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FACULTY OF MECHANICAL ENGINEERING

FAKULTA STROJNÍHO INŽENÝRSTVÍ

INSTITUTE OF SOLID MECHANICS, MECHATRONICS AND BIOMECHANICS

ÚSTAV MECHANIKY TĚLES, MECHATRONIKY A BIOMECHANIKY

DESIGN AND OPTIMIZATION OF THE HYDRAULIC SUSPENSION MODEL

NÁVRH A OPTIMALIZACE MODELU HYDRAULICKÉHO ODPRUŽENÍ

BACHELOR'S THESIS

BAKALÁŘSKÁ PRÁCE

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Assignment Bachelor's Thesis

Institut:	Institute of Solid Mechanics, Mechatronics and Biomechanics
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As provided for by the Act No. 111/98 Coll. on higher education institutions and the BUT Study and Examination Regulations, the director of the Institute hereby assigns the following topic of Bachelor's Thesis:

Design and optimization of the hydraulic suspension model

Brief Description:

The chassis suspension system is an integral part of every vehicle, designed to ensure the best possible tyre to road contact and ride comfort. Particularly on racing vehicles, the suspension behaviour must be tuned to avoid unwanted body oscillations during sharp manoeuvres.
One possible suspension solution is a system of separate hydraulic circuits that control the behaviour of the individual modes of suspension oscillation (heave, pitch, roll, warp).
The aim of this work is to create a simulation model of this hydraulic suspension and to optimize its parameters for different driving scenarios.

Bachelor's Thesis goals:

1. Conduct a research in the field of hydraulic suspension of chassis, its modelling and optimization of parameters.
2. Select a suitable tool to simulate the motion of the vehicle on the road.
3. Create a simulation model of the hydraulic suspension of the vehicle, with the input data from the aforementioned driving simulator.
4. Optimize the parameters of the suspension model to maximize its stability and response speed.
5. Test and evaluate the behaviour of the proposed model for different driving scenarios.

Recommended bibliography:

ZHANG, Nong; SMITH, Wade A. a JEYAKUMARAN, Jeku. Hydraulically interconnected vehicle suspension: background and modelling. Online. Vehicle system dynamics. 2010, roč. 48, č. 1, s. 17-40. ISSN 0042-3114.

Deadline for submission Bachelor's Thesis is given by the Schedule of the Academic year2023/24

In Brno,

L. S.

prof. Ing. Jindřich Petruška, CSc. Director of the Institute	doc. Ing. Jiří Hlinka, Ph.D. FME dean

Summary

This thesis focuses on design and optimization of a decoupled hydraulic suspension model with goal of achieving desired dynamical behaviour of a car. Suspension model is created in Matlab, Simulink. This thesis describes the process of designing the suspension model, finding input into its simulation, determining output of the simulation and process of its optimization. At the end, drawing conclusions from the optimization results.

Abstrakt

Tato práce se zaměřuje na návrh a optimalizaci modelu odděleného hydraulického odpružení s cílem dosáhnout požadovaného dynamického chování automobilu. Model zavěšení je vytvořen v prostředí Matlab, Simulink. Tato práce popisuje proces návrhu modelu zavěšení, zjištění vstupů do jeho simulace, určení výstupu simulace a proces jeho optimalizace. Nakonec vyvození závěrů z výsledků optimalizace.

Keywords

Suspension, modeling, optimalisation, automotive

Klíčová slova

Odpružení, modelování, optimalizace, automobilový průmysl

Bibliographic citation

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Statement of authorship

I declare that this thesis is original and was crated independently by me. I am stating that I did not use any other sources then listed in bibliography and declare no violation of author's rights.

In Brno, may 2024

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I would like to thank my family for supporting me in this study.

Matej Zrnčík

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1. Introduction

Suspension is a fundamental part of every automobile, it translates and dampens forces acting on a car's wheels into the car's body and vice versa. Car’s suspension is mainly loaded by dynamical forces created by accelerating the car's body and by roughness of a road surface.

Upper mentioned dynamical forces cause three main types of oscillations of the car's body relative to its suspension. These oscillations are heave (when all four wheels of the car are compressed by the same amount), pitch (rotation around car's lateral axis present mostly while braking and accelerating) and roll (rotation around car's longitudinal axis present mostly while driving through a corner). In conventional cars used for everyday commute, the goal of the suspension is to dampen road surface roughness and thus, make the ride more comfortable.

This thesis, however, is about suspension of a performance car whose goal isn’t making the ride more comfortable but to maximize contact with the road surface and to make the car as stable as possible and by that making the car go faster around a track. Today most performance cars use conventional suspension which can be described as each wheel of the car having its own spring and damper connecting it to the car's body and having a antiroll bar between the front and rear pair of wheels respectively. This form of suspension brings a big disadvantage, modifying characteristics of any suspension device (springs, dampers) will affect suspensions response to all the oscillation modes and thus making a suspension setup a difficult task often times leading to compromises instead of desired results.

This disadvantage can be solved by implementing a decoupled suspension system. A system like this will decouple the main oscillation modes from each other, so each and every mode is sprung and damped separately, making it easier to adjust the suspension for ideal ride behaviour. This can be achieved by creating a hydraulic suspension with hydraulic circuits arranged in a specific way so that it isolates individual oscillation modes. Besides that, this type of hydraulic suspension has also a advantage of occupying much less volume inside a car's chassis than a conventional suspension and also brings possibility of placing certain parts of the suspension anywhere inside the car, thanks to hydraulic lines transferring force independently of distance. This allows much more freedom while designing a car's chassis.

Now the question is, what should the upper mentioned ideal ride behaviour be in this context. In thesis assignment there is stated, that ideal ride behaviour is when the suspension will go into a stable position in a fastest possible time with minimum oscillations after applying load to it. In the process of creating this thesis, it was revealed that in this context ideal ride behaviour is rather minimal movement of the cars body relative to the road surface, this will benefit the aerodynamic efficiency of the car, next criterium is having force acting on a cars tyres as stable as possible, this will improve grip between the car and the road.

The goal of this thesis is to design, and model decoupled hydraulic suspension system, simulate its behaviour and to optimize its parameters of spring stiffness and damping coefficient for each oscillation mode to achieve upper mentioned criteria of a car's ideal ride behaviour.

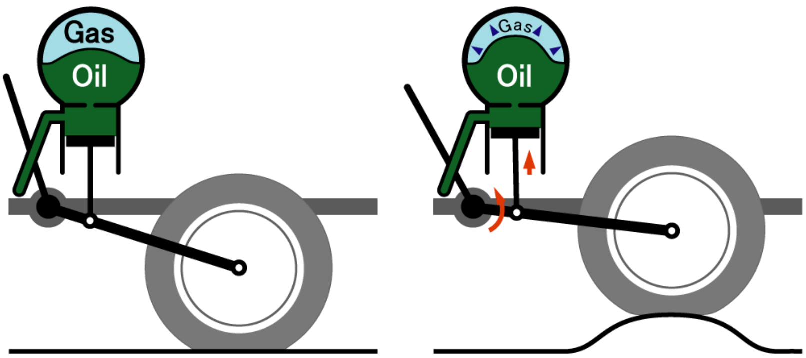
2. Research

2.1. Hydraulic suspension

Hydraulic suspension has been used in cars for almost seventy years now [1]. It replaces spring and dampers in conventional suspension with hydraulic cylinders and actuators, which uses hydraulic fluid to create a better ride quality for the car. There can be multiple motivations for having a hydraulic suspension in a car, in most cases this type of suspension is trying to improve ride comfort in passenger luxury cars, but it can be also used in performance cars to make the car faster and more responsive, or in more rare cases it can be used to save up space inside the car's chassis because hydraulic suspension usually occupies much less volume than conventional suspension.

The first implementation of a hydraulic suspension into a car was for Citroën DS19 (Pic. 2.1) that debuted in 1955 with its Suspension oléopneumatique (Pic. 2.2). This suspension system used hydraulic fluid to directly transfer force from the wheels onto a membrane inside a sphere which has nitrogen gas on the other end of the membrane. Compressibility of this gas is used as a spring in this system. This system was developed with the goal of making the ride more comfortable and to allow the car to change ride height on rough terrain on French post second world war roads [2].

Pic. 2.1 Citroen DS19 [1]

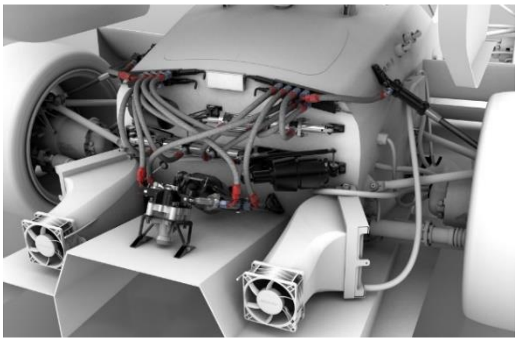


Pic. 2.2 Suspension oléopneumatique [2]

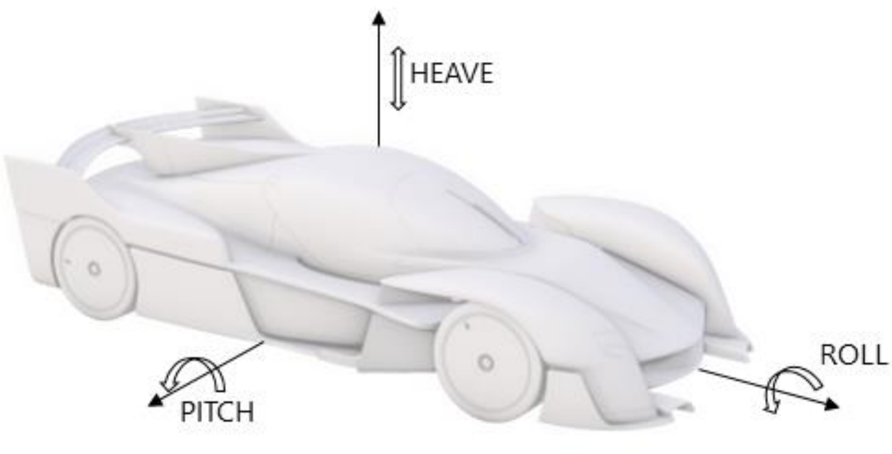
Another well known implementation of a hydraulic suspension, this time to improve performance of a car, was for 1993 Williams FW15C Formula1 car (Pic. 2.3). This suspension system has electronically operated hydraulic actuators at each wheel of the car so that the car's body can be as stable as possible relative to the road surface. This meant that the aerodynamic devices of the car could work more efficiently compared to a car with conventional suspension that has much more movement, therefore compromising the aerodynamics. An advantage like this meant that the car was able to go multiple seconds faster around a track than a conventional car [3].

Pic. 2.3 Detail of Williams FW15C Formula 1 car hydraulic suspension [19]

The last example of hydraulic suspension being used to improve car's performance is hydraulic decoupled suspension of Swiss formula student team AMZ Racing from 2017 (Pic. 2.4) [4]. This suspension system has hydraulic circuits arranged between its springs and dampers in a way, that it isolates oscillation modes of the car which are heave, pitch and roll (Pic. 2.5). This will make the suspension easily adjustable for desired dynamical behaviour of the car as mentioned in the thesis introduction.



Pic. 2.4 AMZ Racing hydraulic suspension [5]



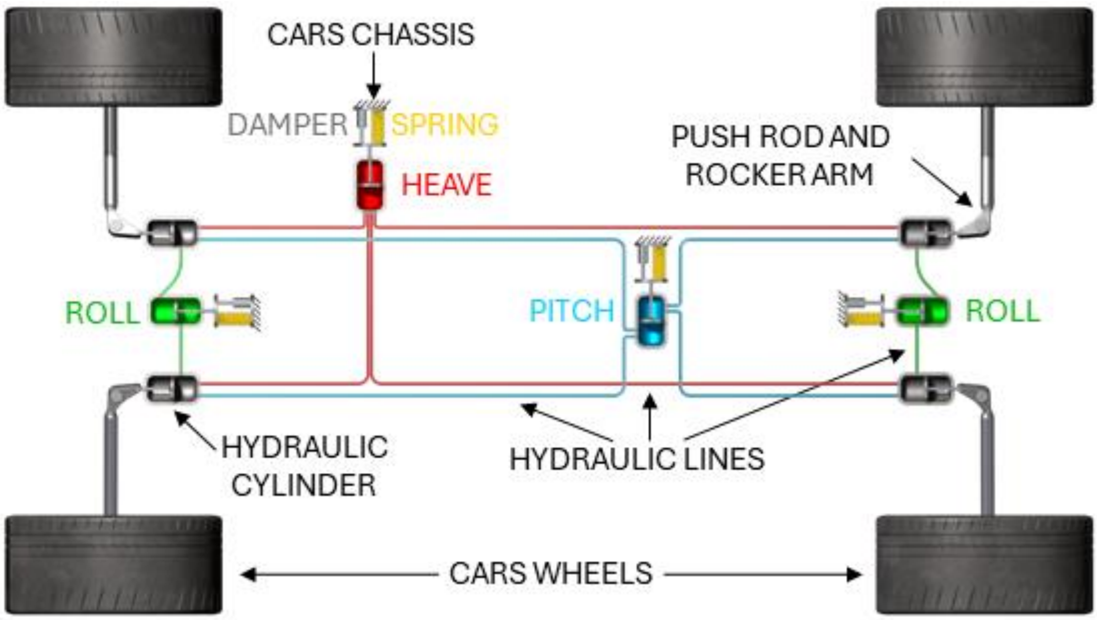
Pic. 2.5 Oscillation modes of a car

2.2. Suspension modelling

As mentioned in chapter 2.1 formula student team AMZ Racing created a decoupled hydraulic suspension system that isolates each individual oscillation modes of the car (Pic. 2.5). This means that when, for example, the car is braking, it rotates around its lateral axis creating pitch movement. In conventional suspension, this movement would load all four springs and dampers of the car, which are there also to compensate heave and roll. Therefore, setting up the conventional suspension to have certain response in pitch movement would mean that heave and roll response would be also affected, creating a need to compromise the response between pitch, heave and roll. AMZ Racing suspension has individual spring and damper for each of these oscillation modes, so when the car is braking only pitch spring and damper are loaded while load on springs and dampers for heave and roll doesn't change. This creates a possibility to adjust response for each oscillation mode independently of each other, making the suspension easier to adjust for desired dynamical characteristics. Proof of this improved dynamical behaviour can be found in source [5].

