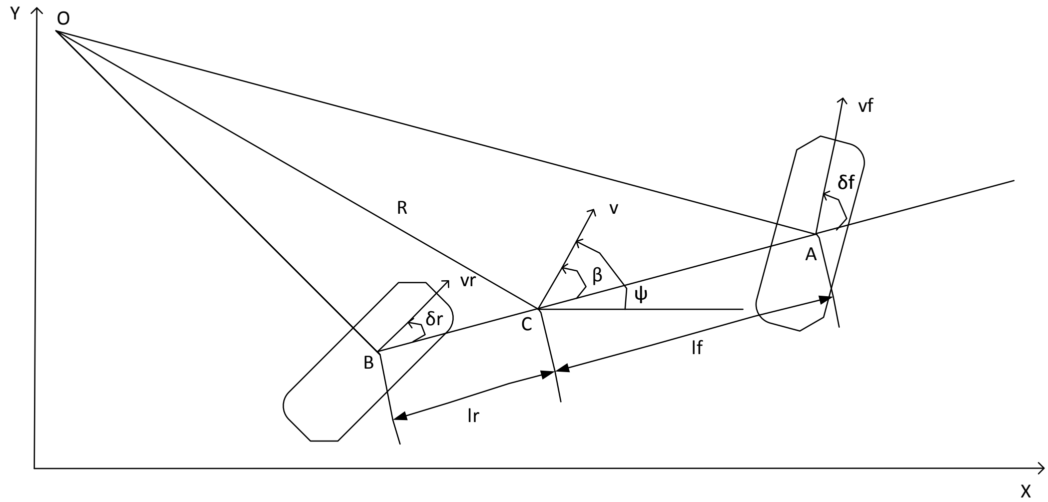


Figure 3.1: Kinematic model of lateral vehicle motion: δ_f is the angle made by the front wheel with respect to the vehicle's longitudnal axis, δ_r is the angle made by the rear wheel with respect to the vehicle's longitudnal axis, centre of gravity is at point C, distance between the front wheel and C is l_f and the distance between C and the rear wheel is l_r.

$$\dot{Y} = V sin(\psi + \beta) \tag{3.2}$$

$$\dot{\psi} = \frac{V cos(\beta)}{L}(tan(\delta_f) - tan(\delta_r)) \tag{3.3}$$

where δ_f and δ_r are the steering angles for the front and rear wheels respectively, and L is the wheelbase of the vehicle.

The slip angle β can be obtained as

$$\beta = tan^{-1}\left(\frac{l_f tan(\delta_r) + l_r tan(\delta_f)}{L}\right) \tag{3.4}$$

where l_f and l_r are distances of centres of front and rear wheels from centre of gravity respectively and $l_f + l_r = L$.

In the bicycle model, left and right front wheels are represented by one front wheel, and left and right rear wheels are represented by one rear wheel. However, in an actual four-wheel vehicle, the left and right steering angles will not be exactly equal because the radius of the path each of these wheels travels is different. If δ_f and δ_r are the average angles made by the front and rear wheels of a 4-wheel vehicle respectively, the corresponding steering angles made by the inner (δ_{fi}, δ_{ri}) and outer wheels (δ_{fo}, δ_{ro}) can be calculated using equations:

$$\delta_{fo} = \delta_f - \frac{\delta_r^2 w}{L} - \frac{2wL}{4R^2 - w^2} \tag{3.5}$$

$$\delta_{fi} = 2\delta_f - \delta_{fo} \tag{3.6}$$

$$\delta_{ro} = \delta_{fo} - \frac{L}{R + \frac{w}{2}} \tag{3.7}$$

$$\delta_{ri} = 2\delta_r - \delta_{ro} \tag{3.8}$$

where $R = \frac{L}{\delta_f - \delta_r}$ is centre of gravity's path radius and w is the average track width of the vehicle. Track width is the distance between the centerline of two road wheels on the same axle, each on the other side of the vehicle [13]. The above equations are based on Ackerman steering geometry and their derivation can be found in appendix A.

This chapter went through the modelling of a 4-wheel vehicle and obtained its equations of motion. In the next chapter, the focus is narrowed down to the main component of the rear axle steer-by-wire system, i.e. the rear axle actuator.

Chapter 4

Modelling of rear axle actuator

The rear axle actuator is a hydraulic system consisting of a motor controller, a hydraulic pump and a hydraulic cylinder. Based on the torque request, the motor controller generates the required torque from an electrical motor and outputs it to the hydraulic pump. The pressure developed across the pump is applied directly across a double-acting hydraulic cylinder. The translational displacement of the hydraulic cylinder piston results in steering of the rear axle through mechanical linkages. An angle sensor measures the angle of the rear axle and feeds it back to the controller for correction and reference tracking.

As mentioned earlier in section 1.3, motor control is not included in modelling of rear axle actuator in this thesis. Instead, torque request from the controller is directly input to the hydraulic pump in the simulation environment.

The rear axle actuator was modelled in two different ways. One was using Simscape, an environment within Simulink for modelling and simulating physical systems. The second method was mathematical modelling of the hydraulic system using first principles. The two models were compared to verify if the fundamental working of the Simscape model is in accordance with first principles.

4.1 Simscape modelling of the hydraulic system

The Simscape model of the hydraulic system used in rear axle actuator can be seen in figure 4.1. The model consists of a hydraulic pump that can rotate in both directions and a double-acting hydraulic cylinder.

Table 4.1 lists some of the known specifications of the pump and cylinder that are used in the Simscape model. The input to this model is pump torque. The calculation for the range of input torque can be found in appendix B.

A mass-spring-damper system is used to simulate the mechanical load. An ideal translational motion sensor is used to measure the load displacement which

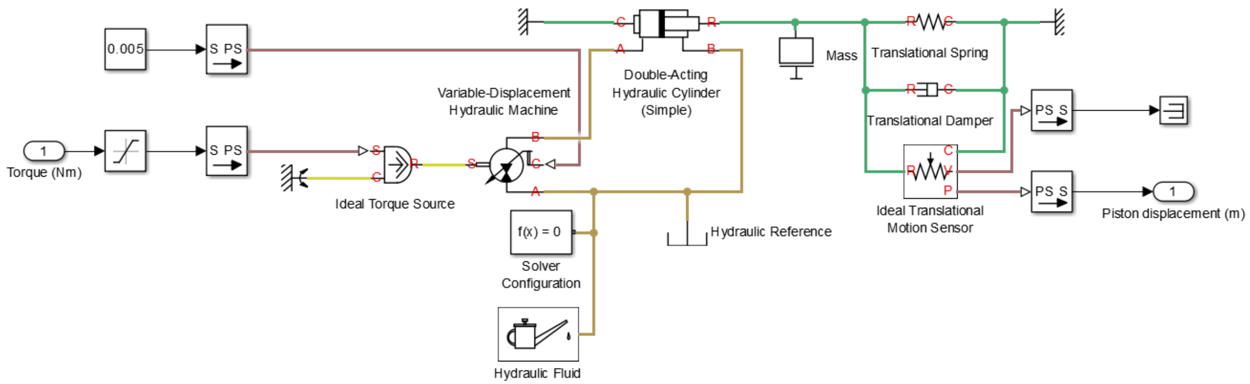


Figure 4.1: Simscape model of the hydraulic system used in the rear axle actuator: The hydraulic system consists of a hydraulic pump and a hydraulic cylinder. A mass-spring-damper system simulates the mechanical load. Pump torque is the input to the system and the piston/load displacement is the output measured by an ideal translational motion sensor. Since Simscape is a physical environment, the inputs to the Simscape model have to first be converted from Simulink to physical system (S→ PS). Similarly, the output of Simscape is converted from physical system to Simulink (PS→ S).

Parameter	Value
Pump displacement per revolution, V_g	$0.000006m^3$
Nominal Pressure	$180bar$
Nominal Angular Velocity	$1500rpm$
Mechanical-hydraulic efficiency, η_{mh}	0.75
Moment of inertia	$0.0006kgm^2$
Piston area of cylinder, A	$0.01075m^2$
Steering unit rack travel	$0.0812m$
Force efficiency, η_f	0.75

Table 4.1: Hydraulic pump and cylinder specifications

is the model output. The load displacement is proportional to piston displacement.

The disturbances in this system will be mainly due to road conditions and side wind. Under test conditions and low vehicle speeds, both these disturbances may be considered to be negligible. It is likely that there will be some measurement noise from the angle sensor which may be removed using a low pass filter. Therefore, the use of an ideal sensor in the model is acceptable.

Open loop input-output behaviour of the Simscape model in figure 4.1 for a sine input can be seen in figure 4.2. There are a few observations that can be made from the input-output behaviour. Firstly, the piston displacement saturates at $+0.0406m$ and $-0.0406m$. This is because the cylinder stroke length is $0.0812m$ (see table 4.1). Secondly, when the input torque is zero, the slope of the displacement curve is also zero, i.e. it is at a maxima or minima. On the other hand, when the magnitude of input torque is maximum, the magnitude of slope of the displacement curve is also maximum. This implies that zero torque results in zero velocity (i.e. the piston is stationary) and maximum torque in maximum velocity. This behaviour suggests that the hydraulic system has been modelled correctly.

In order to design a controller for the plant model, it is necessary to first linearize the non-linear Simscape model. For continuous linear time-invariant systems, the standard state-space representation is given by:

$$\dot{\mathbf{x}} = \mathbf{Ax} + \mathbf{Bu}$$

$$\mathbf{y} = \mathbf{Cx} + \mathbf{Du}$$

where $\mathbf{x}$ is the vector of state variables $(n \times 1)$, $\dot{\mathbf{x}}$ is the time derivative of state vector $(n \times 1)$, $\mathbf{u}$ is the input or control vector $(p \times 1)$, $\mathbf{y}$ is the output vector $(q \times 1)$, $\mathbf{A}$ is the system matrix $(n \times n)$, $\mathbf{B}$ is the input matrix $(n \times p)$, $\mathbf{C}$ is the output matrix $(q \times n)$, $\mathbf{D}$ is the feedforward matrix $(q \times p)$.

