POLITECNICO DI TORINO

DIMEAS - Department of Mechanical and Aerospace Engineering

MASTER OF SCIENCE

IN

AUTOMOTIVE ENGINEERING

Master’s Thesis

Design of Emergency Brake System for Formula Student Driverless car

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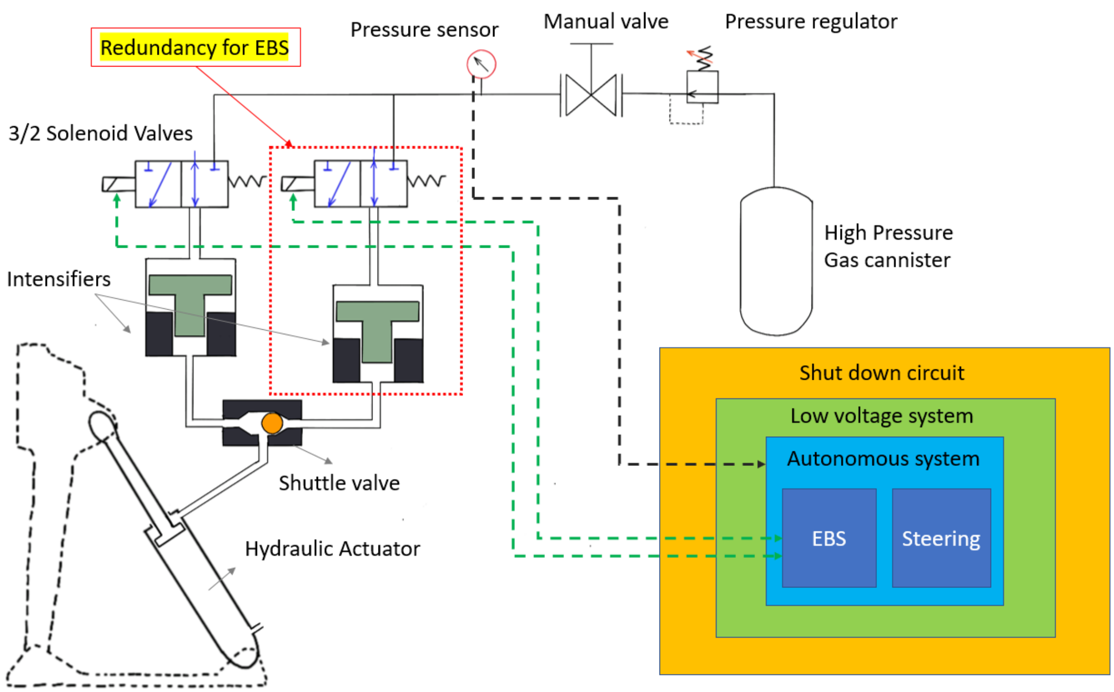
Abstract

The Formula Student competition for Driverless Vehicle (DV) class requires the students to develop a car that can autonomously make its way around a cone track. To ensure safety of such a vehicle without a driver, an Emergency Brake System (EBS) is required. The Emergency Brake System (EBS) shall ensure safe stoppage of the vehicle when any predefined failure modes get triggered.



This thesis focuses on the design and development of the Emergency brake system (EBS) for the Formula Student car (SC19) which is predominantly developed as a car with driver. The primary braking of the Driverless vehicle (DV) will be performed by "Brake-by-wire" while the Emergency brake system (EBS) is designed exploiting the already existing hydraulic brakes (service brake) of the vehicle. Behind the brake pedal is an arrangement of a hydraulic actuator driven by a Hydro-pneumatic intensifier, solenoid actuated valves and a pressure-regulated high-pressure gas cannister. When engaged, the hydraulic actuator connected to the brake pedal pulls it, thus imitating a driver applying brakes manually. Owing to the minimal space available inside the cockpit and the Formula Student regulations, the EBS is designed as a hydro-pneumatic system whose combination provides the required pressure output as well as a compact assembly to be mounted inside the cockpit.

The design specifications of the EBS comes from several design parameters in various vehicle systems such as Brake pedal gain and travel, Brake overtravel switch actuation, Brake master cylinders, balance bar, brake calliper and brake pad, brake discs and tires in addition to vehicle weight, load transfer while braking and the test surface conditions. The brake pedal along with pedal box base plate is modified to accommodate the actuator and FEA is performed to ensure the limit of safe stresses. The overall system has been developed using CATIA™, Stress analysis using Altair Hyperworks and Microsoft Excel calculator for the brake force calculation. All calculations, design and analyses are performed complying to Formula Student regulations 2020.

Acknowledgement

I am grateful for the opportunity to work on Formula student Driverless project and I feel obliged to sincerely thank Prof. Nicola Amati and Prof. Andrea Tonoli for their continuous support and learnings.

I would like to thank Edoardo Corse for his insights regarding the work, a special thanks to Sarath for his accompany and support in the work.

Grateful for a loving family and friends.

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Abbreviations

ABS - Antilock braking system, 9

ADAS - Advanced driver assistance systems, 9

AIR - Accumulator Isolation Relay, 33

AS - Autonomous System, 35

ASMS - Autonomous System Master Switch, 33

CV - Combustion Vehicle, 32

DV - Driverless Vehicle, 31

EBS - Emergency Brake System, 1

ESP - Electronic stability programme, 9

EV - Electric vehicle, 32

FS - Formula Student, 31

HPC - High Pressure Cannister, 42

LVMS - Low voltage Master Switch, 33

LVS - Low Voltage System, 36

RES - Remote Emergency System, 33

SDC - Shut Down Circuit, 33

Chapter 1

Introduction

Automobiles plays a very important role in our lives making us capable of moving from one place to another in order to fulfil the necessities. Over the years, necessities increased and so did the inventions. It was during the 1800s when Germany and France invented and perfected the first working models of our today's automobile. Over the span of the last 100 years, automobiles saw one of the most rapid advancements. Number of cars on the road increased and so did the number of accidents. Safety thus became a matter of highest priority for every car manufacturers and new inventions in active safety and passive safety systems minimise road accidents and it's after effects as they are one of the top causes of death worldwide. Autonomous cars can help to reduce these problems, as they can achieve better traffic flow and to be more efficient. This gives a high potential to reduce fuel consumption and emissions caused by traffic and reduce the risk of accidents among other benefits. Features like ABS, ESP, automated braking and lane keep assistance are examples of such systems that aim to assist the driver. These are referred to as Advanced Driver Assistance Systems (ADAS) and can be seen as stepping-stones in the transition towards Autonomous Driving (AD). By replacing the driver, a large amount of accidents could be avoided due to the nature of human behaviour. Formula student has introduced a new class called Formula student driverless in which students has to build a car which can transition through a coned track autonomously. Competitions like these help students to have a better understanding of Autonomous vehicles.

The aim of this thesis is to design the EBS (Emergency brake system) for the Squadra Corse (SC19) car which was primarily build as a manually driven car as a part of converting it to a driverless vehicle

Figure 1 Squadra Corse car SC19

1.1 Thesis Outline

The thesis is organized as follows.

- Chapter 2: It explains about braking in ideal conditions and how the braking in real conditions is accomplished, followed by the brake architecture of SC19 which a combination of Hydraulic braking and Regenerative braking and explanation about the brake components in SC19.
- Chapter 3: It explains about the regulations regarding Emergency Brake system, followed by design concepts to put such a system in the vehicle, and finally the logic for EBS design achieved using a Hydro-pneumatic arrangement.
- Chapter 4: It deals with the estimation of the pedal force which must be applied by the actuator of the EBS, and different cases have been considered for the force selection and correspondingly the deceleration for all the possible cases.
- Chapter 5: This chapter deals with the calculations regarding the placement of actuator behind the brake pedal and calculations pertaining to different EBS solutions.
- Chapter 6: CAD drawings and FEM analysis of Brake pedal and support

Chapter 2

This chapter explains about braking in ideal conditions and how the braking in real conditions is accomplished, followed by the brake architecture of SC19, which is a combination of Hydraulic braking and Regenerative braking and explanation about the brake components in SC19.

Braking

2.1 Braking in ideal conditions

As defined in [1], Ideal braking is the condition in which all wheels will brake with the same longitudinal force coefficient μ_x, the study of the braking forces which the vehicle can exert follows closely that, with the only obvious difference that braking force, as well as the corresponding force coefficient and the longitudinal slip, are negative. The vertical forces between the vehicle and the ground can be computed using the equations remembering that also the acceleration is now negative.

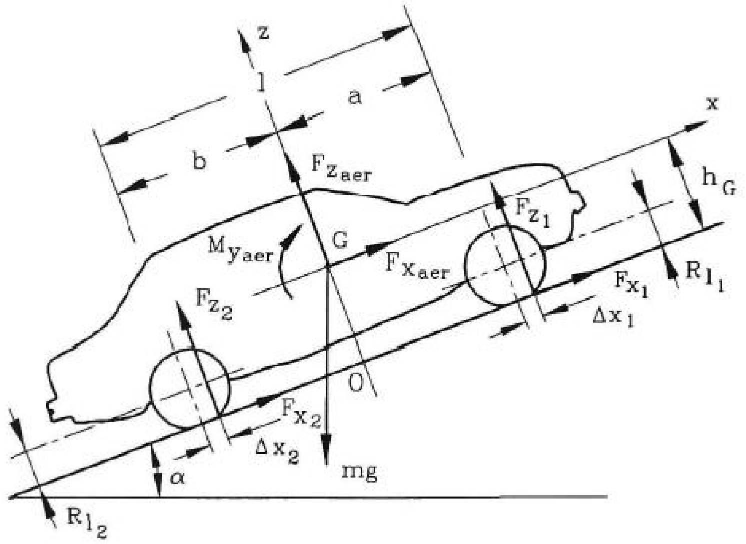


Figure 2 Forces acting on a vehicle moving on an inclined road.

The total braking force F_x is thus

$$F_x = \sum_{\forall i} \mu_{x_i} F_{z_i} \ ,$$

where the sum is extended to all the wheels. The longitudinal equation of motion of the vehicle is then

$$\frac{dV}{dt} = \frac{\sum_{\forall i} \mu_{x_i} F_{z_i} - \frac{1}{2}\rho V^2 S C_X - f \sum_{\forall i} F_{z_i} - mg\sin(\alpha)}{m} \ ,$$

where m is the actual mass of the vehicle and not the equivalent mass, and a is positive for uphill grades. The rotating parts of the vehicle are directly slowed down by the brakes and hence do not enter the evaluation of the forces exchanged between vehicle and road. They must be accounted for when assessing the required braking power of the brakes and the energy which must be dissipated.

In a simplified study of braking aerodynamic drag and rolling resistance can be neglected, since they are usually far smaller than braking forces. Also, rolling resistance can be considered as causing a braking moment on the wheel more than a braking force directly on the ground. As in ideal braking all force coefficients fix is assumed to be equal, the acceleration is,

$$\frac{dV}{dt} = \mu_x \left[g\cos(\alpha) - \frac{1}{2m}\rho V^2 S C_Z \right] - g\sin(\alpha) \ .$$

In case of level road, for a vehicle with no aerodynamic lift, Equation reduces to

$$\frac{dV}{dt} = \mu_x g$$

The maximum deceleration in ideal conditions can be obtained by introducing the maximum negative value of μ_x,

The assumption of ideal braking implies that the braking torques applied on the various wheels are proportional to the forces F_z, if the radii of the wheels are all equal. As will be seen later, this can occur in only one condition, unless some sophisticated control device is implemented to allow braking in ideal conditions.

If μ_x can be assumed to remain constant during braking, the motion of the vehicle occurs with constant acceleration, and the usual formulae hold

$$t_{V_1 \to V_2} = \frac{V_2 - V_1}{|\mu_x| g} , \qquad s_{V_1 \to V_2} = \frac{V_2^2 - V_1^2}{2|\mu_x| g}$$

The time and the space to stop the vehicle from speed V are then

$$t_{stop} = \frac{V}{|\mu_x| g} , \qquad s_{stop} = \frac{V^2}{2|\mu_x| g} .$$

The time needed to stop the vehicle increases linearly with the speed while the space increases quadratically.

To compute the forces Fx the wheels must exert to perform an ideal braking manoeuvre, forces Fz on the wheels must be computed first. This can be done using the formulae seen in above section. However, for vehicles with low aerodynamic vertical loading, as all commercial and passenger vehicles, except for racers and some sports cars, aerodynamic loads can be neglected. Also drag forces can be neglected and, in case of a two-axle vehicle, the equations reduce to

$$F_{z_1} = \frac{m}{l} \left[gb\cos(\alpha) - gh_G \sin(\alpha) - h_G \frac{dV}{dt} \right]$$

$$F_{z_2} = \frac{m}{l} \left[ga\cos(\alpha) + gh_G \sin(\alpha) + h_G \frac{dV}{dt} \right]$$

Then,

$$\frac{dV}{dt} = \frac{\mu_{x_1} F_{z_1} + \mu_{x_2} F_{z_2}}{m} - g\sin(\alpha)$$

since the values of μ_x are all equal in ideal braking, the values of longitudinal forces Fx are

$$F_{x_1} = \mu_x F_{z_1} = \mu_x \frac{mg}{l}\left[b\cos(\alpha) - h_G\mu_x\right]$$

$$F_{x_2} = \mu_x F_{z_2} = \mu_x \frac{mg}{l}\left[a\cos(\alpha) + h_G\mu_x\right]$$

By eliminating μ_x using above equations, the following relationship between F_{X1} and F_{X2} is readily obtained

$$(F_{x_1} + F_{x_2})^2 + mg\cos^2(\alpha)\left(F_{x_1}\frac{a}{h_G} - F_{x_2}\frac{b}{h_G}\right) = 0$$

The plot of equation in F_{X1}, F_{X2} plane is a parabola whose axis is parallel to the bisector of the second and fourth quadrants if a = b. The parabola is thus the locus of all pairs of values of F_{X1} and F_{X2} leading to ideal braking.

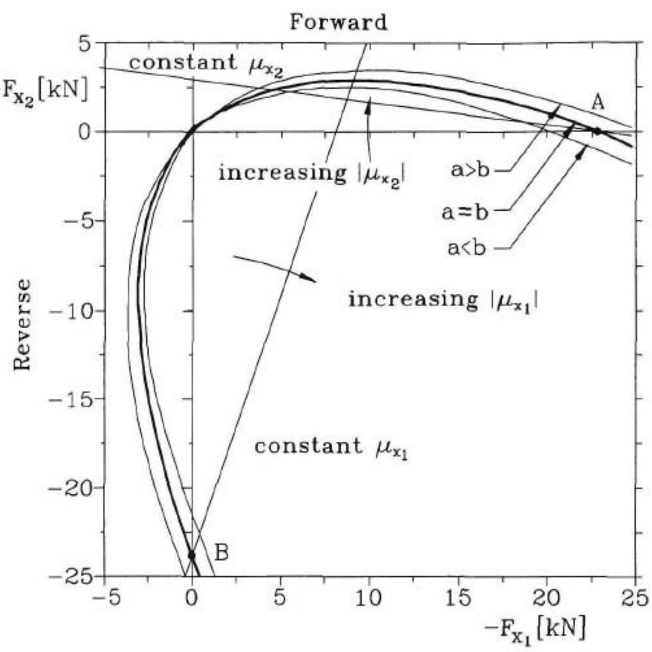


Figure 3 Braking in ideal conditions.

Relationship between F_{X1} and F_{X2} for vehicles with the centre of mass at mid-wheelbase, forward and backward of that point. Plots obtained with m = 1000 kg; I = 2.4 m, ha = 0.5 m, level road.

On the same plot it is possible to draw the lines with constant μ_{X1}, μ_{X2} and acceleration. On level road, the first two are straight line passing respectively through points B and A, while the lines with constant acceleration are straight lines parallel to the bisector of the second quadrant.

Note that the forces are related to each axle and not to each wheel: In the case of axles with two wheels their values are then twice the values referred to the wheel.