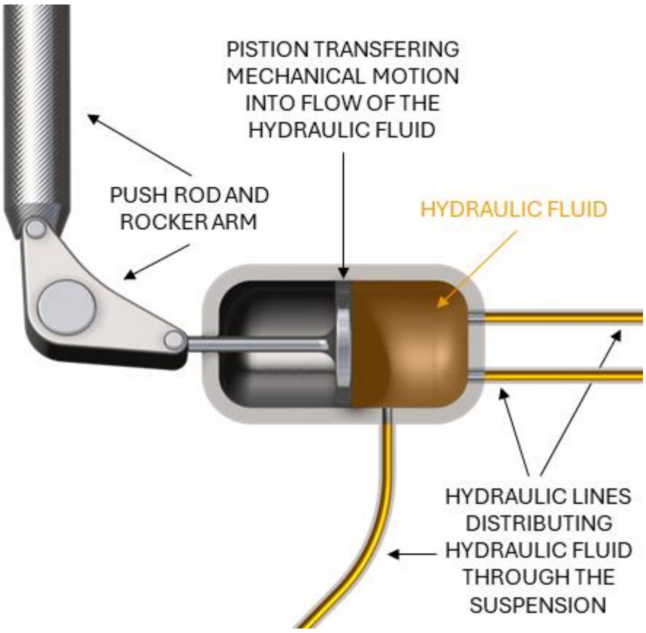
Suspension idea from team AMZ was used as main inspiration for this thesis. Explanation video from KYLE.ENGINEERS [6] showed hydraulic circuits of this suspension in detail and made inner workings of this suspension system clear. Using these sources, design of this thesis suspension model was set (Pic. 2.6). This thesis suspension model is going to be built on this design and simulated in Simulink.

It can be seen, in the Pic. 2.6, that this design doesn't include warp oscillation mode, unlike AMZ Racing suspension. That is because this suspension will be simulated only on flat surface, considering that the model car is perfectly symmetrical. Which means that the car's centre of weight is exactly in the middle of its wheelbase and in the middle of its track. Because of this symmetry, roll on front and rear axle will always be loaded by the same amount, therefore no warp load will be created in these simulations.



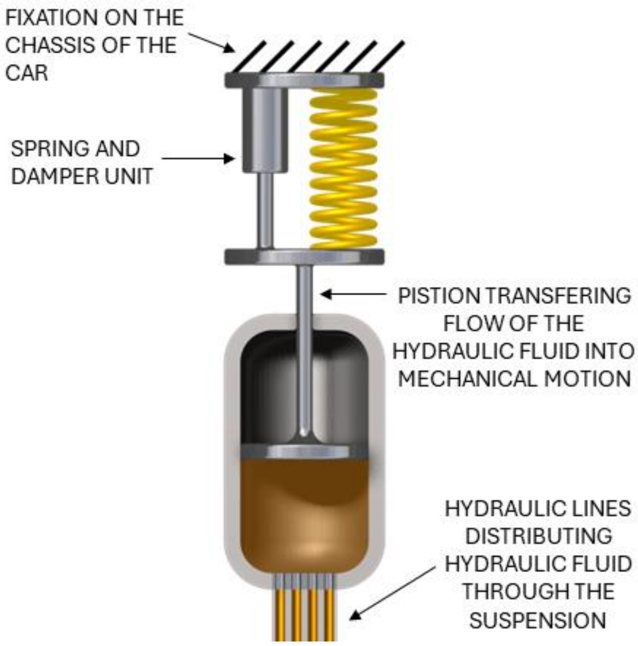
Pic. 2.6 Design of the decoupled hydraulic suspension [6]

On pic. 2.7, which is a close up of a pic. 2.6, visualization of the hydraulic cylinder that transfers force from the wheels into the hydraulic circuits and vice versa can be seen.



Pic. 2.7 Detail of hydraulic cylinder connected to push rod

On pic. 2.8, which is a close up of a pic. 2.6, a hydraulic cylinder that transfers force from the hydraulic circuit onto a spring and damper unit can be seen. This spring and damper unit is fixed to a car's chassis. Spring and damper unit like this is present for every oscillation mode of the car (heave, pitch, roll) and its spring stiffness and damping coefficient will be tuned in the optimization.



Pic. 2.8 Detail of hydraulic cylinder connected to spring and damper unit

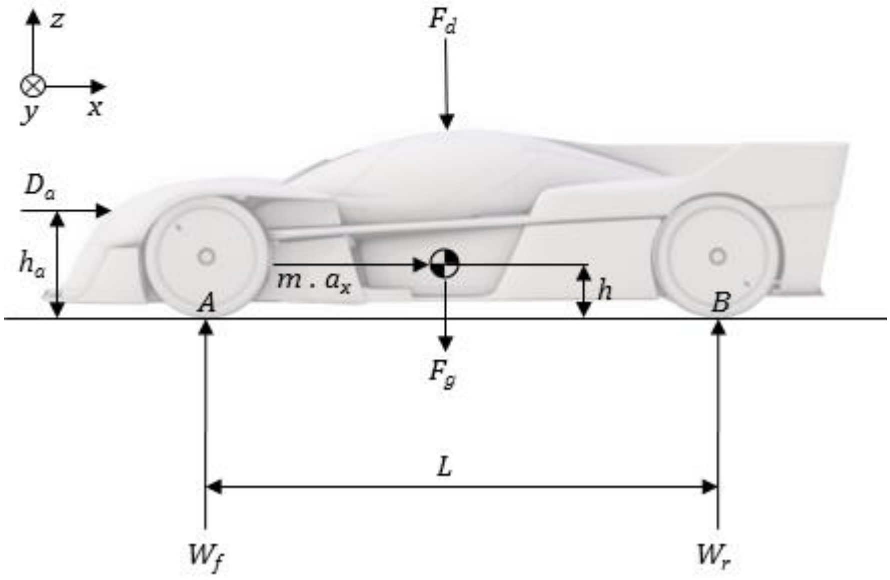
2.2.1 Finding the input for the simulation

Performance car suspension goal is to maximize grip of the tyres with road surface, and it’s achieving it by absorbing load created by cars body accelerations in lateral and longitudinal directions and also by absorbing disturbances from road surface unevenness. Therefore, when looking into sources of input for suspension simulation, dynamical forces created by cars inertia and kinematic excitation of the wheels from road surface unevenness are the ones to focus on.

The best approach of getting these dynamical forces data would be to apply an accelerometer to a real car, do some test runs around a track and then calculate these forces from gathered data of longitudinal and lateral acceleration in m*s^-2 over time. Unfortunately, this wasn’t possible to do while working on this thesis, so an alternative had to be found. Doing a computer simulation of a car driving around a track is probably the least cost and time expensive option. There are multiple software that will simulate a car driving around the track, for example VI-CarRealTime [7], but these softwares tend to be really complex, usually require subscription and lots of detailed car data which aren’t easily available.

After doing some research, OptimumLap software was found. This software simulates a car driving on a given trajectory as a point mass. By that neglecting dynamical transient effects, but the software description states that the simulations match real life data [8]. In this software only much less car data are needed because of point mass simulation approach and the data that are needed are much more easily accessible. OptimuLap works as follows, required car data like its mass, tyre parameters and aerodynamic parameters needs to be set, then a car's driving trajectory needs to be created in the software or can be imported. After these two tasks a simulation can be started and the software outputs resulting data like position, time, speed, accelerations, aerodynamic data and so on. For this thesis, OptimumLap data of longitudinal acceleration, lateral acceleration, aerodynamic drag force and aerodynamic downforce will be used to calculate dynamical load for each wheel of the car.

For these calculations basic equations for car dynamics will be used, considering a symmetrical car, these equations can be even more simplified. Longitudinal forces (Pic. 2.9) will be calculated using equations 2.3 and 2.6 [9]. Lateral forces (Pic. 2.10) will be calculated using equations 2.9 and 2.12 [10].



Pic. 2.9 Forces acting on the car in longitudinal direction [9]

$$\sum M_A = 0 \tag{2.1}$$

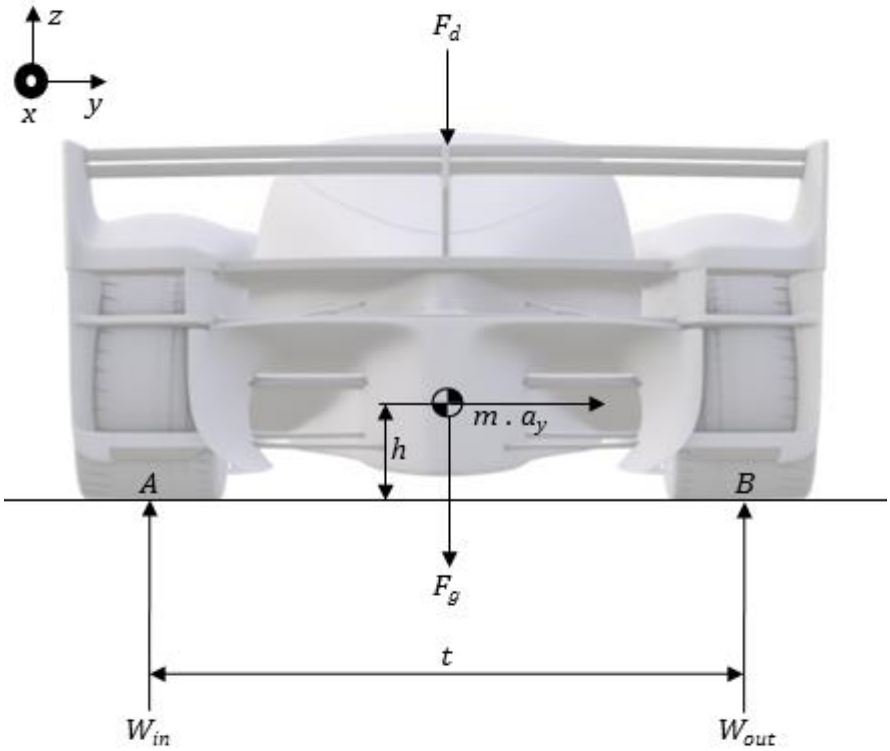
$$W_r L - F_d \frac{L}{2} - D_a h_a - m a_x h - F_g \frac{L}{2} = 0 \tag{2.2}$$

$$W_r = \frac{1}{L}\left(F_d \frac{L}{2} + D_a h_a + m a_x h + F_g \frac{L}{2}\right) \tag{2.3}$$

$$\sum M_B = 0 \tag{2.4}$$

$$-W_f L + F_d \frac{L}{2} - D_a h_a - m a_x h + F_g \frac{L}{2} = 0 \tag{2.5}$$

$$W_f = \frac{1}{L}\left(F_d \frac{L}{2} - D_a h_a - m a_x h + F_g \frac{L}{2}\right) \tag{2.6}$$



Pic. 2.10 Forces acting on the car in lateral direction [10]

$$\sum M_A = 0 \quad (2.7)$$

$$W_{out}t - F_d\frac{t}{2} - F_g\frac{t}{2} - ma_y h = 0 \quad (2.8)$$

$$W_{out} = \frac{1}{t}\left(F_d\frac{t}{2} + F_g\frac{t}{2} + ma_y h\right) \quad (2.9)$$

$$\sum M_B = 0 \quad (2.10)$$

$$-W_{in}t + F_d\frac{t}{2} + F_g\frac{t}{2} - ma_y h = 0 \quad (2.11)$$

$$W_{in} = \frac{1}{t}\left(F_d\frac{t}{2} + F_g\frac{t}{2} - ma_y h\right) \quad (2.12)$$

Equations symbols meaning:

