

Group Project

Zentrack Chassis Group

Final Report

M5X00400

31/03/2019

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Introduction

Zentrack is a privately-owned independent manufacturer of open top road going and race ready sports cars intending to go into production in May of 2019. It will offer a single vehicle, with multiple chassis variants for different situations, and a variety of optional extras, including different powertrain options.

The task given to us by the owner of Zentrack, John Avery, was to evaluate and improve upon the vehicle's performance capability, concentrating specifically on the chassis and suspension components. The scope of project was limited to a few specific suspension components and the central, passenger cell, section of the chassis. Improvements were required to be road legal modifications that would keep the car FIA and Motorsport UK (MSA) compliant. John asked us to make these improvements with the goal of making this car a competitor for both the Lotus Elise, and in VLN endurance races, whilst keeping to his conservative pricing goals.

Organization

The organisation of the team was primarily split up according to the sections of the car that we would be working on, with a few people working across multiple components/areas. Sam, Suriya and Harry took on the task of stress testing and improving the design of the chassis, uprights and wishbones. Oly and Zhou looked at the suspension and steering components. Guo looked at the brakes. Steve

measured the rotational inertia of various components. Leo created a simulation model. Sun evaluated the car and tires.

For everyone apart from Steve, Leo and Sun, the project was split into three primary tasks. The first was to evaluate the current performance of the individual components. The next task was to propose improvements to the designs or alternative part choices to be used on the car in future. The final task was to evaluate the proposed changes.

Leo, Steve and Sun's jobs were split into two main tasks. The first was to evaluate the overall performance of the car, both from a lap times and a drivability point of view. The second task was to re-evaluate the overall performance using the improved designs from the rest of the team.

Whilst group tasks and meetings were organised as a team, team meetings were chaired and minuted by Leo.

The planned time frame for the completion of individual tasks can be seen from the Gantt Chart in figure 1.

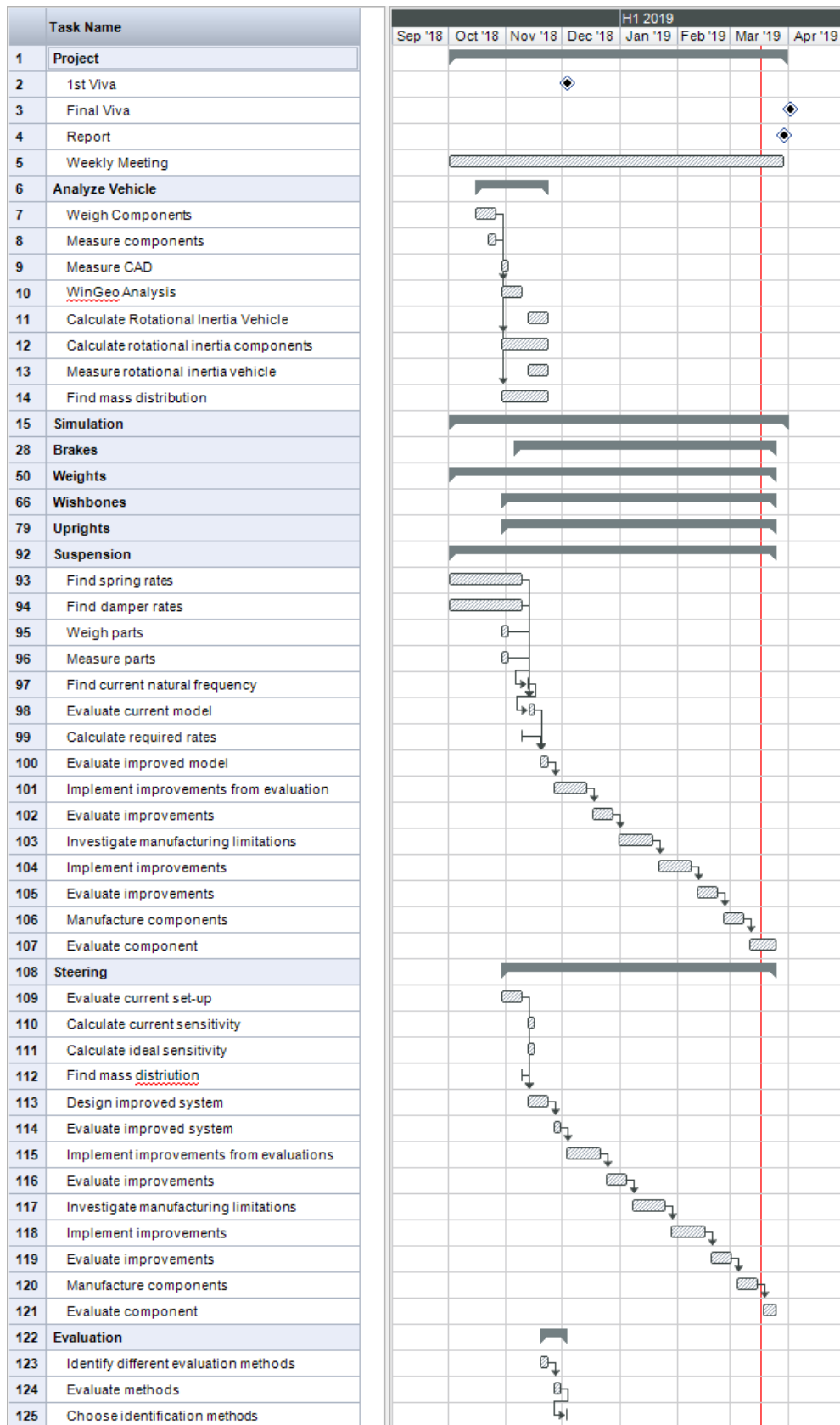


Figure 1: Group Gantt Chart

Quan Zhou

Work content

Task	Description	Start date	End date
Carry out suspension research	Find the suspension features and find appropriate articles and papers to help me optimize the suspension	12/10/18	12/11/18
Carry out suspension calculation	After finding out the theoretical foundation, carry out the calculation before modeling	12/11/18	12/12/18
Build and optimize the model	After calculation, model based on the calculating result	12/12/18	01/02/19
Carry out suspension experiment	Carry out suspension experiment to find all required parameters	01/02/19	15/02/19
Carry out suspension design and optimize	Based on the suspension design to decide how to optimize the suspension	15/02/19	01/03/19
Carry out suspension simulation and testing	Based on the optimization method to simulate the vibration result	01/03/19	10/03/19
Carry out suspension analyzation(conclusion)	Based on simulation result to analyze	10/03/19	01/04/19
Carry out steering test	Carry out steering test to find out the performance of steering system	12/10/18	01/02/19
Carry out steering calculation	Carry out steering calculation to evaluate the performance	01/02/19	01/03/18
Carry out steering conclusion	Based on calculation to give the suggestion to optimize the steering system	01/03/18	01/04/19

Carry out suspension research

My first task is to optimize the suspension. In order to reach the target, I need to search for useful papers and articles may do some help to my work. So I spend nearly one month to search for the literature. The useful literature review is as followed:

The rough road surface is the root cause of the vibration of the vehicle, so it is necessary to establish a mathematical model to describe road roughness. Xie used computer simulation of random road surface excitation as input, and established the road surface roughness model by MATLAB/Simulink simulation software for computer simulation.[1] Liu built a Simulink simulation model for Class B Road based on pseudo white noise method.[2] Zheng used the probability and statistics theory and the central limit theorem to establish a mathematical model of the uneven road surface spectrum based on the power spectral density of the principle of harmonic superposition.[3] Zhang's research summarizes the triangular series method and the Fourier inverse transformation method for road roughness.[4]

How to evaluate the vibration is an important index of evaluating the performance of suspension. In order to accurately evaluate the ride comfort of the vehicle, Zhang uses the RMS value of the vertical weighted acceleration of the passenger to evaluate the index of the flatness. [4] Zhang summarized two internationally accepted evaluation criteria for ride

comfort and three quantitative calculation methods for passenger comfort.[5]

The core is to establish a reasonable mathematical model for calculating the real-time vibration acceleration of the car body. Zhang constructed a five-degree-freedom vehicle model of human-vehicle-road, and analyzed the interaction between human and vehicle roads by transfer matrix method.[4] Xie performed mechanical analysis to obtain the system vibration differential equation and built the four-degree-freedom vehicle model by Simulink.[1] Liu, through the dynamic analysis of vehicles, established the eight-degree-freedom vehicle model, calculated the eight-degree- freedom coupled vibration equation of the vehicle four-point random excitation, and used the block matrix method to derive the transfer function of the vehicle vibration system.[6] Zhang's research summarizes the two-degree-freedom, the four-degree-freedom, five-degree-freedom, the seven-degree-freedom, seventeen-degree-freedom vehicle model for vibration analysis of each part.[5]

Carry out suspension calculation

After researching for useful information, I choose to establish four-degree-freedom vehicle model to calculate. Considering our chair is fixed on the chassis, I think four-degree-freedom model is easier to calculate than more degrees freedom models.

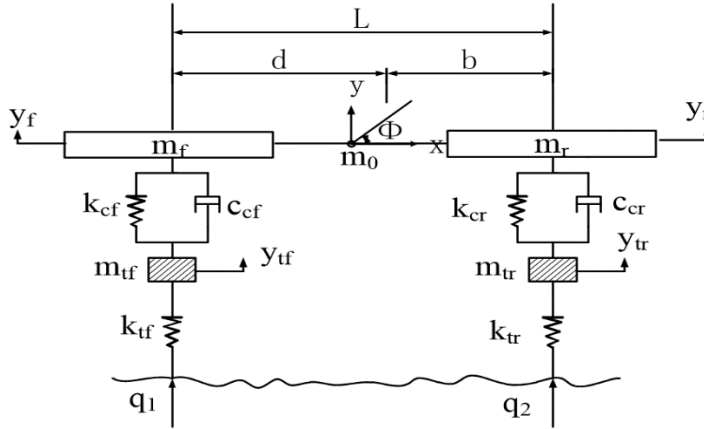


Figure 1. four-degree-freedom vehicle model

Taking the vehicle body as the research object, the front and rear moments are balanced:

$$m_f \ddot{y}_f + m_0 \ddot{y}_c \frac{b}{L} + k_{cf}(y_f - y_{tf}) + c_{cf}(\dot{y}_f - \dot{y}_{tf}) = 0 \quad (1)$$

$$m_r \ddot{y}_r + m_0 \ddot{y}_c \frac{d}{L} + k_{cr}(y_r - y_{tr}) + c_{cr}(\dot{y}_r - \dot{y}_{tr}) = 0 \quad (2)$$

Where: $y_f, y_r, z_c, y_{tf}, y_{tr}$ represent the vertical vibration displacement of the concentrated mass on the front and rear axles, the center of mass of the vehicle, and the non-suspended distribution mass of the front and rear axles; k_{cf}, k_{cr} are the stiffness of the front and rear axle suspensions, respectively. ; c_{cf}, c_{cr} are the damping coefficients of the front and rear

suspension dampers; L, d, b are the distance between the wheelbase and the center of mass to the front and rear axles.

Take the front unsprung mass and the rear unsprung mass as the research objects:

$$m_{tf}\ddot{y}_{tf} + k_{cf}(y_{tf} - y_f) + c_{cf}(\dot{y}_{tf} - \dot{y}_f) + k_{tf}(y_{tf} - q_1) = 0 \quad (3)$$

$$m_{tr}\ddot{y}_{tr} + k_{cr}(y_{tr} - y_r) + c_{cr}(\dot{y}_{tr} - \dot{y}_r) + k_{tr}(y_{tr} - q_2) = 0. \quad (4)$$

Where: q_1, q_2 are the front and rear wheel road surface unevenness excitation.

Taking the vehicle body as the research object, the force is balanced in the vertical direction, and the centroid moment is balanced:

$$m_0\ddot{y}_c + k_{cf}(y_f - y_{tf}) + c_{cf}(\dot{y}_f - \dot{y}_{tf}) + k_{cr}(y_r - y_{tr}) + c_{cr}(\dot{y}_r - \dot{y}_{tr}) = 0 \quad (5)$$

$$m_0\rho_y^2\ddot{\phi} + bk_{cr}(y_r - y_{tr}) + bc_{cr}(\dot{y}_r - \dot{y}_{tr}) - dk_{cf}(y_f - y_{tf}) - c_{cf}(\dot{y}_f - \dot{y}_{tf}) = 0 \quad (6)$$

Where: ϕ is the longitudinal angular vibration angular displacement of the vehicle body (counterclockwise is positive).

Establish a state equation and an output equation, where the state variable vector is selected as:

$$\begin{aligned} X &= [x_1 \ x_2 \ x_3 \ x_4 \ x_5 \ x_6 \ x_7 \ x_8]^T \\ &= [y_{tf} \ y_{tr} \ y_c \ \phi \ \dot{y}_{tf} \ \dot{y}_{tr} \ \dot{y}_c \ \dot{\phi}]^T \\ \dot{X} &= [\dot{x}_1 \ \dot{x}_2 \ \dot{x}_3 \ \dot{x}_4 \ \dot{x}_5 \ \dot{x}_6 \ \dot{x}_7 \ \dot{x}_8]^T \\ &= [\dot{y}_{tf} \ \dot{y}_{tr} \ \dot{y}_c \ \dot{\phi} \ \ddot{y}_{tf} \ \ddot{y}_{tr} \ \ddot{y}_c \ \ddot{\phi}]^T \end{aligned}$$

Input vector: $u=[q_1 \ q_2]^T$

The output vector is:

$$\begin{aligned} Y &= [y_1 \ y_2 \ y_3 \ y_4 \ y_5 \ y_6 \ y_7 \ y_8]^T \\ &= [\ddot{y}_f \ \ddot{y}_r \ y_f - y_{tf} \ y_r - y_{tr} \ k_{tf}(y_{tf} - q_1) \ k_{tr}(y_{tr} - q_2) \ (\ddot{y}_r d + \ddot{y}_f b)/L \ (\ddot{y}_r - \ddot{y}_f)/L]^T \end{aligned}$$

Establish the following equations of state equation and output equations:

$$\begin{cases} \dot{X} = AX + Bu \\ Y = CX + Du \end{cases}$$

Where A,B,C,D are:

$$B = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ \frac{k_{tf}}{m_{tf}} & 0 \\ 0 & \frac{k_{tr}}{m_{tr}} \\ 0 & 0 \\ 0 & 0 \end{bmatrix} \quad D = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ -k_{tf} & 0 \\ 0 & -k_{tr} \\ 0 & 0 \\ 0 & 0 \end{bmatrix}$$

$$A = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ a_{5,1} & 0 & a_{5,3} & a_{5,4} & a_{5,5} & 0 & a_{5,7} & a_{5,8} \\ 0 & a_{6,2} & a_{6,3} & a_{6,4} & 0 & a_{6,6} & a_{6,7} & a_{6,8} \\ a_{7,1} & a_{7,2} & a_{7,3} & a_{7,4} & a_{7,5} & a_{7,6} & a_{7,7} & a_{7,8} \\ a_{8,1} & a_{8,2} & a_{8,3} & a_{8,4} & a_{8,5} & a_{8,6} & a_{8,7} & a_{8,8} \end{bmatrix}$$

Where:

$$\begin{aligned} a_{5,1} &= -\frac{k_{tf}+k_{cf}}{m_{tf}}, a_{5,3} = \frac{k_{cf}}{m_{tf}}, a_{5,4} = -\frac{dk_{cf}}{m_{tf}}, a_{5,5} = -\frac{c_{cf}}{m_{tf}}, a_{5,7} = \frac{c_{cf}}{m_{tf}}, a_{5,8} = -\frac{dc_{cf}}{m_{tf}}, \\ a_{6,2} &= -\frac{k_{tr}+k_{cr}}{m_{tr}}, a_{6,3} = \frac{k_{cr}}{m_{tr}}, a_{6,4} = \frac{bk_{cr}}{m_{tr}}, a_{6,6} = -\frac{c_{cr}}{m_{tr}}, a_{6,7} = \frac{c_{cr}}{m_{tr}}, a_{6,8} = \frac{bc_{cr}}{m_{tr}}, a_{7,1} = \frac{k_{cf}}{m_0}, \\ a_{7,2} &= \frac{k_{cr}}{m_0}, a_{7,3} = \frac{k_{cf}+k_{cr}}{m_0}, a_{7,4} = \frac{dk_{cf}-bk_{cr}}{m_0}, a_{7,5} = \frac{c_{cf}}{m_0}, a_{7,6} = \frac{c_{cr}}{m_0}, a_{7,7} = -\frac{c_{cf}+c_{cr}}{m_0}, a_{7,8} = \frac{dc_{cf}-bc_{cr}}{m_0}, \\ a_{8,1} &= -\frac{dk_{cf}}{I}, a_{8,2} = -\frac{bk_{cr}}{I}, a_{8,3} = \frac{dk_{cf}-bk_{cr}}{I}, a_{8,4} = -\frac{d^2k_{cf}+b^2k_{cr}}{I}, a_{8,5} = -\frac{dc_{cf}}{I}, \\ a_{8,6} &= \frac{bc_{cr}}{I}, a_{8,7} = \frac{dk_{cf}-bk_{cr}}{I}, a_{8,8} = -\frac{d^2c_{cf}+b^2c_{cr}}{I} \end{aligned}$$

$$C = \begin{bmatrix} c_{1,1} & c_{1,2} & c_{1,3} & c_{1,4} & c_{1,5} & c_{1,6} & c_{1,7} & c_{1,8} \\ c_{2,1} & c_{2,2} & c_{2,3} & c_{2,4} & c_{2,5} & c_{2,6} & c_{2,7} & c_{2,8} \\ -1 & 0 & 1 & -d & 0 & 0 & 0 & 0 \\ 0 & -1 & 1 & b & 0 & 0 & 0 & 0 \\ k_{tf} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & k_{tr} & 0 & 0 & 0 & 0 & 0 & 0 \\ c_{7,1} & c_{7,2} & c_{7,3} & c_{7,4} & c_{7,5} & c_{7,6} & c_{7,7} & c_{7,8} \\ c_{8,1} & c_{8,2} & c_{8,3} & c_{8,4} & c_{8,5} & c_{8,6} & c_{8,7} & c_{8,8} \end{bmatrix}$$

Where

$$\begin{aligned} c_{1,1} &= \frac{k_{cf}}{m_0} + \frac{d^2k_{cf}}{I}, c_{1,2} = \frac{k_{cr}}{m_0} - \frac{dbk_{cr}}{I}, c_{1,3} = -\frac{k_{cf}+k_{cr}}{m_0} - \frac{d^2k_{cf}+db^2k_{cr}}{I}, \\ c_{1,4} &= \frac{dk_{cf}-bk_{cr}}{m_0} - \frac{d^3k_{cf}+db^2k_{cr}}{I}, c_{1,5} = \frac{c_{cf}}{m_0} + \frac{d^2c_{cf}}{I}, c_{1,6} = \frac{c_{cr}}{m_0} - \frac{dbc_{cr}}{I}, \\ c_{1,7} &= -\frac{c_{cf}+c_{cr}}{m_0} - \frac{d^2c_{cf}-dbc_{cr}}{I}, c_{1,8} = \frac{dc_{cf}-bc_{cr}}{m_0} + \frac{d^3c_{cf}+db^2c_{cr}}{I}, c_{2,1} = \frac{k_{cf}}{m_0} - \frac{dbk_{cf}}{I}, \\ c_{2,2} &= \frac{k_{cr}}{m_0} + \frac{b^2k_{cr}}{I}, c_{2,3} = \frac{k_{cf}+k_{cr}}{m_0} + \frac{dbk_{cf}-b^2k_{cr}}{I}, c_{2,4} = \frac{dk_{cf}-bk_{cr}}{m_0} - \frac{d^2bk_{cf}+b^3k_{cr}}{I}, \end{aligned}$$

$$\begin{aligned}
c_{2,5} &= \frac{c_{cf}}{m_0} - \frac{dbc_{cf}}{l}, c_{2,6} = \frac{c_{cr}}{m_0} + \frac{b^2c_{cr}}{l}, c_{2,7} = -\frac{c_{cf}+c_{cr}}{m_0} + \frac{dbc_{cf}-b^2c_{cr}}{l}, \\
c_{2,8} &= \frac{dc_{cf}-bc_{cr}}{m_0} + \frac{d^2bc_{cf}+b^3c_{cr}}{l}, c_{7,1} = \frac{k_{cf}}{m_0}, c_{7,2} = \frac{k_{cr}}{m_0}, c_{7,3} = \frac{k_{cf}+k_{cr}}{m_0}, \\
c_{7,4} &= \frac{dk_{cf}-bk_{cr}}{m_0}, c_{7,5} = \frac{c_{cf}}{m_0}, c_{7,6} = \frac{c_{cr}}{m_0}, c_{7,7} = \frac{c_{cf}+c_{cr}}{m_0}, c_{7,8} = \frac{dc_{cf}-bc_{cr}}{m_0}, c_{8,1} = -\frac{dk_{cf}}{l}, \\
c_{8,2} &= \frac{bk_{cr}}{l}, c_{8,3} = \frac{dk_{cf}-bk_{cr}}{l}, c_{8,4} = -\frac{d^2k_{cf}+b^2k_{cr}}{l}, c_{8,5} = -\frac{dc_{cf}}{l}, \\
c_{8,6} &= \frac{bc_{cr}}{l}, c_{8,7} = \frac{dk_{cf}-bk_{cr}}{l}, c_{8,8} = -\frac{d^2c_{cf}+b^2c_{cr}}{l}
\end{aligned}$$

Introduce A,B,C,D to equation (7), I can work out every element of X.

$$\begin{aligned}
X &= [x_1 \ x_2 \ x_3 \ x_4 \ x_5 \ x_6 \ x_7 \ x_8]^T \\
&= [y_{tf} \ y_{tr} \ y_c \ \phi \ \dot{y}_{tf} \ \dot{y}_{tr} \ \dot{y}_c \ \dot{\phi}]^T
\end{aligned}$$

After we get every element of X, we can get the value of velocity of the gravity center of vehicle body y_c . Take the derivative of the velocity, we can get the acceleration of gravity center $\dot{y}_c$ which is exact $a_w(t)$.

Build and optimize the model

1 Road roughness model

The input of the five-degree-freedom vehicle model is mainly the road roughness excitation $q(t)$ and parameters of the vehicle, so establishing a reasonable road model helps to obtain accurate simulation results. Through a large number of literatures[2], the white noise method has the advantages of small calculation, fast running speed and high calculation accuracy. Therefore, this paper chooses the white noise method to simulate the road model.

Road roughness refers to the deviation of the road surface from the ideal plane. It affects the power, comfort and safety of the vehicle. The power spectral density of the road is often used to describe the statistical characteristics of road roughness. The international standard ISO8608[7] and the national standard GB7031-86[3] suggest that the fitting expression of the road power spectral density $G_q(n)$ is:

$$G_q(n) = G_q(n_0) \left(\frac{n}{n_0} \right)^{-W} \quad (8)$$

Where: $G_q(n_0)$ is the road surface roughness coefficient, which is the value of the road surface power spectral density at the reference spatial frequency n_0 , in $m^2 \cdot m = m^3$; n represents the spatial frequency of several wavelengths per m length m^{-1} ; n_0 is the reference spatial frequency.

According to the road power spectrum density, the road roughness is divided into 8 levels, as shown in the following table, and the geometry mean of the corresponding root mean square value $q_{rms}(\sigma_q)$ of the road roughness in the range of $0.011m^{-1} < n < 2.83m^{-1}$ is listed.

Under normal circumstances, the numerical range of road roughness is: wavelength $\lambda = 0.1 \sim 100\text{m}$, amplitude $A = 1 \sim 200\text{mm}$ road roughness classification as shown in the table 1.

Table 1. 8-level-classification of Road Roughness.

Road Level	Roughness Level			
	$n_0 = 0.1\text{m}^{-1}, 0.011\text{m}^{-1} < n < 2.83\text{m}^{-1}$			
	$G_q(n_0)/10^{-6}\text{m}^3$			
	Lower Limit	Geometric Mean	Upper Limit	Geometric Mean
A	8	16	32	3.81
B	32	64	128	7.61
C	128	256	512	15.23
D	512	1024	2048	30.45
E	2048	4096	8192	60.9
F	8192	16384	32768	121.8
G	32768	65536	131072	243.61
H	131072	262144	524288	487.22

The method proposed in this report takes B-level road as an example. For reference to the literature [4], using the pseudo white noise method, the mathematical model of a single wheel subjected to road roughness excitation in the time domain can be described by (9) :

$$\dot{q}(t) = 2\pi f_0 \sqrt{G_q(n_0)} W(t) - 2\pi f_0 q(t) \quad (9)$$

In equation (9), $q(t)$ is the road roughness excitation; $G_q(n_0)$ is the road surface roughness coefficient; $W(t)$ is the random simulated white noise; f_0 is the cut-off frequency.

Because the probability of a passenger car driving on a highway or a road with good road conditions is high, here is a B-level road as an example:

$$\begin{cases} G_q(n_0) = 64 \times 10^{-6}\text{m}^{-3} \\ f_0 = 0.1\text{Hz} \end{cases}$$

According to the road roughness model and the initially set parameters, the road roughness excitation $q(t)$ can be obtained in the Simulink environment.

I built a road roughness Simulink model to simulate the rough road according to equation (9)

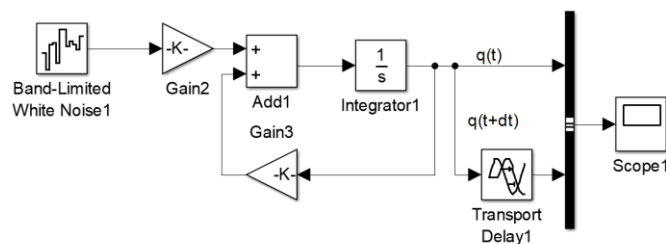


Figure 2. Road roughness Simulink model

As is shown in figure 2, this report chooses to simulate road roughness excitation based on Simulink environment. There are two main outputs for road roughness, the front wheel road roughness $q(t)$ and the rear wheel road roughness $q(t + T)$. Rear wheel output lags front wheel output for T . Assuming that the front and rear wheel axles are on the same rut, the front and rear road roughness displacement outputs q_1 and q_2 are only one time lag in time Δt , where $\Delta t = L/v$, v is the vehicle speed. The hysteresis function can be implemented using the Transport Delay module.

2 Single wishbone beam independent suspension model

If the car does not move, the car is in a static stress state, that is, the ground normal reaction force on a tire is $\frac{(G_s + G_u)}{2}$, G_s is the suspension mass gravity, G_u is Non-suspension mass gravity; then add an upward micro-element force $\Delta F'_z$ on the tire, thereby causing the micro-component displacement Δs_t of the wheel in the vertical direction and the micro-element displacement Δs_s of the spring along its center. The spring force also increases ΔQ , $\Delta Q = k_s \Delta s_s$, k_s is the spring stiffness.

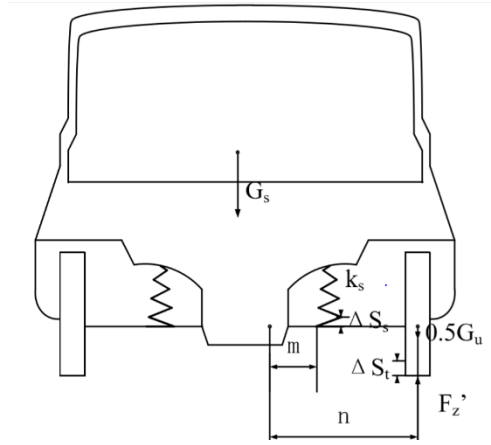


Figure 3. Single wishbone beam independent suspension model.

As can be seen in figure 3:

$$\frac{\Delta s_s}{m} = \frac{\Delta s_t}{n} \quad (10)$$

Where m is the distance from the center of the spring to the hinge point of the wishbone beam; n is the length of the wishbone beam.

In addition, according to the torque balance:

$$\Delta F'_z n = \Delta Q m = k_s \Delta s_s m \quad (11)$$

According to (14)(15):

$$\Delta F'_z = k_s \frac{m}{n} \Delta s_s = k_s \left(\frac{m}{n} \right)^2 \Delta s_t \quad (12)$$

One side suspension stiffness is:

$$\frac{\Delta F'_z}{\Delta s_t} = k_s \left(\frac{m}{n} \right)^2 \Delta s_t \quad (13)$$

The whole suspension stiffness is:

$$k_c = 2k_s \left(\frac{m}{n} \right)^2 \quad (14)$$

According to (18), the front suspension stiffness is:

$$k_{cf} = 2k_{0f} \left(\frac{m_f}{n_f} \right)^2 \quad (15)$$

The rear suspension stiffness is:

$$k_{cr} = 2k_{0r} \left(\frac{m_r}{n_r} \right)^2 \quad (16)$$

Road roughness excitation $q(t)$, the stiffness of vehicle front suspension and rear suspension k_{cf}, k_{cr} , and vehicle parameters are available now.

3 Five-degree-freedom vehicle model

I built the four-degree-freedom vehicle model according to the suspension calculation.

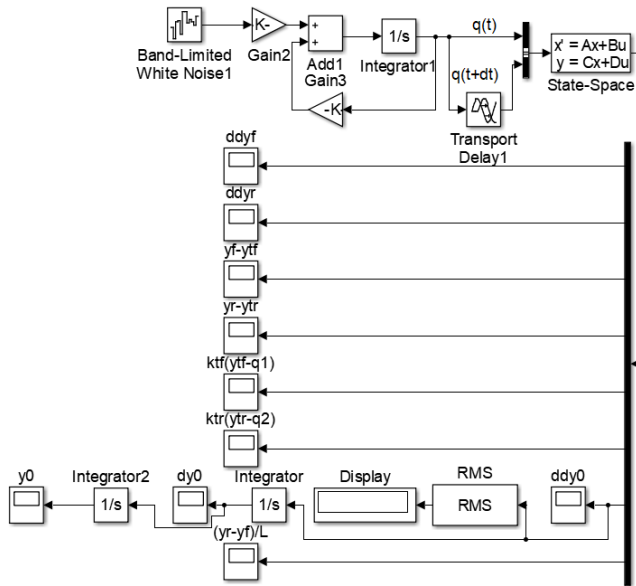


Figure 4. Four-degree-freedom vehicle Simulink model

After I introduced parameters of vehicle, I can get the acceleration of vehicle body

$$a_w(t) \text{ and the RMS value of the acceleration } a_w = \left[\frac{\int_0^T a_w^2(t) dt}{T} \right]^{\frac{1}{2}}.$$

4 Some parameters calculated

In order to get some parameters I need, I need to calculate the parameters, such as A, C matrix.

```
function [A,B1,B2]=qiuAB1B2(kcf,kcr,ccf,ccr,mtf,mtr,m,J)
mp=53;
kp=19030;
ktf=571420;
ktr=571420;
cp=170;
a=0.183;
b=0.947;
d=1.353;
B1=[0;0;0;ktf/mtf;0;0;0;0;0;0;0];
B2=[0;0;0;0;0;ktr/mtr;0;0;0;0;0];
A=[0 1 0 0 0 0 0 0 0 0;-kp/mp -cp/mp 0 0 0 0 kp/mp cp/mp -kp*a/mp -cp*a/mp;0 0 0 1 0 0 0 0 0 0;0 0 -(kcf+ktf)/
format short g;

end
```

Figure 5. MATLAB programme

Experiment of suspension

In order to get the parameters I need (all the parameters I used are from the appendix), the damping coefficient can be only got from experiment data. So I and Leo and Sam went to workshop to test the damping coefficient.

Time	Force	Displacement	Velocity	Time	Force	Displacement	Velocity												
0.00000	-569.78 N	12.56 mm	42.73 mm/s	0.00000	-249.60 N	12.59 mm	49.57 mm/s	0.00000	13.28 N	12.68 mm	40.27 mm/s	0.00000	0.12 N	12.71 mm	40.07 mm/s	0.00000	11.46 N		
0.00032	-567.44 N	12.56 mm	42.16 mm/s	0.00032	-195.69 N	12.62 mm	48.24 mm/s	0.00032	8.59 N	12.69 mm	38.18 mm/s	0.00032	9.50 N	12.70 mm	37.99 mm/s	0.00032	18.50 N		
0.00064	-532.28 N	12.57 mm	42.16 mm/s	0.00064	-151.16 N	12.65 mm	46.15 mm/s	0.00064	13.28 N	12.67 mm	36.28 mm/s	0.00064	18.87 N	12.68 mm	36.47 mm/s	0.00064	27.87 N		
0.00096	-473.69 N	12.57 mm	41.78 mm/s	0.00096	-108.97 N	12.64 mm	43.87 mm/s	0.00096	17.96 N	12.67 mm	34.76 mm/s	0.00096	28.25 N	12.71 mm	34.95 mm/s	0.00096	39.59 N		
0.00128	-401.03 N	12.57 mm	41.40 mm/s	0.00128	-87.88 N	12.63 mm	42.16 mm/s	0.00128	27.34 N	12.68 mm	33.05 mm/s	0.00128	39.96 N	12.69 mm	33.24 mm/s	0.00128	48.96 N		
0.00160	-326.03 N	12.57 mm	41.03 mm/s	0.00160	-62.10 N	12.65 mm	39.69 mm/s	0.00160	39.06 N	12.67 mm	31.15 mm/s	0.00160	54.03 N	12.69 mm	31.53 mm/s	0.00160	58.34 N		
0.00192	-255.72 N	12.56 mm	40.46 mm/s	0.00192	-38.66 N	12.65 mm	37.61 mm/s	0.00192	53.12 N	12.69 mm	29.63 mm/s	0.00192	68.09 N	12.69 mm	30.20 mm/s	0.00192	67.71 N		
0.00224	-185.41 N	12.56 mm	39.51 mm/s	0.00224	-10.54 N	12.67 mm	35.14 mm/s	0.00224	69.53 N	12.68 mm	27.92 mm/s	0.00224	75.12 N	12.69 mm	28.68 mm/s	0.00224	81.78 N		
0.00256	-122.13 N	12.57 mm	37.80 mm/s	0.00256	8.21 N	12.65 mm	32.67 mm/s	0.00256	83.59 N	12.65 mm	26.40 mm/s	0.00256	82.15 N	12.69 mm	27.16 mm/s	0.00256	88.81 N		
0.00288	-72.91 N	12.57 mm	36.09 mm/s	0.00288	36.34 N	12.64 mm	30.20 mm/s	0.00288	92.96 N	12.69 mm	25.07 mm/s	0.00288	93.87 N	12.67 mm	25.64 mm/s	0.00288	93.49 N		
0.00320	-35.41 N	12.58 mm	33.81 mm/s	0.00320	64.46 N	12.65 mm	27.73 mm/s	0.00320	104.68 N	12.70 mm	23.55 mm/s	0.00320	103.24 N	12.67 mm	23.93 mm/s	0.00320	106.53 N		
0.00362	4.43 N	12.57 mm	31.91 mm/s	0.00362	83.21 N	12.65 mm	25.64 mm/s	0.00362	118.74 N	12.72 mm	22.41 mm/s	0.00362	110.28 N	12.70 mm	22.03 mm/s	0.00362	114.59 N		
0.00394	44.27 N	12.59 mm	29.82 mm/s	0.00394	99.62 N	12.63 mm	23.55 mm/s	0.00394	130.46 N	12.70 mm	20.89 mm/s	0.00394	119.65 N	12.69 mm	20.51 mm/s	0.00394	123.96 N		
0.00436	74.74 N	12.58 mm	27.35 mm/s	0.00436	118.37 N	12.61 mm	21.65 mm/s	0.00436	137.49 N	12.70 mm	19.37 mm/s	0.00436	133.71 N	12.69 mm	18.61 mm/s	0.00436	133.34 N		
0.00468	107.55 N	12.57 mm	25.07 mm/s	0.00468	139.46 N	12.63 mm	19.37 mm/s	0.00468	149.21 N	12.68 mm	17.47 mm/s	0.00468	147.77 N	12.65 mm	16.71 mm/s	0.00468	142.71 N		
0.00498	145.05 N	12.57 mm	22.41 mm/s	0.00498	160.55 N	12.65 mm	17.85 mm/s	0.00498	163.27 N	12.62 mm	15.57 mm/s	0.00498	152.46 N	12.62 mm	14.62 mm/s	0.00498	156.77 N		
0.00531	189.58 N	12.57 mm	19.56 mm/s	0.00531	186.33 N	12.64 mm	15.76 mm/s	0.00531	182.02 N	12.64 mm	13.68 mm/s	0.00531	161.84 N	12.61 mm	12.73 mm/s	0.00531	168.49 N		
0.00564	227.08 N	12.57 mm	16.71 mm/s	0.00564	214.46 N	12.61 mm	14.05 mm/s	0.00564	193.74 N	12.63 mm	11.97 mm/s	0.00564	175.90 N	12.62 mm	10.83 mm/s	0.00564	180.21 N		
0.00597	255.21 N	12.59 mm	14.43 mm/s	0.00597	247.27 N	12.64 mm	12.54 mm/s	0.00597	205.46 N	12.61 mm	10.26 mm/s	0.00597	189.96 N	12.61 mm	9.12 mm/s	0.00597	189.59 N		
0.00630	288.02 N	12.56 mm	12.16 mm/s	0.00630	275.40 N	12.61 mm	10.64 mm/s	0.00630	219.52 N	12.61 mm	8.74 mm/s	0.00630	204.02 N	12.59 mm	7.41 mm/s	0.00630	205.99 N		
0.00664	320.83 N	12.57 mm	9.88 mm/s	0.00664	298.83 N	12.60 mm	9.31 mm/s	0.00664	238.27 N	12.59 mm	6.84 mm/s	0.00664	218.09 N	12.57 mm	5.51 mm/s	0.00664	217.71 N		
0.00697	351.30 N	12.57 mm	8.17 mm/s	0.00697	326.96 N	12.61 mm	7.98 mm/s	0.00697	249.99 N	12.57 mm	5.32 mm/s	0.00697	227.46 N	12.55 mm	3.80 mm/s	0.00697	231.77 N		
0.00730	388.80 N	12.55 mm	6.46 mm/s	0.00730	359.77 N	12.58 mm	6.84 mm/s	0.00730	271.09 N	12.54 mm	3.42 mm/s	0.00730	241.52 N	12.54 mm	2.28 mm/s	0.00730	241.15 N		
0.00763	433.33 N	12.56 mm	4.94 mm/s	0.00763	380.86 N	12.57 mm	5.70 mm/s	0.00763	285.15 N	12.53 mm	1.52 mm/s	0.00763	255.59 N	12.54 mm	0.76 mm/s	0.00763	250.52 N		
0.00799	475.52 N	12.55 mm	3.42 mm/s	0.00799	411.33 N	12.56 mm	4.18 mm/s	0.00799	301.55 N	12.51 mm	0.19 mm/s	0.00799	264.96 N	12.49 mm	-0.95 mm/s	0.00799	264.59 N		
0.00830	513.02 N	12.53 mm	2.28 mm/s	0.00830	439.46 N	12.53 mm	2.47 mm/s	0.00830	320.30 N	12.48 mm	-1.52 mm/s	0.00830	279.02 N	12.47 mm	-2.66 mm/s	0.00830	271.62 N		
0.00863	536.45 N	12.53 mm	1.14 mm/s	0.00863	472.27 N	12.53 mm	1.14 mm/s	0.00863	336.71 N	12.45 mm	-3.23 mm/s	0.00863	293.09 N	12.45 mm	-4.18 mm/s	0.00863	283.34 N		
0.00895	564.58 N	12.50 mm	0.19 mm/s	0.00895	500.39 N	12.49 mm	-0.76 mm/s	0.00895	355.46 N	12.41 mm	-4.94 mm/s	0.00895	302.46 N	12.41 mm	-5.51 mm/s	0.00895	292.71 N		
0.00907	599.99 N	12.40 mm	-1.14 mm/s	0.00907	539.89 N	12.40 mm	-0.47 mm/s	0.00907	371.87 N	12.40 mm	-5.84 mm/s	0.00907	311.84 N	12.40 mm	-7.41 mm/s	0.00907	305.60 N		

Figure 6. Experiment data

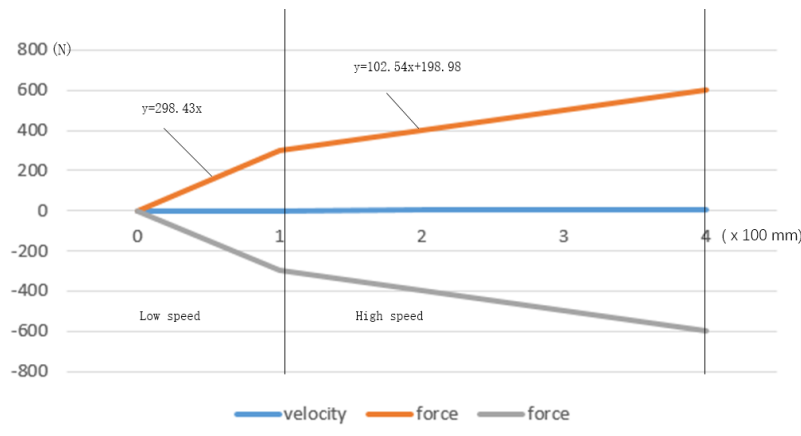


Figure 7. processed graph

Figure 7 above is the fitting curve of the experimental data. According to the definition of damping coefficient, we can know the gradient of the curve is the damping coefficient. At low speed, the damping coefficient is about $2984.3 \text{ N} \cdot \text{s/m}$. At high speed, the damping coefficient c_{cf} and c_{cr} is about $1025.4 \text{ N} \cdot \text{s/m}$.

Design optimization of suspension



Figure 8. Suspension

Based on the design of the suspension, the spring stiffness is nearly a constant value. Therefore, I choose to find an ideal spring stiffness k_{cf} and k_{cr} to minimize the vibration of the vehicle body. Meanwhile, find the acceleration signal of vehicle body under different damping coefficient.

Suspension simulation and testing

As is said in the build and optimize simulation model section, I have built road roughness model and four-degree-freedom vehicle model.

The result of road roughness model is as followed:

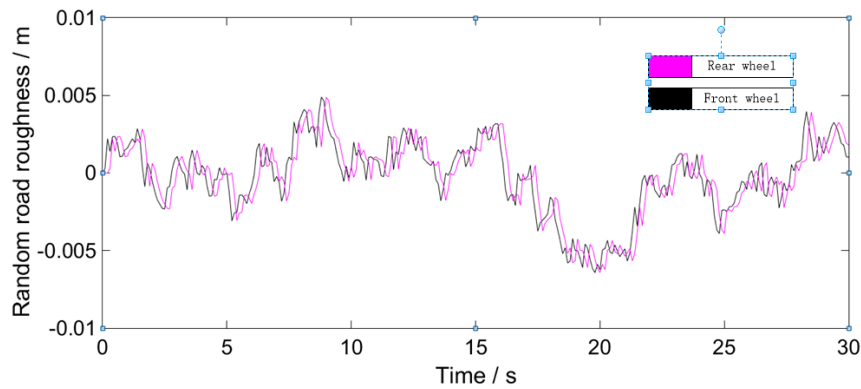


Figure 9. Road roughness excitation

Then we get the road roughness excitation $q(t)$.

Take the example of spring stiffness is 50000N/m and damping coefficient is 2984.3N • s/m (low speed), the result of four-degree-freedom Simulink model is as followed:

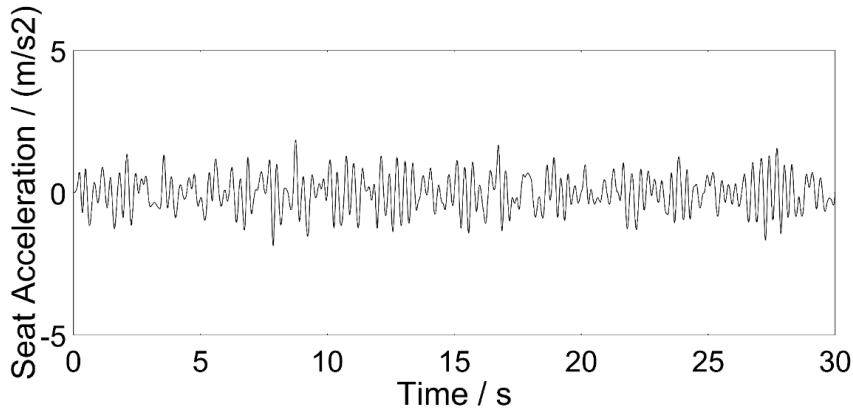


Figure 10. Acceleration of vehicle body under 50000N/m

And according to the value shown in Simulink, the RMS value is about 0.08882m/s^2 .

I have simulated acceleration of vehicle body under 60000N/m, 70000N/m, 80000N/m and some other values.

The RMS value of acceleration of vehicle body under different spring stiffness is as followed table:

Table 2. acceleration of vehicle body under different spring stiffness

Spring stiffness (N/m)	Vehicle body acceleration RMS value (m/s^2) $a_w = \left[\frac{\int_0^T a_w^2(t) dt}{T} \right]^{\frac{1}{2}}$
50000	0.08882
60000	0.06459
70000	0.03773
80000	0.04326
90000	0.06132

According to the result of the table, we can see the best front spring stiffness is 70000 (N/m). Using the same method, I can get the best rear spring stiffness is 90000(N/m).

Suspension analyzation

Since the suspension we use is a traditional suspension, the stiffness of the suspension is almost immutable, so my task is to find the best spring stiffness parameters and to measure and calculate the parameters related to the suspension. According to the above simulation results, it can be found that the ideal front spring stiffness of the Zentrak is 70000N/m and ideal rear spring stiffness is 90000N/m. The damping coefficient is 2984.3N•s/m in low speed while the damping coefficient is 1025.4 N•s/m.

Steering test



Figure 11. steering test

As for the steering part, our aim is to find the ideal steering ratio. So we went to the workshop to experiment the actual steering ratio.

The maximum steering angle of the wheel is about 28.7 and 27.5.

The maximum steering angle of the wheel is about 28.7 and 27.5. The steering wheel is about 1.8 turns lock-to-lock. So the rough steering ratio is 20:1.

Steering calculation

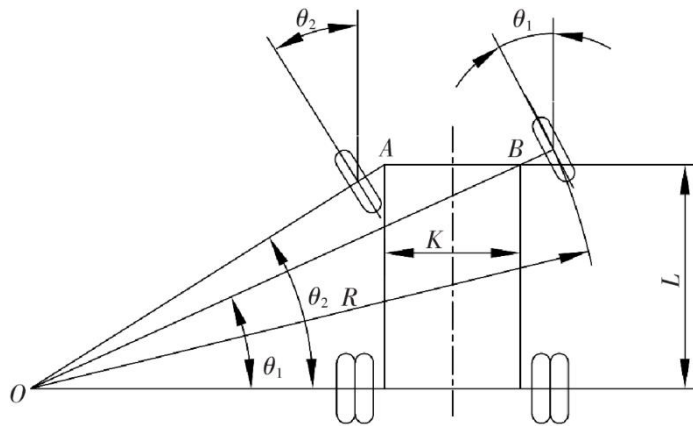


Figure 12. Ackerman model

According to the Geometric relationship:

$$\cot\theta_1 - \cot\theta_2 = K/L \quad (17)$$

where: θ_1 is the right front wheel steering angle; θ_2 is the left front wheel steering angle; K is the center distance of the left front wheel kingpin; L is the distance between the front wheel axis and the rear wheel axis.

Steering conclusion

After introducing the parameters of appendix to the equation (9), I found the initial set can perfectly meet the requirement of Ackerman model and the steering ratio is sensitive enough. So we think there is no need to change the initial set of steering system. So we spend more effort in suspension part.

Suriya Krishna

ABSTRACT

The pursuit for faster lap times around a track never ends. Motorsport is constantly evolving with up to the minute technology and cars are getting lighter and more potent without compromising on reliability and efficiency. This project is about improving the overall performance by reducing weight and finding new methods for making components lighter but also maintaining a middle ground between lightness and reliability and is based on the Zentrack car. Various parameters will be evaluated like the chassis, brakes, steering setup, suspension setup and will be fine-tuned by utilizing CAD software and a Simulator. The car will be setup in a such a way that it feels at home on a public road and at a racetrack. This project will aid us in learning technical knowledge about various materials and their properties, weight distribution, designing components, manufacturing standards and simulation. Success will be measured by how the vehicle performs on track compared to the stock vehicle and how content the owner of the car is after driving it.

PROJECT ORGANISATION - TEAM MEMBERS ROLES AND RESPONSIBILITIES

The team consists of nine members. We conduct group meetings every week and discuss various ideas for the project. The points shared in every meeting is recorded to keep track of milestones.

Team Roles:

Leo May – Team leader

- Responsible for the CAD model and evaluating the SIM model.

Harry Farnhill

- Responsible for the component analysis and evaluating the accuracy of the SIM model.

Oliver Duane

- Responsible for the CAD coordinates, fine tuning of the spring rates and dampers, evaluating coordinates on wingco and deciding on the brake setup.

Sam Treleaven

- Responsible for the stress testing of the chassis on CAD, working on wishbones to make it lighter without compromising the strength, analysing components and weighing of the car with different parameters.

Quan Zhou

- Responsible for the steering setup, fine tuning the spring rates and dampers and designing uprights.

Zhangwei Guo

- Responsible for evaluating brake setup.

Steven Wilkinson

- Responsible for calculating inertia of wheels and tyre setup.

Sun Zhaocong

- Responsible for identifying different evaluation methods.

PERSONAL ROLE

My role in this project is working on the uprights to make it as light as possible without compromising on its ability to withstand high levels of stress while acceleration, deceleration and cornering. The uprights are a component in a car's suspension which is used to form a connection between the upper and lower suspension arms. They are also used to mount the brake discs, the caliper and the wheel.

The stock uprights are weighed and measured first to set a baseline. The design of the uprights is evaluated and is altered on solidworks and then re-evaluated to find out if it has become lighter and has not compromised on the strength of the component. This is done by applying various forces using the stress analysis feature available on Solidworks. Once the stress analysis is complete, the manufacturing limitations, costs and the appropriate material is calculated out and decided on. The design gets refined before it is machined and finished using the chosen material in the workshop using specialist machines and tools.

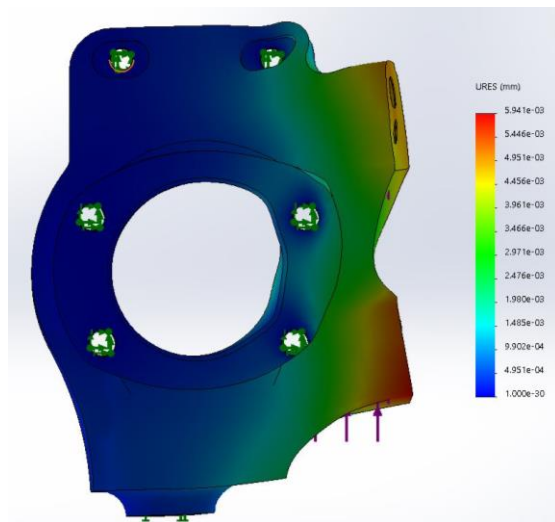
To redesign an upright, I should be able to understand the forces, stress and strain which will act on the car under varying situations, the manufacturing standards and cost limitations, operating specialist equipment and tools and a set of skills to operate Solidworks.

SUMMARY OF WORK COMPLETED

EXPERIMENTS, ANALYSIS, DESIGN, DEVELOPMENT AND TESTING

The uprights have received changes in their design to reduce weight without compromising on strength. For applying design changes, the CAD modelling software Solidworks was used.

The stock uprights are made of aluminium alloy and weigh 1.79 kilos. To attain a baseline, the stock uprights were stress tested under braking, acceleration and cornering conditions with a standard force of 2000 newtons.