Using the MATLAB command `linmod`, the linear state-space and transfer function representations of the Simscape model are obtained. The resulting linearized

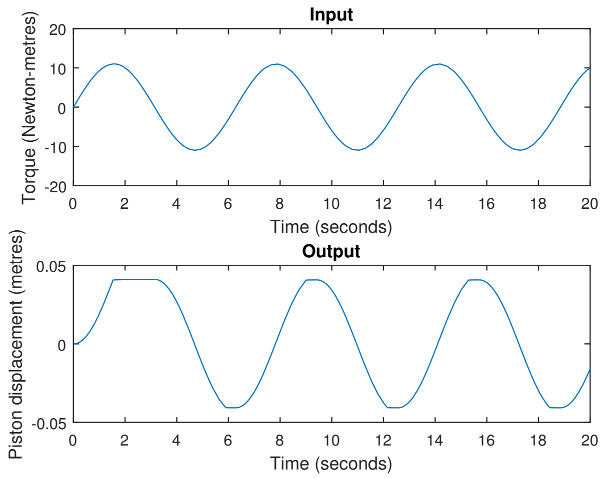


Figure 4.2: Open loop simulation of the Simscape model: The input to the system is pump torque and the output is piston/load displacement. Sine input evaluates the input-output behaviour over the entire permissible range of pump torque. The initial displacement of the piston is half-way between the two ends of the cylinder which keeps the load in position zero.

model is of order 15. To facilitate analysis, the linear model had to be simplified. This was done using the MATLAB command `minreal` which eliminates uncontrollable or unobservable state in state-space models, or cancelling or near-cancelling pole-zero pairs from transfer functions. The resulting simplified model is a second-order linear model. The matrices A, B, C and D obtained for the reduced model are as follows:

$$A = \begin{bmatrix} 0 & 1 \\ -0.4545 & -1181.9 \end{bmatrix} \tag{4.1}$$

$$B = \begin{bmatrix} 0 \\ 5.1170 \end{bmatrix} \tag{4.2}$$

$$C = \begin{bmatrix} 1 & 0 \end{bmatrix} \tag{4.3}$$

$$D = \begin{bmatrix} 0 \end{bmatrix} \tag{4.4}$$

From equations 4.3 and 4.4, it can be inferred that the first state variable of the system is actually the measured hydraulic piston displacement. Also, from equation 4.1, it can be inferred that the second state variable is the velocity of the hydraulic piston.

By computing the eigen values of matrix A using the MATLAB command eig(A), the stability of the system can be determined. Eigen values are obtained to be -0.0004 and -1181.9. Since they both have negative real parts, the system is stable. It can also be noted that there is one very slow pole (-0.0004) and one very fast pole (-1181.9).

The transfer function after pole-zero cancellation is also obtained using the MATLAB commands linmod and minreal. The transfer function of the system, from input torque, T, to piston displacement, x is:

$$\frac{X(s)}{T(s)} = \frac{5.117}{s^2 + 1181.9s + 0.4545} \tag{4.5}$$

The poles of the transfer function in equation 4.5 are the same as the eigen values of matrix A.

The translational displacement of the piston is mapped to the angle made by the rear axle using a linear relationship as shown in figure 4.3. The transfer function from input torque, T, to measured rear axle angle, δ_r is:

$$\frac{\Delta_r(s)}{T(s)} = \frac{72.5905}{s^2 + 1181.9s + 0.4545} \tag{4.6}$$

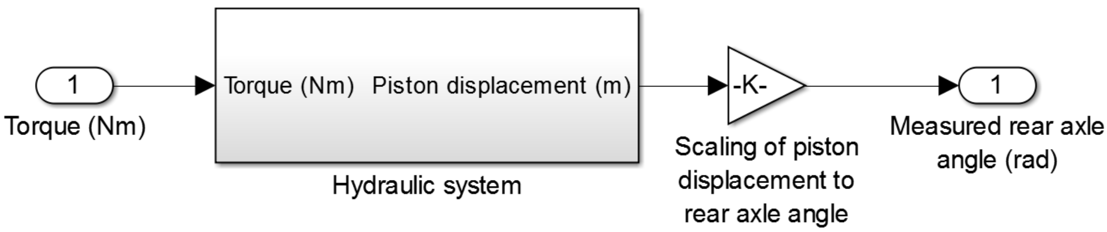


Figure 4.3: Rear axle actuator model with piston/load displacement converted to rear axle angle.

Since the conversion from piston displacement to rear axle angle has been realized only using a scaling operation, there is no change in the poles or eigen values of the system. The matrices A, B and D in the state-space representation remain unchanged. However, the scale factor is reflected in the output matrix, C, which for the system in figure 4.3 is obtained as:

$$C = \begin{bmatrix} 14.1862 & 0 \end{bmatrix} \tag{4.7}$$

Therefore, equations 4.1, 4.2, 4.7 and 4.4 form the state-space of the system in figure 4.3. In the next section, the hydraulic system is modelled using first principles and compared with the Simscape model.

4.2 Modelling of the hydraulic system using first principles

The input to the actuator system is the torque, T, supplied to the hydraulic pump. From the pump data sheet,

$$T = \frac{1.59 \times V_g \times \Delta p}{100 \times \eta_{mh}} \tag{4.8}$$

where V_g is the displacement per revolution in cm^3, Δp is the differential pressure in bar and η_{mh} is the mechanical-hydraulic efficiency.

The pressure differential, Δp, created across the pump is the input for the hydraulic cylinder. Therefore, equation 4.8 is rewritten as

$$\Delta p = \frac{100 \times \eta_{mh} \times T}{1.59 \times V_g} \tag{4.9}$$

The piston force, F_p, is given by

$$F_p = \Delta p \times A \times \eta_f \tag{4.10}$$

where A is the piston area in m^2 and η_f is the force efficiency of the cylinder.

If the mechanical load that is actuated by the piston movement is approximated to be a mass-spring-damper system, equation 4.10 can be rewritten as

$$m\frac{d^2x}{dt^2} + b\frac{dx}{dt} + kx = \Delta p \times A \times \eta_f \tag{4.11}$$

where m is the weight of the mechanical load, b is the damping coefficient, k is the spring coefficient and x is the translational displacement of the hydraulic piston in m. The values of m, b and k used here are the same as the values used in the Simscape model; these are $2200kg$, $2600000Ns/m$ and $1000N/m$.

Substituting equation 4.9 in equation 4.11 with appropriate unit conversion and using the values of all known constants from table 4.1, the transfer function of the system is obtained as

$$\frac{X(s)}{T(s)} = \frac{4.148}{s^2 + 1181.8s + 0.4545} \tag{4.12}$$

Comparing equations 4.5 and 4.12, it is evident that the denominators are essentially the same. This means that the system poles are in the same location in both cases and will therefore have similar response characteristics. The slight difference in the transfer function gain might be due to the fact that a physical system is never ideal. The Simscape model takes into account a lot of factors such as different volumetric and mechanical pump efficiency coefficients, the choice of hydraulic fluid and its viscosity, system temperature, etc. The effect of these factors on system behaviour has been ignored in the first principles analysis.

Since the Simscape environment replicates the non-linear physical system more accurately than a first principles model, the linearized model obtained from Simscape was used for designing a controller and the non-linear Simscape model was used for testing the controller performance as described in chapter 5.

Chapter 5

Controller design for the rear axle actuator

Essentially, a feedback compensator can be designed in two ways: output feedback and state-feedback. Both these techniques were used to design a controller for the rear axle actuator. These control designs, their results and analysis are covered in the next sections. But before that, the goals of controller design have to be defined.

5.1 Goals of controller design

A controller is designed to meet the following time-domain performance specifications:

- 90% rise time of $100ms$ for 1^o amplitude
- 2% settling time of $150ms$
- Maximum overshoot of 2% and no oscillatory behaviour
- Zero or nearly zero steady-state error
- Maximum input torque of magnitude $11Nm$ to the hydraulic pump

The hydraulic system would undergo slow time variation due to ageing and wear of components. However, this does not result in a qualitatively different behaviour. On a day-to-day basis, it is effectively time-invariant but from year to year, the parameters may change. Therefore, in addition to meeting the above specifications, the control system must be robust and have the ability to meet all the performance specifications despite a decrease in pump effectiveness down to 70%.

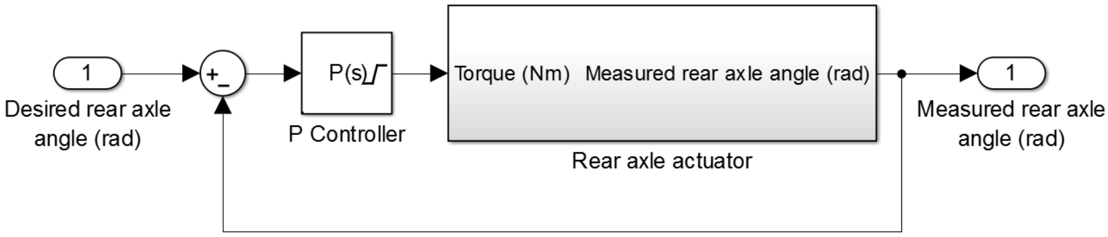


Figure 5.1: Closed loop rear axle actuator system with a P controller.

5.2 PID controller design

Simplicity of design and implementation was the motivation behind the choice of evaluating variants of a proportional-integral-derivative (PID) controller. A PID controller is a control loop output feedback mechanism widely used in industrial control systems. The need for P, I and D actions were evaluated one by one. First step was to design a proportional controller.

5.2.1 P controller

Figure 5.1 shows how the proportional controller is implemented. The controller output is saturated at $11Nm$. Simulink Control Design was used to tune the controller and find the optimal P value. A screenshot of PID tuner, seen in figure 5.2, exemplifies how the slider for response time and robustness can be varied to find the optimal gain values that would satisfy the controller requirements specified in section 5.1.

Tuned design parameter values obtained from PID tuner in Simulink for the P controller are listed in table 5.1. It can be observed that the closed loop system poles have been shifted away from the imaginary axis which increases the system response time.

P	409.073
Rise time, T_r	$84ms$
Settling time, T_s	$151ms$
Overshoot	0%
Steady-state error	0.1%
Closed loop system poles	-25.7, -1156.2

Table 5.1: P controller parameters

Step response of the closed loop non-linear system with P controller is shown in figure 5.3. The response parameters vary slightly from values in table 5.1 because

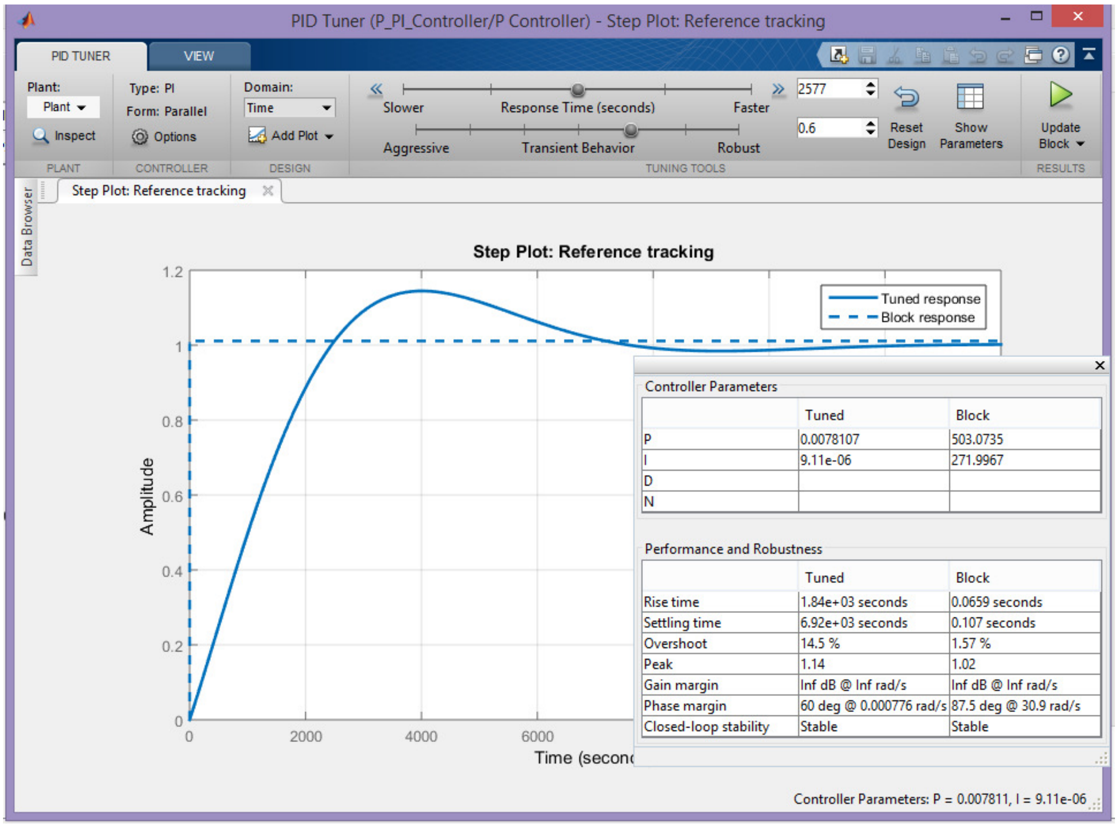


Figure 5.2: Screenshot of PID tuner.

the values listed in the table are for the linearized model. The control signal generated for step input is shown in figure 5.8 and is below the upper limit of $11Nm$.