W_f is force acting on cars front wheels pair

W_r is force acting on cars rear wheels pair

W_{in} is force acting on cars front left and rear left wheels

W_{out} is force acting on cars front right and rear right wheels

F_d is aerodynamic downforce

D_a is aerodynamic drag force

F_g is gravity force

m is cars mass

a_x is longitudinal acceleration

a_y is lateral acceleration

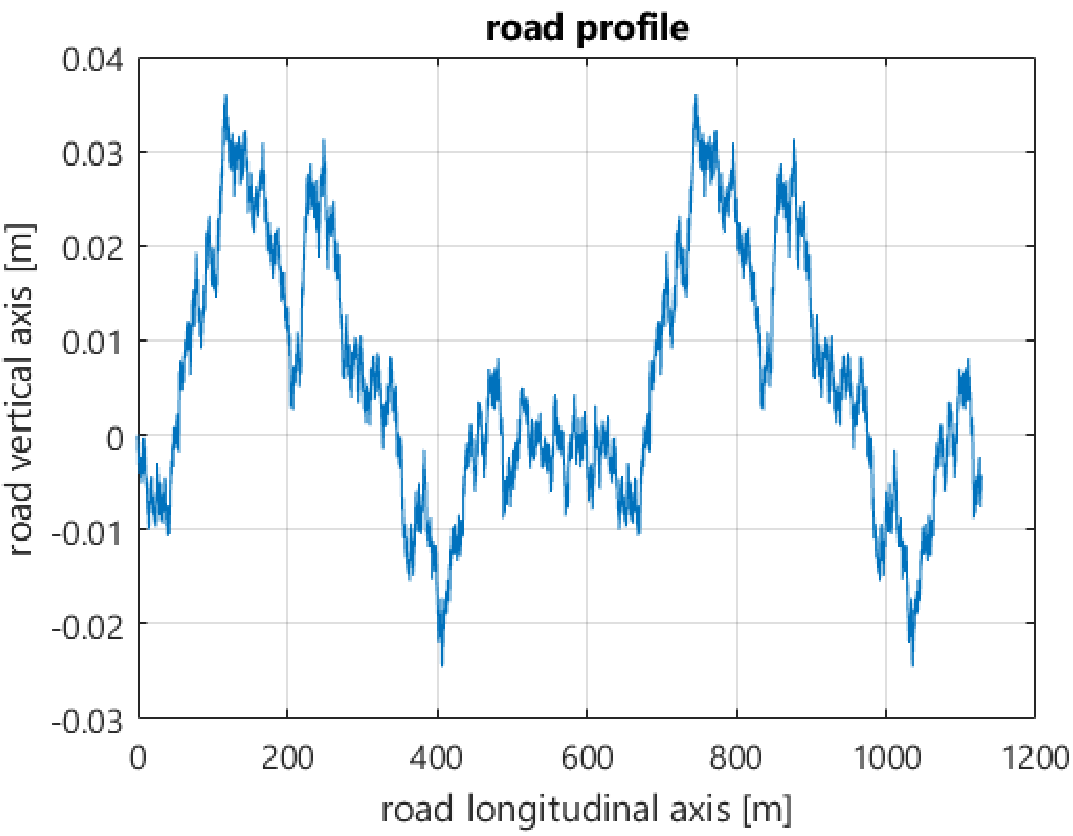
h is cars centre of gravity height

h_a is cars aerodynamic frontal centre of pressure

L is cars wheelbase length

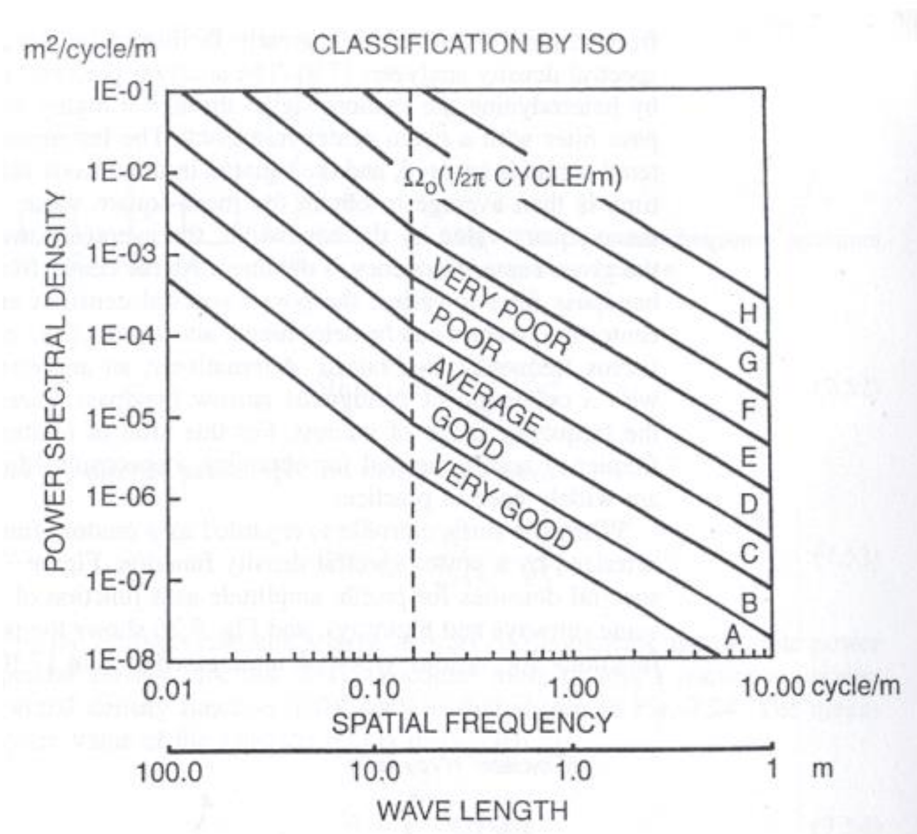
t is cars track length

Then the input of kinematic excitation of the car's wheels from road surface unevenness is needed. First idea was to find freely accessible road profile data on the internet. These data are created by accurately measuring the vertical dimension of the road over the horizontal dimension of the road in meters, creating a profile that is similar to what is shown in pic. 2.11. After finding that this kind of data are very hard to access, an alternative source had to be found. This alternative source is a Matlab script for generating a random typical road surface profile [11]. An example of randomly generated profile by this script can be seen on pic. 2.11. This script allows user to set parameters like the length of generated profile and resolution of this profile in data points per meter of its length, which is very helpful for its implementation in this simulation.

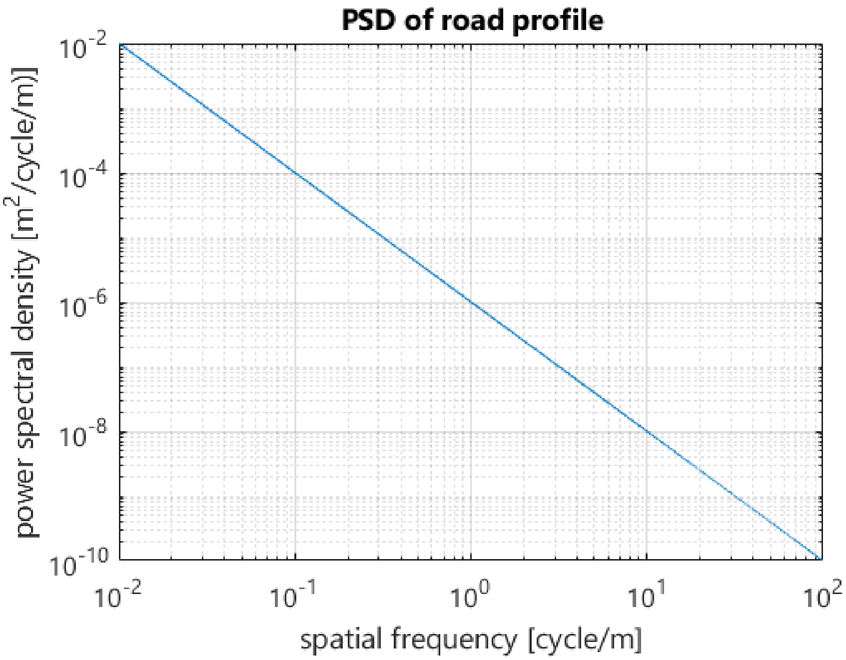


Pic. 2.11 Road profile generated by Matlab script [11]

To prove that road profile shown in pic. 2.11 is useful representation of a real road surface, power spectral density comparison can be used [12]. On the pic. 2.12 power spectral density for different road surface qualities can be seen. On the pic. 2.13 power spectral density of the generated road profile from pic. 2.11 can be seen, this diagram is generated by the same script that also generates the road profile from pic. 2.11. Comparing these two diagrams, it's visible that the generated road profile falls into the spectral density range of a real average road profile, drawing a conclusion that the generated road profile is realistic enough to be used in the simulation.



Pic. 2.12 ISO road surface classification by power spectral density [12]



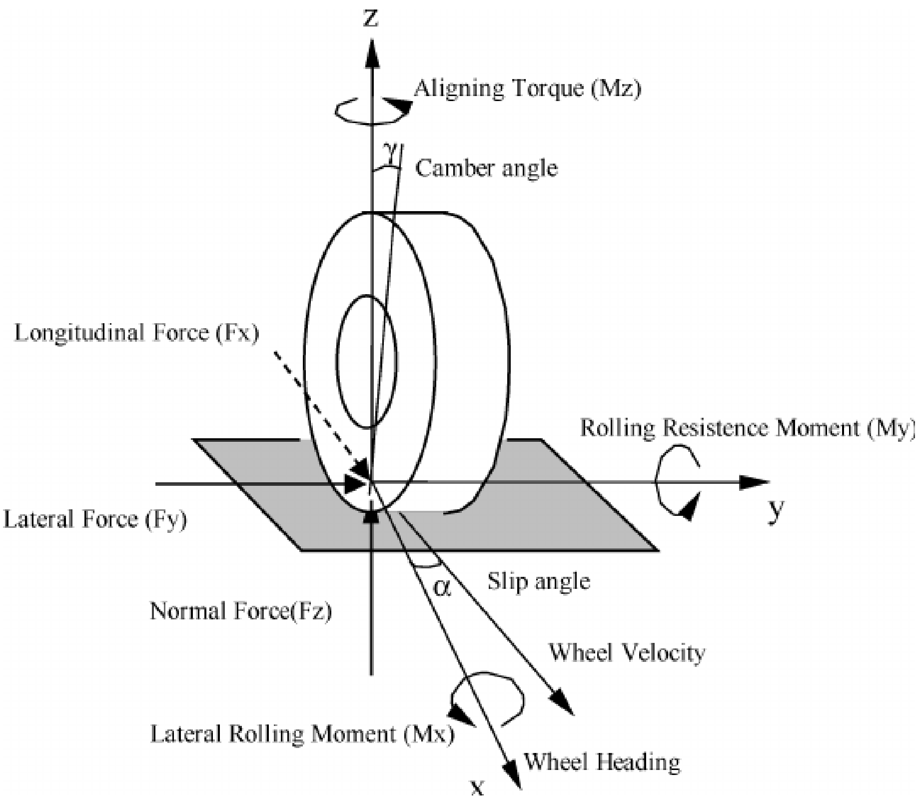
Pic. 2.13 Power spectral density of the generated road profile [11]

2.2.2 Determining the output of the simulation

Output variables of this simulation that are going to be used in the optimization are.

Positions of springs and dampers for the individual oscillation modes. These positions are going to be measured as amount of displacement in metres from base position, which occurs when the load on the spring is zero Newtons. Positions are negative when the spring and damper are compressed from its base position and positive when they are extended from this position. Positions of spring and damper for each individual oscillation mode are always the same because the spring and damper are connected in one assembly like in conventional coilover.

Next output are forces acting on the car's tyres, respectively wheels. In real life, there are multiple types of forces acting on a car's tyre (Pic. 2.14) [13]. But in order to make this simulation simpler, focus is mainly on normal force. This force is measured, in this simulation, by multiplying the amount of the tyre deflection in meters in its vertical axis and the tyre's vertical stiffness in newtons over meters. It gains positive values when the tyre is compressed and zero value when the tyre doesn’t experience any load, i.e. it’s lifted off the road surface.



Pic. 2.14 Forces acting on cars tyre [13]

2.3. Optimization and its goals

Optimization in the context of car suspensions can be done with various motivations. Usually for conventional cars, suspension optimization is aiming to increase comfort and stability of the car. This is mostly done by adjusting the amount of the suspensions stiffness and damping, or by adjusting its kinematics.

The focus of this thesis, however, is on performance car suspension optimization, where other factors are desirable. In most performance car suspensions, the main goal is to maximize grip, i.e. friction between the tyres and the road surface. Grip of the car's tyre is directly dependent on its normal force. Higher normal force means higher friction force, which gives higher grip.

While figuring out how to achieve this goal, the first idea was to make a suspension that goes into stable position in the least amount of time with minimum oscillation, as written in the thesis assignment. Later, it was revealed that this attribute wouldn't necessarily maximize the car's grip, but only make the car's body very stable. Which means that most of the forces that act on the car's body would be dissipated in the car's tyres, making the tyre load varied and thus making the car less stable and predictable. After some consultations, it was revealed that to achieve this goal of maximizing grip, the suspension should be optimized to give tyre load that is as constant as possible. This, however, would result in suspensions springs and dampers dissipating most of the forces that are acting on the car, which creates a car that has a lot of movement of its body relatively to the road surface. While this factor is good for the tyres of the car, it negatively affects other very important part of a performance car, which is aerodynamics. Majority of performance cars try to achieve aerodynamic downforce values as high as possible, because having more downforce acting on the car means that higher normal force is created on the tyres, which means higher grip. Aerodynamic devices in a car with lot of body movement tends to give varying and unstable downforce values, which creates unstable normal force, which means unstable grip.

So, the real goal of car's suspension optimization is a compromise between having the car body with minimal movements relative to the road surface and tyre loads that are as constant as possible.

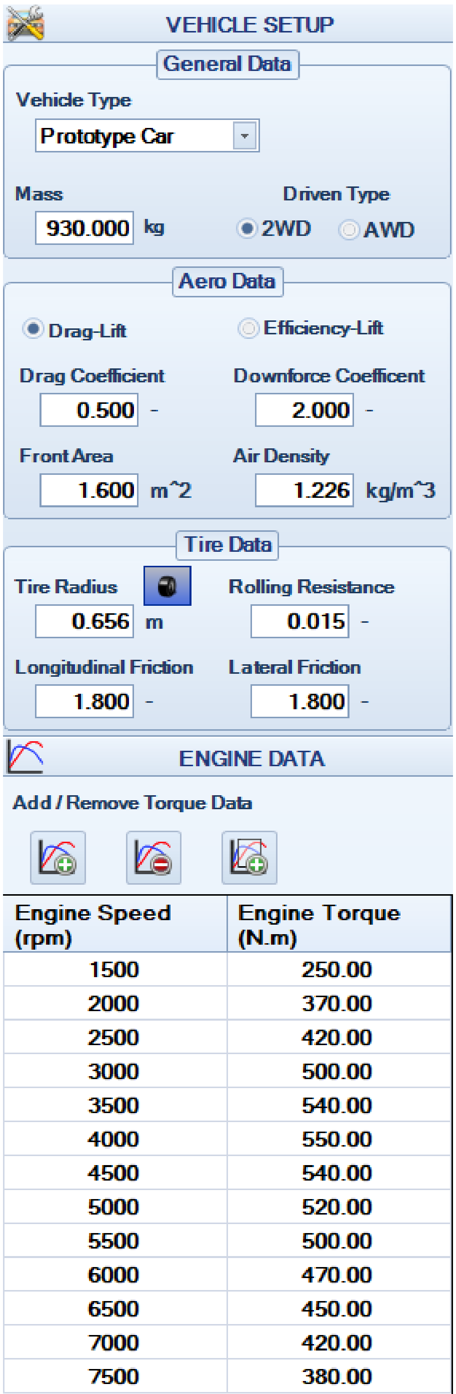
To execute this optimization for this thesis, Simulink response optimization looks as a most efficient tool. This tool is part of a control systems toolbox for Matlab Simulink, and it optimizes given parameters to meet given design requirements or to find local minimum of a function using various optimization algorithms [14]. Plus it's easy to use, and it's built in the Simulink workspace where the simulation is running, making it even more convenient.

3. Approach and results

3.1. Creating input for the simulation

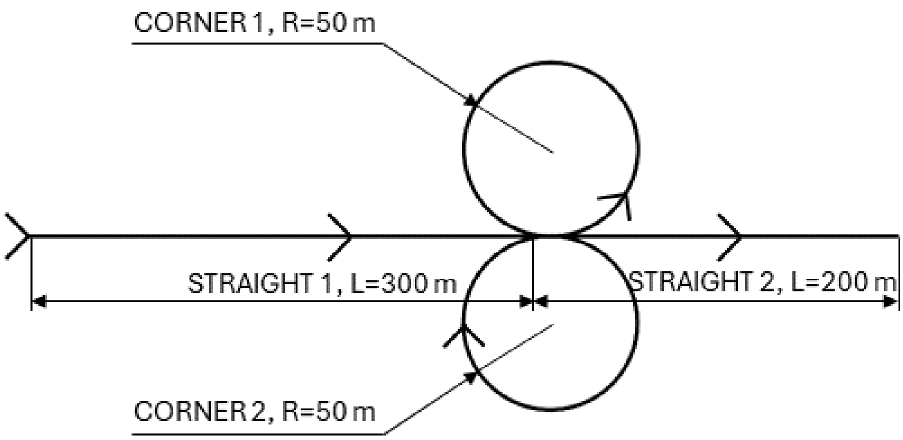
The first step of the simulation input creation is to execute the OptimumLap simulation mentioned in chapter 2.2.1. Firstly, a model car for the OptimumLap simulation needs to be set. For this task, a LeMans Prototype car, Oreca 07 (Pic. 3.1) will be used as source of this model car data. In Pic. 3.2 all car data given to OptimumLap can be seen [15] [16] [17].