The moment to be applied to each wheel is approximately equal to the braking force multiplied by the loaded radius of the wheel: If the wheels have equal radii the same plot holds also for the braking torques. If this condition does not apply the scales are just multiplied by two different factors and the plot is distorted but remains essentially unchanged.

To perform a more precise computation, the rolling resistance should be accounted for, which is a small correction, and the torque needed for decelerating the rotating inertias should be added. This correction is important only in the case of driving wheels and braking in low gear, but in this case the braking effect of the engine, which is even more important and has opposite sign, should be considered.

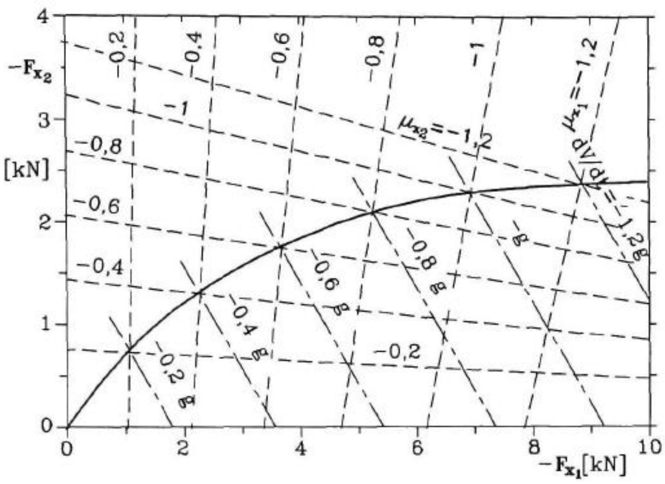


Figure 4 Ideal Braking - Enlargement of the useful zone with constant μ_{X1}, μ_{X2} and acceleration.

As already stated, the law linking F_{XI} to F_{X2}, i.e. M_{bl} to M_{b2} to allow braking in ideal conditions depends on the mass and the position of the centre of mass. For passenger vehicles it is possible to plot the lines for the minimum and maximum load and to assume that all conditions are included between them; for industrial vehicles the position of the centre of mass can vary to a larger extent, and a larger set of load conditions should be considered.

2.2 Braking in actual conditions

The relationship between the braking moments at the rear and front wheels is in practice different from that stated in order to comply the conditions to obtain ideal braking and is imposed by the parameters of the actual braking system of the vehicle.

A ratio,

$$K_B = \frac{M_{b_1}}{M_{b_2}}$$

between the braking moments at the front and rear wheels can be defined. If all wheels have the same radius, its value coincides with the ratio between the braking forces3. For each value of the deceleration a value of KB which allows the braking to take place in ideal conditions can be easily found from the plot of Fig. 4.28.

KB depends on the actual layout of the braking system, and in some simple cases is almost constant. In case of hydraulic braking systems, the braking torque is linked to the pressure in the hydraulic system by a relationship of the type

$$M_b = \epsilon_b(Ap - Q_m)$$

where tb, sometimes referred to as efficiency of the brake, is the ratio between the braking torque and the force exerted on the braking elements and hence has the dimensions of a length, A is the area of the pistons, p is the pressure and Qm is the restoring force due to the springs, when they are present.

The value of KB is thus

$$K_B = \frac{\epsilon_{b_1}(A_1 p_1 - Q_{m_1})}{\epsilon_{b_2}(A_2 p_2 - Q_{m_2})}$$

or, if no spring is present as in the case of disc brakes,

$$K_B = \frac{\epsilon_{b_1} A_1 p_1}{\epsilon_{b_2} A_2 p_2}$$

In the case of disc brakes eb is almost constant and is, as a first approximation, the product of the average radius of the brake, the friction coefficient and the number of braking elements acting on the axle, as braking torques are again referred to the whole axle. In this case, if the pressure acting on the front and rear wheels is the same, the value of K_B is constant and depends only on geometrical parameters.

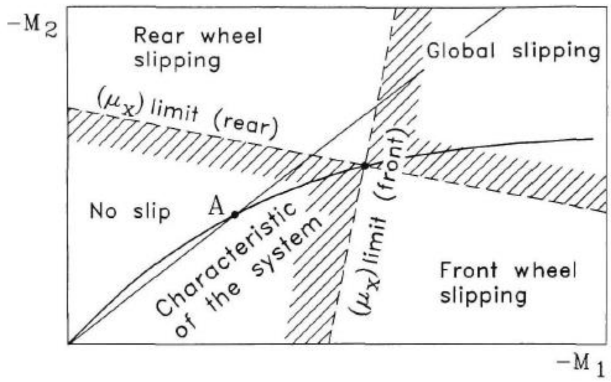


Figure 5 Conditions for ideal braking

Conditions for ideal braking, characteristic line for a system with constant KB and zones in which the front or the rear wheels lock. In the case shown the value of (µX1) limit is high enough to cause sliding beyond point A.

2.3 Brake architecture of SC19

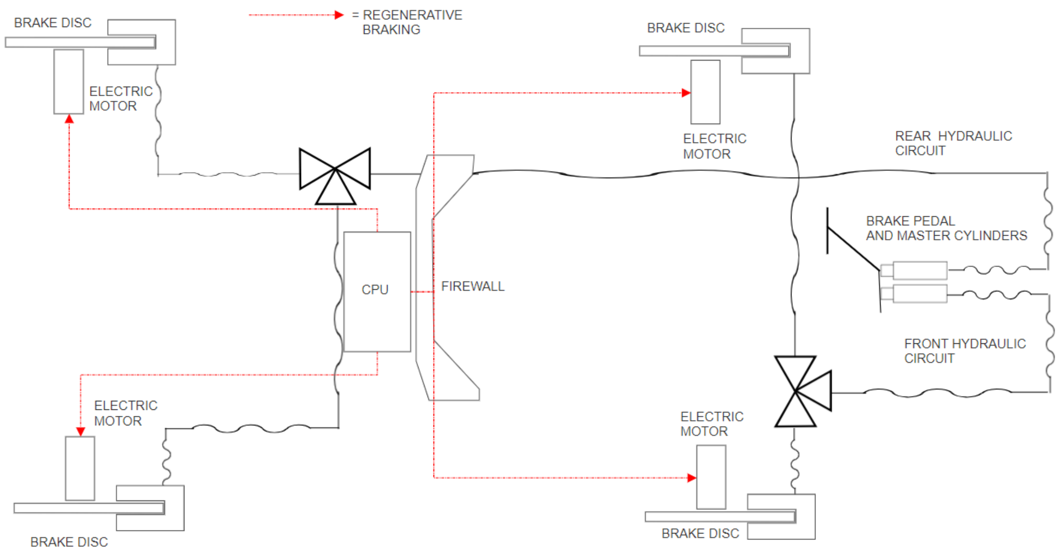


Figure 6 Braking scheme of SC19

The braking system of SC19 consist of combined a Hydraulic braking as well as with regenerative braking using electric motors in the wheel hubs. There is a strain-gauge present in the pedal box assembly so that 90% of pedal travel is used to have braking with regenerative braking phenomenon and the remaining pedal travel is exploited to have braking with hydraulic circuit. 200Nm of braking torque is applied for the front tires and 70 Nm of braking torque is applied for the rear tires, Regenerative braking completely exploited before hydraulics. Maximum energy recovery.

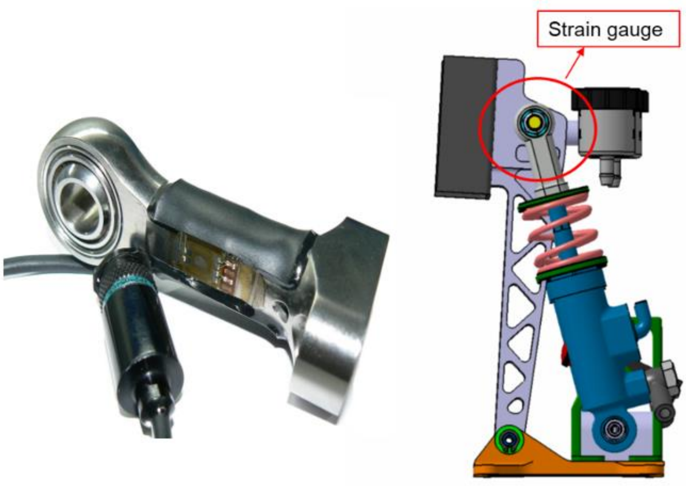


Figure 7 Strain gauge and its placement

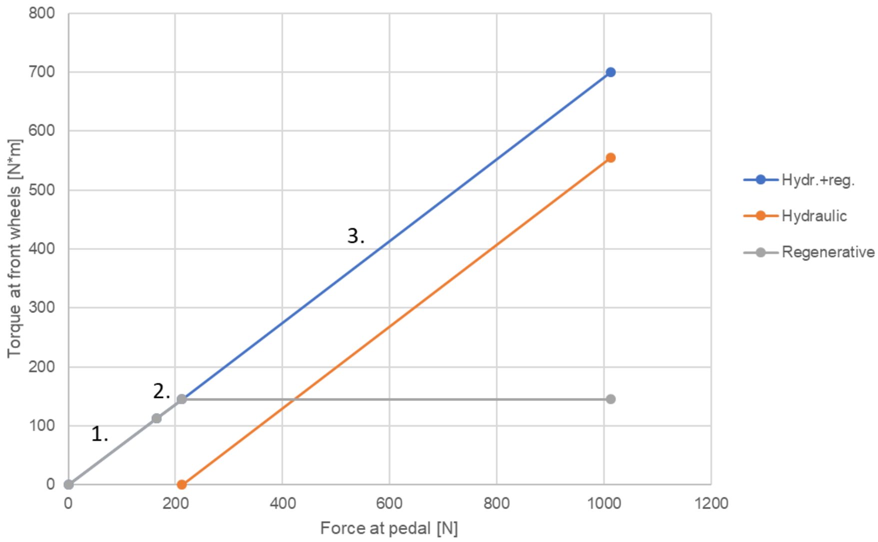


Figure 8 Regenerative braking and Hydraulic braking with pedal force and travel

In the figure shown above

Region 1 - Only regenerative braking, pedal doesn't move due to preload.

Region 2 - Pedal start moving but no hydraulic contribution. Still only regenerative.

Region 3 - After regenerative 100%, Hydraulic braking

2.4 Hydraulic Brake circuit of SC19

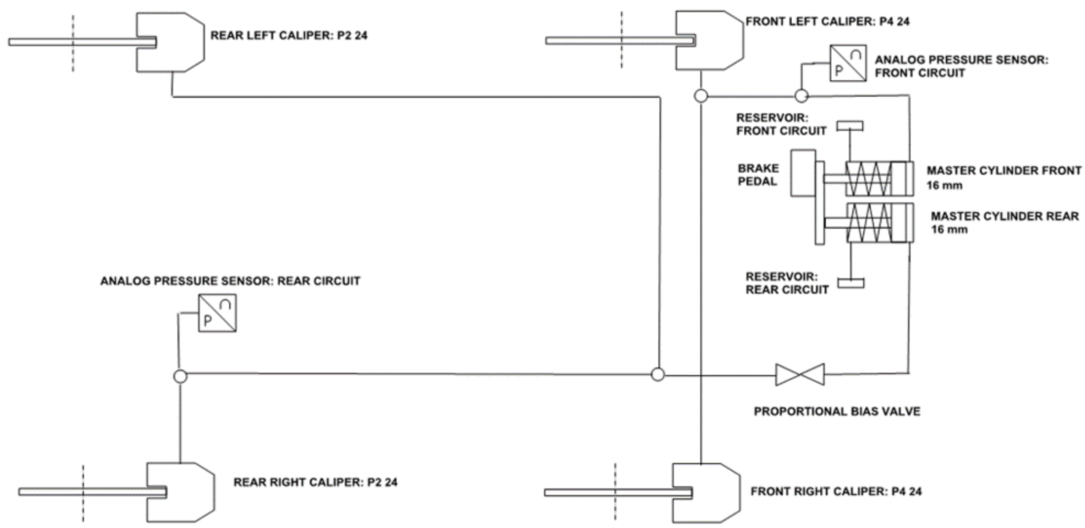


Figure 9 hydraulic brake circuit arrangement used for SC19

Consists of brake pedal box containing brake pedal, master cylinders, fluid reservoirs, a balance bar, thereby connecting to the hydraulic lines to the brake callipers and thereby to the brake rotors (brake disks). Both the master cylinders for front and rear are same so that to have to balance bar at 50:50 repartition, and torque repartition is obtained using Different brake rotors and callipers.

Torque repartition

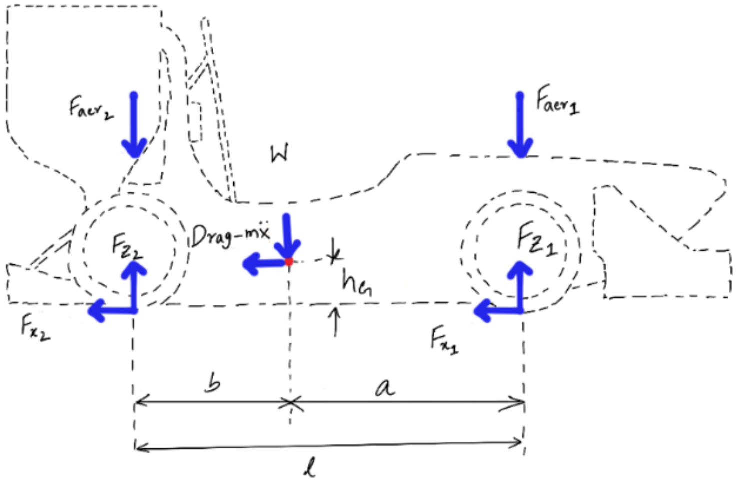


Figure 10 Forces and moment equilibrium

[1]Resolving the forces and moments we get the following equations,

$$\begin{cases} F_{x_1} + F_{x_2} + F_{x_{aer}} - mg\sin(\alpha) = m\dot{V} \\ F_{z_1} + F_{z_2} + F_{z_{aer}} - mg\cos(\alpha) = 0 \\ F_{z_1}(a + \Delta x_1) - F_{z_2}(b - \Delta x_2) + mgh_G\sin(\alpha) - M_{aer} + |F_{x_{aer}}|h_G = -mh_G\dot{V} \end{cases}$$

The second and third equations can be solved in the normal force acting on the axes, yielding

$$\begin{cases} F_{z_1} = mg\dfrac{(b - \Delta x_2)\cos(\alpha) - h_G\sin(\alpha) - K_1V^2 - \dfrac{h_G}{g}\dot{V}}{l + \Delta x_1 - \Delta x_2} \\ F_{z_2} = mg\dfrac{(a + \Delta x_1)\cos(\alpha) + h_G\sin(\alpha) - K_2V^2 + \dfrac{h_G}{g}\dot{V}}{l + \Delta x_1 - \Delta x_2} \end{cases}$$

Where,

$$\begin{cases} K_1 = \dfrac{\rho S}{2mg}\left[C_x h_G - lC_{M_y} + (b - \Delta x_2)C_z\right] \\ K_2 = \dfrac{\rho S}{2mg}\left[-C_x h_G + lC_{M_y} + (a + \Delta x_1)C_z\right] \end{cases}$$

As in F1 race cars Δ_{x1} and Δ_{x2} are same, so neglecting Δ, in the calculation for simplification and

$$F_{x1} = \mu_{x1} * F_{z1} \quad \text{and} \quad F_{x2} = \mu_{x2} * F_{z2}$$

Torque for the front wheels,

$$C_{FA}[Nm] = (F_{x1\,perwheel} * loaded\ radius) + (moment\ of\ inertia_{per\ wheel} * \alpha_A)$$

Where,

$$\alpha_A\left[\frac{rad}{s^2}\right] = \frac{maximum\ deceleration}{loaded\ radius\ of\ wheel}$$

$$maximum\ decelration = \frac{F_{aer\_drag} + 2 * (F_{x1\,perwheel} + F_{x2\,perwheel})}{Total\ mass}$$

And torque for the rear wheels,

$$C_{Fp}[Nm] = (F_{x2\,perwheel} * loaded\ radius) + (moment\ of\ inertia_{per\ wheel} * \alpha_P)$$

$$Torque\ repartition\ front = \frac{(C_{FA})}{(C_{FA}+C_{Fp})}$$

With above calculations for speeds ranging from 10 km/h to 120 km/h, the Brake torque repartition was approximately coming around 68:32,

Braking force on the disks

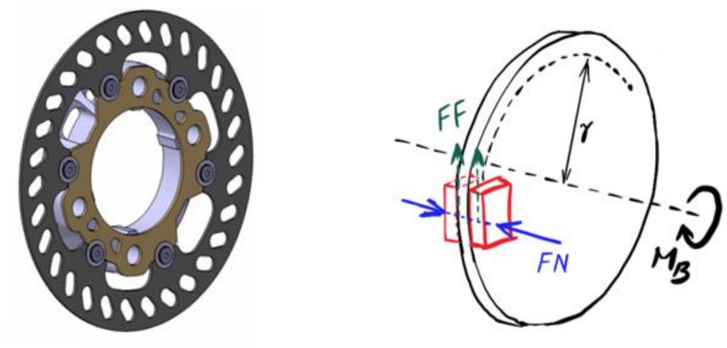


Figure 11 Brake rotor or brake disc

Minimum Braking force which must be applied on the front brake discs

$$FF_A[N] = \frac{C_{FA}}{Front\ brake\ radius}$$

Minimum Braking force must be applied on the rear brake discs

$$FF_P[N] = \frac{C_{FP}}{Rear\ brake\ radius}$$

We decided to go with front and rear brake radius of 94 mm and 83 mm respectively

Normal force on the disk calliper

Normal force on the front calliper

$$FN_A[N] = \frac{FF_A}{\mu_{front\ brake\ pad}}$$

Normal force on the front calliper

$$FN_P[N] = \frac{FF_P}{\mu_{Rear\ brake\ pad}}$$

Where μ is the coefficient of friction on the brake discs which was obtained by telemetry to be 0.4, but under higher temperature conditions the coefficient of friction of the discs reduces to 0.34

Disc callipers and Master cylinders for the front and the rear

master cylinder is a control device that converts force (commonly from a driver's foot) into hydraulic pressure. This device controls slave cylinders located at the other end of the hydraulic system.