The next step was to evaluate the controller performance with different inputs. Figure 5.3 also shows the performance of the controller for a sine input and for an input sequence that approximates a typical steering manoeuvre executed when taking turns. As can be seen, there is a delay in the output following the input due to the small steady-state error accumulating, which could be improved by increasing the proportional constant. But that would result in very high control signals. Since the output of the controller is saturated at $11Nm$, this would in turn give rise to oscillations. Therefore, it is better to minimize the steady state error using integral action.

5.2.2 PI controller

Again using the Simulink Control Design toolbox, the P and I parameters were tuned for the linearized model. The values can be seen in table 5.2. It can be observed that the designed controller introduces an additional pole and zero in the closed loop system. The closed loop system poles being away from the imaginary axis and the zero being very close to the imaginary axis both contribute to a faster response time. The step response is seen in figure 5.4. Again as mentioned before the step response varies slightly from values in table 5.2. Now, it is worth noting that the PI response is much better than P in terms of rise time and settling time owing to the higher control signal as seen in figure 5.8. But it takes a long time for the steady state error to reduce to zero. As a result, the PI controller performance is better than that of P controller when the set point is changing, but it has a small error when the input is held constant for short periods of time. This can be observed in figure 5.4. Overall though, the performance is better than P controller.

P	503.0735
I	275.9967
Rise time, T_r	$65.9ms$
Settling time, T_s	$107ms$
Overshoot	1.57%
Steady-state error	$\approx 0\%$
Closed loop system poles	-0.6, -31.2, -1150.2
Closed loop system zeros	-0.5407

Table 5.2: PI controller parameters

The purpose of D action is to reduce overshoot and oscillations, both of which

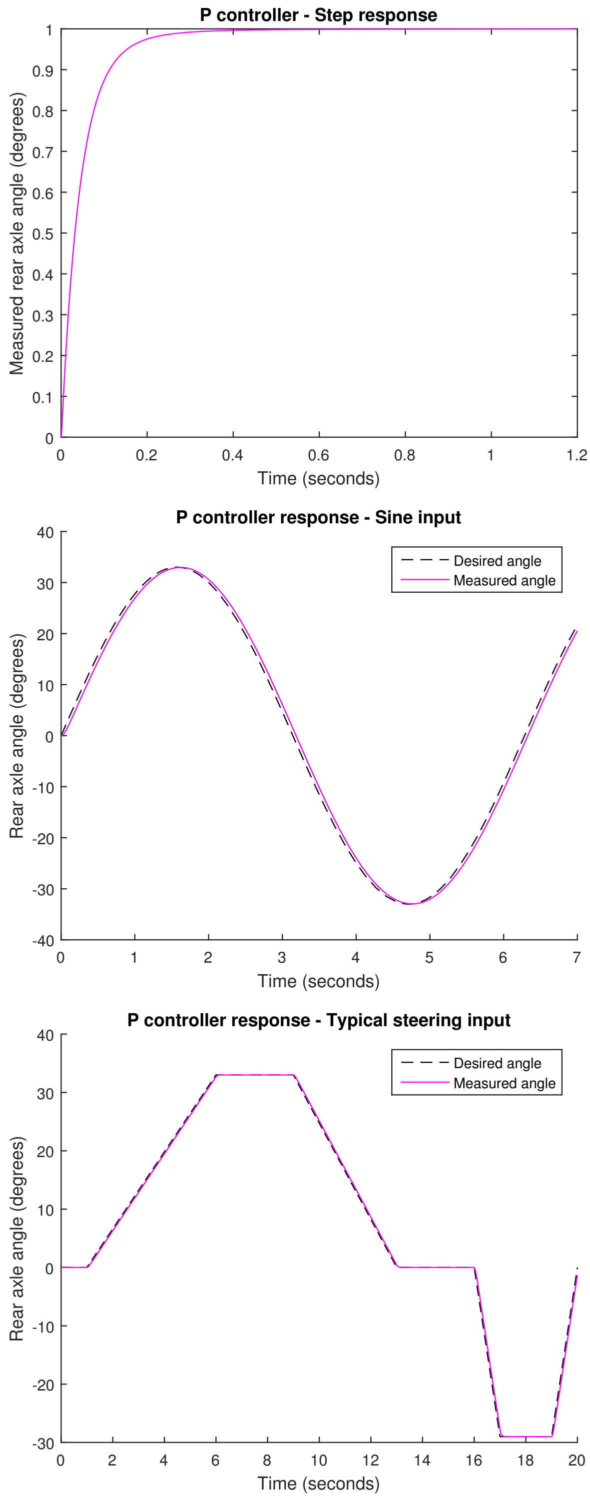


Figure 5.3: Performance of P controller for different inputs: There is a delay in the output following the input due to a small steady-state error getting accumulated.

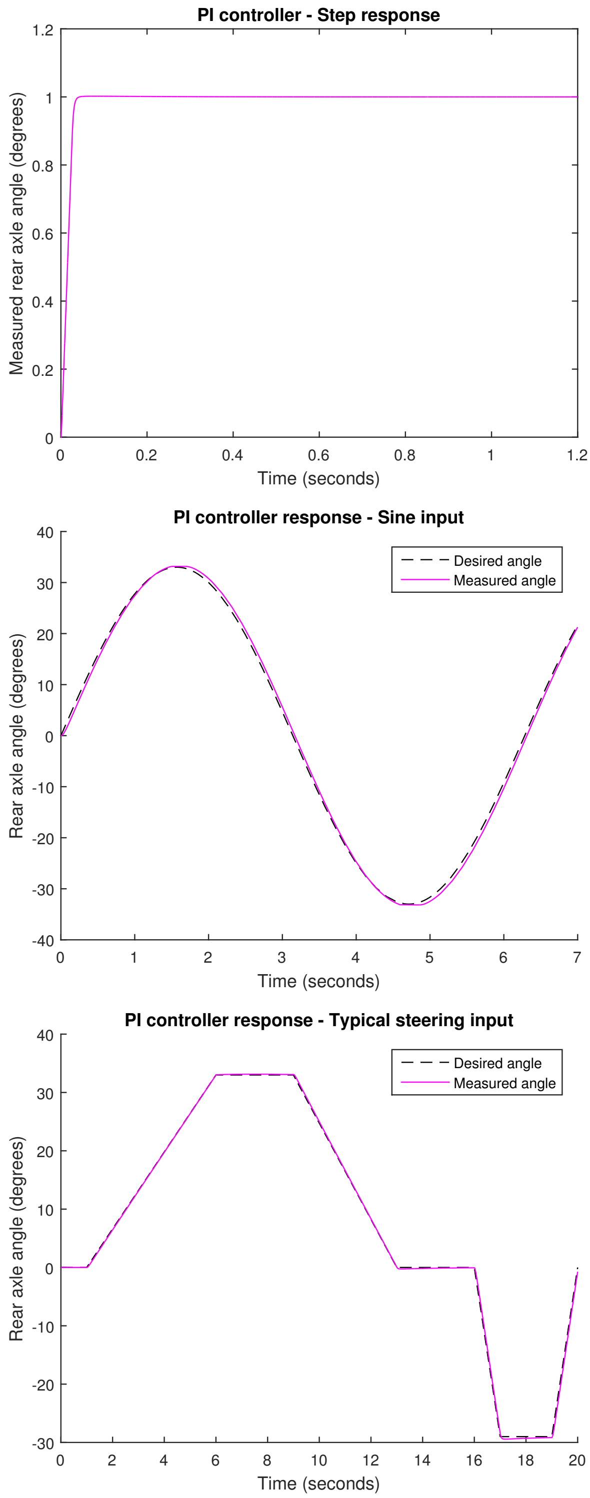


Figure 5.4: Performance of PI controller for different inputs: Steady-state error takes a long time to reduce to zero resulting in an error when the input is held constant.

are not a problem in this case. Adding a D action therefore would be of little use. Instead, a state feedback controller design was explored.

5.3 State feedback control design using pole placement

The advantage with state feedback is that it gives complete control over placement of the closed loop poles or eigenvalues, which control the characteristics of the response of the system.

5.3.1 Controllability and Observability

A system is controllable if there exists a control input, u(t), that transfers any state of the system to zero in finite time. An LTI system is controllable if and only if its controllabilty matrix, $\mathbf{P}_c$, has full rank.

$$\mathbf{P}_c = [B|AB|A^2B|...|A^{n-1}B]$$

The rank of the controllability matrix of an LTI model can be determined in MATLAB using the commands `rank(ctrb(A,B))` or `rank(ctrb(sys))`. The rank of the controllability matrix of the second-order linearized model obtained from the non-linear Simscape model and represented by equations 4.1, 4.2, 4.3 and 4.4 is equal to 2. Therefore, the system is controllable.

A system is observable if the initial state, $x(t_0)$, can be determined from the system output, $y(t)$, over some finite time $t_0 < t < t_f$. For LTI systems, the system is observable if and only if the observability matrix, $\mathbf{Q}$, has full rank.

$$\mathbf{Q} = \begin{bmatrix} C \\ CA \\ CA^2 \\ \vdots \\ CA^{n-1} \end{bmatrix}$$

The observability of an LTI model can be determined in MATLAB using the command `rank(obsv(A,C))` or `rank(obsv(sys))`. The rank of the observability matrix of the linear system represented by equations 4.1, 4.2, 4.7 and 4.4 is equal to 2. Therefore, the system is observable.

5.3.2 Pole placement

Figure 5.5 shows a general block diagram of a single-input, single-output state feedback control design for a system with zero feedforward matrix. K is the control matrix.