Pic. 3.1 Oreca 07 prototype car used for gathering data for model car



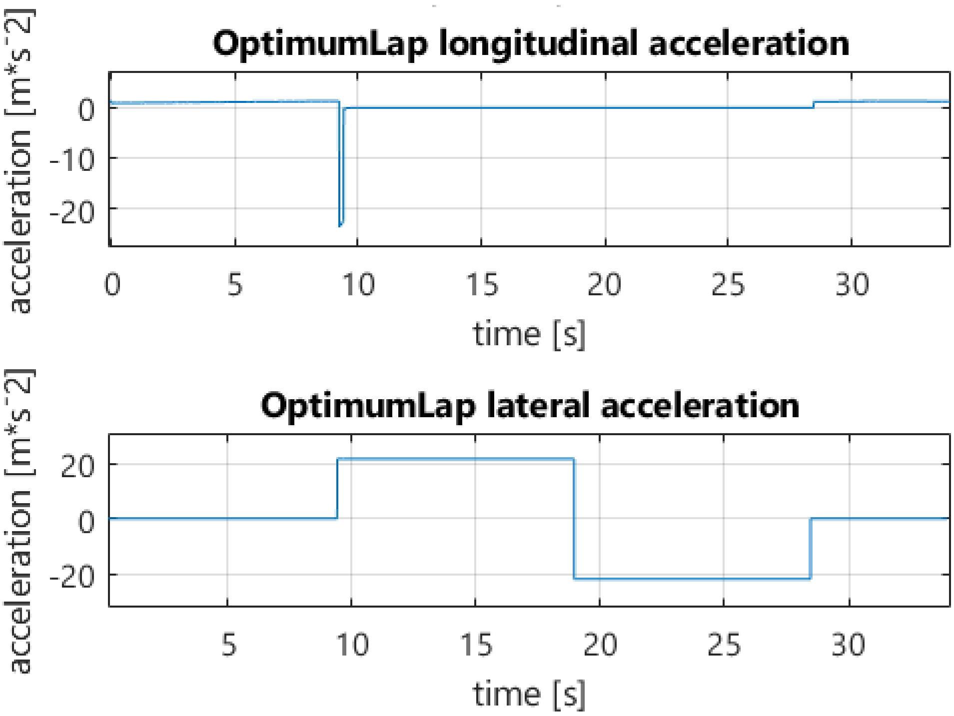
Pic. 3.2 Simulated car in OptimumLap software

Next thing needed for the OptimumLap simulation is the car's driving trajectory. For this objective, a typical skidpad was chosen (Pic. 3.3). This skidpad should be good enough to test suspension for loads from longitudinal and lateral acceleration while keeping the simulation time relatively short and giving simple results that are easy to read.

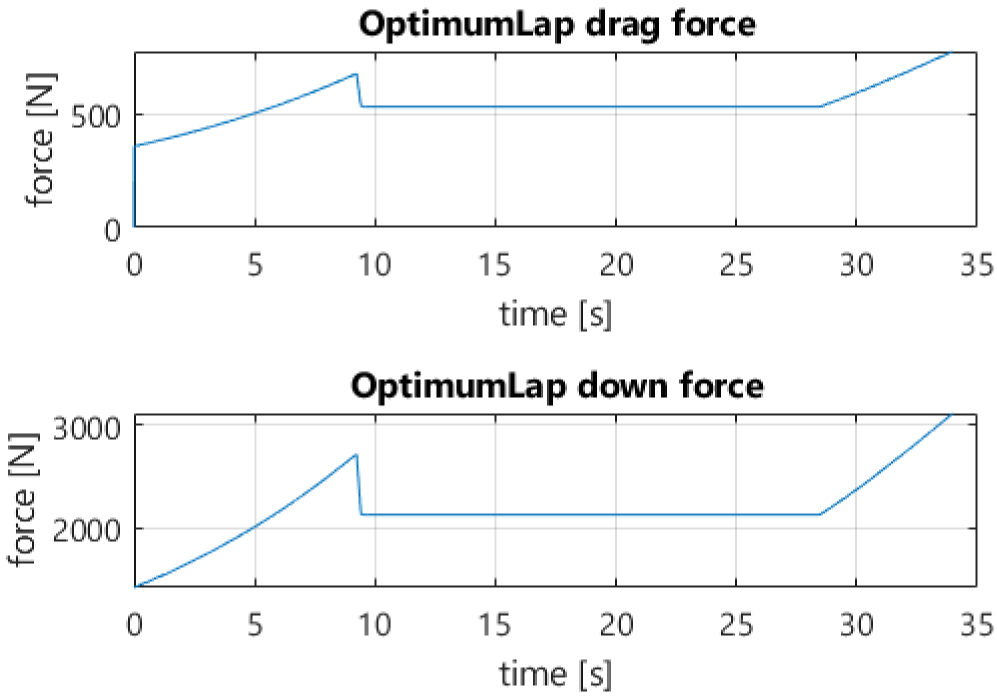


Pic. 3.3 Simulated trajectory in OptimumLap software

On the pic. 3.4 and 3.5, exported data from successful OptimumLap simulation run can be seen. Pic. 3.4 show values of the longitudinal and lateral accelerations over time acting on the car's centre of mass while driving around the upper mentioned trajectory. These accelerations data look a bit questionable with their smoothness and big gains of different values in very short amount of time. In real life, these data tend to be more noisy and gains in acceleration values tends to be much more smooth. Because of very limited input data sources, it was decided to believe these data from OptimumLap software. Pic. 3.5 show values of aerodynamic drag force and aerodynamic downforce that are acting on the car's centre of mass. These four sets of data will be used to calculate forces acting on the individual wheels of the car by using equations 2.3, 2.6, 2.9 and 2.12 from chapter 2.2.1.

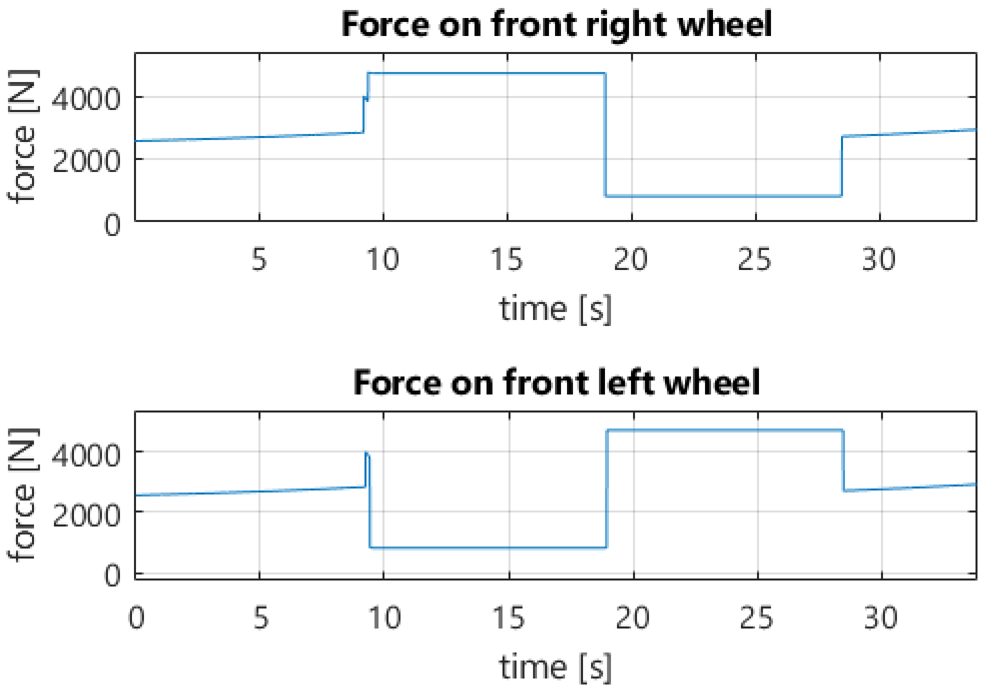


Pic. 3.4 Longitudinal and lateral acceleration data from OptimumLap simulation plotted against time

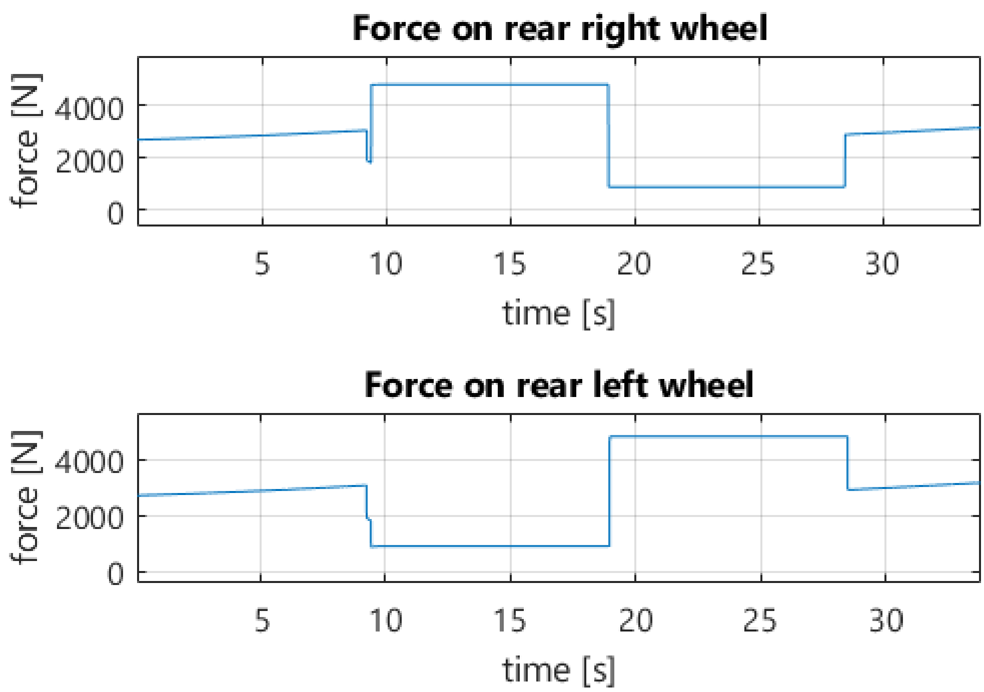


Pic. 3.5 Aerodynamic drag force and down force data from OptimumLap simulation plotted against time

On the pic. 3.6 and 3.7, dynamical forces acting on individual wheels of the car can be seen. It is obvious that these forces are superimposition of the data from pic. 3.4 and 3.5. In all four plots, at circa eight second, a spike in these data can be seen. This spike is caused by the car's sudden transition from braking in the straight line to executing a left-hand turn. Realism of this event is a bit questionable like mentioned in the article above.

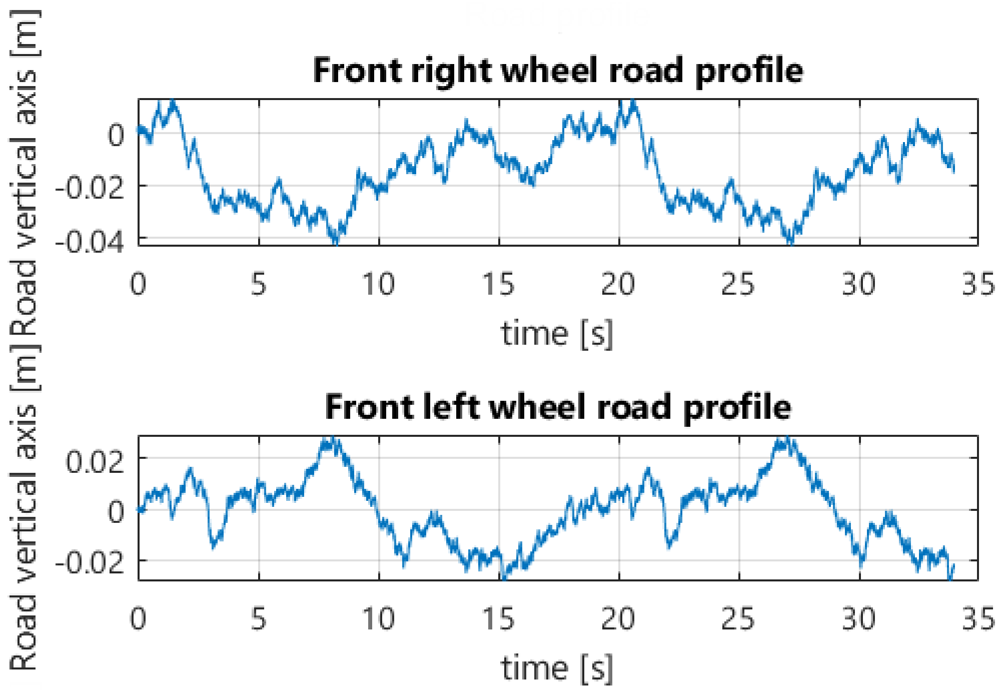


Pic. 3.6 Dynamical forces for front wheels of the car plotted against time

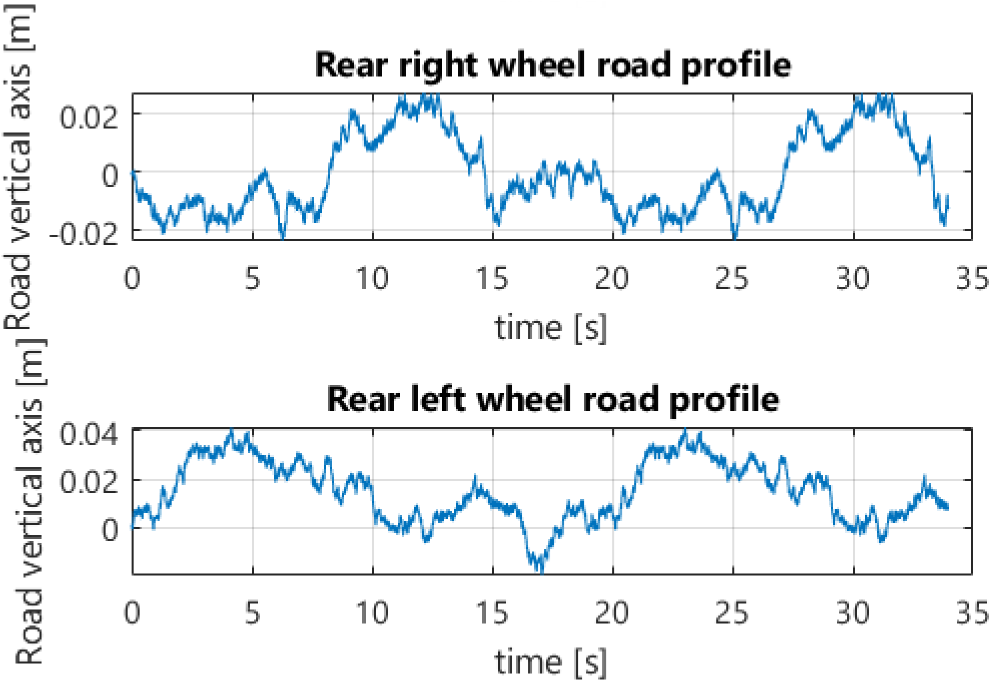


Pic. 3.7 Dynamical forces for rear wheels of the car plotted against time

On the pic. 3.8 and 3.9, kinematic excitation from road surface acting on the individual tyres of the car can be seen. Source of these figures is described in chapter 2.2.1. These data are plotted against time while in pic. 2.11 the road profile is generated with length dimension on its horizontal axis. That's because the Matlab script that generated these road profiles generates them against length, but Simulink needs time dependent data as an input for the simulation. So the data were converted from length dependent to time dependent in a Matlab script by the use of car's speed and simulated time from OptimumLap simulation data.