Figure 12 Master cylinders of the hydraulic brakes

As piston(s) move along the bore of the master cylinder, this movement is transferred through the hydraulic fluid, to result in a movement of the slave cylinder(s). The hydraulic pressure created by moving a piston (inside the bore of the master cylinder) toward the slave cylinder(s) compresses the fluid evenly, but by varying the comparative surface area of the master cylinder and each slave cylinder, one can vary the amount of force and displacement applied to each slave cylinder, relative to the amount of force and displacement applied to the master cylinder.

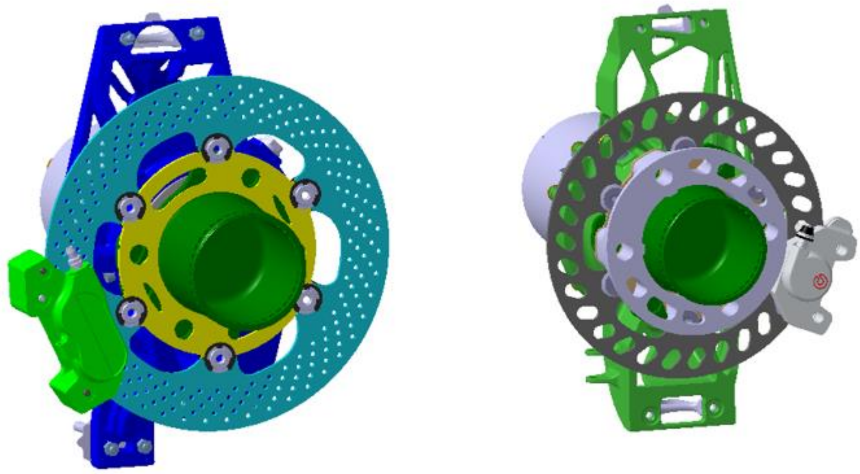


Figure 13 Brake callipers for the front (4 pistons)-left and rear (2 pistons)-right

Number of pistons on the calliper in the front are twice as the number of pistons in the rear. Four pistons in the front calliper and two pistons in the rear calliper, also the brake rotor diameters for the front brakes are bigger than the rear brake rotors.

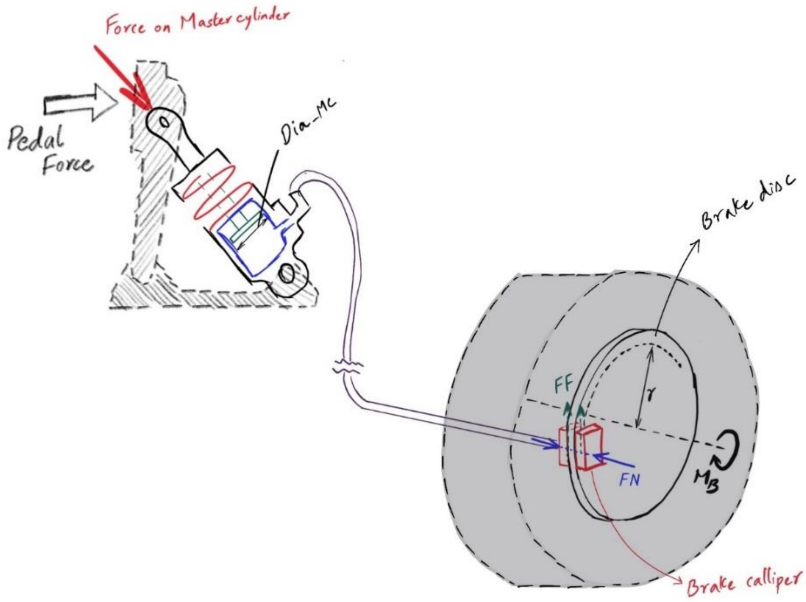


Figure 14 Brake architecture of a Hydraulic brake system

$$calliper\ piston\ area = \frac{Normal\ force\ on\ calliper[FN_A\ or\ FN_P]}{Pressure\ in\ circuit[Front\ or\ rear]}$$

$$Pressure\ in\ the\ circuit[front\ or\ rear] = \frac{Force\ on\ the\ master\ cylinder}{Area\ of\ the\ master\ cylinder}$$

$$Input\ force\ [for\ combined\ on\ the\ master\ cylinders] = Pressure_{Front\_MC} * Area_{Front_{MC}} + Pressure_{Rear\_MC} * Area_{Rear\_MC}$$

$$Input\ force[N] = spring\ preload + Input\ force\ [for\ combined\ on\ the\ master\ cylinders]$$

Disc callipers are selected to have the pressure in the both the Master cylinders of the front and rear as close as possible (to keep the balance bar at 50:50 repartition) under maximum braking conditions (i.e., deceleration of 1.8g)

As regards a study about the torque distribution has been done which is a study about the longitudinal dynamic of the vehicle in braking conditions, we obtained the total torque repartition was about 65:35 front to the rear,

Once the maximum braking torque for each wheel has been established, the hydraulic pressures inside the circuits have been evaluated considering as mentioned above for maximum deceleration of 1.8g, passing from the torque to the tangential force acting on the disc, then to the normal force and finally, dividing for the pistons area of the callipers.

A thorough study has been done to select the brake callipers and the Master cylinders, the brake callipers were found to be P4 24 for the front and P2 24 for the rear to have the required Brake torque repartition

And master cylinders of 16mm diameter for both front and rear brake circuits were selected to have balance bar kept at 50:50

Balance bar

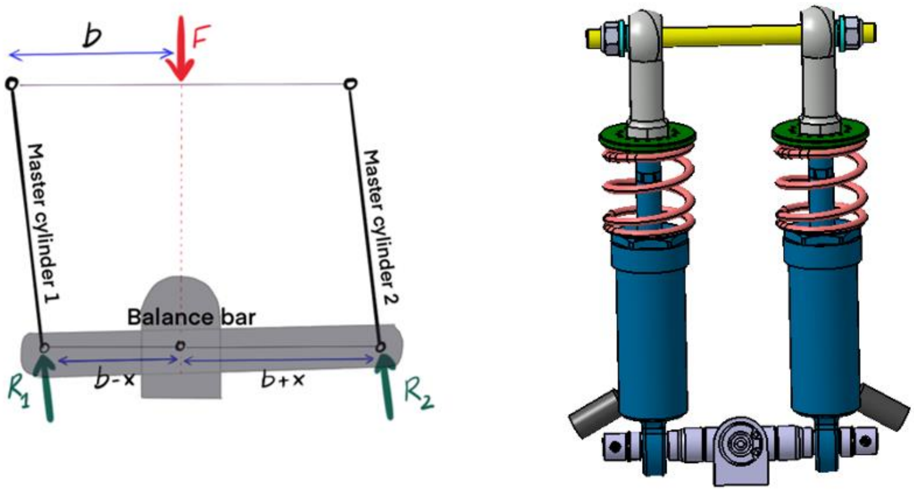


Figure 15 balance bar adjustment

$$Balance\ bar\ repartition_{front\ to\ rear} = \frac{Pressure_{Front\_MC}}{Pressure_{Front\_MC} + Pressure_{Rear\_MC}}$$

The balance bar is an adjustable lever (usually a threaded rod), that pivots on a spherical bearing and uses two separate master cylinders for the front and rear brakes. Most balance bars are part of a pedal assembly that also provides a mounting for the master cylinders. When the balance bar is cantered, it pushes equally on both master cylinders creating equal pressure, given that the master cylinders are of the same size bore. When adjusted as far as possible toward one master cylinder it will push approximately twice as hard on that cylinder as the other.

To set up the balance bar, thread the master cylinder pushrods through their respective clevises to obtain the desired position. Threading one pushrod into its respective clevis means threading the other one out the same amount.

Balance bar was initially kept at 50:50 but it can be further calibrated for our required brake repartition.

Brake pedal gain

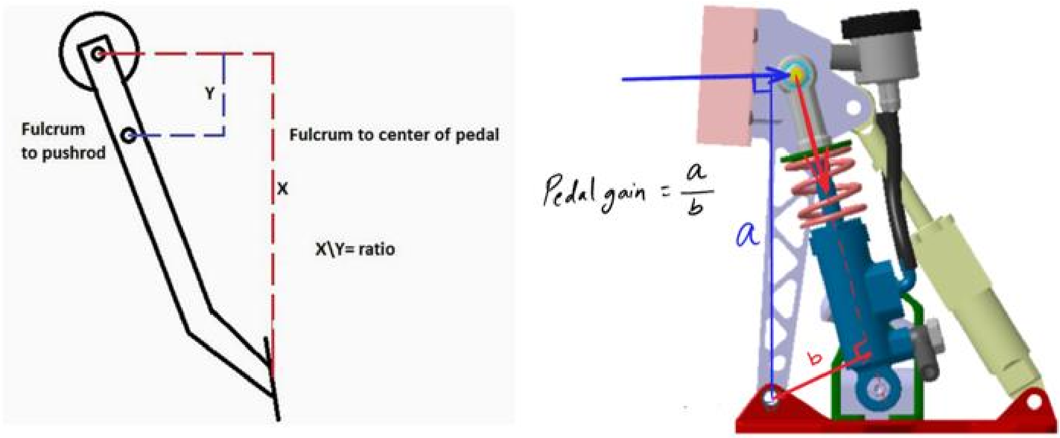


Figure 16 Pedal gain

$$Pedal\ force[N] = \frac{Input\ force}{pedal\ gain}$$

Brake pedal arrangement is a critical issue to be addressed with its geometrical arrangement which will directly affect the drivers foot effort for the pedal travel which is given by the pedal ratio (pedal gain)

Pedal ratio (pedal gain) is the ratio of leverage with the pedal applies to the master cylinder. To determine the pedal ratio we need to measure the height of the pedal to the pivot point then divided the measurement of the pivot point to the lower arm that controls the rod to the master cylinder. For comfortable braking effort the pedal gain should be between 3 to 6, higher the pedal gain lower force will be requested by the driver and a higher pedal displacement is a drawback of it, and lower the pedal gain higher force have to be applied by the driver with lower pedal displacement. The master cylinder is arranger to have a steep slope so that we can add an actuator behind making a pedal gain of 3.

Chapter 3

This chapter explains about the regulations regarding Emergency Brake system, followed by design concepts to put such a system in the vehicle, and finally the logic for EBS design achieved using a Hydro-pneumatic arrangement.

Emergency Brake System (EBS) for SC19

3.1 Regulations regarding the implementation of EBS according to rulebook FS2020

Brake system (T6) [2]

T6.1 Brake System - General

T6.1.1 The vehicle must be equipped with a hydraulic brake system that acts on all four wheels and is operated by a single control.

T6.1.2 The brake system must have two independent hydraulic circuits such that in the case of a leak or failure at any point in the system, effective braking power is maintained on at least two wheels. Each hydraulic circuit must have its own fluid reserve, either by the use of separate reservoirs or by the use of a dammed reservoir.

T6.1.3 A single brake acting on a limited-slip differential is acceptable.

T6.1.4 “Brake-by-wire” systems are prohibited. [DV ONLY] In autonomous mode, it is allowed to use “brake-by-wire”. In manual mode, T6.1.1 applies.

T6.1.5 Unarmoured plastic brake lines are prohibited.

T6.1.6 The brake system must be protected from failure of the drivetrain, see T7.3.2, from touching any movable part and from minor collisions.

T6.1.7 In side view any portion of the brake system that is mounted on the sprung part of the vehicle must not below the lower surface of the chassis.

T6.1.8 The brake pedal and its mounting must be designed to withstand a force of 2 kN without any failure of the brake system or pedal box. This may be tested by pressing the pedal with the maximum force that can be exerted by any official when seated normally.

T6.1.9 The brake pedal must be fabricated from steel or aluminium or machined from steel, aluminium or titanium.

T6.1.10 [EV ONLY] The first 90% of the brake pedal travel may be used to regenerate brake energy without actuating the hydraulic brake system. The remaining brake pedal travel must directly actuate the hydraulic brake system, but brake energy regeneration may remain active.

Brake Over-Travel Switch (BOTS) (T6.2) [2]

T6.2.1 A brake pedal over-travel switch must be installed on the vehicle as part of the shutdown circuit, as in EV6 or CV4.1. This switch must be installed so that in the event of a failure in at least one of the brakes circuits the brake pedal over-travel will result in the shutdown circuit being opened. This must function for all possible brake pedal and brake balance settings without damaging any part of the vehicle.

T6.2.2 Repeated actuation of the switch must not close the shutdown circuit, and it must be designed so that the driver cannot reset it.

T6.2.3 The brake over travel-switch must be a mechanical single pole, single throw switch, commonly known as a two-position switch, push-pull or flip type, it may consist of a series connection of switches.

Emergency Brake System (EBS) (DV3) [2]

DV3.1 Technical Requirements

DV3.1.1 All specifications of the brake system from T6 remain valid.

DV3.1.2 The vehicle must be equipped with an EBS, that must be supplied by LVMS, ASMS, RES and a relay which is supplied by the SDC ([EV ONLY] parallel to the AIR, but must not be delayed/[CV ONLY] parallel to fuel pump relay).

DV3.1.3 The EBS must only use passive systems with mechanical energy storage. Electrical power loss at EBS must lead to a direct emergency brake manoeuvre (keep in mind T11.3.1!).

DV3.1.4 The EBS may be part of the hydraulic brake system. For all components of pneumatic and hydraulic EBS actuation not covered by T6, T9 is applied.

DV3.1.5 When the EBS is part of the hydraulic brake system, the manual brake actuation (by brake pedal) may be deactivated for autonomous driving.

