بسم الله الرحمن الرحيم

PPU College Of Engineering and Technology

The Home of Competent Engineers and Researchers

Mechanical Engineering Department

Graduation Project

Develop Yaw Control System for Ground Vehicles

Project Team

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Hebron-Palestine

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Dedication

To

My Mother

A strong and gentle soul who taught me to trust in Allah, believe in hard work and that so much can be done with little

My Father

For earning an honest living for us and for supporting and encouraging me to believe in myself

Acknowledgment

We would like to thank our supervisor of this project, Dr. Momen Sugayyer for his valuable guidance and advice. He inspired us greatly to work in this project. His willingness to motivate us contributed tremendously to our project. Besides, we would like to thank the staff of mechanical engineering department especially Eng. Abdalkarim Mohtasib and our colleagues Ammar Haliqa and Ibrahim Hroub for helping us in every possible way to complete this project.

Abstract

Vehicles' Active Stability Control is intended to achieve safe and comfortable drivability response on roads. Vehicle stability usually depends on forces and vehicle interaction, in addition to many other factors such as tire-road friction, dry or wet road, steering angle and vehicle speed. Thus, it is difficult for the normal drivers to achieve optimal performance under these limits without assistance form control onboard systems, since the driver is not able to know the friction between the tire and the road and have the required response in time, which might cause more danger to the vehicle and driver.

The proposed project aims to develop a Yaw Control System, which forms a solution for this problem that will prevent the vehicle from spinning, drifting and rolling over.

This project will address this engineering issue by considering vehicle lateral and longitudinal dynamics on roads, obtaining a model for a testing prototype (sophisticated toy vehicle), design and apply a controller for the target toy vehicle (PID controller, Microcontroller, etc..), which simulate the real vehicle but at small scale, finally, the obtained results will be used for simulating real scale vehicle. This, of course, will helps in decreasing accidents and save people lives when the target control system is adopted and applied.

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CHAPTER 1

Introduction

1.1 Introduction

As the worldwide use of automobiles increases rapidly, it has become even more important to develop vehicles that provide safe and comfortable transportation. To meet these diverse and often conflicting requirements, automobiles are increasingly relying on electromechanical systems that employ sensors, actuators and feedback control.

A variety of driver assistance systems are being developed by automotive manufacturers to automate mundane driving operations, reduce driver burden and thus reduce highway accidents, like yaw stability control systems.

Vehicles' Active Stability Control is intended to achieve safe and comfortable drivability response on roads. Vehicle stability usually depends on forces and vehicle interaction, in addition to many other factors such as tire-road friction, dry or wet road, steering angle and vehicle speed. Thus, it is difficult for the normal drivers to achieve optimal performance under these limits without assistance from control onboard systems, since the driver is not able to know the friction between the tire and the road and have the required response in time, which might cause more danger to the vehicle and driver.

1.2 Project idea

The project idea is to develop a yaw control system to improve lateral stability for a toy vehicle, in other words, to develop a controller to make it follow the desired path in response to a steering input, especially when the vehicle is cornering on a road where the coefficient of friction between the road and the tires is small, then the vehicle might not follow the desired path and spinning will occur, so the yaw control system will restore the bath of the vehicle as much as possible to the nominal motion expected by the driver. Fig 1.1 shows the advantage of using yaw control system.

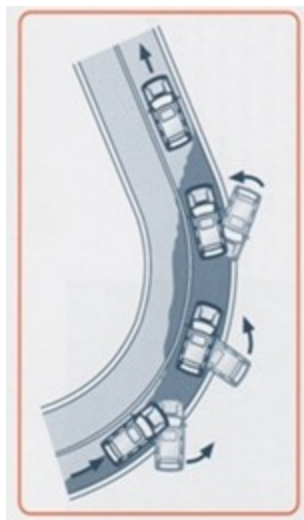


Figure 1.1 the functioning of yaw control system.

1.3 Recognition of the Need

The proposed project aims to develop a Yaw Control System, which forms a solution that will prevent the vehicle from spinning, drifting and rolling over. The project will address this engineering issue by considering vehicle lateral and longitudinal dynamics on roads, obtaining a model for a testing prototype (sophisticated toy vehicle), design and apply a controller for the target toy vehicle (PID controller, Microcontroller, etc..), which simulate the real vehicle but at small scale, finally, the obtained results will be used for simulating real scale vehicle. This, of course, will helps in decreasing accidents and save people lives when the target control system is adopted and applied.

1.4 Project Objectives

1- Develop a Yaw Control System, which will prevent the vehicle from spinning, drifting and rolling over.
2- Obtain a model for a testing prototype (sophisticated toy vehicle)
3- Design and apply a controller for the target toy vehicle.
4- Simulating this controller on a real scale vehicle.

1.5 Literature Review

Integrated vehicle dynamics control via coordination of active front steering and rear braking [1]
This paper investigates the coordination of active front steering and rear braking in a driver-assist system for vehicle yaw control. This study focuses on two main methods to control the yaw moment in order to improve vehicle handling and stability. The first one is the Active Steering (AS) system that regulates the tire slip angle and affects the vehicle handling behavior by directly modulating the generation of lateral tire forces. The second one is the Direct Yaw Control (DYC) that utilizes differential braking forces between the left and the right sides of the vehicle to produce the required corrective yaw moment.
During high lateral acceleration due to the inherent nonlinear characteristics and tire saturations, AS is no longer able to produce enough lateral force by steering to hold on the vehicle in a turn. In other words, AS cannot keep the vehicle under control when the handling limit is reached and consequently, AS performance is limited within the linear vehicle handling region (low to mid-range lateral accelerations). On the other hand, DYC is shown to be effective in both vehicle linear/nonlinear regions; however, it is only desirable for limit handling rather than for normal driving situations. This is due to the braking effect that wears out the tire and interferes with the longitudinal vehicle dynamics. Moreover, DYC causes the vehicle to slow down significantly, and this may be objectionable and not desirable for the driver.
The main goal of the proposed control system is to make the actual yaw rate, to follow the desired yaw rate. In other words, the controller must track the reference yaw rate intended by the driver through driving the tracking error between the actual and desired yaw rate to zero. Another purpose of the controller is to limit the vehicle side slip angle, to be within an acceptable region to prevent vehicle spin.
The present work deals with the design of a new vehicle chassis control scheme that integrates and coordinates rear braking and front steering. The control scheme is built on a MIMO (Multi Input Multi Output) gain scheduled controller worked out on the basis of a 2-DOF (Degree-Of-Freedom) bicycle linear planar vehicle model. Two control inputs are considered (the steering angle and the yaw moment). The whole hierarchical structure of the controller, designed in two layers:

a- The upper-level controller defines the amount of the active steer angle, and the corrective yaw moment, needed to achieve a good tracking of the yaw-rate set-point.
b- The lower-level controller converts the stabilizing yaw moment generated by the upper-level controller into an effective braking torque, and decides which wheel must be braked to counter act the undesired yaw motion.

Generalized predictive control of yaw dynamics of a hybrid brake-by-wire equipped vehicle [2]. This paper presents a theoretical development and experimental results of a vehicle yaw stability control system based on generalized predictive control (GPC) method, this method is expected to improve the yaw rate as it tries to take control action before yaw rate error grows to a level requiring larger control action. The controller tries to predict the future yaw rate of the vehicle and then takes control action at present time based on future yaw rate error. The proposed controller utilizes the insight into the yaw rate error growth when the automobile is in an under steer or over steer condition on a low friction coefficient surface in a handling maneuver. The yaw stability control method presented in this paper has advantages over the more conventional methods in that it provides more responsive and smoother control of the yaw stability due to its predictive capability. A hybrid brake-by-wire equipped vehicle was used to experimentally verify the proposed control algorithm. Experimental results show that the vehicle can be effectively stabilized in an over steer/under steer condition on a packed snow surface using the predictive controller. The vehicle speed could be kept at relatively higher value throughout the slalom maneuver.

LTV-MPC for Yaw Rate Control and Side Slip Control with Dynamically Constrained Differential Braking [3]. This paper presents a novel vehicle lateral dynamic control approach. A differential braking control law based on vehicle planar motion has been designed using a two-degrees-of-freedom vehicle model. On the basis of the estimate of tire longitudinal forces we estimate the range of lateral forces, which the tire can exert. Using these constraints a model predictive control (MPC) based on a two-track model is designed in order to stabilize the vehicle. The performances are estimated comparing the results with standard maneuvers. The simulations were performed on three different Carsim datasets cars that represent different typologies: light cars, over steering rear traction sport cars, SUV cars. The use of an MIMO controller based on MPC techniques with dynamical constraints yields the desired differential braking.

Robust vehicle yaw control using an active differential and IMC techniques [4]. This paper introduces a robust non-parametric approach to improve vehicle yaw rate dynamics by means of a rear active differential. The design of the feedback controller is performed using an enhanced internal model control (IMC) technique that is able to handle in an effective way both robustness and control variable saturation issues. Increasing the degrees of freedom for the structure to two has improved the transient behavior a feed forward control. Improvements on under steering characteristics, stability in demanding conditions such as m-split braking and damping properties in reversal steer and low friction step steer maneuvers are shown through simulation results performed on an accurate 14 degrees of freedom non-linear model of a car. Simulation results performed on an accurate model of the considered vehicle demonstrate the effectiveness of the proposed control structure. In particular, it has been shown that the achieved performances are very close to the target under steering objectives; a highly damped behavior in reversal steer and step steer maneuvers has been obtained; stability is guaranteed in presence of demanding driving conditions like m-split braking and resonance peak has been significantly reduced in the frequency response.

Lane-keeping assistance control algorithm using differential braking to prevent unintended lane departures [5]. This paper describes a hierarchical lane keeping assistance control algorithm for a vehicle. The proposed control strategy consists of a supervisor, an upper-level controller and a lower-level controller. The supervisor determines whether lane departure is intended or not, and whether the proposed algorithm is activated or not. The upper-level controller is designed to compute the desired yaw rate for the lane departure prevention, and for the guidance with ride comfort. The lower-level controller is designed to compute the desired yaw moment in order to track the desired yaw rate, and to distribute it into each tire's braking force in order to track the desired yaw moment. The slip angles of the left and right wheels are assumed to be small enough that the lateral tire force is assumed to be proportional to the slip angle at the corresponding tire. Based on this assumption, the vehicle model used in this research describes error with respect to road (lateral position error and yaw rate error), the control input is the yaw moment generated to achieve the desired lane keeping, so the driver's steering input is considered as disturbance input.

Yaw stability control of a steer-by-wire equipped vehicle via active front wheel steering [6]. A novel yaw stability control algorithm with active front wheel steering control of a vehicle equipped with a steer-by-wire system is presented in this paper. The dynamics of vehicle is described by the bicycle model using steering angle as the control input. They considered the steering control system design that decouples the yaw dynamics and lateral motion of the vehicle. These decoupled equations are then used to design a second order controller for yaw stabilization and damping with more tunable parameters to achieve optimal performance which is defined as achieving maximum lateral acceleration without yaw instability and rollover in a steering maneuver. The control objective is to provide the decoupled yaw dynamics equation more freedom to be tuned via the second order controller structure.

In this project, vehicle lateral and longitudinal dynamics on road will be considered to model the ground vehicle and apply a controller using differential braking method that utilizes differential braking forces between the left and the right sides of the vehicle to produce the required corrective yaw moment.

1.6 Time Plan

The time plan explains the stages in designing and building the system components. The section includes the first table that shows the activities and task scheduling for the first semester, while the second table shows the tasks for the second semester.

Table 1-1: Timetable for the first semester time (week)

Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Selection of the project															
Search about information															
Search for previous project															
Analysis of the machine															
Evaluating and reviewing															
Writing and typing															
Final Edit															

Table 1-2: Timetable for the second semester time (week)

Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Software tesign															
Software testing															
Hardware testing															
Interface system															
System testing															
Documentation															

1.7 Risk Management

The implementation of any project may face many risks during each stage of the project, such as: determining and analyzing the system requirements, designing, implementing and testing the whole system. This section illustrates the problems that might occur during the implementation.

1.7.1 Technology Risk

Technology risks can be classified as hardware and software risks:

- **Hardware Risks Include:**

1- Malfunctions of expensive components such as the RC car yaw sensors or any other sensitive electrical components.

2- Facing a difficulty in finding or purchasing some of the components that are not available in our country and the consequent of wasted time to wait for the components.

- **Software Risks Include:**

The program may follow undesirable algorithm, but this project algorithm was clear and understandable.

1.7.2 Risks Avoidance

- Be careful when using hardware components and using them according to their specifications.
- Use the right steps in working safely during the project development.
- Purchasing extra spare parts of the sensitive components, so alternative components will be replaced when any problem occurs.

Chapter 2

Yaw Stability Model

2.1 Introduction

2.1.1 The functioning of a stability control system

Vehicle stability control systems that prevent vehicles from spinning and drifting out have been developed and recently commercialized by several automotive manufacturers. Such stability control systems are also often referred to as yaw stability control systems or electronic stability control systems.

A yaw rotation is a movement around the vertical axis of a vehicle that changes the direction the vehicle is facing, to the left or right of its direction of motion. The yaw rate or yaw velocity of a car is the angular velocity of this rotation, it is commonly measured in degrees per second or radians per second. Fig 2.1 shows the direction of yaw motion in a car.