Pic. 3.8 Road profile for front wheels of the car plotted against time



Pic. 3.9 Road profile for rear wheels of the car plotted against time

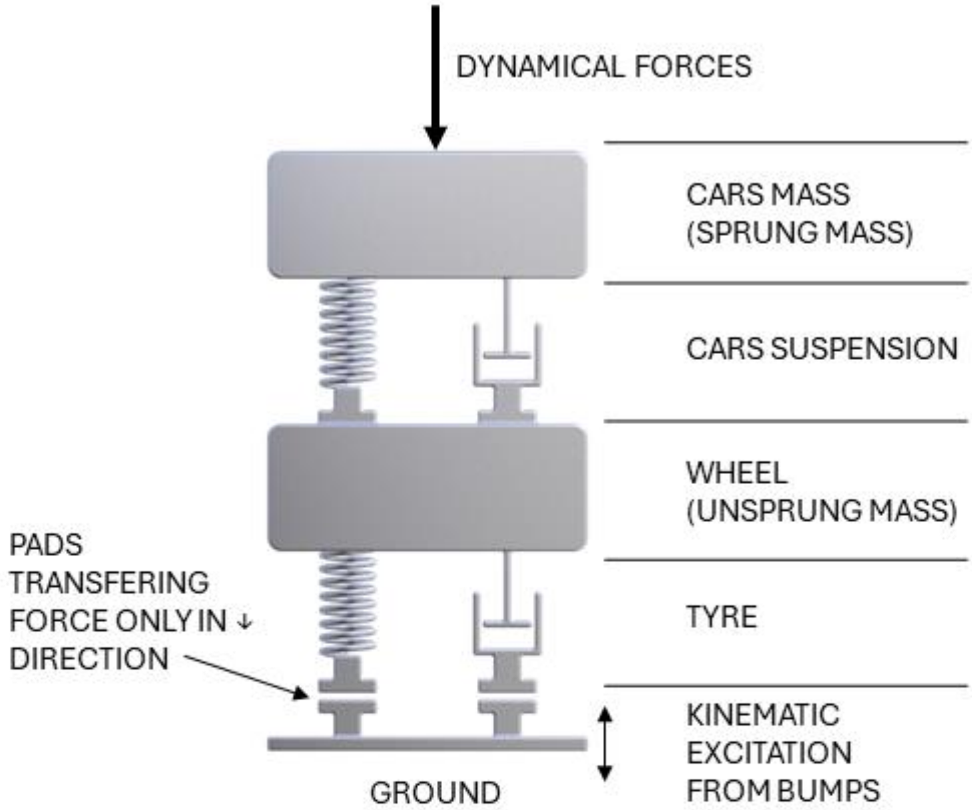
3.2. Creating the Simulink model

Model of this suspension will be simulated in Matlab Simulink, using mostly Simscape hydraulic library, multibody library and mechanical library. Hydraulic fluid inside this suspension will be considered as isothermal and incompressible, so there are no losses in the transferred force and no parasitical damping of the fluid.

In this thesis, the suspension is modelled for a completely symmetric car, which means that its centre of gravity lies exactly in the middle of its wheelbase and in the middle of its track. This means that every wheel of the car carries the same amount of weight. Chassis of the car is considered as infinitely stiff, so there is no chassis flexing further loading the suspension and creating asymmetry. This also means that front and rear roll springs will always be loaded by the same amount.

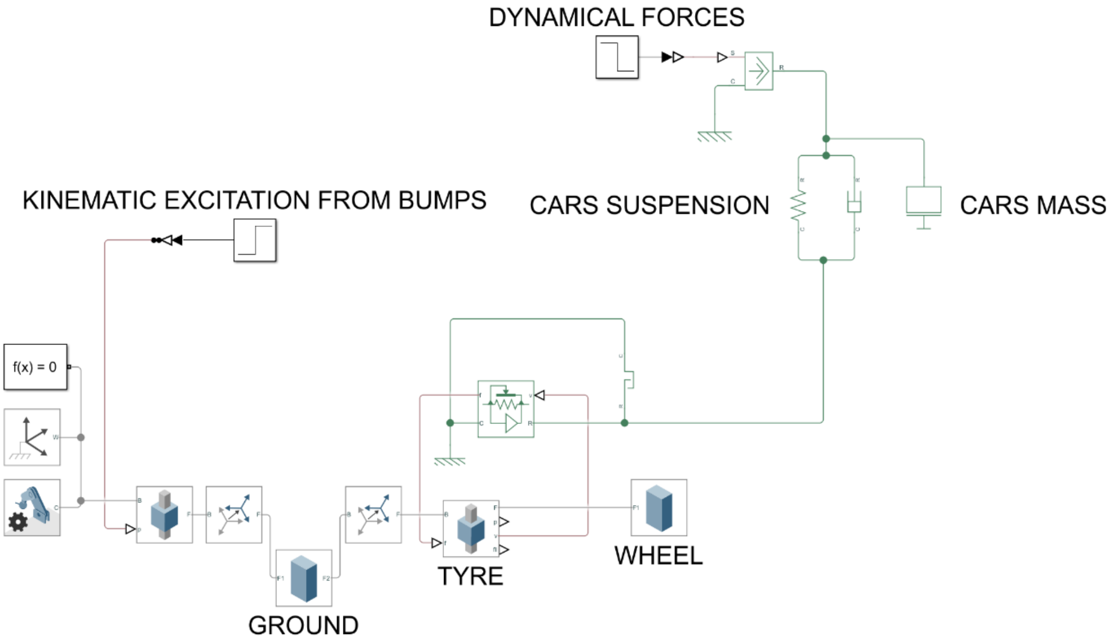
3.2.1. Base model of the suspension

To start creating a model of the suspension in Simulink, some base model on which a final model will be built is needed. This base model can be seen on pic. 3.10.



Pic. 3.10 Basic quarter car model [20]

Note that pic. 3.10 depicts only quarter car model, that means only one out of four wheels of the car. A full car model would consist of four of this quarter car models, one for each wheel of the car while the sprung mass would be equally divided between them, if a fully symmetric car is considered. This model is only one dimensional, this means that ground, wheel, and cars mass in the pic. 3.10 has one degree of freedom and can only move in vertical direction. A base model like this is replacing rotational motion of a real suspension by translational, therefore neglecting certain dynamical transient effects, but it should be realistic enough for this simulation while keeping the Simulink model simple and easy to work with. A vertical tyre stiffness and damping coefficient values are needed for the tyre model of this Simulink model. Stiffness of 275 kN/m and damping coefficient of 750 N/m*s were chosen according to source [18].

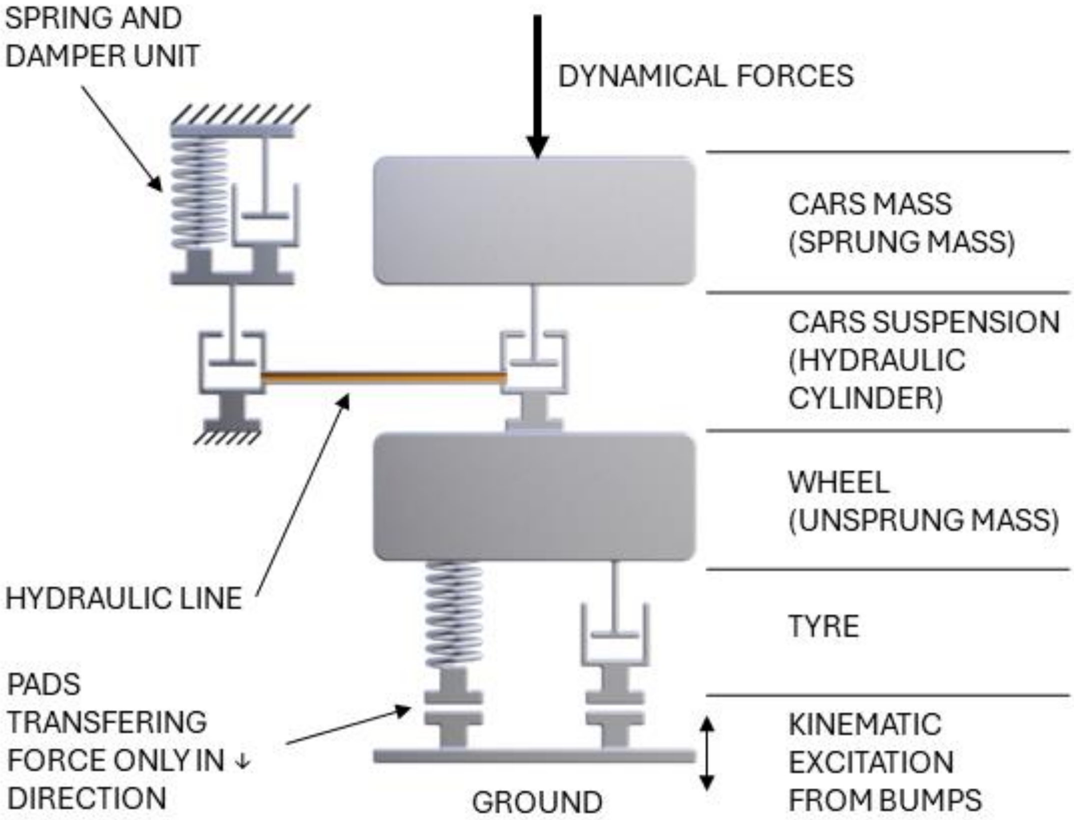


Pic. 3.11 Simulink model of the basic quarter car model

On pic. 3.11, Simulink model of the basic quarter car model from pic. 3.10 can be seen.

3.2.2. Hydraulic model of the suspension

After the base model was created, it needed to be modified to work with the hydraulic part of the suspension. A simplified depiction of hydraulics being implemented into a base model from pic. 3.10 can be seen on pic. 3.12. It can be seen that the conventional spring and damper from the base model was replaced by a hydraulic cylinder which is connected to another hydraulic cylinder. These two cylinders directly transfers forces via hydraulic line which is infinitely stiff and doesn't consider hydraulic losses, from the rest of the model to the spring and damper unit. This spring and damper unit is fixed to a car's chassis. Spring and damper unit like this, is present for every oscillation mode of the car (heave, pitch, roll) and its spring stiffness and damping coefficient will be tuned in the optimization.