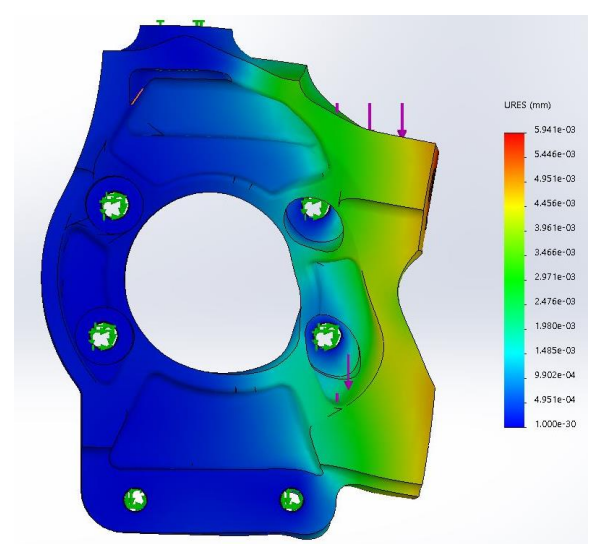
Braking tests:

Front view

Top wishbone mount view

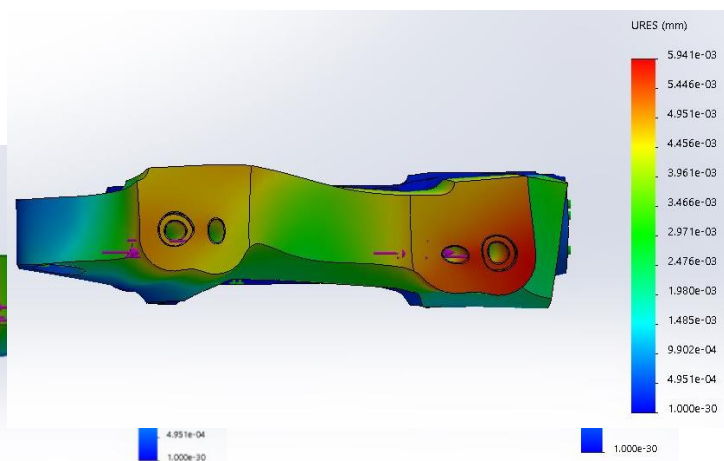
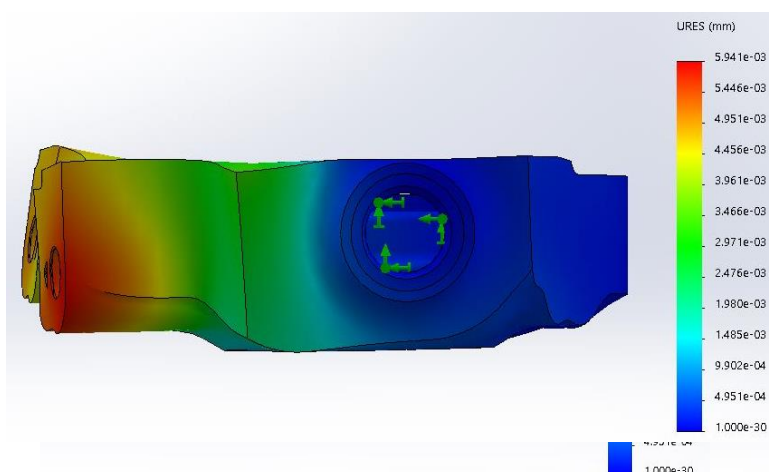
Caliper mount view

Acceleration tests:

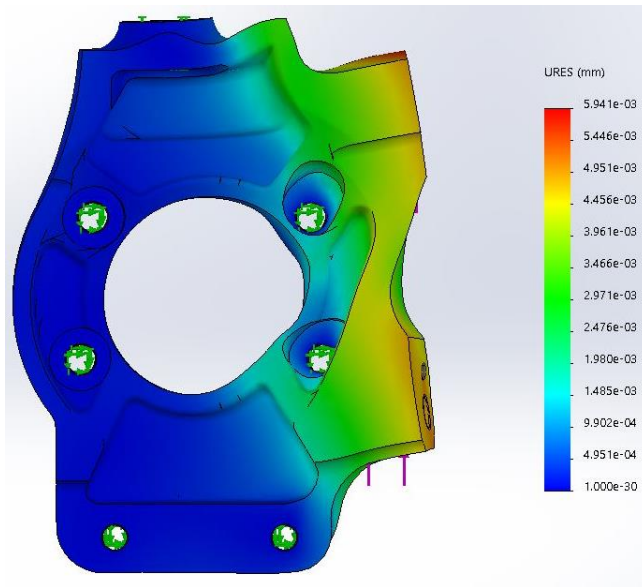


Rear view

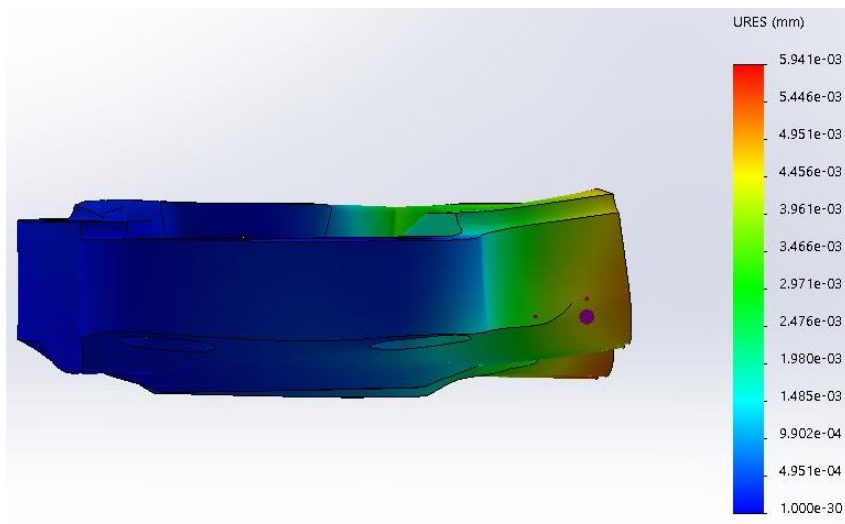
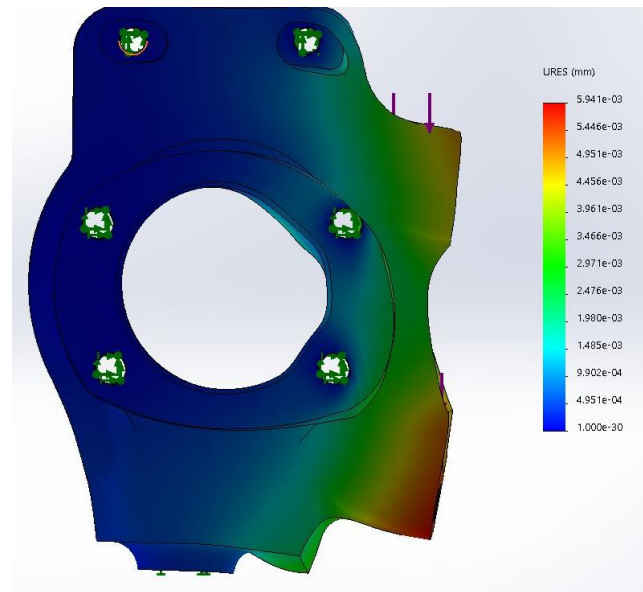
Bottom wishbone mount view



Bottom wishbone mount



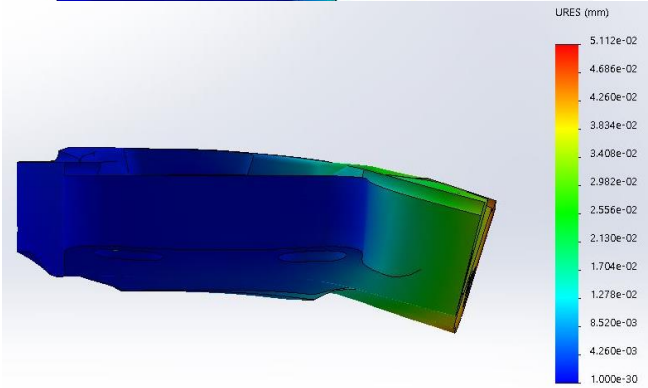
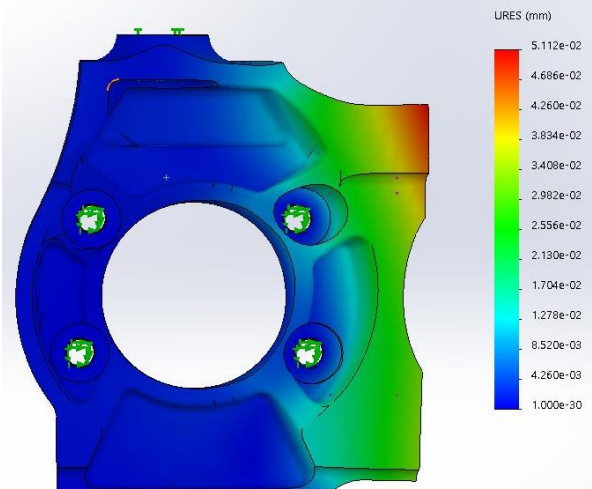
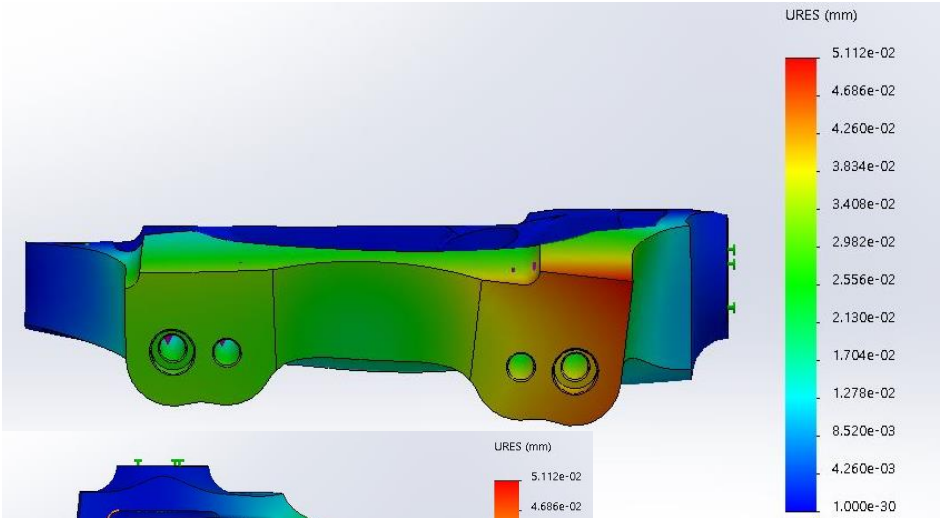
Caliper mount view

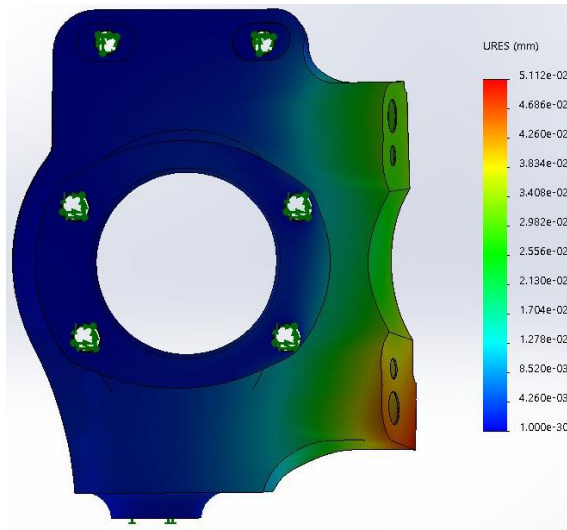


Rear view

Front view
Top wishbone mount view

Cornering tests:





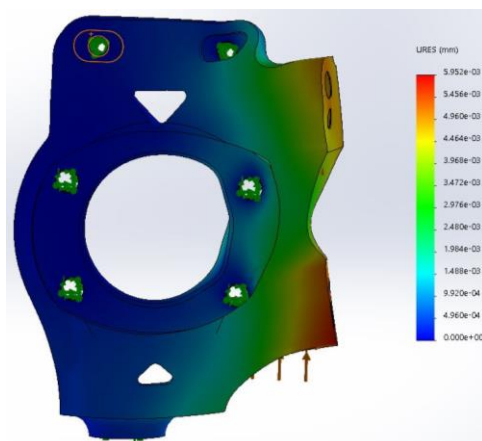
Front view

After completing the stress analysis on the stock design, I started tweaking the design to reduce weight. Three design variations were carried out on the same stock aluminium material and then was carried out on the uprights with magnesium alloy.

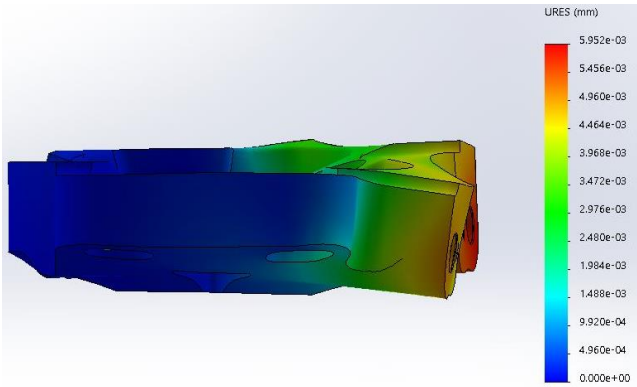
Design 1 with aluminium alloy:

The first design variation that I carried out was removing material near the top and bottom wishbone mounts on the front face of the upright. I decided to go with a triangle cutout first. The triangle was drawn on the front face and then extruded to remove the material. To give a smooth finish, I used the fillet tool (2mm) on the inner edges of the triangle. This dropped the weight from 1.79 kilos to 1.77 kilos, a saving of 20 grams from stock. In the braking test for stress analysis, there was almost zero change in strength compared to stock, but in the acceleration test, there was a slight decrease in strength in the bottom end of the upright and an increase in strength near the top hole for the brake mount section. The cornering test also showed a slight decrease in strength at the bottom section.

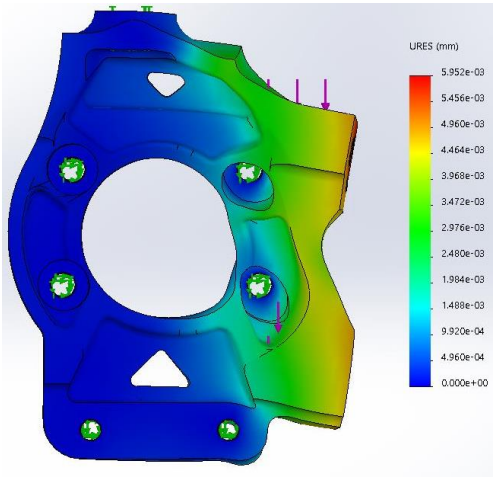
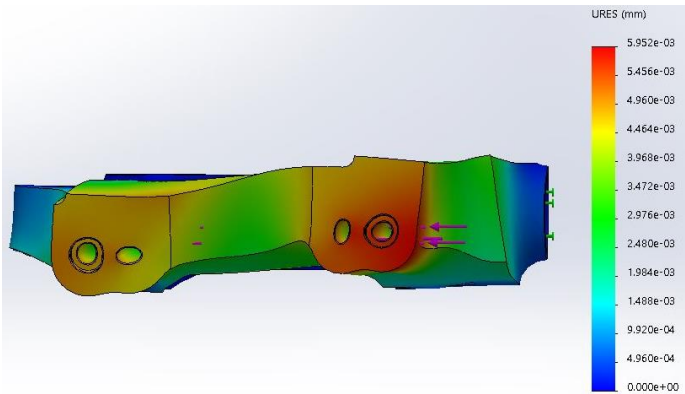
Brake test:



Front view

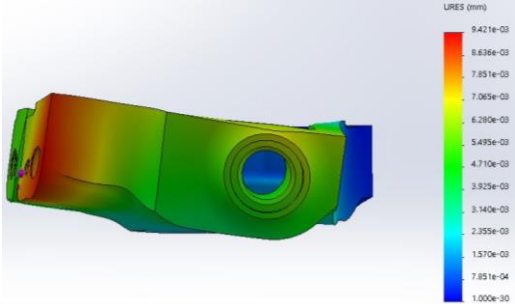


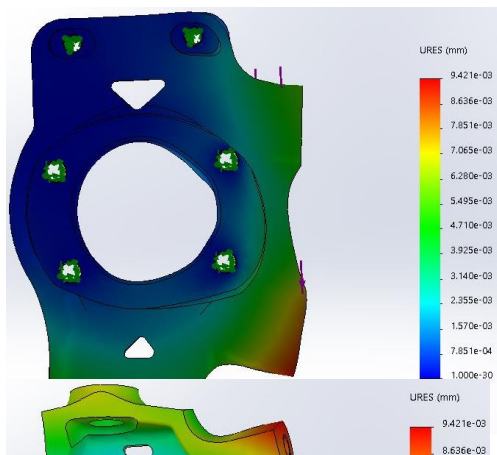
Top wishbone mount view



Caliper mount view

Rear view

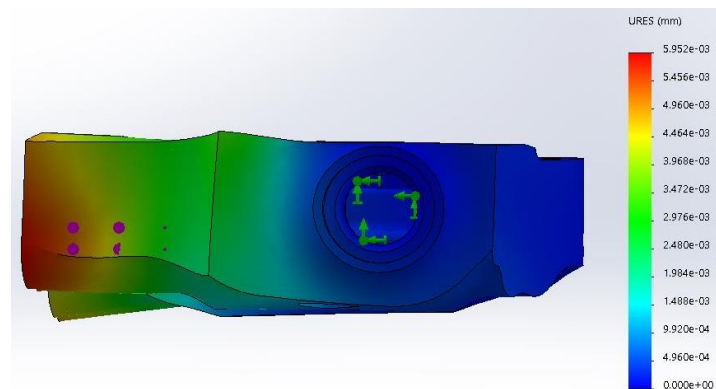
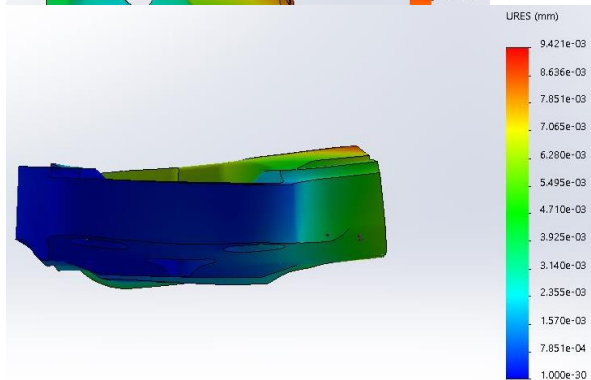
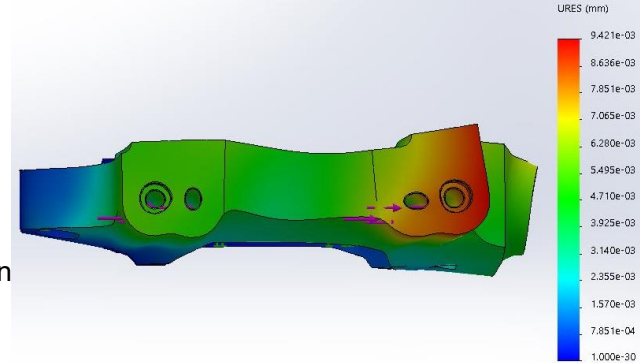




Caliper moun

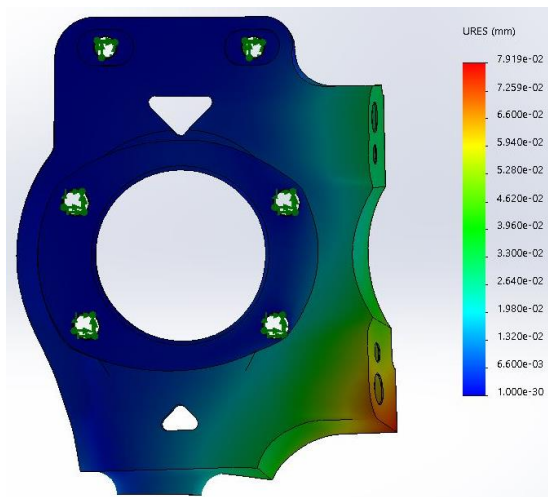
Front view

Rear view

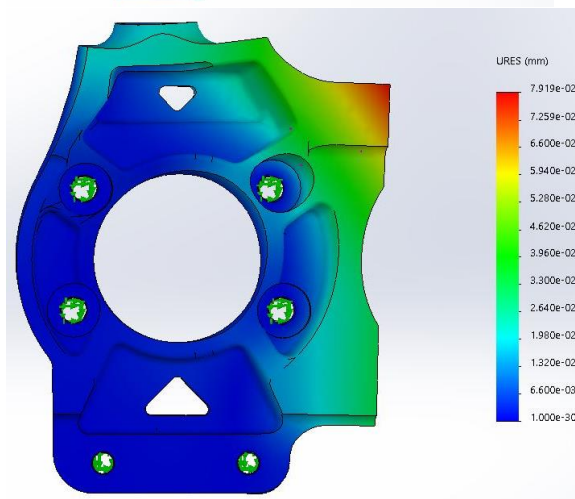


Top wishbone mount view

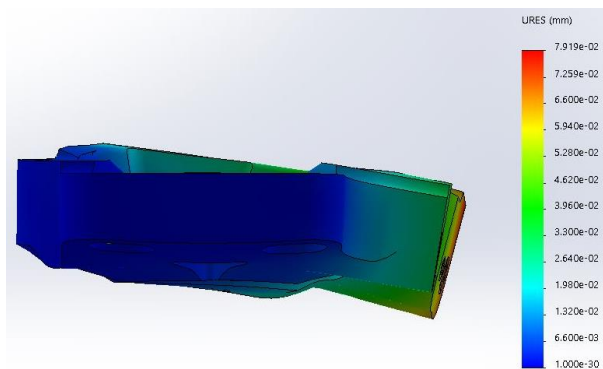
Cornering test:



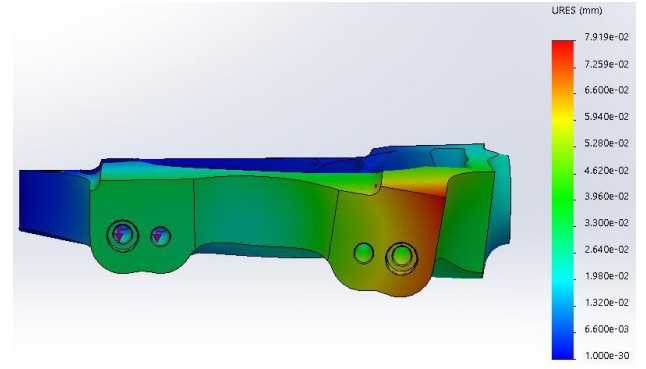
Front view



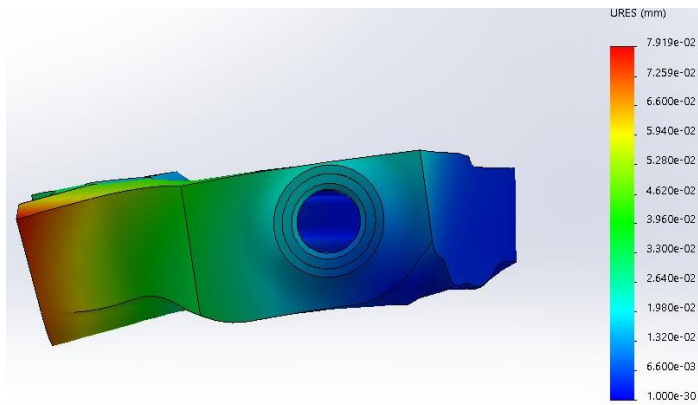
Rear view



Top wishbone mount view



Caliper mount view

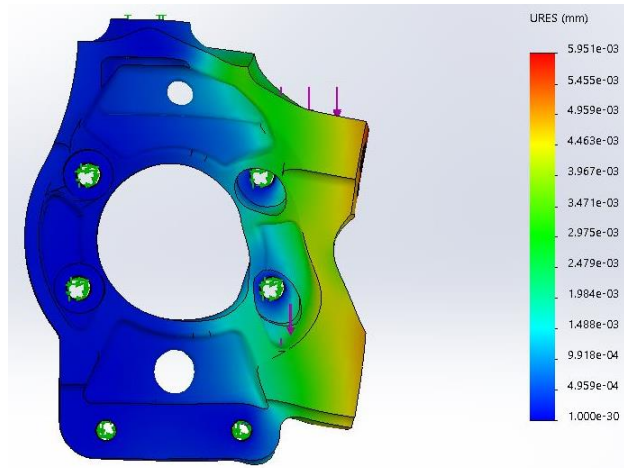
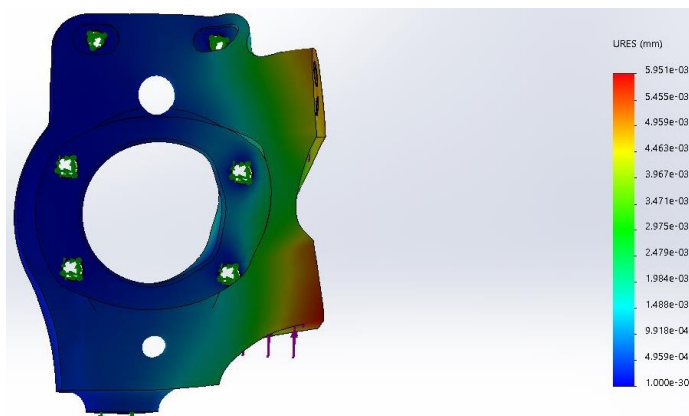


Bottom wishbone mount view

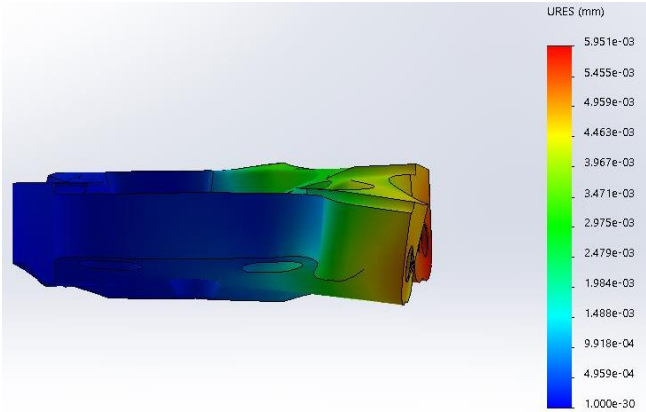
Design 2 with aluminium alloy:

In this second design change, the triangle cutout was replaced with a circular cutout in the front face. The weight remained unchanged from the first design at 1.77 kilos. This design change displayed better results than the first design variation as there was almost zero change in strength compared to stock in the braking, acceleration and cornering tests whereas the first design variation with the triangle cutout showed a decrease in strength.

Braking tests:

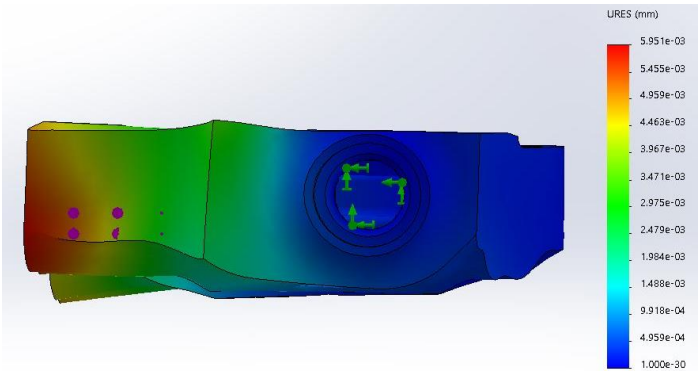
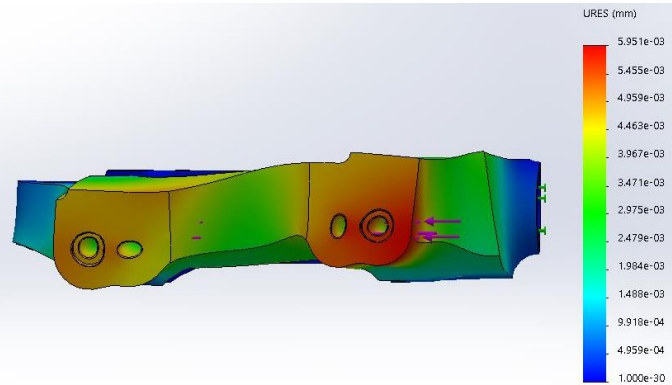


Front view



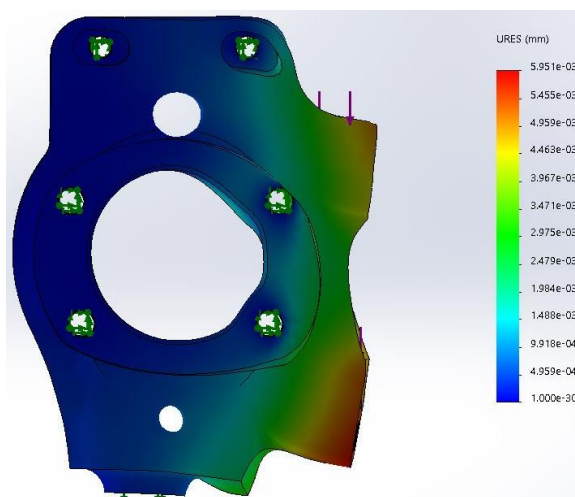
Rear view

Caliper mount view
Top wishbone mount view

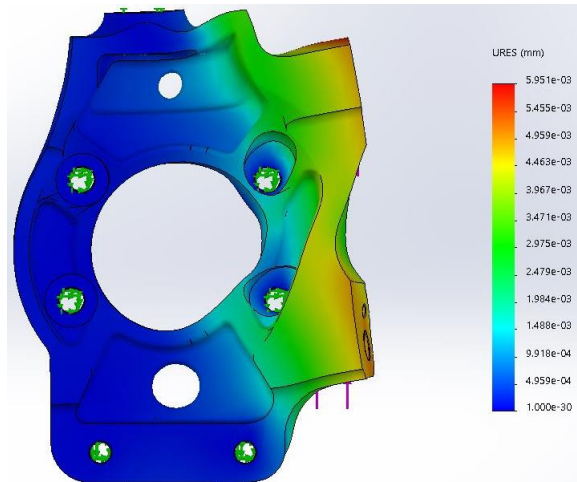


Bottom wishbone mount view

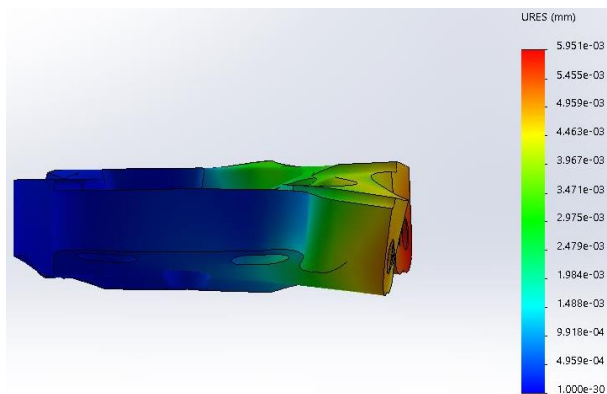
Acceleration tests:



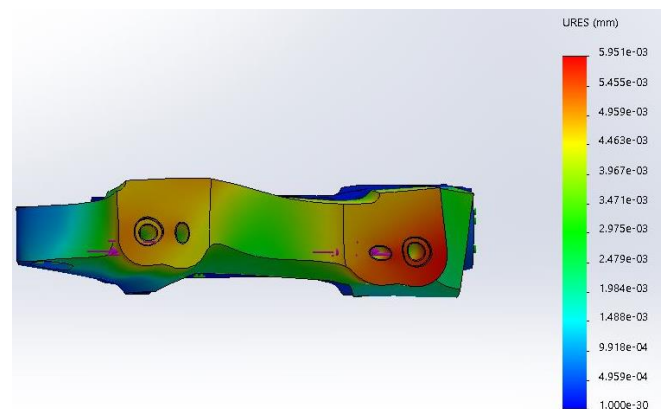
Front view



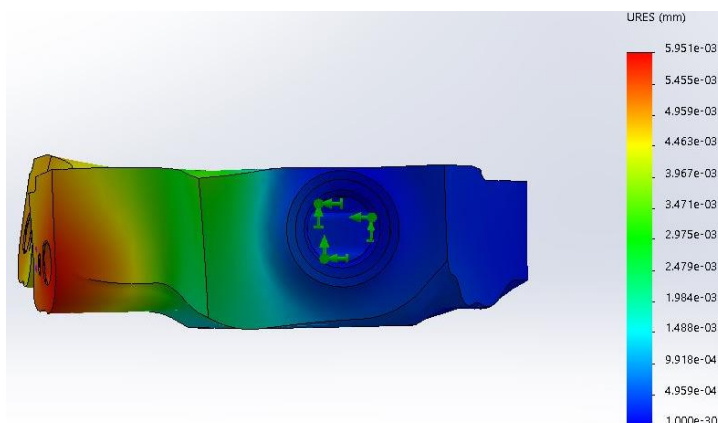
Rear view



Top wishbone mount view

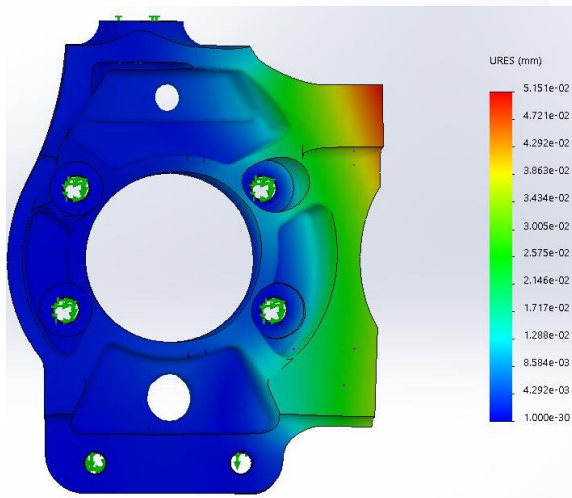
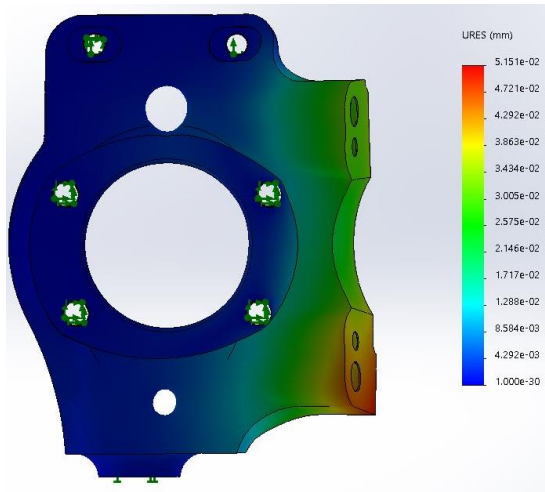


Caliper mount view



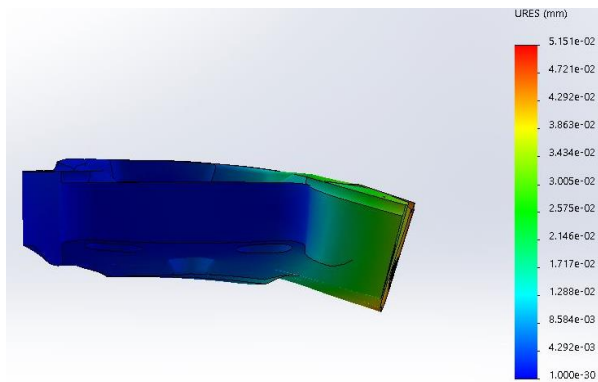
Bottom wishbone mount view

Cornering tests:

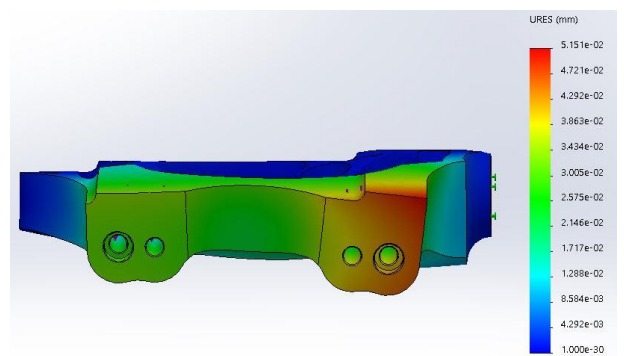


Front view

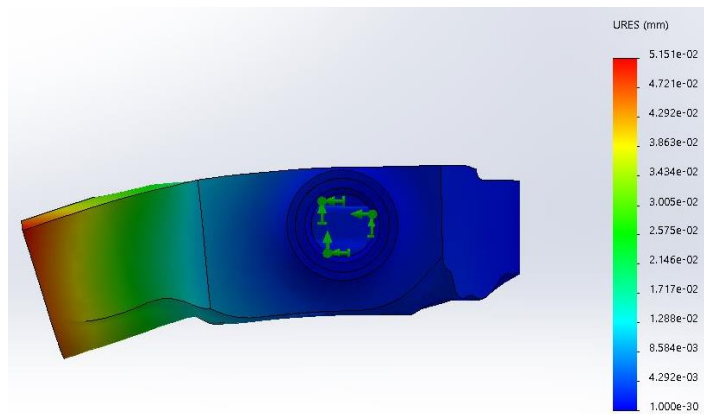
Rear view



Top wishbone mount view



Caliper mount view

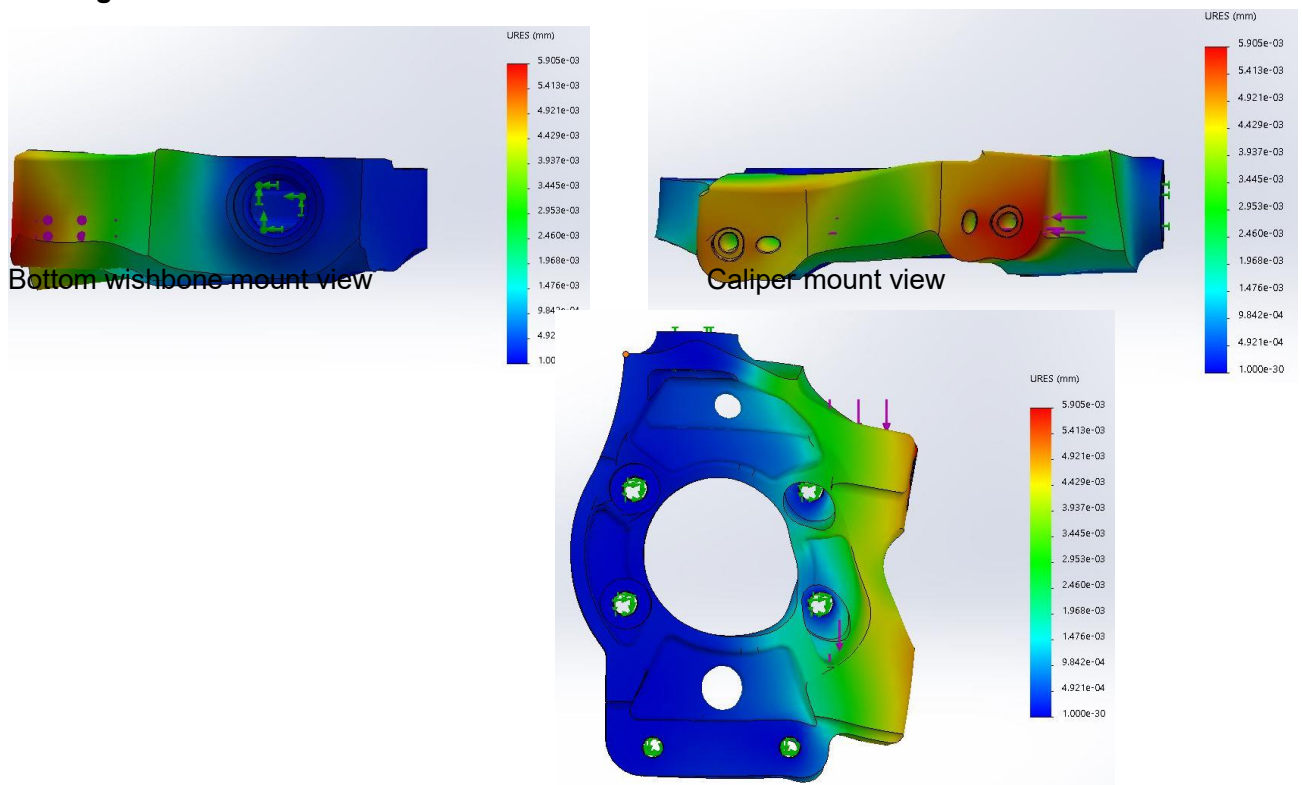


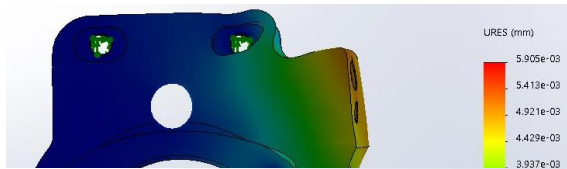
Bottom wishbone mount view

Design 3 with aluminium alloy:

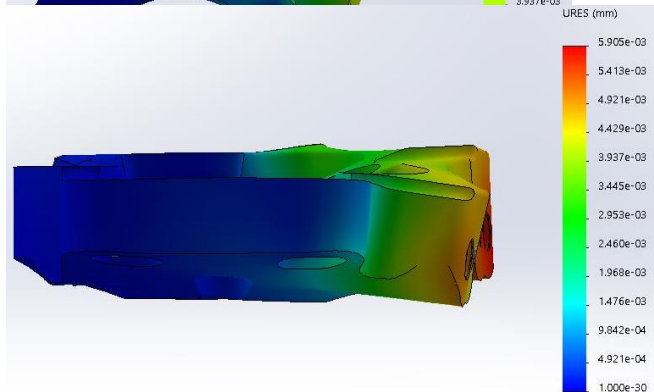
This is the last design change using aluminium alloy. Design 2 was taken and the edges on the brake caliper mount was chamfered and a square cutout was made on the face opposing the brake caliper mounts. The weight was further reduced from 1.77 kilos to 1.76 kilos, a saving of 10 grams. This change did not cause a decrease in strength. So far, the weight has been reduced without compromising on strength.

Braking tests:



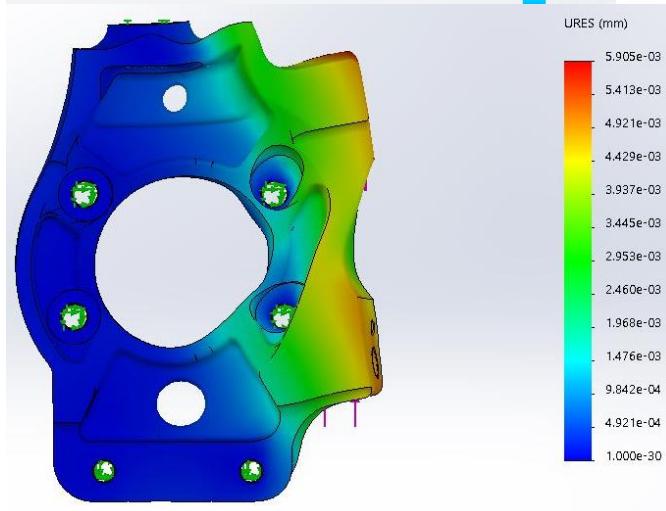
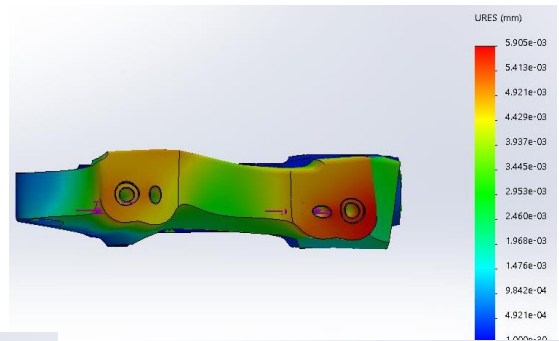
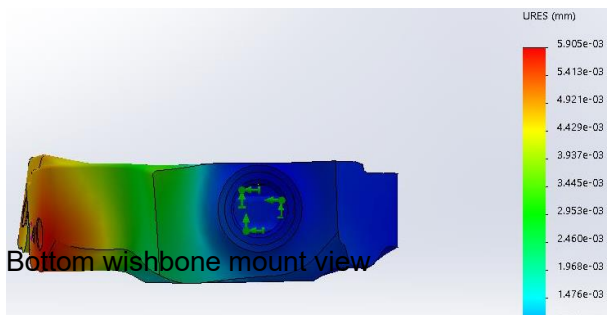


Front view
Rear view



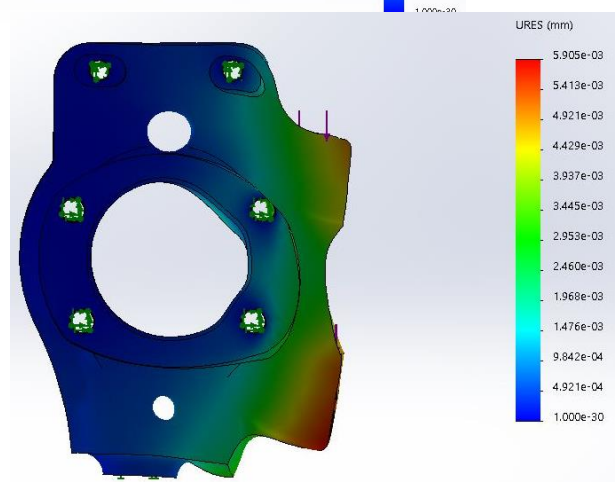
Top wishbone mount view

Acceleration tests:

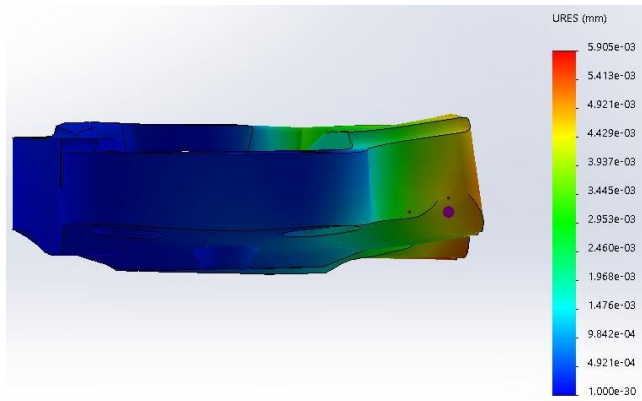


Caliper mount view

Rear view



Front view



Top wishbone mount view

Magnesium alloy:

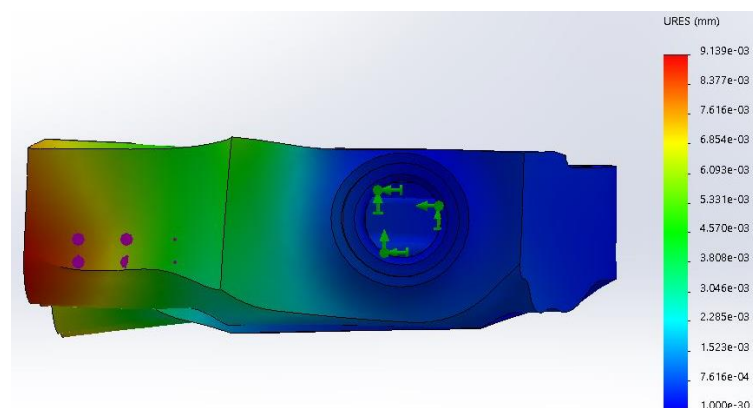
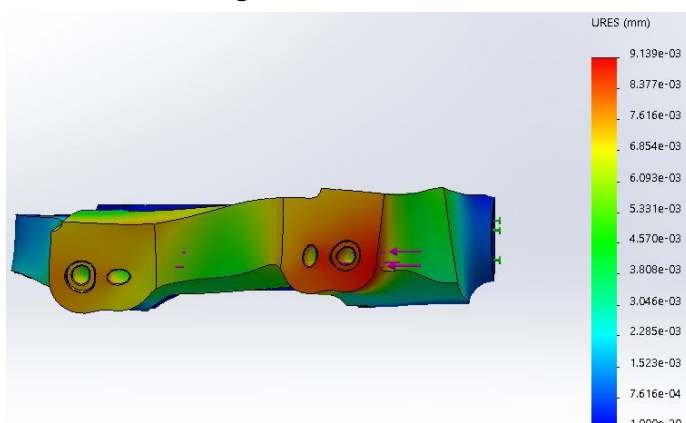
Compared to aluminium alloy, magnesium alloy is much lighter as it is softer but it also flexes more than aluminium. Magnesium alloy, as a metal alone is also more expensive to procure. It requires higher temperatures for formability and requires special coatings to protect it from corrosion. On the plus side, Magnesium is comparatively easier to machine and is more resistant to dents. The dies and machining tools used to create magnesium parts last longer and requires less energy for machining as it is a softer metal. This material was applied to the upright and tested as the main goal is to reduce weight. This design variations that were carried out on aluminium alloy were used on magnesium alloy to get a direct comparison between the two materials.

(Diecasting, n.d.) (Phys.org, n.d.) (Meridian, n.d.) (Research Gate, n.d.)

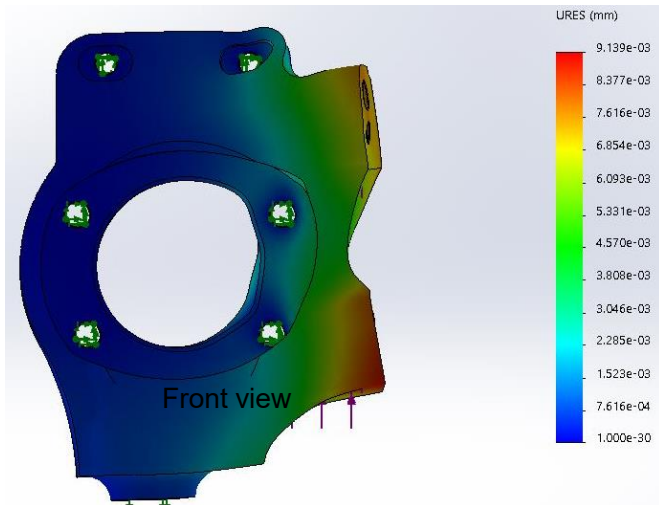
Stock design with magnesium alloy:

Changing the material from aluminium alloy to magnesium caused a drastic decrease in weight from 1.79 kilos down to 1.13 kilos. The tests carried out showed that magnesium alloy flexes more under braking, acceleration and cornering.