Figure 2.1 Yaw motion in a vehicle

Fig 2.2 schematically shows the function of a yaw stability control system. In this figure, the lower curve shows the trajectory that the vehicle would follow in response to a steering input from the driver if the road were dry and had a high tire-road friction coefficient. In this case the high friction coefficient is able to provide the lateral force required by the vehicle to negotiate the curved road. If the coefficient of friction were small or if the vehicle speed were too high, then the vehicle would not follow the nominal motion expected by the driver – it would instead travel on a trajectory of larger radius (smaller curvature), as shown in the upper curve of Fig 2.2.

The function of the yaw control system is to restore the yaw velocity of the vehicle as much as possible to the nominal motion expected by the driver. If the friction coefficient is very small, it might not be possible to entirely achieve the nominal yaw rate motion that would be achieved by the driver on a high friction coefficient road surface. In this case, the yaw control system might only partially succeed by making the vehicle's yaw rate closer to the expected nominal yaw rate, as shown by the middle curve in Fig 2.2.

These systems are designed to detect a difference between the vehicle response requested by the driver and the actual response of vehicle. When the differences are detected, the system intervenes to correct the path of the vehicle.

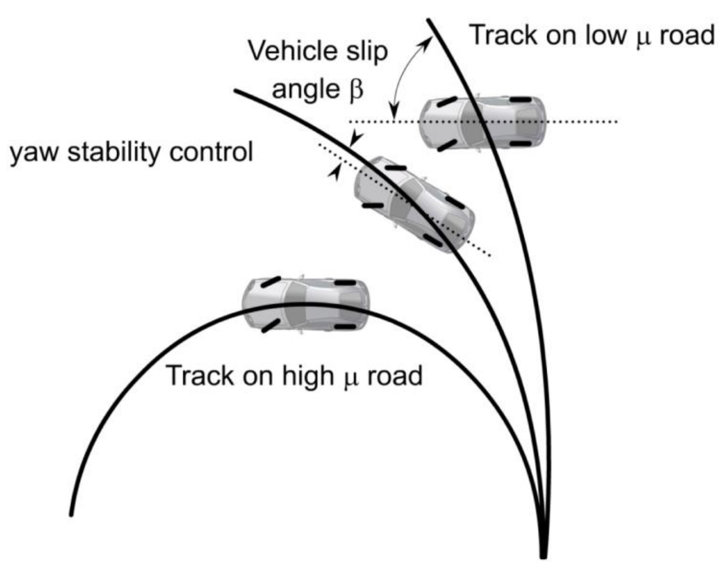


Figure 2.2 The functioning of yaw control system

2.1.2 The need for a vehicle stability control

The motivation for the development of yaw control systems comes from the fact that the behavior of the vehicle at the limits of adhesion is quite different from its nominal behavior. At the limits of adhesion, the slip angle is high and the sensitivity of yaw moment to changes in steering angle becomes highly reduced. At large slip angles, changing the steering angle produces very little change in the yaw rate of the vehicle. This is very different from the yaw rate behavior at low frequencies. On dry roads, vehicle maneuverability is lost at vehicle slip angles greater than ten degrees, while on packed snow; vehicle maneuverability is lost at slip angles as low as 4 degrees.

Due to the above change of vehicle behavior, drivers find it difficult to drive at the limits of physical adhesion between the tires and the road. First, the driver is often not able to recognize the friction coefficient change and has no idea of the vehicle's stability margin. Further, if the limit of adhesion is reached and the vehicle skids, the driver is caught by surprise and very often reacts in a wrong way and usually steers too much. Third, due to other traffic on the road, it is important to minimize the need for the driver to act thoughtfully. The yaw control system addresses these issues by reducing the deviation of the vehicle behavior from its normal behavior on dry roads and by preventing the vehicle slip angle from becoming large.

2.1.3 State-of-the-art Technology

Many companies have investigated and developed yaw control systems during the last ten years through simulations and on prototype experimental vehicles. Some of these yaw control systems have also been commercialized on production vehicles. Examples include the BMW DSC3 and the Mercedes ESP.

Yaw Stability control could have variety of names. These names include VSA (vehicle stability assist), VDC (vehicle dynamics control), VSC (vehicle stability control), ESP (electronic stability program), ESC (electronic stability control) and DYC (direct yaw control).

2.2 Kinematic model of lateral vehicle motion

Under certain assumptions described below, a kinematic model for the lateral motion of a vehicle can be developed. Such a model provides a mathematical description of the vehicle motion without considering the forces that affect the motion. The equations of motion are based purely on geometric relationships governing the system.

Consider a bicycle model of the vehicle as shown in Fig 2.3 (Wang and Qi,2001) [7]. In the bicycle model, the two left and right front wheels are represented by one single wheel at point A. Similarly the rear wheels are represented by one central rear wheel at point B. The steering angles for the front and rear wheels are represented by δ_f and δ_r respectively. The model is derived assuming both front and rear wheels can be steered. For front-wheel-only steering, the rear steering angle δ_r can be set to zero .The center of gravity (c.g.) of the vehicle is at point C. The distances of points A and B from the c.g. of the vehicle are l_f and l_r respectively. The Wheelbase of the vehicle is $L = l_f + l_r$.

The vehicle is assumed to have planar motion. Three coordinates are required to describe the motion of the vehicle: X , Y and ψ. (X,Y) are inertial coordinates of the location of the c.g. of the vehicle while ψ describes the orientation of the vehicle. The velocity at the c.g. of the vehicle is denoted by V and makes an angle β with the longitudinal axis of the vehicle. The angle β is called the slip angle of the vehicle.

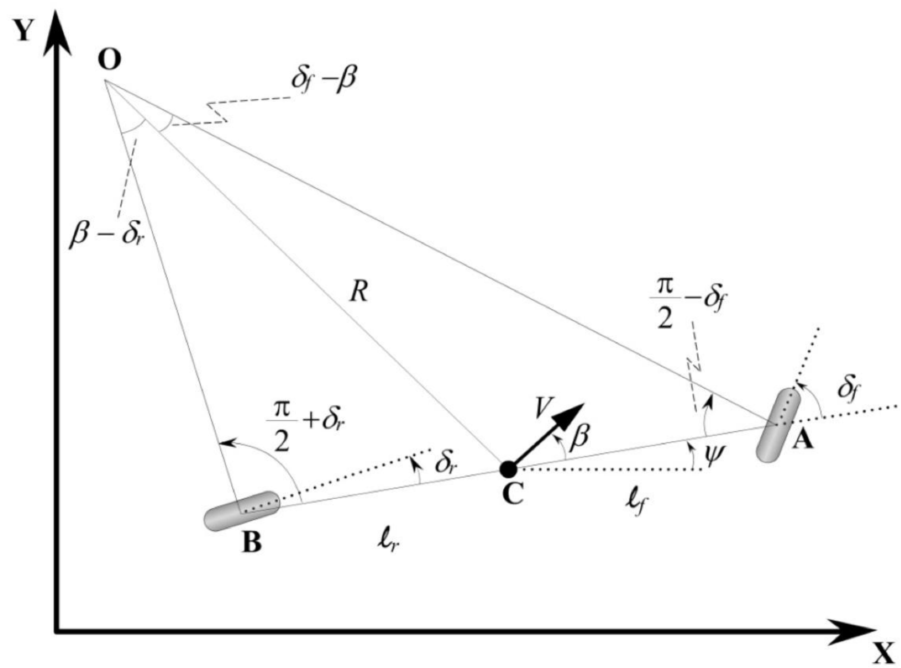


Figure 2.3: Kinematics of lateral vehicle motion

The major assumption used in the development of the kinematic model is that the velocity vectors at points A and B are in the direction of the orientation of the front and rear wheels respectively. In other words, the velocity vector at the front wheel makes an angle δ_f with the longitudinal axis of the vehicle. Likewise, the velocity vector at the rear wheel makes an angle δ_r with the longitudinal axis of the vehicle. This is equivalent to assuming that the "slip angles" at both wheels are zero.

The point O is the instantaneous rolling center for the vehicle. The point O is defined by the intersection of lines AO and BO which are drawn perpendicular to the orientation of the two rolling wheels.

The radius of the vehicle's path R is defined by the length of the line OC that connects the center of gravity C to the instantaneous rolling center O. The velocity at the c.g. is perpendicular to the line OC. The direction of the velocity at the c.g. with respect to the longitudinal axis of the vehicle is called the slip angle of the vehicle β .

The angle ψ is called the heading angle of the vehicle. The course angle for the vehicle is $\gamma = \psi + \beta$.

2.3 Dynamic Bicycle model of lateral vehicle

A "bicycle" model of the vehicle with two degrees of freedom is considered, as shown in Fig 2.4. The two degrees of freedom are represented by the vehicle lateral position y and the vehicle yaw angle ψ. The vehicle lateral position is measured along the lateral axis of the vehicle to the point O that is the center of rotation of the vehicle. The vehicle yaw angle ψ is measured with respect to the global X-axis. V denotes the longitudinal velocity of the vehicle at the c.g.

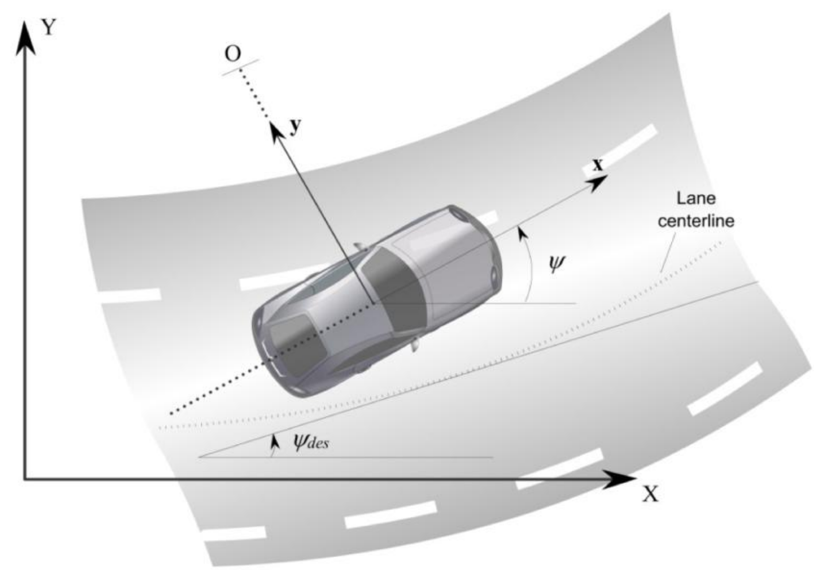


Figure 2.4 Lateral Vehicle Dynamics

Applying Newton's law for motion along the Y-axis,

$$ma_y = F_{yf} + F_{yr} \tag{2.1}$$

Where a_y is the inertial acceleration of the vehicle at the center of gravity (c.g) in the direction of the y-axis and F_{yf} and F_{yr} are the lateral tire forces of the front and rear wheels respectively. a_y contributes for the acceleration $\ddot{y}$ which is due to motion along the y axis and the centripetal acceleration $V_x\dot{\psi}$. Hence

$$a_y = \ddot{y} + V_x \dot{\psi} \tag{2.2}$$

Substituting from Eq. (2.1) into Eq. (2.2), the equation for the lateral translational motion of the vehicle is obtained as

$$m(\ddot{y} + \dot{\psi} V_x) = F_{yf} + F_{yr} \tag{2.3}$$

Moment balance about the z-axis yields the equation for the yaw dynamics as

$$I_z \ddot{\psi} = l_f F_{yf} - l_r F_{yr} \tag{2.4}$$

Where l_f and l_r re the distances of the front tire and the rear tire respectively from the c.g. of the vehicle.

The next step is to model the lateral tire forces F_{yf} and F_{yr} that act on the vehicle. Experimental results show that the lateral tire force of a tire is proportional to the "slip-angle" for small slip-angles. The slip angle of a tire is defined as the angle between the orientation of the tire and the orientation of the velocity vector of the wheel as in Fig 2.5. the slip angle of the front wheel is

$$\alpha_f = \delta - \theta_{Vf} \tag{2.5}$$

Where θ_{Vf} is the angle that the velocity vector makes with the longitudinal axis of the vehicle and δ is the front wheel steering angle. The rear slip angle is similarly given by

$$\alpha_r = -\theta_{Vr} \tag{2.6}$$

The lateral tire force for the front and rear wheels of the vehicle can be written as

$$F_{yf} = 2C_{\alpha f}(\delta - \theta_{Vf}) \tag{2.7}$$

$$F_{yr} = 2C_{\alpha r}(-\theta_{Vr}) \tag{2.8}$$

Where the proportionality constant C_α is called the cornering stiffness of the tires, δ is the front wheel steering angle and θ_V is the tire velocity angle. Where V is the velocity vector. The factor 2 accounts for the fact that there are two front and rear wheels.

To calculate θ_{Vf} and θ_{Vr}, the following relations can be used:

$$\tan(\theta_{Vf}) = \frac{V_y + l_f \dot{\psi}}{V_x} \tag{2.9}$$

$$\tan(\theta_{Vr}) = \frac{V_y - l_r \dot{\psi}}{V_x} \tag{2.10}$$

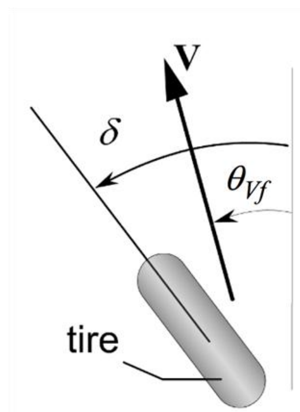


Figure 2.5 Tire slip-angle

Using small angle approximations and using the notation $V_y = \dot{y}$.

$$\theta_{Vf} = \frac{\dot{y}+l_f\dot{\psi}}{V_x} \tag{2.11}$$

$$\theta_{\square\square} = \frac{\dot{y}+l_r\dot{\psi}}{V_x} \tag{2.12}$$

2.4 Dynamic Model in terms of yaw rate and slip angle

In Fig 2.6 , vehicle side slip angle β is defined as the angle between the longitudinal axis of the vehicle and the orientation of vehicle velocity vector, and $r \equiv \dot{\varphi}$ is the yaw rate of the vehicle body. The lateral dynamics of the vehicle is controlled by the front wheel steering angle δ.