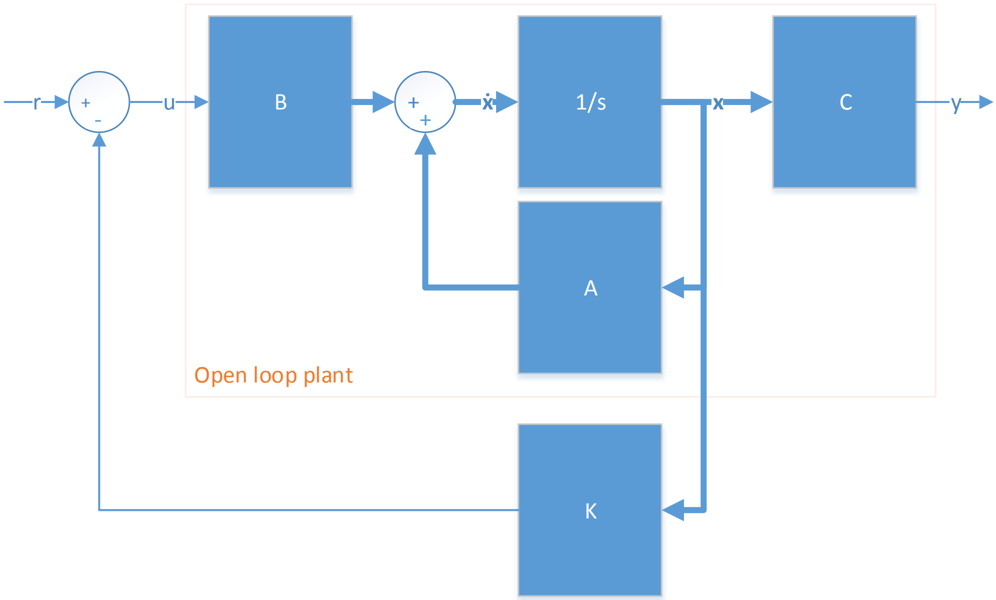


Figure 5.5: Single-input, single-output full-state feedback system: K is the control matrix.

The state-space equations for the closed-loop feedback system are

$$\dot{\mathbf{x}} = (\mathbf{A} - \mathbf{BK})\mathbf{x} + \mathbf{Br}$$

$$\mathbf{y} = (\mathbf{C} - \mathbf{DK})\mathbf{x} + \mathbf{Dr}$$

The dimensions of matrix K are such that the dimension of matrix BK is the same as the dimensions of matrix A. The stability and time domain performance of the closed-loop feedback system are determined primarily by the location of the poles/eigenvalues of the matrix $(A - BK)$. Since the matrices A and BK are both 2 by 2 matrices, there will be 2 poles for the system. By choosing an appropriate K matrix, these closed-loop poles can be placed anywhere. Therefore, the first step is to decide where the closed-loop poles are to be placed.

The general form of the transfer function of a second-order system is

$$G(s) = \frac{k_{dc}{\omega_n}^2}{s^2 + 2\zeta\omega_n s + {\omega_n}^2} \tag{5.1}$$

where k_{dc} is the DC gain, ζ the damping ratio and ω_n the natural frequency of the system.

The percent overshoot, $\%OS$ is directly related to the damping ratio by the following equation:

$$\zeta = \frac{-ln(\%OS/100)}{\sqrt{\pi^2 + (ln(\%OS/100))^2}} \tag{5.2}$$

As per the controller requirements listed earlier, the $\%OS$ should be 2% or less. Substituting the value of $\%OS$ as 0.02 in equation 5.2, we get $\zeta = 0.8$

The 2% settling time, T_s, is related to the damping ratio and natural frequency by the following equation:

$$T_s = \frac{3.9}{\zeta\omega_n} \tag{5.3}$$

Plugging in the values of $150ms$ and 0.8 for T_s and ζ respectively in equation 5.3, we get $\omega_n = 325$.

The desired poles of the closed-loop system with observer are then obtained by using the values of ζ and ω_n, as obtained above, in equation 5.1 and equating the denominator to zero. The poles thus obtained are:

$$p1 = -260 + 390i$$

$$p2 = -260 - 390i$$

The MATLAB function `place(A, B, [p1 p2])` is then used to find the control matrix, K, which will give the desired poles. Thus,

$$K = \begin{bmatrix} 42935 & -129.3529 \end{bmatrix}$$

The reference input $r(t)$ needs to be scaled appropriately to make it comparable to $Kx(t)$. This scale factor is called $\bar{N}$ and can be calculated from MATLAB by using the function `rscale`.

$$\bar{N} = 3026.6$$

5.3.3 Observer design

Since the states representing piston displacement and velocity cannot be measured directly in the real system, an observer is to be built to estimate the states, while measuring only the output y. The Simulink model for a state-feedback controller with observer can be seen in figure 5.6.

The observer is basically a copy of the plant; it has the same input as the plant. It is constructed using the same state-space matrices as the plant and an additional term which is the observer gain, L. The observer gain compares the actual measured output y to the estimated output $\hat{y}$; this will cause the estimated states $\hat{x}$ to approach the values of the actual states x.

The error dynamics of the observer are given by the poles of $(A - LC)$. The observer gain L is found by using the MATLAB function `place(A', C', [p1 p2])'`.

$$L = \begin{bmatrix} -46.6579 \\ 70632 \end{bmatrix} \tag{5.4}$$

Figure 5.7 shows the controller performance for different inputs. Again, as in the case of P/PI controllers, the control matrix and observer gain here are designed based on the linearized model but the simulations in figure 5.7 have been carried out by applying the controller to the non-linear physical model. The step response thus obtained has a rise time of $163.69ms$ and nearly zero steady-state error. The control matrix and observer gain are relatively high and therefore they give high control signals at the instant the step input is applied, as seen in figure 5.8. Saturation of the control signal at $11Nm$ results in a momentary oscillation also reflected in the step response. But the state feedback controller adapts quickly unlike P/PI controllers with high gain values and does not sustain those oscillations. Therefore, in totality, it performs much better than the output feedback controllers. The output follows the input at all times without any noticeable delay for the sine input and for the typical steering input.

In addition, all three control designs were tested for robustness by changing the efficiency values of the hydraulic pump. There was no noticeable change in control performance of any of the controllers on doing so, which indicates that all the controllers are robust. However, based on the simulation plots in figures 5.3, 5.4 and 5.7, it is evident that the state feedback controller has the best performance. Therefore, it is the state feedback controller that is used in the Simulink implementation of the entire rear axle steer-by-wire system and the vehicle simulator.

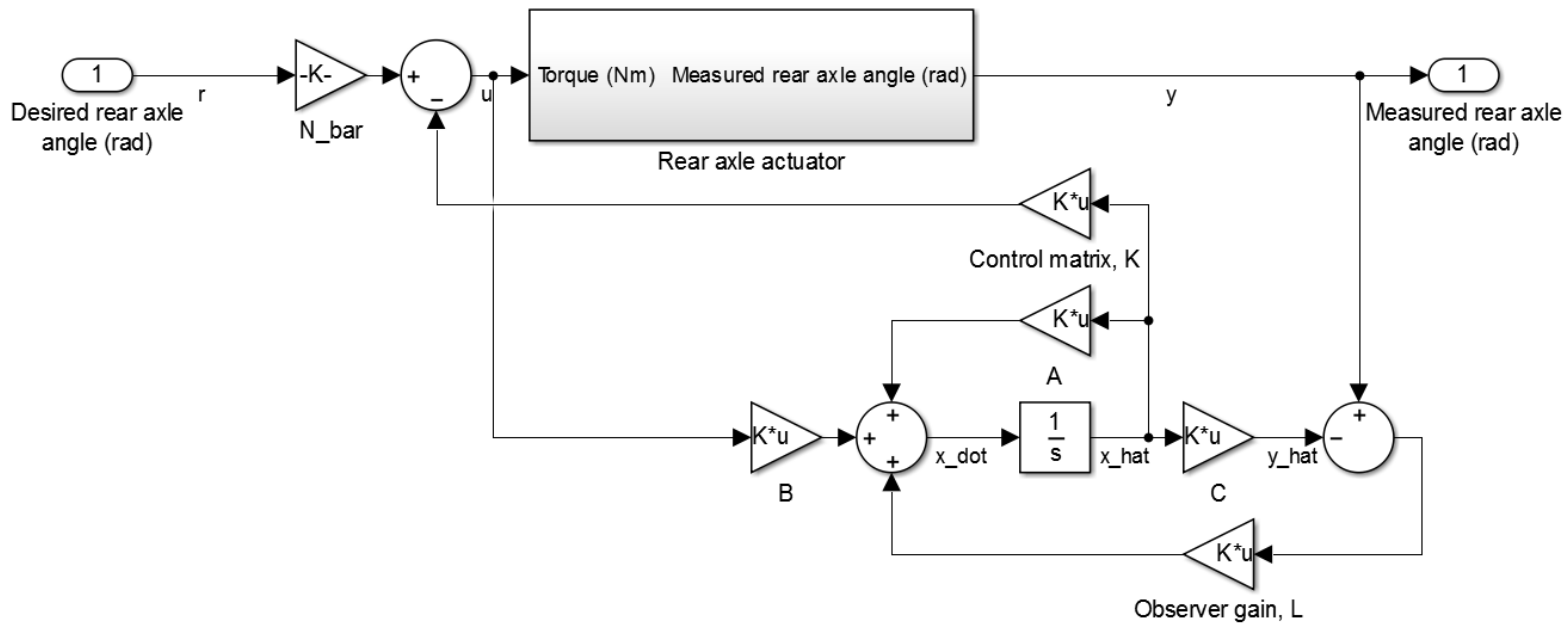


Figure 5.6: Simulink implementation of state feedback controller with observer.