DV3.1.6 The EBS must be designed so that any official can easily deactivate it. All deactivation points must be in proximity to each other, easily accessible without the need for tools/removing any body parts/excessively bending into the cockpit. They must be able to be operated also when wearing gloves.

DV3.1.7 A pictographic description of the location of the EBS release points must be clearly visible in proximity to the ASMS. The necessary steps to release the EBS must be clearly marked (e.g. pictographic or with pull/push/turn arrow) at each release point. This point must be marked by a red arrow of 100mm length (shaft width of 20mm) with “EBS release” in white letters on it.

DV3.1.8 The use of push-in fittings is prohibited in function critical pneumatic circuits of the EBS and any other system which uses the same energy storage without proper decoupling.

Functional Safety (DV3.2) [2]

DV3.2.1 Due to the safety critical character of the EBS, the system must either remain fully functional, or the vehicle must automatically transition to the safe state in case of a single failure mode.

DV3.2.2 The safe state is the vehicle at a standstill, brakes engaged to prevent the vehicle from rolling, and an open SDC.

DV3.2.3 To get to the safe state, the vehicle must perform an autonomous brake manoeuvre described in section DV3.3 and IN6.3.

DV3.2.4 An initial check must be performed to ensure that EBS and its redundancy is able to build up brake pressure as expected, before AS transitions to “AS Ready”.

DV3.2.5 The tractive system is not considered to be a brake system.

DV3.2.6 The service brake system may be used as redundancy if two-way monitoring is ensured.

DV3.2.7 A red indicator light in the cockpit that is easily visible even in bright sunlight and clearly marked with the lettering “EBS” must light up if the EBS detects a failure.

EBS Performance (DV3.3) [2]

DV3.3.1 The system reaction time (the time between entering the triggered state and the start of the deceleration) must not exceed 200 ms.

DV3.3.2 The average deceleration must be greater than 8 m/s2 under dry track conditions.

DV3.3.3 Whilst decelerating, the vehicle must remain in a stable driving condition (i.e. no unintended yaw movement). This can be either a controlled deceleration (steering and braking control is active) or a stable braking in a straight line with all four wheels locked.

DV3.3.4 The performance of the system will be tested at technical inspection, see IN6.3.

Driverless Inspection EBS Test (IN6.3) [2]

IN6.3.1 The EBS performance will be tested dynamically and must demonstrate the performance described in DV3.3.

IN6.3.2 The test will be performed in a straight line marked with cones similar to acceleration.

IN6.3.3 During the brake test, the vehicle must accelerate in autonomous mode up to at least 40 km=h within 20m. From the point where the RES is triggered, the vehicle must come to a safe stop within a maximum distance of 10 m.

IN6.3.4 In case of wet track conditions, the stopping distance will be scaled by the officials dependent on the friction level of the track.

Autonomous System Master Switch (ASMS) (DV2.2) [2]

DV2.2.1 Each DV must be equipped with an ASMS, according to T11.2.

DV2.2.2 The ASMS must be mounted in the middle of a completely blue circular area of 50mm diameter placed on a high contrast background.

DV2.2.3 The ASMS must be marked with “AS”.

DV2.2.4 The power supply of the steering and braking actuators must be switched by LVMS and ASMS

DV2.2.5 When the ASMS is in “Off” position, the following must be fulfilled:

• No steering, braking and propulsion actuation can be performed by request of the autonomous system.

• The sensors and the processing units can stay operational.

• The vehicle must be able to be pushed as specified in A6.7.

• It must be possible to operate the vehicle manually as a normal CV or EV.

DV2.2.6 It is strictly forbidden to switch the ASMS to the “On” position if a person is inside the vehicle.

DV2.2.7 After switching the ASMS to the “On” position, the vehicle may not start moving and the brakes must remain closed (“AS ready” state, Figure 21) until a “Go” signal is sent via the RES (“AS driving” state, Figure 21).

DV2.2.8 The ASMS must be fitted with a “lockout/tagout” capability to prevent accidental activation of the AS. The ASR must ensure that the ASMS is locked in the off position whenever the vehicle is outside the dynamic area or driven in manual mode.

Autonomous State Definitions (DV2.4) [2]

DV2.4.1 The AS must implement the states and state transitions as shown in Figure 21.

DV2.4.2 The AS must not have any other states or transitions.

DV2.4.3 Numbered steps within an AS state machine transition (see Figure 21) must be checked in the given order. The vehicle must only perform a state-transition if all conditions are fulfilled. Until the transition is complete the ASSIs must indicate the initial state.

DV2.4.4 The steering actuator can only have the following states:

• “unavailable”: power supply of the actuator is disconnected, manual steering is possible

• “available”: power supply is connected, and the actuator can respond to commands of the AS according to DV2.3.1.

DV2.4.5 The service brake can only have the following states:

• “unavailable”: power supply of the actuator is disconnected, manual braking is possible

• “engaged”: prevents the vehicle from rolling on a slope up to 15%

• “available”: responds immediately to commands from the AS For the state transition of the service brake actuator no manual steps (e.g. operating manual valves / (dis-)connecting mechanical elements) are allowed.

DV2.4.6 The EBS can only have the following states:

• “unavailable”: the actuator is disconnected from the system/the energy storage is de-energized, emergency brake manoeuvre is not possible.

• “armed”: will initiate an emergency brake manoeuvre immediately if the SDC is opened or the LVS supply is interrupted

• “activated”: brakes are closed and power to EBS is cut. Brakes may only be released after performing manual steps.

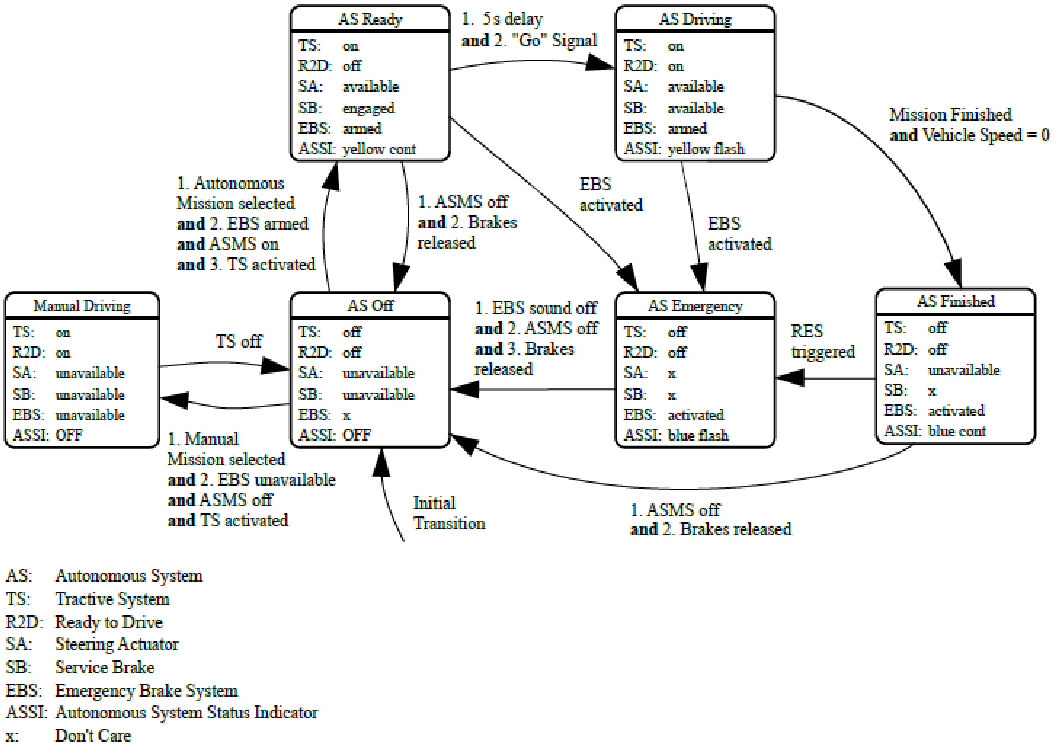


Figure 17 Autonomous system (AS) state machine

Remote Emergency System (RES) (DV1.4) [2]

DV1.4.1 Every vehicle must be equipped with a standard RES specified in the competition handbook.

The system consists of two parts, the remote control and the vehicle module.

DV1.4.2 The RES must be purchased by the team.

DV1.4.3 The RES has two functions:

• When the remote emergency stop button is pressed, it must trigger the DV Shutdown Circuit (SDC) defined in DV1.5.

• Race-control-to-vehicle communication:

– The race control can send a “Go” signal to the vehicle

– The “Go” signal replaces green flags

DV1.4.4 The RES vehicle module must be directly integrated in the vehicle’s SDC with one of its relays hard-wired in series to the shutdown buttons.

DV1.4.5 The antenna of the RES must be mounted unobstructed and without interfering parts in proximity (other antennas, etc.).

Shutdown circuit (DV1.5) [2]

DV1.5.1 The drivetrain-specific requirements for the SDC (see CV4.1 or EV6) remain valid for DV.

DV1.5.2 If the SDC is opened by the Autonomous System (AS) or the RES, it has to be latched open by a non-programmable logic that can only be reset manually (either a button outside of the vehicle, in proximity to the ASMS, or via LVMS power cycle).

DV1.5.3 The SDC may only be closed by the AS, if the following conditions are fulfilled:

• Manual Driving: Manual Mission is selected; the AS has checked that EBS is unavailable (No EBS actuation possible).

• Autonomous Driving: Autonomous Mission is selected, ASMS is switched on and enough brake pressure is built up (brakes are closed).

Compressed gas systems and high-pressure hydraulics (T 9) [2]

T9.1 Compressed Gas Cylinders and Lines

T9.1.1 Any system on the vehicle that uses a compressed gas as an actuating medium must comply with the following requirements:

• The working gas must be non-flammable.

• The gas cylinder/tank must be of proprietary manufacture, designed and built for the pressure being used, certified and labelled or stamped appropriately.

• A pressure regulator must be used and mounted directly onto the gas cylinder/tank.

• The gas cylinder/tank and lines must be protected from rollover, collision from any direction, or damage resulting from the failure of rotating equipment.

• The gas cylinder/tank and the pressure regulator must be located within the rollover protection envelope T1.1.14, but must not be located in the cockpit.

• The gas cylinder/tank must be securely mounted to the chassis, engine or transmission.

• The axis of the gas cylinder/tank must not point at the driver.

• The gas cylinder/tank must be insulated from any heat sources.

• The gas lines and fittings must be appropriate for the maximum possible operating pressure of the system.

T9.2 High Pressure Hydraulic Pumps and Lines

T9.2.1 The driver and anyone standing outside the vehicle must be shielded from any hydraulic pumps and lines with line pressures of 2100 kPa or higher. The shields must be steel or aluminium with a minimum thickness of 1mm. Brake lines are not considered as high-pressure hydraulic lines.

3.2 Design concepts for the development of Emergency Brake System EBS

Since the Emergency brake system must be actuated only once if there is a power loss to the Low voltage system of the vehicle or if the race marshal triggers the stoppage actuation using a remote (RES – Remote Emergency System). Therefore, exploiting the already existing service brake for the emergency brake action is a good idea.

Brake actuation using an electrically driven servo motor with a cable drive

This kind of arrangement was not realised given to the fact that cables have to be pre-tensioned and one big reason for not using any electric actuator is due to the fact that EBS has to automatically engage when there is a loss of power to the Autonomous system or Low voltage system of the vehicle and the motors are electrically driven so decided to have an actuator behind the brake pedal hinged to the base and other end attached to the brake pedal top, so that it can pull and swivel at the same time, which could be either a Hydraulic or pneumatic.

Brake actuation using an Electric linear actuator

Given to the FS rules the EBS must engage when there is electrical power loss, so all electric actuators have been avoided. And, electric actuators are comparatively slow with respect to hydraulic or pneumatic actuators, electric actuators do not fulfil the FS regulations to have the brake actuation within 200 milli seconds.

Brake actuation using a Pneumatic Actuator

A preliminary design was done to have a pneumatic actuator with a cannister and pressure regulator, all commercially available pneumatic actuators works at a maximum input pressure of 10 bar and some special pneumatic actuators works at input pressure of 17 bar, in both the cases the pneumatic actuator bore size was very high for our required power rating, bore sizes of minimum 60 mm was coming out of calculations.

Brake actuation using a hydraulic actuator

Hydraulic actuators can be used under higher operating pressures, hence hydraulic actuators can provide higher forces with bore of smaller size. In our design arrangement we need a hydraulic actuator capable of providing the required force at minimum actuation rate of 200 milli seconds.

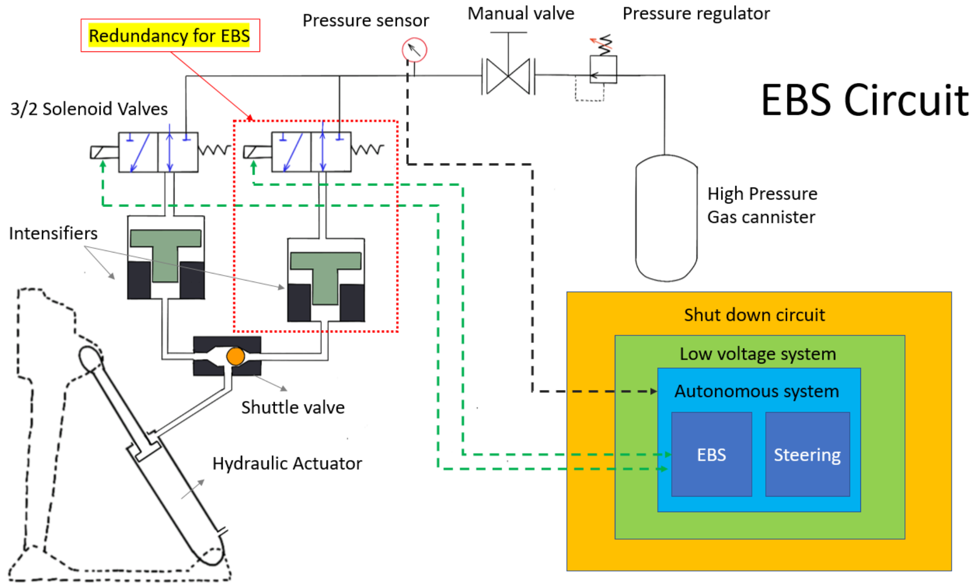
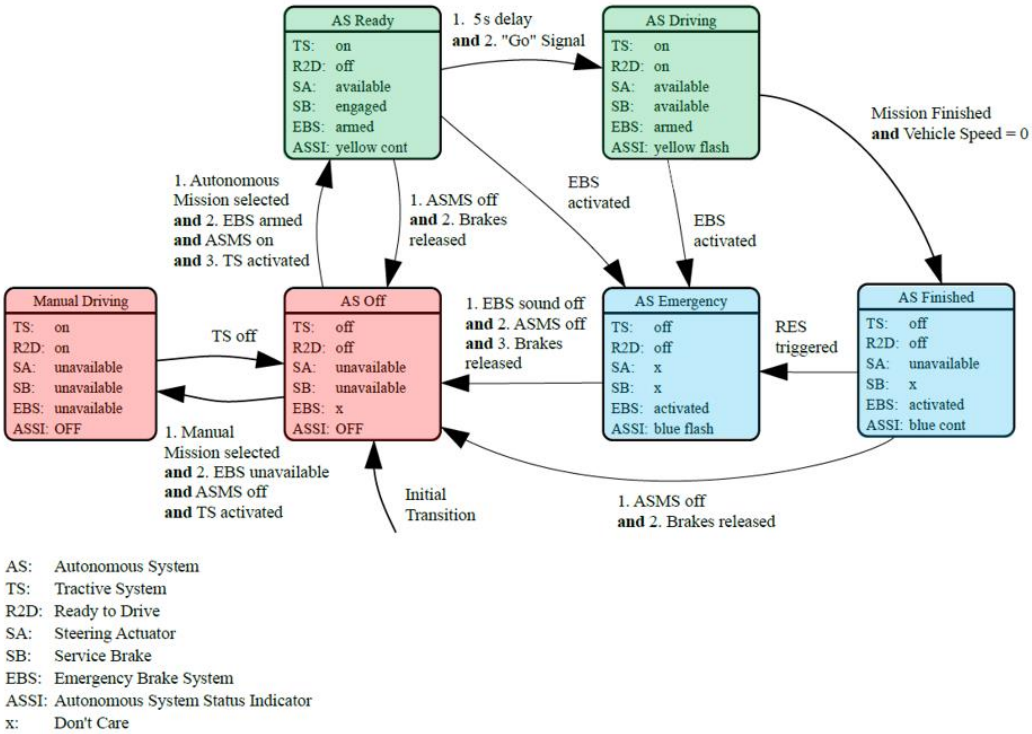


Figure 18 EBS Layout

Since we do not have a pump to provide the hydraulic pressure, it was decided to go with the use of a high-pressure gas cannister with Intensifiers (The Hydro-Pneumatic intensifier consists of a double acting Pneumatic Cylinder and a Hydraulic high-pressure chamber. The Pneumatic Cylinder piston rod is forced into the hydraulic chamber resulting in high-pressure oil displacement). Intensifiers can be actuated using solenoid actuated valves (two position three-way valves). Since the brake pedal has to be actuated only as it is an Emergency brake actuation which will bring the vehicle to a complete stoppage.