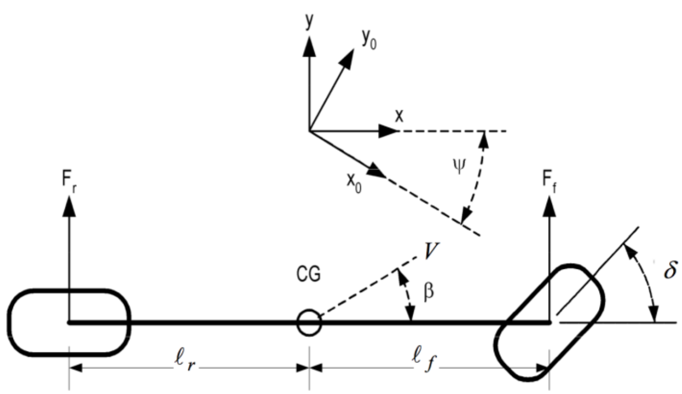


Figure 2.6: Single Track model for vehicle dynamic

Using the body side slip angle β and the yaw rate of vehicle body $r \equiv \dot{\varphi}$ as state variables, the vehicle lateral dynamics can then be described by the following differential equations

$$mV_x\left(\frac{d\beta}{dt} + \dot{\varphi}\right) = mV_x\left(\frac{d\beta}{dt} + r\right) = F_{yf} + F_{yr} \tag{2.13}$$

$$I_z\ddot{\varphi} = I_z\dot{r} = l_fF_{yf} - l_rF_{yr} \tag{2.14}$$

where m is vehicle mass, V_x is vehicle longitudinal velocity, F_{yf} , F_{yr} are front and rear tire forces, respectively, I_z is yaw moment of inertia, and l_f , l_r are distances from CG (center of gravity) to front and rear tires, respectively. For small tire slip angles, the lateral tire forces can be approximated as a linear function of tire slip angle. The front and rear tire forces and tire slip angles are defined as follows:

$$F_{yf} = C_{\alpha f}\alpha_f \, , \alpha_f = \delta - \theta_{Vf} = \delta - \beta - \frac{l_f r}{V_x} \tag{2.15}$$

$$F_{yr} = C_{\alpha r}\alpha_r \, , \alpha_r = -\theta_{Vf} = -\beta - \frac{l_r r}{V_x} \tag{2.16}$$

where $C_{\alpha f}$ and $C_{\alpha r}$ are the cornering stiffness of the front and rear tires respectively. Substituting (2.13) and (2.14) into (2.11) and (2.12) yields the following description for the vehicle lateral dynamics:

$$\frac{d\beta}{dt} = -r + \frac{C_{\alpha f}}{mV_x}\left(\delta - \beta - \frac{l_f r}{V_x}\right) + \frac{C_{\alpha r}}{mV_x}\left(-\beta + \frac{l_r r}{V_x}\right) \tag{2.17}$$

$$\frac{dr}{dt} = \frac{l_f C_{\alpha r}}{I_z}\left(\delta - \beta - \frac{l_f r}{V_x}\right) - \frac{l_r C_{\alpha r}}{I_z}\left(-\beta + \frac{l_r r}{V_x}\right) \tag{2.18}$$

The State Space Model

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \frac{-(C_f + C_r)}{mV_x} & \frac{-(C_f l_f - C_r l_r)}{mV_x^2} - 1 \\ \frac{-(C_f l_f - C_r l_r)}{I_z} & \frac{-(C_f l_f^2 + C_r l_r^2)}{I_z V_x} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} \frac{C_f}{mV_x} \\ \frac{C_f l_f}{I_z} \end{bmatrix} \delta \tag{2.19}$$

3Chapter

Yaw Stability Control

3.1 Introduction

A trend in modern vehicles is the application of active safety system used to improve vehicle handling, stability and control. The development of chassis control systems is still the object of intense research activities from both industrial and academic sides.

Safety of ground vehicles requires the improvement of yaw stability by active control the basic idea is to assist the vehicle handling to be close to linear vehicle characteristics that are familiar to the driver and to restrain the vehicle lateral dynamics to be within a stable handling region in aggressive maneuvers.

3.2 Yaw control systems

3.2.1 Types of yaw control systems

Several actuators, such as active suspension, active steering and active braking could be used for vehicle stability control. An active suspension system, by controlling the wheel load, may improve the lateral dynamics of the vehicle. An active steering system, by controlling the steering angles of the wheels, has great influence on the lateral behavior of the vehicle. Finally, an active braking system like the Direct Yaw Control (DYC), by using differential braking, is very effective for lateral stability of the vehicle.

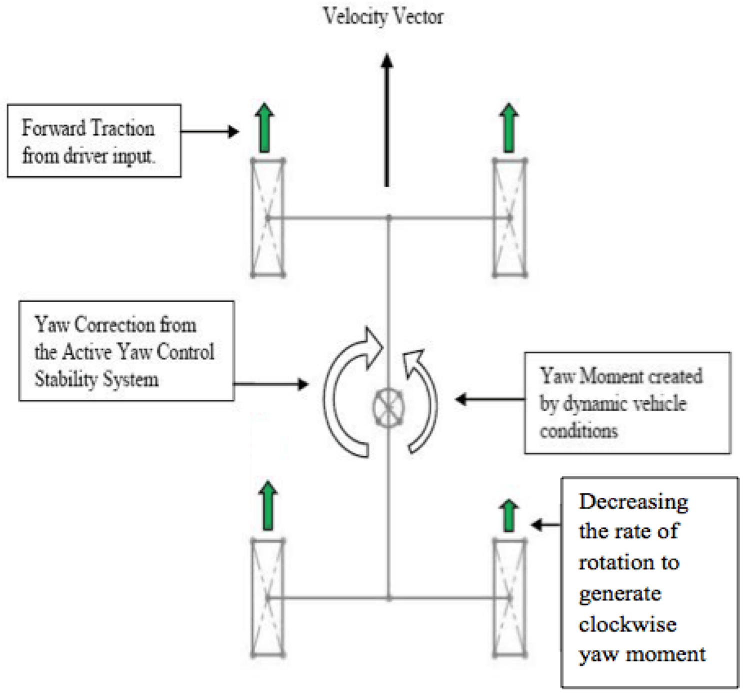


Figure 3.1 Generation of corrective yaw moment using differential drive

3.3.1 Vehicle body equations

The 2-DOF (degree of freedom) classical linear bicycle model discussed in chapter 2 is used. By adding the corrective yaw moment term (needed to achieve the desired yaw rate) to the model:

$$\frac{d\beta}{dt} = -r + \frac{C_{\alpha f}}{mV_x}\left(\delta - \beta - \frac{l_f r}{V_x}\right) + \frac{C_{\alpha r}}{mV_x}\left(-\beta + \frac{l_r r}{V_x}\right) \tag{3.1}$$

$$\frac{dr}{dt} = \frac{l_f C_{\alpha r}}{I_z}\left(\delta - \beta - \frac{l_f r}{V_x}\right) - \frac{l_r C_{\alpha r}}{I_z}\left(-\beta + \frac{l_r r}{V_x}\right) + \frac{1}{Iz} Mz \tag{3.2}$$

The state space model

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} \frac{-(C_f + C_r)}{mV_x} & \frac{-(C_f l_f - C_r l_r)}{mV_x^2} - 1 \\ \frac{-(C_f l_f - C_r l_r)}{I_z} & \frac{-(C_f l_f^2 + C_r l_r^2)}{I_z V_x} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{I_z} \end{bmatrix} M_z + \begin{bmatrix} \frac{C_f}{mV_x} \\ \frac{C_f l_f}{I_z} \end{bmatrix} \delta \tag{3.3}$$

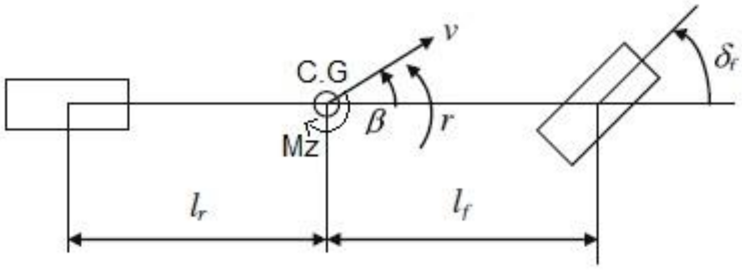


Figure 3.2: bicycle model showing the generated yaw moment

3.4 Control design

3.4.1 Robust Tracking Controller

Using this controller, the system will be able to track a desired input, which is in our case (the desired yaw rate) at any value with the ability to overcome disturbances[8].

First, we will discuss the principle of tracker controller, consider the following process:

$$\dot{x} = Ax + Bu + Ep \tag{3.4}$$

$$y = Cx + Du \tag{3.5}$$

Where:

- $x \in R^n$: The state vector.
- $u \in R^m$: The input vector.
- $P \in R^m$: The disturbance vector.
- $A \in R^{n \times n}$: The system matrix.
- $B \in R^{n \times m}$: The input matrix.
- $E \in R^{n \times m}$: The disturbance matrix.
- $C \in R^{r \times n}$: The output matrix.
- $D \in R^{r \times m}$: The feed-forward matrix.
- $y \in R^r$: The output vector.

Control low

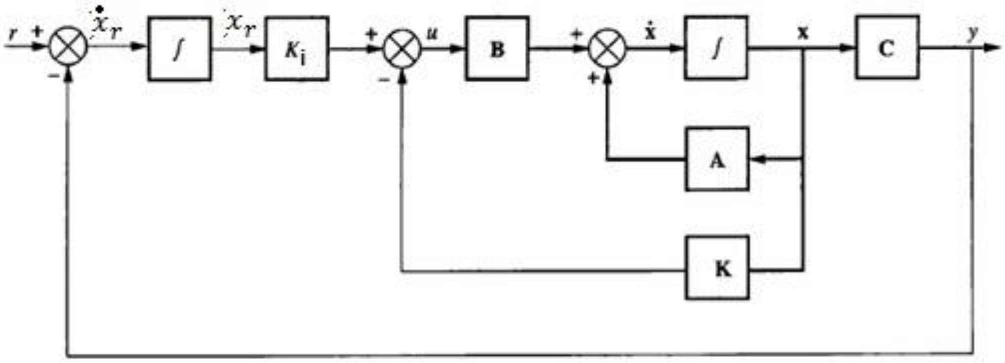


Figure 3.3: General Simulink model for the robust tracking controller

$u = kx + k_i x_r$ (3.6)

$x_r = r - Cx$ (3.7)

Representing Eq. (3.8) and (3.11) in compact form

$$\begin{bmatrix} \dot{\boldsymbol{x}} \\ \dot{x}_r \end{bmatrix} = \begin{bmatrix} \boldsymbol{A} & \boldsymbol{0} \\ -\boldsymbol{C} & 0 \end{bmatrix} \begin{bmatrix} \boldsymbol{x} \\ x_r \end{bmatrix} + \begin{bmatrix} \boldsymbol{B} \\ 0 \end{bmatrix} u + \begin{bmatrix} \boldsymbol{0} \\ 1 \end{bmatrix} r \tag{3.8}$$

$$\dot{X}_e = A_e X_e + B_e u + E_e r \tag{3.9}$$

Representing equation Eq. (3.10) in compact form

$$u = -[\boldsymbol{k} \quad -k_i] \begin{bmatrix} \boldsymbol{x} \\ x_r \end{bmatrix}$$

$$u = -k_e X_e$$

$$\dot{X}_e = A_e X_e + B_e(-k_i X_e) + E_e r$$

$$\dot{X}_e = (A_e - B_e k_e) X_e + E_e r$$

If the pair (A_e, B_e) is controllable, then we can k_e to place the eigenvalues of the entire control system to get the desired response

Now, making a controller design for our system using the state space model described in equation in the form of $\dot{X}_e = A_e X_e + B_e u + E_e r$ for the tracker, now we must check the controllability for our system, by applying the controllability matrix method in Matlab

$\gg C_{M=}ctrb(A_e, B_e);$

$\gg rank(C_M);$

We found that the rank is 3 (full rank), so the pair (A_e, B_e) is controllable.

To get the desired response, let the design region ranges between (8 to 10) rad/s for the natural frequency (ω) and from (0.7 to 0.9) for the damping ratio (ζ), so the poles are placed at:

P_e = [-6.4000 + 4.8000i -6.4000 - 4.8000i -10.0000] due to the desired responses.