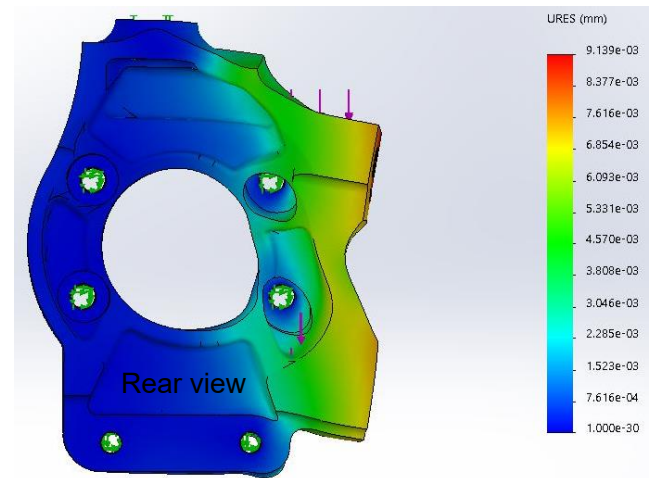
Braking test:



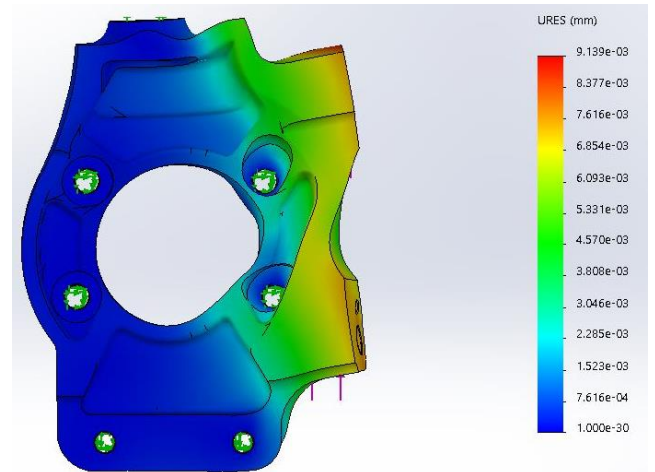
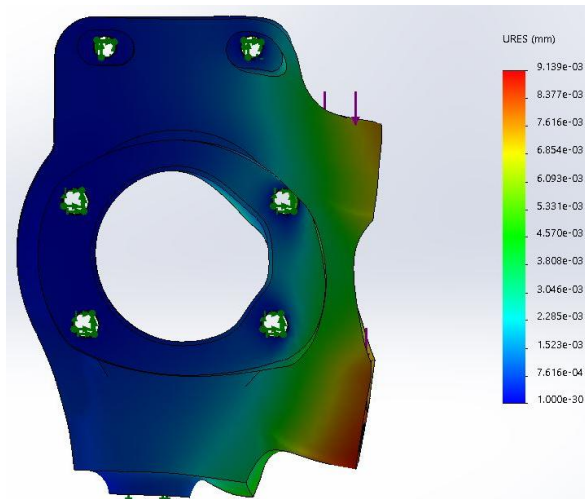
Caliper mount view



Bottom wishbone mount view

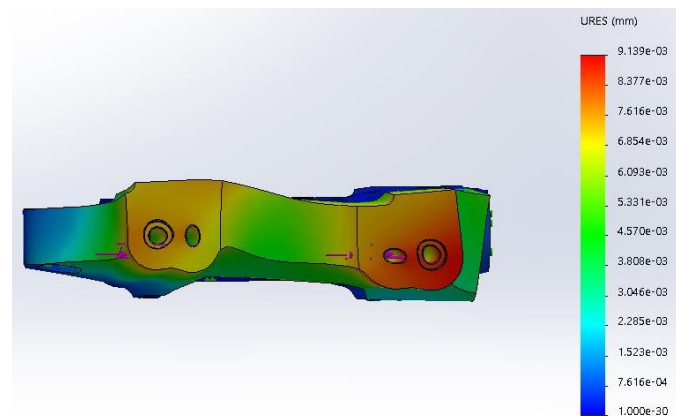
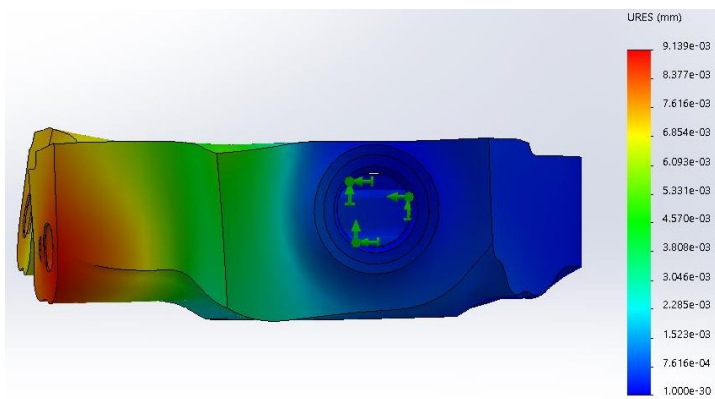


Acceleration test:



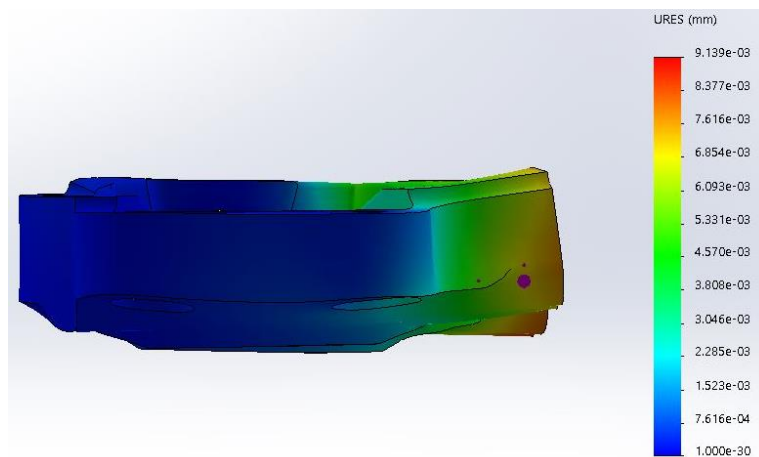
Front view

Rear view



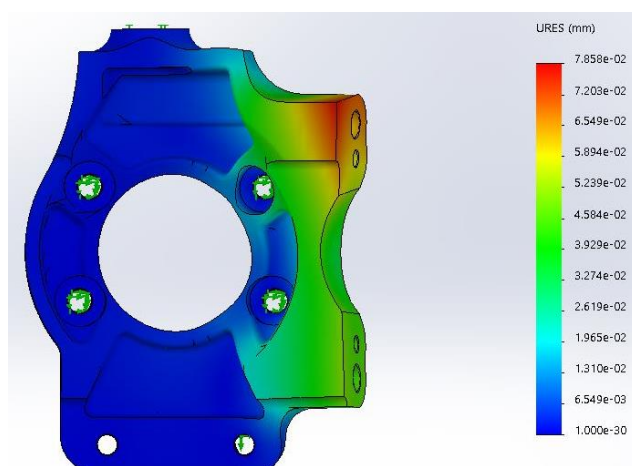
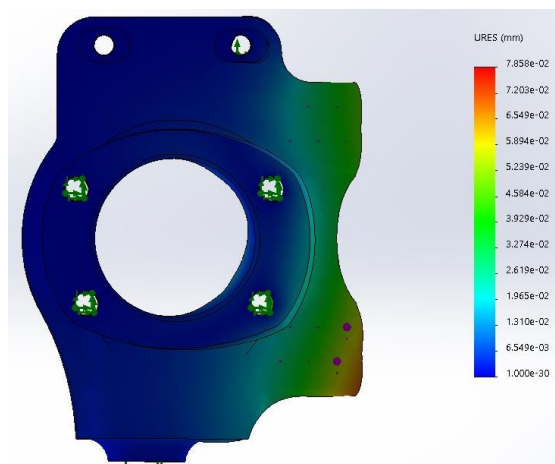
Bottom wishbone mount view

Caliper mount view

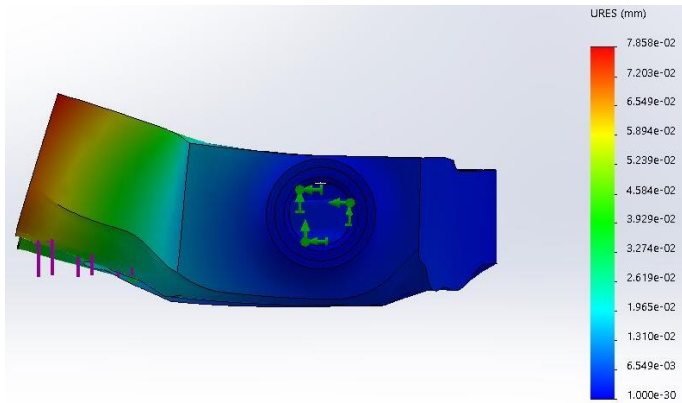


Top wishbone mount view

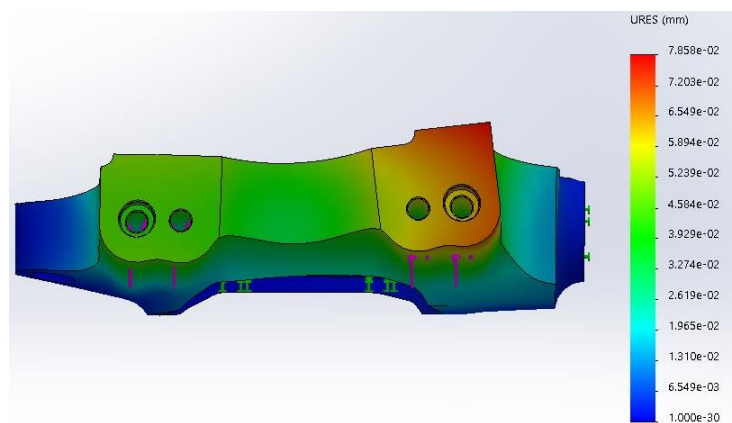
Cornering test:



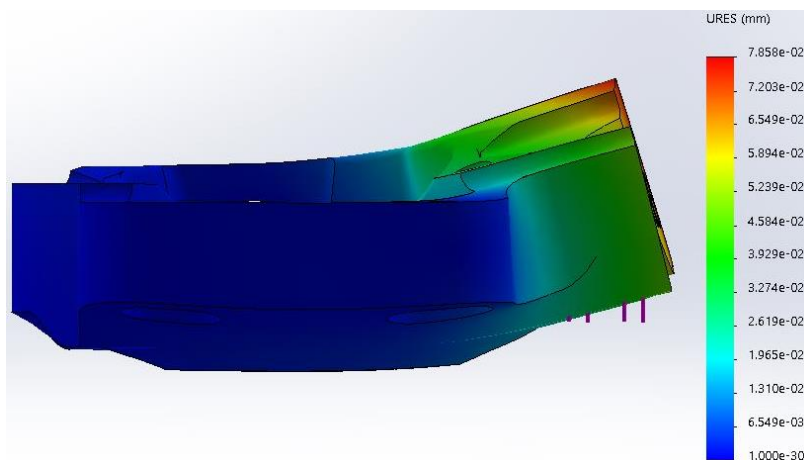
Front view



Rear view



Bottom wishbone mount view



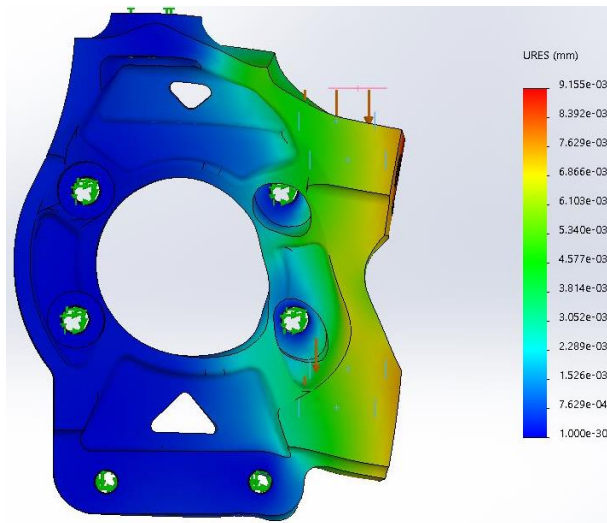
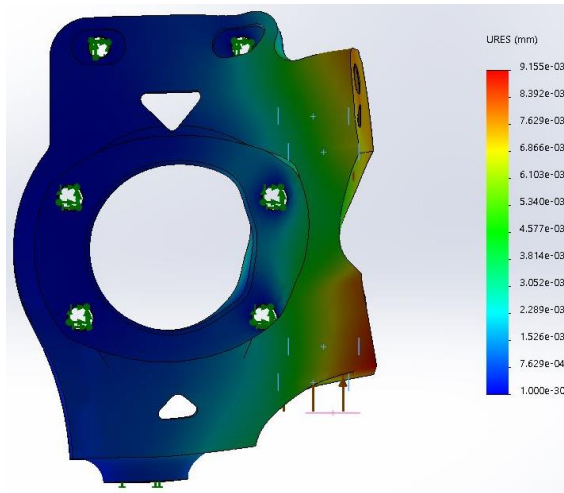
Caliper mount view

Top wishbone mount view

Design 1 with magnesium alloy:

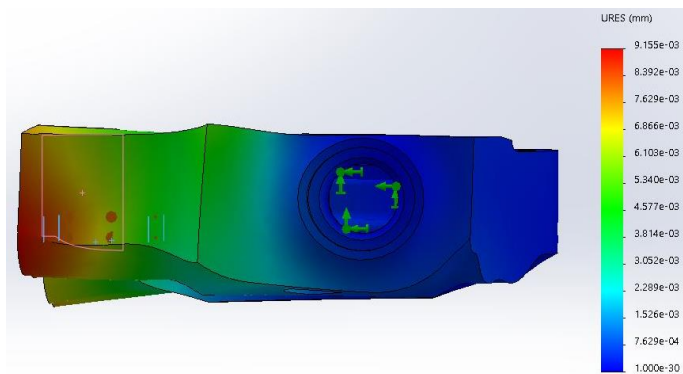
With this design, the weight was reduced from 1.13 kilos to 1.11 kilos, a saving of 20 grams.

Braking tests:

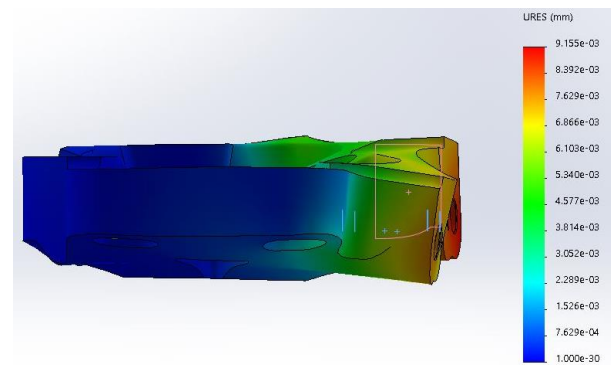


Front view

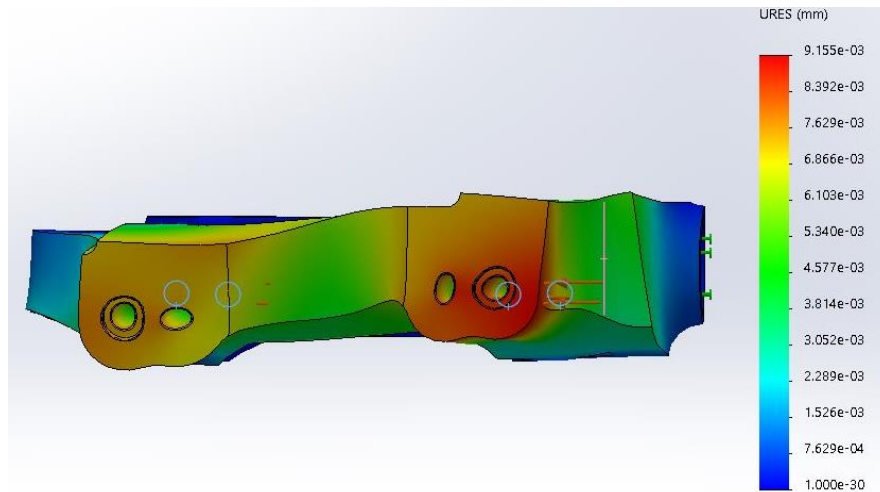
Rear view



Bottom wishbone mount view

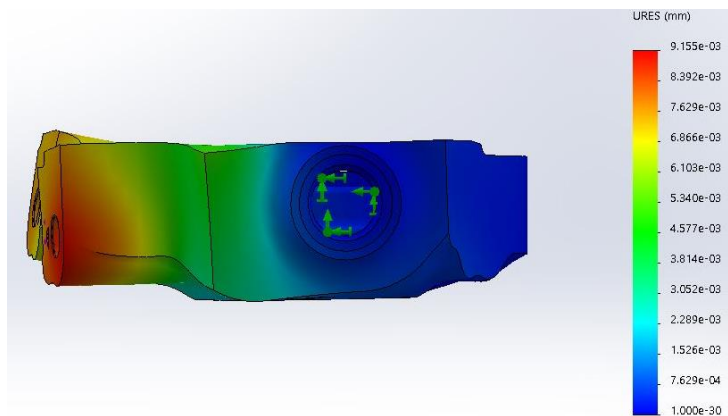


Top wishbone mount

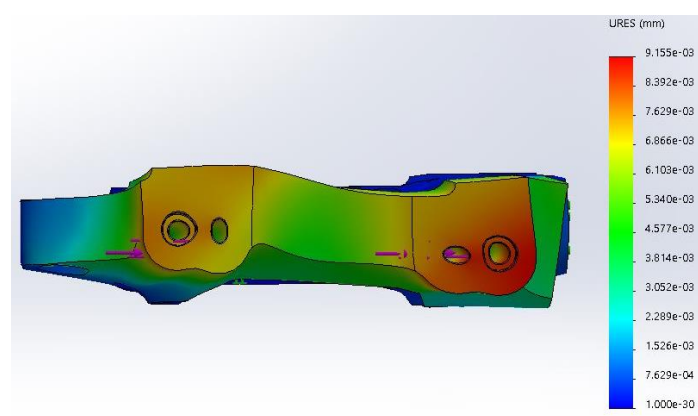


Caliper mount view

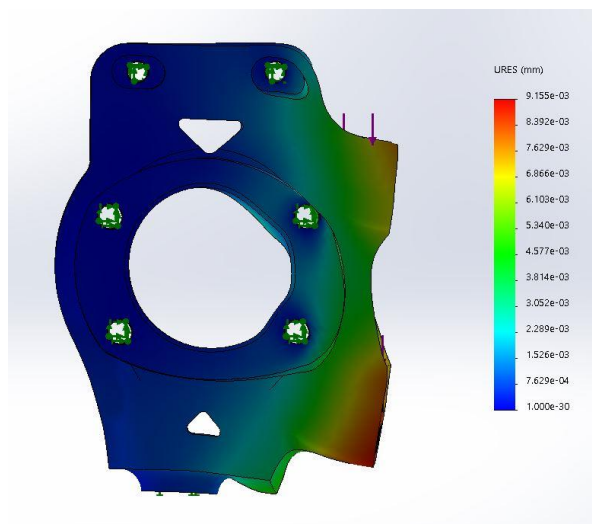
Acceleration tests:



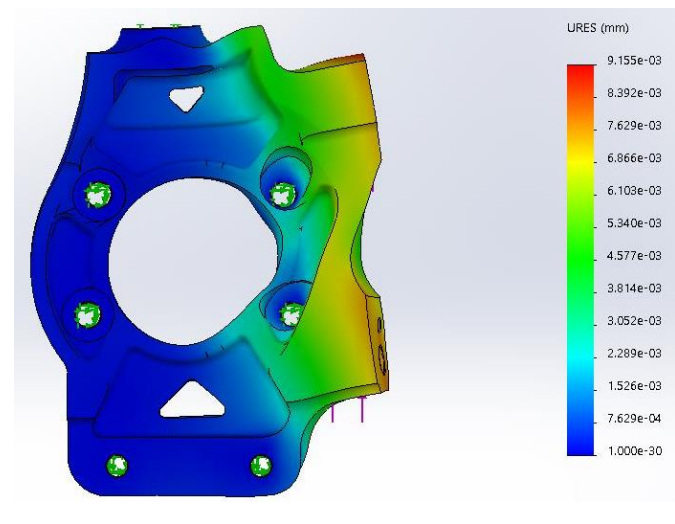
Bottom wishbone mount view



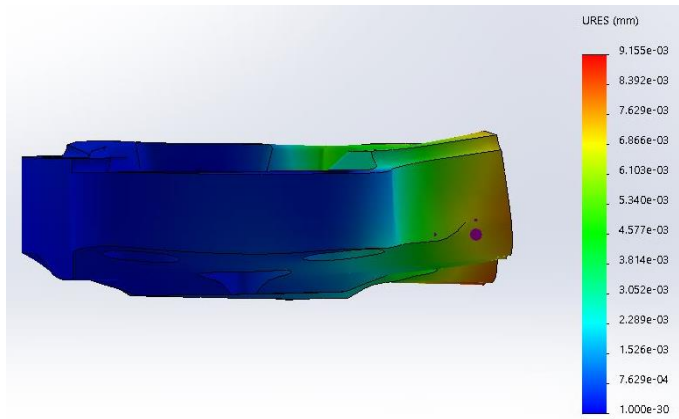
Caliper mount view



Front view

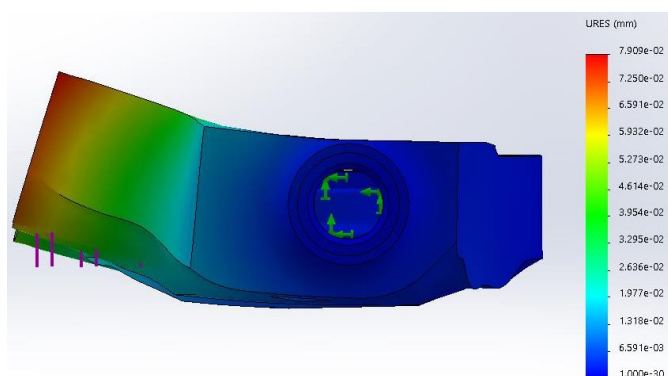
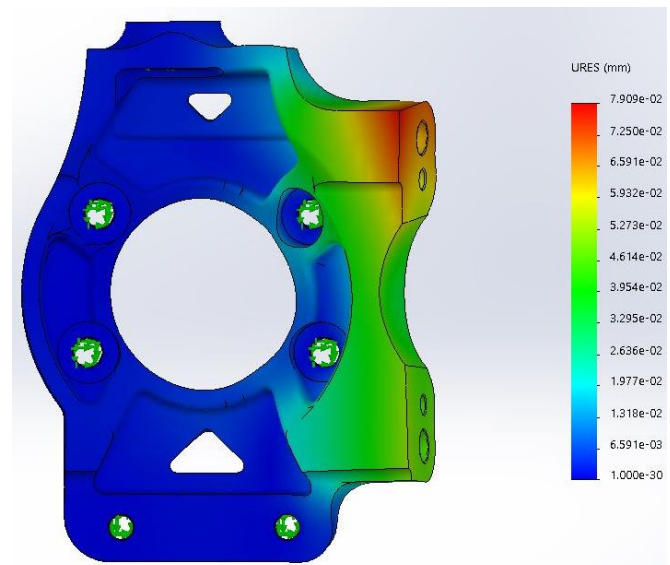
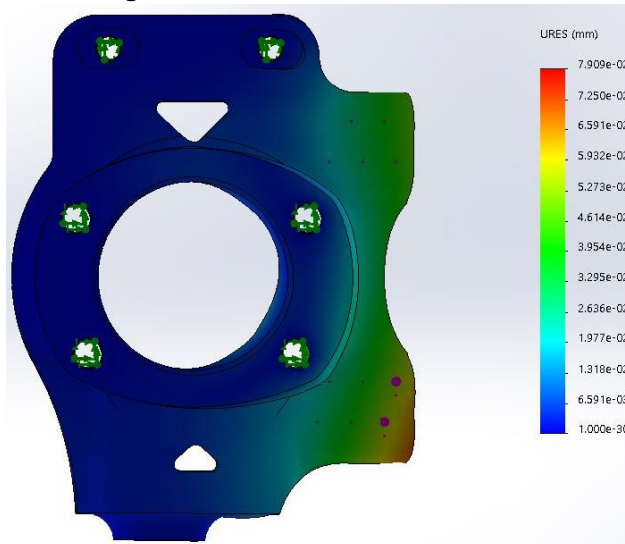


Rear view

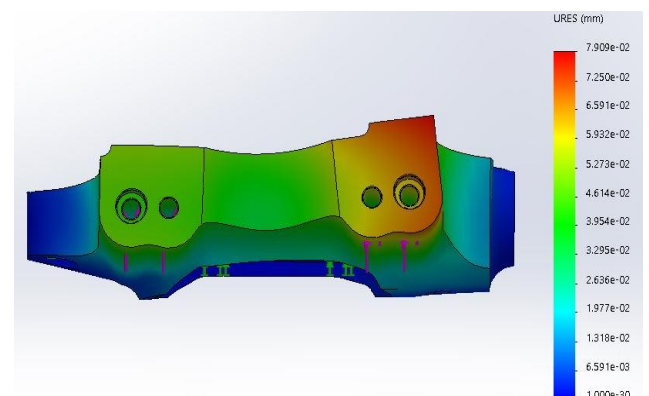


Top wishbone mount view

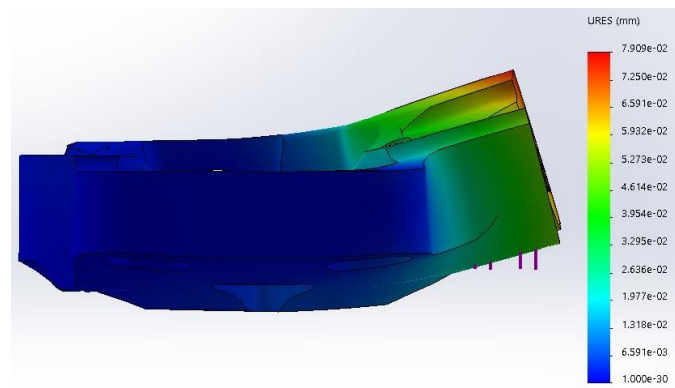
Cornering tests:



Front view
Bottom wishbone mount view



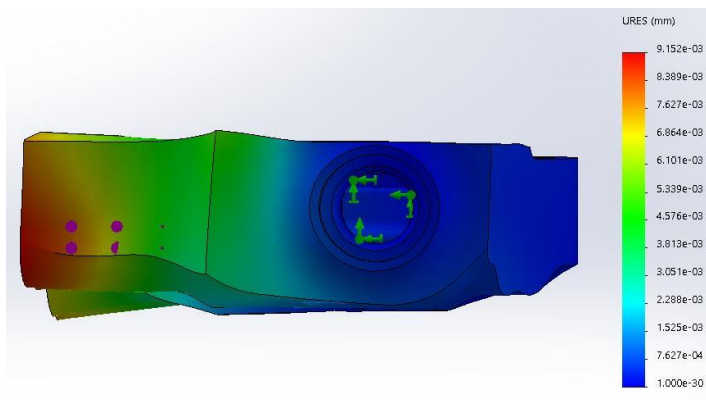
Rear view
Caliper mount view



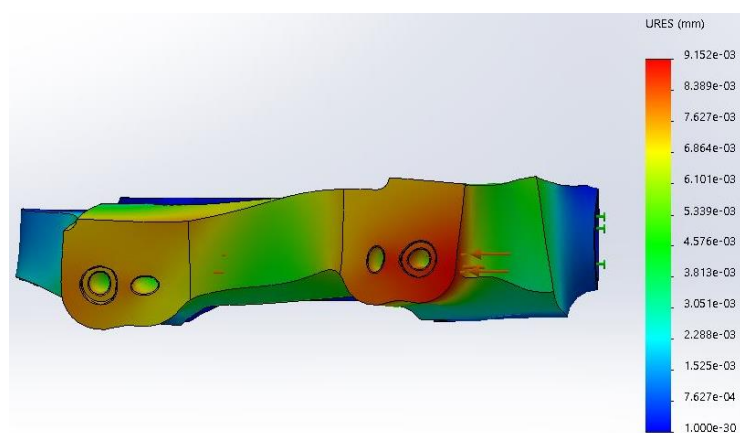
Top wishbone mount view

Design 2 with magnesium alloy:

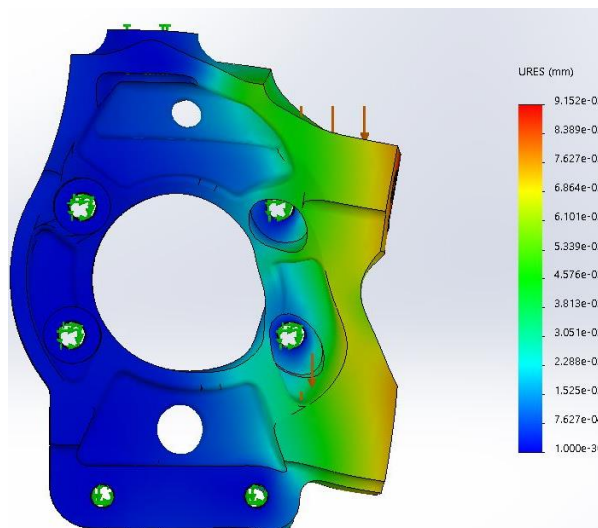
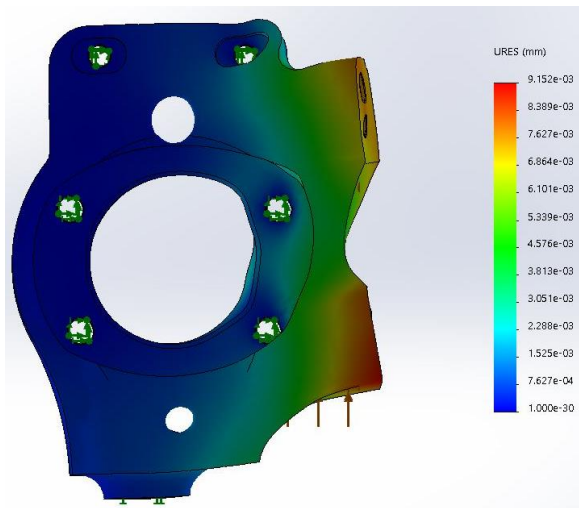
Braking tests:



Bottom wishbone mount view



Caliper mount view

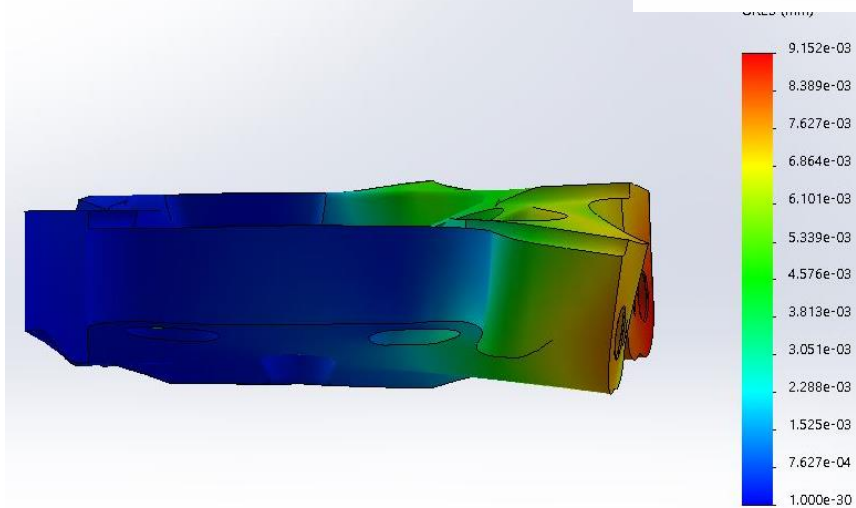
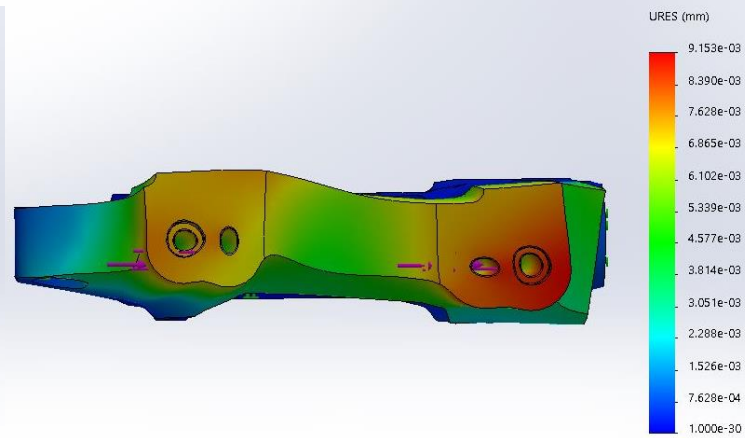
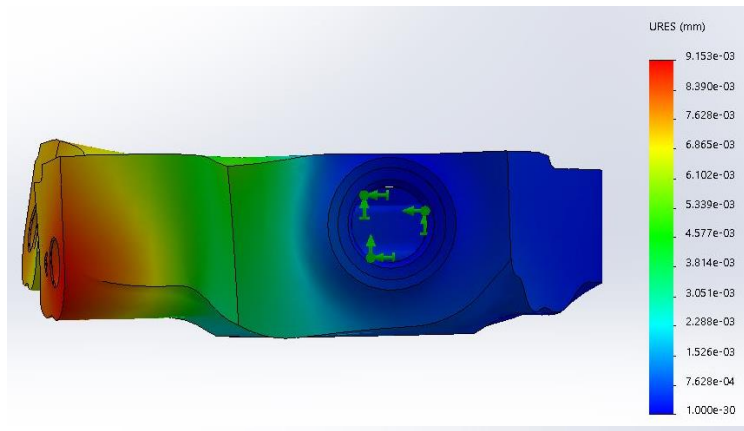


Front view

Top wishbone mount view

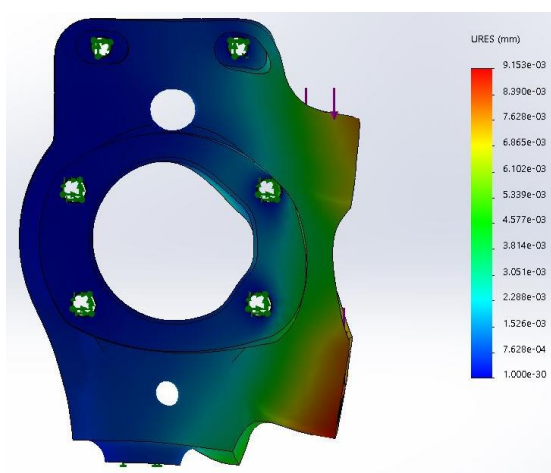
Rear view

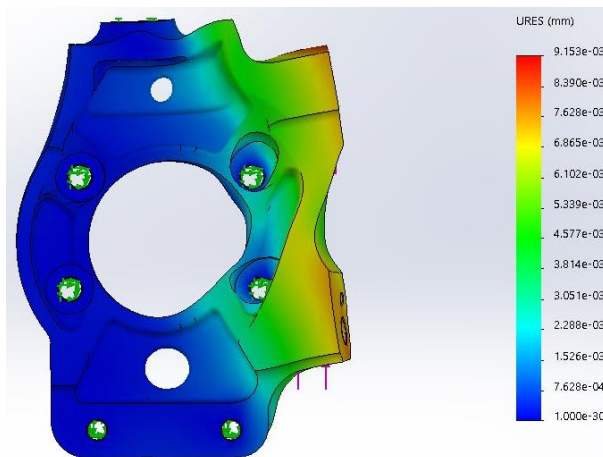
Acceleration tests:



Bottom wishbone mount view

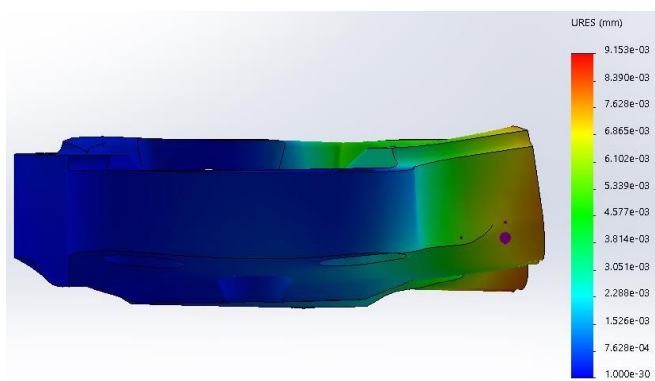
Caliper mount view





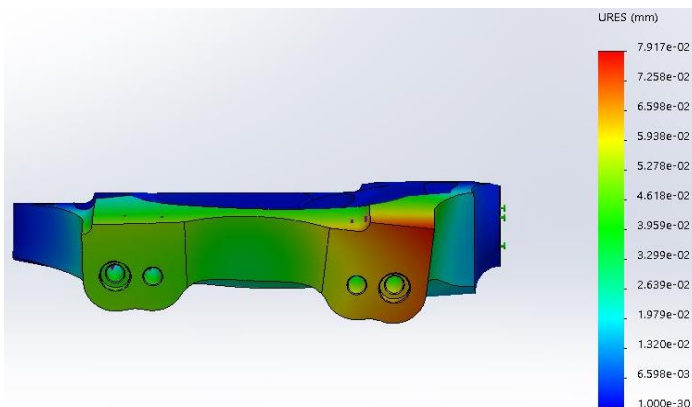
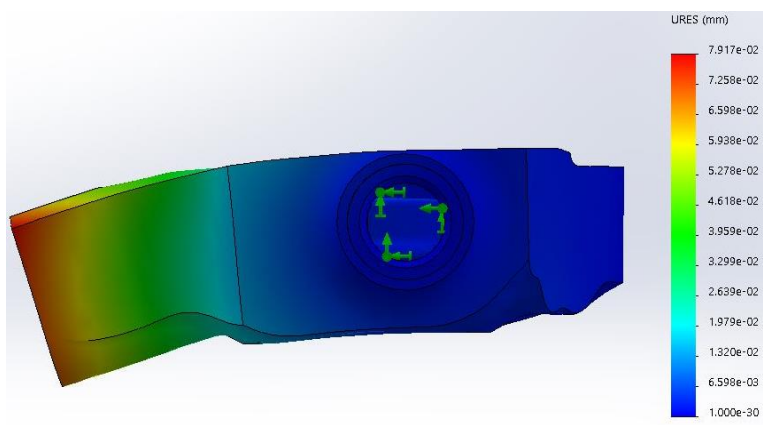
Front view

Rear view

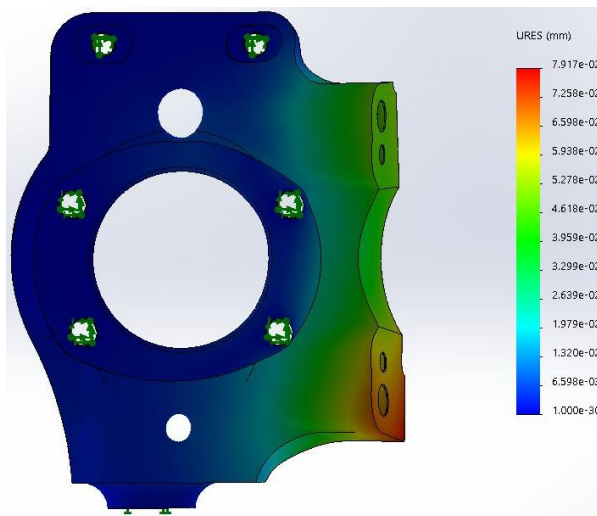


Top wishbone mount view

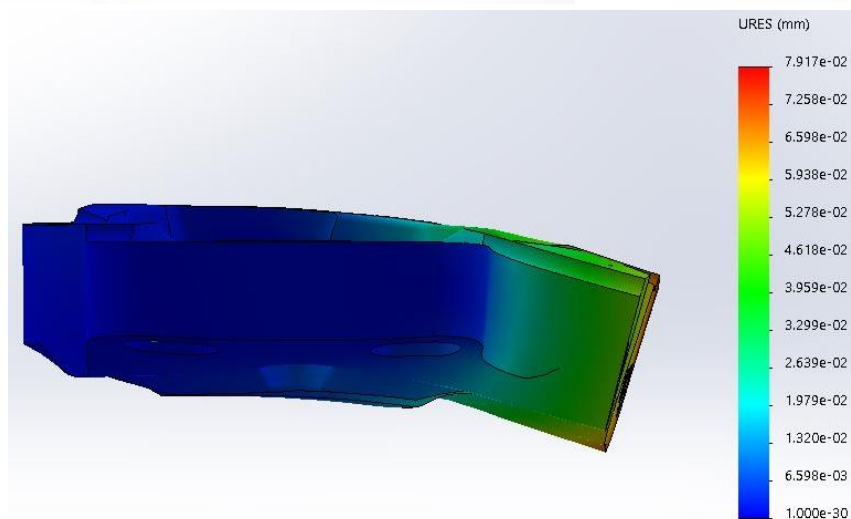
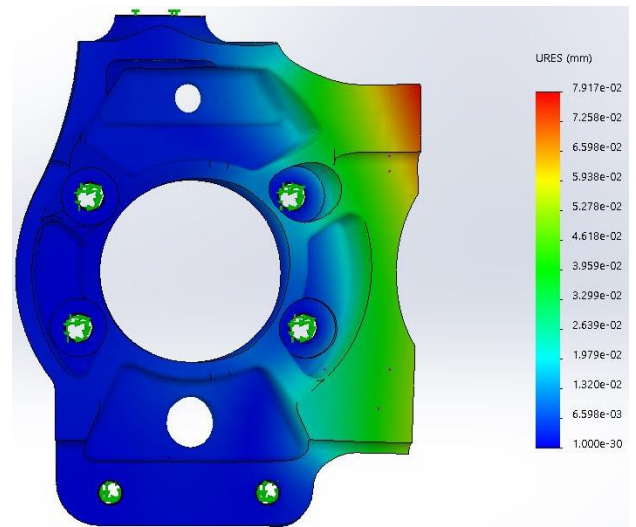
Cornering tests:



Bottom wishbone mount view
Front view



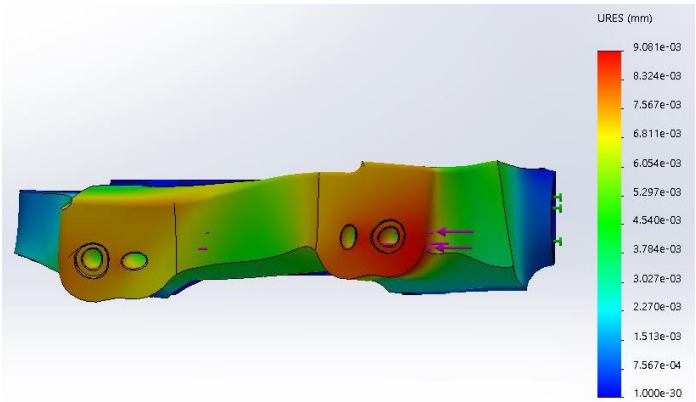
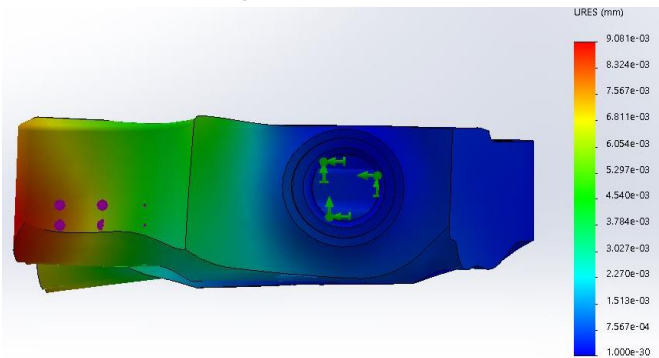
Caliper mount view
Rear view



Top wishbone mount view

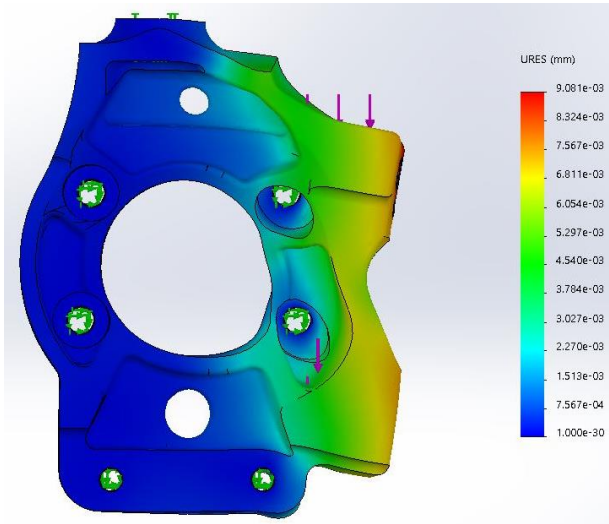
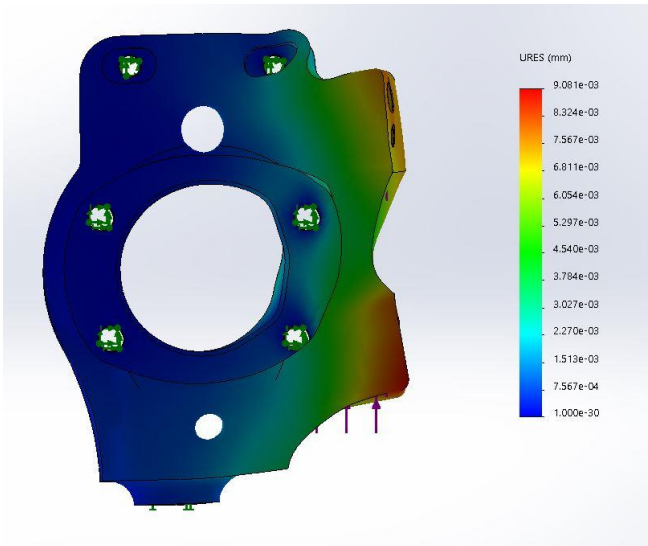
Design 3 with magnesium alloy:

Braking tests:



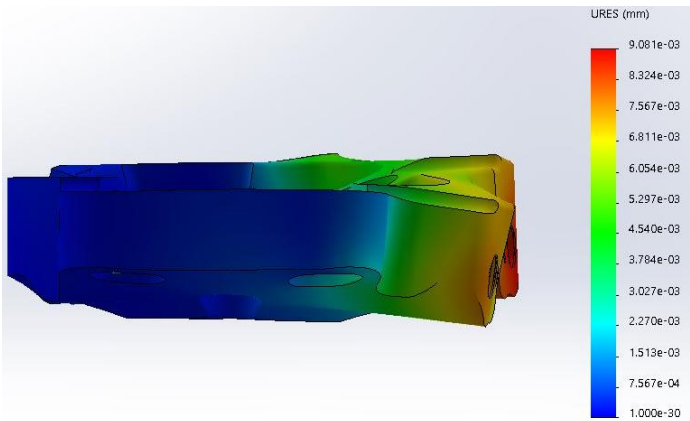
Bottom wishbone mount view

Caliper mount view



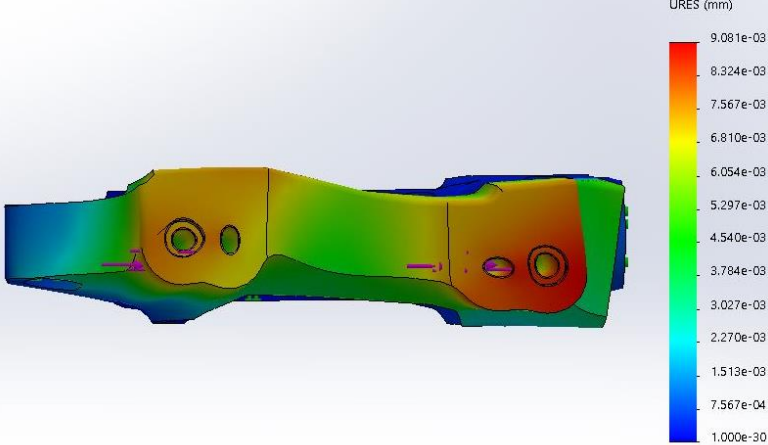
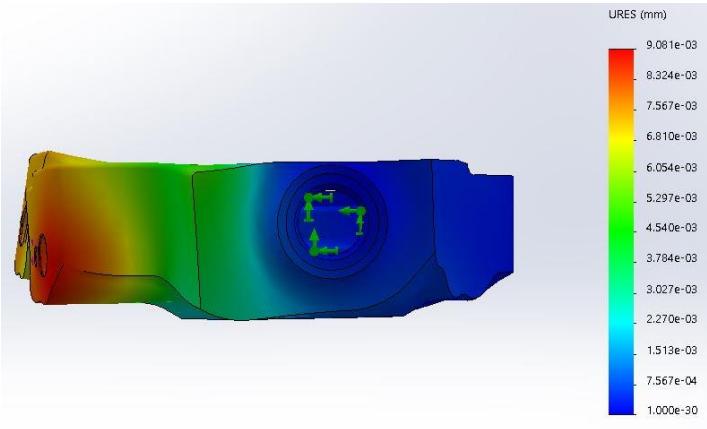
Front view

Rear view



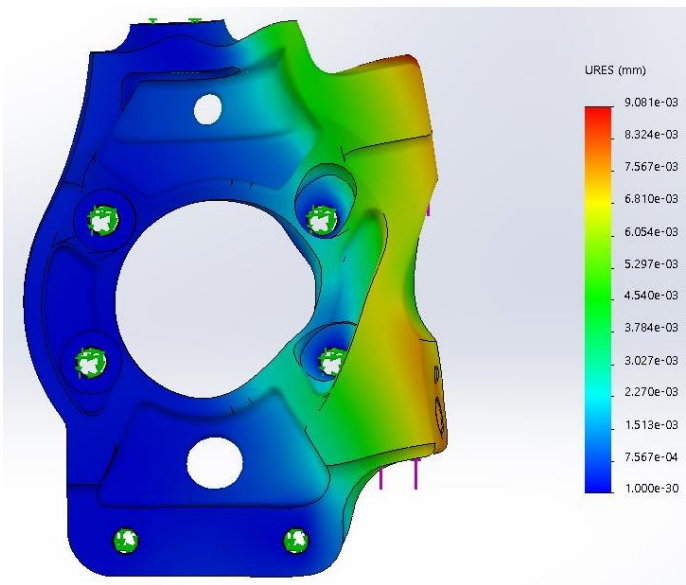
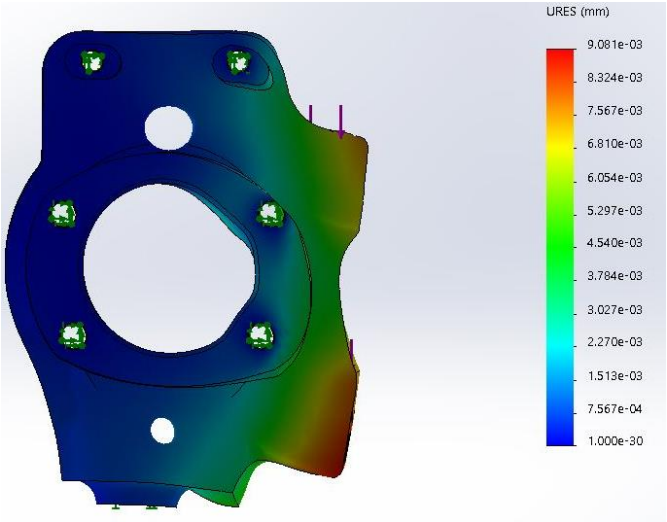
Top wishbone mount view

Acceleration tests:



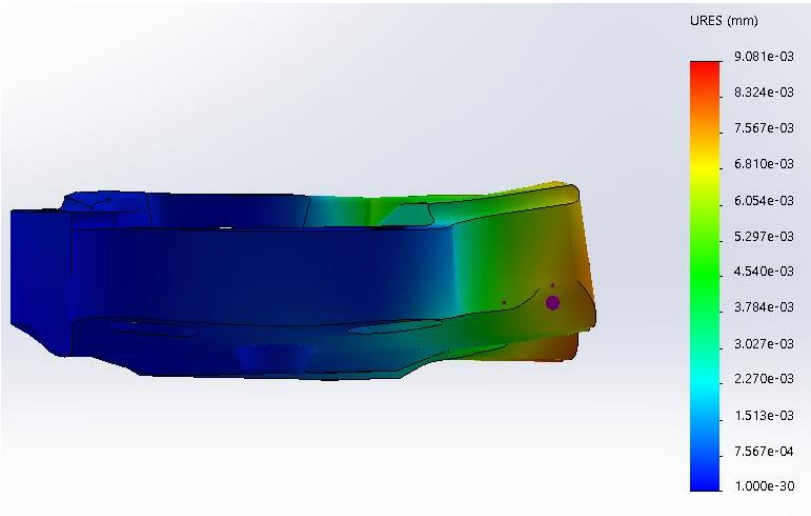
Bottom wishbone mount view

Caliper mount view



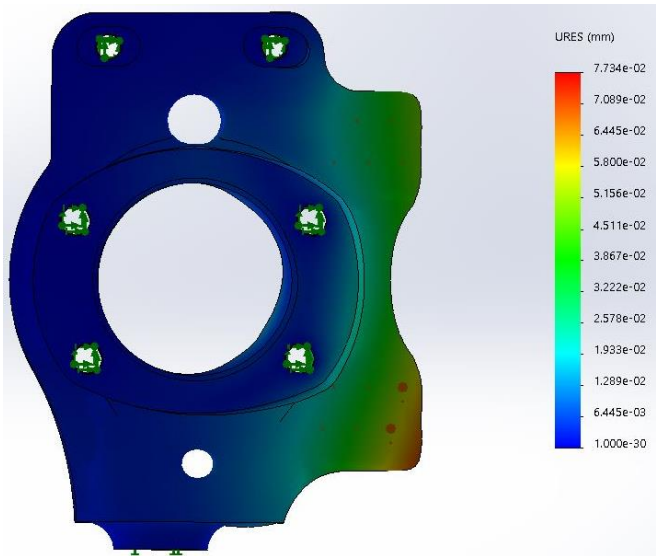
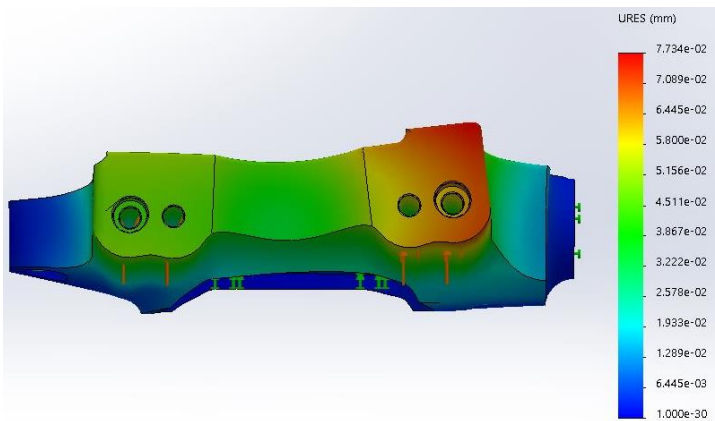
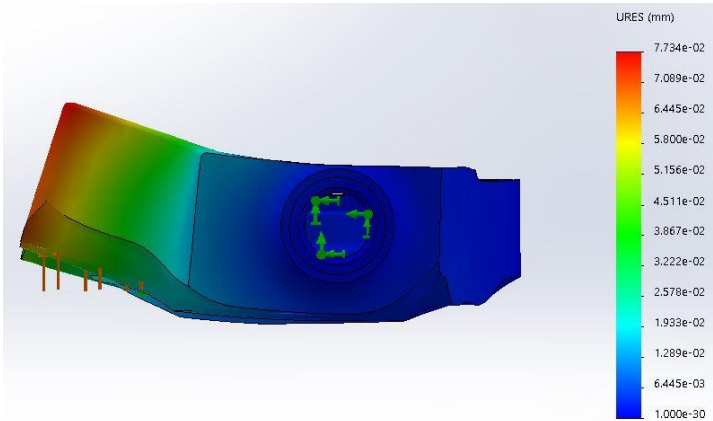
Front view

Rear view

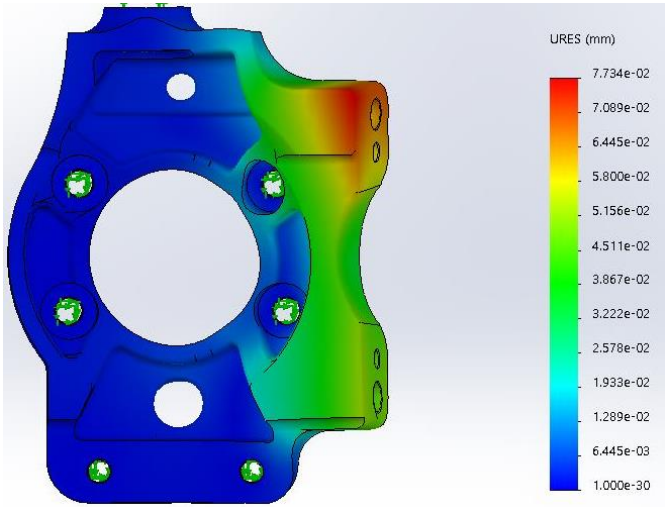


Top wishbone mount view

Cornering tests:

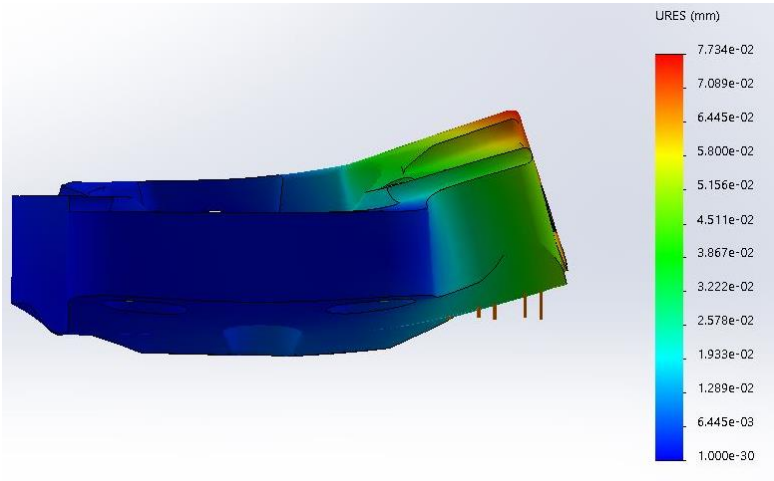


Bottom wishbone mount view
Front view



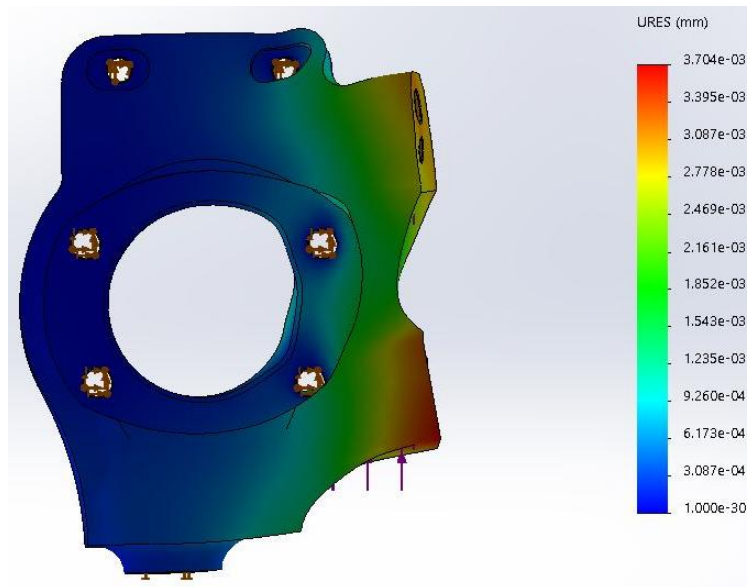
Caliper mount view
Rear view

Top wishbone mount view



Titanium

Titanium was also used as a material for the upright as it is stronger than aluminium and tests proved that, but the weight had increased drastically from 1.79 kilos to 2.97 kilos for the stock upright design. As the goal here was to decrease weight, the idea of using titanium was dropped.