3.3 EBS logic

Formula Student Driverless cars are equipped with an Emergency Brake System (EBS). This is actuated via the Remote Emergency System (RES) or during any loss of voltage to the LVS (Low Voltage System) and at the end of completion of Autonomous driving (AS finished).



In order to guarantee the safety of the autonomous vehicles in the operation and handling for all parties concerned, the team must fulfil some special requirements. Each vehicle must be equipped with a so-called RES (Remote Emergency System), which fulfils two functions. By means of this remote control, the required Emergency Brake System (EBS) can be triggered and the vehicle can be stopped in emergency situations. At the same time, the RES control system enables the "Go" signal to be sent to the vehicle at the start of the dynamic disciplines. Furthermore, all FSD vehicles are equipped with different coloured signal lamps, which indicate the respective operating states of the vehicle. In autonomous mode, a yellow signal is illuminated, whilst a blue light indicates the status of the RES. These systems must be tested during the Driverless Inspection.

Due to rule DV3.1.3 of FS2020 a passive system with mechanically stored energy must be used for the EBS. This led to the Hydro-pneumatic system for the brake solution.

The EBS has been designed to maintain the states illustrated in figures below, The HPC (High pressure gas cannister) will be inserted when the vehicle is turned off, and hence the electric normally opened EBS valve still open. Consequently, the manual on/off valve needs to be closed during insertion of the Cannister, as seen in Figure. The yellow line describes the pressure flow in the system, pressure regulator is used to manipulate the high pressure from the cannister to the operating pressure of the EBS valves

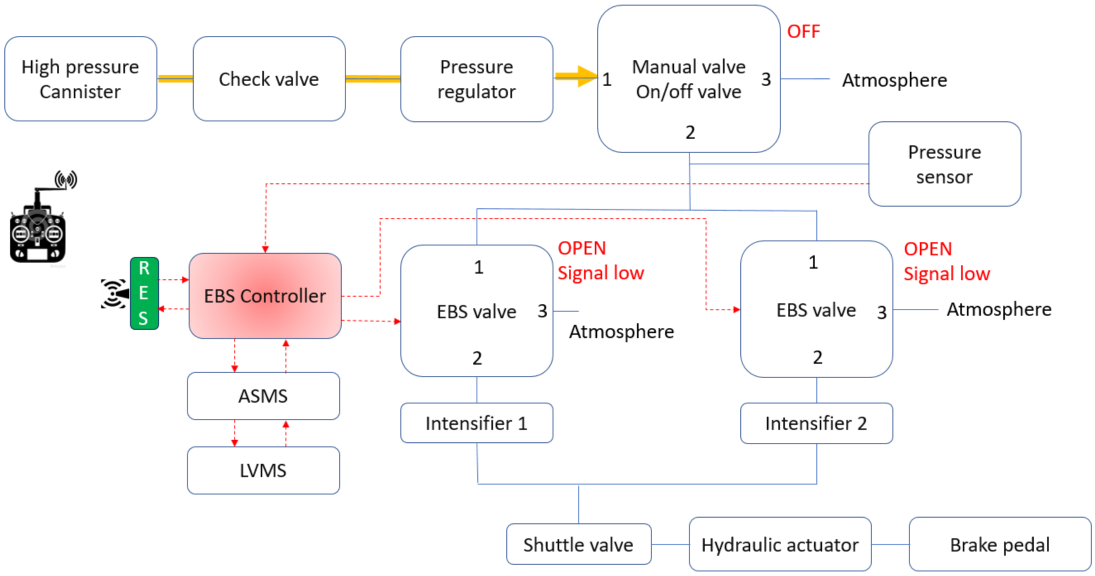


Figure 19 Pressure flow after inserting High pressure gas cannister

When the vehicle has been started and no errors have been detected, the EBS valve is supplied by the low-voltage battery and is closed. At this moment, the manual on/off valve can be opened to put the EBS in "armed" mode, seen in Figure, the manual valve is connected to two EBS valves (2/3 solenoid actuated valves).

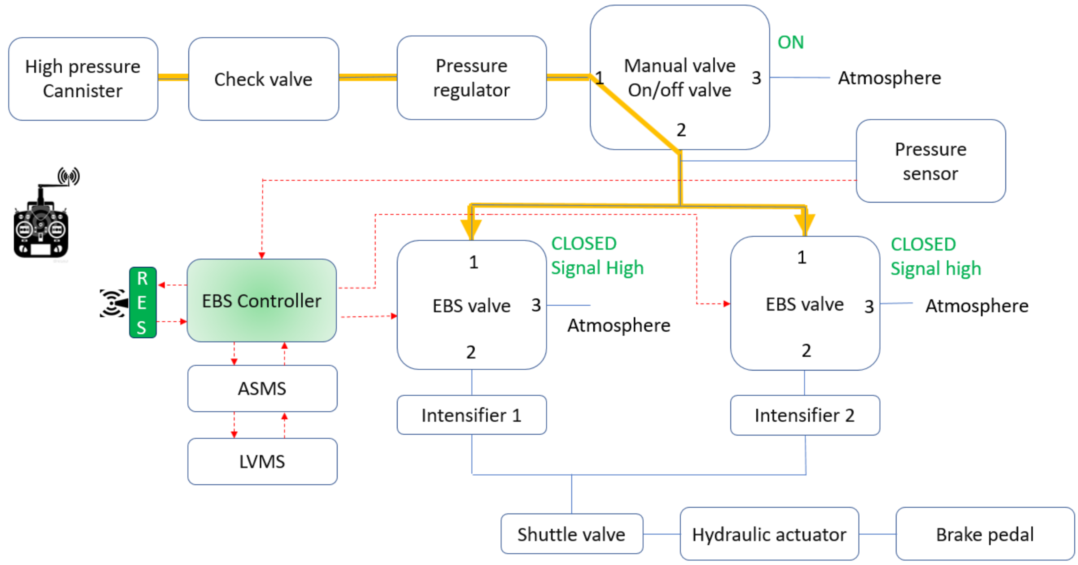


Figure 20 Pressure flow when EBS is armed.

As soon as either an error is detected, low-voltage supply cuts off, the EBS is triggered by RES (Remote emergency system) or the EBS is triggered in the initial check-up, the EBS valve will open and release the pressure to engage the brakes. Figure shows the pressure flow at this state, the two EBS valves are connected to two intensifiers (which converts the Pneumatic pressure at around 10 bar to a higher hydraulic pressure around 100 bar), the intensifiers are connected to the Hydraulic actuator by a shuttle valve (allows only one pressure to flow through the circuit).

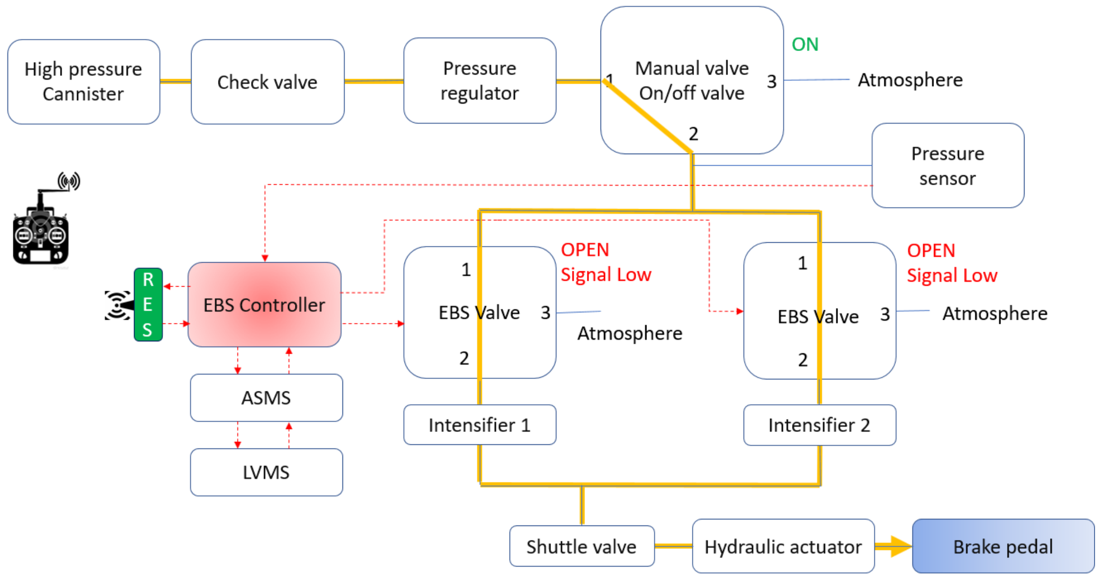


Figure 21 Pressure flow when the EBS has been triggered.

The functionality of the EBS will be tested either by a circuit marshal or during the initial check-up by triggering the EBS and checking the hydraulic brake pressure. This is done by opening the EBS valve to trigger the brake, followed by closing the valve in order to release the pressure to atmosphere and transition back to EBS "armed"-mode1. The pressure flow when releasing the brake after initial check-up is seen in Figure below.

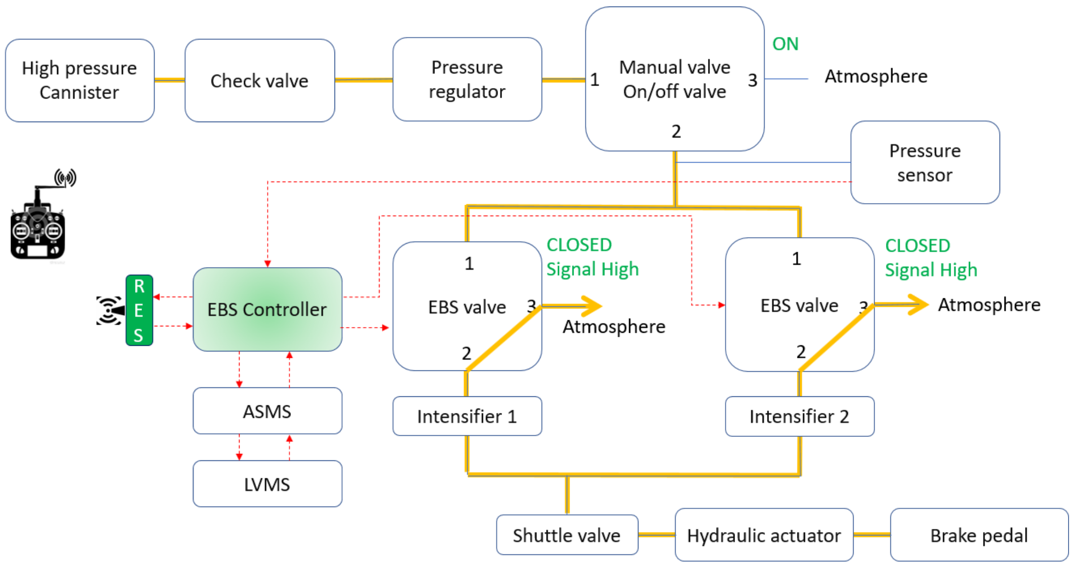


Figure 22 Pressure flow when releasing pressure after the initial checkup.

If the EBS is triggered after the initial check-up due to an error or power loss while driving on the circuit, there is no possibility to close the EBS valve by putting the signal to high again. Instead the pressure needs to be released manually by first closing the first manual on/off valve in order to cut the pressure supply. After this, the pressure still present in the system can be released by opening the second manual on/off valve. See Figure below.

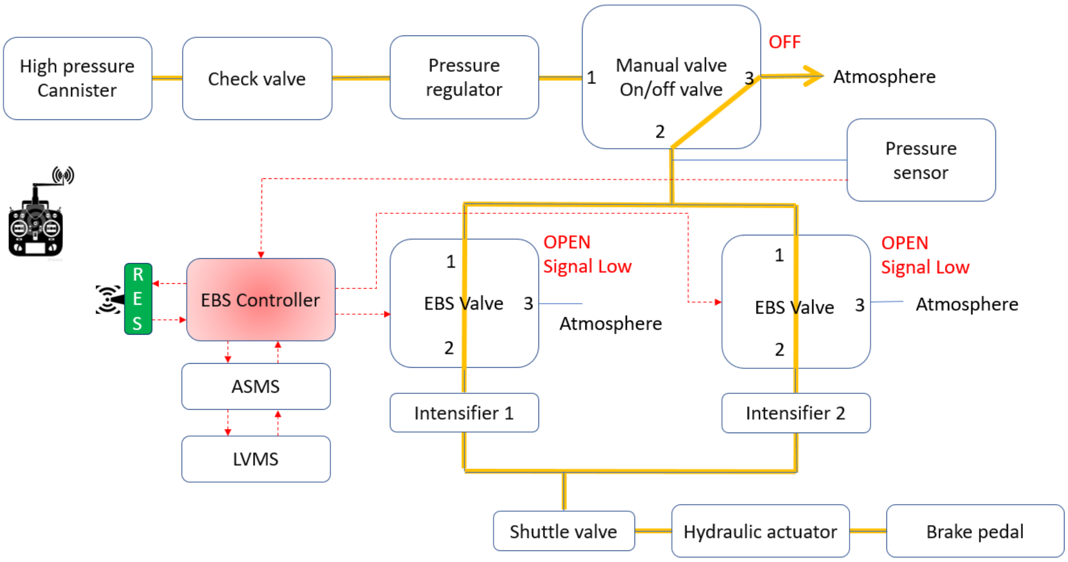


Figure 23 Pressure flow when the EBS has been disabled in order to release the brakes.

Chapter 4

This chapter is about estimating the pedal force which must be applied by the actuator of the EBS, and different cases have been considered for the force selection and correspondingly the deceleration for all the possible cases.