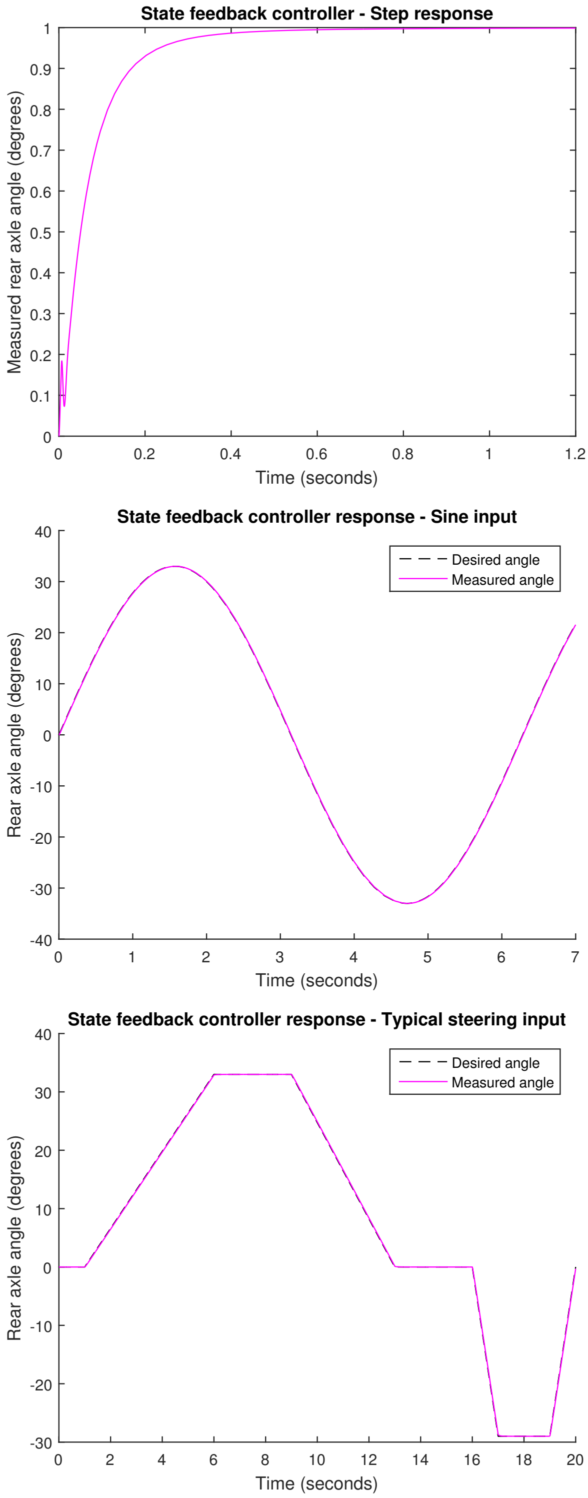


Figure 5.7: Performance of state feedback controller for different inputs.

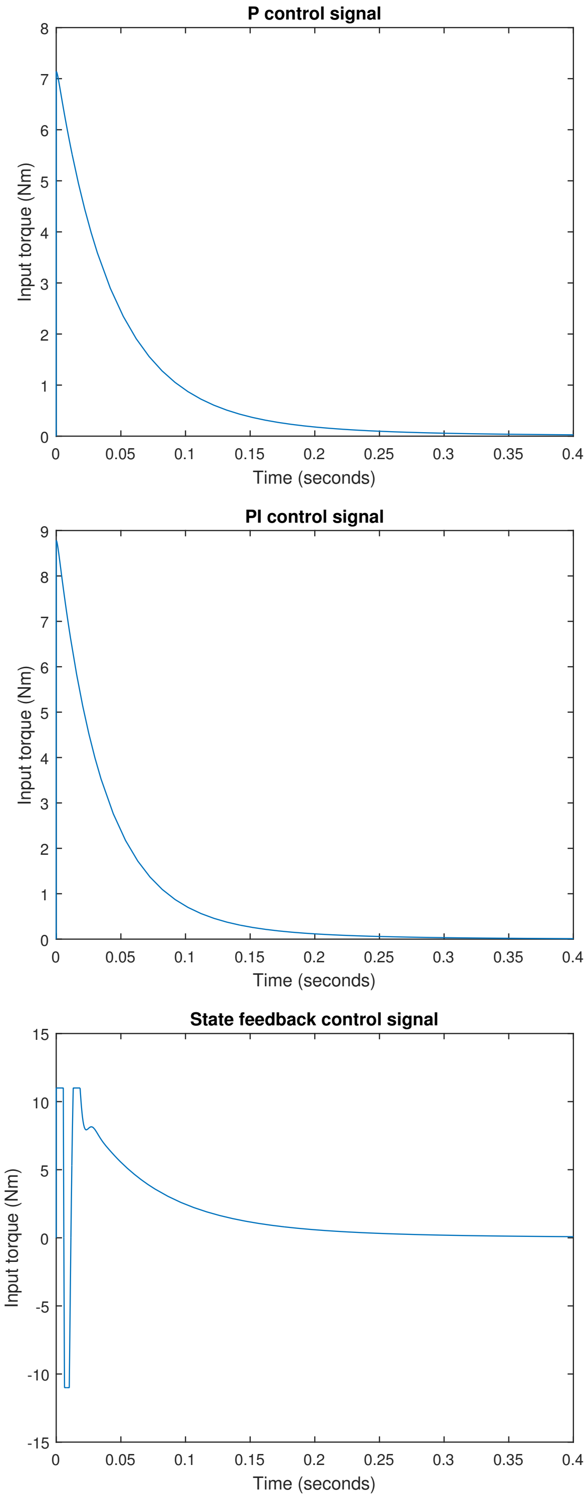


Figure 5.8: Control signal generated by different controllers for step input.

Chapter 6

Simulation of controlled rear axle steering

So far, lateral dynamics of a 4-wheel vehicle and a rear axle actuator system have been modelled as explained in chapters 3 and 4, and a controller has been designed for the rear axle actuator. This is a good time to have a look at figures 2.1 and **??** again. Before the different blocks can be integrated together and the vehicle movement simulated, assigning of a steering mode and calculating the rear axle angle need to be taken care of.

6.1 State machine for mode switching

A state diagram for switching between the different steering modes can be seen in figure 6.1. The rear steering mode, being more of a concept for the future, has been excluded from the state diagram.

The state diagram in figure 6.1 was realized using the Stateflow environment in Simulink. Figure 6.2 shows a test plot of the state machine. While normal and crab modes are always activated upon driver request, clamp mode requires an additional condition of speed limit to be satisfied before it can be activated. The speed limit for clamp mode was arbitrarily set to $8m/s$ (approximately $30km/h$).

Once the steering mode has been assigned, the rear axle angle is calculated based on the steering input from the driver. In the front steering mode, the rear axle angle is always zero. For the clamp and crab modes, a 1:1 relation between the front and rear axle angles is used. When the front axle angle exceeds the maximum rear axle angle limit, the rear axle angle is saturated at its maximum.

Now that all the blocks outlined in figures 1 and 2 are tested and ready, it is time to put them all together and see the results.

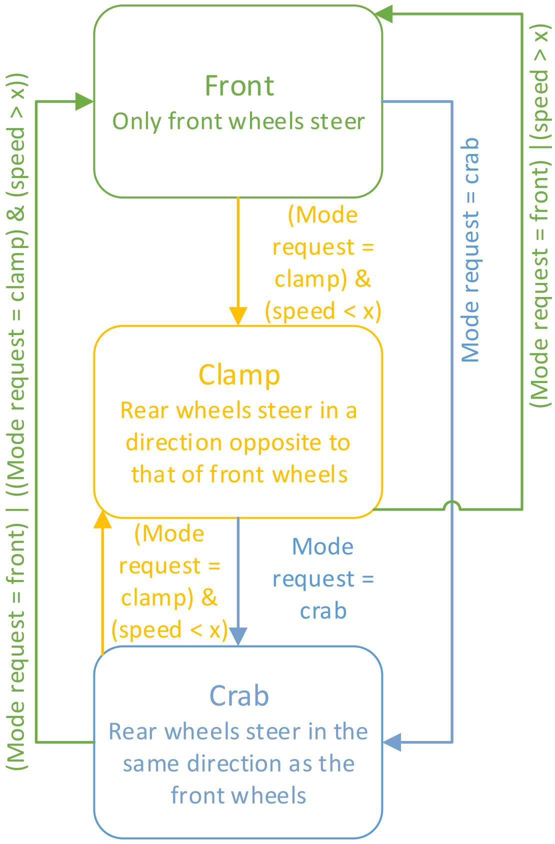


Figure 6.1: State diagram for switching between different steering modes.

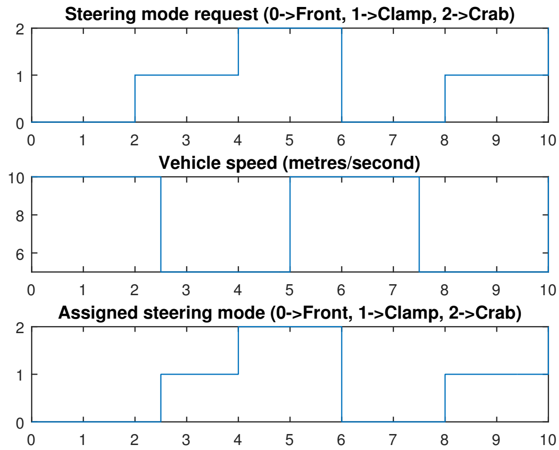


Figure 6.2: Simulation of the state diagram for mode switching: The clamp mode is not activated unless it satisfies the low vehicle speed requirement, while the other two modes only need driver input.

6.2 Vehicle simulator

A snapshot of the implementation of the entire control system, including the simulator, in Simulink can be seen in figure 6.3. The dimensions of a typical heavy vehicle, such as a city bus, are listed in table 6.1. These values were used for building the simulator using the vehicle dynamics equations discussed earlier in chapter 3.

Vehicle length	$12m$
Vehicle width	$2.5m$
Wheel diameter	$0.8m$
Distance between axles	$6m$
Maximum front axle steering angle	35^o
Maximum rear axle steering angle	29^o
Distance between front axle and front end of vehicle body	$2.6m$
Distance between rear axle and rear end of the vehicle body	$3.4m$
Front track width	$2m$
Rear track width	$2m$

Table 6.1: Vehicle dimensions used in the simulator

The simulator has three inputs – steering mode input, steering angle input and throttle input. Interfacing a joystick with push-buttons to provide the inputs for the simulator helped in getting a feel of the different modes. However, to compare the different modes and illustrate the advantages of the clamp and crab steering modes, it was easier to provide inputs to the simulator from within Simulink.

Figure 6.4 shows a plot of a typical heavy vehicle making a circle with its front axle steered to a maximum of 35^o in front and clamp steering modes. The magenta and green lines indicate the trail of the innermost and outermost points on the vehicle body as it goes in a circle.

As per the legal requirements, any $12m$ long vehicle should be able to turn in a swept circle of inner radius $5.3m$ and outer radius $12.5m$ as shown in figure 6.5 [14]. Looking again at figure 6.4, it can be seen that the legal requirement is satisfied in front steering mode. In fact, this legal requirement is part of the reason why $12m$ long vehicles are designed to have a wheelbase of $6m$.

A vehicle with a longer wheelbase usually rides smoother both on and off the road. The lengthy distance between the front and rear wheels means that riding over any bumps creates less of a pitch in the vehicle's horizontal ride line compared to a short wheelbase vehicle. This in turn provides a more comfortable ride for the passengers inside the vehicle [1].

In figure 6.4, the clamp steering mode turns much more than that required by legal requirements, which is very good since it greatly enhances the vehicle's manoeuvrability. But suppose the focus was on making travel more comfortable for the passengers as would be the case for an inter-city bus driving on a highway. Then rear axle steering and clamp mode provide the option to increase the wheelbase of the vehicle for the same vehicle length and still be able to satisfy the legal requirement for turning radius. Figure 6.6 demonstrates a $12m$ long vehicle with an increased wheelbase of $10m$ in clamp steering mode making a circle within the boundaries of legal turning radius requirements.

The manoeuvre of changing lanes is illustrated in figure 6.7 for front and crab steering modes. A clear advantage with the crab mode is that the road length covered and the time taken in changing lanes is shorter compared to that of front steering mode for the same vehicle speed.