Conclusion:

The tests show that magnesium is not a suitable material to be used for uprights. Even though it is much lighter than aluminium, it is less stiff making it prone to breaking under stress. Titanium is stronger and stiffer than aluminium but it is heavier, thus titanium is not suitable for this application as weight saving is the goal. Aluminium is the best option as it combines the best of both worlds, lightness and strong enough for this application.

Samuel Treleavern

This is a report of the work I carried out with a review of my work and myself. In this group project my main role was to redesign the Wishbones for the car. The four key aspect of how I carried out my work were the research I conducted to see how wishbones can be improved, the investigation into the current designs of the wishbones, the designing of adjustments and improvements and the analysis of how these would improve the overall performance.

Work Content

There are three section of work which I carried out for the project; helping evaluate the initial vehicle, rebuilding the CAD model and suggesting and designing wishbone improvements. See Appendix A for my Gantt Chart Task List.

Group Vehicle Evaluation

When the vehicle arrived in the workshop, the task of evaluating was carried out by the whole group. I specifically helped with removing and refitting the suspension components, weighing the vehicle and calculating the spring and damper rates. Removing and refitting the suspension components just consisted of unbolting components and then refitting them in the correct place when they had been measured and weighed. The suspension components analysed were the front left of the car and the rear right of the car as well as other important parts such as the steering column and the pedal box.

Once the components had been refitted, I helped the team with weighing the total vehicle. This checked the mass of the car to help with the weight distribution calculations which were carried out by Harry.

The final part of the vehicle analysis I helped with was to calculate the spring and damper rates. This consisted of removing them and evaluating their performance on the damper dyno. This data could then be used by Leo for the simulation model and Zhou for his suspension model.

Rebuilding the CAD Model

The rebuilding of the CAD model was a role I picked up at the beginning of the project. A CAD model of the car was supplied by Zentrack, but it was largely inadequate. It had many issues; all the components were fixed and so didn't move as the model should. This was due to the model being made up of sub-assemblies imported into other assemblies. This meant that data couldn't be collected for the simulation model created by Leo and other data couldn't be collected by the rest of the group. For this reason, I took

on the role of rebuilding the whole model to allow my team to be able to complete the project.

I started rebuilding it by renaming all the individual parts supplied to us. The names they had were the basic files name, making it hard to know which part was which when using the model. The renaming process also made it easier when I imported the masses of each part.

The next stage was to remake assemblies as parts and importing them into a new assembly to create one big assembly of the car. This was done because if a sub-assembly was imported into an assembly its parts stay fixed, for example this was the case for the anti-roll bar drop link, originally the ball joint wouldn't move but after being made as separate parts and imported it move and functioned as it should.

The final stage for me was to add the weights we measured as a group. This allowed the model to be correctly analysed and stress tested. Now, the team could use the model to collect any of the data they needed.

Another team member, Harry, made new parts for weight distribution and added them to the model.

Wishbone Improvements

The wishbone improvements were my main task for the project. I worked independently to understand, research, design, and evaluate the wishbones for the Zentrack car.

The first step was to evaluate the design of the current wishbones. For this I took the bottom wishbone for the front of the car and went to carry out a stress test on it.

Unfortunately, it failed due to the wishbones being made up from several parts which didn't fit together correctly so interference was occurring so the wishbone wouldn't mesh.

To overcome this issue a lot of time was spent rebuilding them so there was no interference whilst remaining accurate models of the original wishbones. This meant several measurements had to be made as well as angles measured to ensure they were correct. To make sure the wishbone mated correctly most of its parts needed remanufacturing so the pipes would fit flush against each other.

Once this was finished, I could move onto designing improvements for better performance. This consisted of researching what the best materials and shapes for track car wishbones were and where the location of the pick-up points should be. Further research was carried out to consider the manufacturing limitations of small-scale production. Once the research had been carried out, I could apply it to the design of the wishbones.

Once this design was finalised, I could move onto analysing the new model and hopefully building a prototype. Unfortunately, due to time restraints, the prototype wasn't created.

Research

Research is a key element for this project as the main goal is to understand new things working both individually and as a team. Online research was the most common form I used. Research was a very important part for the wishbone task, it was used in developing the current wishbone designs, learning how to carry out stress analysis, finding alternative materials to make wishbones out of and figuring out the manufacturing processes and limitations of small-scale productions. Websites such as the *FEA Analysis of Double Wishbone Suspension System* in Blogspot (Albuquerque, 2018) helped in my research to broaden my knowledge of how wishbones work and how I could apply the FEA stress test to the wishbone CAD model. Another form of online research which heavily influenced my project was the use of journal articles such as the *Finite Element Analysis and Topography Optimization of Lower Arm of Double Wishbone Suspension Using Abacus and Optistruct* in the International Journal of Engineering (Darge *et al*, 2014). Journal articles like this one gave me solid knowledge to further my understanding of how to apply forces affecting the wishbone and aided my understanding of carrying out the FEA stress analysis.

I also accessed a lot of technical reports online which carried out similar procedures as I was planning. Reports like the research paper on *Design and Analysis Double Wishbone Suspension System using Finite Element Analysis* by Thacker and Bhatt (2015) helped me with the general design ideas but also to find more information on different materials which could be used for the new wishbones. Papers like this one helped me get an idea of issues which may arise and to strengthen my understanding of wishbones.

Another helpful research function was the Solidworks help function and tutorials (SolidWorks, 2018). These allowed me to teach myself how to carry out the FEA stress analysis as well as how to interpret and understand the results. This allowed me to then apply what I learnt to this project.

Investigation

Investigations were only used in my wishbone task. The investigation carried out were FEA stress analyses. Two types of investigation were carried out to see how the wishbones would act under braking and bump force. These were carried out using Solidworks simulation study, the forces applied were 6300N and 10962N. For the braking force the figure 6300N came from the maximum deceleration, 12m/s^2 , given by another team member, Guo, and multiplied by half the mass of the car, 350kg. then finally multiplied by 1.5 to account for extra force for safety as you don't want it snapping.

$$\text{Max Deceleration} \times \frac{1}{2} \text{mass of car} \times 1.5 = \text{braking force}$$

$$12 \times \frac{700}{2} \times 1.5 = 6300\text{N}$$

The force for bump, 10962N, came from the maximum bump force, 7308N, given by a different team member, Zhou and this was multiplied by 1.5 as well.

The results are as follows.

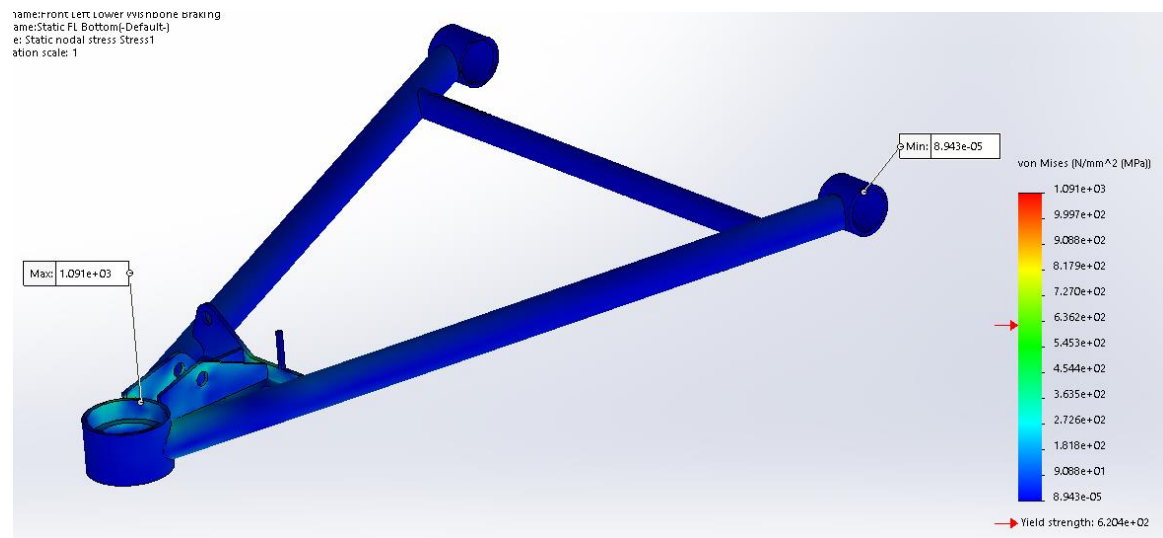


Figure 1 - Bump Stress Results

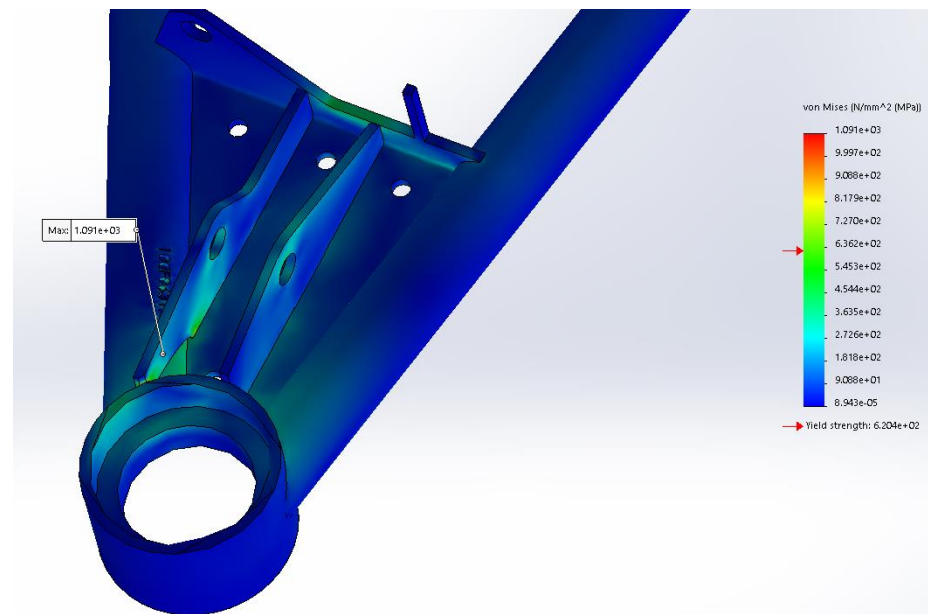


Figure 2 – Bump Stress Results 2

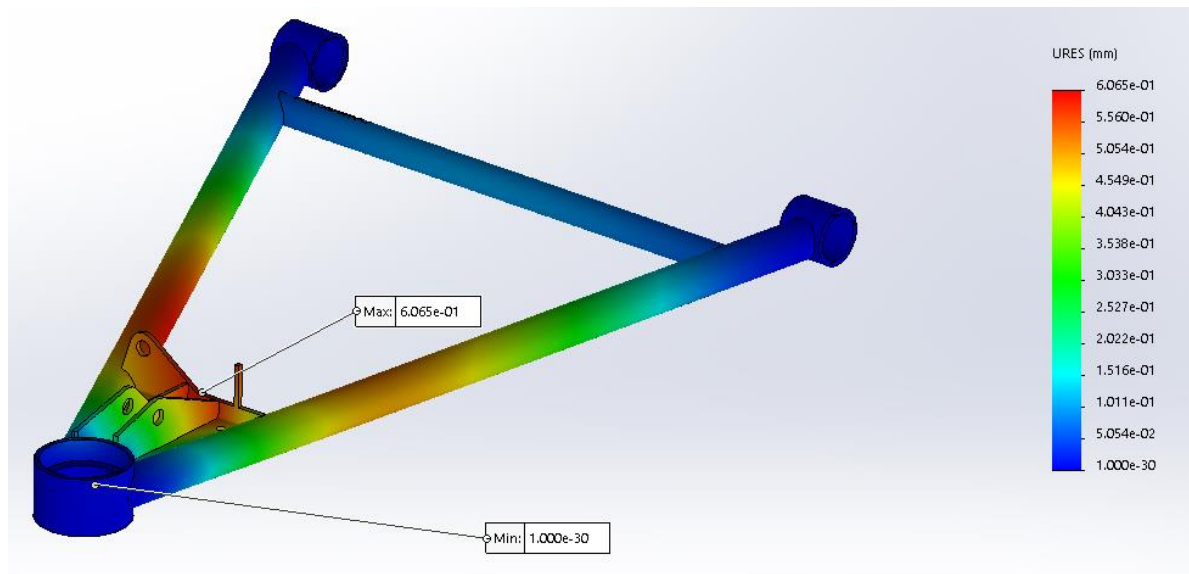


Figure 3 - Bump Displacement Results

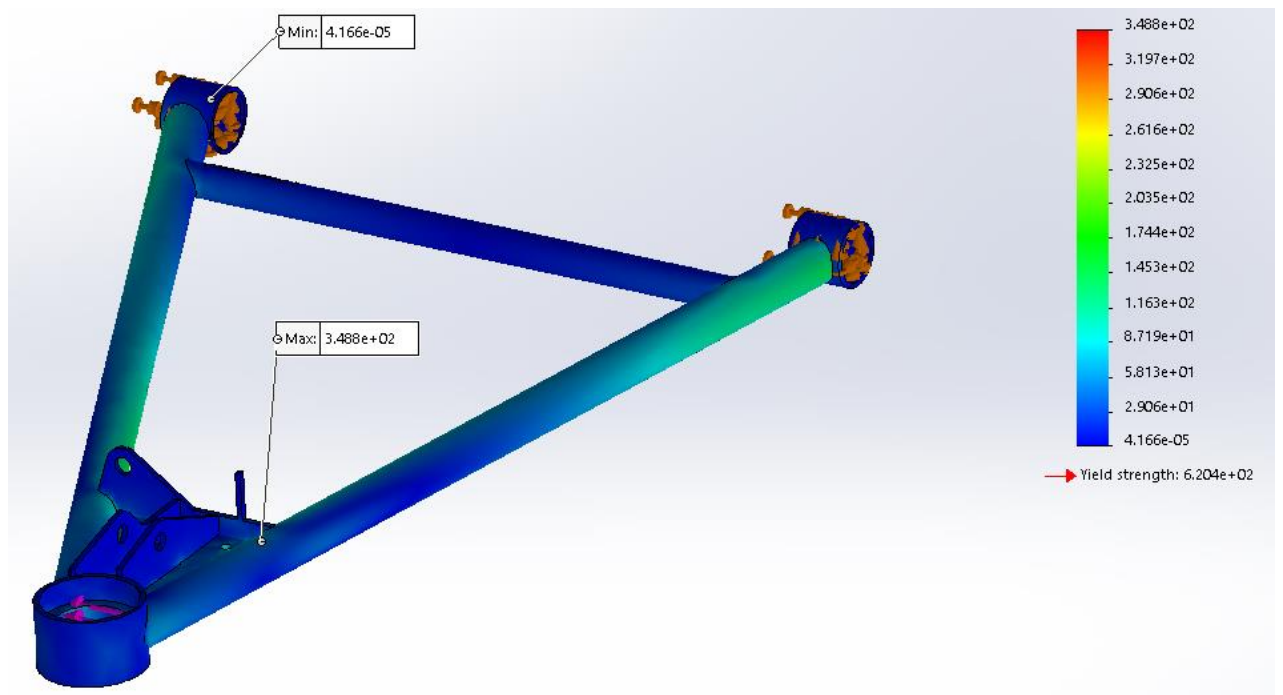


Figure 4 - Break Stress Results

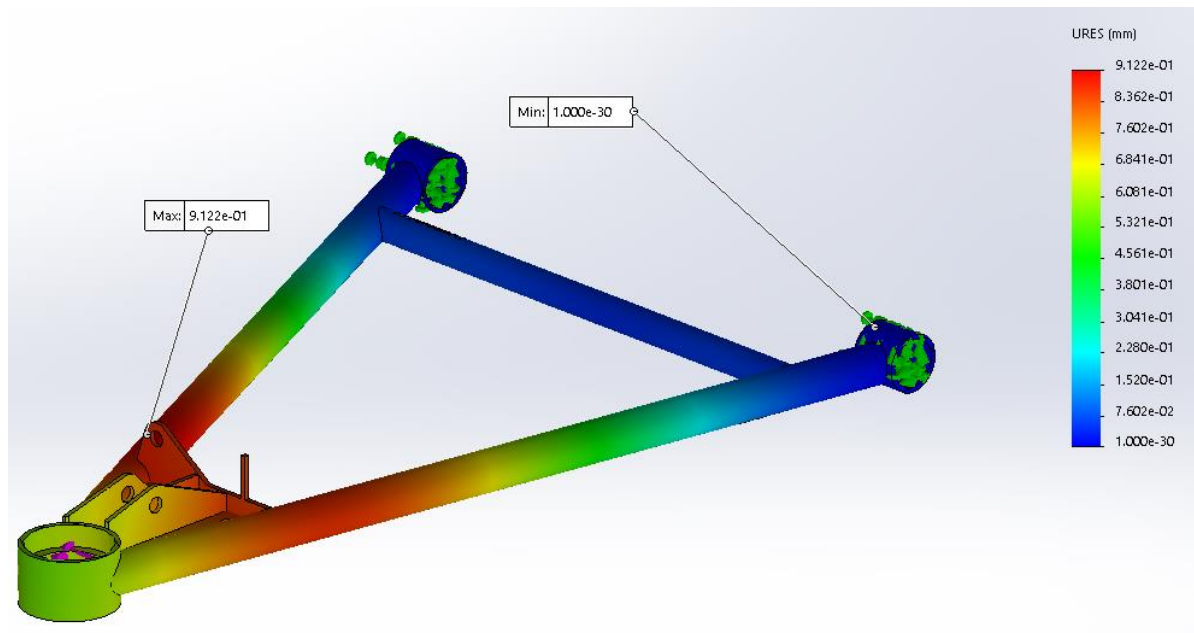


Figure 5 - Break Displacement Results

As you can see the wishbones are currently well built with no overly weak points that will break. For the braking test the max stress didn't exceed the yield strength, as you can see in figure 4. More images of the bump stress test can be found in Appendix B. However, the yield strength, $6.204 \times 10^2 N/mm^2$, is exceeded, for the bump test, with a max stress of $1.091 \times 10^3 N/mm^2$. This can be seen in figure 1. You can see in figure 2 where the weak point is, and this point is where reinforcement needs to be made. There is excess material that doesn't get effected so this could be removed to reduce mass. Another solution could be to make them out of a different material which will reduce mass but retain the current rigidity. The reduction in mass will benefit the cars setup as it'll reduce both the sprung and unsprung mass.

Design and Analysis

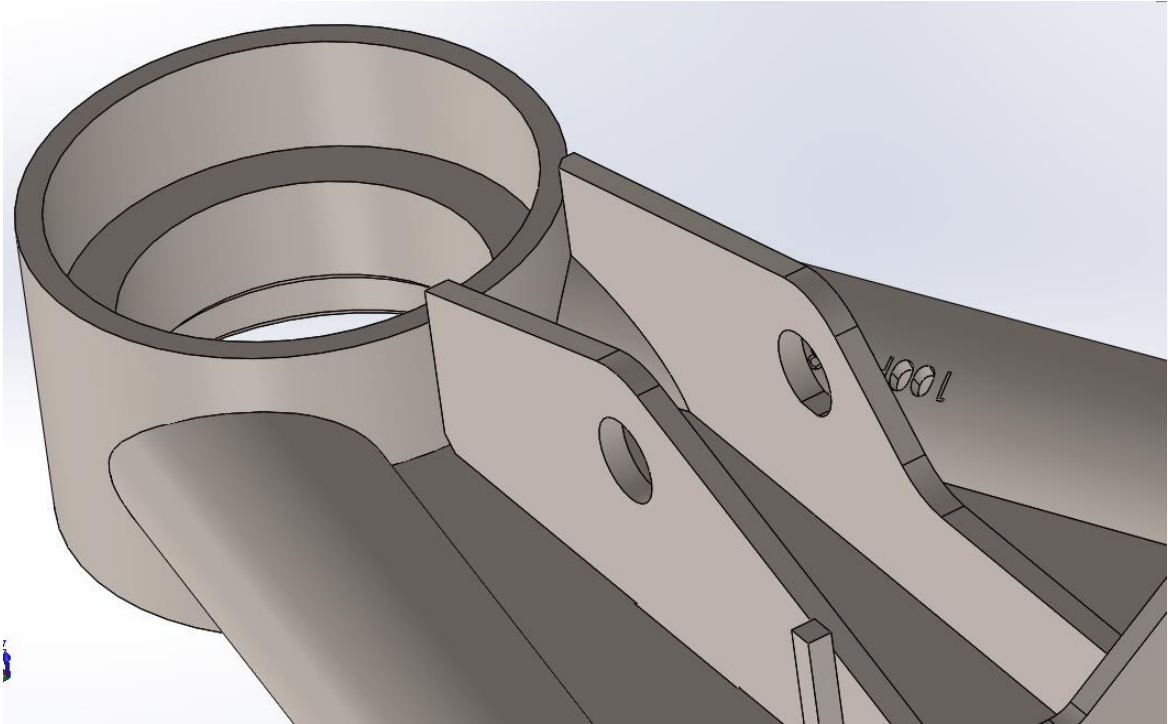


Figure 6 – New Part Close Up

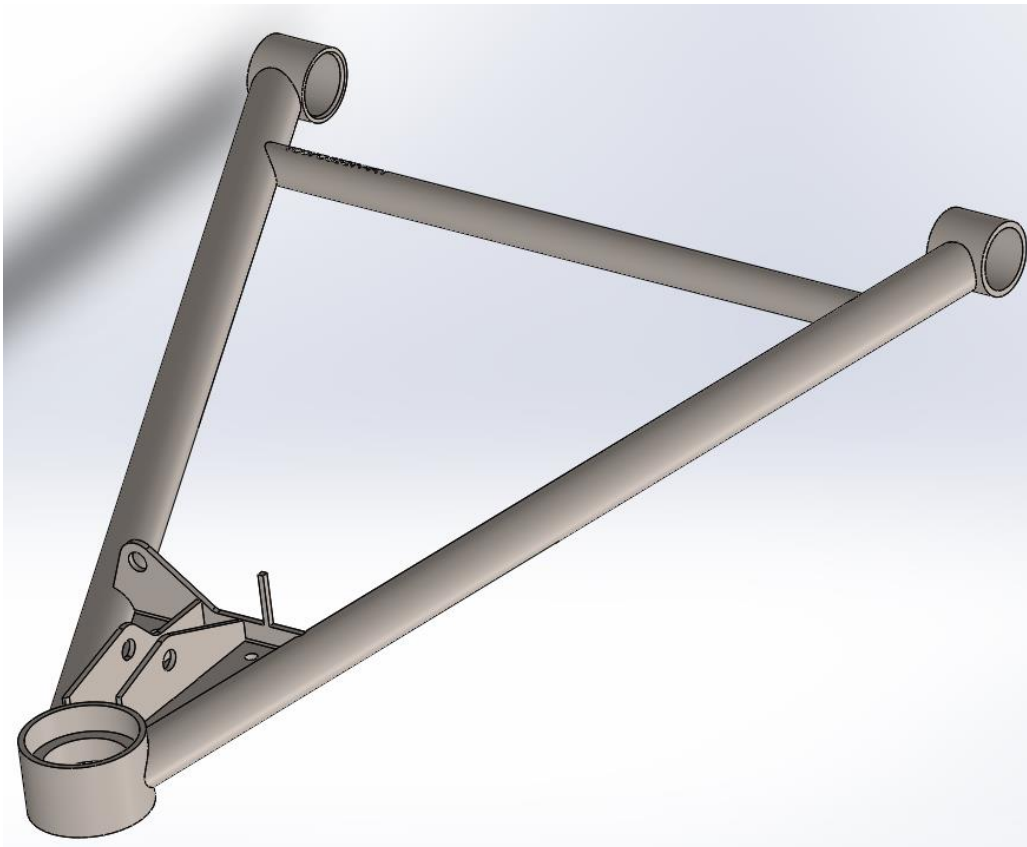


Figure 7 – New Design Screenshot

The new design wasn't very different to the original, as can be seen in figure 7, as the original was a well designed wishbone. The improvement was done by making one of the push rod mounting brackets slightly longer with more material so it would spread the load across a larger area, see figure 6, thus hopefully reducing the maximum stress to less than the yield strength.

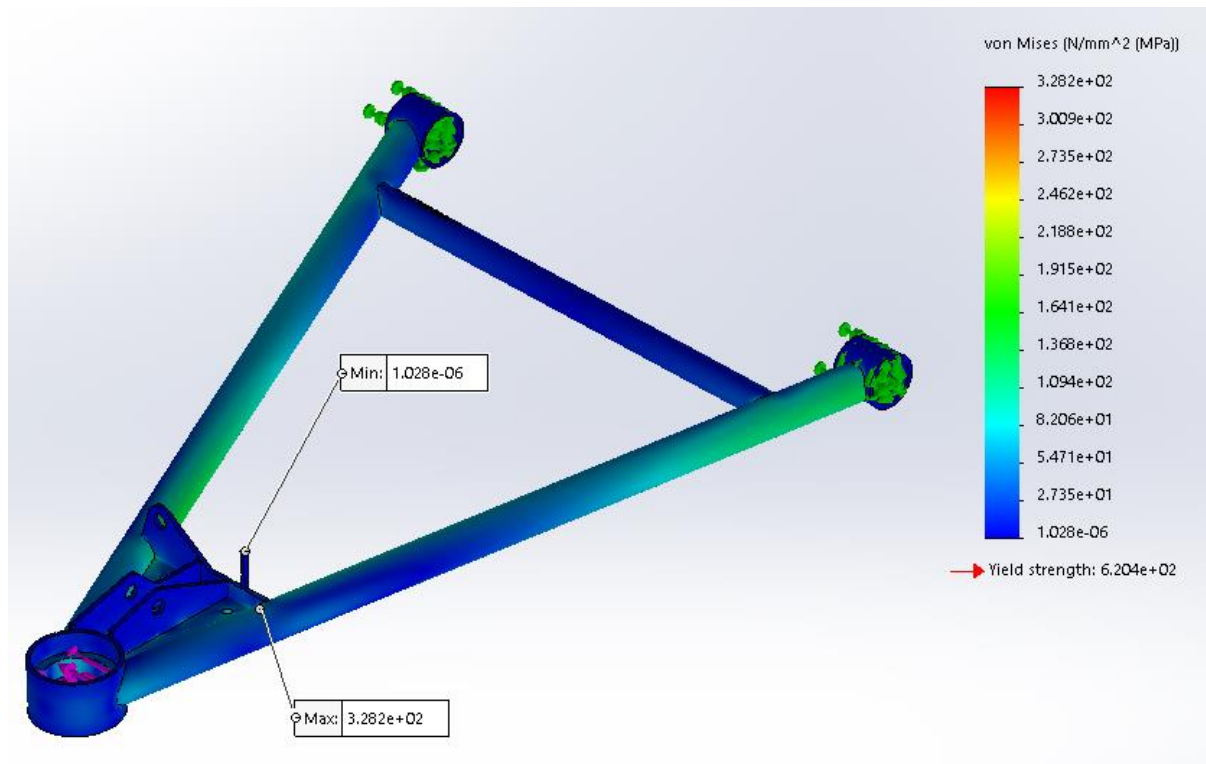


Figure 8 - New Braking Stress Results

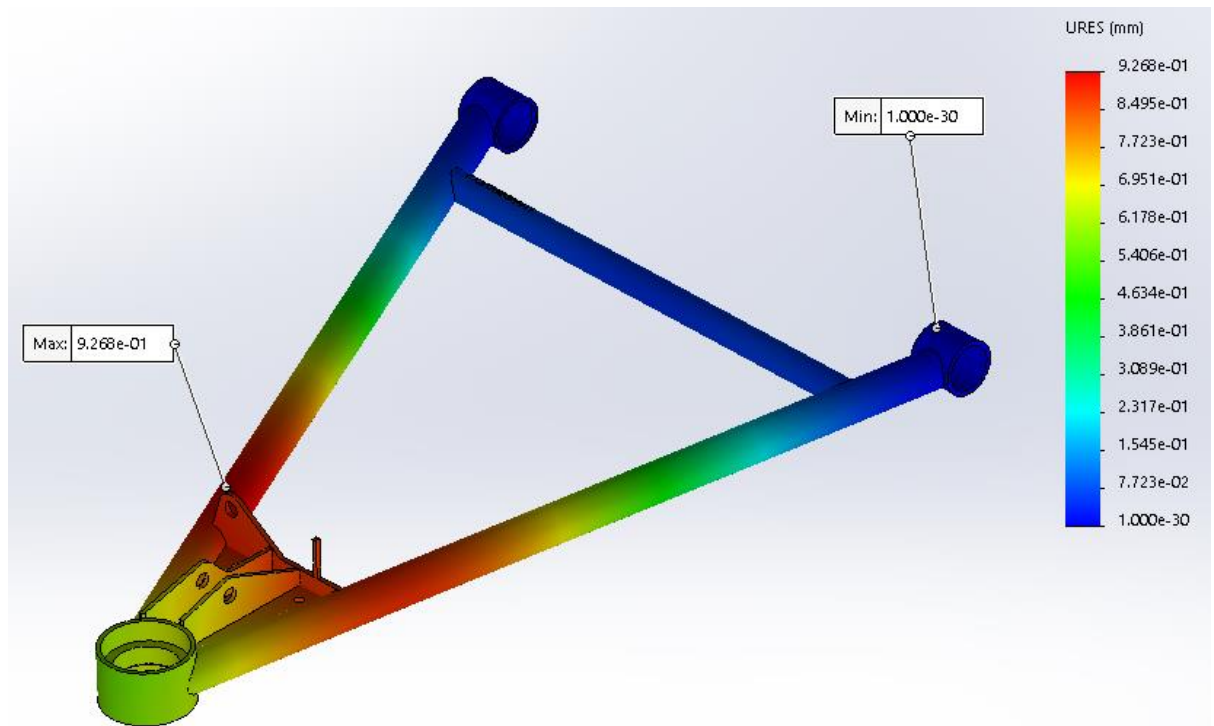


Figure 9 - New braking displacement

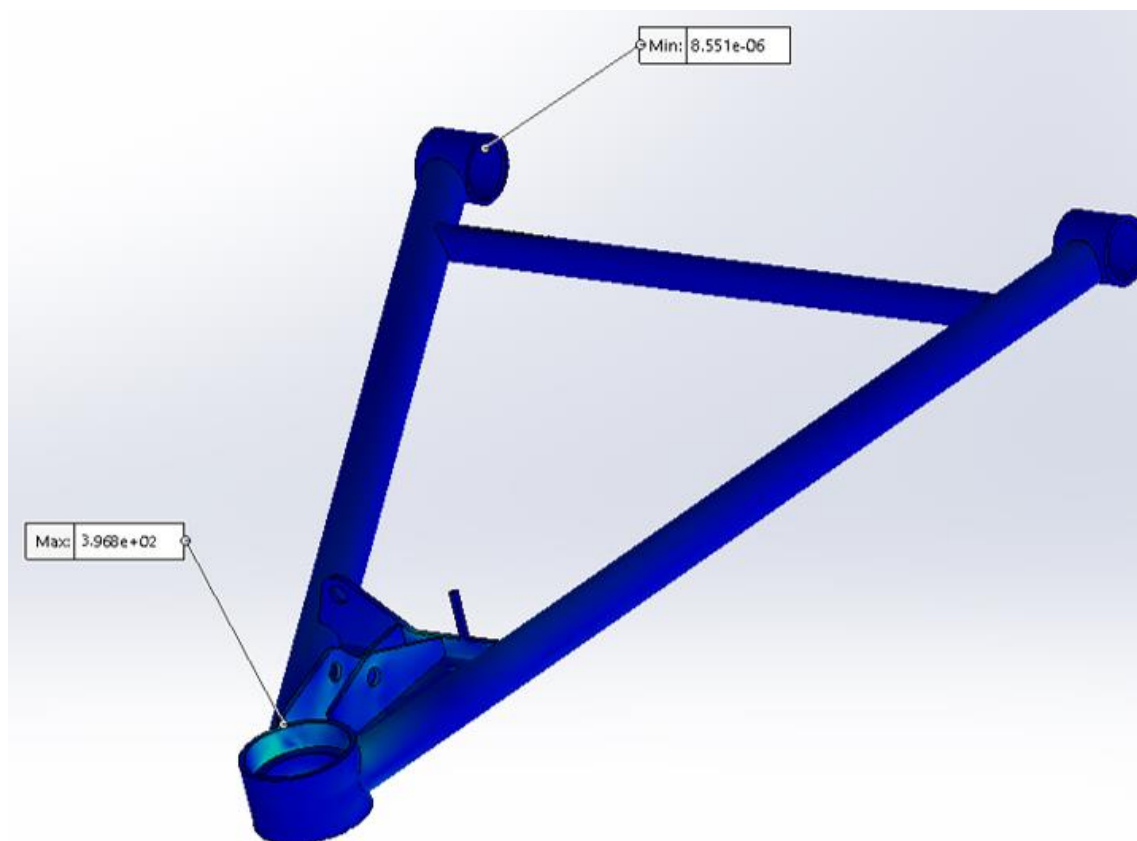


Figure 10 - New bump stress

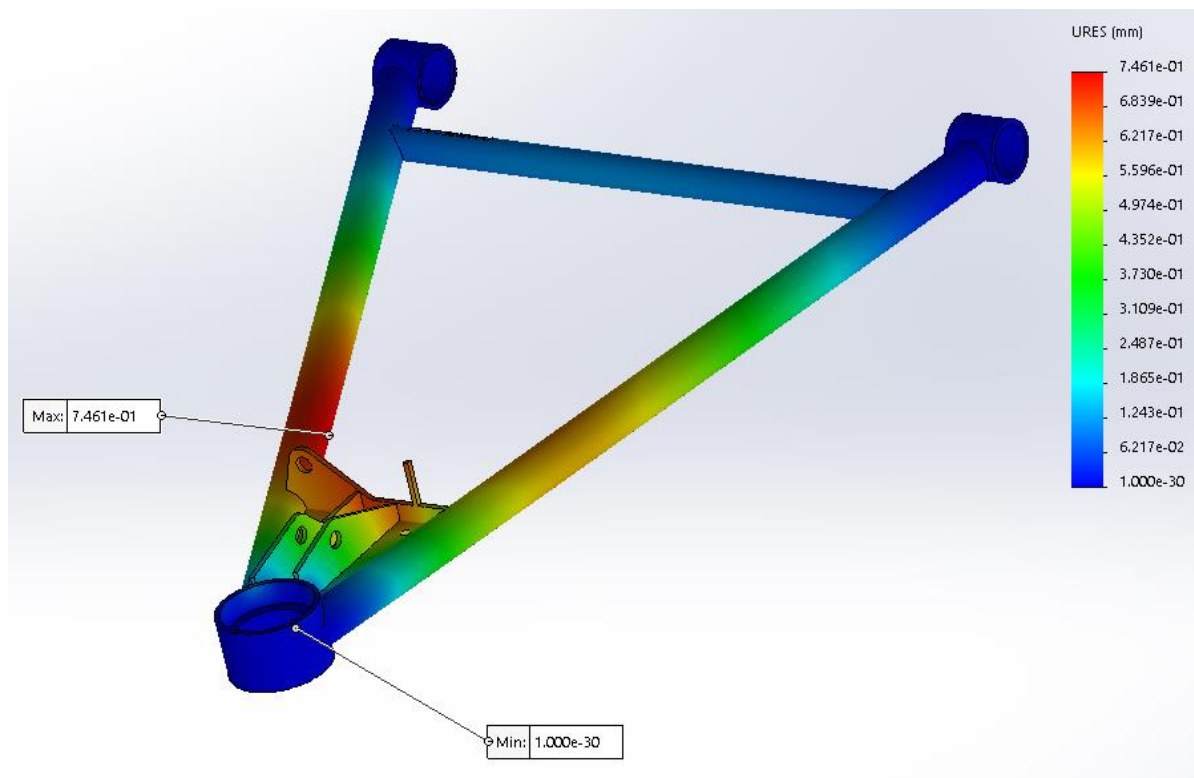


Figure 11 - New bump displacement

The new wishbone design responded much better than the old one. This was due to the maximum stress for both the braking and the bump test not exceeding the yield strength. In fact, it was nearly half the value at $3.282 \times 10^2 \text{ N/mm}^2$ for braking and $3.968 \times 10^2 \text{ N/mm}^2$ for bump. This means the new wishbone design will be able to withstand all the forces its expected to see for normal competition use and won't break. The point of maximum displacement also moved for the bump test, away from the pushrod mounting plate towards the front tube of the wishbone. For the brake test the maximum displacement stayed in a similar place, which was expected.

Review

As previously mentioned, the newly designed wishbones weren't too different to the originals. This was expected as the originals are most likely lotus wishbones so would have been thoroughly tested and analysed to high standard. Further improvement could be made if the wishbones were made in house rather than outsourcing. Making them out of Chromium Molybdenum Steel (Chrome Moly) would reduce the mass as well as retaining, if not improving, the rigidity of the wishbones. Other materials could be considered if money was no object or if there was a larger budget, such as carbon fibre. However, the customer made it clear that they wanted everything done as cheap as possible so this was not investigated as this would have been an uneconomical use of time during the project. With a no limit budget, a composite material would be a good starting point for considering a stronger material.

During this project the original time plan was not kept to. This was due to underestimating how long jobs would take as well as not accounting for problems and issues that could arise. Throughout the project there were several time consuming problems, these all related to the CAD. Originally it took a long time to receive the CAD of the car from Zentrack delaying our whole start to the project. When it was finally received, it was incomplete and badly put together. This could have been due to the conversion process between different software's, from CATIA to Solidworks. Because of this, I spent time at the start of the project rebuilding the complete CAD model instead of investigating the wishbones. Once the CAD model was completed the wishbone analyses could begin, however, another issue arose where the CAD of the wishbones wasn't created and mated correctly meaning more time had to be spent on rebuilding these. These problems resulted in the whole project being delayed and the prototype work for the wishbones couldn't be carried out in the timeframe set out. If this could have been done this would have been a good way of verifying the CAD model and the accuracy of the stress test. This would have then allowed further designs to be made to make a final design to present to the customer.

Zhangwei Guo

Abstract.

The goal for the group project is to evaluate and upgrade the Zentrack (Figure 1) which is a civilian vehicle to make it possible to participate even win the champion in a future competition.



Figure 1. Zentrack

When it comes to my personal part, I am responsible for testing and improving the braking system for the Zentrack from the braking pedal, cylinders, pipes, calipers, pads to disc. During the process of the research, I have used some principles from *Principle of automotive* (Yu Z. , 2009) as well as equations from *Analysis of the braking process of the front and rear brake braking forces with fixed ratios on various road surfaces* (Yu C. , 2016).

Made the calculation in the Excel after finishing the built of model in the Solidworks.

I have found that the existing braking system (Figure 2) is different from any other civilian vehicles on account of the strong brakes after the evaluation. They are quite suitable for common competitions but I still need to make the upgrades to ensure they are much stronger so that the driver could start braking as late as possible.



Figure 2. Brake

Personal Roles

For the braking system, first thing I had to finish is to split the tasks into Measure Vehicle, Create 3D model, Make Calculation, Search and Decide how to improve, Find suitable components for the new system and Evaluate the improvement. After that, I made a comprehensive measurement for each part in the existing braking system and then use 3D modeling software like Solidworks to build the models like discs(Figure 3). After that, I can make some simulations on CAD (Computer Aided Design) by these models and they can give me a possible upgrade optimization plan with the help of the literatures and magazines found. Finally, I have

to finish the calculation in Excel (Figure 4) to make sure that the new braking system is well enough for the competition.

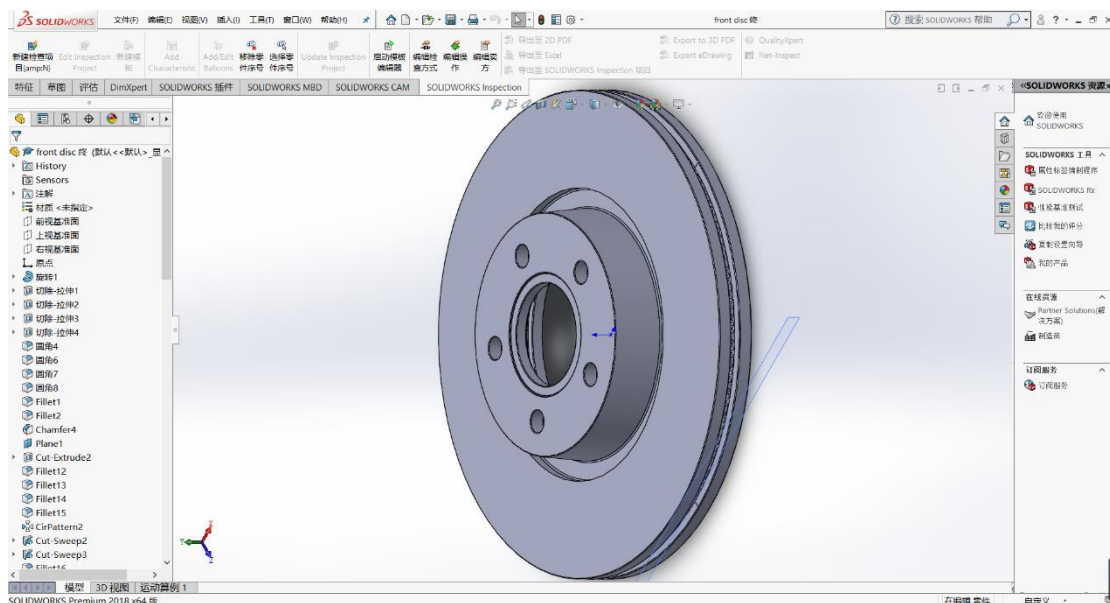


Figure 3. Disc in Solidworks

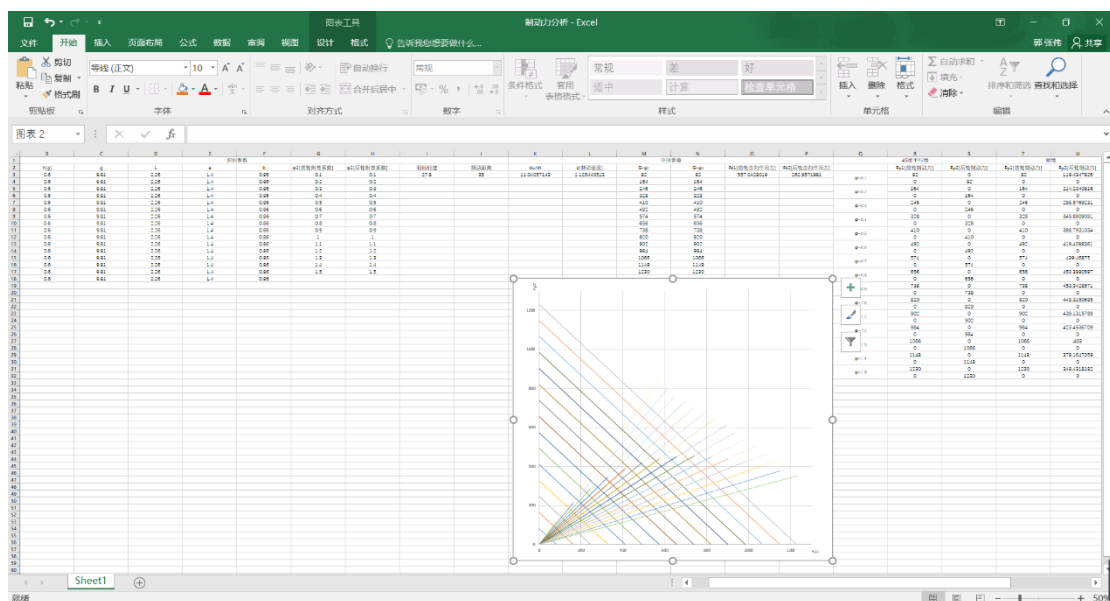
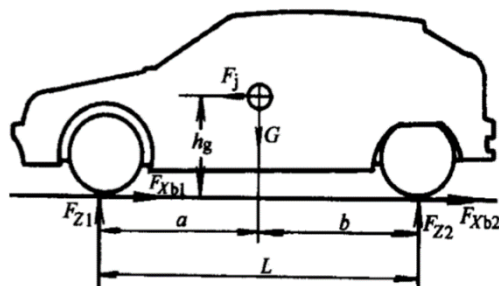


Figure 4. graph in Excel

Principles of Researches

Before analyzing the braking force distribution ratio, it is necessary to understand the opposite force acting on the front and rear wheels during braking.



(Yu Z. , 2009)

This picture shows the force of the car when it brakes on a horizontal road. The figure ignores the rolling resistance moment of the car, the air resistance and the moment of inertia generated when the rotating mass decelerates. In addition, the analysis does not take the process of rolling the wheel while braking into account and the adhesion coefficient only takes a fixed value.

According to the figure, fomula was got when the torque is obtained for the grounding point of the rear wheel as follows:

$$F_{z1}L = Gb + m \frac{du}{dt}h_g$$

(Yu C. , 2016)

In the equation, F_{z1} is the opposite force of the ground to the front wheel; G is the gravity of the car; b is the distance from the center of mass of the car to the centerline of the rear axle; m is the mass of the car; h is the height of the car's centroid; du/dt is the deceleration.