Pedal force calculation

4.1 Estimation of Pedal force from deceleration

Vehicle data for brake force calculation

1. Vehicle mass (Kg) – 187 (without Autonomous driving components - Steering, EBS, camera etc)
2. Mass of Autonomous driving components – around 3 kg
3. Driver mass (Kg) – 75
4. Wheel base(mm) –1525
5. X_G (mm) – 808.25
6. Z_G centre of gravity(m) – 247

Hydraulic Torque repartition

1. Hydraulic torque repartition front to rear = 65:35

Electric torque

1. Electric torque repartition front to rear = Hydraulic torque repartition front to rear

Aerodynamics Data

1. Frontal area (m^2) -1.16
2. Density of Air (kg/m^3) - 1.2
3. C_X – 0.9655
4. C_{ZA} – -1.504
5. C_{ZP} – -1.696

Tyre data

1. Tire Continental C16 - 205/470 R13
2. Nominal wheel radius Front or rear(mm) – 242
3. Loaded radius front or rear(mm) – 237.16

4. μ-x Tire front or Rear – 1.8
5. Moment of inertia front or rear(kg-m^2) – 0.27

Brake Data

1. Brake disc radius front(mm) – 94
2. Brake disc radius rear(mm) – 83
3. Coefficient of friction(μ_{pad}) front or rear – 0.4

Calliper information

1. Font calliper – P4-24
   No of pistons – 4 – Diameter 24 mm – Area – 1809 mm^2
2. Rear calliper – P2-24
   No of pistons – 2 – Diameter 24 mm – Area – 905 mm^2

Master Cylinders information

1. Diameter of Master cylinder front – 16 mm
2. Diameter of Master cylinder rear – 16 mm

Pedal box

1. Pedal gain = 3
2. Spring preload = 280 N
3. Balance bar repartition (front to rear) = 50:50(can also be calibrated for further optimisation, max 60:40)

Pedal force calculation

Front and rear hydraulic pressure are estimated by doing torque balance at the wheel level

$$Pedal\ force_{with\ springs} = Pedal\ force_{Without\ spring\ preload} + Spring\ preload$$

Where,

$$Pedal\ force_{Without\ spring\ preload} = \frac{Front\ Hydraulic\ pressure * Area_{MC\ front} + rear\ Hydraulic\ pressure * Area_{MC\ rear}}{9.81 * pedal\ gain}$$

Where,

$$Front\ Hydraulic\ pressure = \frac{Braking\ torque_{front} + Inertia\ torque_{Front} - Electric\ Brake\ torue_{front}}{Radius\ of\ the\ calliper_{front} * Friction_{brake\ pad} * Calliper\ piston\ area_{front}}$$

$$Rear\ Hydraulic\ pressure = \frac{Braking\ torque_{Rear} + Inertia\ torque_{Rear} - Electric\ Brake\ torue_{rear}}{Radius\ of\ the\ calliper_{Rear} * Friction_{brake\ pad} * Calliper\ piston\ area_{Rear}}$$

Where,

$$Braking\ torque_{front\ or\ rear} = \frac{deceleration * 9.81 * mass * Torque\ repartition_{front\ or\ rear} * Wheel\ radius}{2}$$

$$Inertia\ torque = deceleration * 9.81 * Inertia * wheel\ radius$$

$$Electric\ brake\ torque_{front\ or\ rear} = \frac{Electric\ torque * Electric\ torque\ repartition_{Front\ or\ rear} * Transmission\ ratio_{front\ or\ rear}}{2}$$

4.2 Estimation of deceleration from pedal force

Once the pedal force is determined for various driving scenarios then the deceleration must be found with respect to them all.

$$Deceleration = \frac{2 * (Ground\ froce\ front + Ground\ froce\ Rear)}{Total\ vehicle\ mass * 9.81}$$

Where,

$$Ground\ froce_{front\ or\ rear} = \frac{(Hydraulic\ brake\ torque + Electric\ brake\ torque - Inertia\ torque)_{front\ or\ rear}}{wheel\ radius_{front\ or\ rear}}$$

Where,

$$Inertia\ torque = deceleration * 9.81 * Inertia * wheel\ radius$$

$$Electric\ brake\ torque_{front\ or\ rear} = \frac{Electric\ torque * Electric\ torque\ repartition_{Front\ or\ rear} * Transmission\ ratio_{front\ or\ rear}}{2}$$

$$Hydraulic\ brake\ torque_{front\ or\ rear} = \frac{Force\ on\ Master\ cyliinder_{Front\ or\ rear}}{Area\ master\ cylinder_{front\ or\ rear}} * calliper\ piston\ area_{front\ or\ rear} * \mu_{pad\ front\ or\ rear} * Brake\ disc\ radius_{front\ or\ rear}$$

Where,

$$Force\ on\ Master\ cylinder_{front\ or\ rear} = (pedal\ force - spring\ preload) * 9.81 * pedal\ gain * balance\ bar\ repartition_{front\ or\ rear}$$

Where,

$$Torque\ repartition\ front\ = \frac{(C_{FA})}{(C_{FA} + C_{Fp})}$$

4.3 Pedal force for different cases

Cases of brake forces

Estimating the pedal force is done for different scenarios of braking, same has to be considered before selecting the desired Hydraulic actuator which will satisfy our brake force requirement

1. Regenerative braking + driver + spring preload on brake pedal
2. Regenerative braking + without driver + spring preload on brake pedal
3. Regenerative braking + without driver + without spring preload on brake pedal
4. Without Regenerative braking + driver + spring preload on brake pedal
5. Without Regenerative braking + without driver + spring preload on brake pedal
6. Without Regenerative braking + without driver + without spring preload on brake pedal

Note:

1. According to DV2.2.6 of the FSG 2020 regulations [2], the driver cannot be inside the car while the car is in autonomous mode, but brake force calculations have to be done considering the EBS test can be done with a driver inside to apply brakes manually if the EBS doesn’t work.
2. Regenerative braking will not be present during power loss to the system, but regenerative braking can be active in case of the EBS actuation done with the RES (remote emergency system)

Case no. 5 (Without Regenerative braking + without driver + spring preload on brake pedal) and case no.6 (Without Regenerative braking + without driver + without spring preload) are the cases to be considered to have the pedal force estimation, since the driver won’t be present in the vehicle during autonomous driving and during EBS (Emergency brake system) actuation the driver weight is to be neglected, and EBS should act in case of power loss to the LV (Low voltage system) and hence the contribution of regenerative braking is also disregarded. And thereby calculating the pedal force.

Further with this pedal force, deceleration for the cases of “without driver” are found to make sure the vehicle will satisfy the deceleration of 0.8g and complete vehicle stoppage in 10m from an initial velocity of 40 kmph according to the FS rules.

4.4 Pedal force results

ESTIMATE OF PEDAL FORCE FROM THE DECELERATION (Considering various conditions of braking)		Without Regen		
		Driver	Without Driver	
		with Springs		Without Springs
		Vehicle + driver + spring preload	Vehicle + spring Preload	Vehicle (No driver and spring preload)
Deceleration	g	1.8	1.8	1.8
Vehicle		190	190	190
Driver		75	0	0
Vehicle mass	kg	265	190	190
Total torque repartition front	%	65	65	65
Friction tire	/	1.8	1.8	1.8
Wheel radius	m	0.237	0.237	0.237
Inertia	kg*m^2	0.27	0.27	0.27
Elecrtic torque	Nm	0	0	0
Electric torque repartition fror	%	65	65	65
Front transmission ratio	/	14.8	14.8	14.8
Rear transmission ratio	/	14.8	14.8	14.8
Front rotor radius	m	0.094	0.094	0.094
Rear rotor radius	m	0.083	0.083	0.083
Friction brake pad	/	0.4	0.4	0.4
Caliper pistons front area	mm^2	1809.6	1809.6	1809.6
Caliper pistons rear area	mm^2	904.8	904.8	904.8
Front hydraulic pressure	bar	53.13834632	38.14619178	38.14619178
Rear hydraulic pressure	bar	64.98367681	46.69852817	46.69852817
Front master cylinder diamete	mm	16	16	16
Rear master cylinder diamete	mm	16	16	16
Spring preload	kg	0	0	0
Pedal gain ratio	/	3	3	3
Pedal force input without spri	kg	80.65851773	57.93542277	57.93542277
Spring stiffness	N/m	26754	26754	0
Spring preload	kg	15	15	0
Pedal force input with springs	kg	108.8399652	86.11687027	57.93542277
Pedal force input with springs	N	1067.720059	844.8064974	568.3464974

Figure 24 Results – Pedal force Without regenerative braking

ESTIMATE OF PEDAL FORCE FROM THE DECELERATION (Considering various conditions of braking)		With regen		
		Driver	Without driver	
		with Springs		Without Springs
		Vehicle + driver + spring preload	Vehicle + spring Preload	Vehicle (No driver and spring preload)
Deceleration	g	1.8	1.8	1.8
Vehicle		190	190	190
Driver		75	0	0
Vehicle mass	kg	265	190	190
Total torque repartition front	%	65	65	65
Friction tire	/	1.8	1.8	1.8
Wheel radius	m	0.237	0.237	0.237
Inertia	kg*m^2	0.27	0.27	0.27
Elecrtic torque	Nm	25	25	25
Electric torque repartition fror	%	65	65	65
Front transmission ratio	/	14.8	14.8	14.8
Rear transmission ratio	/	14.8	14.8	14.8
Front rotor radius	m	0.094	0.094	0.094
Rear rotor radius	m	0.083	0.083	0.083
Friction brake pad	/	0.4	0.4	0.4
Caliper pistons front area	mm^2	1809.6	1809.6	1809.6
Caliper pistons rear area	mm^2	904.8	904.8	904.8
Front hydraulic pressure	bar	35.46516829	20.47301374	20.47301374
Rear hydraulic pressure	bar	43.4286237	25.14347507	25.14347507
Front master cylinder diamete	mm	16	16	16
Rear master cylinder diameteı	mm	16	16	16
Spring preload	kg	0	0	0
Pedal gain ratio	/	3	3	3
Pedal force input without spri	kg	53.87188732	31.14879236	31.14879236
Spring stiffness	N/m	26754	26754	0
Spring preload	kg	15	15	0
Pedal force input with springs	kg	82.05333482	59.33023986	31.14879236
Pedal force input with springs	N	804.9432146	582.029653	305.569653

Figure 25 Results - Pedal force with regenerative braking

4.5 Deceleration results

Pedal forces of case 5 and case 6 is selected to obtain the required deceleration for the cases without driver, and all the cases satisfy the required minimum deceleration according to the FS rules.

ESTIMATION OF THE DECELERATION FROM THE FORCE TO THE PEDAL (without driver and with springs)					
		spring preload + no regen	no Spring preload + no regen	spring preload + regen	no spring preload + regen
Pedal force	**kg**	**86.12**	**86.12**	**86.12**	**86.12**
Springs preload	kg	28.00	0.00	28.00	0.00
Pedal gain ratio	/	3.00	3.00	3.00	3.00
Diametro Master cylinder fron	mm	16.00	16.00	16.00	16.00
Diametro Master Cylinder rea	mm	16.00	16.00	16.00	16.00
Balance bar repartition front	%	50.00	50.00	50.00	50.00
Caliper pistons area front	mm^2	1809.56	1809.56	1809.56	1809.56
Caliper pistons area rear	mm^2	904.78	904.78	904.78	904.78
Rotor radius front	m	0.09	0.09	0.09	0.09
Rotor radius rear	m	0.08	0.08	0.08	0.08
Wheel radius	m	0.24	0.24	0.24	0.24
Inertia	kg*m^2	0.27	0.27	0.27	0.27
Friction pad-rotor	**/**	**0.40**	**0.40**	**0.40**	**0.40**
Friction tyre	/	1.80	1.80	1.80	1.80
Total hydraulic torque repartit	%	69.37	69.37	69.37	69.37
Total electric torque	Nm	0.00	0.00	25.00	25.00
Electric torque repartition [fro	%	69.37	69.37	69.37	69.37
Front transmission ratio	/	14.80	14.80	14.80	14.80
Rear transmission ratio	/	14.80	14.80	14.80	14.80
Vehicle total mass	kg	180.00	180.00	180.00	180.00
Ground force front	N	1136.89	1725.49	1678.40	2267.00
Ground force rear	N	454.52	714.38	693.59	953.45
Total torque repartition [front	%	0.71	0.71	0.71	0.70
Deceleration	**g**	**1.80**	**2.76**	**2.69**	**3.65**

Figure 26 Results - Deceleration from pedal force (With spring preload)

ESTIMATION OF THE DECELERATION FROM THE FORCE TO THE PEDAL (without driver and without springs)					
		spring preload + no regen	no Spring preload + no regen	spring preload + regen	no spring preload + regen
Pedal force	**kg**	**57.94**	**57.94**	**57.94**	**57.94**
Springs preload	kg	28.00	0.00	28.00	0.00
Pedal gain ratio	/	3.00	3.00	3.00	3.00
Diametro Master cylinder fron	mm	16.00	16.00	16.00	16.00
Diametro Master Cylinder rear	mm	16.00	16.00	16.00	16.00
Balance bar repartition front	%	50.00	50.00	50.00	50.00
Caliper pistons area front	mm^2	1809.56	1809.56	1809.56	1809.56
Caliper pistons area rear	mm^2	904.78	904.78	904.78	904.78
Rotor radius front	m	0.09	0.09	0.09	0.09
Rotor radius rear	m	0.08	0.08	0.08	0.08
Wheel radius	m	0.24	0.24	0.24	0.24
Inertia	kg*m^2	0.27	0.27	0.27	0.27
Friction pad-rotor	**/**	0.40	0.40	0.40	0.40
Friction tyre	/	1.80	1.80	1.80	1.80
Total hydraulic torque repartit	%	69.37	69.37	69.37	69.37
Total electric torque	Nm	0.00	0.00	25.00	25.00
Electric torque repartition [fro	%	69.37	69.37	69.37	69.37
Front transmission ratio	/	14.80	14.80	14.80	14.80
Rear transmission ratio	/	14.80	14.80	14.80	14.80
Vehicle total mass	kg	180.00	180.00	180.00	180.00
Ground force front	N	544.50	1133.10	1086.02	1674.62
Ground force rear	N	192.98	452.85	432.06	691.92
Total torque repartition [front	%	0.74	0.71	0.72	0.71
Deceleration	**g**	**0.84**	**1.80**	**1.72**	**2.68**

Figure 27 Results - Deceleration from pedal force (Without spring preload)

Chapter 5

This chapter deals with the calculations regarding the placement of actuator behind the brake pedal, calculations for solution using Pneumatic actuator which is not compatible due to oversize of the actuator and the calculations for using a Hydro-Pneumatic arrangement for the EBS.

Calculation for EBS components

5.1 Calculations for actuator placement

Any actuator can have specifications as

1. Force that the actuator provides during pulling – depends upon inclination
2. Stroke of the actuator – depends upon pedal travel

The above specifications becomes a function of geometry in our pedal box arrangement on how the actuator is placed, it more or less looks like an inversion of slider crank mechanism, to estimate the exact force and stroke of the actuator a kinematic study of the hard points have to be done either using a drawing tool like Catia or using basic geometrical formulas.