Applying the place method to find the gains to obtain the desired poles locations we used the following Matlab commands:

$\gg k_e = place\ (A_e, B_e, P_e);$

$\gg k = k_e[1{:}\,2];$

$\gg k_i = k[3];$

3.4.2 Simulation

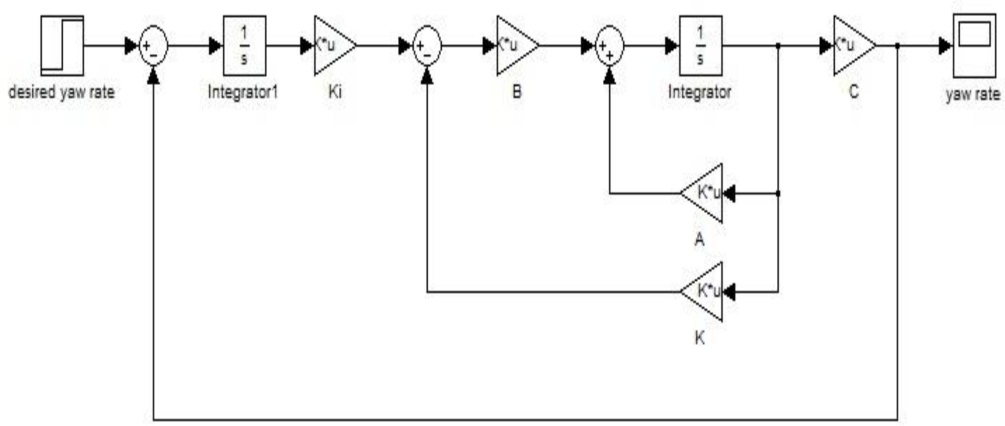


Figure 3.4: Yaw Control Model using Robust Tracking Controller (Simulink).

In this simulation a longitudinal speed of 30 m/s is used. The road is initially straight and then becomes circular with a radius of 1000 m starting at a time of 1 second. The corresponding desired yaw rate can be calculated from $r_{des} = \frac{V_x}{R} = 0.03\ rad/s$. The desired yaw rate is shown in Figure 3-1 and is a step input from 0 to 0.03 rad/s at 1 second. The actual yaw rate (output) is shown in Figure 3-2.

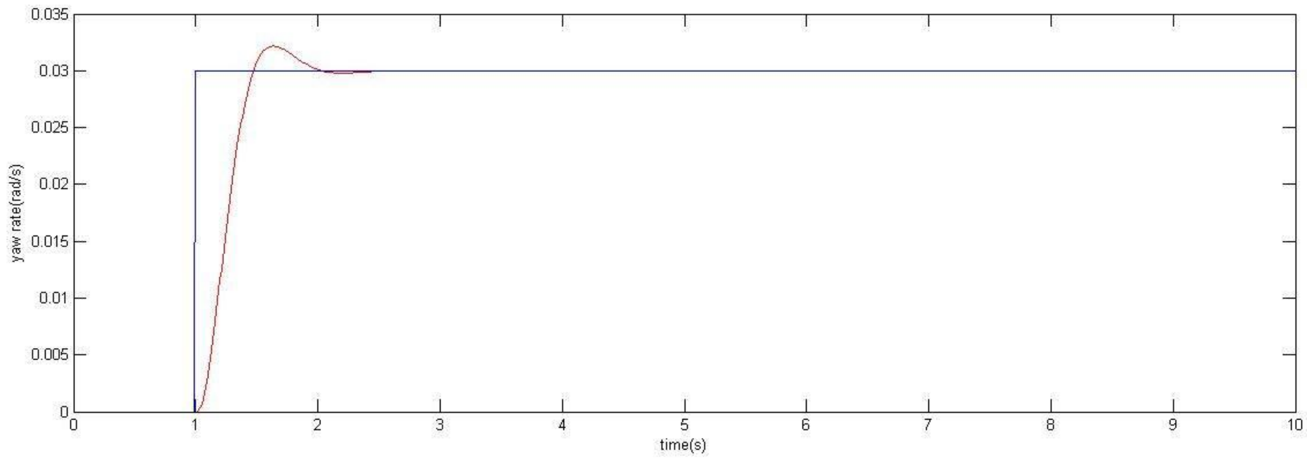


Figure 3.5: Desired Yaw rate for step input (Simulation)

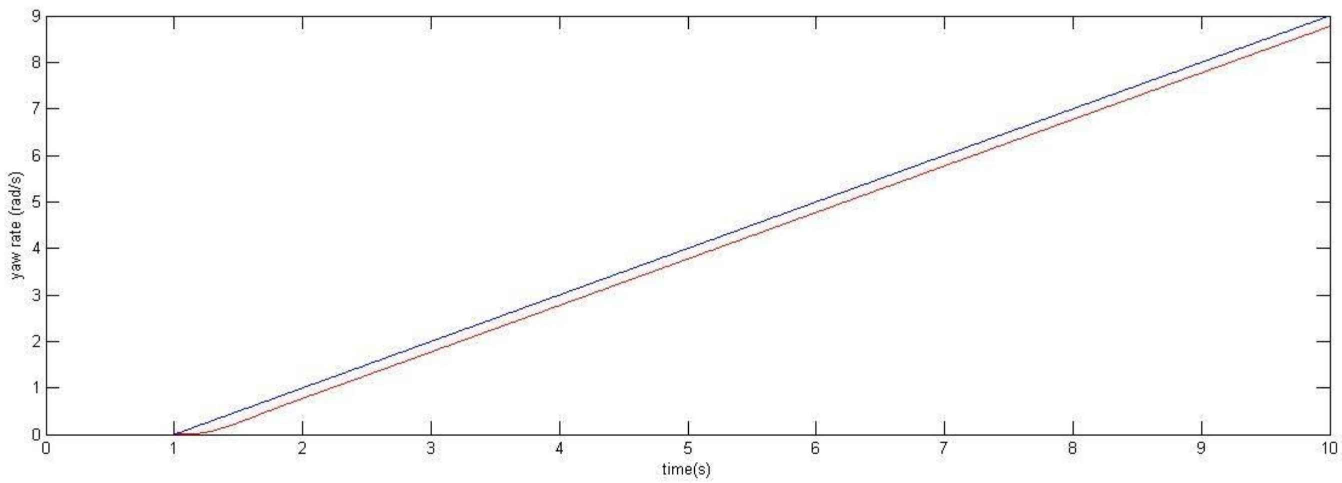


Figure 3.6: Desired Yaw rate for ramp input (Simulation)

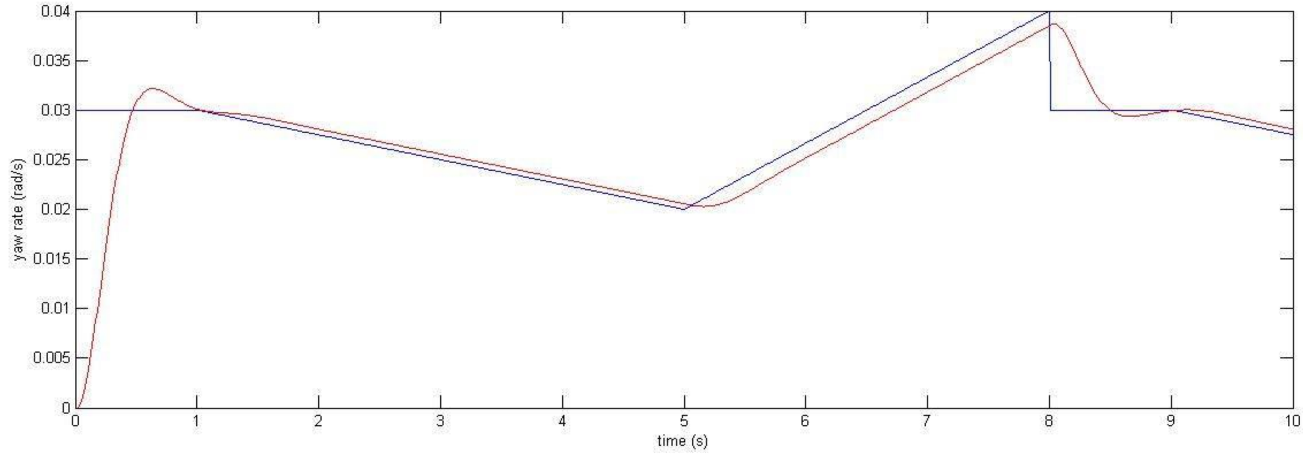


Figure 3.7: Desired Yaw rate for generated signal (Simulation)

Chapter 4

Hardware Architecture

4.1 Introduction

It is a project need to identify the system’s physical components and to gather information about the related component to find the rules that dominate these relations. Hardware architecture allows understanding how these components should fit into system architecture and provides important information needed for software design, development and integration. For controlling the yaw stability of the car, we chose to have a remote controlled car as a model for the project. This car will have close specification to a real car , such as the existence of suspension system on the wheels and having a rubber tires that have a friction with the surface of the road.

4.2 System architecture

In System architecture each part that will be used on the Remote Controlled car will be mentioned, described and explained.

4.2.1 Remote Controlled Car (RC Car)

The RC car shown if Fig 4.1 will be used as the controlled model. The car was chosen to have close specifications to a real car so the controller can be adjusted and developed and be applied for a real scale car as a future work.
RC car is a self-powered car through four, 1.5 V batteries, transmitter and receiver to control the movement and direction of the car.
The transmitter that came with the car will be used since it works in good function.

Figure 4.1 RC Car with remote control

4.2.1.1 Finding the rotational moment of inertia (J) of the car.

It is essential to know the moment of inertia of RC car in order to know the desired torque required to give a certain angular acceleration about the axis of rotation. To calculate the moment of inertia, we hanged the car to a steel stand, where the hanging point was passing through the center of gravity of the car, as shown in fig (4.2).

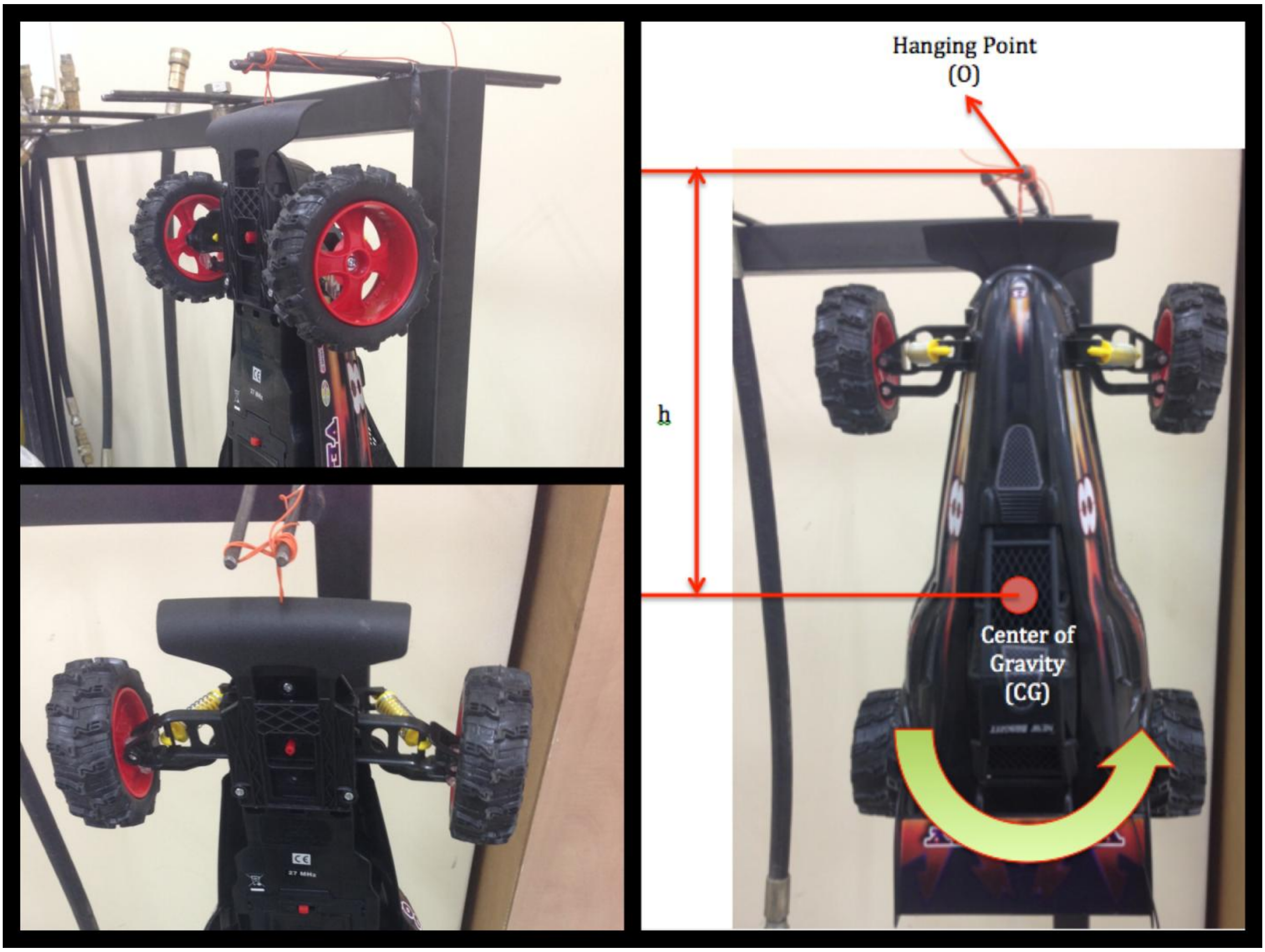


Figure 4.2 Moment of Inertia Test

The car was hanged on the stand were no oscillation occurred, then the car was given a pulse to oscillate around fixing point, after that we took the time of twenty cycles of oscillation. This trial was made twenty times, the values were as shown below:

Table 4.1: Readings of time for moment of inertia test

Number of trial	Time for 20 cycles of oscillation (s)	Number of trial	Time for 20 cycles of oscillation (s)
1	21.11	11	21.11
2	21.13	12	21.3
3	21.26	13	21.41
4	21.12	14	21.23
5	21.29	15	21.1
6	21.22	16	21.26
7	21.29	17	21.21
8	21.28	18	21.31
9	21.28	19	21.18
10	21.14	20	21.21
SUM			424.44 sec

Dividing the sum of time for all trials over the number of trials, we get the time for one trial.