Take the torque on the grounding point of the front wheel to get the following formula on the same way.

$$F_{z2}L = Ga - m \frac{du}{dt}h_g$$

(Yu C. , 2016)

In the formula, F_{z2} is the normal reaction force of the ground to the rear wheel; a is the distance from the center of mass to the center line of the front axle.

If the brakes are applied on the road surface with different adhesion coefficients and the front and rear wheels are locked, the opposite forces acting on the front and rear wheels are like these:

$$\begin{cases} F_{z1} = \frac{G}{L}(b + \frac{h_g}{g} \frac{du}{dt}) \\ F_{z2} = \frac{G}{L}(a - \frac{h_g}{g} \frac{du}{dt}) \end{cases}$$

(Yu Z. , 2009)

When the front and rear wheels are locked at the same time

during braking, the use of the attachment conditions and the directional stability of the vehicle are both advantageous. On road with any adhesion coefficient, the conditions for front and rear wheels are locked at the same time are like these: the sum of the braking forces of the front and rear wheel brakes is equal to the adhesion force and the braking force of the front and rear wheel brakes is equal to the respective adhesion force.

The fomulas are as follows:

$$\begin{cases} F_{\mu 1} = F_{Z1}\varphi \\ F_{\mu 2} = F_{Z2}\varphi \end{cases} \quad \text{且} \quad F_{\mu 1} + F_{\mu 2} = G\varphi \quad (\text{Yu Z. ,}$$

Principle of Automobile, 2009)

Or

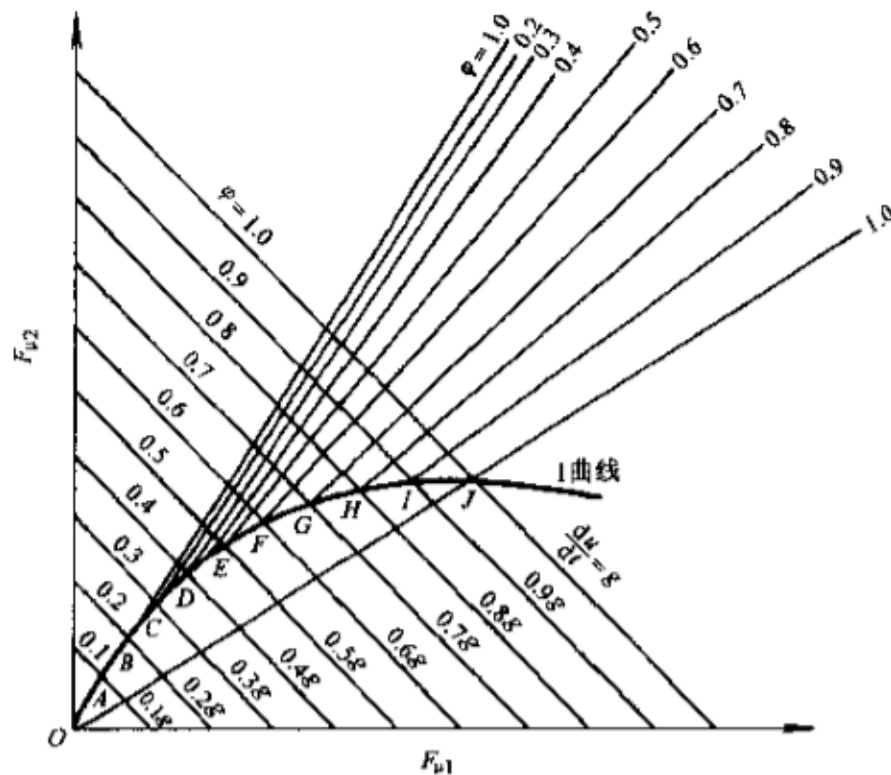
$$\begin{cases} F_{\mu 1} + F_{\mu 2} = G\varphi \\ \frac{F_{\mu 1}}{F_{\mu 2}} = \frac{b + \varphi h_g}{a - \varphi h_g} \end{cases} \quad (\text{Yu Z. , Principle of Automobile, 2009})$$

We can get this fomula after eliminating the variable road surface adhesion coefficient.

$$F_{\mu 2} = \frac{1}{2} \left[\frac{G}{h_g} \sqrt{b^2 + \frac{4h_g L}{G} F_{\mu 1}} - \left(\frac{Gb}{h_g} + 2F_{\mu 1} \right) \right]$$

(Yu Z. , Principle of Automobile, 2009)

The curve drawn by this equation is the relationship between the front and rear braking force while the front and rear wheels are locked at the same time - the ideal front and rear wheel brake braking force distribution curve.



(Yu Z. ,

Principle of Automobile, 2009)

This curve is a distribution curve of the front and rear brake braking forces when the pedal force is increased to the front and rear wheels being dragged. However, the front and rear brake

braking forces of the car cannot be distributed according to the requirements of the curve. During the braking process, wheels on one axle are often locked first, and as the pedal force increases, wheels on the other axle are locked. This process requires the rear axle to be locked first and then the front axle to be locked to improve the stability and safety of the vehicle. Therefore, the brake braking force distribution scheme tends to select a point below the curve.

The ratio of the front and rear braking force of our vehicle is a certain value. The ratio of the common front brake braking force to the total brake force of the vehicle indicates the proportion of the distribution, called the brake braking force distribution coefficient, as shown in the following formula:

$$\beta = \frac{F_{\mu 1}}{F_{\mu}}$$

(Yu Z. ,

Principle of Automobile, 2009)

Each physical quantity can be expressed as follows:

$$F_{\mu 1} = \beta F_{\mu}, F_{\mu 2} = (1 - \beta) F_{\mu}$$

$$\frac{F_{\mu 1}}{F_{\mu 2}} = \frac{\beta}{1 - \beta}$$

(Yu Z. , Principle of Automobile, 2009) (Yu Z. , Principle of Automobile, 2009)

Since the braking force of the front and rear brakes is a fixed value, the front and rear wheels can be locked at the same time to achieve the best braking effect only when braking on a road surface with a certain adhesion coefficient, that is, a synchronous adhesion coefficient.

The synchronous adhesion coefficient can be obtained by using the graph method or by an analytical method. The following formula is obtained after finishing the calculation which assumes that it brakes on the surface of the synchronous adhesion coefficient and the front and rear wheels are locked at the same time.

$$\frac{\beta}{1 - \beta} = \frac{b + \varphi_0 h_g}{a - \varphi_0 h_g}$$

$$\varphi_0 = \frac{L\beta - b}{h_g}$$

(Yu

Z. , Principle of Automobile, 2009)

In the formula, L is the wheelbase of the car, L=a+b

In the course of the research, I chose the braking distance of the car's 100-0km/h as the evaluation standard of the braking system. The target of my evaluation and upgrade of the vehicle is to make the braking distance less than 40 meters.

The calculation of the braking distance of the car involves many aspects. But I only chose the following widely used formula to finish it.

$$s = \frac{1}{3.6} \left(\tau_2' + \frac{\tau_2''}{2} \right) u_{a0} + \frac{u_{a0}^2}{25.92 a_{bmax}}$$

(Yu

Z. , Principle of Automobile, 2009)

It can be seen from this formula that the main factors

determining the braking distance of the vehicle are: the time of the brake action, the maximum brake deceleration and the initial braking speed. It is obvious that the greater the adhesion or brake braking force, the lower the starting brake speed, and the shorter the braking distance.

Process of Researches

The tasks which need to be completed after the entire task process has been divided become clear and unambiguous. Firstly, the entire brake system was split and an accurate measurement was performed to obtain a large amount of raw data. Then use the obtained raw data to update the model of some parts in Solidworks to provide a complete vehicle model for the subsequent virtual test.

After extensively querying the literature and magazines and combining with the discussion with teachers at the weekly meeting to deepen the understanding of the car's braking system and braking process, it began to use some of the original formulas found to push to the calculation to obtain high practicality which means to make it convenient to input the formula for calculation into the computer. Using these formulas to complete the preliminary processing calculation of part of the original data.

After completing the preliminary processing, the results are analyzed for component-by-component analysis to seek an optimized upgrade plan. The initial brake system is evaluated to see if it meets the requirements of the competition and the goals established. At the same time, the optimized upgrade plan is evaluated, the cost of implementing the plan is calculated and the feasibility is verified. Finally, the upgraded brake system is

compared with the initial brake system to verify the effectiveness and feasibility of the optimized upgrade solution to determine the final solution.

Analysis

There are two main difficulties in the analysis process: using which brake system and the determination of the aerodynamic force.

There are two main solutions for the brake system: one is the front and rear brake braking force proportional variable braking system and the front and rear brake braking force proportional fixed braking system. Our existing vehicle uses the later one with a fixed brake ratio of the front and rear brakes. The system is cheaper and has a simple mechanical structure and high stability. However, because the brake force ratio of the front and rear brakes is fixed, there is only one stable adhesion coefficient, which is not suitable for roads with complicated road conditions.

The other one is a brake system with a variable ratio of front and rear brakes. This solution can be achieved in a variety of ways, but considering the cost and stability factors in harsh environments, combined with the analysis of the MCR team's braking scheme, the knob pull-type horizontal braking force distribution adjustment structure is not only cheap and easy to implement, the mechanical structure is also simple and stable. This

solution changes the cable pulling horizontal brake force distribution module by the knob in the cockpit, thereby realizing the change of the distribution of the front and rear braking forces. Compared with other variable front and rear brake force distribution schemes, the biggest advantage of this scheme is that it can regulate the distribution of the front and rear brake braking forces in a certain interval. This is beneficial for the technician to perform corresponding simulation calculations for the road conditions and find the optimal braking point which can delay the start of braking time as late as possible. That is beneficial to shorten the lap time.

In determining the aerodynamic force, since two prototypes are not equipped with the body shell components, the external appearance of the vehicle body and its corresponding aerodynamic characteristics cannot be accurately judged. At the same time, I got the model of the Aston Martin DB9 racing car on the Vehicle Dynamic course. Through the simulation experiment of that model, a series of aerodynamic parameters like CL_f , CL_r and CdA of the model are analyzed and applied to the calculation of the aerodynamic force for Zentrack.

With the downward pressure of the front and rear of the vehicle body, the curve of the front and rear downforce with the

speed is plotted. For the case of the conventional curve deceleration and cornering, only 60-100km/h is used as the reference interval. The front and rear downforce values at 85 km/h were selected as the front and rear downforce forces for the entire braking process after analyzing the curve.

For the horizontal resistance in the direction of the vehicle body, the horizontal resistance of the vehicle body is plotted with speed in which 60-100km/h is also used as the reference interval. The horizontal resistance value at 85km/h is selected as the reference horizontal resistance throughout the normal braking process from this curve.

Conclusion.

After analyzing the existing brake system, the brake system is a brake system with a fixed braking force ratio of the front and rear brakes. The front brake cylinder has an inner diameter of 2.1 centimeters and an area of 3.46 centimeter square. The rear brake cylinder's inner diameter is 1.8centimeters, the area is 2.54 centimeter square. The adjustment ratio of the horizontal adjuster is front/rear: 50%/50%. Each front brake caliper is equipped with two sides each side three pistons which internal diameters are 25millimeter, 25 millimeter, 30 millimeter. The total area of the brake pistons is 33.8 centimeter square for one front brake and the total area of the front brake pistons is 67.5 centimeter square.

Each rear brake is equipped with two sides each side two pistons with inner diameters of 30millimeter and 25millimeter respectively, and the total area of each rear brake is 24 centimeter square. The total area of the rear brake pistons is 47.9 centimeter square. The friction coefficients between brake pads and discs are tentatively set at front 0.45 and rear 0.40. On this condition, the braking force distribution ratio is 53.8% for the front brake and 46.2% for the rear brake. The maximum deceleration can be 9.12 meter per second square. The required braking distance from 100km/h to 0 is 42.8669meter under these circumstances.

For the upgrade optimization of the prototype vehicle, it can be started from two aspects. On the one hand, the components of the brake system can be reinforced. On the other hand, the proportional relationship of the brake force distribution of the front and rear brakes can be changed.

For the reinforcement of the brake system, change the mechanical structure between the brake pedal and the brake master cylinder, so that the lever ratio is further increased, and a larger brake hydraulic pressure can be obtained. This method is the easiest one, but it also means that the structure needs to be equipped with a strong enough brake pedal. It is also possible to replace the brake master cylinder with a smaller cylinder diameter

to obtain a higher brake hydraulic pressure under the same brake pedal force. However, both approaches require attention to the maximum brake fluid pressure that the brake caliper and pipe can withstand. Finally, you can replace brake calipers and discs with higher quality to improve braking performance and stability.

In this project, only the replacement of the brake master cylinder is simulated and the results are compared and analyzed.

Replace the rear brake master cylinder with the inner diameter of the front brake master cylinder of 2.1 centimeter, and the cylinder cross-sectional area of 3.46 centimeter square. At this time, although the maximum braking force of the rear brake system is reduced from 4517.6425N to 3319.0843N. If the original system braking force distribution scheme is maintained, the maximum braking deceleration of the vehicle will increase to 9.43 meter per second square, an increase of 4.78%, and the braking distance of 100-0 km/h will be shortened to 40.9122meter, with a drop of 4.56%, but still failed to reach the expected target of 40metres.

To change the ratio of brake force distribution, adjust the horizontal brake force distribution adjuster to front/rear = 52/48 while maintaining the same components. On this condition, the ratio of front and rear braking force of the entire brake system is

front 55.85. % and after 44.15%. The maximum braking deceleration that the braking system can withstand increases to 10.6 meter per second square, an increase of 16.23%, and the braking distance of 100-0 km/h is shortened to 36.3965m. The decline was 15.09%, reaching the expected target.

Areas of concern.

When I first thought about how to upgrade and optimize the brake system of the prototype car, a comprehensive upgrade plan for both the control mode and the braking mode of the brake system came to mind. Equipped with electronic brake pedal and horizontal acceleration sensor to capture the brake signal and

horizontal acceleration, each wheel is equipped with an electronic brake cylinder. After receiving the brake signal processed by the processor, the dynamic sub-pump independently completes the braking process according to the optimal braking force distribution ratio relationship in order to achieve the best braking performance.

Unfortunately, during the discussion with the teacher about the program, I was told that this program did not meet the requirements of the competition. The race regulations only allow the change of the brake force distribution ratio of the front and rear brakes of the vehicle. The ratio of the brake force distribution of the left and right brakes of the vehicle is not allowed to change. Therefore, the adjustment scheme of the MCR fleet for the distribution ratio of the brake force of the front and rear brakes of the vehicle is selected.

However, the magnitude and direction of acceleration can be captured by the sensor and the calculation can be finished by the processor. The braking performance of the vehicle during most braking processes can be improved and the braking distance can be shortened. This brake scheme has been reflected in the latest model 01 PHEV (Figure 5) of geely's link&co (PHEV, 2019).



Figure 5. 01 PHEV

It uses four channels to independently control each wheel for braking. It can intelligently adjust the torque of different wheels according to the driving situation of the vehicle, and help the driver to make the vehicle safe, stable and smooth on corners, ramps, separated roads and snow and ice. drive. 01 PHEV is a new energy electric vehicle, and when the brake system works with Bosch iBooster, it can achieve almost 100% braking energy recovery, which can not only increase the cruising range of electric vehicles by up to 15.4%, but also Effectively reduce the fuel consumption and carbon dioxide emissions of hybrid vehicles. It can be seen that this brake system can not only improve the

driving safety, stability and smoothness of the vehicle, but also contribute to reducing carbon emissions and protecting the natural environment.

Harry Farnhill

Work content

Weight distribution

One of my areas to study was to analyse and improve the weight distribution. What this task meant was that the car needed to be stripped as far as we could go and weigh every individual component which would be listed in an excel spreadsheet. This would show the obvious areas to improve and to breakdown the areas of the car which needed to be improved. The car would also need to be weighed on the mobile scales so we could find out the weight distribution with the rolling chassis we were provided. This would show the distribution on each wheel so we could find the weight distribution front to back as well as left to right.

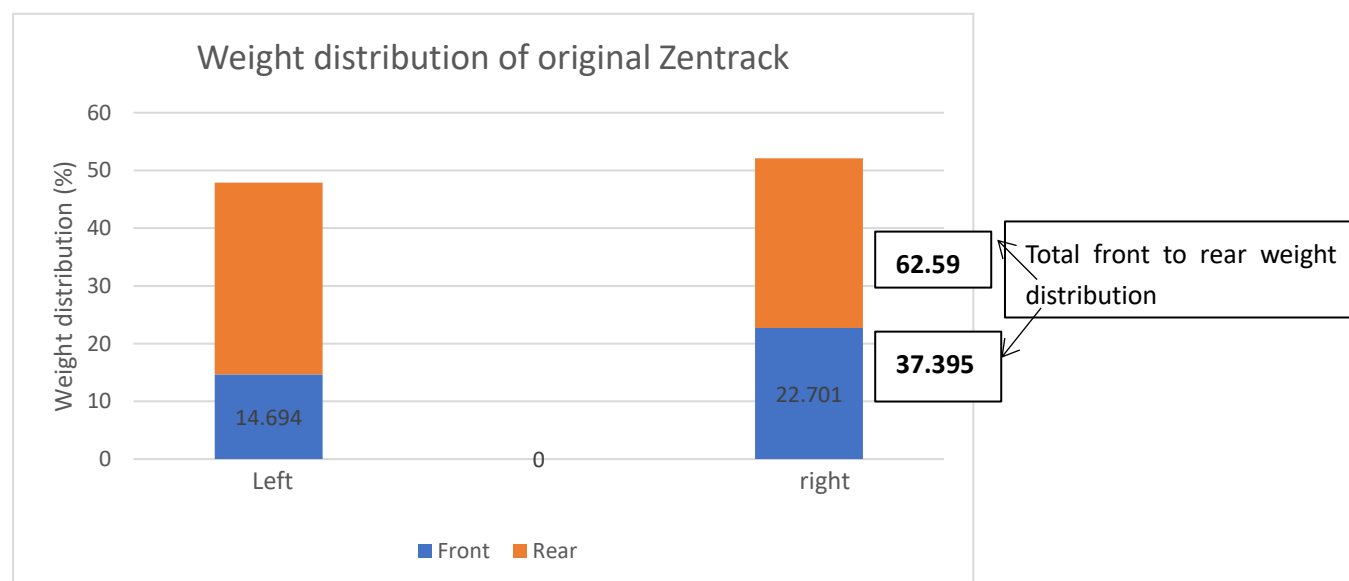
Stress testing

Another area that I would study would be the structural design of the chassis by stress testing the passenger cell to show displacement under force and to reduce the mass by 10%. This would involve re doing the CAD model that was provided as it meshed and there were gaps between roll bars and was inadequate to stress test.

Research carried out

Weight distribution

The research that I conducted was to fully understand the effects of weight distribution of a mid-engined vehicle and compare that to what's Johns' requirements of the Zentrack were. The weight distribution of the Zentrack was 38:62 from front to rear with an empty fuel tank. This means that car would oversteer under corner entry and mid corner. This is because there is greater mass over the rear axle which can generate more inertia when rotating around the corner. (J.Rowley, n.d.) This would push the rear of the car away from the corner causing oversteer. The handling can be unpredictable and difficult to handle mid corner as the driver would need to control oversteer which is traditionally harder to control than understeer. However, if the car could be designed that the oversteer could be manageable and controlled the car can be a lot of fun to drive and easier to handle. (L.Milliken, 1995) It can also mean that there would be better traction out of corners providing more grip and can help rotate the car around the corner acting like rear wheel



steering but only if it can be controlled. (Smith, 1984) From the research I Carried out the ideal weight distribution that I aimed for would be 55:45 rear to front, this would keep the characteristics of a mid-engine car but would be more predictable to drive. (L.Milliken, 1995)

Stress testing

The research that I carried out for stress testing the passenger cell was to look at images of race car roll cage designs and passenger cells. A lot of the passenger cells in road and race cars are built into the car so I researched was MSUK and FIA roll cages for GT3 race cars. What I found was that all roll cages had 2 cross bars intersecting each other. A triangle shape has the highest torsional strength and impact strength so tried to integrate as many triangles as I could into the passenger cell model but still make space for the other components to fit around the chassis as well as keeping the mass down.

(L.Milliken, 2002)



Intersecting cross bars forming a triangle

The big difference between the Zenos chassis and the Zentrack was that there is large aluminium beam running through the middle of the Zenos and transverse at the back of the passenger cell. This provided the main structural integrity to the chassis which means losing this piece could massively weaken the whole structure for the chassis. Therefore I decided that I needed to apply 5G in the x,y and z co-ordinates to find the structural integrity of the chassis to make sure that it would have the strength to pass NCAP frontal impact tests. (Anon., 2011) (Singh, 2015) A 30-mph impact would register 20G for a full-scale car, therefore I chose to apply 5G just for the passenger to cell as this wouldn't

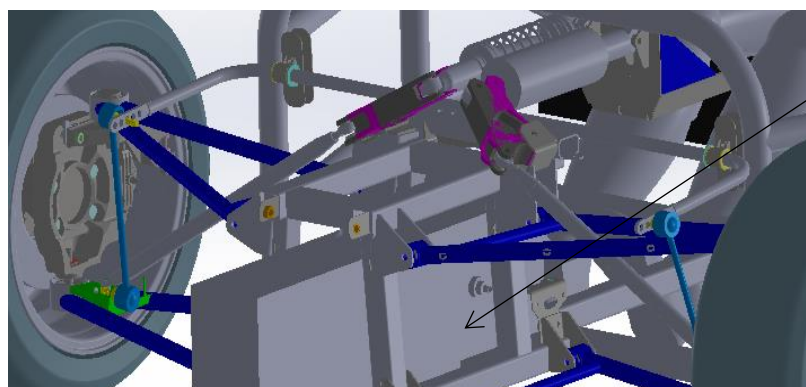


The aluminium structure in the Zenos E10

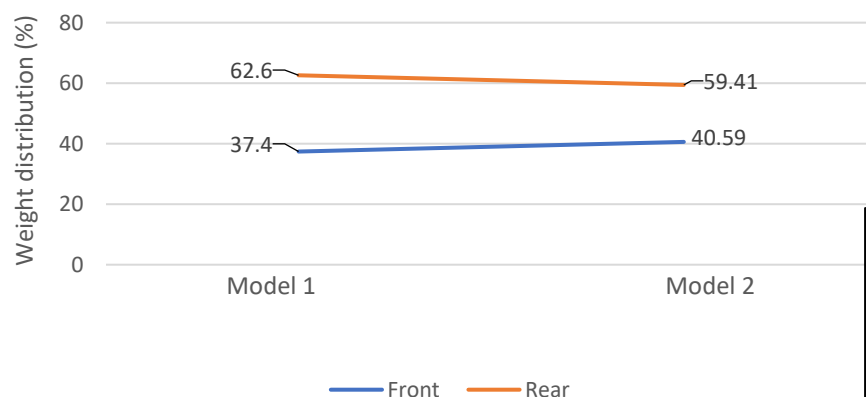
completely snap the passenger cell but would find the weak areas to redesign the chassis. (Sethupathi, 2018)

Designs and development carried out

Some of the initial designs carried out were to relocate the fuel tank and the radiators. The wheelbase of the Zentrack was 2600mm which meant that the rear of the car was very compacted with the engine and gearbox over the rear axle which means there wasn't much room for fuel tank in the rear of the car. John would like an option to run a Zentrack into MSUK endurance racing which means that the fuel tank would need to have a capacity of 60 litres. The only room available to have a 60-litre fuel tank would be in front of the front suspension as for MSAUK fuel tank lines and pipes could not be in the passenger cell. (R.Davis, 2019)The weight distribution improved to 59:41 rear to front with a 20-litre fuel capacity tank, the reason for using a 20 litre initially was to evaluate the difference before designing a 60-litre fuel tank. (Anon., n.d.)What this meant is that when the fuel tank was full, the weight distribution would be further forward than when the tank was empty. This affect would be worse with 60 litre fuel tank so the characteristics would vary with fuel load. This could be combined with a manual anti roll bar adjust in the driver's cockpit for racing so when the fuel tank becomes empty the car could oversteer more, therefore softening the anti-roll bar at the back would reduce oversteer. (C.Dixon, 1996)

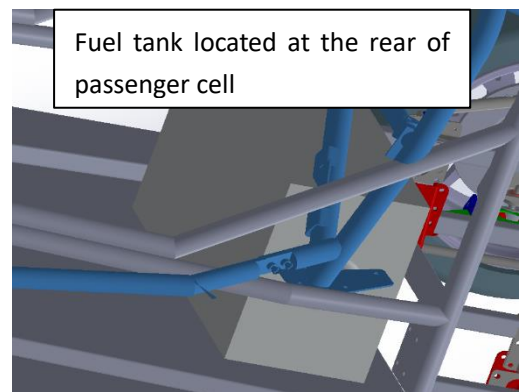
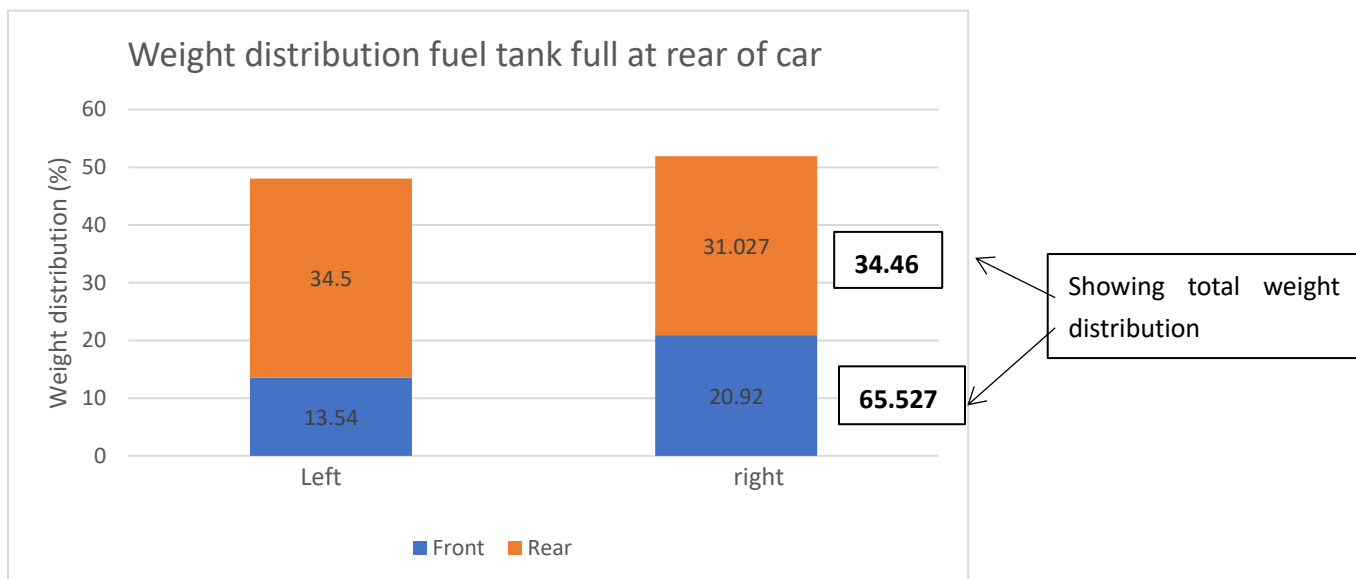
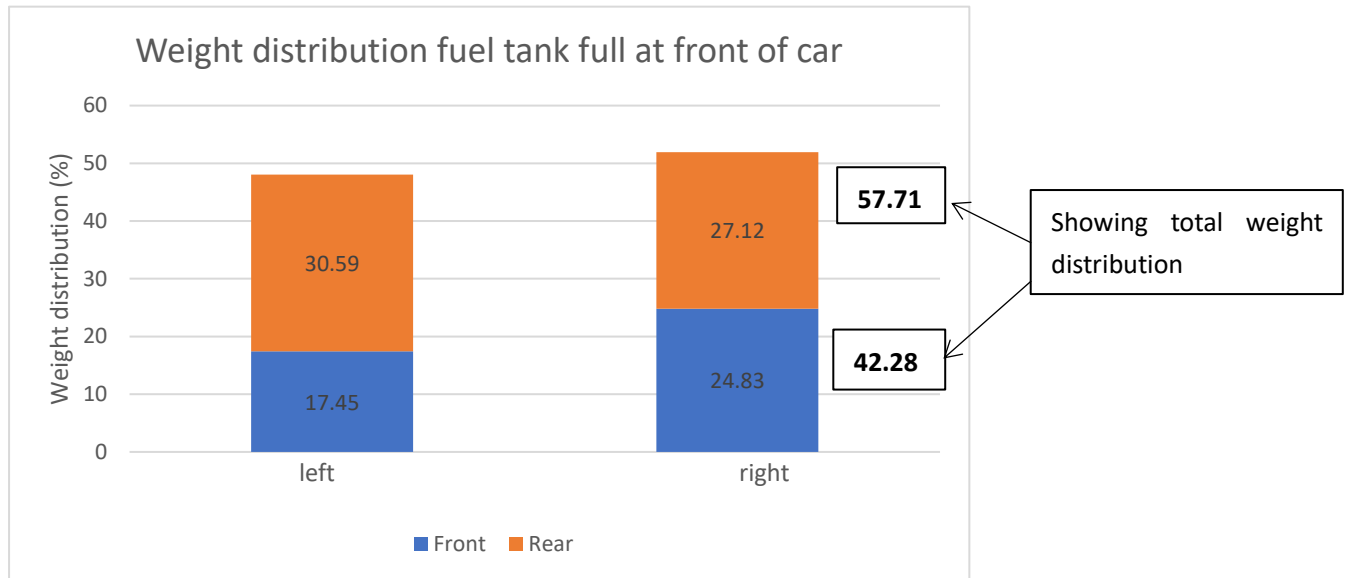


Comparison of weight distribution between Model 1 and model 2



Model 1: original Zentrack
Model 2: 20 litre fuel tank located at the front of the car, Battery also at the front of the car

Below is a graph showing the effect on weight distribution between locating a full 60 litre fuel tank at the front or the back of the car.



- There is more space for the fuel tank at the front compared to the rear where there is not enough space
- The passenger cell and chassis would need to be redesigned to fit a 60 litre fuel tank
- There would need to be a firewall and more structure to protect the fuel tank at front in the event of a crash

Stress testing

From stress testing the chassis in Solidworks I looked at the very high loading areas to find ways to reduce the load because that would be first area to fatigue and snap. (S.S.Sane, n.d.) The way I did this was by integrating crossbars into the chassis design to provide more strength under very impact and to improve torsional strength. (Yu, 2017) The main areas that needed to be development was the main cell along the side of the chassis and the backside of the passenger cell. When using 5G of force this was the major area of displacement and there was too much material that was being wasted. (Babu, 2012) With the CAD model that was provided I was aware of how much freedom I had to redesign the cross bars, so they didn't intersect with the roll cage and other components. However, the MSUK and passenger cell intersected on the original model so I assumed this would not be the final design and I redesigned the passenger cell to increase the strength regardless if it meshed with other components as there were not the final designs.



Testing carried out

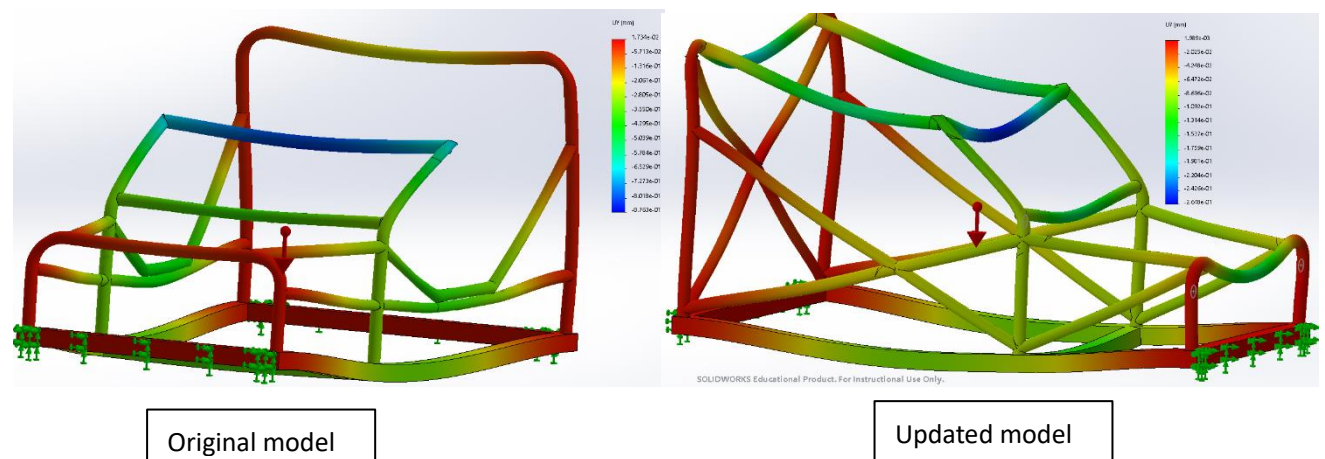
Weight distribution

The testing carried out had to be virtual and theoretical calculations because we didn't have the money or resources to develop the components in the timeframe we had for the project. The dimensions were first calculated by measuring the distance from the wishbones to the bodywork and the floor to the bodywork, this was to find as much available space we could work with. The fuel tank, battery and radiators would then be designed in CAD with the correct material and mass properties, the fuel tank in CAD would have the mass properties with a full fuel tank and an empty tank so I could calculate the weight distribution for when the tank is full and empty as this would be useful for the simulator model

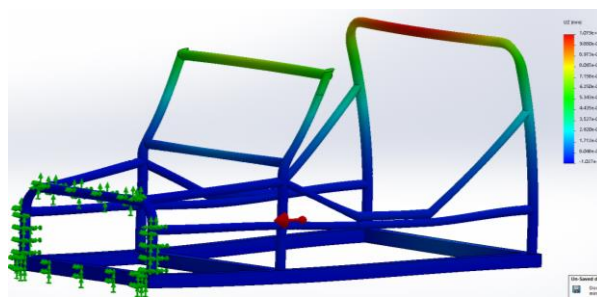
Stress testing

The stress testing of the chassis that was carried out was applying 5G in the x,y and z place. The stress test below shows displacement in the axis that the force was acting on, for example, 5G side impact will show displacement in the Z-axis. (Singh, 2015). For the properties of steel refer to the appendices

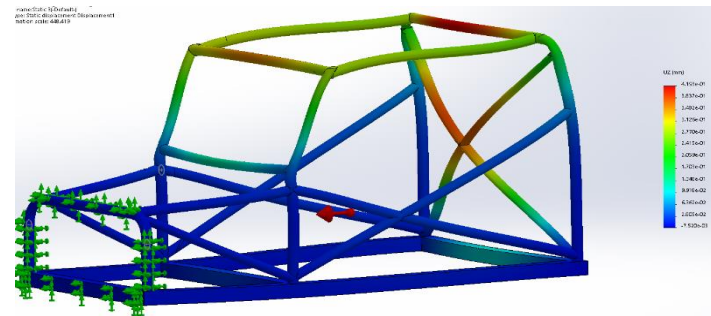
5G acting on the roof



5G rear impact

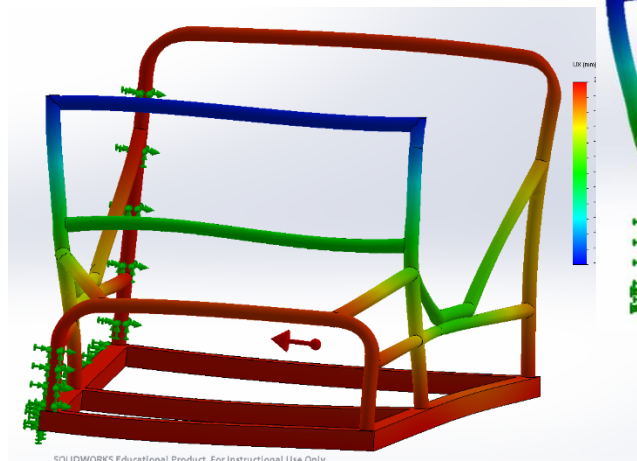


Original model

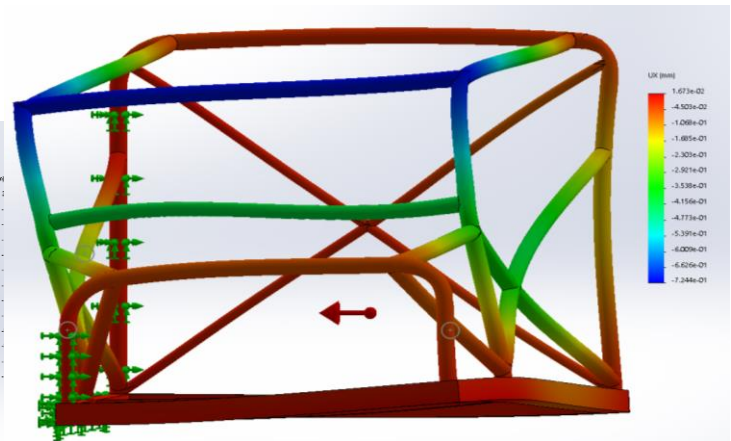


Updated model

5G side impact



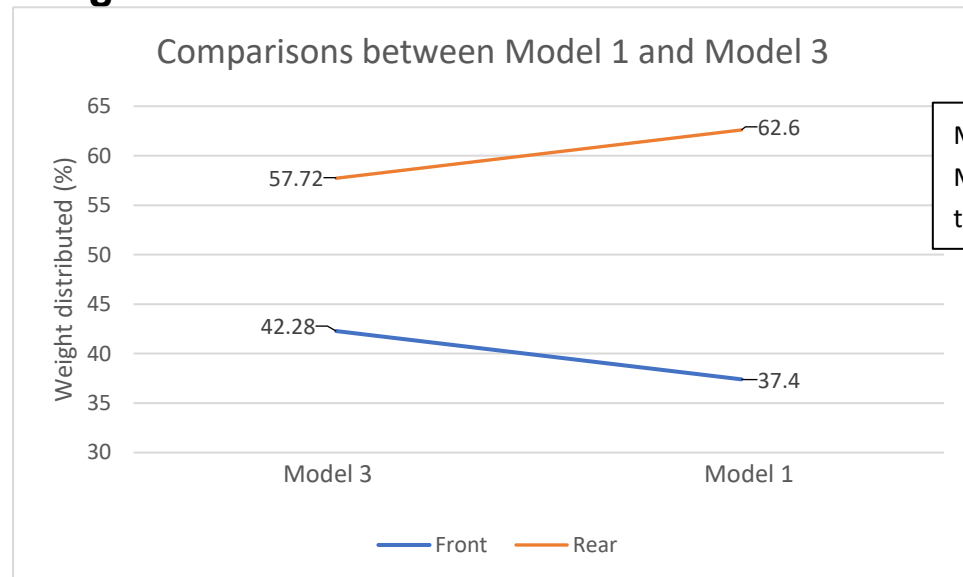
Original model



Updated model

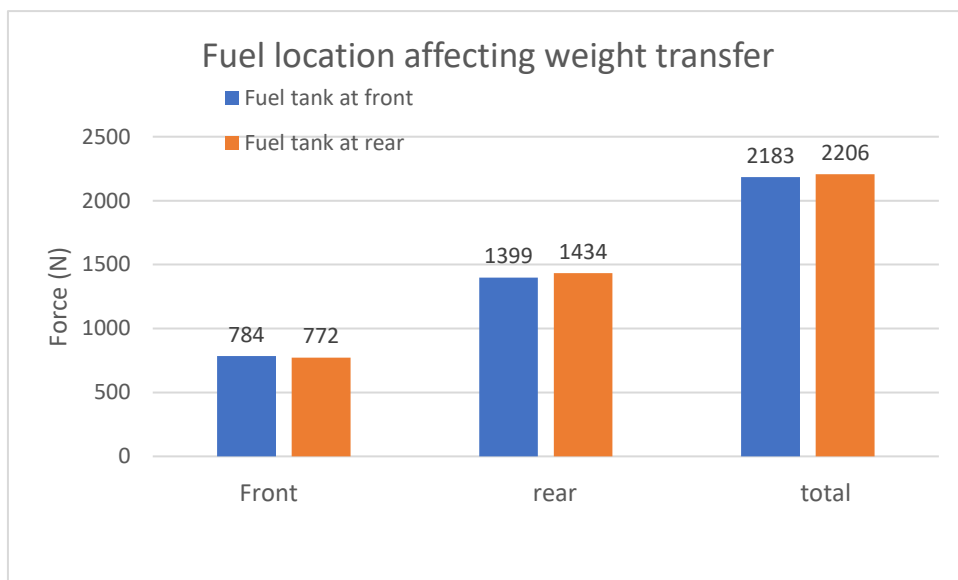
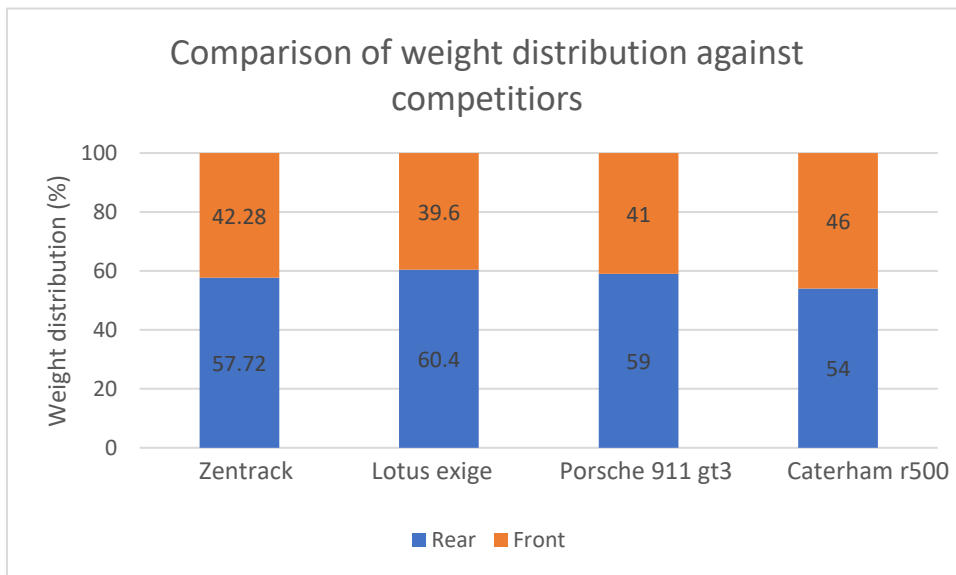
Analysis carried out

Weight distribution



From what I have found is that locating the fuel tank at the front of the car improved the weight distribution and got closer to the aim of 55:45 rear to front weight distribution but would vary with fuel load. Another location which was considered was a fuel tank to be located in between the passenger cell and the engine. The effect of the fuel tank becoming empty would have a smaller effect at the rear because the engine and gearbox weighed at 150kg whereas there is only front suspension over the front axle which weighed under 25kg. However, the weight distribution would be further towards the rear which could cause the car to oversteer more under braking and cornering. (L.Milliken, 2002) If the car was set up with the anti-roll bar softer at the rear and suspension running softer at the back, there would be more traction so this could reduce the effect of a greater rear to front weight distribution. (Bastow, 2004) However, for novice/amateur drivers which would be likely to buy a Zentrack it could get complicated trying to set up the car and not affect the balance of the car dramatically.

Below is a graph showing how the updated Zentrack would compare against rival track day cars and locating the fuel tank would affect weight transfer



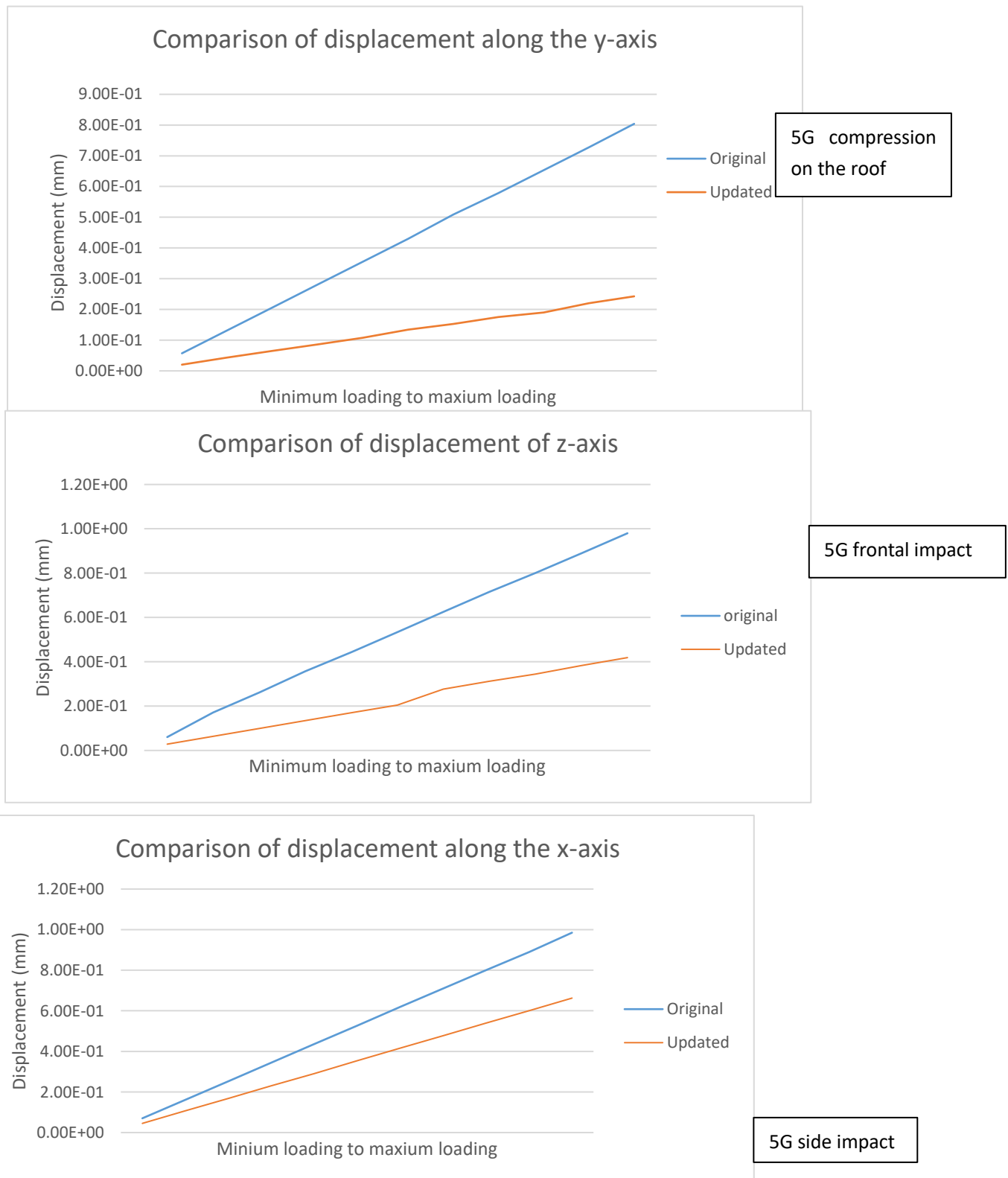
Actual weight transfer

- More weight transfer in total from the fuel tank been located at the rear
- The front could understeer more with the fuel tank at the front
- The rear could grip more and oversteer less with the fuel tank at the front

Stress testing

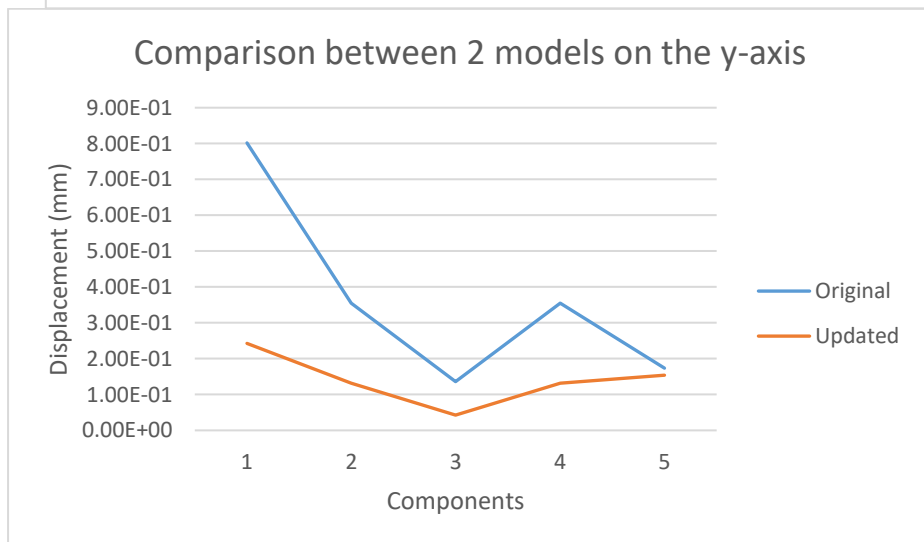
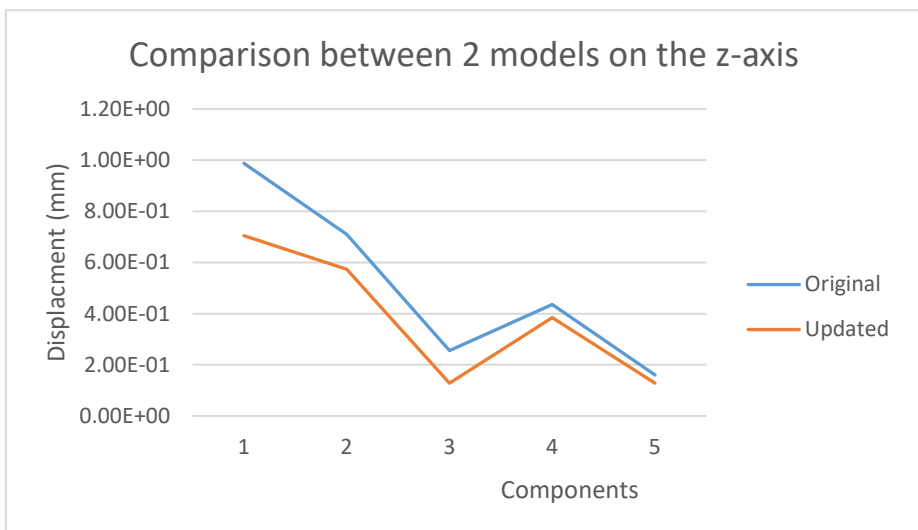
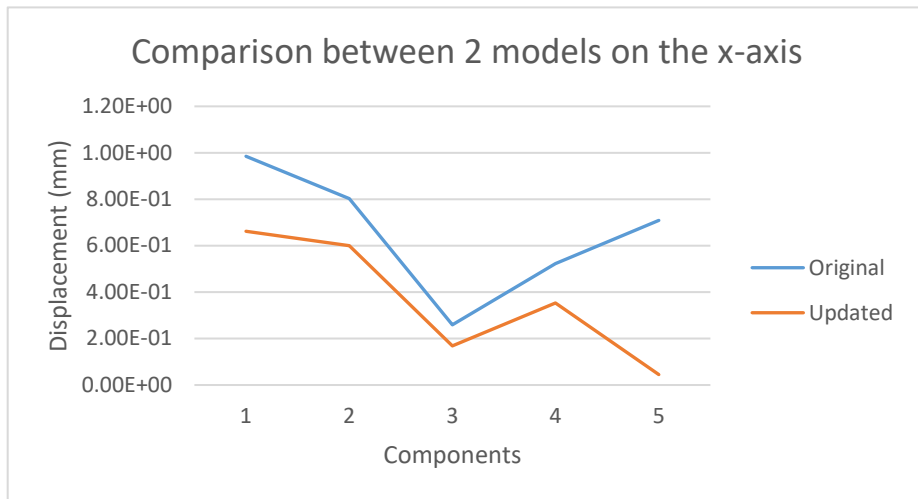
When I stress tested the original model, I found that there was lot of displacement from the frontal and rear impacts in the main area where the passengers would be sitting. These were the main areas that I wanted to change by intersecting cross bars on the side and introducing side roof bar to be attached between the A and B pillar. This would

connect the 2 main areas of the car and keep the chassis more rigid during very hard braking and cornering as it would provide more strength at the top of the chassis. (Singh, 2015) The graphs below show the maximum and minimum displacement under 5G of force between the original passenger cell and the updated passenger cell.