5.1.1 Skeleton study for the estimation of hard points of the Actuator placement

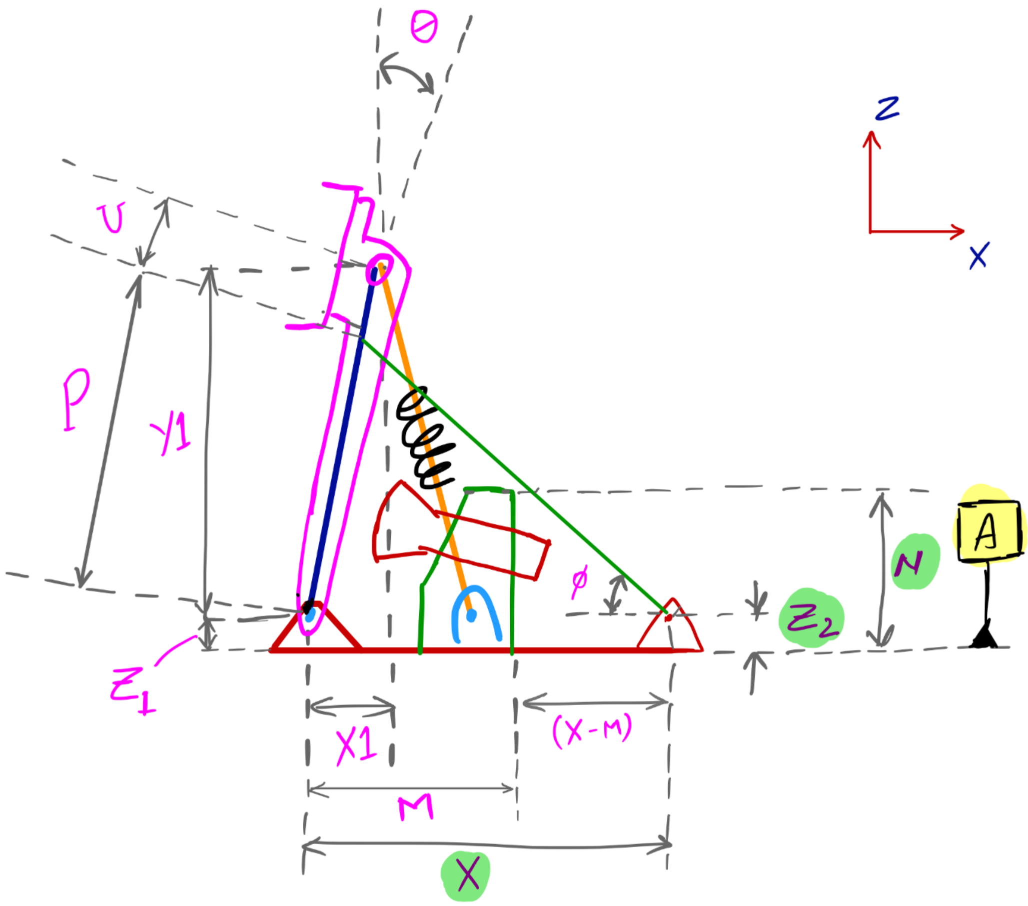
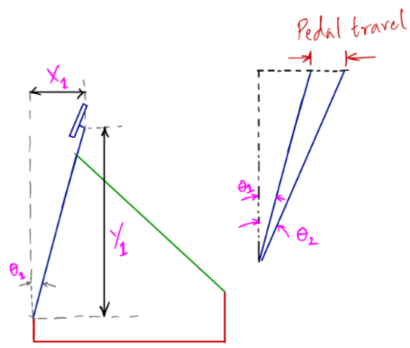


Figure 28 Skeleton of Pedal box – hard points for Actuator

This study was done to estimate the length of the support bracket and length of the actuator in compressed and extended position.

Geometrical calculations are formed to set the dimensions X, Z_2 and N as variables and find the inclination of the actuator and the stroke of the actuator and the length of the support bracket.

Geometry of the Assembly (Hard points)



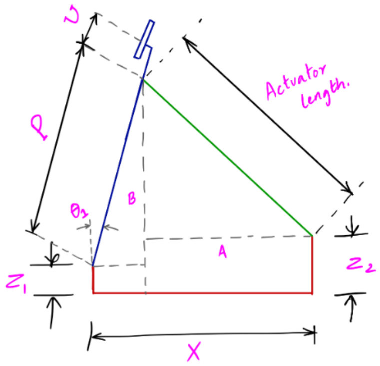
length_brakepedal = sqrt(X1^2 + Y1^2)

P = length_brakepedal – U

theta1 = atan(X1/Y1)*180/pi

theta2 = asin((X1+travel)/length_brakepedal)*180/pi

Actuator length at Initial position of the pedal



A1 = X - (P*cos((90-theta1)*pi/180))

B1 = (P*sin((90-theta1)*pi/180)) + Z1 - Z2

Actuator_length_extended= sqrt(A1^2 + B1^2)

Full braking position of the pedal

A2 = X - (P*cos((90-theta2)*pi/180))

B2 = (P*sin((90-theta2)*pi/180)) + Z1 - Z2

Actuator_length_compressed= sqrt(A2^2 + B2^2)

Angles of the Actuator

phi(minimum to avoid interferance)= atan((N-Z2)/(X-M))*180/pi

phi1(angle in relaxed position) = atan(B1/A1)*180/pi

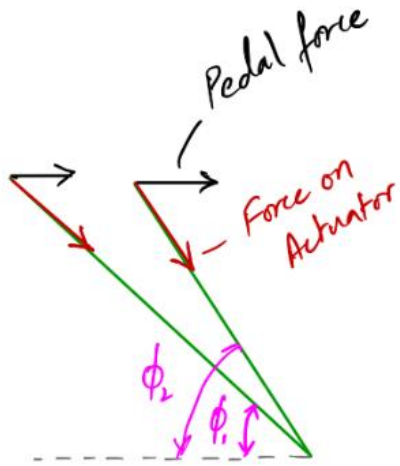
phi2(angle in compressed position) = atan(B2/A2)*180/pi

Force and Stroke requirement

Stroke of the actuator (exact) = (Actuator_length_extended - Actuator_length_compressed)

min_force = Force/cos(phi1*pi/180)

max_force = Force/cos(phi2*pi/180)



5.1.2 Estimation of the stroke and actuator placement using CAD drawings

Stroke length estimation

Using Catia drawing environment, the stroke and force requirement is done to have a matching with the skeleton study

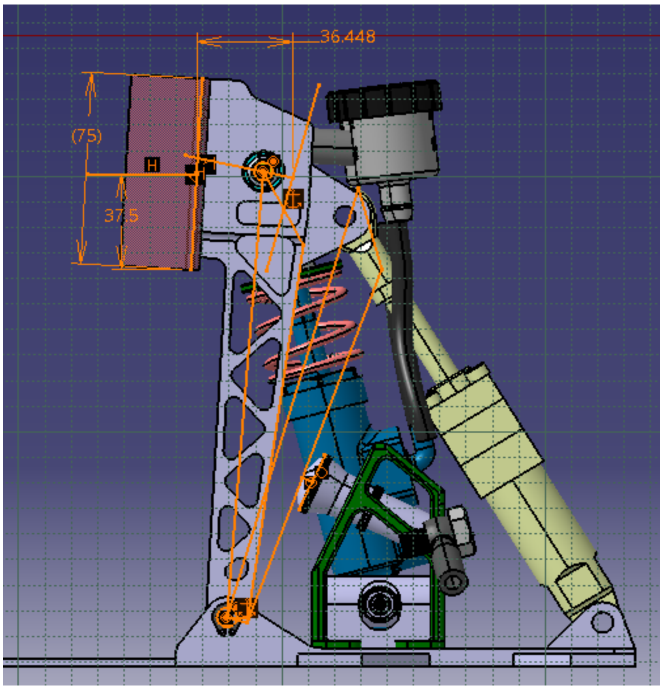


Figure 29 Pedal travel for BOTS actuation

Above figure shows the pedal travel required for the pressing of the BOTS (Brake over travel switch) in case of loss of pressure in the Brake line. the maximum brake travel is 25 mm for full brake actuation and 38 mm is the brake travel required for the actuation of the BOTS which is connected to the shutdown circuit.

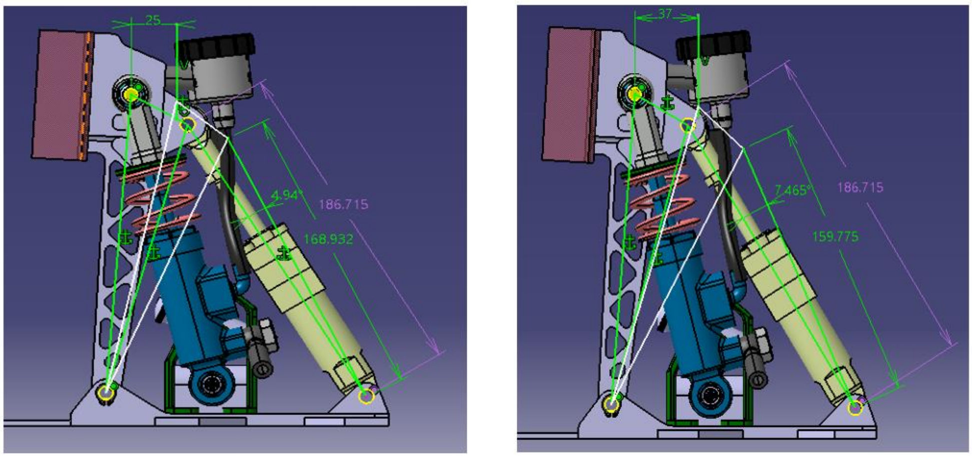


Figure 30 Actuator stroke for full braking and BOTS actuation

The above figure shows that for a 25 mm of the pedal travel we require the actuator stroke should be 186.715 – 168.992 = 18 mm and for a pedal travel of 37 mm (required for BOTS actuation) we need a stroke of 186.715 – 159.775 = 27 mm and with tolerance of 2 mm, we selected an actuator with 29~30 mm.

5.1.3 Force multiplying factor for the actuator

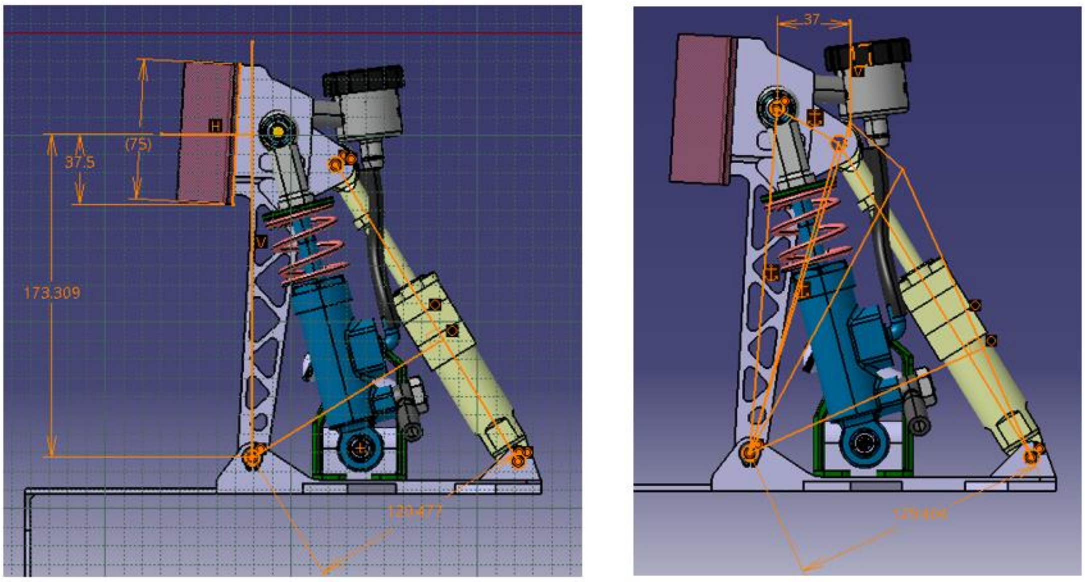


Figure 31 Force by the actuator (Amplification factor of the actuator)

Actuator amplifying factor can be estimates a one maximum value and a minimum value as the actuator will be compressed and swivel around the hinged point.

Actuator amplification factor (Maximum) = 173.3/120.5 = 1.44

Actuator amplification factor (Minimum) = 173.3/129.4 = 1.34

Force to be provided by the actuator = pedal force * Actuator amplification factor

Input for the pedal		
Pedal travel to activate kill switch(mm)	37	mm
stroke of the actuator(mm) for corresponding ped	27	mm
Force on pedal by driver for max braking (N)	845	N
Amplification factor for the actuator	1.44	
Force[N] that has to be provided by the Actuator	1216.8	N

Load factor of the Brake actuator due to actuation rate

According to the Formula student regulations the brake actuation has to be happened with a maximum time of 200 milli seconds, and due to such high actuation rated hydraulic actuators cannot provide the desired force, hence they have to be calibrated with a load factor of 60 percent [3]

$$Theoretical\ load = \frac{Actual\ load}{0.6}$$

Constraints		
Actuation time(<= 200 ms) FSD regulation	150	ms
Load on actuator	1217	N
Stroke of the actuator(with tolerance)	29	mm
Piston dia of actuator	20	mm
Rod dia of actuator	10	mm

5.2 Calculations for Pneumatic actuator

The bore of Pneumatic actuator is larger (around 60 mm) to accommodate in the brake assembly to provide the required power rating, both in both the pull and push solution hence the solution with pneumatic actuator is disregarded.

5.2.1 Pull solution

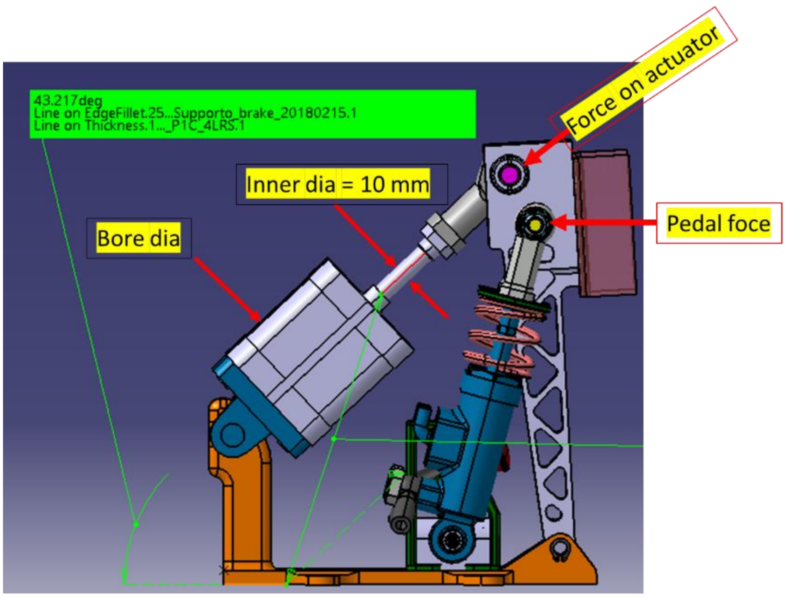


Figure 32 Pneumatic actuator

With spring preloading (Pull actuator)			
Considering Actuation time < 200ms, load factor 60 %	Pedal force with spring preloading	845	N
	Amplification factor for the actuator	1.44	
	Pneumatic Presssure	1	Mpa
	Force	1216.8	N
	Load factor	0.6	60%
	Theoretical load	2028	N
	Stroke(27 mm + 2 mm tolerance)	29	mm
	Rod inner dia	10	mm
	Bore dia	61	mm

Figure 33 Bore calculation for pneumatic actuator (Pull with spring preload)

Without spring preloading(Pull actuator)			
Considering Actuation time < 200ms, load factor 60 %	Pedal force without spring preloading	569	N
	Amplification factor for the actuator	1.44	
	Pneumatic Presssure	1	Mpa
	Force	819.36	N
	Load factor	0.6	60%
	Theoretical load	1365.6	N
	Stroke(27 mm + 2 mm tolerance)	29	mm
	Rod inner dia	10	mm
	Bore dia	52	mm

Figure 34 Bore calculation for pneumatic actuator (Pull without spring preload)