τ_{20} = 424.44/20= 21.222 sec.

Each trial consisted of 20 cycles, dividing the time of one trial by the number of cycles, we get the time for one cycle.

Divide τ_{20} by 20.

$\tau = \tau_{20}/20$.

τ =21.222/20 = 1.0611 sec.

Using the following relationship that determines the rotational moment of inertia around a point passing through the center of gravity of an object.

$$J_{CG} = \frac{mgh\tau^2}{4\pi^2} - mh^2 \quad (4.1)$$

We get that **J**= 0.00709 Kg.m^2

Where:

The mass of the car (m) = 0.641 Kg.

Gravitational force (g) = 9.81 m/sec^2

The distance between the hanging point and center of gravity (h)= 0.232 m.

4.2.2 Arduino Mega Microcontroller

The chosen microcontroller is ARDUINO Mega as shown in Fig 4.3. This microcontroller will be used to have the reading of the actual and desired yaw rate form the accelerometer, then compare it in order to exert a force that will give a steady state error equal to zero. Arduino Mega has many advantages that make it the preferred choice such as: user friendly, open source codes which are available for many applications that can be downloaded from the company's website, simulation program, the available shields for many components and one of them is suitable for the wireless modules which allows the connection of modules easily. Other advantages are the Upgradability, the modularity and the last important thing is that, it does not need a dedicated programmer as it can be programmed through the same cable that is used for the PC connection. The Arduino Mega is a microcontroller board based on the ATmega1280. It has 54 digital input/output pins (of which 14 can be used as PWM outputs), 16 analog inputs,4 UARTs (hardware serial ports), a 16 MHz crystal oscillator, a USB connection, a power jack, an ICSP header, and a reset button. It can be operated using USB cable , AC-DC adapter or battery.

Figure 4.3: Arduino Mega Microcontroller

Table 4.2: Specifications of Arduino Mega

Microcontroller	ATmega 1280
Operating Voltage	5V
Input Voltage (recommended)	7-12V
Input Voltage (limits)	6-20V
Digital I/O Pins	54 (15 of them provide PWM)
Analog Input Pins	16
Flash Memory	128 KB of which 4KB used by boot loader
SRAM	8 KB
EEPROM	4 KB
Clock Speed	16 MHz

4.2.3 Gyroscope (MPU-6050)

For the purpose of sensing the yaw angle of the car while it is motion, Gyroscope MPU-6050 shown in Fig 4.4, was used since it is a small (6mm x 3mm x 1.45mm), thin low power, complete 9-axis gyroscope with I^2C output. The MPU-60X0 features three 16-bit analog-to-digital converters (ADCs) for digitizing the gyroscope outputs and three 16-bit ADCs for digitizing the accelerometer outputs with precision tracking of ±250°/sec (dps) and a user-programmable accelerometer full-scale range of ±2g.

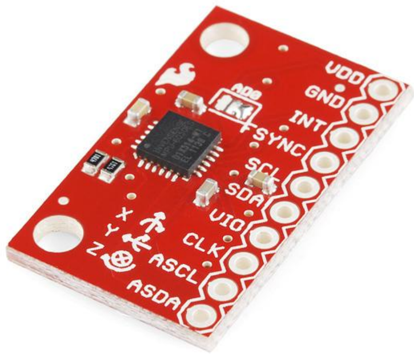


Figure 4.4: MPU 6050 Gyroscope

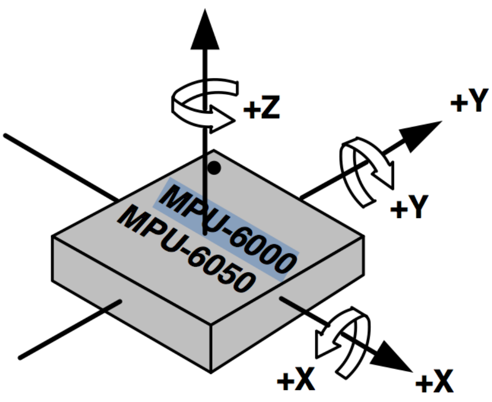


Figure 4.5: Orientation of Axes of Sensitivity and Polarity of Rotation

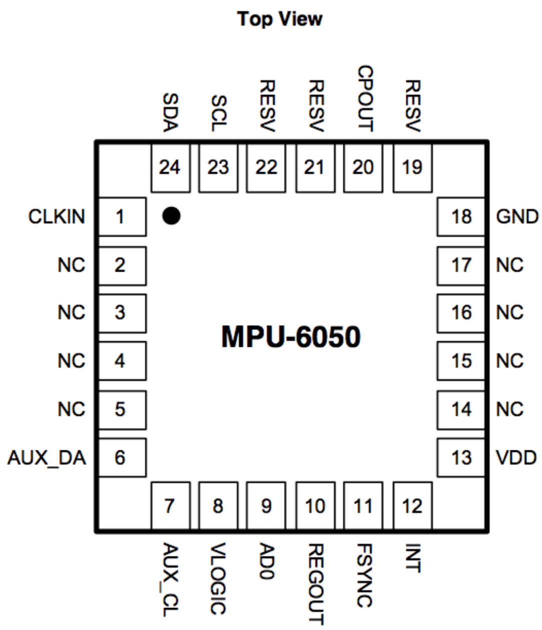


Figure 4.6: QFN Package 24-pin, 6mm x 3mm x 0.9mm

4.2.4 DC Motor (FA-130)

The DC motor shown in fig. 4.7 is used for the movement of the car. Two DC motors were used on the rear wheels in order to generate a differential drive. When the speed of both motors are equal, the car will move in a straight line, and when the speed of the right wheel is greater than the left wheel, the car will rotate to the right, and when the speed of the left wheel is greater than the right wheel, the car will rotate to the left, as shown if Fig 4.8.

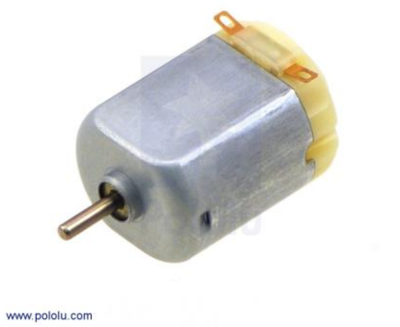


Figure 4.7: FA-130 DC Motor

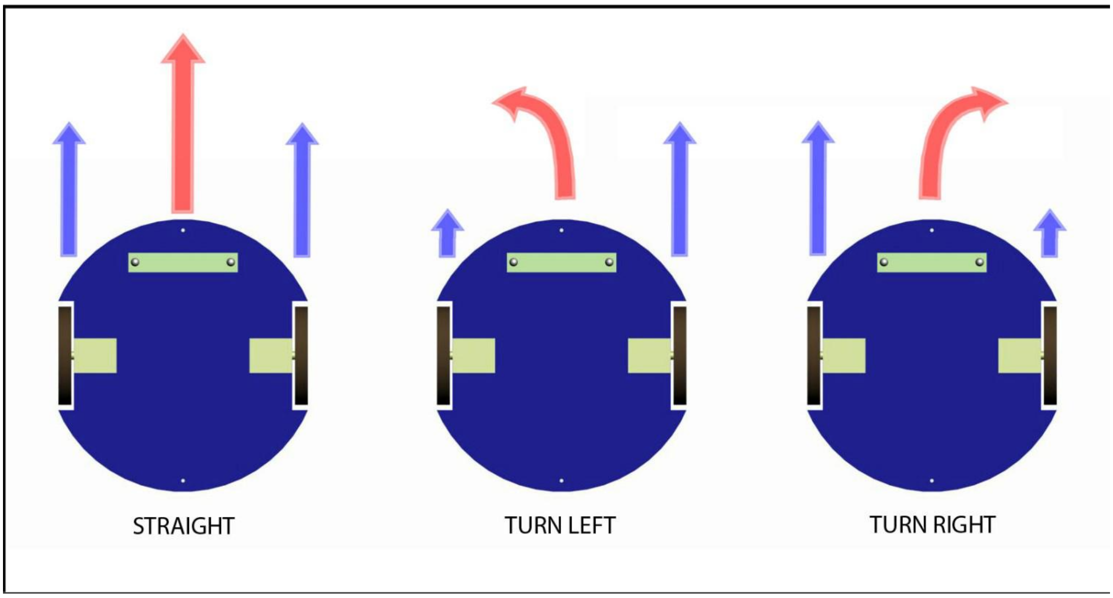


Figure 4.8: The three cases of twin motor speed

MODEL	VOLTAGE		NO LOAD		AT MAXIMUM EFFICIENCY					STALL		
	OPERATING RANGE	NOMINAL	SPEED	CURRENT	SPEED	CURRENT	TORQUE		OUTPUT	TORQUE		CURRENT
			r/min	A	r/min	A	mN·m	g·cm	W	mN·m	g·cm	A
FA-130RA-2270	1.5~3.0	1.5V CONSTANT	9100	0.20	6990	0.66	0.59	6.0	0.43	2.55	26	2.20

Table 4.4: Specifications of the chosen DC motor

4.2.5 Double Gearbox Kit

The main feature of double gearbox kit shown in Fig 4.9, is that it has two motors with their gearboxes so it can give a speed for the two rear wheels independently, This kit can give a clockwise or counter clockwise rotation around the vertical axis of the car through giving less rate of rotation to one wheel than the other wheel. The double gearbox will be installed on the rear wheels of the car.
This Kit has 12.7:1 gear ratio, generating 9 mN.m torque, since FA-130 DC motor can not generate enough torque to move the car.

Figure 4.9 Double Gearbox kit

4.2.5.1 Double Gearbox Kit installation

The process of installation started with removing the old gearbox and motor of the RC car, then the double gearbox kit was attached with two screws on a steel sheet that was fixed on the body of the RC car as shown in fig 4.10. The diameter of the hexagonal axle of the gearbox was smaller than the axle of the wheel, so a connection was made between these two axles using the piece shown in fig 4.11.