Below is table showing the displacement under 5G of force in that component area from the graphs on the next page.

Component	Number
Front Roof bar	1
A pillar	2
B pillar	3
Passenger side cross bar	4
Rear roof bar	5



The graphs above focus on the structural areas that were designed to improve the stiffness of the chassis. The results show that:

- The updated chassis reduces flex

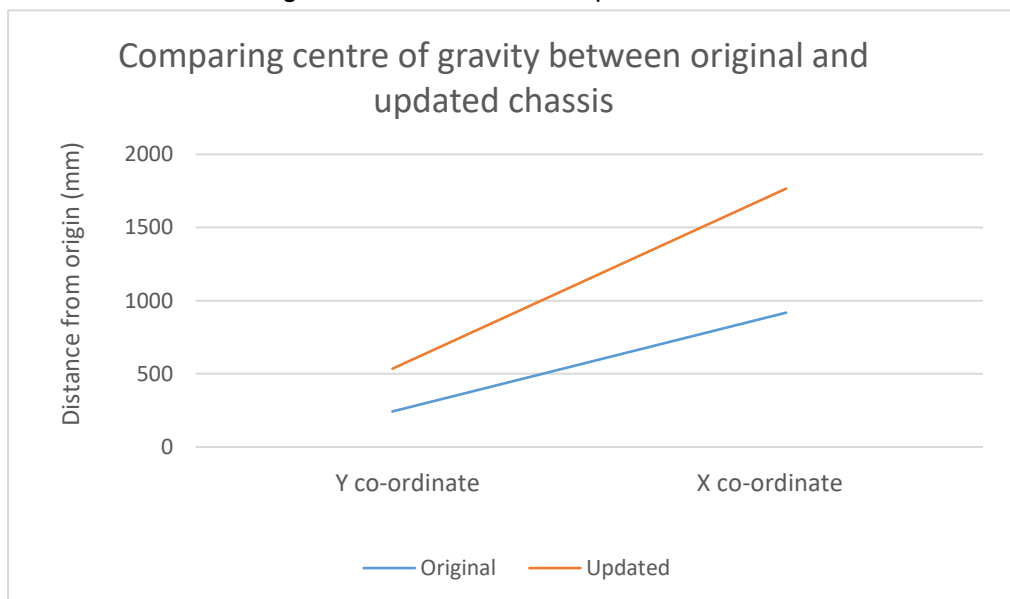
- The front and rear roof bar has displaced the smallest amount compared to the original model
- More material has been used which increases costs and mass
- The greatest improvement in strength seems to be from compression acting on the roof on the Zentrack in y-axis

The table below shows the percentage difference between the original chassis and updated chassis, the larger the percentage the bigger the difference between the displacement, the higher the percentage the greater improvement in reducing the flex.

Co-ordinate	Difference in displacement (%)
x	32.87
y	69.63
z	58.26

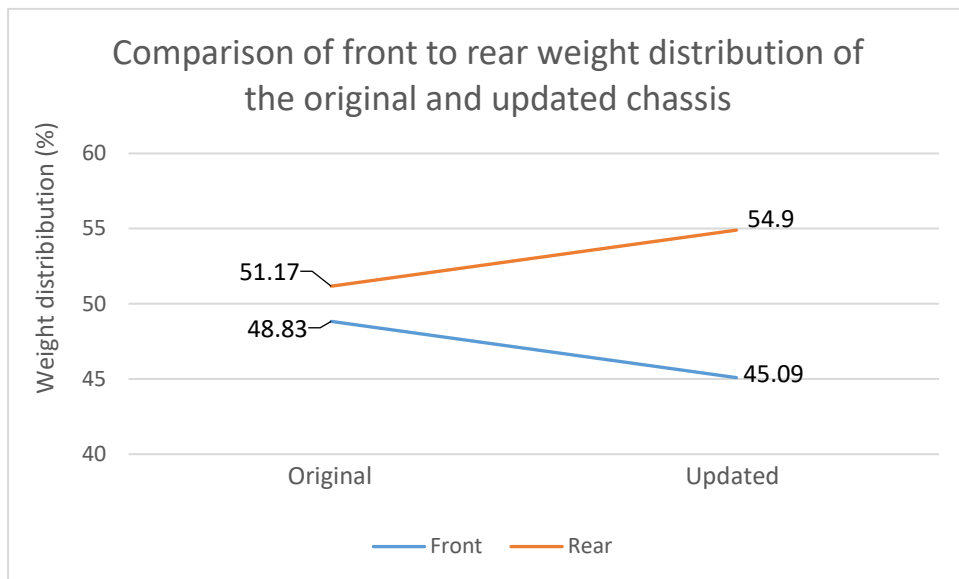
Weight distribution of the updated chassis

The graph below compares the centre of gravity between the original and updated chassis, the y co-ordinate is the centre of gravity height from the floor of the chassis and the x co-ordinate is cog distance from the back plate of the chassis.

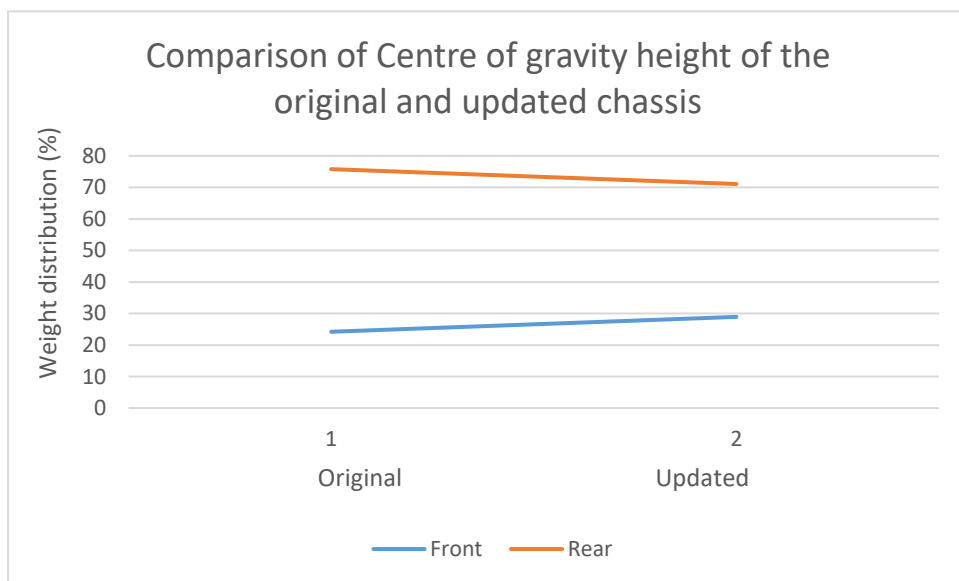


- The centre of gravity height is now 50mm higher because of adding the roof bar connecting the A and B pillar.
- The centre of gravity is 70mm further back because of the reduced material at the front and adding more intersecting cross bars at the rear of the chassis

The graphs on the next page show the difference in front to rear weight distribution and top to bottom weight distribution between the original and updated chassis



- The updated chassis has moved the weight distribution further back by 3.5% compared to original but increased strength by at least 30%.
- The chassis weighs 53 kg so only takes up 7.361% of the total weight of the car so would only have a small effect on weight distribution but doesn't get

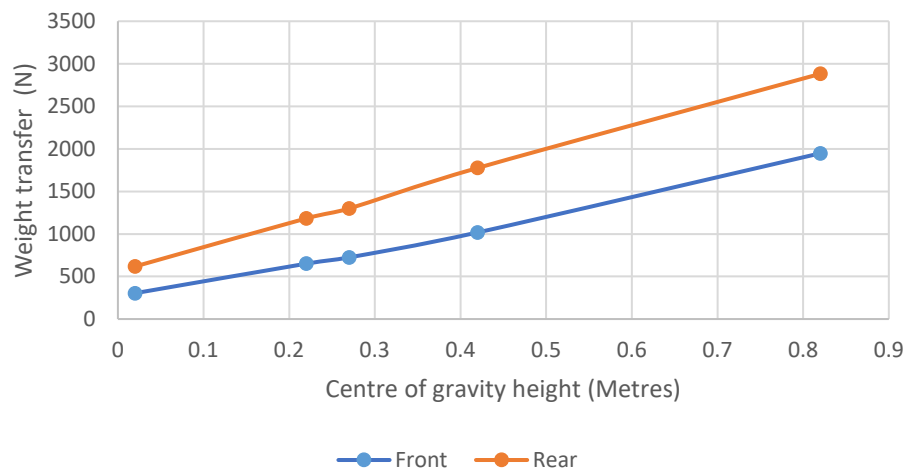


any closer to the 55:45 aim.

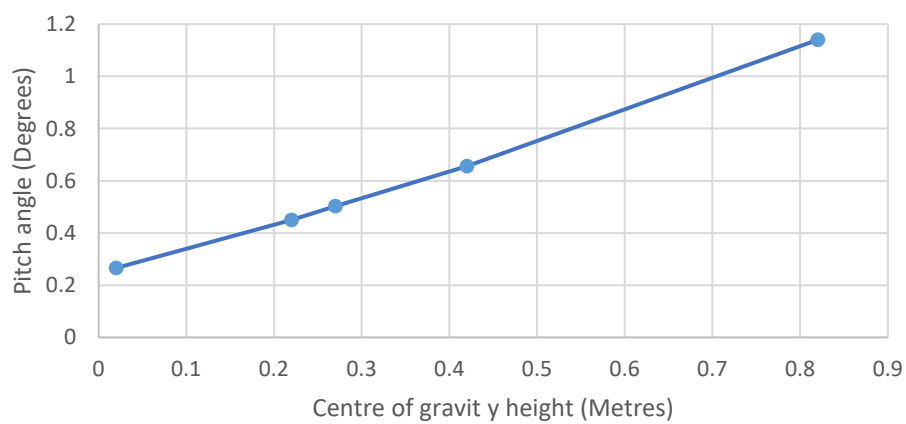
- The centre of gravity height has increased by 4.5% after adding the roof bar connecting the A and B pillar together which has improved the strength but decreased handling performance.

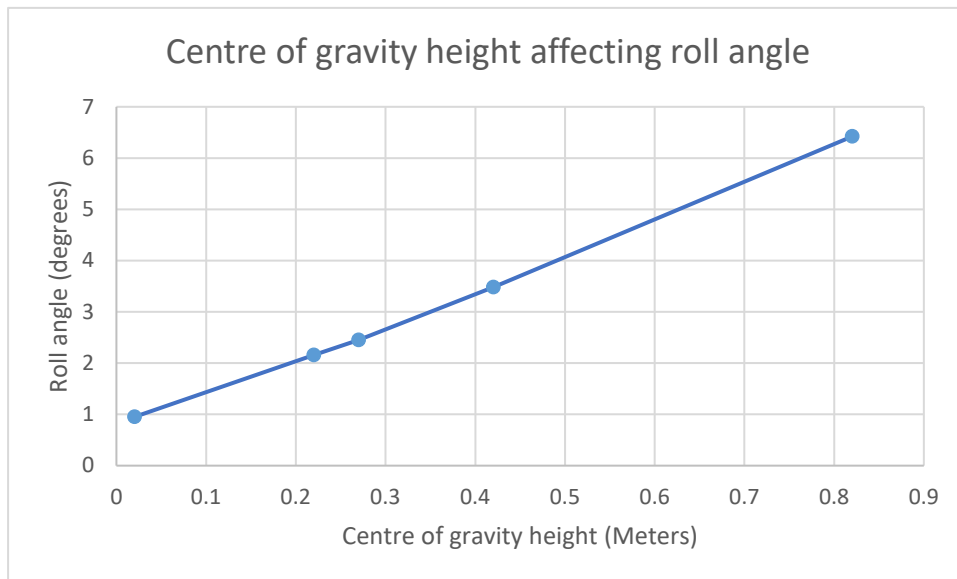
The graphs on the next page show how the centre of gravity height affects weight transfer, roll and pitch from the simulator model.

Centre of gravity height affecting weight transfer



Centre of gravity height affecting pitch under accelerating





The graphs show:

- The higher the centre of gravity the greater load transfers to the outside tyres when cornering which is more likely to increase understeer and oversteer.
- Roll angle is proportional to weight transfer as it increases at the same rate.
- The higher the centre of gravity height the increased traction when accelerating as there is greater force on the rear tyres maximising the contact patch due to weight transfer. However, the front tyres would overload under braking and which can cause understeer.
- The lowest centre of gravity height would be the ideal set up as it reduces weight transfer and applies maximum force to each tyre without overloading providing maximum grip.

Achievements and progress

Weight distribution

From the weight distribution tables, the location of the fuel tank and battery at the front of the car improves the weight distribution however the fuel tank can vary the weight distribution as the fuel burns in the tank. This means there needs to be further development of the relocated components to improve the weight distribution and to reduce the mass of the chassis but keeping the same strength. What this means is by relocating the fuel tank to improve weight distribution, the cars weight distribution will vary a lot with fuel load. The difference between the fuel tank when its full and empty will vary by 10% which means that the car will start to oversteer more as the fuel tank gets lighter. For inexperienced drivers this could be difficult to control and could be unpredictable. Therefore, it would be recommended for the track day model to have a manual anti roll

bar adjust within the cockpit for the front and rear. So, when the fuel tank becomes lighter the driver can stiffen the front and soften the rear to control oversteer.

Stress testing

Initially the importance of stress testing the passenger cell was to improve performance but as the new passenger cell doesn't have the central aluminium beam the priority became to make sure the passenger cell would be safe and strong enough from a side impact but also to make sure the chassis wouldn't flex during braking and cornering.



Aluminium beam providing structure after a crash

However, trying to connect as many areas together meant that the mass increased from 47.76kg to 53.56 kg which would equate to 0.5 seconds per lap slower. The centre of gravity moved higher and further back which was not intended and makes it more difficult to get to closer to the 44:55 aim, however the chassis has become more rigid as it had reduced by at least 30% less displacement than the original chassis. (Sethupathi, 2018) Therefore, going forward there will need be a compromise between the 2, further considerations could be:

- Using a different material such as aluminium, titanium or carbon fibre
- Reducing the size of material to thinner metal, however this could weaken the structure significantly
- Making the chassis 5-10% smaller to reduce the amount of material needed.

Review of the project

Weight distribution

The ideal weight distribution would be 50:50 but the handling dynamics can be subjective to the driver's ability and style. For example, a driver would like the car to oversteer more with a small degree of yaw into the corner to act like rear wheel steering with driver

wanting to have more of a 60:40 rear to front weight distribution so we had to listen to the direction John gave us of what people are likely to drive his cars. This meant it had to be a compromise between the two but what makes the Zentrack unique was that it was an affordable Mid-engine car which meant it still needed to handle like a mid-engine car otherwise it would be too like Caterham's or hot hatch race cars.

Strength testing

The chassis displacement decreased as it flexed less under high loading, but the weight increased by 7kg. This could be as I was focused too much on the safety and flex of the chassis that there could be too much unnecessary material to maintain the strength required. With more direction from the customer we could of go down the route of a lighter chassis or strength testing, but I am assumed that the Zentrack has not been NCAP crash tested and doesn't have the aluminium beam in the Zenos and impact testing became the highest priority over the mass of the chassis.

Cornelius May

Overview

The aim for my section of the group project was to create a simulation model of the existing Zentrack vehicle. This would be done using the BPS simulator. There were multiple reasons for creating a simulation model. The first reason this simulation was required, was to evaluate the performance of the existing design at the beginning of the project. This was to allow us to understand how the car performs on track, without needing to take up the time of a car. The next reason this was required is to evaluate the effectiveness of modifications and changes, that would be suggested by the rest of the team throughout the project. The simulator would allow the group to evaluate the effects making these changes have, on drivability and lap times. Finally the simulation would support in suggesting investigating further changes, that were not covered by the areas the rest of the group were already covering.

Creating the Simulation Model

In order to create the first simulation model there were a number of steps required. The issues that needed resolving were my lack of understanding of how to program the simulator and the lack of available vehicle data.

rFactor

Our simulator, BPS, is based on rFactor, which is an open platformed, flexible program designed to allow the user to run any type of four wheeled vehicle they would like through modding (Rfactor.net, 2019).

Since rFactor is designed to allow its users to create and modify vehicles to suit their requirements, its programming is relatively straight forward. Figure 1 shows the filing structure used.

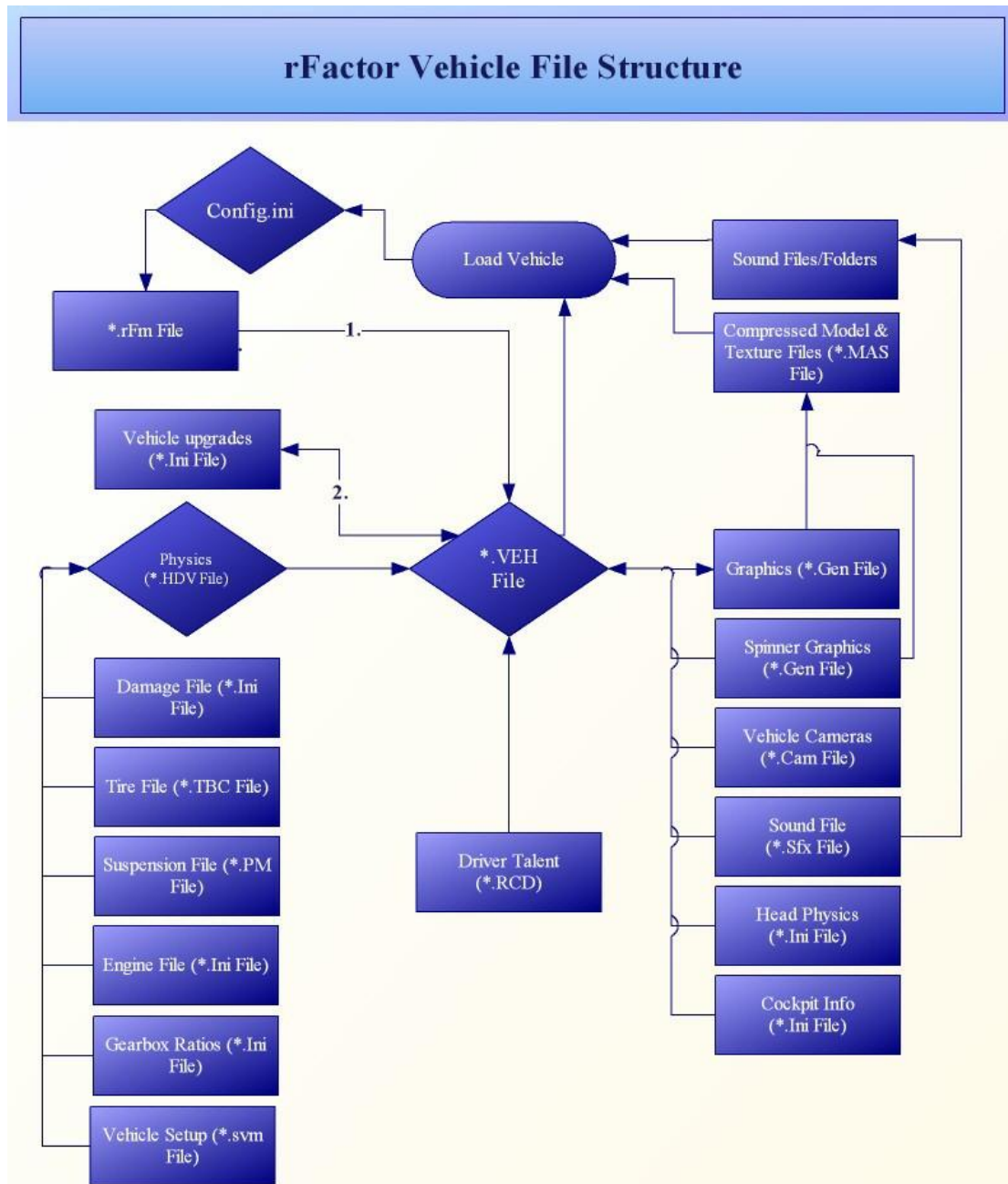


Figure 1: rFactor filing structure (Operation.arma.free.fr, 2019)

Each available vehicle in the game requires its own version of each of these files. Within each file is a list of parameters and as a total, this defines how the vehicle behaves.

Research showed that there are two ways to go about programming a new vehicle for the simulator. It is possible to begin completely from scratch, however the preferred method is to begin with an existing model, that is as similar as possible to your vehicle, and modify the vehicle parameters to suit your needs. The reason for this is that it can be easy to miss out important parameters or even entire files. Using

an existing model means that it is possible to simply go down a list of parameters and change each value to suit your car.

Whilst programming the simulator, a few other advantages of beginning with an existing model became apparent. Whilst filling in data, it is often the case that not all the data is available the first time you attempt to program the simulator. Not starting from scratch means that the model will still run and the available settings can be evaluated without all the data being available. Furthermore if data is never going to be available, starting with a model that is similar to your car, means that you can use the parameters from that vehicle instead, giving an approximation to work with.

Zentrack

With a basic understanding of how to program rFactor it is possible to begin collecting data for your model. Going through each file it is possible to make a list of the parameters required, before going to the car, or the CAD model and finding that data.

Model development

To begin with, a number of different vehicle models were evaluated for their suitability as a basis model for modification. Rear engined, rear wheel drive were the primary requirements, closely followed by engine size and power output to match the Zentrack.

The obvious choice would have been a Lotus Ellise as this is who Zentrack sees as their primary competitor. A vehicle model was found, however the model was not editable. For this reason the model was kept for the purposes of comparison and evaluation, but could not be modified to suit our car.

After a number of vehicles were tested for suitability, the group set its sights on the Junior 2WD Buggy (Anon, 2019). In many aspects this car was not particularly suitable. The real world application of this buggy was completely different to the intended use of the Zentrack, making its suspension geometry, and weight distribution completely different. Due to the availability of this data from Zentrack, it was determined that this would not cause an issue and that the pros of the Junior Buggy model were what was required. The model came with a 5 speed, rear wheel drive, 1.6L drivetrain that roughly matched what was expected for use in the Zentrack.

The availability of data will inevitably define the order in which a car model is built up or, in our case, modified. At the beginning of the project, a rolling chassis, without engine or bodywork, was the only model that was available for evaluation. For this reason the first area that was programmed into the simulator was the chassis geometry. The chassis was stripped down, and each component measured and weighed individually input into the suspension.pn. To simplify this process a 3D modelling programme (Anon, 2019) was used. Once the suspension geometry was programmed and individual component masses input, the chassis was reassembled and weighed in its entirety. This gave a first estimate of the center of mass location, using the available data. At this point, spring and damping rates were also measured, and ranges programmed into the simulator.

When a rolling chassis with an engine and some preliminary body work became available, the center of mass measurements were repeated and updated in the simulation model.

Towards the end of the project it became apparent that engine and transmission data would not become available. To allow testing to commence, this data was copied from the Lotus files as rough estimates, that would allow us to evaluate other areas of the model.

Evaluating a simulation model

Evaluation methods

Creating a simulation can be a complicated and lengthy process, so once the model is thought to be completed, it is necessary to evaluate the accuracy of the data received. Since this project is based on a real car of which prototypes are available, the easiest way to do this is to create a scenario on the simulator that can be replicated in real life.

An example of this is using comparing track data, by driving the simulated model on a track, and then taking the real car to the same real world track. This however causes a number of issues. For example, weather and climate are controlled in the simulator which is not possible on the real track. Even if this was accounted for, the same driver might behave differently in the car, than they would in the simulator. For this reason, an effective means of evaluation is to use a set of pre-specified ISO standards that evaluate the behaviour of the car whilst requiring little to no skill by the driver. Normally this data would be compared and used to find errors in the simulation until a reasonable margin of error is achieved, however due to this data not being available this step was not able to be completed.

To compare the repeatability, and accuracy of track testing vs. maneuver testing, a few drivers participated in a comparison and repeated identical maneuvers.

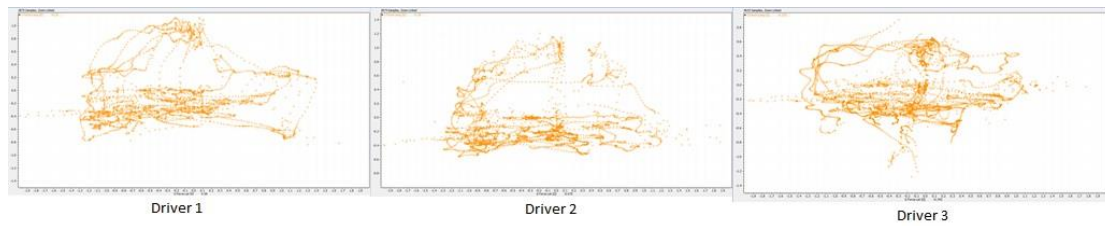


Figure 2: Frictional Ellipse

The use of the frictional ellipse shows how well each driver can keep the car at its limits. It also shows how varied the ability and performance of these three drivers are. Since all 3 drivers drove the same vehicle on the same track with the same amount of experience in that vehicle, they might be expected to perform similarly. Despite their average lap times being only 5.63% apart, their frictional ellipses show very different driving styles. Since driver 2s ellipse has the smallest lighn, this is the most constant driver.

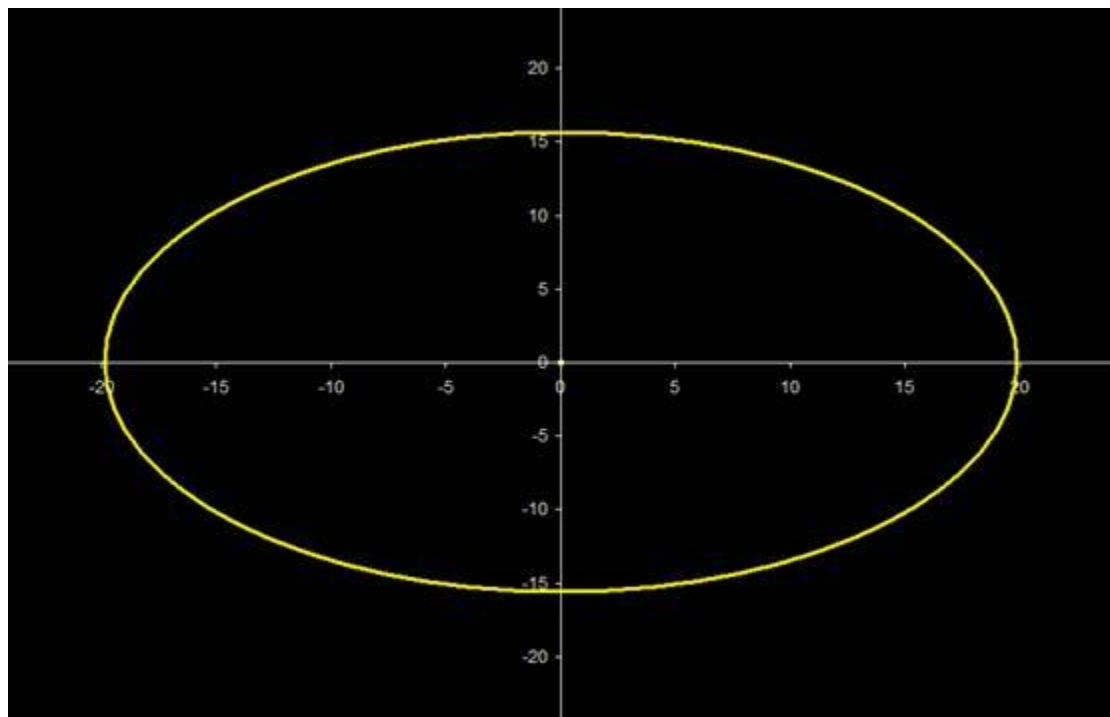


Figure 3: Perfect frictional ellipse (Macgracing.co.uk, 2019)

Figure 3 shows what a perfect frictional ellipse should look like. Since none of the ellipses in figure 2 look particularly similar to this, even driver 2 does not provide perfectly repeatable results. As a result, a different method to evaluate the data was investigated.

There are a number of ISO standards set out for vehicle manufacturers. The primary function of these standards, such as ISO 3888-2 is to prove the safety of these vehicle in certain circumstances (Car Engineer, 2019). Drivability can be defined as the smoothness and

steadiness of acceleration, in a straight line and when cornering (www.dictionary.com, 2019). ISO 4138 is another standard, that can also be used as a repeatable means to evaluate the drivability of the car. It can show us how the car behaves under cornering, allowing us to decide which of these characteristics makes for a “smooth and steady” driving experience, i.e. the car behaves the way the driver expects it to. Since our area of interest revolved around the suspension and chassis, ISO 4138 was chosen as the ideal method to evaluate the effect changes would have.

Using the data collected, we can look at a few vehicle characteristics, to decide which would best suit the customer. The first such characteristic is the Understeer of the car. From road cars, most amateur drivers are used to a car that understeers, meaning a greater steering input is required to achieve a turning radius due to the front wheels slipping.

Results

The simulator had two purposes. The first was to investigate the effect suggested improvements would have and the other was to research further areas of improvement.

Dampers

One area that was looked at was the dampers. Although the Zentrack already had a set of dampers, that John was not keen on changing, these dampers were adjustable, and which setting should be recommended to customers had not yet been investigated. This seemed like the perfect starting point for the simulator to investigate the effect of damping distribution on the car.

To evaluate the dampers, two aspects of the vehicle behaviour were investigated. The first was its maximum braking performance, since the dampers can affect how much grip the car has, and the second was the weight transfer under these conditions, since the dampers can affect the weight transfer.

The dampers had a very limited effect on the braking performance of the car, as figure 4 shows. Since this change is unlikely to be noticeable by the target market of the Zentrack, the weight transfer becomes a more important characteristic to look at.

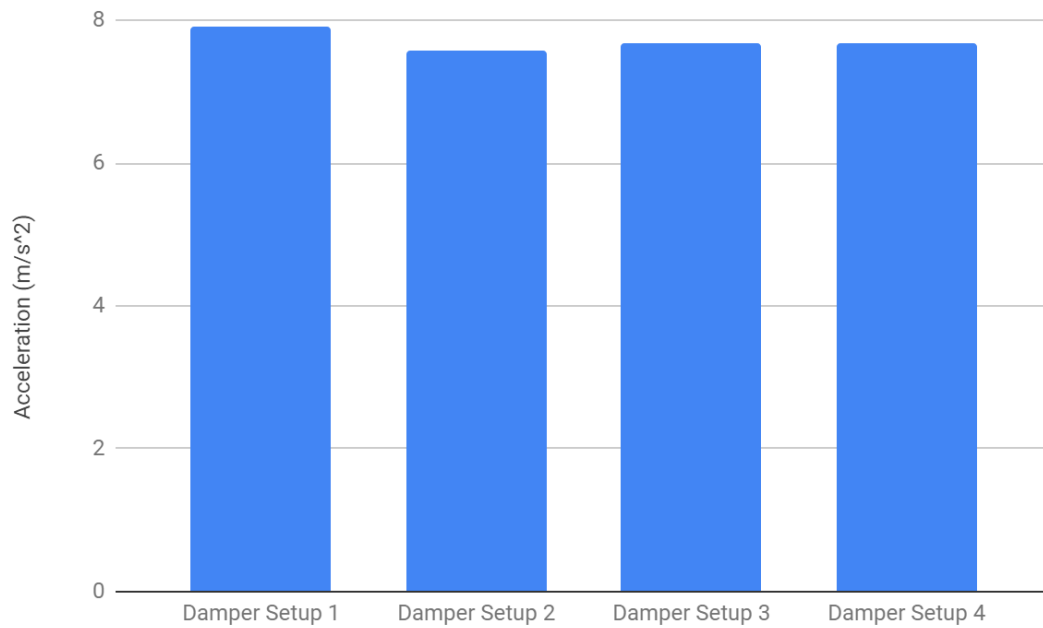


Figure 4: Acceleration at each damper setup

The weight transfer is also not affected a large amount by the change in setup. The graph shows that the lowest amount of weight transfer occurs at low front and rear damping settings. Since amateur drivers have a tendency to break under cornering, the data suggests that low damper settings front and rear should be used, since this causes the least weight transfer, so the car will maintain more grip.

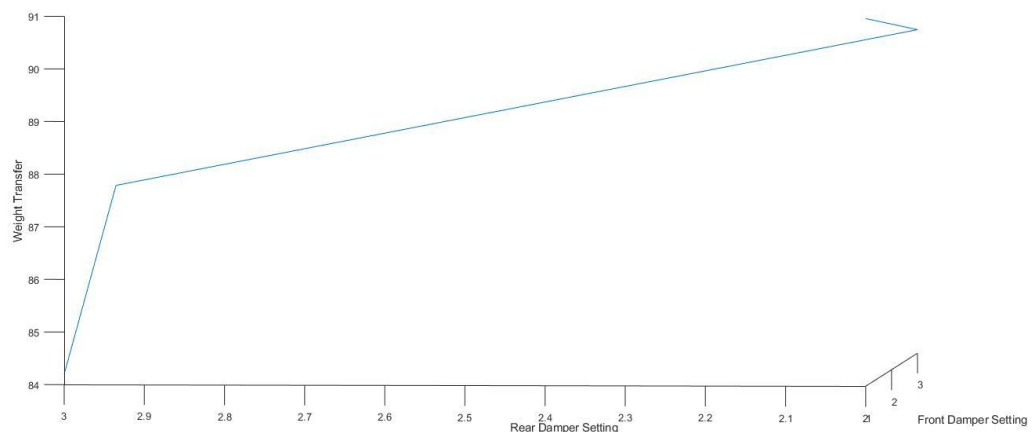


Figure 5: Weight transfer at each damper setup

Anti-Roll Bar

The group wanted to evaluate the effect of adding a rear anti-roll bar to the vehicle. The vehicle packaging would allow the use of one, however due to cost, this option

had not previously been investigated by Zentrack. Nevertheless, Zentrack was interested in knowing how this upgrade would affect drivability and lap times, to help them decide if the change would be worth the extra cost.

Since understeer can be used to evaluate the drivability of the car, this was measured at each rear anti-roll stiffness.

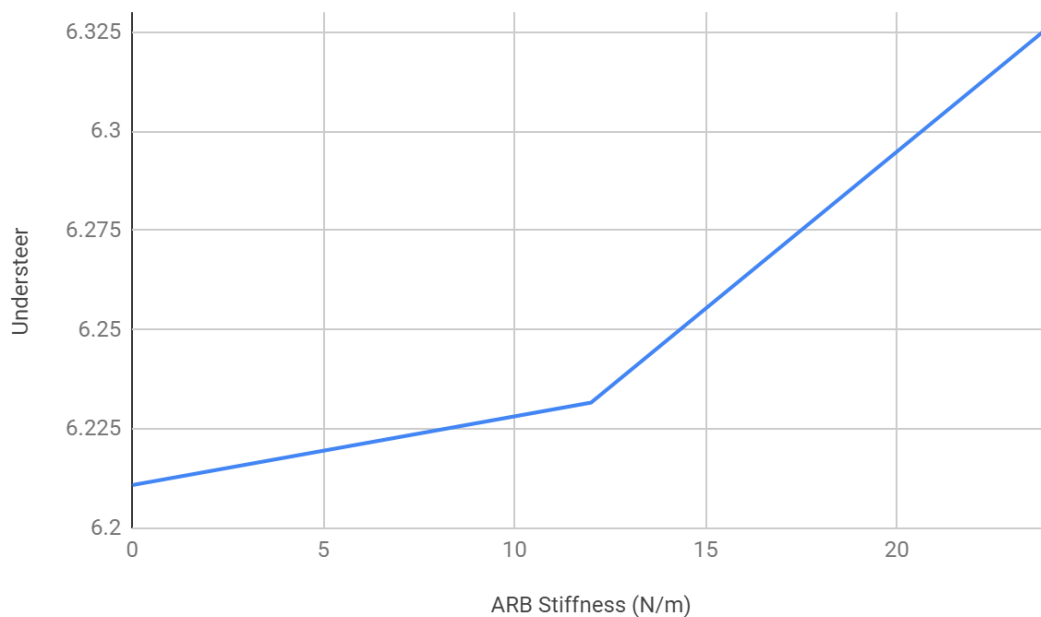


Figure 6: Understeer vs. ARB Stiffness

This shows that the best value of understeer is achieved at the lowest value of anti-roll bar. However Figure 7 also shows that the increased anti-roll bar stiffness improves the Magic Number (weight transfer distribution). For this reason, a combination of ARB and damper settings should be used for the best handling, to suit the driver.

Springs

The next area that was looked at, was to evaluate the effectiveness of changes suggested by the team. Zhou, was in charge of calculating the ideal spring rate for the car, with the aim of improving drivability and lap times. The suggested settings of 70 N/m front and 90 N/m rear was compared with the original settings of 62 N/m front and rear.

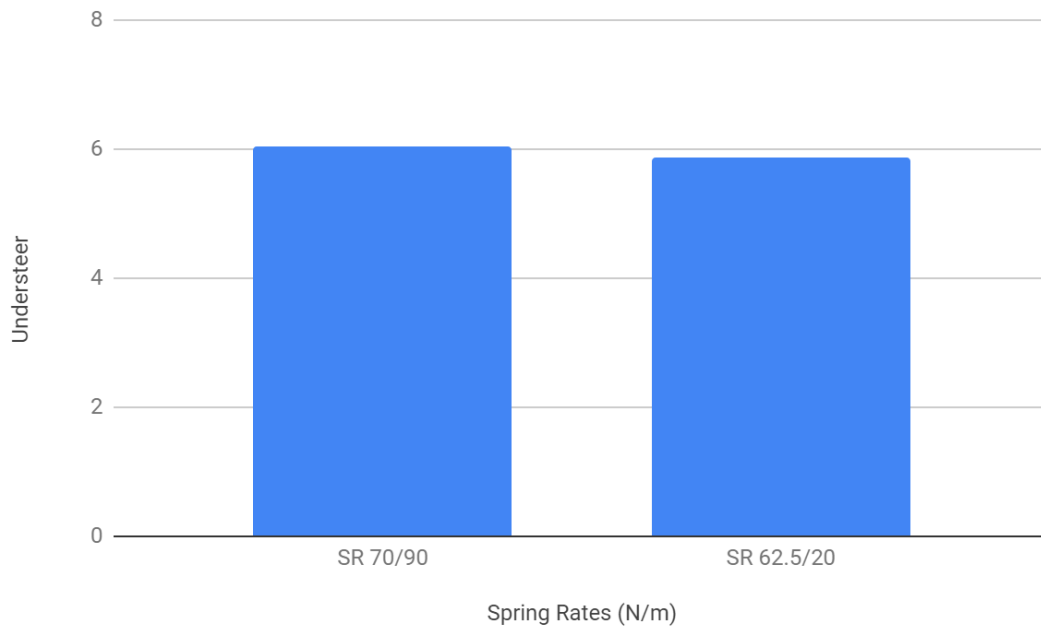


Figure 7: Understeer at each Spring Rate

The results show that these springs would increase the Understeer of the car. Since the current vehicle has an issue with Oversteer, this would make the car easier for amateur drivers to use.

Center of Mass

Finally, the simulation was used to investigate the effect of moving the centre of mass. This was an area requested for investigation by Zentrack. The task, which Harry investigated, was to move the centre of mass forward, to improve the rearward biased weight distribution. The figure below shows how this affected Understeer and Magic Number. It shows an increased Understeer and Magic Number, suggesting improved drivability for amateur drivers.

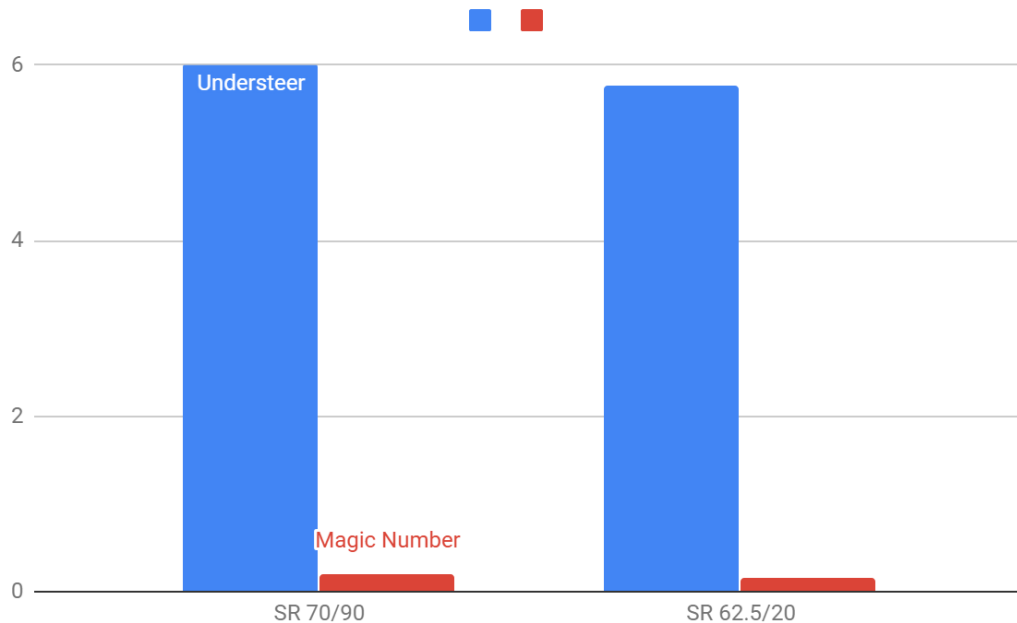


Figure 8: Magic Number and Understeer vs. Spring Rate

Evaluation

The simulation has allowed the group as an entirety, to justify suggested changes to the customer. It has enabled the evaluation of changes, to help the group understand the effect their changes would have. Without it, changes can only be understood on a component level, whereas the simulation allowed the group to see the effect the changes have on the entire vehicle.

The usefulness of this section was severely limited by the availability of data. Car data was limited, meaning that a large amount of areas had to be assumed or approximated. Due to the lack of access to track and dyno data, it was not possible to compare the simulation model to the real world, meaning that information received could only be used for comparative evaluation (better or worse) and not for absolute lap time improvements.

Despite these issues, the simulation was an important tool, as without it a deep technical understanding would have been required to help convince Zentrack of the effectiveness of our suggestions.

Oliver Duane

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Background

As one of Jeremy Clarkson's "Terrible 10 Stinkers" of 2015-16, the Zenos obviously had a lot of room for improvement (Clarkson, 2016).

After Clarkson's review, the company went into administration and was acquired by the international serial con-man and owner of AC Cars, Alan Lubinsky (AM-Online, 2008; Car Magazine, 2002; Farago, 2009; Stagno-Navarra, 2012; Yardley & Motavalli, 2006).

John Avery, presumably working for Mr Lubinsky, requested that the University help develop the car.

The latest version of the Zenos, called Zentrack, will apparently be ready for hire on June 1st according to Mr Avery's website performancecarclub.co.uk.

Analysis

As no performance data was provided, the Clarkson review, a video of a test-drive resulting in an horrific accident, and a video of "The Stig" were three starting points to analyse the characteristics of this car (Clarkson, 2016; ZENOS E10 BRAKES FAIL @ 150MPH: In car footage released by Stephen Singleton, 2017; TGTv s23: ride onboard with Stig in a Zenos E10 S: Top Gear, 2017).

Oliver's primary focus was to improve handling via the optimisation of suspension and brakes. The addition of a rear Anti-Roll Bar and brake bias control was considered to increase adjustability.

Some concerns regarding handling were:

Wishbone length

Differences in wishbone lengths at the front and rear will potentially cause imbalanced characteristics due to:

- the roll axis pitching and yawing undesirably, creating lateral forces during bump;

- unequal changes in camber, camber forces and the size of the tyre's contact patches (Dixon, 1991).

The Centre of Gravity and Weight Transfer

The friction coefficient of a tyre tends to reduce as load increases (see Figure 12)

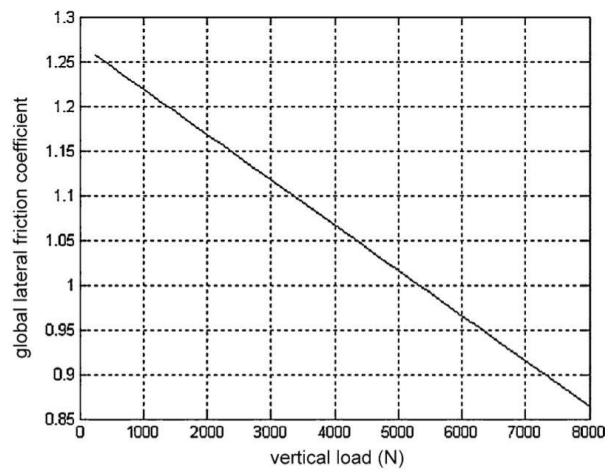


Figure 12 - Lateral friction coefficient / Vertical Load (Guiggiani, 2014)

Although the rate of decrease is unlikely to be linear (as it appears in Figure 12) due to the complexities of tyre modelling, it was assumed that it is an adequate assumption for a basic calculation of how weight transfer affects the total available grip. Using this assumption, a calculator was made to estimate changes in grip (See Figure 13).

Static tyre load (N)			Outer wheel			Inner wheel			
	3000	W.T. (N)	Load (N)	Fr. coef	max lat F (N)	Load	Fr. coef	Max lat F (N)	Total max lat. F (N)
Fr. Coef. at static load	1.12	0	3000	1.12	3360	3000	1.12	3360	6720
Mu loss per 1000N load increase	0.05	250	3250	1.1075	3599.375	2750	1.1325	3114.375	6713.75
W.T. increments	250	500	3500	1.095	3832.5	2500	1.145	2862.5	6695
		750	3750	1.0825	4059.375	2250	1.1575	2604.375	6663.75
		1000	4000	1.07	4280	2000	1.17	2340	6620
		1250	4250	1.0575	4494.375	1750	1.1825	2069.375	6563.75
		1500	4500	1.045	4702.5	1500	1.195	1792.5	6495
		1750	4750	1.0325	4904.375	1250	1.2075	1509.375	6413.75
		2000	5000	1.02	5100	1000	1.22	1220	6320
		2250	5250	1.0075	5289.375	750	1.2325	924.375	6213.75
		2500	5500	0.995	5472.5	500	1.245	622.5	6095
		2750	5750	0.9825	5649.375	250	1.2575	314.375	5963.75
		3000	6000	0.97	5820	0	1.27	0	5820

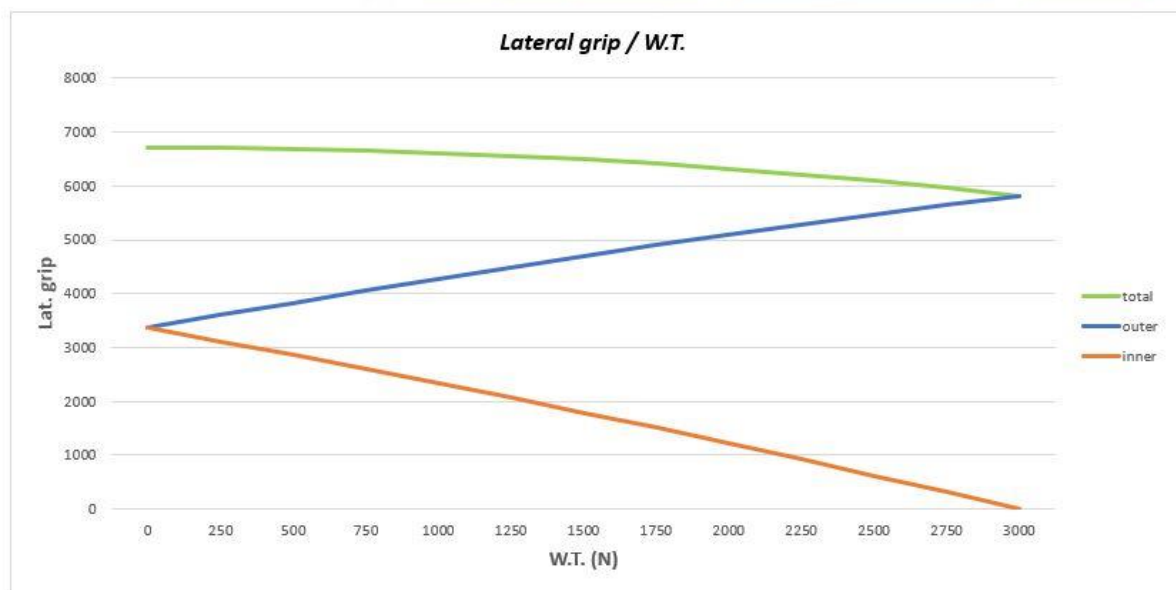


Figure 13 - Lateral Grip / W.T.

The graph shows that total achievable grip decreases non-linearly with an increase in weight transfer.

The problem of weight transfer reducing grip unequally between front and rear could be encountered as the CoG is relatively far back and high up, causing:

- more lateral weight transfer at the rear;
- less rear grip;
- oversteer.

This could be dealt with by softening the rear suspension (reducing rear lateral weight transfer) but would have the negative characteristic of:

- Increasing front-end weight transfer thus reducing front-end grip, especially when braking into a corner as the inside-front wheel would lock-up sooner due to the decrease in vertical load on that wheel;
- Increasing pitch change over bumps (see Natural Frequency below).

The extra lateral inertia at the rear could result in oversteer and cause a pendulum-like effect (due to the delayed counter-steering if the rear regains grip too quickly), potentially inducing uncontrolled yaw oscillation, or “tank-slap”. Porsches’ and Lamborghinis’ seem to be prone to this, and they also have around 40-60 weight distribution. However, they have significantly wider rear tyres which the Zentrack does not, suggesting that the Zentrack may just tend to oversteer into a spin - rather than the rear suddenly gripping and causing the trajectory and yaw of the car to rapidly change direction, faster than the driver counter-steers.

Natural Frequency

It is desirable, when hitting a bump, for front and rear to return to the ground (or “settle”) at the same time so that front and rear tyre loads are more quickly balanced, and chassis pitch changes minimised. This is achieved by the front having a lower / slower natural frequency than the rear. This optimal difference in front / rear natural frequency would be different for every speed but would always require a lower natural frequency at the front. (Wong, 1978).

In Stephen Singleton’s YouTube video, the car goes over a speed bump and appears to have a higher natural frequency at the front - the front of the car’s rise-and-fall, lasts approximately 6 frames and the rear rise-and-fall lasts approximately 9 frames suggesting a significantly softer rear (0:18s) (ZENOS E10 BRAKES FAIL @ 150MPH: In car footage released by Stephen Singleton, 2017).

The problem of pitch oscillations causes imbalance of:

- Cornering – the oscillating pitch would oscillate the front and rear tyre loads, grip and yaw;
- Braking – deceleration ability would be reduced due to the changing load and grip level of the front tyres;
- Acceleration - acceleration ability would be reduced due to the changing load and grip level of the rear tyres;

Brakes

In the same video, the driver is twice heard complaining about the brakes, and Clarkson calls them “the turd in the swimming pool” (Clarkson, 2016).

On one of the models that Mr Avery brought in for inspection, the rear brake calipers were positioned at the top of the brake disk which is very unusual (See Figure 14) (Brake Caliper Location - Explained, 2015). Unless the caliper was designed for this position, it could lead to difficulty bleeding the brakes properly as the position of the bleed nipple may not allow all the trapped air to float up to it.

The rear calipers had a larger piston leading a smaller one which is also unusual, as a larger and more forceful ($F = P \times A$) piston is required on the trailing side of the brake pad to equalise wear by overcoming:

- friction losses caused by the heat and gas that are produced by the leading brake pad;
- the moment produced by the friction, twisting the pad surface away from the disk surface.