A more significant advantage of crab mode is that it provides better yaw stability than front steering mode. The change in yaw rate with vehicle speed in the different cases can be seen in figure 6.8.

The complete design and implementation of the rear axle steer-by-wire system along with simulation results at different stages have now been explained.

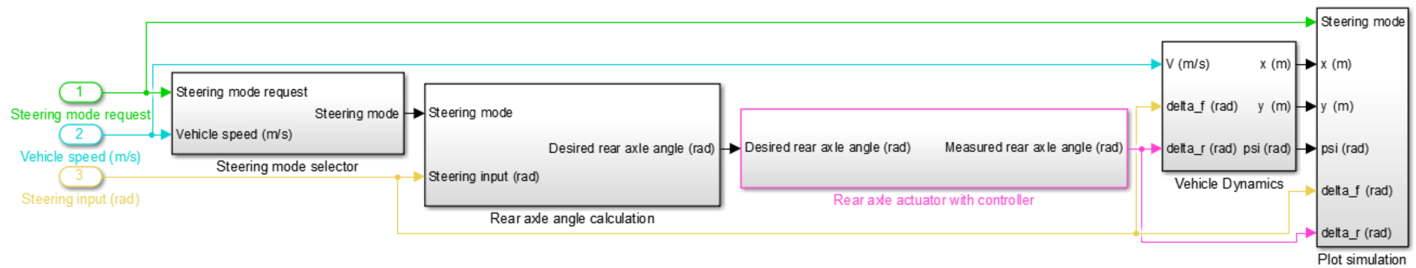


Figure 6.3: Simulink implementation of the complete rear axle steer-by-wire system including the vehicle simulator: There are 3 inputs which maybe provided either from within Simulink or by interfacing with external hardware like a joystick.

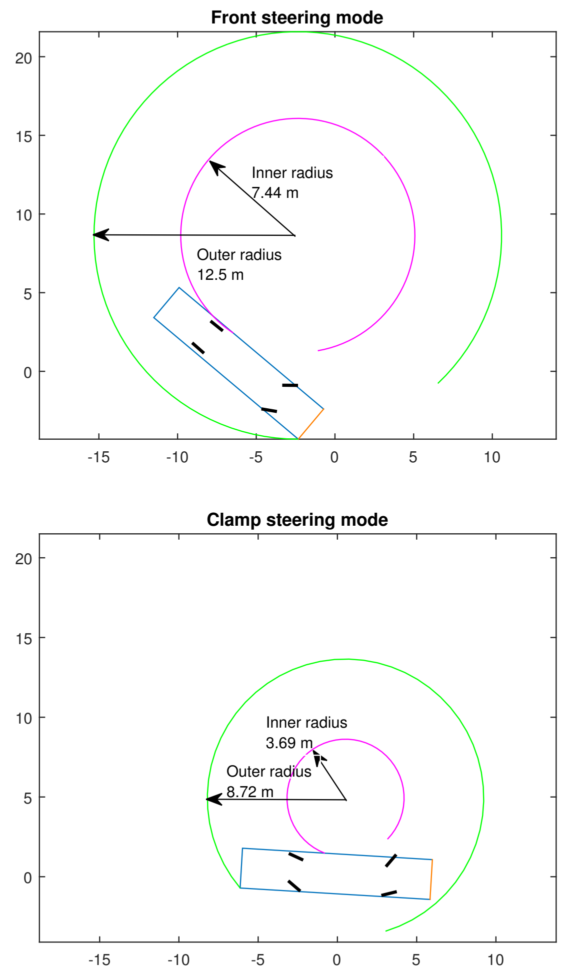


Figure 6.4: Inner (magenta) and outer (green) turning radius of a typical heavy vehicle when the wheels are steered to their maximum in front and clamp steering modes. Red line indicates front end of the vehicle.

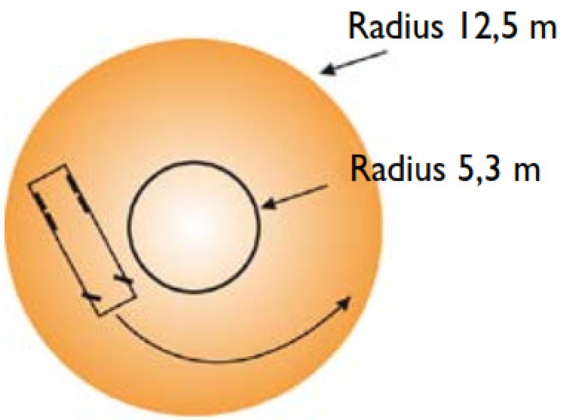


Figure 6.5: Turning in a swept circle legal requirement for any power-driven 12m long vehicle (Transportstyrelsen, 2010).

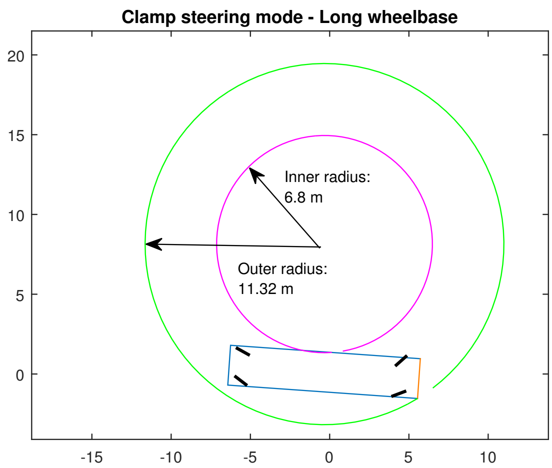


Figure 6.6: Inner (magenta) and outer (green) turning radius of a 12m long vehicle with longer wheelbase in front clamp steering mode. Red line indicates front end of the vehicle.

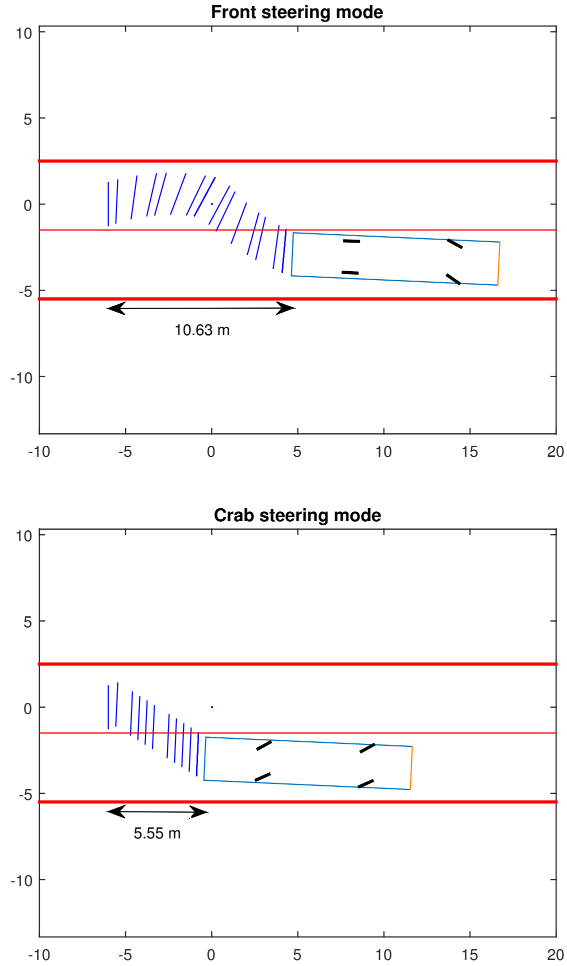


Figure 6.7: Lane changing manoeuvre in a typical heavy vehicle in front and crab steering modes: Not only does the crab mode have better yaw stability but also takes approximately half the time to change lanes as compared to using front steering mode. Simulations of both cases were carried out at a vehicle speed of $5m/s$. Red line indicates front end of the vehicle. The dark blue lines represent the yawing of the vehicle's centre of gravity as it moves from left to right.

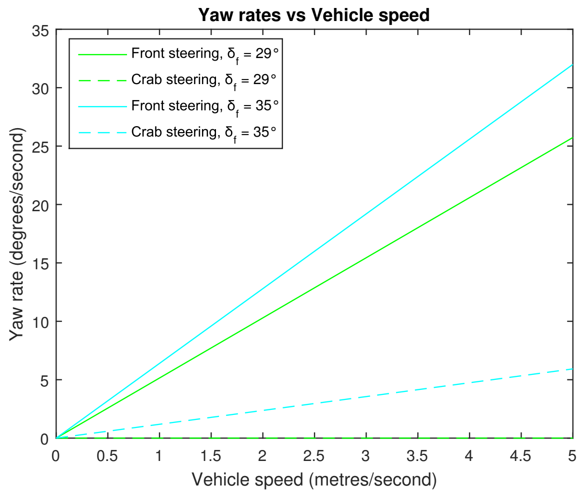


Figure 6.8: Yaw rate vs Vehicle speed for front and crab steering modes: δ_f is the steering input or the front steering angle. Up until a value of $\delta_f = 29^o$, the yaw rate for crab mode is zero due to the 1:1 relation between front and rear axle angles.

Chapter 7

ISO 26262: Road vehicles - Functional safety

Automobile development has seen an increase in technological complexity, software content and mechatronic implementation in recent years. As stated in [6], the continuing trend points to increased risks from systematic failures and random hardware failures making safety a key issue of future automobile development.

According to [15], the safety aspects of any electrical and/or electronic (E/E) system may be divided into:

- **Primary safety:** This refers to hazards that may be caused directly by the system's hardware. It includes dangers from electrocution or electric shock, and from burns or fire.

- **Functional safety:** This is related to the correct functioning of the hardware and software, including aspects concerned with equipment that is directly controlled by the system.

IEC 61508, titled Functional Safety of Electrical/Electronic/Programmable Electronic Safety-related Systems (E/E/PE, or E/E/PES), is an international standard of rules intended to be a basic functional safety standard applicable to all kinds of industry. ISO 26262, titled Road vehicles - Functional safety, is the adaptation of IEC 61508 to comply with needs specific to the functional safety of E/E systems within road vehicles. It also provides a framework within which safety-related systems based on other technologies, e.g. hydraulic, can be considered.

The standard applies to all activities during the safety lifecycle of safety-related systems comprised of electrical, electronic and software components. It provides appropriate requirements and processes that function as guidance to avoid risks. ISO 26262 consists of 10 parts but in this thesis the focus has been only on part 3 - Concept phase.

7.1 Part 3: Concept phase

The requirements for concept phase for automotive applications as per the ISO 26262 standard include:

- Clause 5 Item definition
- Clause 6 Initiation of the safety lifecycle
- Clause 7 Hazard analysis and risk assessment
- Clause 8 Functional safety concept

The definition of an item and many other terms used in ISO 26262 are listed in appendix C.1. Table C.1 gives an overview of the concept phase. Initiation of the safety lifecycle was ignored because its results are not required for the subsequent steps leading up to functional safety concept.