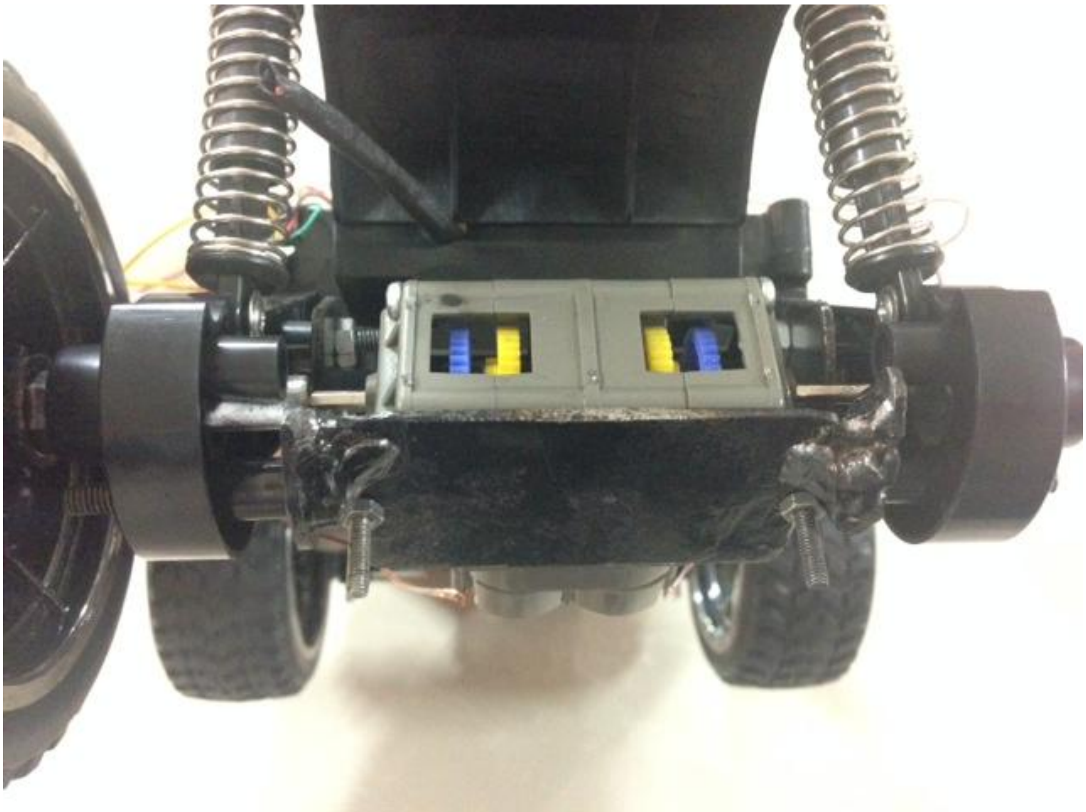


Figure 4.10: Attaching Double Gearbox Kit to RC car

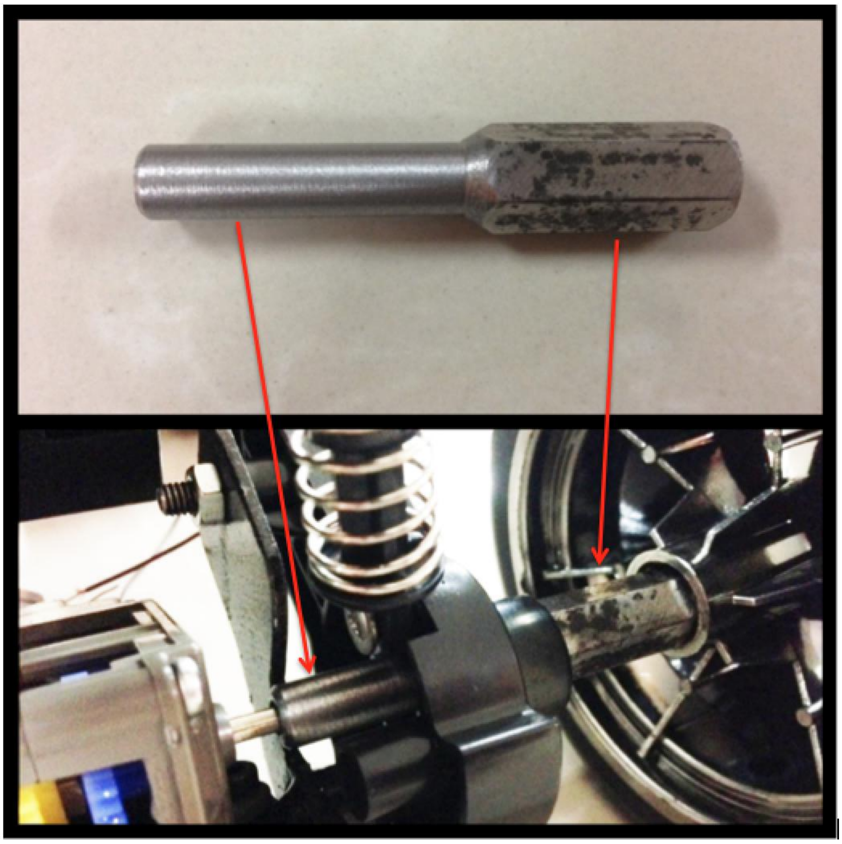


Figure 4.11: Connecting piece between Double gearbox and RC car's wheel

Chapter 5

Discrete Tracking Controller

5.1 Introduction

The next step in the project is to implement the control system on Arduino microcontroller to be used in the car. In this chapter, the continuous controller, which was discussed in chapter 3 will be converted to a discrete controller in order to find the controller gains.

5.2 Discretizing the tracking controller

Now the gain Ke for the discrete tracker controller will be evaluated at a sampling time (Ts) of 0.001 second. To find it, the following steps are required using Matlab:

- Select the desired location of the system poles is Z-domain
  Poles=[-4.2000 + 5.6000i -4.2000 - 5.6000i -10.0000]

- Find the extended discrete matrices of the system

  $$F_e = \begin{bmatrix} F & 0 \\ -C & I \end{bmatrix} \tag{5.1}$$

  $$G_e = \begin{bmatrix} G \\ 0 \end{bmatrix} \tag{5.2}$$

  F and G are the system discrete matrix and the discrete input matrix respectively.

  $$F = Ts * A + I \tag{5.3}$$

  $$G = Ts * B \tag{5.4}$$

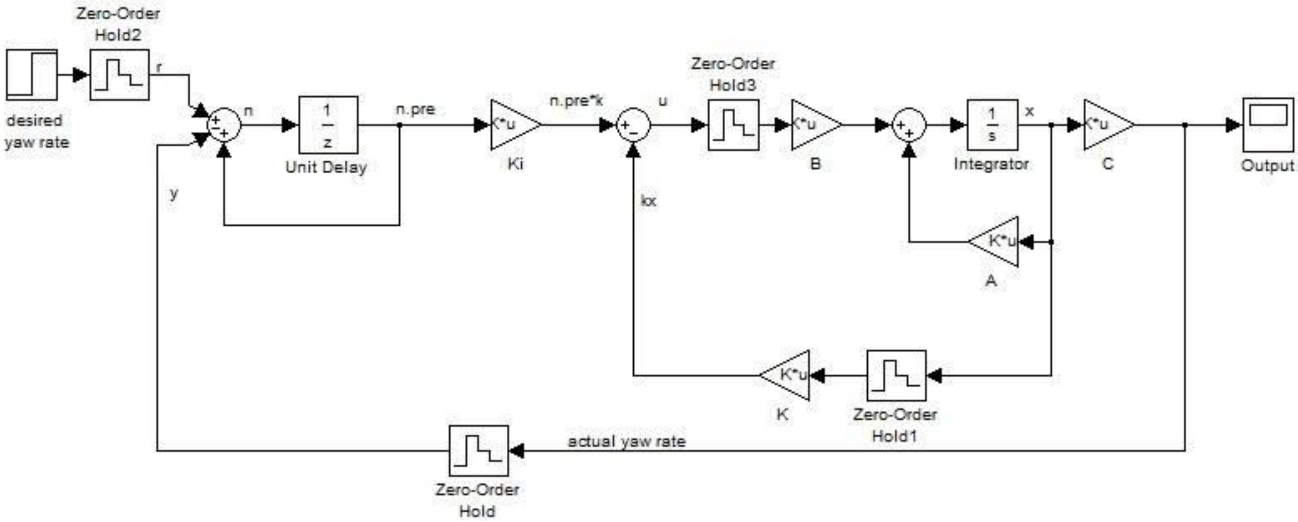
  Where:
  Ts: sampling time
  A: Continuous system matrix
  B: Continuous input matrix
  I: Identity matrix

- Using pole placement method, the gains are

$$Ke = place(F_e, G_e, poles) \tag{5.5}$$
$$K = Ke(1:2) \tag{5.6}$$
$$Ki = Ke(3) \tag{5.7}$$

5.3 Simulation results

For obtaining the simulation results, it is needed to draw the Simulink model as shown if Fig 5.1 and define all its parameters. The response of this model is shown in Fig 5.2 where it was obtained at an input yaw rate equals to 2 rad/s.



Simulink model of discrete traking controller :5.1Figure

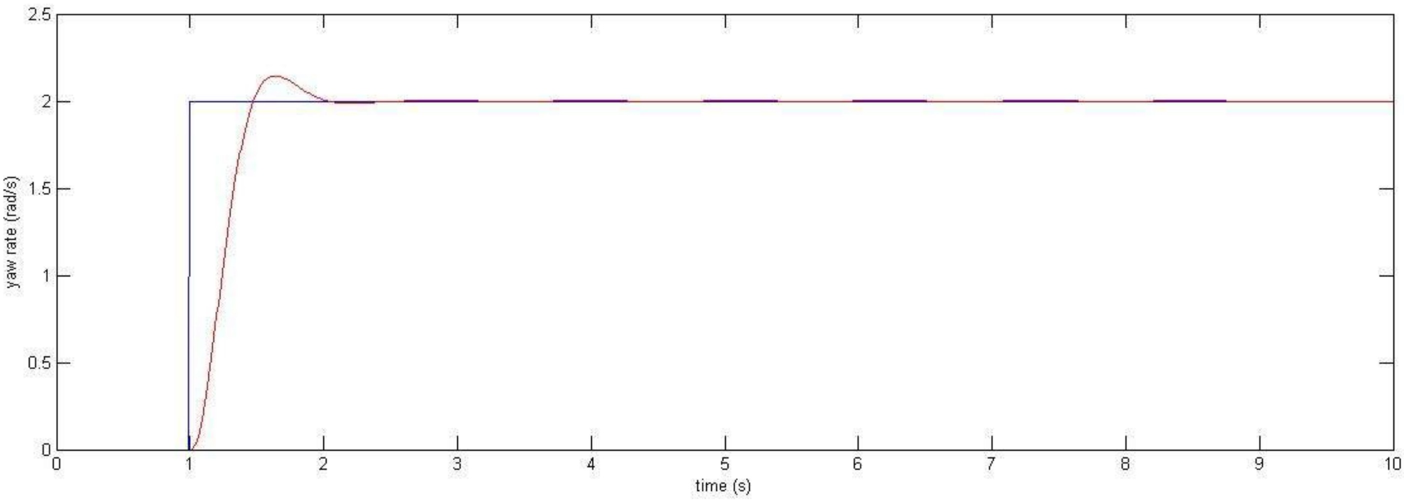


Figure 5.2: Simulink response for step input

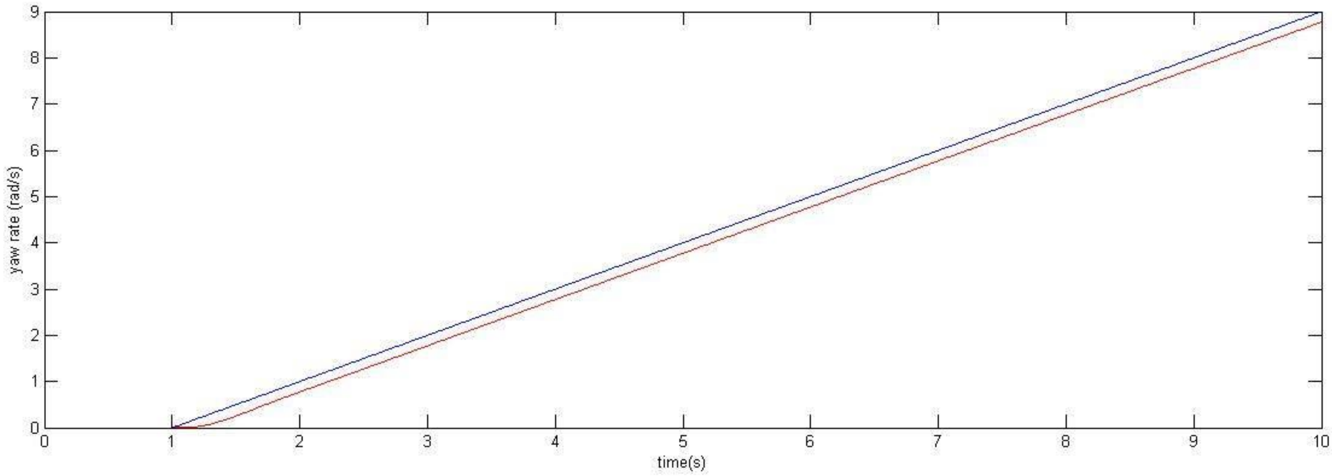


Figure 5.3: Simulink response for ramp input

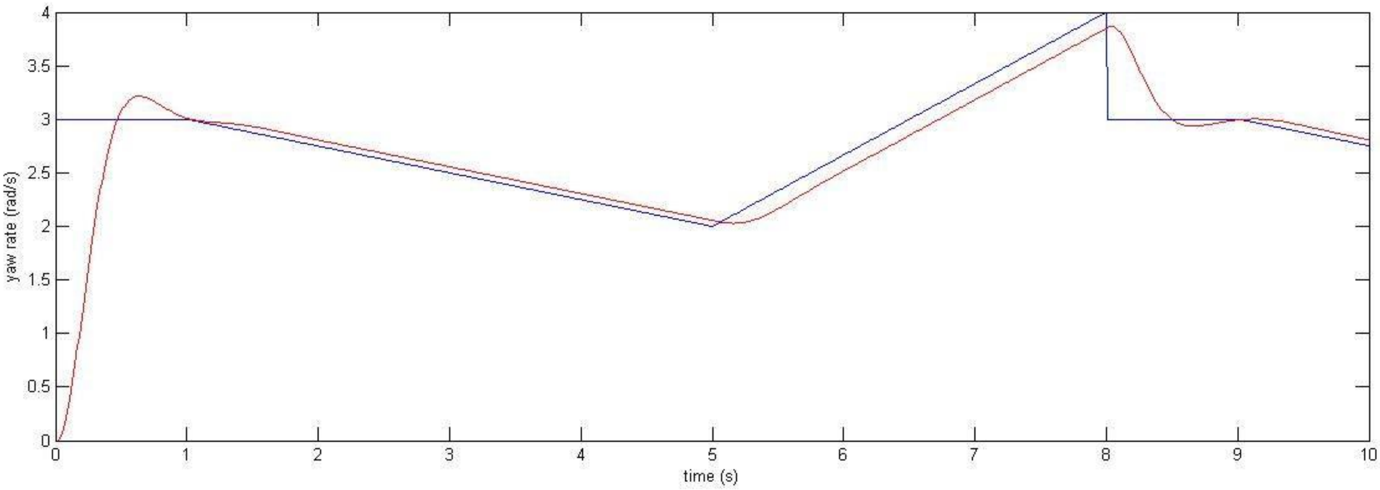


Figure 5.4: Simulink response for generated input

Chapter 6

Software Design

6.1 Introduction

After finding the gains for the discrete controller in the previous chapter, converting the controller to a code in the C-language can be done.

6.2 Implementation

For the hardware, the microcontroller used in the car is Arduino Mega 2560 as mentioned in chapter 4. And for the software, Arduino 1.5.6-r2 software is used to verify and download the code on Arduino Mega as shown if Fig 6.1.