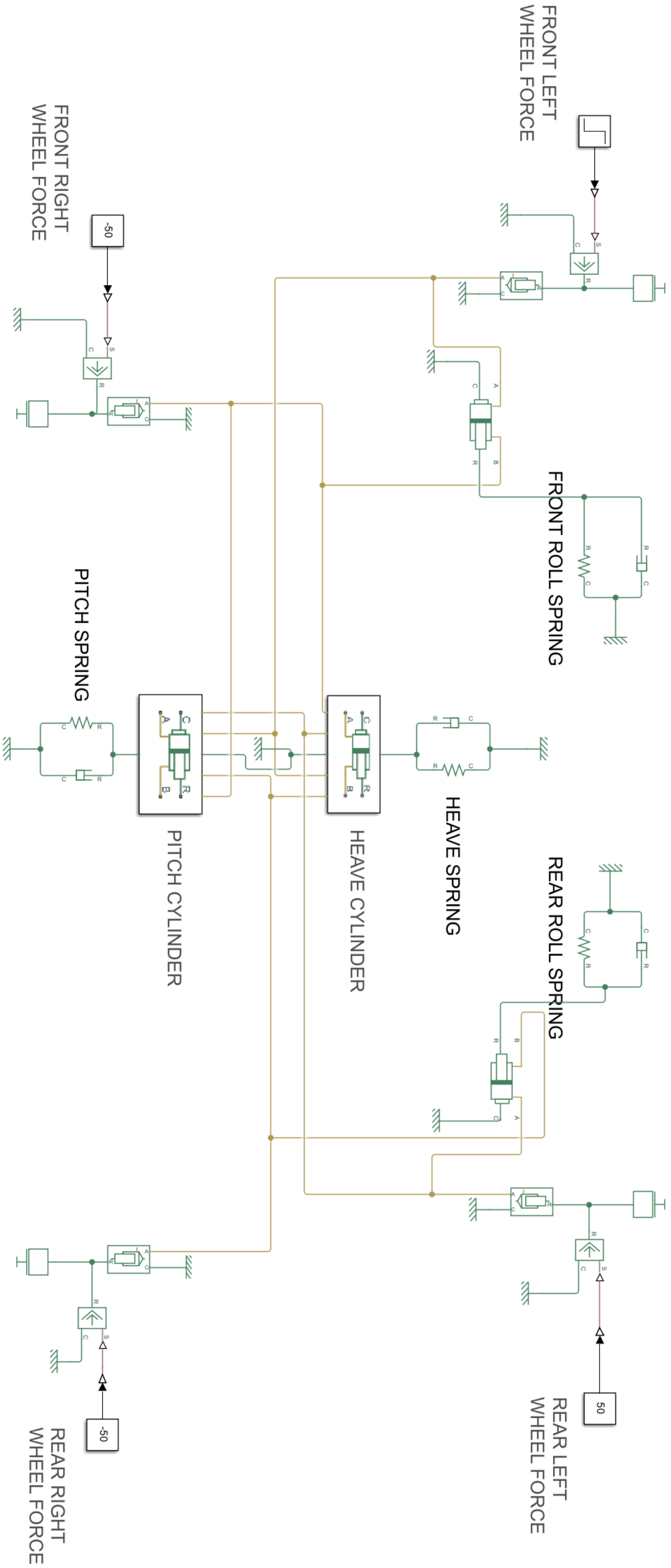


Pic. 3.12 Hydraulic quarter car model

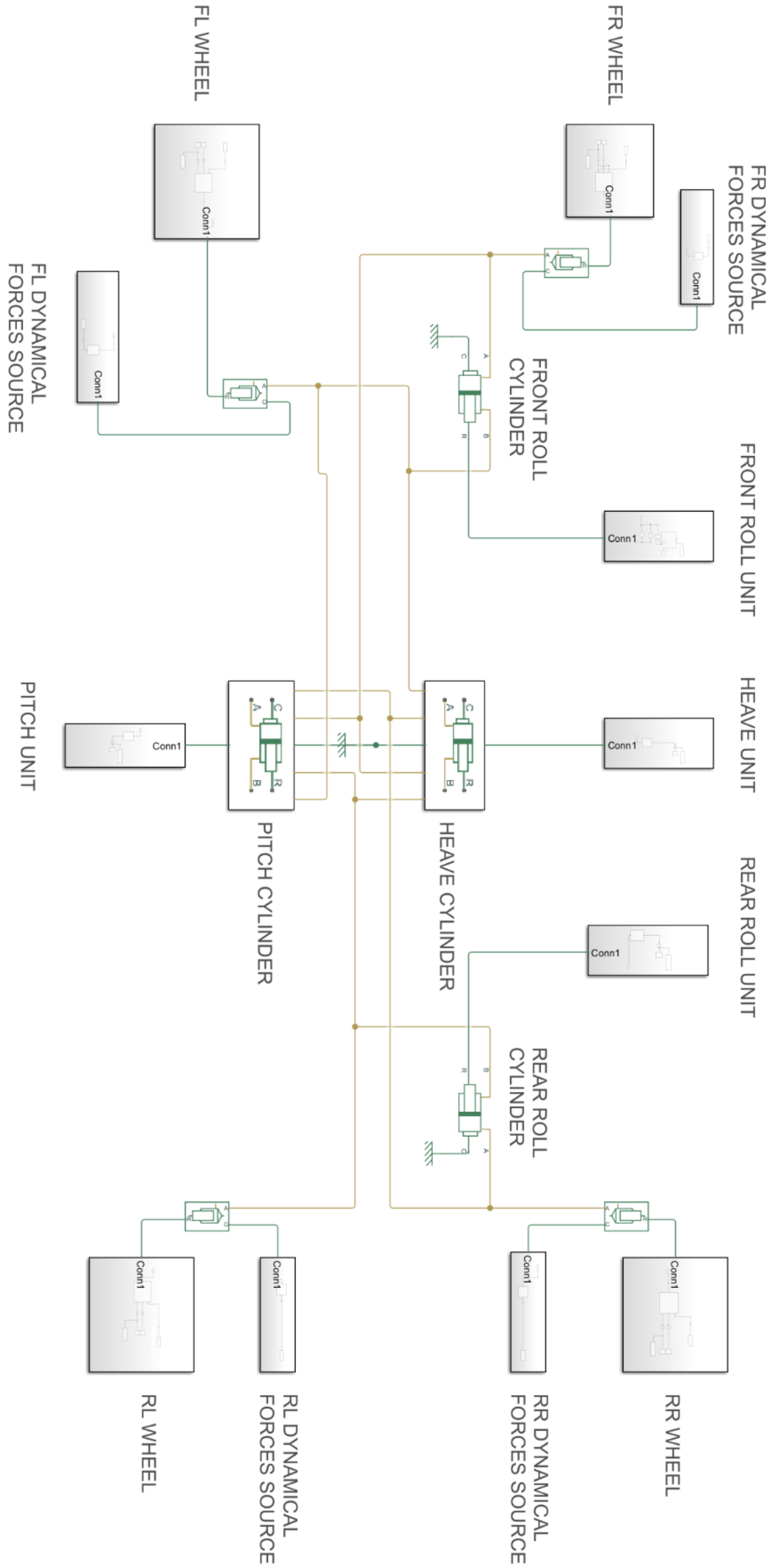
A full car hydraulic model built in Simulink can be seen on pic. 3.13. This model is just for testing the hydraulic part of the suspension, it doesn’t consider cars wheels, tyres and kinematic excitation from bumps.

3.2.3 Final model of the suspension

Merging of the basic quarter car model from pic. 3.11 and hydraulic model from pic. 3.13 creates a final model of the hydraulic decoupled suspension that will be used for the optimization and to determine results of this thesis. The Simulink model, ready for optimization can be seen on pic. 3.14. Output of this model are positions of spring and damper units for individual oscillation modes and forces acting on individual tyres, these data will be used as a input for optimization loss functions. This model is solved by Simulink built in solver ode45.

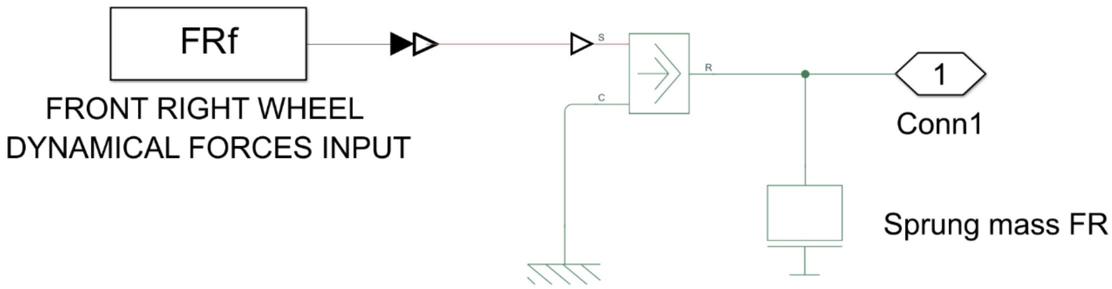


Pic. 3.13 Simulink model of the hydraulic part of the suspension



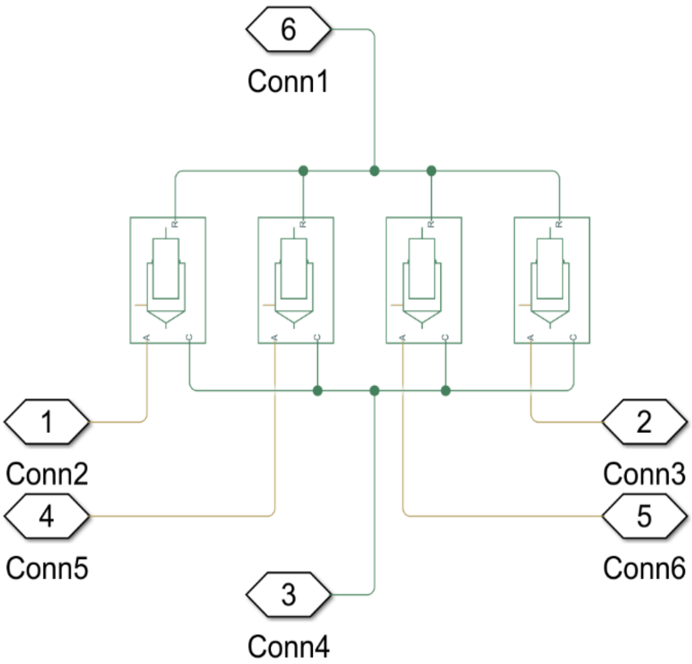
Pic. 3.14 Complete model of the suspension

On pic. 3.15 contents of the dynamical forces source for front right wheel can be seen. The role of this subsystem is to input dynamical forces through Conn 1 into the hydraulic piston of the front right wheel while adding an inertia element that is named Sprung mass FR on the pic. 3.15. This sprung mass element is a Simscape mechanical mass block, and its value is a quarter of the car's weight. Dynamical forces sources are modelled for all wheels analogically.

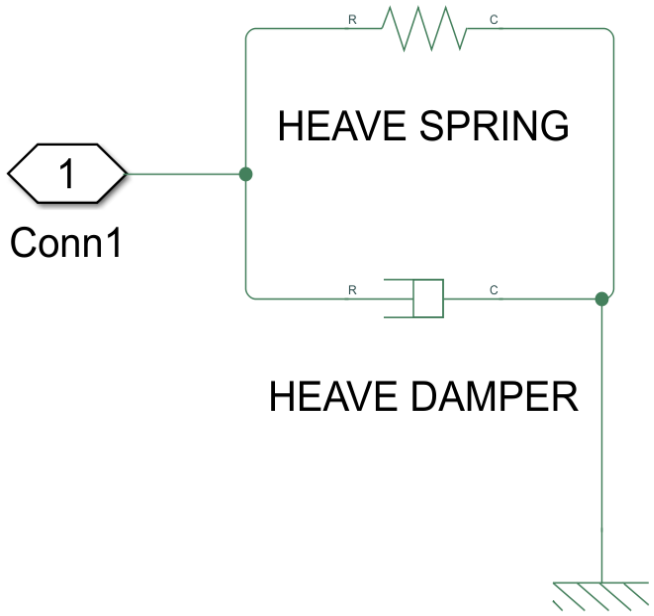


Pic. 3.15 Contents of FR DYNAMICAL FORCES SOURCE subsystem

On pic. 3.16 contents of heave cylinder subsystem can be seen. Unlike in hydraulic suspension design from pic. 2.6, there are four hydraulic cylinders instead of one, this is because Simscape doesn't allow connection of multiple hydraulic lines into a single cylinder, but the force transfer is the same no matter the number of cylinders. Output conn six goes into a heave unit subsystem. On pic. 3.17 spring and damper unit for heave oscillation mode can be seen. Pitch cylinder is modelled analogically to heave cylinder and front roll, rear roll and pitch units are modelled analogically to heave unit.

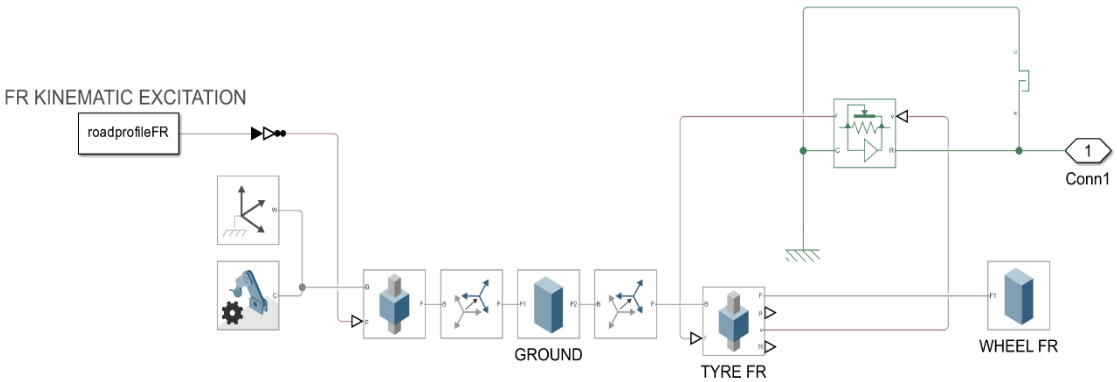


Pic. 3.16 Contents of HEAVE CYLINDER subsystem



Pic. 3.17 Contents of HEAVE UNIT subsystem

On pic. 3.18 contents of a subsystem that represents the front right wheel can be seen. In the upper left corner kinematic excitation from road surface unevenness that inputs into the simulations ground can be seen. Output Conn 1 of this subsystem goes into a hydraulic cylinder of the front right wheel. Every other wheel in this model is modelled analogically.



Pic. 3.18 Contents of FR WHEEL subsystem

3.3. Optimization

As mentioned in chapter 2.3, optimization is going to be done using Simulink response optimization tool. In this tool, variables that are going to be optimized can be chosen from Simulink model workspace, which in this case are spring stiffnesses and damping coefficients for individual oscillation modes of the car. These variables are going to be optimized by gradient descent algorithm to satisfy a given objective. In this case, the objective is to minimize output of a loss functions that are specifically designed for goals of this optimization. There are two types of loss functions designed for this optimization goals.

First one is for the goal of minimizing the car's body movement relative to the road surface. This function takes a resulting data of the individual spring and damper unit positions over time after each run of the simulation and calculates an absolute value of a discrete integral from this data. Resulting value of this integral is an output of the loss function, and its objective is to be minimized. This is performed for every oscillation mode separately.

The second type of loss function is for the goal of having the force that is acting on a car's tyres as constant (stable) as possible. This function takes resulting data of force acting on the tyre over time after each run of the simulation and calculates an absolute value of a discrete derivative for every time step of this data. Then it calculates a mean of these values. Value of this mean is an output of this loss function, and its objective is to be minimized. This is performed for every tyre of the car separately.

After test runs of the optimization it was revealed that the first loss function while minimizing the positions of springs and dampers, it creates a lot of oscillations. These oscillations while having low amplitude are not desired in the goals of this optimization. After this finding, the second type of loss function was also applied to the first goal of this optimization which is minimizing movements of the car's body relative to the road surface.

Another task in this optimization problem was to set upper and lower bound of spring stiffness and damping coefficient values. These bounds can be seen in table 1.

Table 1 Bounds of optimized values

Oscillation mode	Spring stiffness lower bound [N/m]	Spring stiffness upper bound [N/m]	Damping coefficient lower bound [N*s/m]	Damping coefficient upper bound [N*s/m]
Heave	250000	1000000	5000	50000
Pitch	100000	1000000	5000	50000
Roll	100000	10000000	1000	50000

These bounds were chosen by doing many runs of the simulation at different values and examining output data of the simulation by eye, seeing which values give useful results and which not. This was done to speed up the convergence of the optimization.

3.4. Results

Resulting values of spring stiffnesses and damping coefficients are listed in table 2.