Figure 14: Zentrack Rear Brake Caliper

In Clarkson's review he says that the brake pedal "feels like it's connected to not much at all" but that the front wheels are prone to locking-up (Clarkson, 2016). This could be due to a higher proportion of weight transfer at the front, a result of the necessity for a soft rear (as postulated above). (Clarkson, 2016)

Steering

Clarkson also complained of "very fidgety" steering, causing driver fatigue. This could be a result of a large kingpin offset (producing erratic torque in the steering wheel over uneven road surfaces) and no power steering to disguise it. (Clarkson, 2016)

Fuel tank

Mr Avery requested a fuel capacity that was large enough for endurance races (at least

60 litres) to be incorporated into the plans. This would ideally be behind the seats as the change in tank weight would have least effect on handling if it was as close as possible to the centre of gravity of the car. However, due to packaging constraints, it was suggested that an auxiliary tank was placed at the front end. This would lead to further problems such as:

- Changes in handling balance as the fuel mass changes;
- Complications to fuel supply;
- Increased weight of having another tank and associated/necessitated parts;
- Safety implications of fuel being at the front and having to pass the driver/passenger.

Work content

Optimising the braking system

To maximise brake performance during deceleration the correct amount of resistive torque must be applied to each wheel to utilise as much available grip as possible, while maintaining control of yaw and trajectory. The available grip, however, constantly changes in non-static situations due to:

- Tyre load changes due to weight transfer and road surface topography;
- Road surface grip levels;
- Tyre slip, temperature and surface condition.

This would require active control of each wheel which is beyond the realm of possibility in this project. For this reason, compromises must be made to simplify and generalise the requirements, in a way that suits the driver.

As the data available was vague and subject to change, a calculator was devised so that variables (e.g. car dimensions) can be input and required data calculated (e.g. size of front and rear brake disks, master cylinder etc.)
(

Brakes Calculator
(Laterally Symetrical)

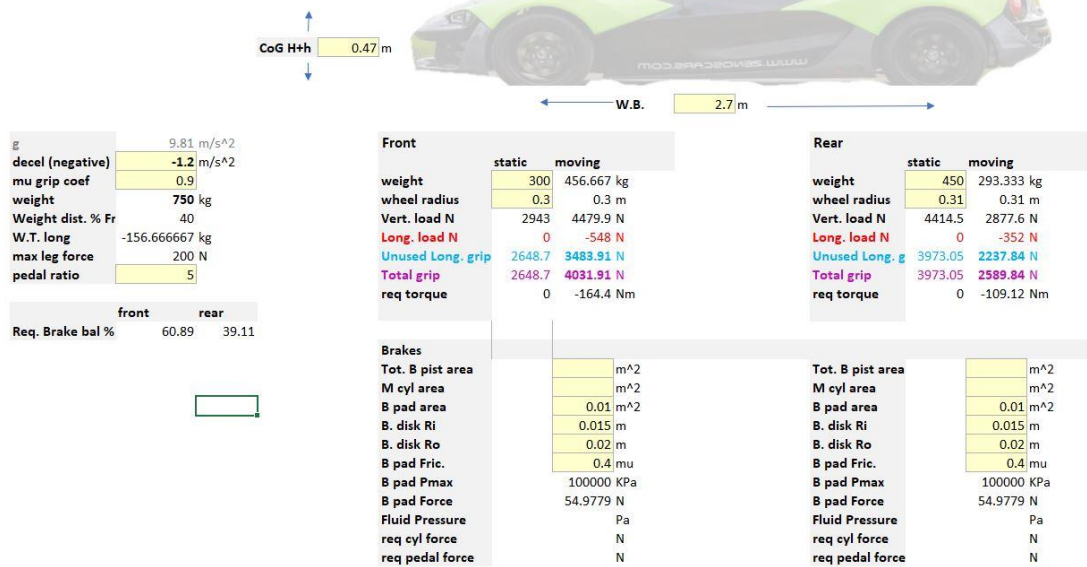


Figure 15 Brakes Calculator

Brakes Calculator
(Laterally Symetrical)

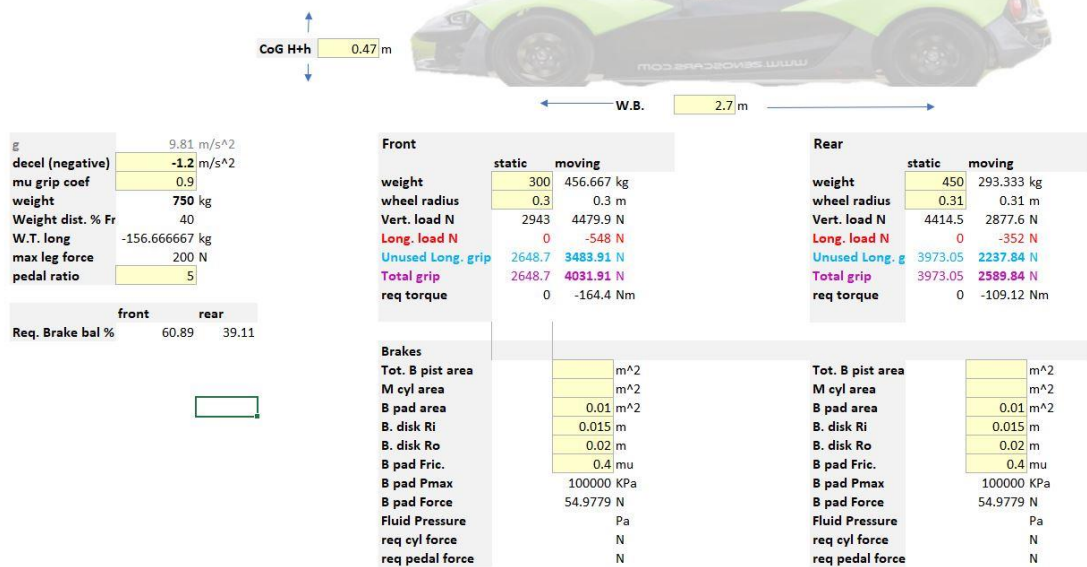


Figure 15.)

Brakes Calculator
(Laterally Symetrical)

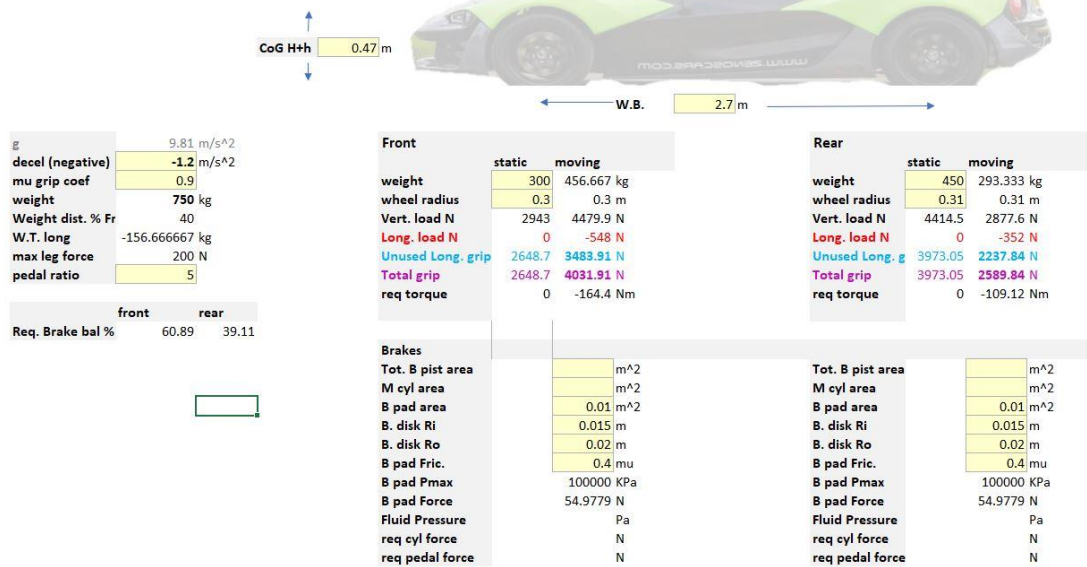


Figure 15 Brakes Calculator

Optimising Suspension

A similar calculator was begun, for calculating the natural frequencies of the suspension
(See Figure 16).

Suspension Calculator

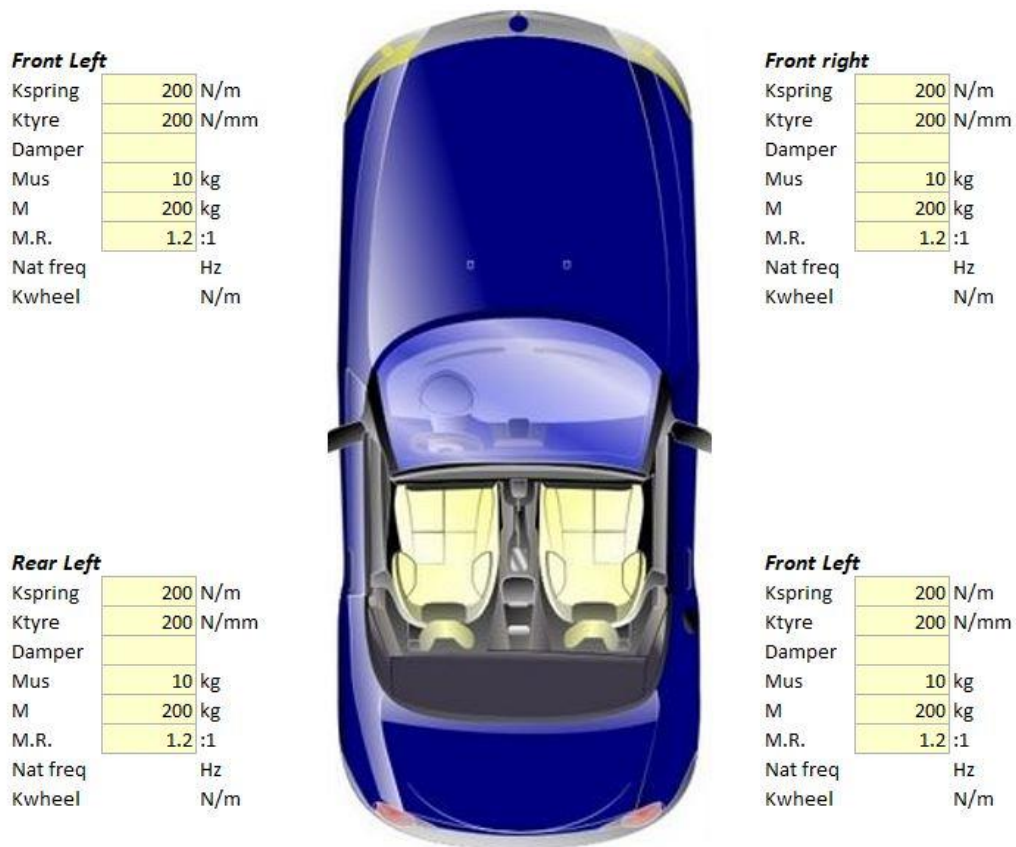


Figure 16 - Suspension Calculator

Measuring improvements

To measure performance improvements, the basic properties of performance needed to be identified. These can be broken down into *steady-state* and *dynamic* performance characteristics.

Steady-state characteristics:

- Longitudinal acceleration/deceleration and maximum speed at which it can be attained;
- Lateral acceleration and maximum speed at which it can be attained;
- Combinations of lateral and longitudinal accelerations/decelerations (i.e. decelerating into a corner) and maximum speeds at which they can be attained.

This data is available as raw data from the data logger and does not require any calculations.

Dynamic characteristics / Jerk – agility and responsiveness:

- Rate of change in longitudinal acceleration/deceleration;
- Rate of change in lateral acceleration;
- Rate of change for combinations of lateral and longitudinal accelerations.

This is more complicated to acquire the data and offers a variety of choices.

Calculating Jerk using derivatives

First, the total G is found (Lateral and longitudinal G are both taken as positive) (See Figure 17).

$\text{sqrt}(\text{sqrt}(\text{sqrt}(\text{sqrt}(\text{'G Force Lat' [G]})))+\text{sqrt}(\text{sqrt}(\text{sqrt}(\text{'G Force Long' [G]}))))))$



Figure 17 - Calculating Total G

To check the derivative function works as expected, the derivative of speed (acceleration) was used to compare with G (See Figure 18).

`derivative(-('Corr Speed' [km/h]/3.6))`



Figure 18 - Testing the Derivative Function

Using the derivative function with “G Force Total” at a rate of 10Hz (this is the rate at which results are continuous i.e. do not revert to zero between steps) (See Figure 19),

`derivative('G force total' [G])`



Figure 19 - Calculating Jerk

Performance graphs

To visualise and quantify overall performance, two 3D graphs were suggested:

- **Steady state performance graph** (see Figure 20) - similar to a friction ellipse, with the velocities of those points displayed on the vertical z axis (both graphs are based loosely on Lotus Exige MoTeC data).

Steady state performance

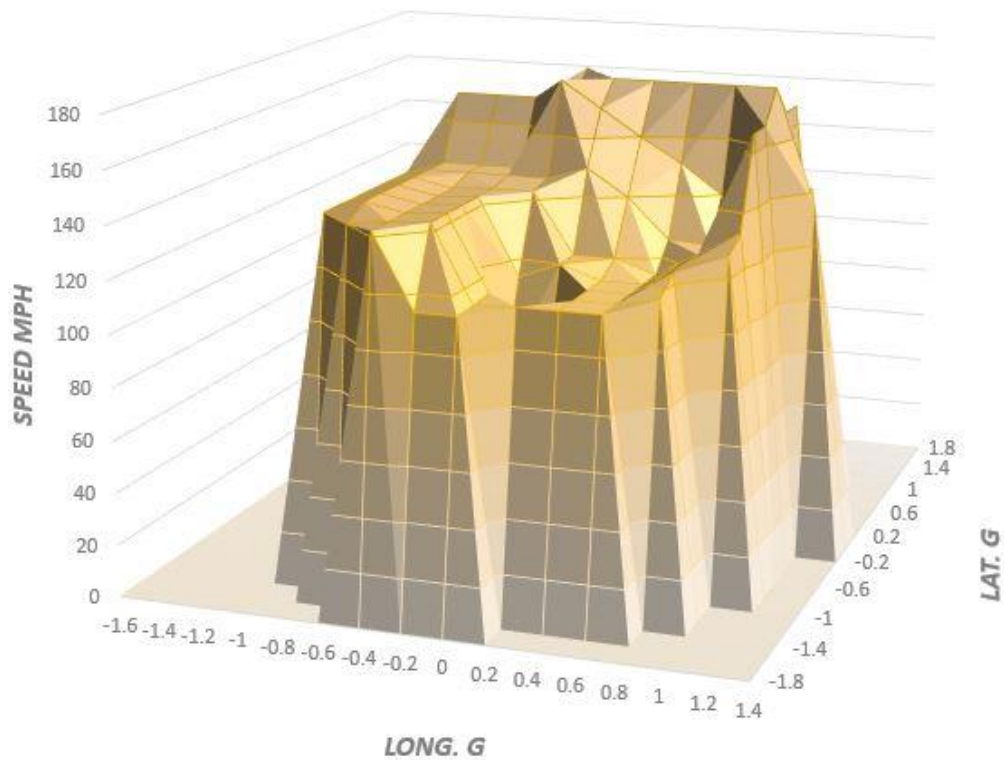


Figure 20 - Steady State Performance Chart

- **Dynamic performance graph** (See Figure 21) - shows accelerations (on the x and y axes) for any given velocity, but instead of *velocity* on the vertical z axis, would show the *maximum rate of change* (or "jerk") in achieving those accelerations.

Dynamic performance

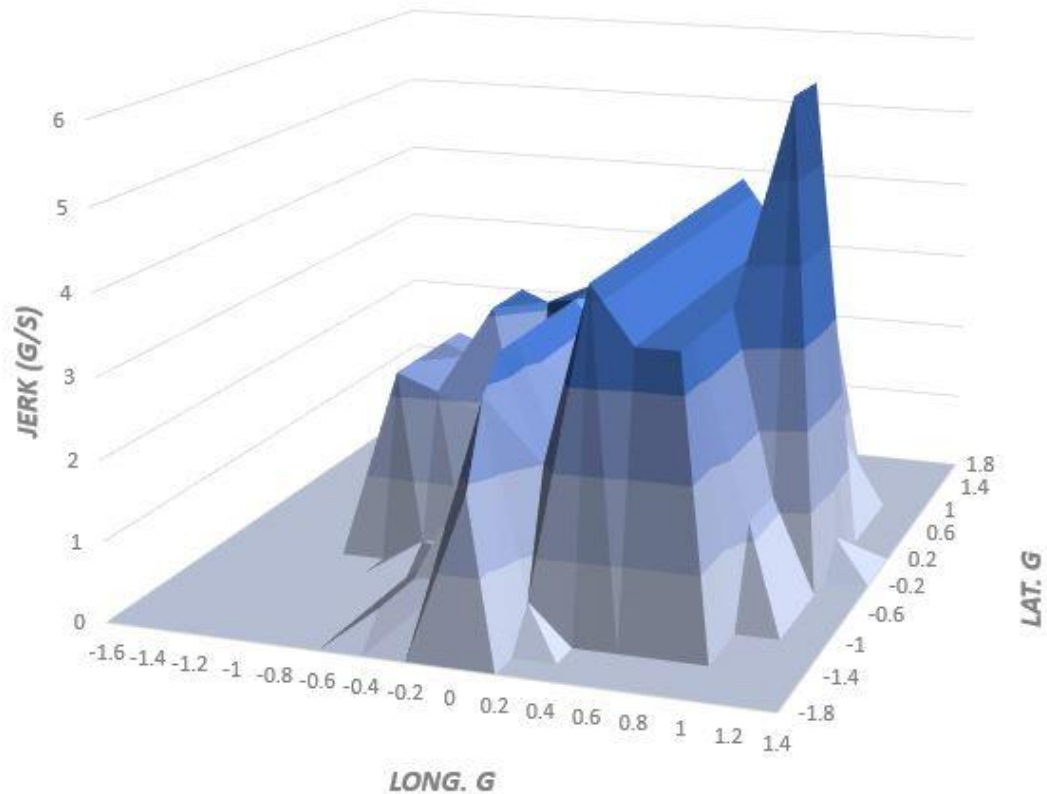


Figure 21 - Dynamic Performance Chart

The resulting volumes of these two 3D graphs would give an indication of the overall performance which could compare changes to the car, without being track-specific.

To gather enough data to populate the charts adequately, specific tests would need to be performed on the simulator using maths channels. The calculated data could then be exported to graphing software such as Matlab.

Testing

The original plan was to:

- Get actual data from a Zentrack on track;
- Simulate the car on the UWTSD simulator;

- Compare / validate / calibrate the sim model;
- Test modifications on the sim model;
- Implement viable modifications to the car and test on track;
- Compare new data with the simulation;
- Compare new data with the original data.

Unfortunately, none of these have happened.

- Mr Avery kept changing the car and failed to provide any performance data;
- A good simulation model has not been built;
- No baseline for performance has been identified;
- Any suggested modifications would be based on basic vehicle dynamics theory;
- Improvements could not be verified.

Contributions to project

- Contributed ideas;
- Attended initial meetings – extenuating circumstances limited involvement later in the project;
- Identified areas for improvement and ways to improve these areas;
- Identified ways of evaluating different areas of performance;
- Collaborative work such as taking measurements and testing dampers.

Further work

- Complete and combine calculators;
- Plot performance charts using Matlab; use more data.

Conclusion

Despite the disappointment of the project not going to plan, it has been a very interesting investigation.

The car appears to have inherent problems that compromise performance such as:

- **Lack of front / rear balance of:**
 - roll centre position, inducing unbalanced lateral forces to the tyres and roll centres during suspension movement;
 - percentage of W.T.-to-weight – The end with most W.T. will have a higher tyre load range (reducing grip) and the inside (unloaded) wheel will lose traction sooner (further reducing grip);
 - lateral grip level due to pitch oscillation – The alternation of load (and therefore grip) during the pitch oscillation could cause yaw oscillation.
 - Weight distribution and tyres – Cars with more weight at the rear often have a similar increase in the size of the contact patch, i.e. a larger diameter and width of tyre at the rear. The Zentrack front and rear tyres are very similar but with different loads. One is unlikely to behave like the other under such different load conditions.
- **The use of a front-engined-road-car engine in a performance car** – As most road cars are not designed for performance, they often understeer for safety reasons. This means that the height of the engine CoG does not matter so much. Using an engine like this at the back therefore increases oversteer. The gearbox and clutch are contained in the same unit which by necessity is very close to the rear axle. The result of this and the high CoG (of the engine), means a disproportionate amount of weight transfer on the rear axle.

Suggested improvements

- Increase rear track width – Reduces rear weight transfer;
- Increase rear wishbone length and shorten front wishbone length – Equalises roll axis behaviour;
- Increase wheelbase – Less pitch oscillation and more time to respond to oversteer; less longitudinal weight transfer;
- Increase rear tyre width (proportional to load) – Accommodates for larger rear loads with increased volume of air in the tyre, reducing pressure differences between front and rear; Increases rear contact patch (proportional to load).
- Reduce kingpin offset. – Reduces torque feedback to the steering wheel.

Steven Wilkinson

Trifilar Pendulum rig.

For my own part of the project I am going to be investigating Rotational inertia and how it affects objects associated with the clients vehicle a Zenos E10 .

I will be conducting an experiment using a trifilar pendulum to rotate components through a prescribed arc then calculating their time period.

With this I will then be able to calculate their moment of inertia and add the results to excel to produce graphic data to show my findings.

Tasks associated with the project

- Researching into Rotational Inertia and its relevance in the automotive industry.
- Plan and execute an experiment to test the components.
- I will be investigating how mass affects automotive components in regards to its inertia and what changes could be made to improve theses.
- How changes in cable length affects the time period of an object.
- Calculate the moment of inertia for the individual components as accurately as possible.

Rotational Inertia in the automotive industry.

Within the automotive industry there are many examples of rotating components that are used in all vehicles. Wheel, brake discs and flywheels are the most commonly used. These are all accelerated and decelerated during their use, it is this change in velocity that reduces available engine power and most of all it adds unwanted mass to a vehicle if the component is over engineered and could be lightened or altered.

An objects Moment of Inertia is its resistance to change in velocity, for example a flywheel with a higher Moment of inertia will take longer to transfer energy and get up to the required RPM that the driver has requested. However a smaller diameter flywheel would take less time and also less power would be wasted in rotating it.

This is why in high performance cars the flywheels have holes in the outer part to bring the rotating mass closer to the centre, thus reducing the Moment of inertia.

The trifilar pendulum

Setting up the rig.

The simplest way to find out a components Moment of inertia would be to construct a trifilar pendulum rig.

A trifilar pendulum rig uses three equal length wires to suspend a base plate off the ground so it can freely rotate around its axis.

The wires used are 3.5mm in thickness as this not only allows them to suspend components of up to 85kg, but they are also thin enough that the wires own torsional movement when it is twisted is ignored in the experiments results.

The wires are put through drilled holes into the base plate at the same equal distance from one another and then are held in place by U clamps. This will stop the plate from moving during any testing.

A second plate that is identical to the base plate is clamped and attached to the supporting posts of the gantry and wires are put through it in the same way as before. Having a second plate to attach the wires to instead of another methods such as fastening the wires to the gantry allows for the wires to remain straight and it reduces the chance that they cause any errors in the results.

To make sure that the base plate was level I used a digital protractor to measure the plate, I allowed for at least 1 degree of tolerance between the three sides. I was able to reduce this to 0.5 of a degree.

Experiment procedure

Each component was weighted before used in the test as I would need the weight to calculate the Moment of inertia. It was interesting to witness how different massed objects rotated under the same prescribed arc.

I then put the object onto the rig making sure it was exactly in the centre of the base plate using a steel ruler measuring the radius away from each point.

Once I was happy with the set up I rotated the plate and mass 180 degrees to load the rig.

As I released the base plate I also started a stop watch to count how many full oscillation the plate made within 30 seconds.

The amount of oscillations made was then divided by 30 to calculate the time period that the component had.

I ran the test three times under the same conditions and used the results to get an average time period. The reason I carried out the test three times was because I felt that this would give a more accurate reading. I noted down the results of all the components

and then added them into an excel document for further analysis.

Development to the rig.

During the initial testing process it was quickly found that the top plate was flexing or twisting as the test was being carried out, this allowed for errors to occur as the rig wasn't stable enough. To combat this, the top plate was directly clamped to the top bar of the gantry thus removing all flex.

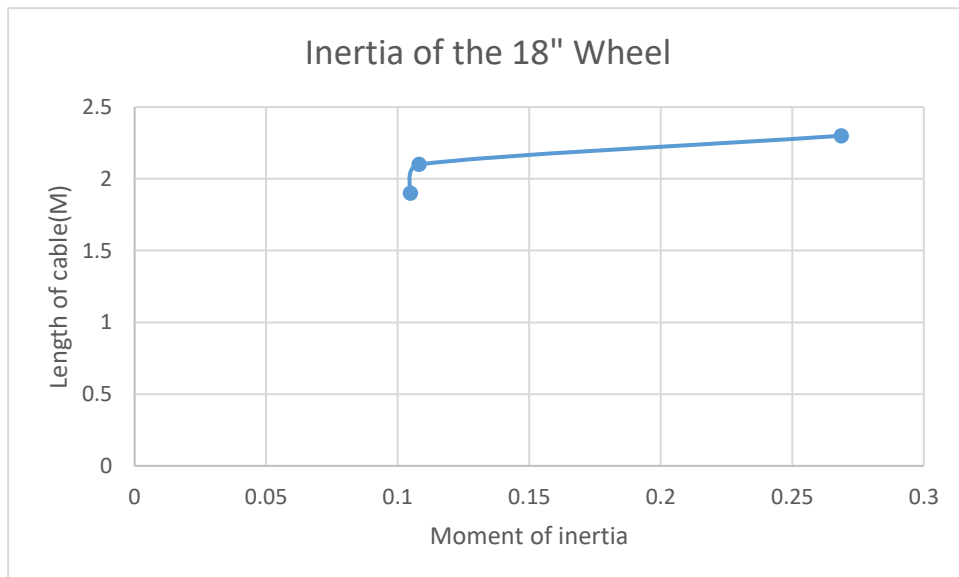
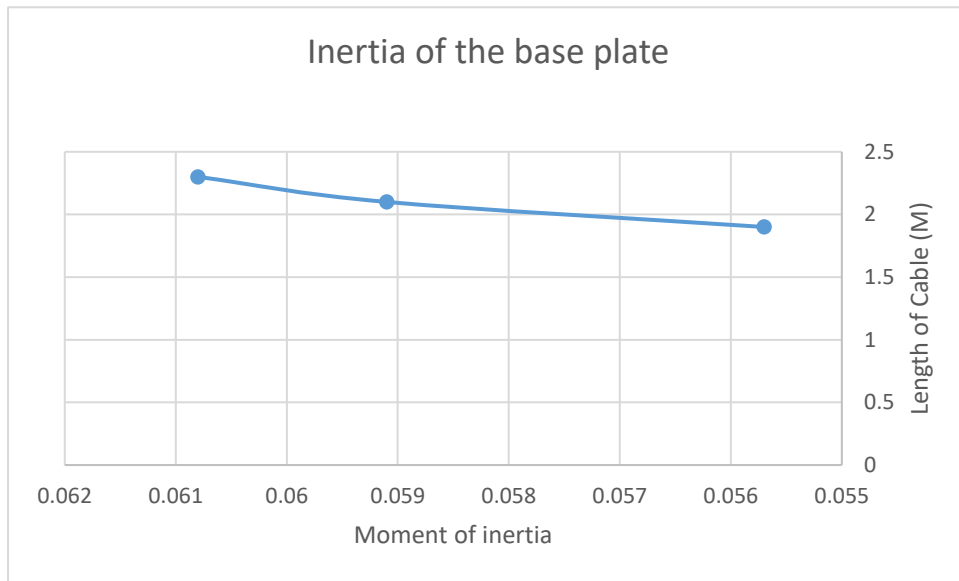
Another development that the rig underwent was that the U clamps were not strong enough and did unfortunately slip. This meant that 6 new stronger clamps were manufactured using a 3.5mm drill hole in M10 bolt and nut. This then resolved the issue. All of the components were retested using the same procedures as the first test attempt.

Changing the length of cable.

I then conducted a test to see if changing the length of the cable would have an effect on the way a mass rotated during a test run.

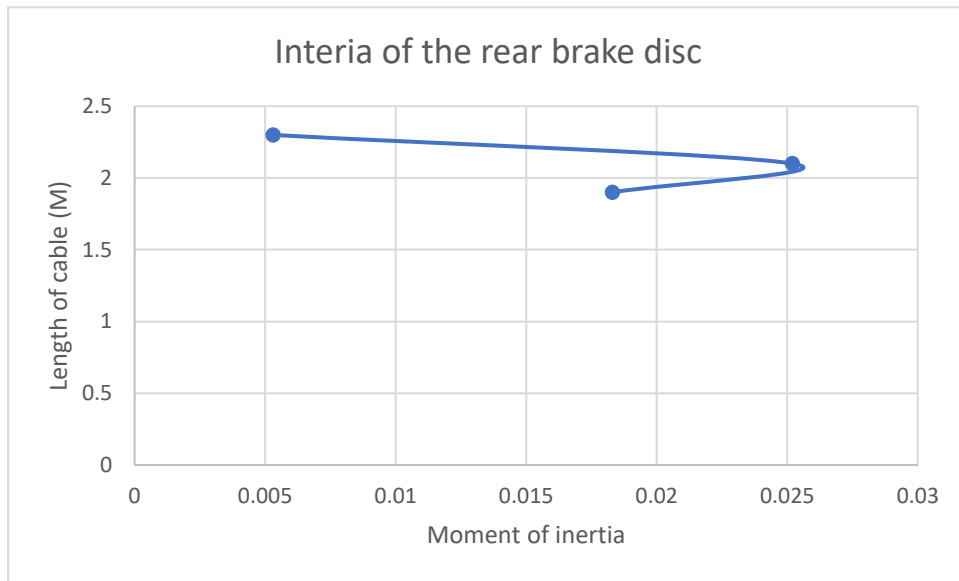
I set my initial cable length at 2.3m which was the highest cable length available in relation to the full extent of the gantry. This I meant that the base plate was allowed to freely be suspended without obstruction.

After testing each component three times to get an average I raised the plate up 20cm increments thus reducing the cable length and restricting the cables ability to twist freely. I had originally planned to reduce the cable length down to 50cm however at 1.7m the plate was oscillating to quickly and thus was deemed unsafe to continue to reduce the cable any further.



The graphs above show that as the cable length decreases the Moment of inertia of the rotating mass also decreases.

When testing the 18" wheel the cable was getting caught by the wheel and affecting the results which you can clearly see on the graph.

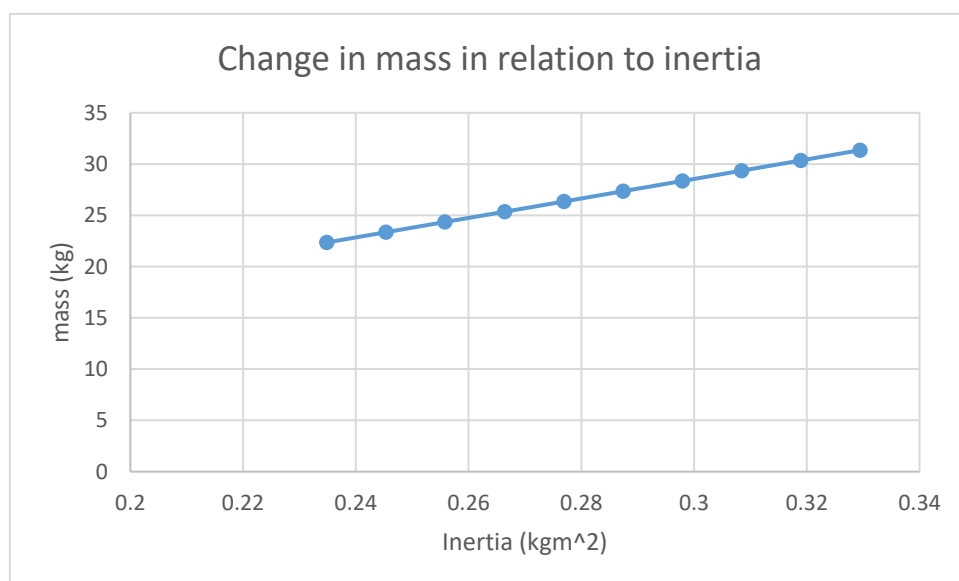


When testing the rear brake disc the results produced an error as when the cable was at length 2.1m and below it started to rotate fast enough for the disk to move.

Mass affecting Inertia.

The second different variable I wanted to investigate during the experiment was how mass affected the inertia of a component that the client was using. This would give me an idea on if reducing the mass of the component actually had any benefits to changing the moment of inertia.

I was not able to conduct the test for real so these results are theoretical, however they do give an insight into what happens when mass is the variable and all of the other time period, length of cable and radius stays constant.



I used the 18" wheel as a good component to test as it was the heaviest in mass. I started with the original mass of 31.35kg and reduced the mass by 1 kg at a time. The results show that it reduced the moment of Inertia by 0.105kgm^2 each time. This did not take into account any changes in time period that could have been present; however it does prove that a reduction in mass also reduces the moment of inertia.

Errors in the test.

There were a number of errors which presented themselves at various points during the tests. These would have all affected the final test results to some degree.

- The shutter door to the workshop the test was being carried out in was constantly in use. This meant that the door was being opened and closed while a few of the tests were being carried out. This would have meant that a breeze would have blow through to the testing area causing an error in the results as it would have slowed down the plate during its rotations.

If I was to carry out these tests again I would ensure that I was alone in the testing area. This would reduce the chances that an unwanted wind would affect the test results.

- The plate might not have been rotated to exactly 180 degrees each time and this would have caused some test runs to have more pre load than others. This would have given a false result on how many oscillations were made during a set time limit. This is down to human error and more care would be needed next time to ensure that the plates are always rotated from 180 degrees each test.

- There could be potential human errors in counting the oscillations, especially when the lengths of cable were reduced thus increasing the time and speed of oscillation. This could be rectified with an electronic device that counts when an object breaks its beam. Another way to rectify this would be to have more than one person counting the oscillations. That way the number of oscillations would match one another's.

- Lag between releasing the plate and starting the stopwatch. Again this is down to human error and care would be needed to try to ensure that the starting of the stop watch matches the time of release of the plate.

- The components may not have been truly in the centre of the base plate? This would affect the results as the mass would not be rotating centrally on the plate.

- The cables could have been different lengths this would have affected how level the base plate was.

- After the rig was set up a number of times the cables began to fray and come apart, this caused issues such as trying to get them through the holes made to clamp them. This

would have affected the weight distribution of the mass as it would not have been oscillating evenly.

- The cables began to unravel and this would have affected the way it restores torque and oscillated, affecting the results as more resistance would have been present.
- The clamping bolts were tightened to different amounts of torque. This would have been a minor error in the calibration of the base plate in potentially raising the plate up.
- Not accurate way to set the height of the base plate and then clamp it to set it. This was very fiddley and would have made small errors in its height adjustments.
- Base plate flexed when a mass was put onto it. This made the plate not completely level when it rotated.
- Errors might have happened because the gantry was slightly swinging during testing procedure.
- The results are not accurate as the time period is not just the component but the plate + component. It is not very efficient and does not give a true indication of the moment of inertia of the component itself.
- The cables got in the way of the component as it was rotating, as the cable was shortened this happened more frequently. This prevented the component from being able to oscillate freely.

Thoughts on the experiment.

I feel the Trifilar pendulum is a good way to obtain quick inertia figures. The rig itself is very simple to build and it's very easy to use. It's very interesting to see how different objects rotate in relation to their mass or the length of cable they are suspended from.

On the other hand, the results that were obtained from my experiment are not very conclusive as there was a lot of errors or faults which lead to a bad set of results. The base plate was a triangle which was easy to manufacture but made it difficult to place components equally and in the centre of it.

The components that were tested needed to be securely fixed into position to reduce or stop any movement while the plate was rotating.

This would also increase the safety of the experiment as I could not increase any higher than 1.9m in cable length due to the components rotating too fast and risk falling off.

Any adjustments to the cable length or the base plate lead to a long time spent recalibrating the level of the plate.

My contribution has benefited towards the progression of the group as my Moment of inertia findings have helped generate the Simulation model of the Zenos. By using an experimented result rather than guessed figures from data found in other sources has made the level of potential accuracy of the model increase. My results can be used as a base model for further improvement and refinement as this has given some direction to continue with should the need arise.

Better communication between the client and the team, if prior knowledge of equipment being taken away by the client was notified then arrangements could have been made so items such as the front wheels and brake discs could have been tested before being taken would have led to more detailed evaluation and comparing the difference between front and rear of the vehicle.

Further work.

If I was to continue with this project I would like to revisit this Rotational inertia experiment and refine it and improve upon it.

- I would firstly use a circular plate to suspend my components on as it is far easier to get more accurate results.

- I would also use electronic timing to improve the accuracy.

- Starting each test run at exactly 180 degrees.

- Improve the clamping design so it did not damage the cables or caused the plate not to be level.

- I would fix each component to the base plate to avoid it from moving and affecting the results.

- Care would also be needed on my part to ensure that human error is at a minus.

Zhaocong Sun

Work Content

My work is divided into 2 parts. The first is about wheels and the second is about evaluation to the whole vehicle.

I have studied the side slip phenomenon and the curve of side slip angle and side slip force. Then I researched on the influence that the tire structure and working conditions will have on the side performance. After measuring the parameters of our vehicle, I have built a model in SIMPACK, and start a simulation to see the lateral dynamic performance. Finally, I proposed some ideas to improve some parameters of the components.

I investigated and concluded the evaluation methods about how the driving performance improves with the changes of all components. For example, brakes, suspension, material, cell construction and so on. Actually, what I want to do is "Evaluate effectiveness of improvements". I have built a SIMULINK/MATLAB model to do the overall driving performance simulation, and see how much we improved the vehicle.

Some graphs and figures are chosen from the software, and analysis for some of them is added.

1 Wheels

When crossing a curve, the vehicle body will roll to the other side. Even if the steering wheel angle is unchangeable, the rear wheel rim will rotate at a small angle, which is perpendicular to the ground axis. This is because when the car is tilted, the front suspension guide bar will move and have deformation. The rotation of the wheel rim plane is called the roll steering. The parameters of wheels include: Elastic side angle, Roll Steer Angle and Compliance Steer Angle, which are worth researching.

The side deviation of tire means direction of the wheel deviating from the wheel plane although the adhesion limit is not reached in time without the lateral elasticity of wheel. The angle between the tire plane and direction is called side angle. The size, form and structural parameters of the tire have a significant effect on the cornering stiffness.

The above theory gives us a basic idea about what kind of research could be carried out. During my work, I will focus on the force and torque on the tire and wheel, and give some suggestions for the producer.

The line stiffness of the suspension means the total elastic resilience of the suspension system of the car. In my simulation of SIMPACK, the damping coefficient is about $2984.3 \text{ N} \cdot \text{s/m}$. At high speed, the damping coefficient is about $1025.4 \text{ N} \cdot \text{s/m}$. The spring stiffness of front wheel is 70000 N/m while the spring stiffness of rear wheel is 90000 N/m . The value of stiffness comes from ZHOU, which may not be very accurate because we couldn't find any other 3-D model or more accurate parameters about the trunk.

1.1 Parameters

Table 1 The parameter used in the calculation

Name	Value	Unit
Stiffness of Front Wheel	70000	N/m
Stiffness of Rear Wheel	90000	N/m
Damper at High Speed	1025.4	$\text{N} \cdot \text{s/m}$
Damper at Low Speed	2984.3	$\text{N} \cdot \text{s/m}$
Mass	As Shown in Appendix	Kg
Velocity	As Shown in Appendix	m/s

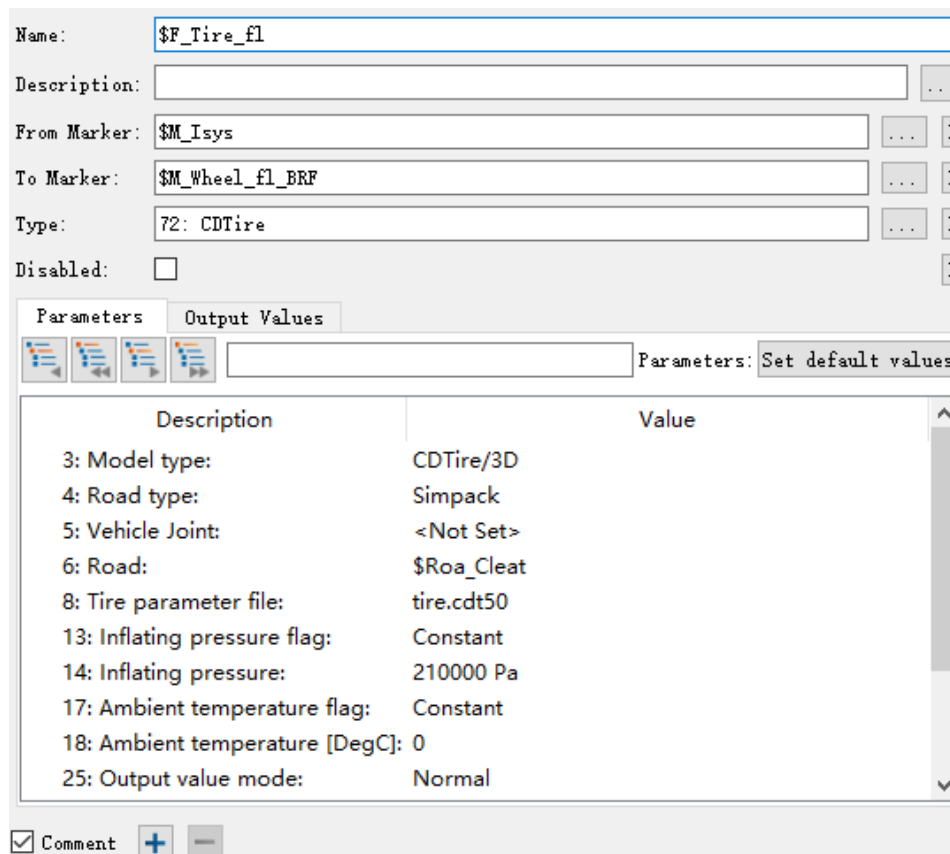
The stiffness and damper comes from ZHOU, and the mass and velocity is measured in the workshop. The value of other parameters is shown in the appendix.

1.2 Simulation

I used SIMPACK to build the model. Before continuing my report, I have to introduce that software. SIMPACK is a professional software for building a vehicle model. Unique vehicle elements for modeling enable us to build dynamic models for cars, trucks, buses, motorcycles, etc. quickly. SIMPACK could be used to analyze the problems of vehicle dynamics in low frequency range and problem of durability in high frequency range. SIMPACK has been widely used in suspension design, bench test, drivetrain, powertrain, steering stability, NVH analysis etc. It could also be linked to another software and carry out combined simulation, such as SIMULINK, MATLAB, and SOLIDWORDS. Additionally, modules from third-party could be integrated. SIMPACK integrates tire mechanics models designed by many companies effectively. Take my simulation as an example, the F-tire model comes

from COSIN Company. The tire force element model used in this report comes from F-tire Company.

Firstly, we have to input some value from the appendix, such as camber, stiffness and so on. Then, I build the tire& wheel model in the SIMPACK. Both of the 1/4 model and entire vehicle model are built to see the forces and torques on the wheel. We just need to input some values and parameters to the software. This is because all the setting and parameter are integrated into the software.



Name:

Description:

From Marker:

To Marker:

Type:

Disabled: ☐

Parameters Output Values

Parameters:

Description	Value
3: Model type:	CDTire/3D
4: Road type:	Simpack
5: Vehicle Joint:	<Not Set>
6: Road:	\$Roa_Cleat
8: Tire parameter file:	tire.cdt50
13: Inflating pressure flag:	Constant
14: Inflating pressure:	210000 Pa
17: Ambient temperature flag:	Constant
18: Ambient temperature [DegC]:	0
25: Output value mode:	Normal

☒ Comment

Figure 1 The parameter settings for the entire vehicle in SIMPACK

From the above figure, we know that we could define the road, where the vehicle will drive at.

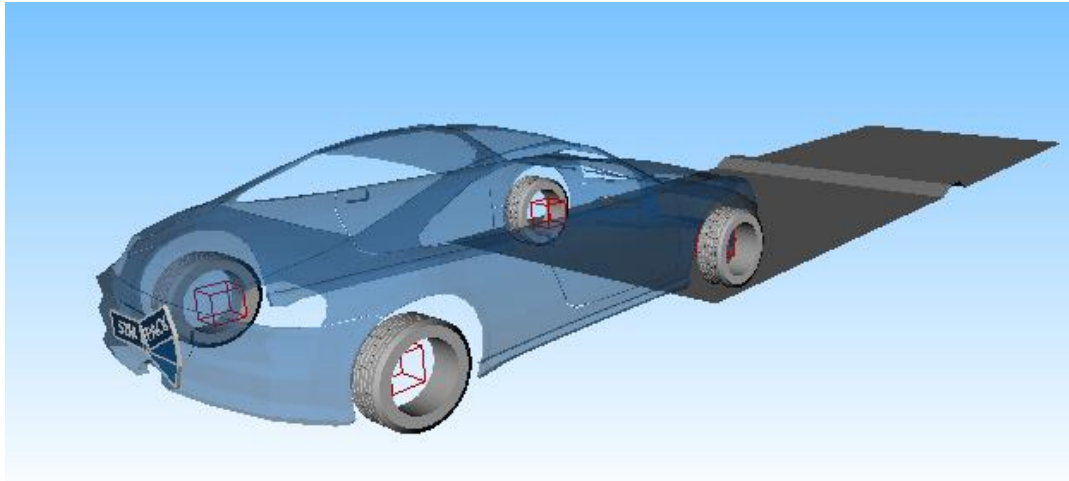


Figure 2 The 3-D entire model

Figure 2 shows the overall shape of the model. I set the road into a bump.

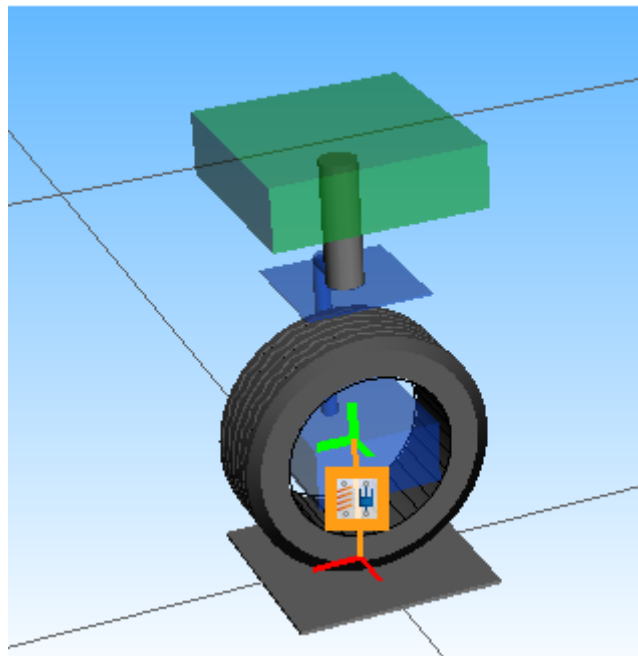


Figure 3 1/4 model 3-D image

In figure3, which is 1/4 model in SIMPACK, I set the support mechanism, spring, damper, wheel and tire. The parameter setting process is the same as former. But maybe we could change to another tire force element model.

1.3 Analysis for the result

After the settings, I ran the software and finish the simulation in 30 seconds. Then, I moved to post-processing

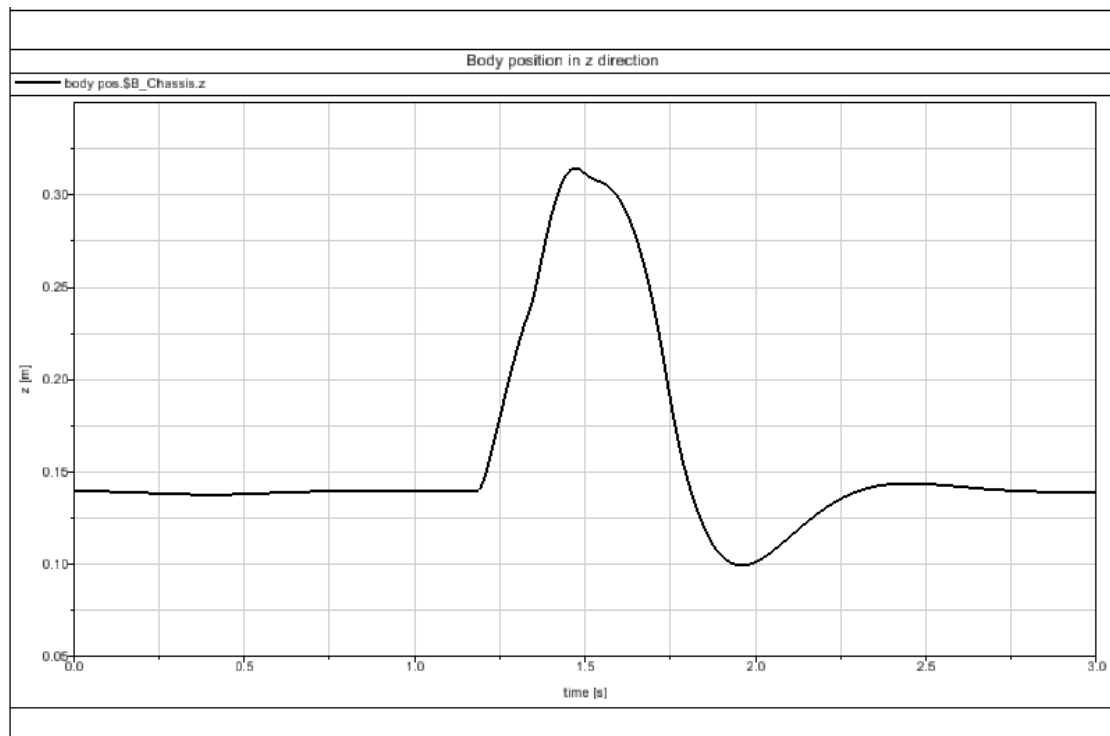


Figure 4 Body position in z direction

Figure 4 shows the body position in z direction. When the vehicle cross a bump, the main body will be lifted first and then be lowered. The tire will recover original pressure and height several seconds after the bump.