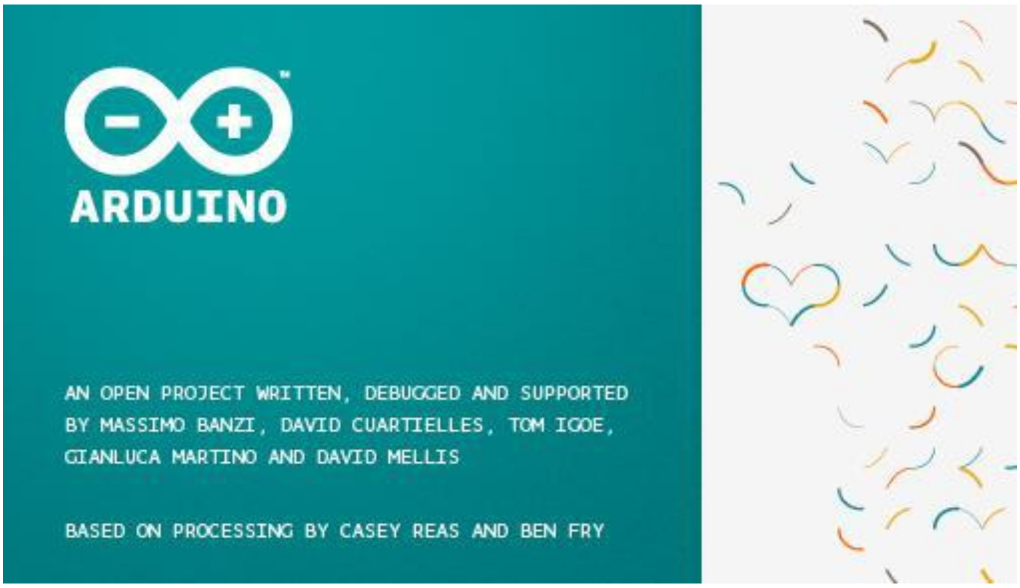


Figure 6.1: Arduino 1.5.6-r2 software

6.2.1 Reading the accelerometer

Since the accelerometer output is in $volts$, the output must be converted to (m/s^2) for the controller. From accelerometer datasheet, zero acceleration equals $1.5g$, that means when there is no acceleration, the accelerometer output will be $1.5g$, where g is a unit of acceleration equal to earth's gravity at sea level (9.81 m/s^2). And as the sensitivity of the accelerometer is 0.3 $(volts/g)$, where the sensitivity is a measure of how much the output of a sensor changes as the input acceleration changes measured in $(volts/g)$; the output of the accelerometer can be transferred to distance as follows:

$$acceleration(m/s^2) = \frac{accelerometer\ output(v) - 1.5(v)}{0.3\ (v/g)} * 9.81 \tag{6.1}$$

6.2.2 Calculating the speed

As the accelerometer gives the acceleration and there is no integration in the Arduino, the following steps must be done to calculate the speed in the Arduino:

- Store the acceleration before on sampling time (previous acceleration) (a_xp).
- Add the current acceleration to the previous acceleration
- Multiply the the value from step 2 by the sampling time divided by 2.
- finally add the value of the previous speed to the value from step 3.

6.3 Program code

This section shows how the ESC is implemented on the microcontroller

A- Calculating the actual yaw rate

To calculate the actual yaw rate, the longitudinal acceleration (a_y) and the lateral speed (V_x) must be known, and then the desired yaw rate can be found by Eq. (2.2) $(\dot{\psi}_{actual} = a_y/V_x)$ where $\ddot{y}$ is assumed to be small, thus it has a neglected effect.

```
a_x=(((xpin-1.5)/0.3)*9.81);       //acceleration(x-axis), longitudinal
v_x=((Ts/2)*(a_x+a_xp))+v_xp; //speed(x-axis), longitudinal
a_y=(((ypin-1.5)/0.3)*9.81);       //acceleration(y-axis), lateral
y_r_actual=a_y/v_x;                 //actual yaw rate
```

B- calculating the desired yaw rate

To calculate the desired yaw rate, the longitudinal acceleration (a_x) and the longitudinal speed (V_x) must be known, then the desired yaw rate can be found by the relation ($\dot{\psi}_{desired} = V_x/R$).

```
a_x=(((xpin-1.5)/0.3)*9.81);        //acceleration(x-axis), longitudinal
v_x=((Ts/2)*(a_x+a_xp))+v_xp;  //speed(x-axis), longitudinal
y_r_desired=v_x/rad_of_road;     //desired yaw rate
```

C- Controller code

After knowing the actual yaw rate, the desired yaw rate, settling time and the gains; the tracker controller can be transferred to a code to write it on the microcontroller

The discrete tracker controller equations are

$$u = K_i X_n(k) - Kx(k) \tag{6.2}$$
$$X_n(k+1) = r(k) - y(k) + X_n(k) \tag{6.3}$$

By transferring these equations to a code, the controller can be written as shown below

```
y_r_desired=v_x/rad_of_road;                          //desired yaw rate
y_r_actual=a_y/v_x;                                    //actual yaw rate
float x1=0.5;                                          //constant slip angle
n=y_r_desired-y_r_actual+np;
F=(0.3918*x1)-(0.1702*y_r_actual)+(0.2088*np); //force
u=(F)*(255/5);                                         //voltage
np=n;
a_xp=a_x;
v_xp=v_x;
```

The output of the controller is voltage (u), this voltage will be connected to two pulse width modulation pins on the microcontroller to feed the DC motors which are connected to the rear wheels of the car. So, if the car is turning right; the voltage will be transferred throw the PWM pin that is connected to the left rear wheel, and if the car is turning left; the voltage will be transferred throw the rear right wheel as shown in the code below

```
 if(dir_forward && dir_right && !dir_left)         //car is cornering to the right
{
  if (u==0){
  analogWrite(motor_right_for, motor_speed);
  analogWrite(motor_left_for, motor_speed);
  }
  if (u>0)
  {
  analogWrite(motor_right_for, motor_speed);
  analogWrite(motor_left_for, u);
  }
```

```
if(dir_forward && !dir_right && dir_left)            //car is cornering to the left
{
 if (u==0){
 analogWrite(motor_right_for, motor_speed);
 analogWrite(motor_left_for, motor_speed);
 }
 if (u>0)
 {
 analogWrite(motor_right_for, u);
 analogWrite(motor_left_for, motor_speed);
 }
}
```

Chapter 7

Electronic Circuit

7.1 Introduction

An electronic circuit is composed of individual electronic components, such as resistors, transistors, capacitors, inductors and diodes, connected by conductive wires or traces through which electric current can flow. The combination of components and wires allows various simple and complex operations to be performed: signals can be amplified, computations can be performed, and data can be moved from one place to another. Circuits in the project are constructed of discrete components connected by individual pieces of wire. In this chapter, the inputs and outputs of the system will be shown.

7.2 Power Supply

For this project, we need a constant DC voltage of 5 Volt to supply both the Arduino microcontroller and the two DC Motors. Hence four 1.2 Volt batteries will be used, giving a total voltage of 4.8 Volt, as shown in Fig 7.1.

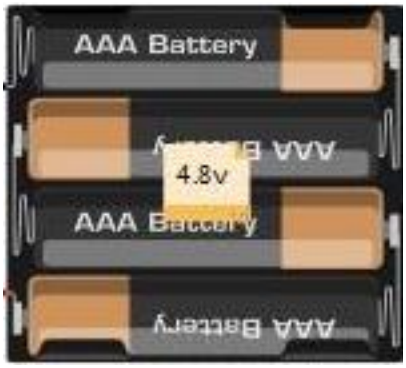


Figure 7.1: 4 X 1.2 Volt AAA batteries

7.3 Interfacing circuit

This section will show the inputs and outputs that will interface with the mechanical structure, inputs and outputs are shown in Fig 7.2.

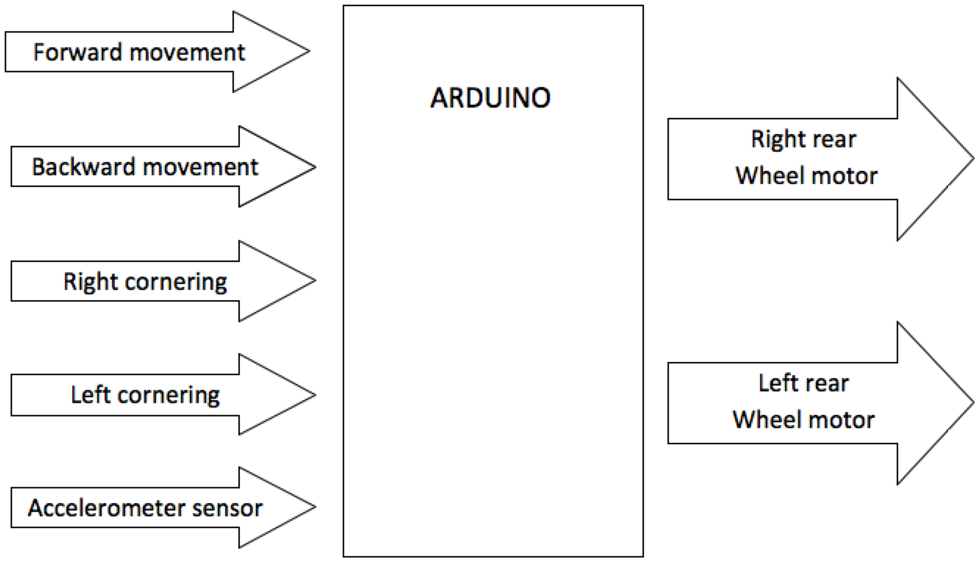


Figure 7.2: Inputs and Outputs of the system

7.3.1 Inputs

The input shown in Fig 7.2 will be explained below:

1) Forward movement signal: is a voltage signal from the car main circuit indicates that the car should move forward.
2) Backward movement signal: is a voltage signal from the car main circuit indicates that the car should move backward.
3) Right cornering signal: is a voltage signal from the car main circuit indicates that the car should move to the right.
4) Left cornering signal: is a voltage signal from the car main circuit indicates that the car should move to the left.
5) Accelerometer sensor: is the part that gives the acceleration to be used in finding the actual and the desired yaw rate.

7.3.2 Outputs

1) Right rear wheel motor.
2) Left rear wheel motor.

The two DC motors are connected to Arduino microcontroller through H Bridge as shown in Fig 7.3, which is an electronic circuit that enables a voltage to be applied across a load in either direction.

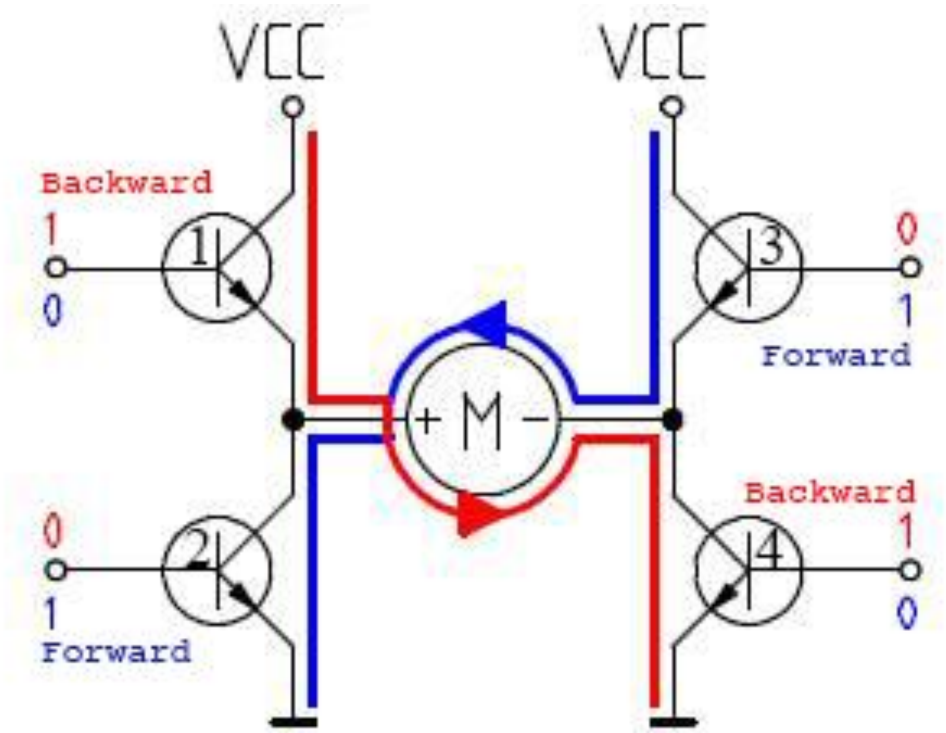


Figure 7.3: Structure of H Bridge

When the Arduino give a signal to forward motion, transistors 2 and 3 will open, and the motor will rotate to make the car move forward. Furthermore, when the Arduino give a signal to backward motion, transistors 1 and 4 will open, causing the car to move backward.

7.4 Wiring Diagram

After determining the electrical components, inputs and outputs. The wiring diagram was drawn as shown in Fig 7.4.