Table 2 Results of the optimization

Oscillation mode	Spring stiffness [N/m]	Damping coefficient [N*s/m]
Heave	995680	17791
Pitch	235640	5872
Roll	1088100	27381

This optimization converged after six iterations of a gradient descent algorithm, ninety-four simulation runs and gave results that can be seen below. Plots of loss functions values over the iterations can be found in this thesis attachment.

Resulting data of spring and damper unit for individual oscillation modes can be seen on pic. 3.19, 3.20 and 3.21. Absolute displacements of these units are just in tens of millimetres at maximum, which is much less than what is expected in conventional suspensions. This means that the car's body is very stable relatively to the road surface, and therefore the aerodynamic devices should show more consistent performance.

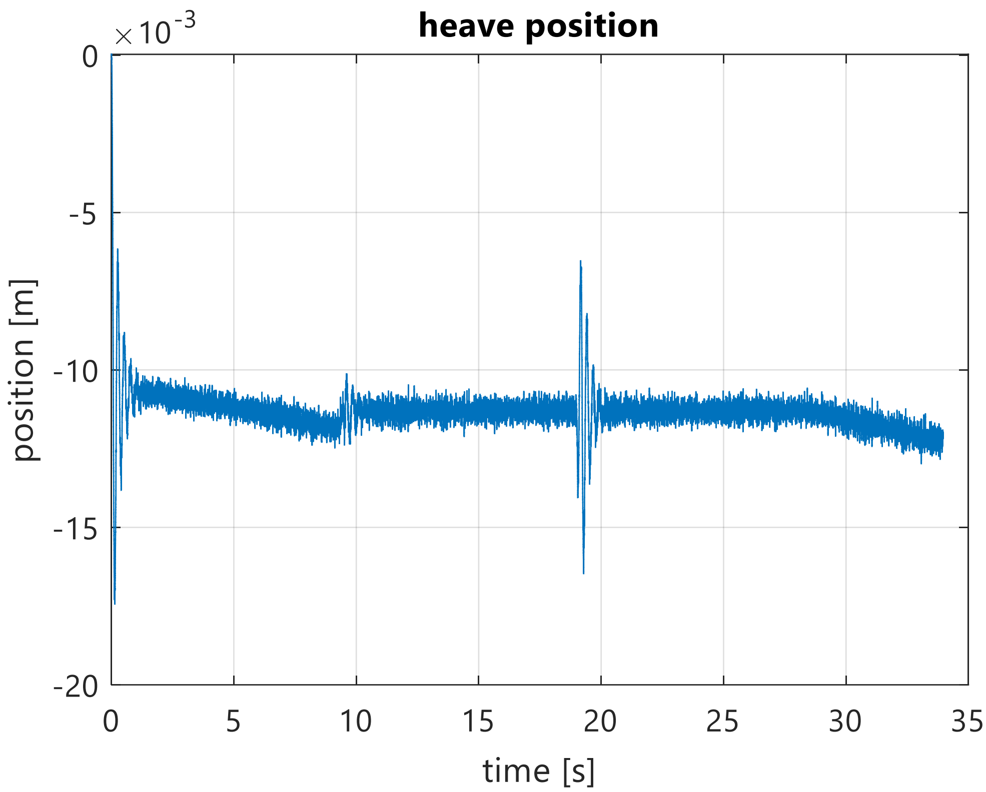
Noise that is present in these data is caused by bumps from road surface and its amplitudes are a bit greater than expected. This may be caused by the design of the model, it doesn't have any damping elements like real suspension would (compressible hydraulic fluid, wishbones, rocker arms, bolt connections). It's transferring the whole force directly from the wheel to the spring and damper.

On the pic. 3.19 an impulse can be seen at circa eighteenth second, this occurs at a time when the simulated car switches from turning left to turning right on the skidpad track. This transition in the car's roll direction, excited heave and created a short oscillation.

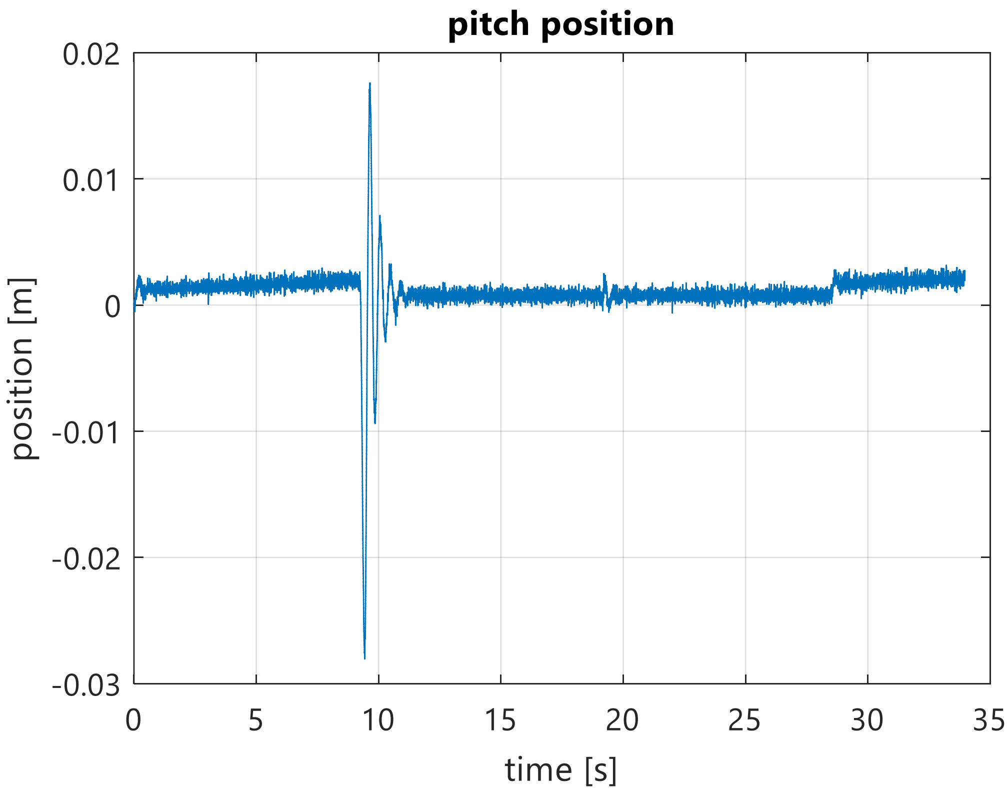
It also can be seen on the pic. 3.19 that in the first ten seconds of the simulation, heave is gaining value, and then it is stabilized and continues to gain value again after thirtieth second. This is because aerodynamic downforce is getting bigger with speed and in the first ten seconds of the simulation the car is accelerating, therefore gaining downforce which is loading the heave spring. After tenth second the car starts cornering at constant speed, so the heave load stays the same and then at thirtieth second the car starts to accelerate again.

On the pic. 3.20 a big spike around ninth second can be seen. This spike is caused by hard braking of the car before the first corner, which created a lot of load on the pitch unit. This spike has an amplitude bigger and oscillation decay time longer than what would be ideal for car's aerodynamics, but it will benefit the car's grip by absorbing extra load from tyres into the suspension. It can also be seen from the pic. 3.20 that like in the pic. 3.19, the pitch load grows at the beginning of the simulation, then stabilizes at a non-zero value and then grows again. This is caused by aerodynamics drag force pushing on car's front centre of pressure and by that loading the pitch unit, this occurs at constant speed too. That's why the pitch position is non-zero even while cornering.

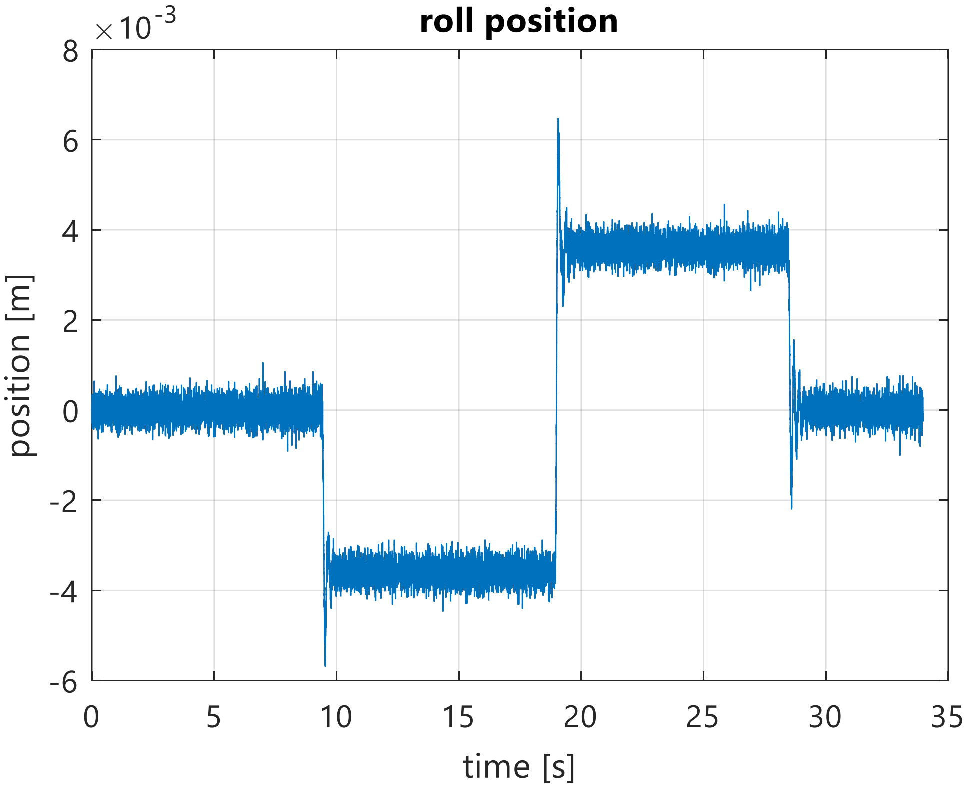
On the pic. 3.21 roll spring and damper position can be seen. It has relatively low amplitudes and short oscillation decay time, which is good for car's aerodynamic performance.