7.1.1 Item definition

All engineering projects start out with a set of requirements from the customer which dictate what tasks need to be performed by the end-product. The first step in the design process is to prepare a requirements document that states the needs of the customer in an unambiguous manner.

The requirements for a given system should include the functions to be performed by that system. Functional requirements describe the service that the system will deliver, e.g. the user interface and how the system responds to inputs from the user. Functional requirements comprise of primary and secondary functionality. While primary functionality is the service delivered in response to normal inputs, secondary functionality is the service delivered in case of incorrect input and/or in the presence of faults and errors.

A separate safety requirements document may also be produced which details what is required of the system to ensure adequate safety including more general system characteristics that have implications for the safety of the system, e.g. reliability, failure modes, positioning of the unit, physical protection required or necessary administrative procedures.

The various functional and safety requirements of the system constitute the item definition, along with a set of more general system characteristics. Given below are functional and safety requirements for the particular rear axle steer-by-wire system under consideration:

Functional Requirements

- A heavy vehicle shall have a steer-by-wire solution implemented for the rear axle in order to enhance the manoeuvrability.
- The different possible steering modes, their functions and conditions for their activation must be clearly defined. The system's performance should be equally efficient in all the different modes.
- The maximum steering angle of the rear axle of a heavy vehicle with rear axle steering shall not exceed 29^o.
- In the future, the front and rear steering may be integrated into the same control system. In that case, it should be possible to control the steering entirely with an electronic HMI device like a joystick. The steering wheel may be present just as a mechanical backup in case of a failure in the steer-by-wire system.

Safety Requirements

- To make sure that the steering system will interpret the driver's intentions correctly in all situations: Suppose a vehicle parked by the side of a road is to get back on the road. The system should either be in the normal or crab mode to perform this manoeuvre. If the vehicle erroneously enters the clamp mode, the rear wheels can go on the pavement and hit the people standing/walking there.
- The speed range for the different modes must be clearly defined and the system should not activate a steering mode unless all conditions for it's activation are satisfied.
- It might be reasonable to assume that a driver would drive the vehicle only in low speed in reverse gear. However, if needed, a safety condition might be implemented in the software for the dependence of steering modes on vehicle speed in reverse also.
- Fail-safe operation: In case of a failure, the vehicle should still be able to maintain a degraded or 'limp home' functionality.
- System integrity: The integrity of a system is its ability to detect faults in its own operation and to inform a human operator. The system may be designed so that it will enter its fail-safe state if there is any uncertainty about the system's correctness. The steering modes should be entirely controlled by the driver if there was any doubt about the correctness of the automatic

operation of the system e.g. based on vehicle speed and/or GPS input. The driver could be informed of a failure through a message on a display.

- Data integrity: It is the ability of a system to prevent damage to its own database and to detect, and possibly correct, errors that do occur. It is of importance when the current state of the system is stored e.g. when using push buttons to change steering modes. A corruption of this data could destroy the system's image of its own state and thus affect its operation. Since transition to a mode may require different conditions to be satisfied, data integrity becomes very important.

7.1.2 Hazard analysis and risk assessment

The purpose of hazard analysis is to identify the hazards associated with a safety-critical system, and all events that may lead to a hazard. Even though described only in the concept phase document of ISO 26262, it is important that hazard analysis be conducted throughout the development lifecycle of a product.

There are a number of techniques for hazard analysis that provide insight into the characteristics of the system under investigation. Among the most widely used analytical techniques are:

- failure modes and effects analysis (FMEA)
- failure modes, effects and and criticality analysis (FMECA)
- hazard and operability studies (HAZOP)
- event tree analysis (ETA)
- fault tree analysis (FTA)

The method chosen here was FMEA. It is a bottom-up modular approach that progressively selects the individual components or functions within a system and investigates their possible modes of failure. It then considers possible causes for each failure mode and assesses their likely consequences. The effects of the failure are determined for the unit itself and for the complete system.

The technique may be applied at various stages of a development project. It is often used early in the lifecycle, when it can be useful in the determination of the required safety integrity level. It can also be applied at a fairly late stage, after much of the design work has been done. Here it may be applied at either a component or functional level. In this thesis, failure modes were decided based on functions rather than components because the analysis had to be carried out at a early phase of the project when the exact hardware components were not finalized.

Because it considers all component/function failures, the FMEA is particularly good at detecting conditions where a single failure can result in a dangerous situation. However, the technique does not normally consider multiple failures. Unfortunately, because the analysis looks at the effects of all component failures, much work is associated with failures that do not result in hazardous conditions. FMEA involves much detailed, demanding work and is therefore expensive to apply to large complex systems in their entirety. Manual FMEA analysis can be time-consuming and there is always a risk for human errors.

Risk assessment predicts the probability and severity of accidents. All hazardous events identified using FMEA (i.e. the failure modes) are classified based on severity, probability of exposure regarding operational situations, and controllability. The classification tables for the hazards as defined in ISO 26262 can be found in appendix C.2.2.

The next step after hazard classification is to determine the automotive safety integrity level (ASIL) of the system as defined in Table C.5. Four ASILs are defined: ASIL A, ASIL B, ASIL C and ASIL D, where ASIL A is the lowest safety integrity level and ASIL D the highest one. In addition to these four ASILs, the class QM (quality management) denotes no requirement to comply with ISO 26262.

A safety goal is then determined for each hazardous event with an ASIL evaluated in the hazard analysis. Safety goals are not expressed in terms of technological solutions, but in terms of functional objectives. They can include features such as the fault tolerant time interval, or physical characteristics (e.g. a maximum level of unwanted steering-wheel torque) if they were relevant to the ASIL determination. The ASIL determined for the hazardous event is assigned to the corresponding safety goal. If similar safety goals are determined, these may be combined into one safety goal and the highest ASIL should be assigned to the combined safety goal.

Table 7.1 shows the FMEA for the rear axle steer-by-wire system. The analysis has been done considering that only normal, clamp and crab modes exist. Table 7.2 details the classification of hazards based on severity, probability of exposure and controllability, and determination of ASIL. Safety goals are then assigned in table 7.3 for the hazards that have ASIL A or higher.

7.1.3 Functional safety concept

To comply with the safety goals, the functional safety concept contains safety measures to be implemented in the item's architectural elements and specified in the functional safety requirements. The functional safety concept addresses fault detection and failure mitigation, transitioning to a safe state, fault tolerance mechanisms, fault detection and driver warning in order to reduce the risk

Failure mode	**Possible cause**	**Local and System effects**
Rear axle not steering in Clamp and/or Crab mode	Loss of communication Failure in the controller Actuator failure Failure in HMI device Battery failure	Negligible
Rear axle steering in the wrong direction in clamp mode	Communication error Controller error Software error	The vehicle goes into crab mode when the driver inputs clamp mode. Dangerous failure.
Rear axle steering not 1:1 with front axle in clamp mode	Communication error Controller error Actuator fault Angle sensor error Software error	Negligible
Rear axle steering in the wrong direction in crab mode	Communication error Controller error Software error	The vehicle goes into clamp mode when the driver inputs crab mode. Dangerous failure.
Rear axle steering not 1:1 with front axle in crab mode	Communication error Controller error Actuator fault Angle sensor error Software error	Negligible
The maximum that the rear axle turns is less than the allowed maximum of 33^o	Controller error Actuator fault Angle sensor error Software error	Negligible
The rear axle turns more than the allowed maximum of 33^o	Controller error Actuator fault Angle sensor error Software error	This could damage or break some mechanical parts/linkages and result in breakdown of the entire vehicle. Dangerous failure.
The vehicle goes into clamp or crab mode without any input from the driver	Software error	Driver has no control over the steering of the vehicle. Dangerous failure.

Table 7.1: FMEA for rear axle steer-by-wire system

Hazard	**Severity**	**Probability of exposure**	**Controllability**	**Integrity level**
Rear axle not steering in clamp and/or Crab mode	S0	E1	C0	QM
Rear axle steering in the wrong direction in clamp mode	S2	E1	C1	QM
Rear axle steering not 1:1 with front axle in clamp mode	S0	E3	C0	QM
Rear axle steering in the wrong direction in crab mode	S2	E1	C1	QM
Rear axle steering not 1:1 with front axle in crab mode	S0	E3	C0	QM
The maximum that the rear axle turns is less than the allowed and designed maximum of 33^o	S0	E1	C0	QM
The rear axle turns more than the allowed maximum of 33^o	S2	E4	C3	ASIL B
The vehicle goes into clamp or crab mode without any input from the driver	S3	E1	C3	ASIL A

Table 7.2: Classification of hazards and ASIL determination

Safety goal	**ASIL**
Maximum level of rear axle steering angle shall be limited to 24^o.	ASIL B
Rear axle shall never steer without any input from the driver.	ASIL A

Table 7.3: Determination of safety goals

exposure time to an acceptable interval (e.g. steer-by-wire fault warning lamp), and arbitration logic to select the most appropriate control request from multiple requests generated simultaneously by different functions.

The functional safety concept for the rear axle steer-by-wire system is as follows:

- It is important that the angular position of the axle is known precisely so that the system can respond appropriately to any input from the driver. Arbitration logic could be used to measure the angle using two different sensors. Redundancy makes it easier to detect and mitigate faults. Additionally, the software should have a very good filter for sensor data that can cancel the effects of noise and disturbances.

- Saturate the desired angle value generated based on driver input to a maximum of 24^o and input only the saturated value to the controller. Saturate the controller output again to make sure that the torque being output to the pump does not exceed the value that would keep the axle at 24^o. There could also be a mechanical stop in place to limit the axle angle.

- If push buttons are used for mode selection, the software should clearly recognize when a button has been pushed intentionally and when it was touched by mistake. Alternately, some other kind of switches may be used to avoid any ambiguity. The software should also store the current state of the system and have mechanisms to prevent data corruption. Additionally, there could be a display on which the driver can read all relevant information about the current state of the system so that he/she knows if the system has not been able to understand his/her intentions correctly.

- If a fault is detected, the system should deactivate rear axle steering immediately.

The above functional safety concept concludes the concept phase functional safety analysis for the rear axle steer-by-wire system.

Chapter 8

Discussion and Conclusions

This thesis work concerned designing a rear axle steer-by-wire system for a heavy vehicle. It included modelling of a hydraulic rear axle actuator system and designing a controller for it. Three different controllers – P, PI and state feedback – were evaluated. The step response of all three controllers satisfied the design criteria. But when tested with other input sequences, the state feedback controller clearly outperformed the other two controllers. The non-linear physical nature of Simscape ensured an accurate model for the rear axle actuator. The way forward is to conduct model-in-the-loop tests of the controller with real hardware and improve the control design. The integration of front and rear axle steering within the same electronic system would play a key role in coordinating the vehicle handling more effectively. For instance, Yih and Gerdes [16] present a method for altering a vehicle's handling characteristics by augmenting the driver's steering command with full vehicle state feedback.