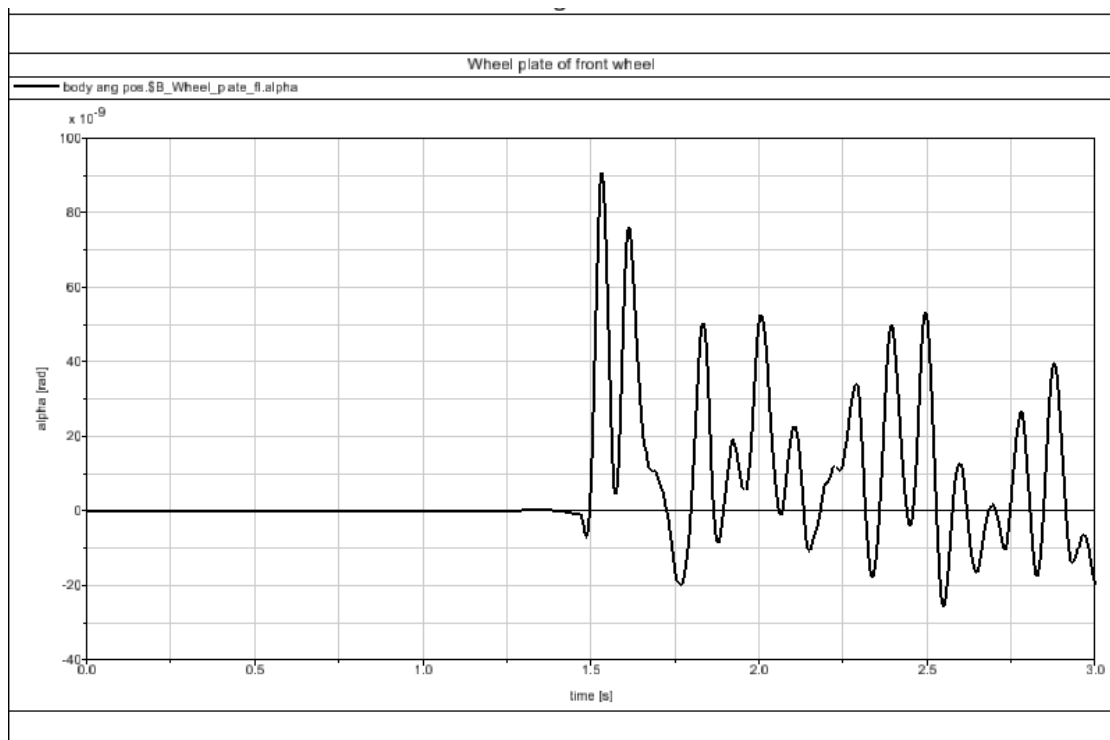


Figure 5 Wheel plate angle of front wheel at alpha

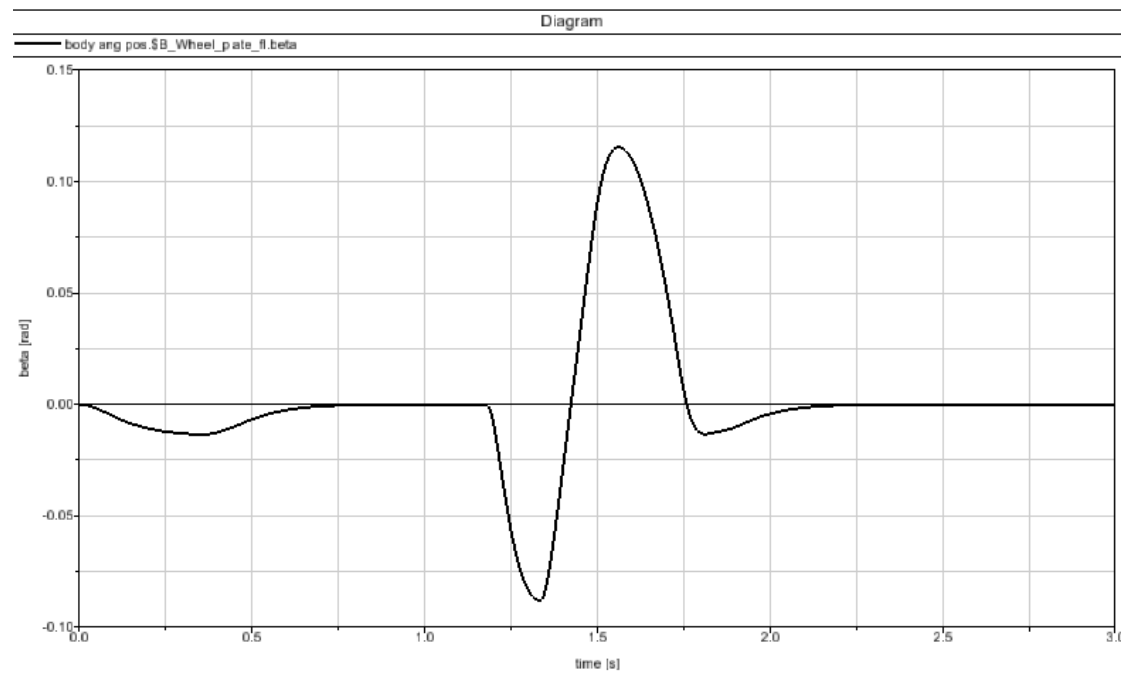


Figure 6 Wheel plate angle of front wheel at beta

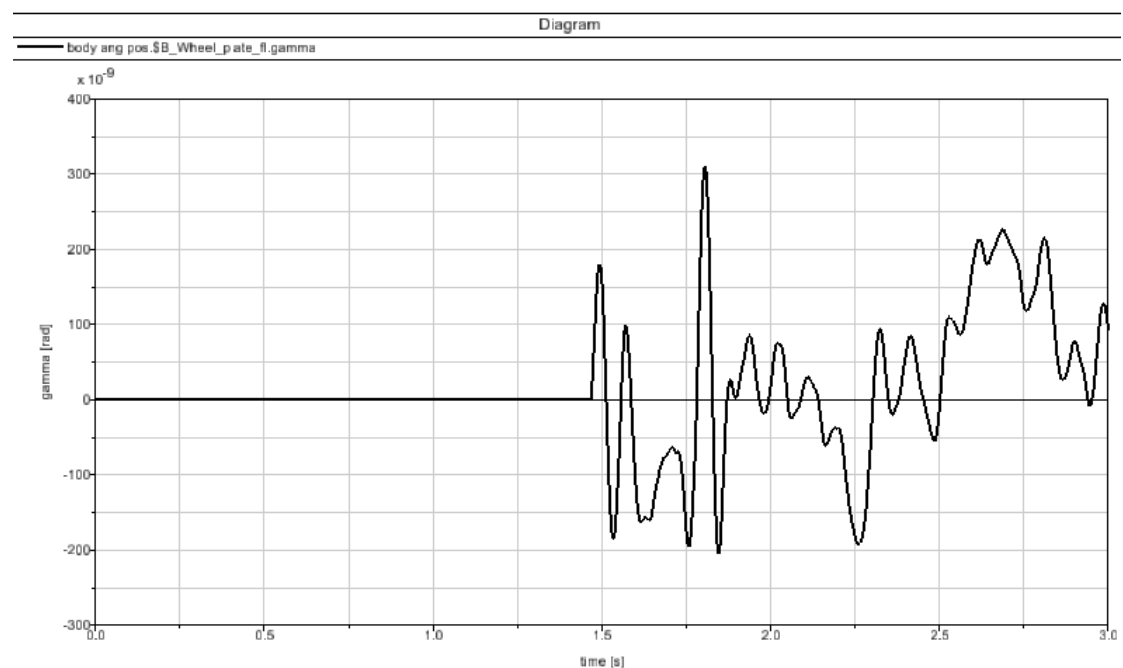


Figure 7 Wheel plate angle of front wheel at gamma

Figure 5-7 shows the wheel plate angle of front wheel in three directions. We could see that, the angle change in x and z direction is very small. But the angle change in y direction is very large, which means the steering wheel will change the direction once when crossing the dump. As we could see, the angle goes back to 0 after 1 second.

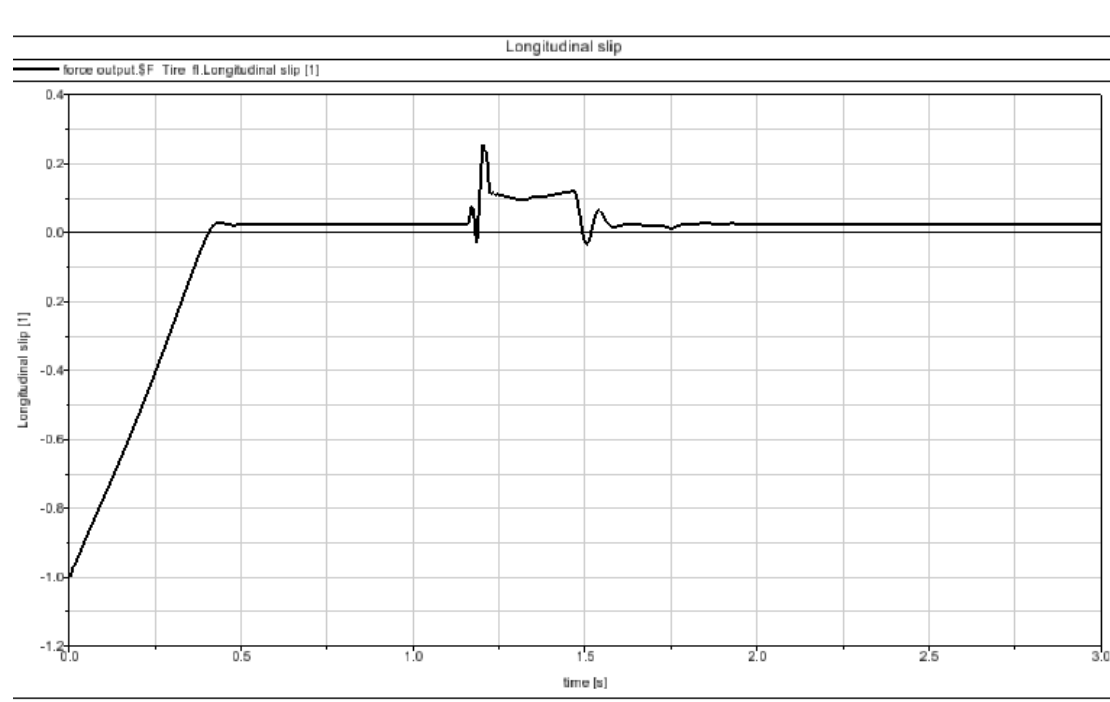


Figure 8 Longitudinal slip for the vehicle

Figure8 shows the longitudinal slip for the vehicle. At the start of crossing the dump, the longitudinal slip reaches the highest value.

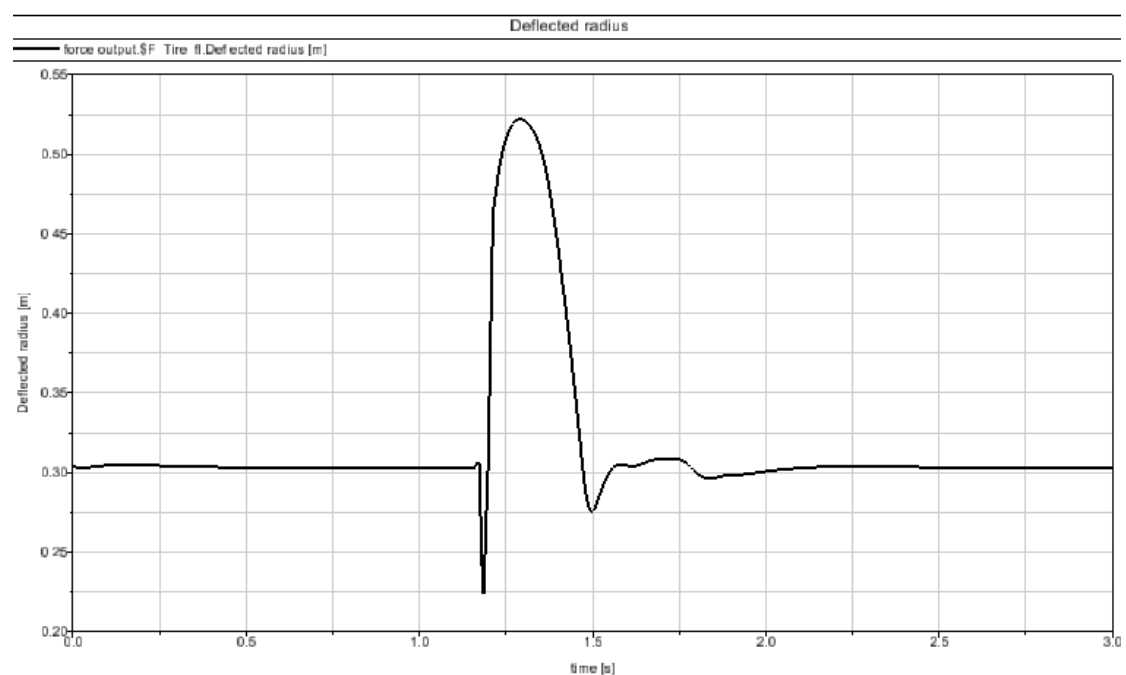


Figure 9 Deflected radius

Figure 9 shows the deflected radius. At the beginning of crossing the dump, the deflected radius dropped a little and went back to the highest value in a very short time.

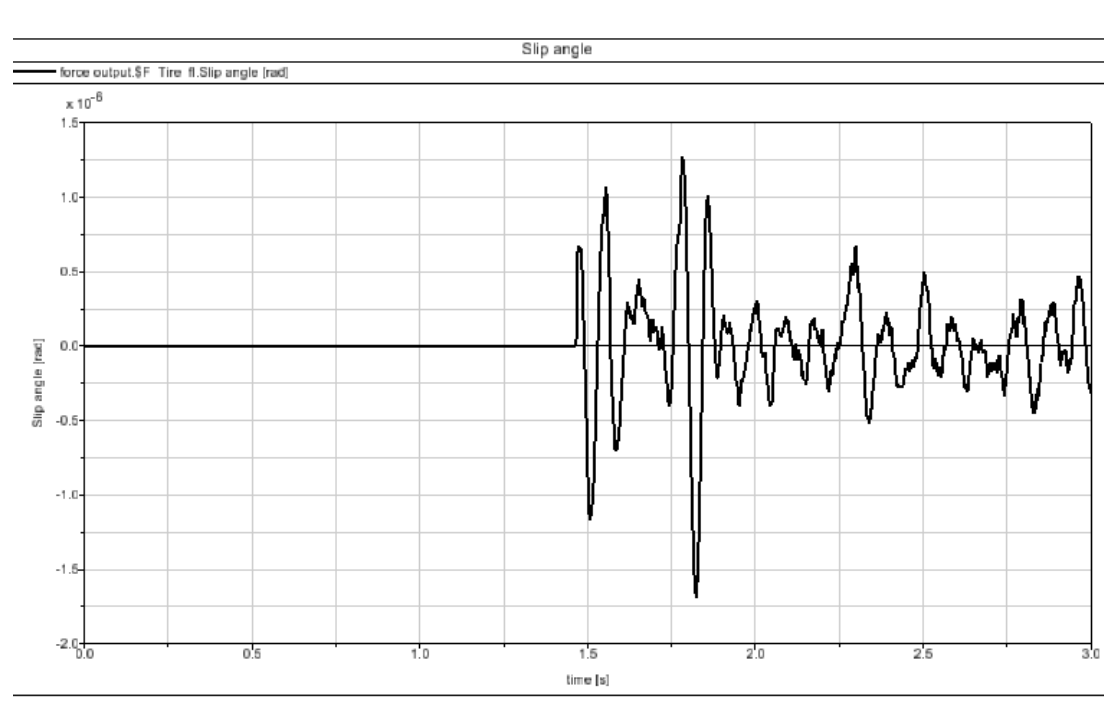


Figure 10 Slip angle

Here comes the slip angle.

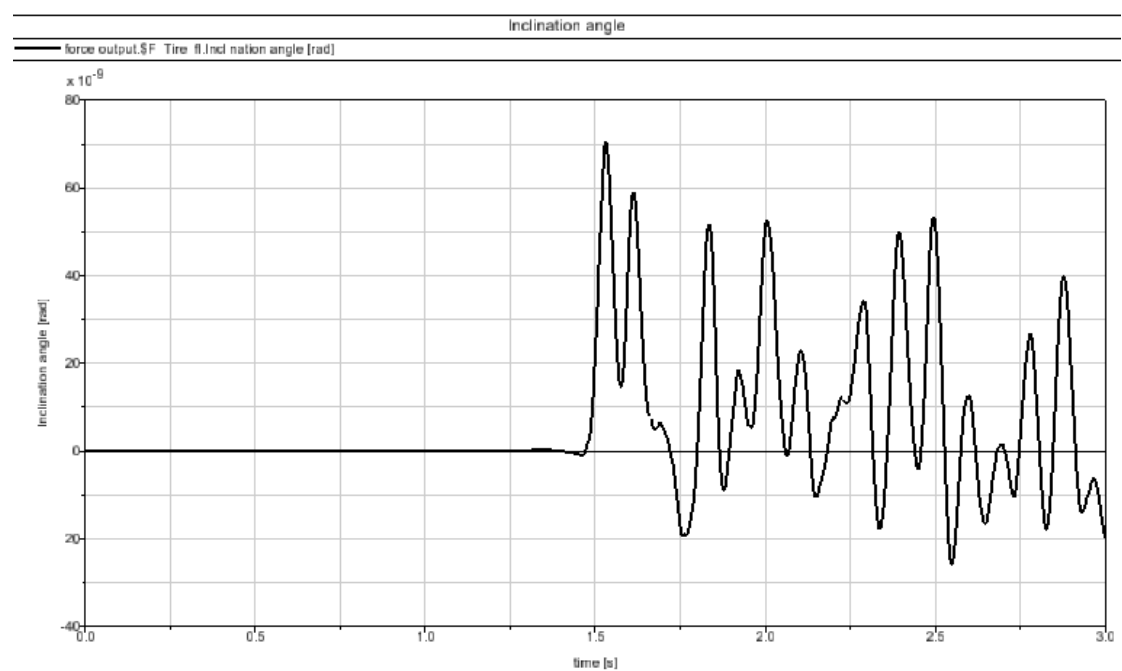


Figure 11 Inclination angle

Figure 11 shows us the inclination angle, most of which is positive.

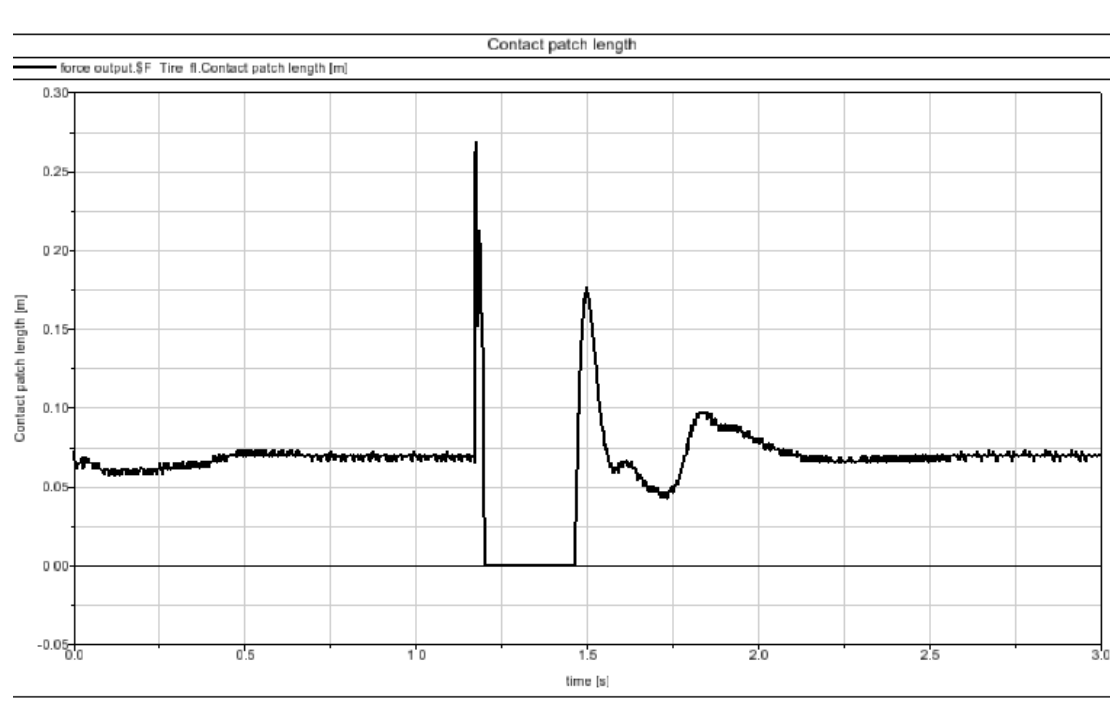


Figure 12 Contact patch length

The contact patch length also changes a lot during the process.

1.4 Suggestions for improvements

Larger tires have higher cornering stiffness. The radial tire has a wide grounding surface and generally has a high lateral stiffness. Steel radial tires are more rigid and have higher stiffness than nylon radial tires. The main measure to increase the cornering stiffness is using wide tires with a small aspect ratio. The cornering stiffness increases with the vertical load, but when the vertical load is too large, the pressure between the tire and the ground area becomes extremely uneven, which reduces the tire cornering stiffness. With the air pressure increasing, the cornering stiffness increases. The stiffness does not change if the air pressure is extremely high.

2 Evaluation

2.1 Total dynamic performance

According to the vehicle dynamics, the vehicle dynamic equation consists of rolling resistance, aerodynamic resistance, engine-braking force, and inertial resistance.

$$G(f_0 + f_1 u_a) \cos \alpha + \frac{C_D A u_a^2}{21.15} + F_z(u_a) + \delta m \frac{du_a}{dt} = 0 \quad (1)$$

Where f_0, f_1 means rolling resistance coefficient; u_a means vehicle speed, km/h; C_D is air resistance coefficient; δ is the rotating mass conversion coefficient, and $\delta=1.03$ to 1.04 when sliding with gear; m is the total mass; α is the slope angle. The engine braking force $F_z(u_a)$ could be described with the expression $F_z(u_a) = k_1 u_a + k_2$, where k_1, k_2 are constants.

Integrate formula (1) and get:

$$\int_0^{u_a} \frac{du_a}{G(f_0 + f_1 u_a) \cos \alpha + \frac{C_D A u_a^2}{21.15} + k_1 u_a + k_2} = \int_0^t -\frac{dt}{\delta m} \quad (2)$$

According to the formula (2), the form of the function of speed-time can be described as:

$$u_a = a_1 \tan(a_2 t + a_3) + a_4 \quad (3)$$

Where a_1, a_2, a_3, a_4 are constants. The least square method is used to fit the curve of function (3). The coefficients in the formula (1) before u_a^2 , u_a , and constant are replaced with j_1, j_2 and j_3 respectively.

$$j_1 u_a^2 + j_2 u_a + j_3 = -\delta m \frac{du_a}{dt} \quad (4)$$

Where j_1 denotes the coefficient of the quadratic item before the air resistance, j_2 denotes the sum of the scale factor of the engine braking force and the scale factor of the rolling resistance, j_3 denotes the constant of the rolling resistance and the constant of the engine braking force.

Put function (3) into formula (1), the result of which is shown as follows:

$$\begin{cases} j_1 a_1^2 + \delta m a_1 a_2 = 0 \\ 2j_1 a_1 a_4 + j_2 a_1 = 0 \\ j_1 a_4^2 + j_2 a_4 + j_3 + \delta m a_1 a_2 = 0 \end{cases} \quad (5)$$

In this formula, solve the ternary system of equations and get the result in formula (6)

$$\begin{cases} j_1 = -\frac{\delta m a_2}{a_1} \\ j_2 = \frac{2\delta m a_2 a_4}{a_1} \\ j_3 = -\frac{\delta m a_2 a_1^2 + \delta m a_2 a_4^2}{a_1} \end{cases} \quad (6)$$

When the driver decelerates, there must be a process of releasing the throttle and taxiing. So the data could be extracted out to calculate the resistance.

2.2 Simulation A and results

The parameters of the trunk could be found in appendix. To carry out Vehicle Resistance Calculation Simulation, we have installed some sensors.

The Hall wheel speed sensor is installed to obtain the velocity by counting the pulse signals from the wheel. Meanwhile, the pulse signals are converted to a voltage signal. When the wheel rotates, the signal shaft rotates synchronously. The magnetic density of the Hall element changes, and generated a Hall voltage. The distance between the Hall sensor and the signal wheel is ensured to be between 0.5mm and 1.5mm when it is installed. Thus, the simulation proves that the error of wheel speed sensor is below 3.4%. Differential Hall Measurement Principle is used to calculate the velocity.

Choose the simulation road as straight and flat as possible, and collect no less than 3 sets of data for each simulation. The rolling resistance coefficient and engine braking force could be calculated when the pickup slides in slow velocity. The aerodynamic resistance coefficient could be calculated when the pickup slides in high velocity. The two experiments correct each other.

Table3 Vehicle resistance parameters of sliding in high speed without gear

Number	Sim1	Sim2	Sim3	Reference
Rolling resistance coefficient	0.014827	0.011996	0.012037	0.0105
Aerodynamic resistance coefficient	0.710548	1.740165	1.47515	0.954

According to the calculation, the average rolling resistance coefficient is 0.012953, and the rolling resistance coefficient of asphalt pavement is 0.0105. The aerodynamic resistance coefficient is 1.2175.

Calculated rolling resistance and aerodynamic resistance have slight fluctuations. Therefore, during simulations, we collected as much data as possible and removed the apparently unreasonable value of the fluctuation. Then the data is averaged to get the final vehicle resistance parameters.

Table4 Vehicle resistance parameters of sliding in high speed

Number	Sim1	Sim 2	Sim 3	Sim 4	Sim5	Reference
Rolling resistance coefficient	0.018	0.009	0.019	0.018	0.018	0.010
Aerodynamic resistance coefficient	1.517	0.339	1.334	1.439	1.490	0.954
Engine braking force factor	18.18	17.44	22.84	18.70	16.38	8.332

The constant in the dynamic equation reflected on the rolling resistance coefficient is 0.018, and the roll resistance coefficient in Table1 is 0.013. Thus the constant of engine braking force that contributes to the constant in the dynamic equation is 0.005. The engine braking force constant is calculated as 125.195N. Previous researchers usually took three points from the velocity-time data into the vehicle dynamics equation to calculate resistance parameters. The simulation shows that error of this method is below 7.9%[11].

In summary, the rolling resistance coefficient of the vehicle is 0.013, the aerodynamic resistance coefficient is 1.44, and the engine braking force is $F=19.9u+125.195$. The results are in good agreement with the values in the automotive theory.

This paper proposed a method for vehicle resistance identification based on vehicle dynamics. The simulation precision of this method is very accurate, and the error is below 1%. For the vehicle resistance recognition experiment, the collected data shows that the rolling resistance coefficient is relatively stable, but the air resistance coefficient and the engine braking force coefficient fluctuate greatly. The future research will focus on improving the precision of the algorithm.

2.3 Simulation B and results

In order to have a research on the overall driving performance, the simulation road is designed and the SIMULINK model has been built.

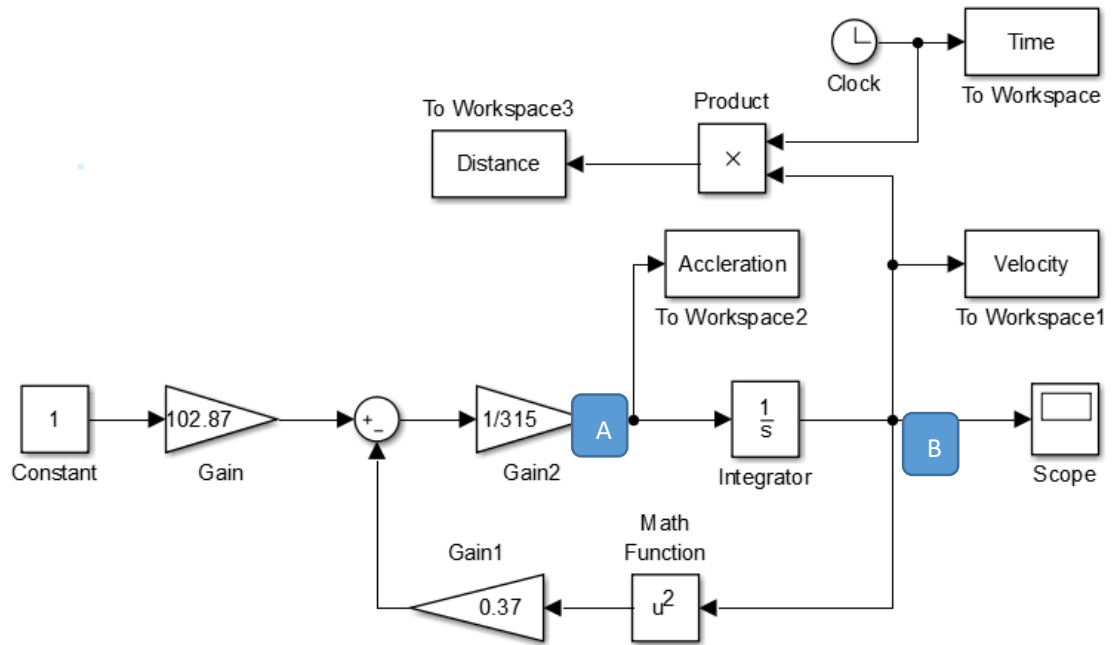


Figure 13 The original SIMULINK model

Build SIMULINK model for this differential equation

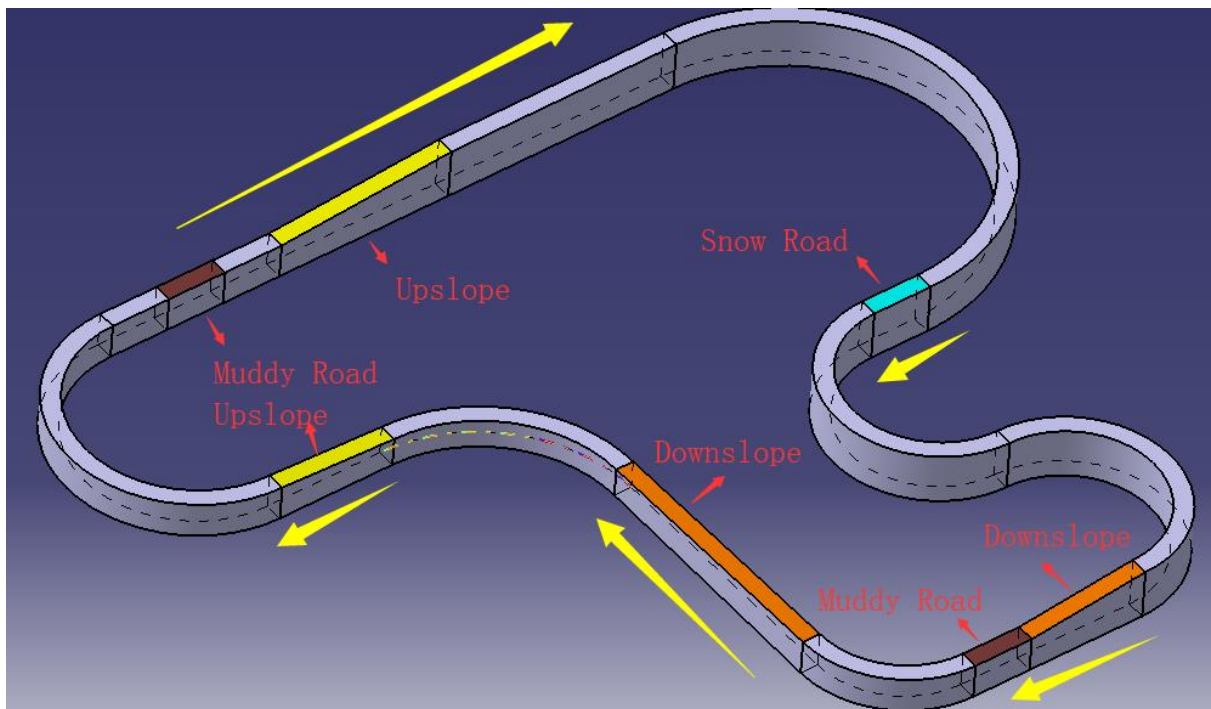


Figure 14 The self-made circuit

For better effect of simulating, the small off road circuit is built as figure2. The direction of travel of the vehicle has been marked with yellow arrow. The muddy road has been marked with brown, whose f is 0.15. Upslope has been marked with yellow and downslope has been marked with orange. The compacted snow road has been marked with blue, whose f

is 0.05. The flat good asphalt or concrete road is the original colour, whose f is 0.015[5]. To simulate the driving progress of the trunk, we need to know the tractive force. The website says that the maximum speed of the buggy is 35-40MPH, which means 56.33-64.37 km/h.

Before building the MATLAB/SIMULINK model, the author brings known parameters into formula (6), and get:

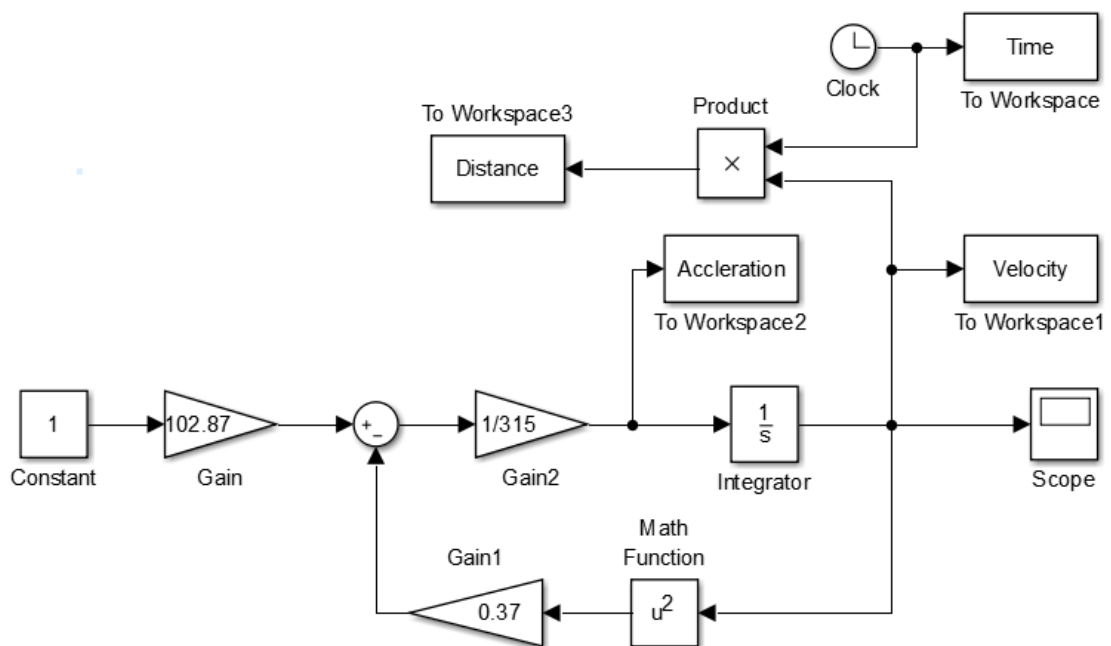


Figure 15 The original SIMULINK model

The author set the time from 0-100s, and the time-velocity graph could be plotted as below

[6].

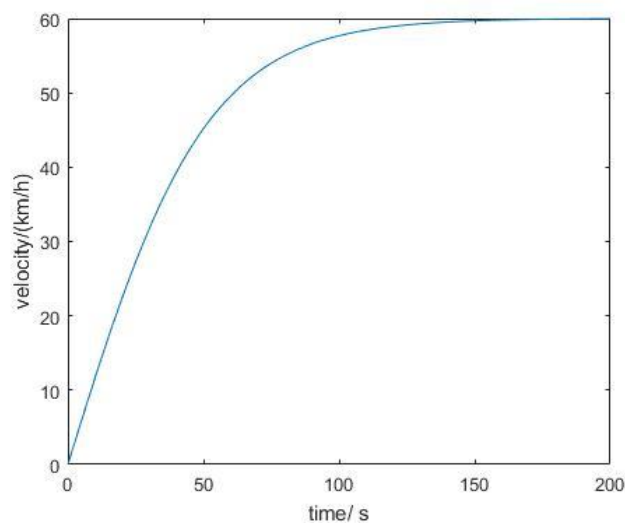


Figure 16 Time-Velocity Graph

In this part, the rolling resistance coefficient is changing with the increasing distance. The

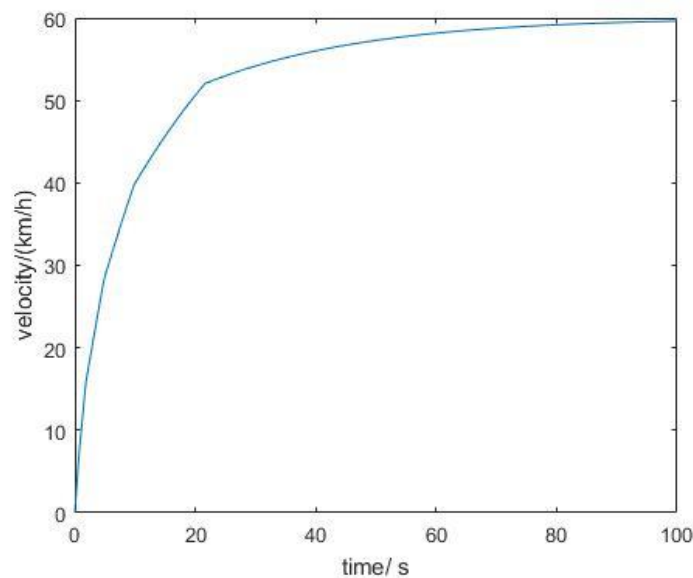


Figure 17 The accelerating progress with CVT's tractive force

principle is the same as before. The author will not go into details

3 Conclusion

As we know, the budget for this truck is limited. Therefore it is important for us to add something like Hydraulic retarder, Electric turbine brake devices, ABS etc.

What we could do in this group project is to measure some parts of the vehicle and do some simulations and calculations. Then we could propose some suggestions based on these results. In my work, I have built 2 models for wheels and overall driving performance.

4 Future work

I have several suggestions and idea for brakes because GUO and I did some measurements in the beginning. I think we should do more tests about how to distribute the front and rear brake braking force. We could make it clear that what is the ideal front and rear brake braking force distribution curve for our vehicle and analyze the braking process of vehicles with fixed ratio of brake force distribution on racing roads. Some improvements about the distribution considering adhesion coefficient and braking efficiency could be made. If necessary, design and add an Auxiliary brake to our vehicle.

Review

The project has allowed us to propose designs for the Zentrack car, that will improve its performance on the road and on the track. The car has become more drivable for the cars target market whilst remaining within their price range.

Given more time, there are other significant changes that would make a more significant impact on the car's performance, and sadly there was not enough time to see any real-world prototypes of any suggested improvements with which to validate the suggested performance gains.

The limitation of budget has meant that some of the most effective improvements were not available to us. As well as this, as a result of Johns determination to keep some components the way they were at the beginning of the project, the overall performance gains we could achieve were restricted from the outset.

Weight improvements, and a stiffer chassis, as well as new springs, dampers and a new braking system will significantly improve performance, as the simulation has shown. Overall, we have been able to make some significant improvements to the Zentrack, that we hope John will implement into his vehicle before it goes on sale.

From the outset the team decided that the dynamic within the group would work best if we did not have one single group leader. Instead, we agreed to each take charge of individual sections and to come together regularly to check up on progress. In hindsight, having a group leader might have made the group dynamic work differently, however this is not necessarily an improvement. Having a group leader would have meant one person fewer working on the technical aspects of the project, however he might have been able to better organise the work done by the rest of the group. We felt that the build-up of the tasks in this project did not require one person to be in charge. Furthermore, this person would have had all the responsibility for the team's progress and might have been blamed when issues occurred. The team build up that we went with has meant that when issues occurred, we were forced to solve them together, rather than blame one person.

In future, we could have seen a significant increase in efficiency of our time, if we had been able to communicate with the customer from an earlier stage in the project. We had already lost a month of our time, before we had signed NDAs and even once this had happened, the sharing of information by the customer was extremely slow, often resulting in required information not being shared at all. As well as information, the required parts for evaluation were also not shared in a timely manner. If we had had a full running vehicle from the outset, along with a complete CAD model, we would have been able to evaluate the existing car far better and could measure our

improvements more effectively. The target of our project was to improve upon the performance of a car, that we never saw in its entirety.

The project's success has been supported significantly by the availability of workshops and equipment. Having full access to tools such as the portable weighing scales has meant that we can make the most of our time. The use of the simulator has allowed us to measure the performance gains of the car, as changes are made by the team. Access to software packages such as Matlab and Solidworks, has allowed us to make better progress, than if we had had to make do without them.

In conclusion, we believe that despite the issues that the team has seen, we have successfully completed the tasks set out by Zentrack. We have improved the performance of the car, whilst keeping it low cost and drivable by amateur drivers. The team worked together very well to reach the goals set out.

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Appendix

Samuel's Appendices

Samuel Appendix A – Original Personal Gantt Chart Task List

#	Task Name		Description	Days	Start	End	Hrs
1	Project			130	01/10/18	01/04/19	160
2	First VIVA		Presentation of work done at half way point	30	03/11/18	03/12/18	20
3	Final VIVA		Presentation of work done at end of project	31	01/03/19	01/04/19	20
4	Report		Final report of whole project	33	24/02/19	29/03/19	20
5	Weekly Meeting		Weekly report of work done and set tasks to be done for following week	130	01/10/18	29/03/19	100
6	Vehicle Analysis		Measuring and weighing current components and calculating rotational inertia	18	15/10/18	02/11/18	44
7	Remove	Suspension Components	Ready for analysis	4	15/10/18	19/10/18	8
8	Weigh		Ready to be inputted into the CAD model	2	19/10/18	21/10/18	4
9	Measure		To verify the CAD model	5	22/10/18	26/10/18	8
10	Refit		Ready to weigh the vehicle	4	26/10/18	30/10/18	8
11	Weigh Vehicle		To check overall mass and for weight distribution calculations	5	29/10/18	02/11/18	4
12	Calculate spring rates		To see current difference between front and rear and decide if it needs changing	5	19/10/18	24/10/18	6
13	Calculate damper rates			5	19/10/18	24/10/18	6
14	CAD		Checking measurements and rebuilding of the CAD model	25	01/11/18	22/11/18	54
15	Rename parts		For ease of building and importing mass	5	01/11/18	06/11/18	6
16	Make parts		Remake assemblies as parts	8	10/11/18	18/11/18	14
17	Make new assemblies		Rebuild CAD model with minimal assemblies	14	06/11/18	22/11/18	30
18	Add weights		To allow the CAD model to be correctly analysed and stress tested	2	06/11/18	08/11/18	4
19	Wishbone		Evaluation and suggestion of improvements of the wishbone design	106	05/11/18	25/03/19	117
20	Evaluate Current Design		Evaluate top and bottom wishbone for front and rear of the car in CAD software through stress testing	5	05/11/18	10/11/18	6
21	Design Improvements		Such as new materials, shapes and location of pick up points (e.g. ARB Droplink)	8	12/11/18	20/11/18	10
22	Evaluate Design Improvements		Carry out stress test in CAD software	9	21/11/18	30/11/18	8

23	Build Prototype of Improvements	Creation of the prototypes	15	03/12/18	18/12/18	15
24	Evaluate Physical Improvements	Analysis and tests of the prototypes	15	19/12/18	03/01/19	10
25	Investigate Manufacturing Limitations	Research manufacturing processes	10	07/01/19	17/01/19	12
26	Design improvements considering manufacturing limitations	Change design if necessary, for larger production, based on previous research	15	21/01/19	05/02/19	18
27	Evaluate Improvements	Carry out stress test in CAD software	10	08/02/19	18/02/19	8
28	Manufacture Component	Build wishbones	25	20/02/19	17/03/19	20
29	Evaluate Component	Analysis and tests of the final product	6	20/03/19	26/03/19	10
Total Hours						345

Samuel Appendix B – Extra Results

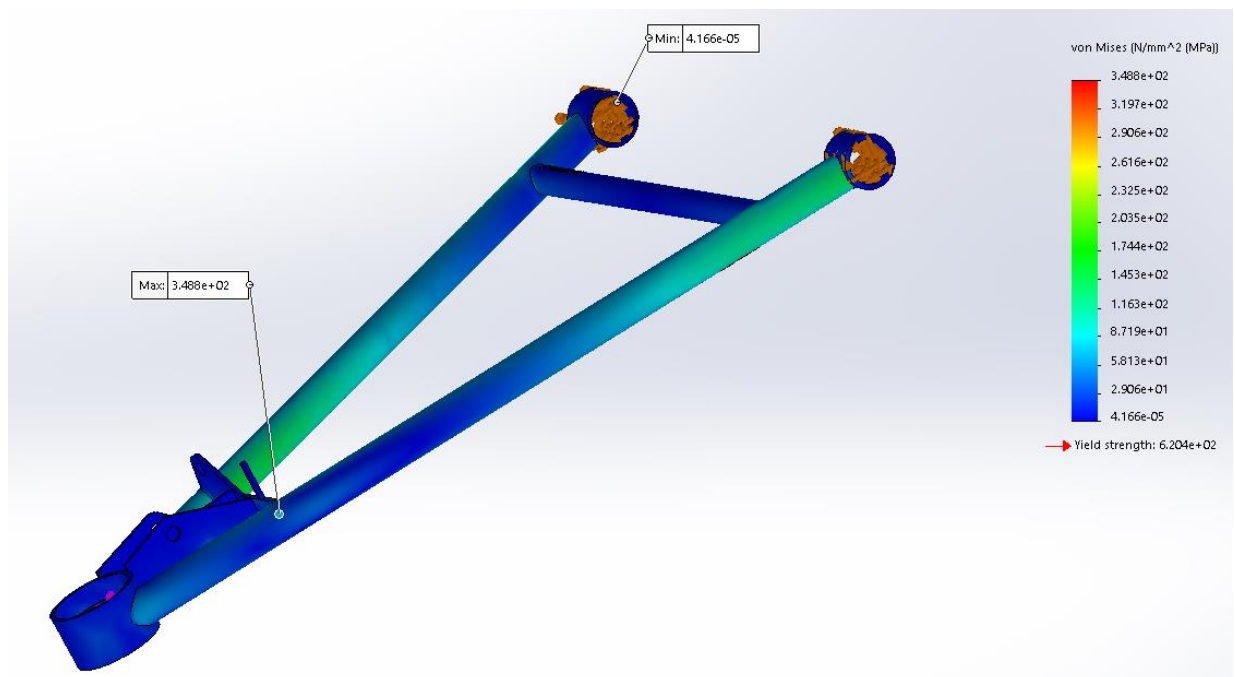


Figure 22 - Break Stress Results 2

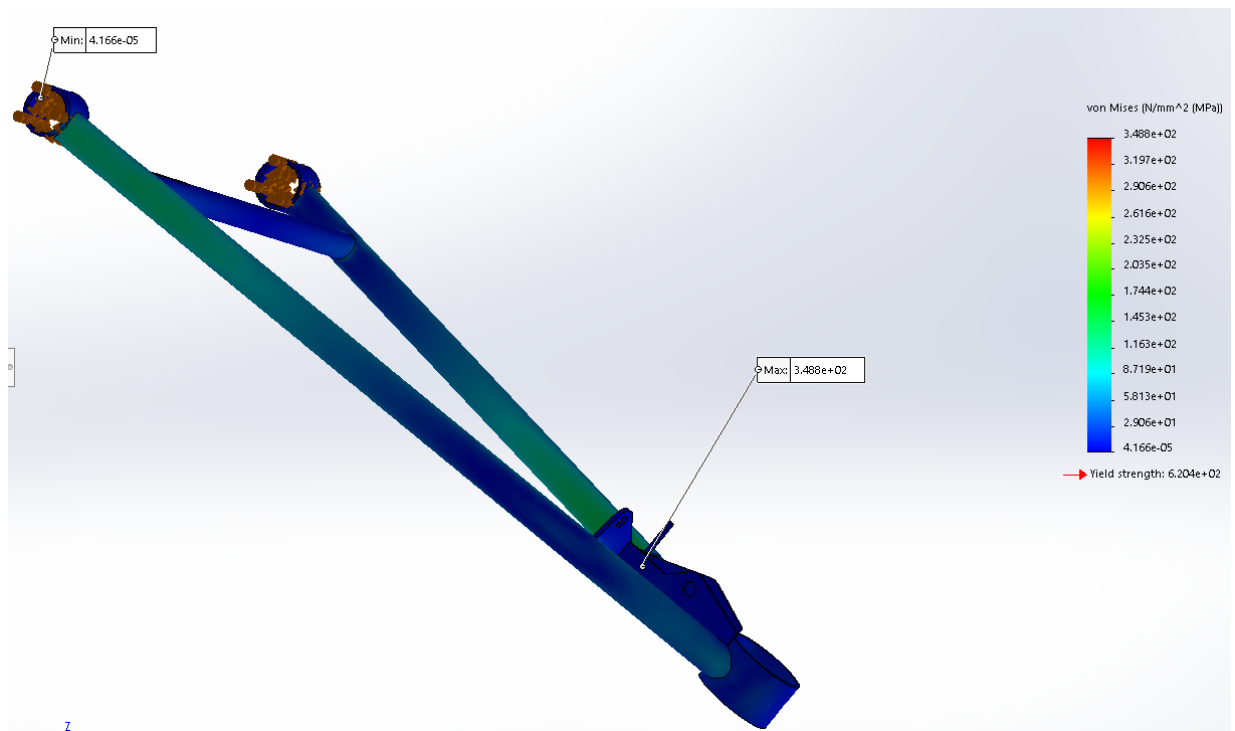


Figure 23 - Break Stress Results 3

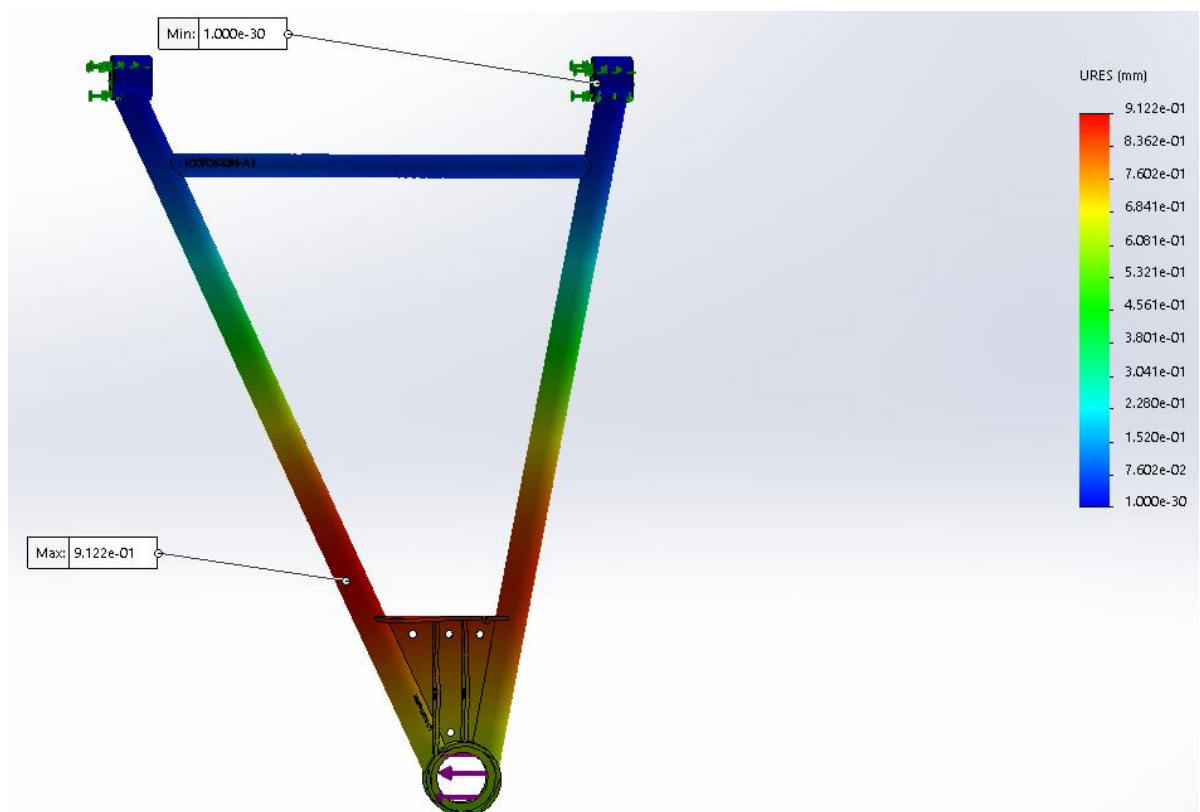


Figure 24 - Break Displacement Results 2

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Property	Value	Units
Mass density	7700	Kg/m ³
Tensile strength	723825600	N/m ²
Yield strength	620422000	N/m ²
Thermal expansion coefficient	0.000013	/k
Thermal conductivity	50	W/(m-k)
Specific heat	460	J/(kg-K)
Elastic modulus	0.00000000021	N/m ²
Poisson's ratio	0.28	N/A
Shear modulus	79000000000	N/m ²