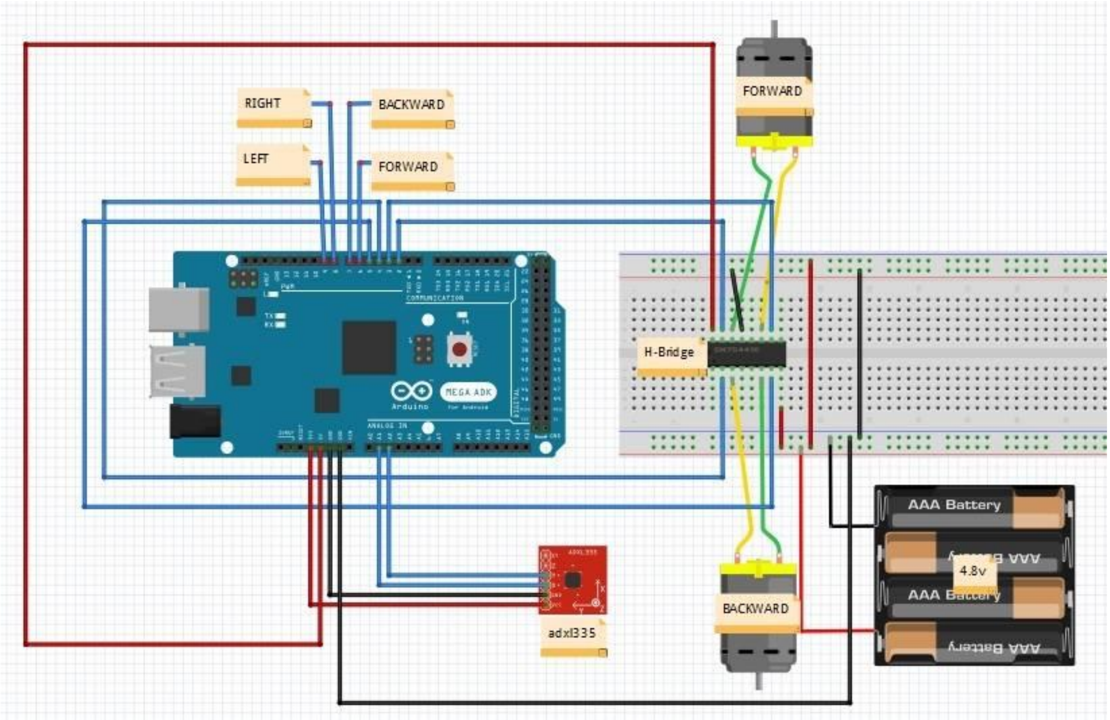


Figure 7.4: Wiring diagram of Yaw Stability Control system

Chapter 8

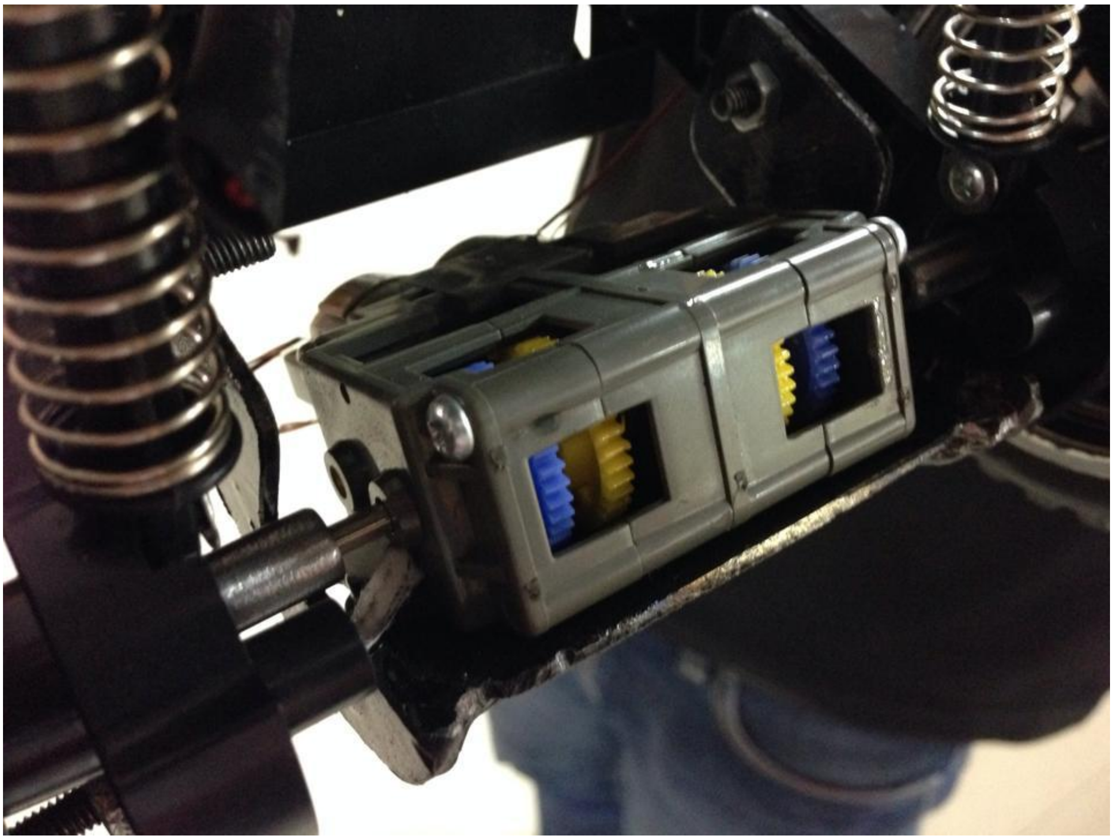
Building

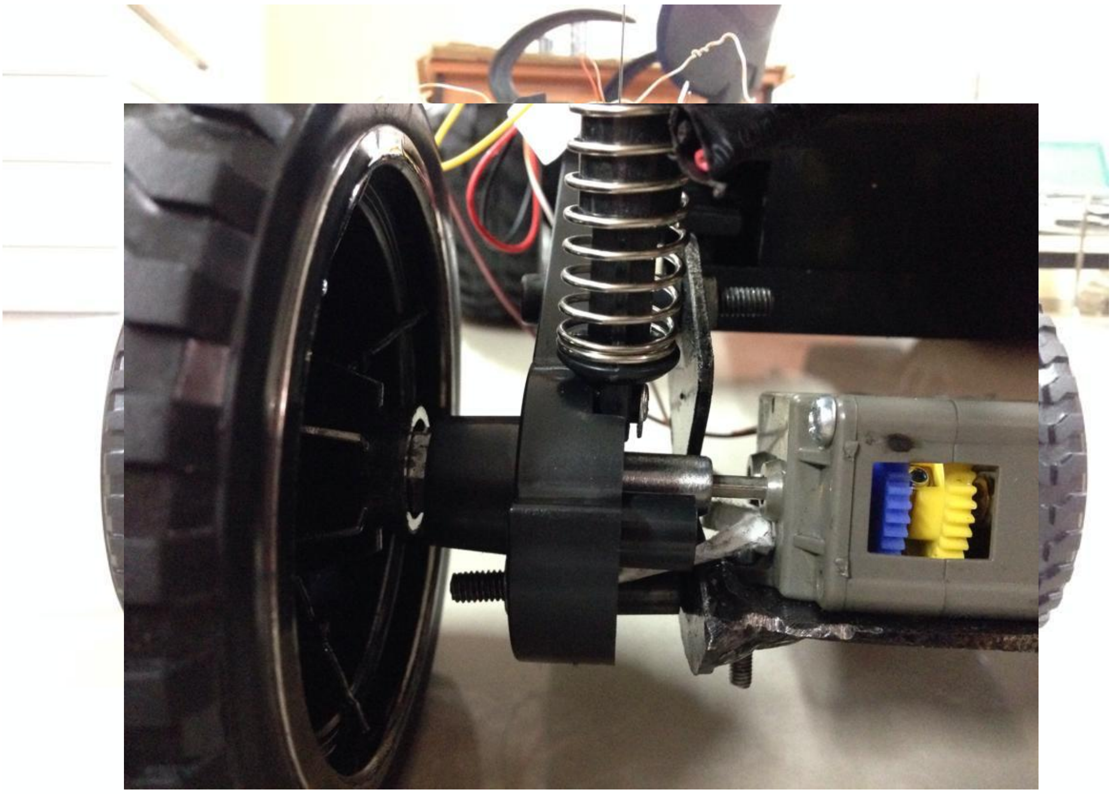
8.1 Introduction

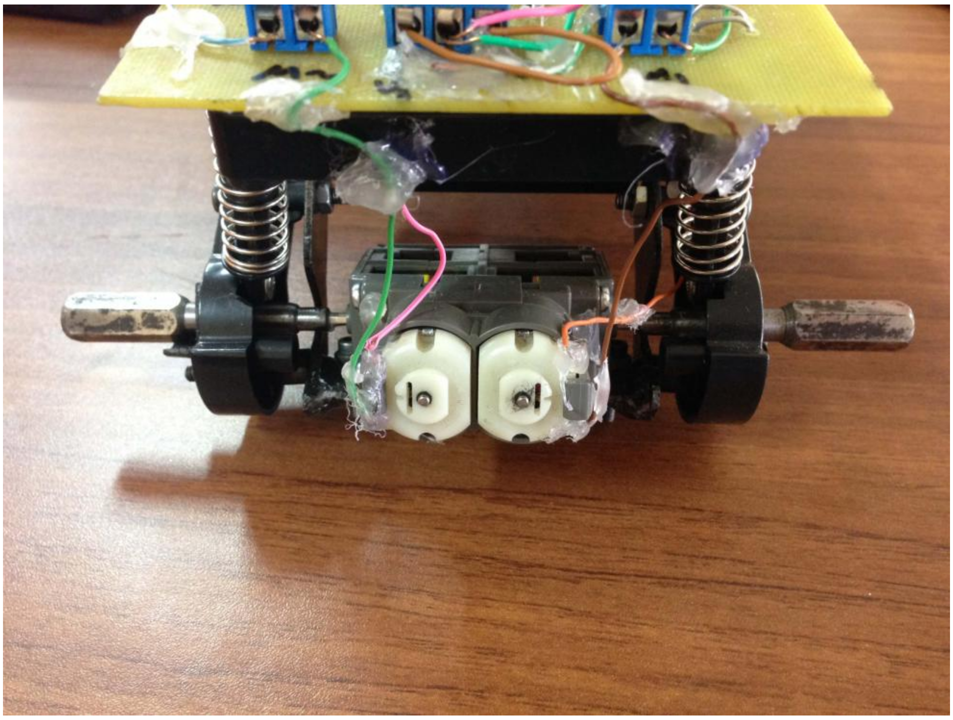
After determining the hardware components, the software design and the electrical circuit design, we started the assembly process, in which all parts were fitted in their suitable position and that the electrical components were installed in a proper way to insure that the car will function properly and safely.

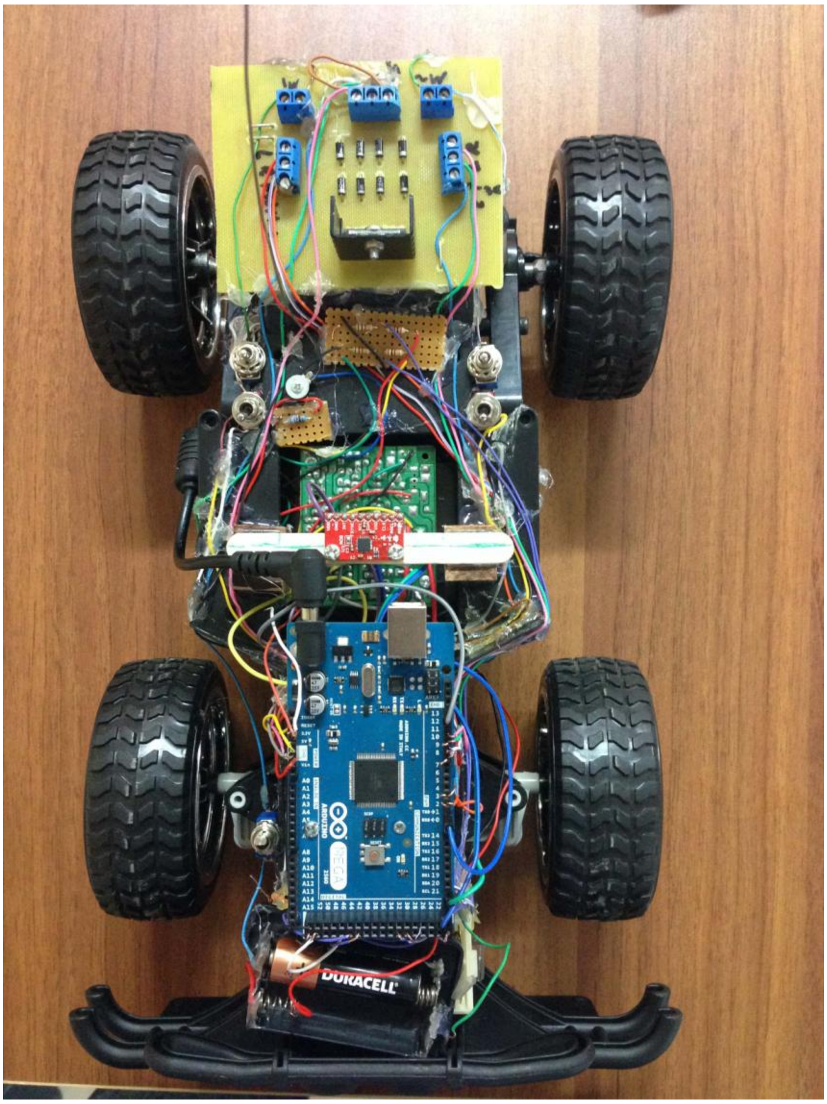
8.2 Final Model

The following figures show the final model of the car after it has been assembled. These steps consisted of attaching double gearbox kit to the car, and insure that it does not prevent the motion of RC car. After finishing the gearbox we started assembling the electrical components (Arduino Mega, accelerometer, motors and H Bridge).





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