Pic. 3.19 Resulting heave position over time

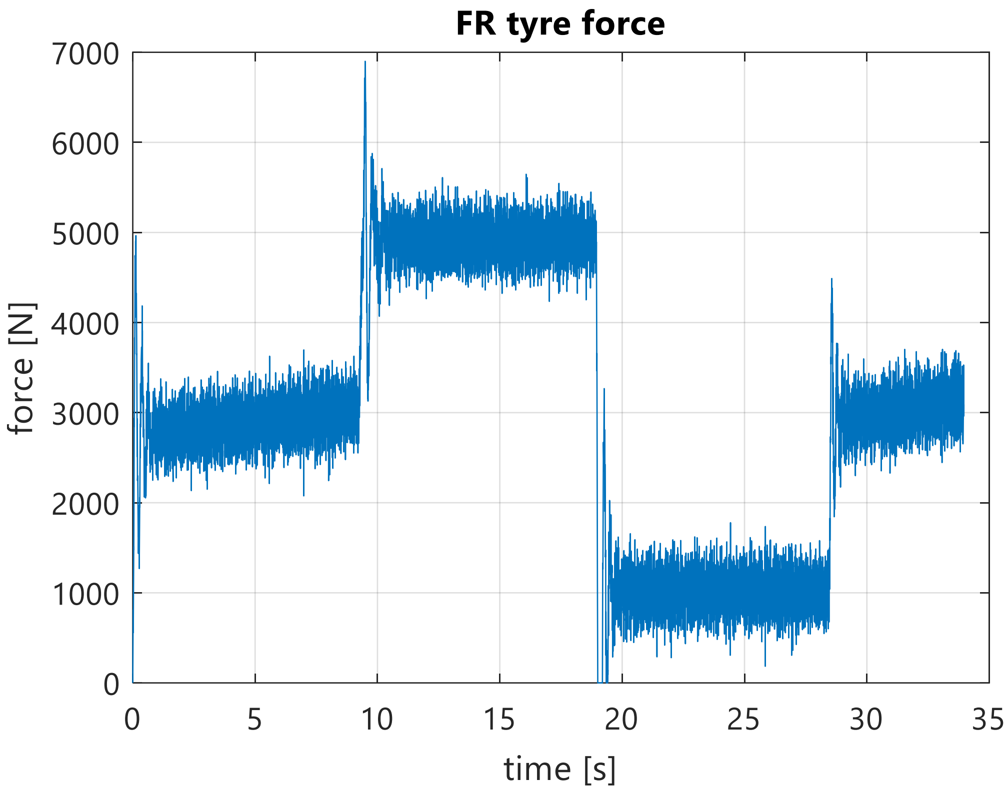


Pic. 3.20 Resulting pitch position over time

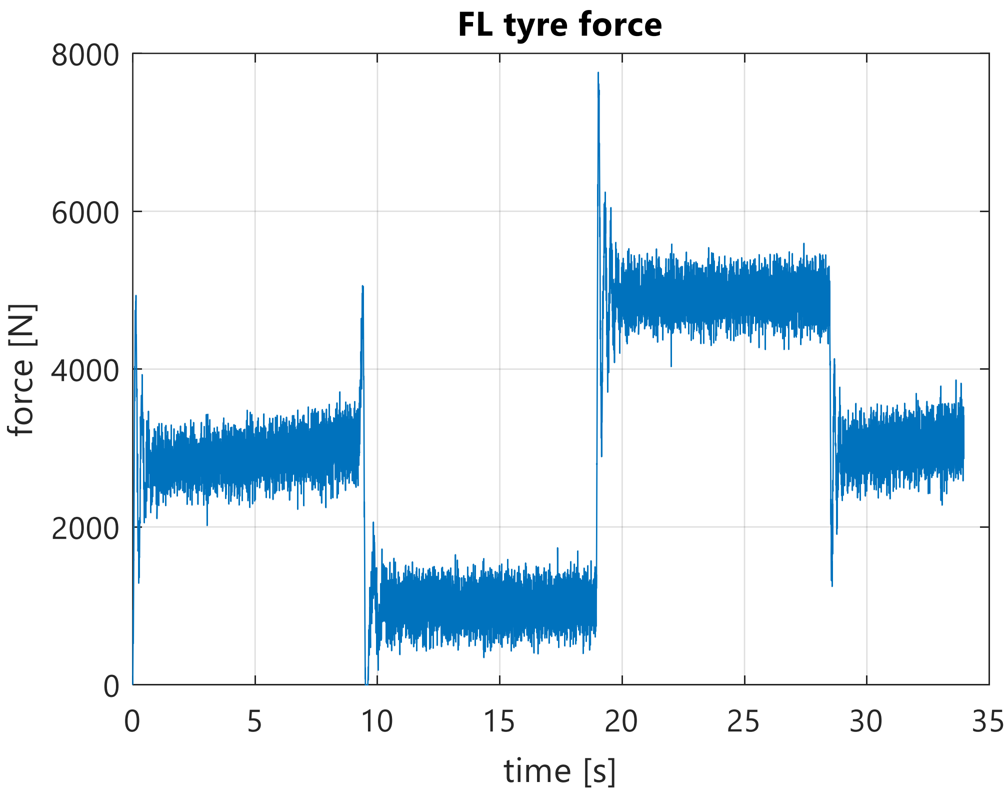


Pic. 3.21 Resulting roll position over time

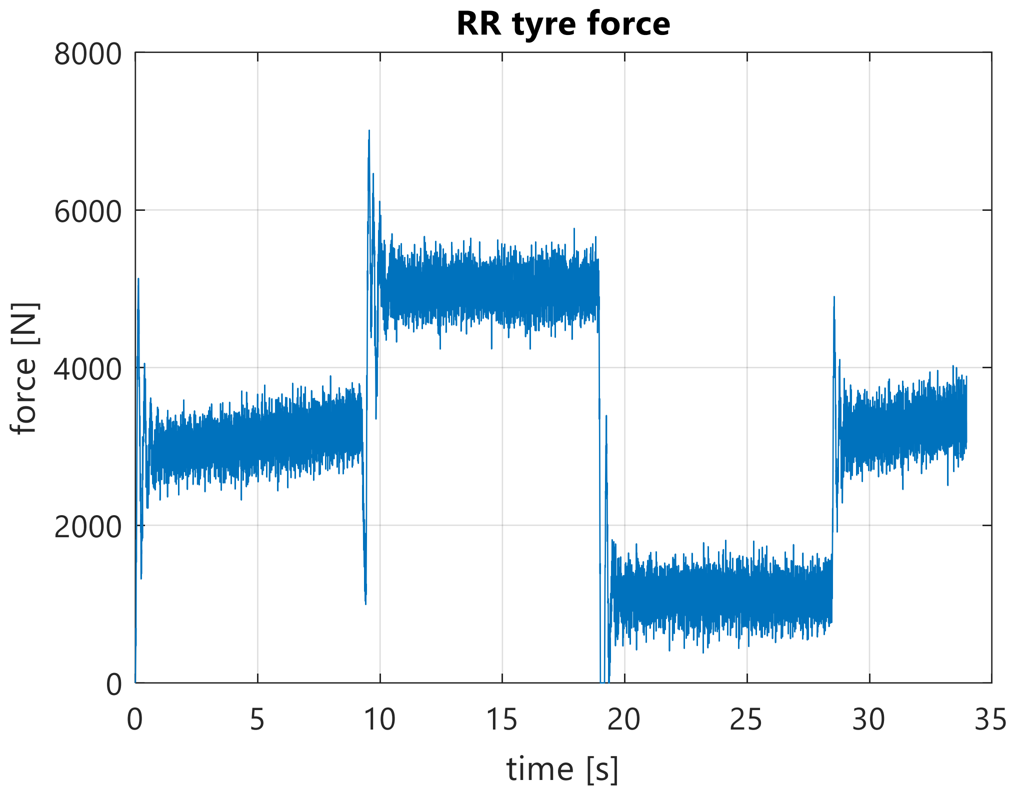
On pic. 3.22, 3.23, 3.24 and 3.25 forces acting on the individual tyres of the car can be seen. A relatively large amount of noise is present in these plots, which is caused by tyres absorbing most of the impacts from road surface unevenness. It can also be seen that these force data reach a value of zero in every plot, which means that the tyre was lifted off the road surface during the simulation. An event like this isn't ideal for the tyre's grip maximization, but it can occur in certain dynamical scenarios. These data also have relatively large spikes when the car switched from turning left to turning right. These spikes are caused by a rather stiff and heavily damped roll oscillation mode, which doesn't absorb much energy. Therefore, the tyres are being more loaded in this event. Overall, these tyre force data are not very pleasing and would need further optimization.



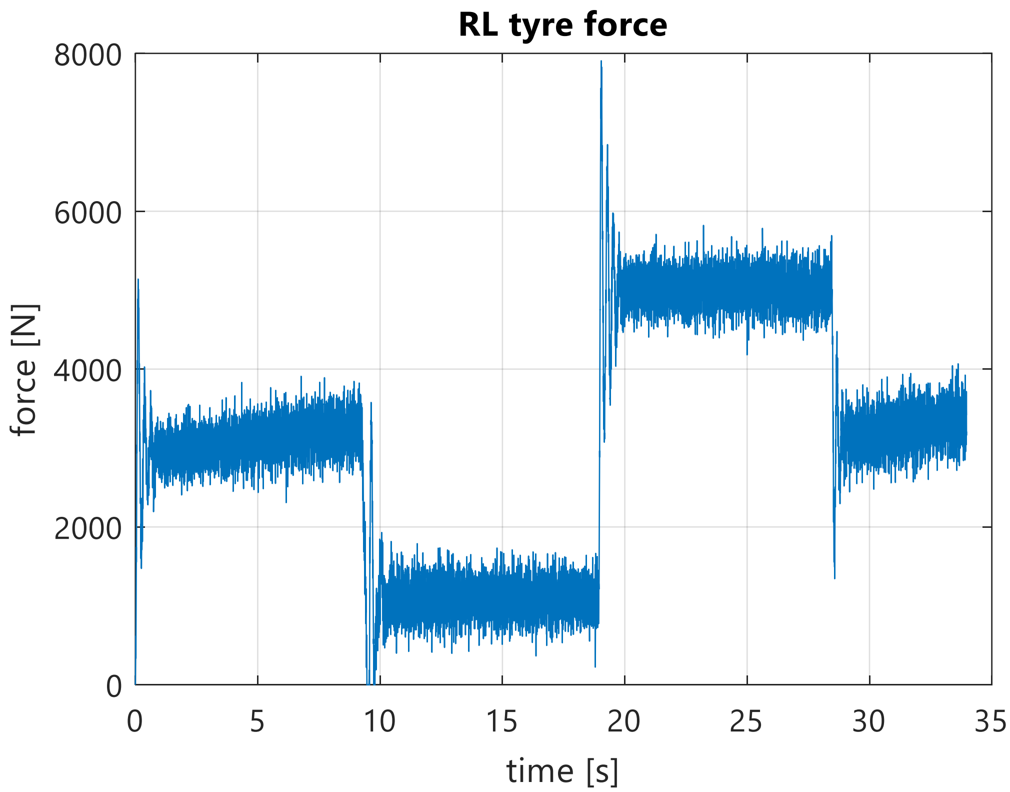
Pic. 3.22 Resulting force on front right tyre over time



Pic. 3.23 Resulting force on front left tyre over time



Pic. 3.24 Resulting force on rear right tyre over time



Pic. 3.25 Resulting force on rear left tyre over time

4. Discussion

When looking at the results, factors that were neglected in this simulation needs to be considered too, since they could have some effect on these results. For example, simulating the suspension as four one dimensional models instead of doing simulation of a three-dimensional model car. This approach substitutes rotational movements of a real suspension by translational, neglecting rotational inertia effects of a real car.

Considering a car that is fully symmetrical with homogenous mass distribution and infinitely stiff chassis. In real life, cars are rarely symmetrical, and they usually experience chassis flexing which add additional load and variables to the suspension setup.

Hydraulic fluid that transfers forces inside the suspension system is considered as incompressible and isothermal, neglecting parasitical damping that would occur in real hydraulic suspension.

Hydraulic lines being considered as dimensionless and infinitely stiff. In real hydraulic suspension, bends in the hydraulic lines and their flexing under load from hydraulic fluid pressure would create additional parasitic damping and force transfer losses.

Tyres are modelled as one dimensional linear spring and damper. Real tyres are loaded in three dimensions and have different nonlinear characteristics in all of these dimensions. Also, kinematic excitation from road surface unevenness is one dimensional in this simulation, while in real this surface unevenness is acting on a three-dimensional contact between tyre and the road surface.

Potential improvements of this simulation would mainly be to model and simulate a three-dimensional car driving through three-dimensional space with improved tyre models, more realistic suspension model and realistic car kinematics that considers car's rotational inertia. This more realistic model could be also improved with use of nonlinear spring and dampers for the suspension, which could bring even more options in the suspension optimization.

5. Conclusion

The goals of this thesis assignment were achieved successfully. A decoupled hydraulic suspension system was designed, modelled in Simulink, and tested for various dynamical scenarios to see if it works according to expectations. Input for this simulation was set to be dynamical forces created by the car's inertia and kinematic excitation acting on the car's tyres caused by road surface unevenness. Dynamical forces input was sourced from OptimumLap simulation software data, which were further processed in Matlab to create a useful simulation input. Kinematic excitation was sourced from an internet found Matlab script and inputted into the simulation. The output of the simulation was set to be positions of the individual oscillation modes and normal force acting on the car's tyres, these variables were used as inputs for individual loss functions for optimization. After validating that the Simulink model of the suspension works correctly, the optimization of its values of spring stiffnesses and damping coefficients was executed by using Simulink response optimization tool.

This optimization gave results that seem to mainly satisfy the optimization goal of having minimum movement of the car's body relative to the road surface, while the other goal of the force loading the tyres being as constant as possible seems to be less optimal. This could be caused by having more weight on the first optimization goal than the second one. So, it depends on the car and its desired dynamical behaviour, when choosing which of one of the two goals should have more weight in the suspension optimization. This thesis clearly showed, that even after creating a complex suspension system with goal of achieving uncompromised ideal dynamical behaviour of a car, compromises are still needed, creating room for new ideas and further optimization.

Bibliography

[1] ‘Citroën’s Hydropneumatic Suspension Has Officially Been Depressurized—Forever – News – Car and Driver’. Accessed: May 14, 2024. [Online]. Available: https://www.caranddriver.com/news/a15354448/citroens-hydropneumatic-suspension-is-being-depressurized-forever/

[2] ‘Citroën’s Amazing Hydropneumatic Suspension – suspensionspot’. Accessed: May 14, 2024. [Online]. Available: https://suspensionspot.com/blogs/news/citroens-amazing-hydropneumatic-suspension

[3] ‘TECH TUESDAY: Why the Williams FW15C remains F1’s technological Tour de Force’. Accessed: May 14, 2024. [Online]. Available: https://www.formula1.com/en/latest/article/tech-tuesday-wizardry-behind-williams-fw15c.6IzOXYuuPYOWyaaGGS4AgS

[4] ‘Hydraulic decoupled suspension – the journey starts with a bold idea » Student Lounge - MATLAB & Simulink’. Accessed: May 14, 2024. [Online]. Available: https://blogs.mathworks.com/student-lounge/2018/02/01/hydraulic-decoupled-suspension-part1/

[5] ‘Hydraulic decoupled suspension – how to get things running smoothly » Student Lounge - MATLAB & Simulink’. Accessed: May 16, 2024. [Online]. Available: https://blogs.mathworks.com/student-lounge/2018/02/07/hydraulic-decoupled-suspension-part2/?s_tid=blogs_rc_2

[6] ‘The Ultimate in Mechanical Grip - How Does Fully Mode Decoupled Suspension Work? - YouTube’. Accessed: May 14, 2024. [Online]. Available: https://www.youtube.com/watch?v=5eh5lZu2JqM&t=721s

[7] ‘VI-CarRealTime | VI-grade’. Accessed: May 16, 2024. [Online]. Available: https://www.vi-grade.com/en/products/vi-carrealtime/

[8] ‘OptimumLap | OptimumG’. Accessed: May 14, 2024. [Online]. Available: https://optimumg.com/product/optimumlap/

[9] P. Porteš, ‘DYNAMIKA VOZIDEL ZATÍŽENÍ NÁPRAV Ústav automobilního a dopravního inženýrství Fakulta strojního inženýrství Vysoké učení technické v Brně’.

[10] P. Porteš, ‘DYNAMIKA VOZIDEL OVLADATELNOST Ústav automobilního a dopravního inženýrství Fakulta strojního inženýrství Vysoké učení technické v Brně’.

[11] ‘Road profile, car model, suspension model - MATLAB Answers - MATLAB Central’. Accessed: May 14, 2024. [Online]. Available:

https://www.mathworks.com/matlabcentral/answers/1668539-road-profile-car-model-suspension-model

[12] P. Porteš, 'DYNAMIKA VOZIDEL JÍZDA PO NEROVNÉ VOZOVCE Ústav automobilního a dopravního inženýrství Fakulta strojního inženýrství Vysoké učení technické v Brně'.

[13] J. P. C. Gonçalves and J. A. C. Ambrósio, 'Road vehicle modeling requirements for optimization of ride and handling', in *Multibody System Dynamics*, Feb. 2005, pp. 3–23. doi: 10.1007/s11044-005-2528-5.

[14] 'Response Optimization - MATLAB & Simulink'. Accessed: May 14, 2024. [Online]. Available: https://www.mathworks.com/help/sldo/response-optimization.html

[15] 'Oreca 07 - Wikipedia'. Accessed: May 14, 2024. [Online]. Available: https://en.wikipedia.org/wiki/Oreca_07

[16] 'Rolling resistance of racing tyre - F1technical.net'. Accessed: May 14, 2024. [Online]. Available: https://www.f1technical.net/forum/viewtopic.php?t=3960

[17] 'Tires - Vehicle Physics Pro'. Accessed: May 14, 2024. [Online]. Available: https://vehiclephysics.com/blocks/tires/

[18] 'Tire vertical damping - F1technical.net'. Accessed: May 14, 2024. [Online]. Available: https://www.f1technical.net/forum/viewtopic.php?p=468756#p468756

[19] 'WILLIAMSFW15'. Accessed: May 14, 2024. [Online]. Available: http://www.gurneyflap.com/williamsfw15.html

[20] Z. Wang, M. Dong, Y. Qin, Y. Du, F. Zhao, and L. Gu, 'Suspension system state estimation using adaptive Kalman filtering based on road classification', *Vehicle System Dynamics*, vol. 55, no. 3, pp. 371–398, Mar. 2017, doi: 10.1080/00423114.2016.1267374.

List of attachments

1. 2024_BP_Zrncik_Matej_228760_Attachment – contains Simulink model of the suspension, running script of the suspension model, input data, iteration data figures, optimization convergence report