The other significant contribution of this thesis was to use lateral vehicle dynamics equations of motion to build a simulator and test the different steering modes. The plots from the simulator demonstrated the turning radius and stability advantages that come with rear axle steering. Concept phase functional safety analysis presented a safety concept that would ensure a safe demonstration of the system. Functional safety is a continuous process and will have to be carried out at every stage in the production cycle.

There are many more things to be taken care of before rear axle steering can be implemented in any production vehicle. As mentioned earlier in section 1.3, mode switching operation was not paid any special attention since the initial tests are to be carried out at low vehicle speed. However, it's not the same if the vehicle was moving at a relatively higher speed. Suppose the vehicle is in crab mode and steered to it's maximum, and the driver changes the steering mode to clamp mode. Even if the driver holds the steering wheel in the same position, the effect on the rear wheels is that they have to turn from a maximum in one direction to the maximum in another direction very quickly and suddenly. The safest way to switch modes might then be to do so only after stopping or considerably slowing down the vehicle. If the mode switching operation is to be made possible while driving the vehicle at higher speeds, which would quite likely be expected of production vehicles in the future, then obviously there must be some kind of safety mechanism to ensure smooth switching between modes.

Further still, it must be noted that one of the major impediments of the proposed rear axle steer-by-wire solution is that the driver needs to get used to driving with different modes. Drivers' acceptance of the technology is very important and hence it is necessary to make it as easy for them as possible. One way could be to make the mode switching process automatic.

Certain vehicles, such as city buses, usually have well-established bus routes in the city and they follow the same route everyday. In such a scenario, it is possible to activate mode switching based on GPS data depending on availability. For example, at certain junctions in the city, it may help the vehicle to be in clamp mode. Then, there could be an automated system that switches the vehicle to clamp mode even if the driver has forgotten to activate clamp mode.

However, one must exhibit caution when implementing such an automated system. The automated system must be allowed to change modes only if the vehicle speed is below a certain limit. Otherwise, it's best to assume that the driver knows what he/she is doing. The system could perhaps display a warning message to the driver when it takes control because the vehicle speed is low enough. Another precaution measure could be to make the rear axle angle smaller when it is GPS activated so that it assists in driving without taking the driver by surprise.

While GPS-based mode switching may be implemented for city buses following a fixed-route everyday, the technique is not universal and may be overly complicated to implement in other heavy vehicles. An alternate solution could be to develop an algorithm which, depending on how much the driver steers and the vehicle speed, determines if switching to crab or clamp mode will improve vehicle handling. For example, if the driver provides large steering input at low speed, it is likely that the vehicle needs more turning angle and the system may switch to clamp mode. On the other hand, for vehicles that are driven more on the highways, e.g. trucks and inter-city buses, crab mode may be activated only at higher speeds for stability. It is essential to collect data and find the right balance between steering wheel angle, vehicle speed and any other input that may be used. Any automatic system will have to be tailored to the needs of different categories of vehicles.

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Appendices

Appendix A

Ackerman turning geometry

Let w be the average track width of the vehicle, and δ_{fo}, δ_{ro}, δ_{fi} and δ_{ri} be the outer and inner steering angles of front and rear wheels respectively. Let the wheelbase L be small compared to the radius R of the vehicle's path. If the slip angle β is small, then equation 3.3 can be approximated by

$$\frac{\dot{\psi}}{V} \approx \frac{1}{R} = \frac{\delta_f - \delta_r}{L} \tag{A.1}$$

or

$$\delta_f - \delta_r = \frac{L}{R} \tag{A.2}$$

Since the radius at the inner and outer wheels are different,

$$\delta_{fo} - \delta_{ro} = \frac{L}{R + \frac{w}{2}} \tag{A.3}$$

$$\delta_{fi} - \delta_{ri} = \frac{L}{R - \frac{w}{2}} \tag{A.4}$$

The average front and rear wheel steering angles are approximately given by

$$\delta_f = \frac{\delta_{fo} + \delta_{fi}}{2} \tag{A.5}$$

$$\delta_r = \frac{\delta_{ro} + \delta_{ri}}{2} \tag{A.6}$$

Equations A.3, A.4, A.5 and A.6 above can be solved to obtain the angles δ_{fo}, δ_{ro}, δ_{fi} and δ_{ri}.

Appendix B

Calculation of input torque of hydraulic pump

Parameter	Value
Pitman arm axle range	90^o
Max. Pitman arm axle torque, M_p	$6408Nm$

Table B.1: Electrical machine specifications

Since Pitman arm axle range is 90^o,

$$\begin{aligned} \text{Circumference of pinion, } \phi &= 4 \times \text{Steering unit rack travel} \\ &= 4 \times 0.0812m \\ &= 0.3248m \end{aligned} \tag{B.1}$$

$$\begin{aligned} \text{Pitch radius of pinion, } r_p &= \frac{\phi}{2\pi} \\ &= \frac{0.3248m}{2\pi} \\ &= 0.0517m \end{aligned} \tag{B.2}$$

$$\begin{aligned} \text{Hydraulic cylinder pressure, } P &= \frac{M_p}{A \times r_p} \\ &= \frac{6408Nm}{0.01075m^2 \times 0.0517m} \\ &= 11529846N/m^2 \end{aligned} \tag{B.3}$$

$$
\begin{aligned}
\text{Input torque of hydraulic pump, } M_m &= \frac{P \times D}{2\pi} \\
&= \frac{11529846 N/m^2 \times 0.000006 m^3}{2\pi} \\
&= 11 Nm
\end{aligned} \tag{B.4}
$$

Appendix C

ISO 26262: Road vehicles - Functional safety

C.1 Part 1: Vocabulary

Here is a list of some important definitions from Part 1 (Vocabulary) of ISO 26262 that have been relevant for this thesis work.

Allocation
Assignment of a requirement to an architectural element.

Architecture
Representation of the structure of the item or functions or systems or elements that allows identification of building blocks, their boundaries and interfaces, and includes the allocation of functions to hardware and software elements.

Automotive Safety Integrity Level (ASIL)
One of four levels to specify the item's or element's necessary requirements of ISO 26262 and safety measures to apply for avoiding an unreasonable residual risk, with D representing the most stringent and A the least stringent level.

Component
Non-system level element that is logically and technically separable and is comprised of more than one hardware part or of one or more software units.

Element
System or part of a system including components, hardware, software, hardware parts, and software units.

Hardware part
Hardware which cannot be subdivided.

Harm
Physical injury or damage to the health of persons.

Hazard
Potential source of harm caused by malfunctioning behaviour of the item.

Hazardous event
Combination of a hazard and an operational situation.

Item
System or array of systems to implement a function at the vehicle level, to which ISO 26262 is applied.

Failure
Termination of the ability of an element, to perform a function as required.

Failure mode
Manner in which an element or an item fails.

Fault
Abnormal condition that can cause an element or an item to fail.

Functional concept
Specification of the intended functions and their interactions necessary to achieve the desired behaviour.

Functional safety
Absence of unreasonable risk due to hazards caused by malfunctioning behaviour of E/E systems.

Malfunctioning behaviour
Failure or unintended behaviour of an item with respect to its design intent.

Operational situation
Scenario that can occur during a vehicle's life.

Random hardware failure
Failure that can occur unpredictably during the lifetime of a hardware element and that follows a probability distribution.

Redundancy
Existence of means in addition to the means that would be sufficient for an element to perform a required function or to represent information.

Risk
Combination of the probability of occurrence of harm and the severity of that harm.

Safety
Absence of unreasonable risk.

Safety case
Argument that the safety requirements for an item are complete and satisfied by evidence compiled from work products of the safety activities during development.

Safety measure
Activity or technical solution to avoid or control systematic failures and to detect random hardware failures or control random hardware failures, or mitigate their harmful effects.

Safety mechanism
Technical solution implemented by E/E functions or elements, or by other technologies, to detect faults or control failures in order to achieve or maintain a safe state (1.102).

Severity
Estimate of the extent of harm to one or more individuals that can occur in a potentially hazardous situation.

Software unit
Atomic level software component of the software architecture that can be subjected to stand-alone testing.

System
Set of elements that relates at least a sensor, a controller and an actuator with one another.

Systematic failure
Failure, related in a deterministic way to a certain cause, that can only be eliminated by a change of the design or of the manufacturing process, operational procedures, documentation or other relevant factors.

Testing
Process of planning, preparing, and operating or exercising an item or an element to verify that it satisfies specified requirements, to detect anomalies, and to create confidence in its behaviour.

Unreasonable risk
Risk judged to be unacceptable in a certain context according to valid societal moral concepts.

C.2 Part 3: Concept phase

C.2.1 Overview and document flow of concept phase

Clause	Objectives	Prerequisites	Work products
5 Item definition	The first objective is to define and describe the item, its dependencies on and interaction with the environment and other items. The second objective is to support an adequate understanding of the item so that the activities in subsequent phases can be performed.	None	5.5 Item definition
6 Initiation of the safety life-cycle	The first objective of the initiation of the safety life-cycle is to make the distinction between a new item development and a modification to a existing item. The second objective is to define the safety lifecycle activities that will be carried out in the case of a modification.	Item definition	6.5.1 Impact analysis, 6.5.2 Safety plan (refined)
7 Hazard analysis and risk assessment	The objective of the hazard analysis and risk assessment is to identify and to categorise the hazards that malfunctions in the item can trigger and to formulate the safety goals related to the prevention or mitigation of the hazardous events, in order to avoid unreasonable risk.	Item definition	7.5.1 Hazard analysis and risk assessment, 7.5.2 Safety goals, 7.5.3 Verification review report of the hazard analysis and risk assessment and the safety goals
8 Functional safety concept	The objective of the functional safety concept is to derive the functional safety requirements, from the safety goals, and to allocate them to the preliminary architectural elements of the item, or to external measures.	Item definition, Hazard analysis and risk assessment, Safety goals	8.5.1 Functional safety concept, 8.5.2 Verification report of the functional safety concept

Table C.1: Overview of concept phase

C.2.2 Clause 7 - Hazard analysis and risk assessment

Classification of hazardous events

Class	**S0**	**S1**	**S2**	**S3**
Description	No injuries	Light and moderate injuries	Severe and life-threatening injuries (survival probable)	Life-threatening injuries (survival uncertain), fatal injuries

Table C.2: Classes of severity

Class	**E0**	**E1**	**E2**	**E3**	**E4**
Description	Incredible	Very low probability	Low probability	Medium probability	High probability

Table C.3: Classes of probability of exposure regarding operational situations

Class	**C0**	**C1**	**C2**	**C3**
Description	Controllable in general	Simply controllable	Normally controllable	Difficult to control or uncontrollable

Table C.4: Classes of controllability

Determination of ASIL

Severity Class	**Probability Class**	**C1**	**Controllability Class C2**	**C3**
S1	**E1**	QM	QM	QM
	E2	QM	QM	QM
	E3	QM	QM	A
	E4	QM	A	B
S2	**E1**	QM	QM	QM
	E2	QM	QM	A
	E3	QM	A	B
	E4	A	B	C
S3	**E1**	QM	QM	A
	E2	QM	A	B
	E3	A	B	C
	E4	B	C	D

Table C.5: Determination of ASIL