5.2.2 Push solution

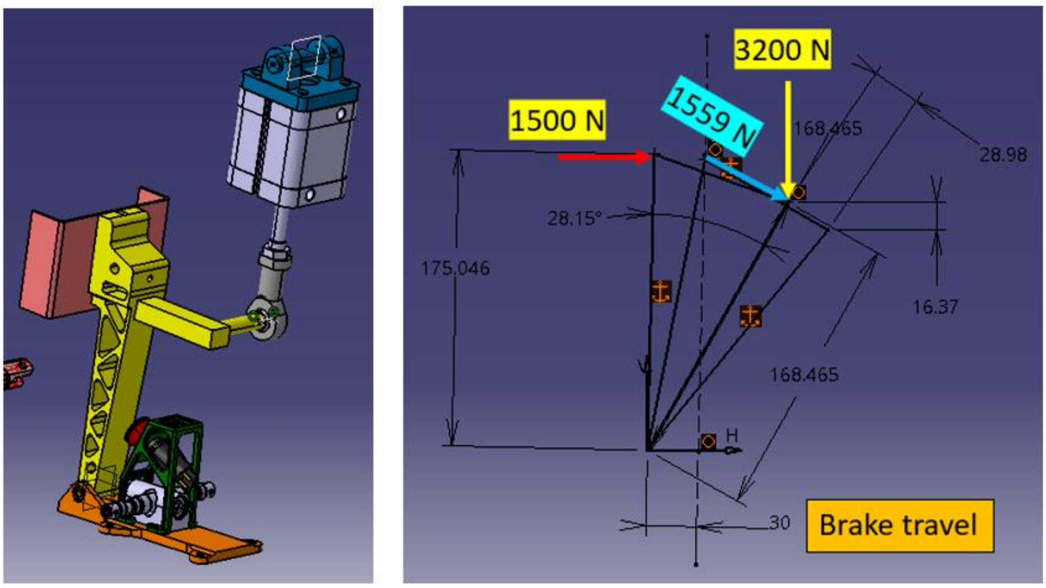
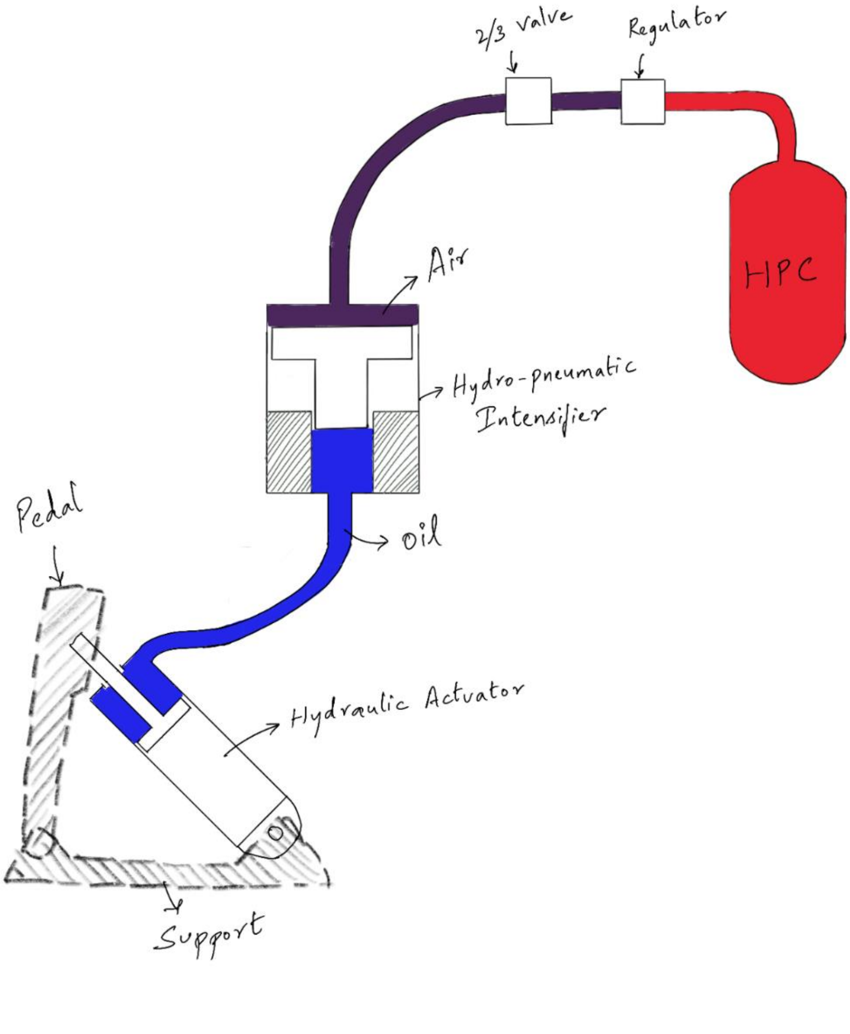


Figure 35 Pneumatic actuator (Push)

Push solution for pneumatic actuator requires a bigger force requirement, hence this cannot be implemented.

5.3 Calculations for Hydro-pneumatic system

Important components for our arrangement of Hydro-pneumatic system are a Hydraulic actuator, Intensifiers (Converts the lower pneumatic pressure into higher hydraulic pressure), solenoid actuated valves (2/3 valves), pressure regulator and a high-pressure gas cannister (around 200 bar).



Hydro-pneumatic calculations are started with selecting the outer and inner diameter of the hydraulic actuator, we selected the OD to be 20 mm and ID to be 10 mm with a stroke of 30 mm, and the oil side diameter of the intensifier is kept at around 70~80 % of the OD of the Actuator, and accordingly the rest of the dimensions like Stroke of the Intensifier, OD of the Intensifier are obtained. [3]

5.3.1 Pressure and volume flow rate in Hydraulic Actuator

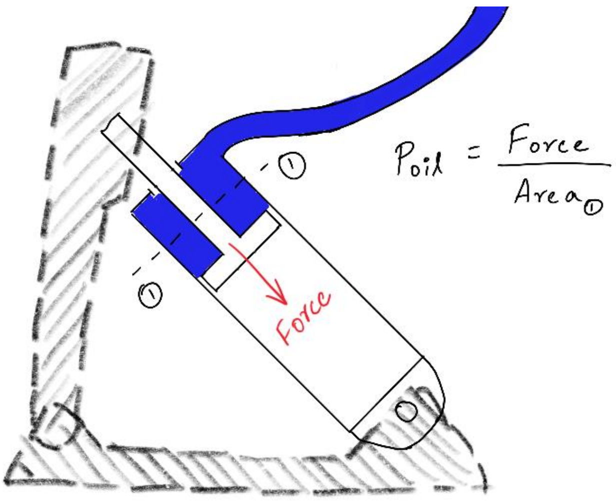


Figure 36 Pressure in Hydraulic actuator

$$Hydraulic\ Pressure = \frac{Theoretical\ load}{Area\ of\ pulling\ side}$$

$$Area_{Pulling} = \frac{\pi(D^2 - d^2)}{4}$$

$$Volume\ flow\ rate_{Actuator} = \frac{Area_{Pulling} * Stroke}{Actuation\ time}$$

$$Volume_{actuator\ durig\ pulling} = Stroke * Area_{Pulling}$$

Actuator		
Description	**Value**	**Unit**
Load factor(based on time of actuation)	0.6	60%
Force(Theoretical) with load facor	2028	N
Area (Pulling)	235.62	mm^2
Volume in the Actuator	6832.98	mm^3
Pressure in the hydraulic actuator	8.61	Mpa
Speed(actuator cylinder)	193.333333	mm/sec
Volume flow rate in actuator	45553.2	mm^3/sec
Time	0.15	s

Figure 37 Hydraulic actuator calculation

Design pressure of the actuator can be as high as 300 bar since the actuator is made of high strength material, according to our calculation the pressure inside the actuator was around 65 to 120 bar based on our force requirement

5.3.2 Intensifiers (Pressure multipliers)

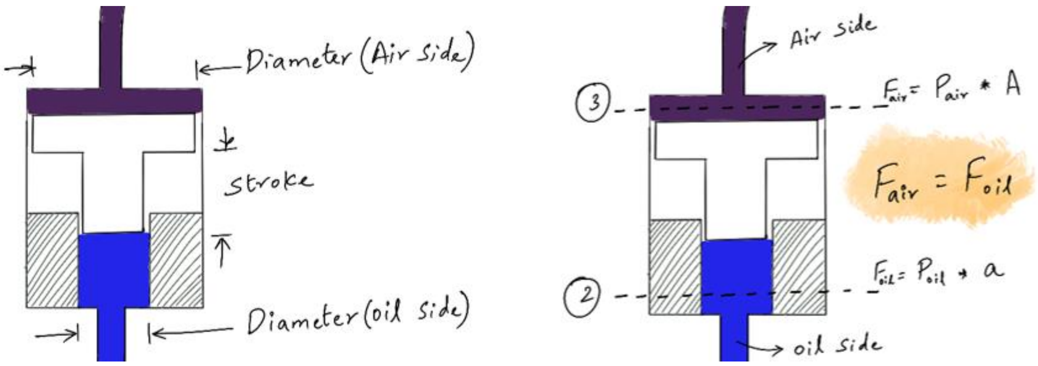


Figure 38 Intensifiers (Pneumatic to hydraulic pressure multiplier)

$$oil\ side\ diameter = 70\ to\ 80\ \%\ of\ the\ OD\ of\ the\ Hydraulic\ actuator$$

$$Stroke_{Intensifier\ oil\ side} = \frac{Volume_{actuator\ durig\ pulling}}{Area_{oil\ side}}$$

Air side diameter is obtained by doing force equilibrium on the air side and the oil side of the intensifier

$$Air\ side\ diameter = \sqrt{\frac{Hydraulic\ pressure}{Pneumatic\ pressure}} * oil\ side\ diameter$$

Since the actuator inlet is connected to the oil side of the intensifier the flow rates remain the same

$$Volume\ flow\ rate_{oil\ side\ intensifier} = Volume\ flow\ rate_{Actuator}$$

$$Speed_{oil\ side} = \frac{Volume\ flow\ rate_{oil\ side\ intensifier}}{Area_{oil\ side}}$$

Since the piston inside the intensifier moves the same distance for each stroke up and down and with the same speed

$$Stroke_{Intensifier\ oil\ side} = Stroke_{Intensifier\ Air\ side}$$

$$speed_{Air\ side} = speed_{oil\ side}$$

$$Volume\ flow\ rate_{oil\ side\ intensifier} = speed_{Air\ side} * Area_{Air\ side}$$

$$Volume_{Air\ side\ per\ stroke} = Area_{Air\ side} * Stroke_{Intensifier\ Air\ side}$$

Constraints		
Pneumatic Pressure	1	Mpa

Intensifier		
Oil side		
Description	**Value**	**Unit**
Diameter(oil side)	14	mm
Area(Oil side)	153.94	sq.mm
Volume(Oil side) Intensifier	6832.98	mm^3
Stroke(oil side)	44	mm
Volume Flow rate oil side (Intensifier)	45553.2	mm^3/sec
Speed oil side piston(Intensifier)	295.91529	mm/sec
Time	0.15	s

Figure 39 Calculations for Intensifier (oil side)

Intensifier		
Air side		
Description	**Value**	**Unit**
Diameter(Air side)	41	mm
Area(Air side)	1320.25	sq.mm
Stroke(Air side)	44	mm
Volume(Air side) per stroke	58091	mm^3
Speed Air side piston(Intensifier)	295.9152917	mm/sec
Volume Flow rate in air side(Intensifier)	390683.44	mm^3/sec
Time	0.15	s

Figure 40 Calculations for Intensifier (Air side)

5.3.3 High pressure Gas canister (Pressure Accumulator)

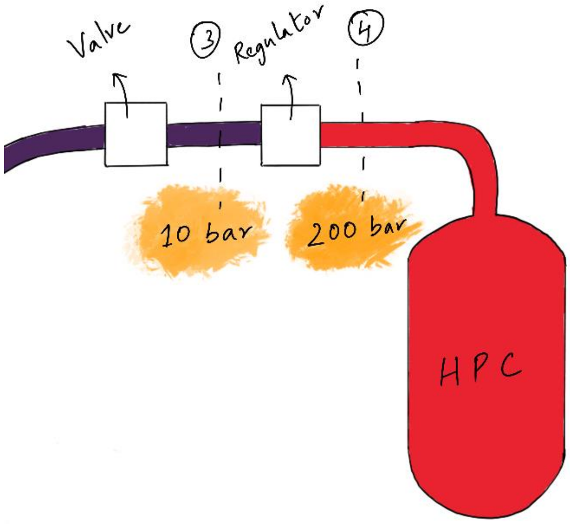


Figure 41 Accumulator (High pressure Gas cannister with Pressure regulator)

High pressure gas cannister must be used based upon the number of times it can be used during testing and in real time application, usually cannisters are selected based upon its pressure and volume, for our design we decided to use a 200-bar pressure and out 0.2 litre volume.

$$Expected\ volume\ at\ 10\ bar = \frac{Cannister\ volume * cannister\ pressure}{Air\ pressure}$$

$$No. of\ times\ can\ be\ used = \frac{Expected\ volume\ at\ 10\ bar}{Volume_{Air\ side\ per\ stroke}}$$

Accumulator		
Description	**Value**	**Unit**
Cannister volume	0.2	litre
Cannister volume	200000	cubic mm
canister pressure	200	Bar
pressure line	10	Bar
expected volume of air at 10 bar	4000000	cubic mm
no of times	69	n°
for 2 intensifiers	34.5	n°

Figure 42 Calculation for Accumulator

Chapter 6

Brake pedal and Support bracket are designed to accommodate the actuator behind and the same have been analysed for safe stresses under both pull (Actuator force) and push conditions (Leg force)

Design of Brake pedal and brake support

6.1 CAD models

Base support

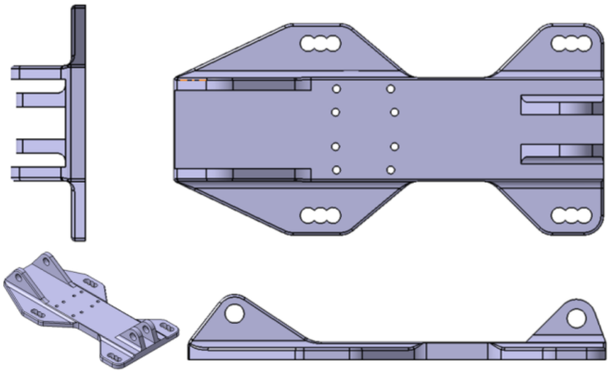


Figure 43 CAD Support bracket

Brake Pedal

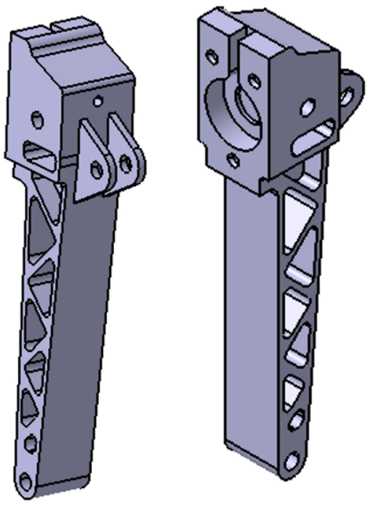


Figure 44 CAD Brake pedal

6.2 FEM analysis

Cases for FEM analysis

1. Push condition (As per FS rules) – 2000 N push
2. Pull Condition (since actuator will be pulling the brake pedal) – 1550 N pull

FEM Analysis shows stresses are considerably below the yield stresses for both the pedal and Base support in both push and pull conditions, a topological optimisation study can be done to further decrease the weight.

6.2.1 Push Condition

As per Formula student rules the pedal box must be designed for the pedal force of minimum 2000 N, and analysis has been done considering the same.

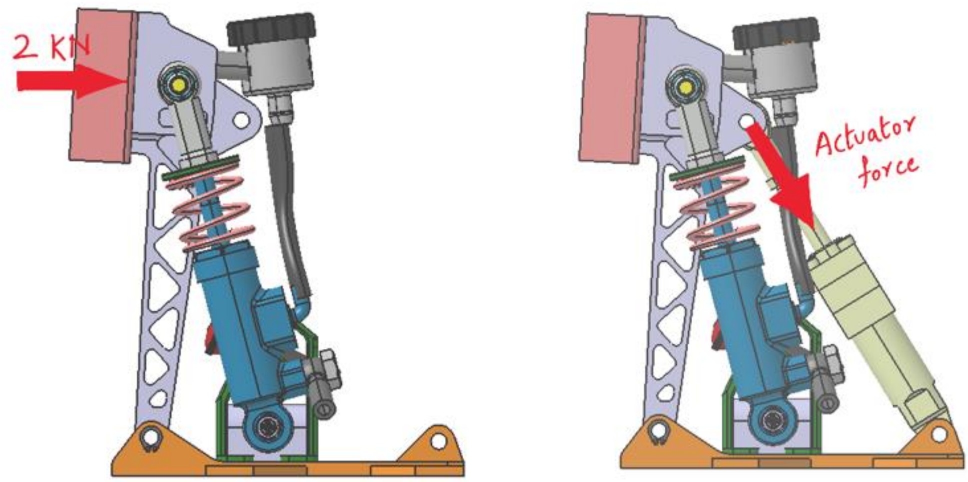


Figure 45 Force application on pedal in XZ plane for push and pull conditions

Force application is as realised as shown in the figure

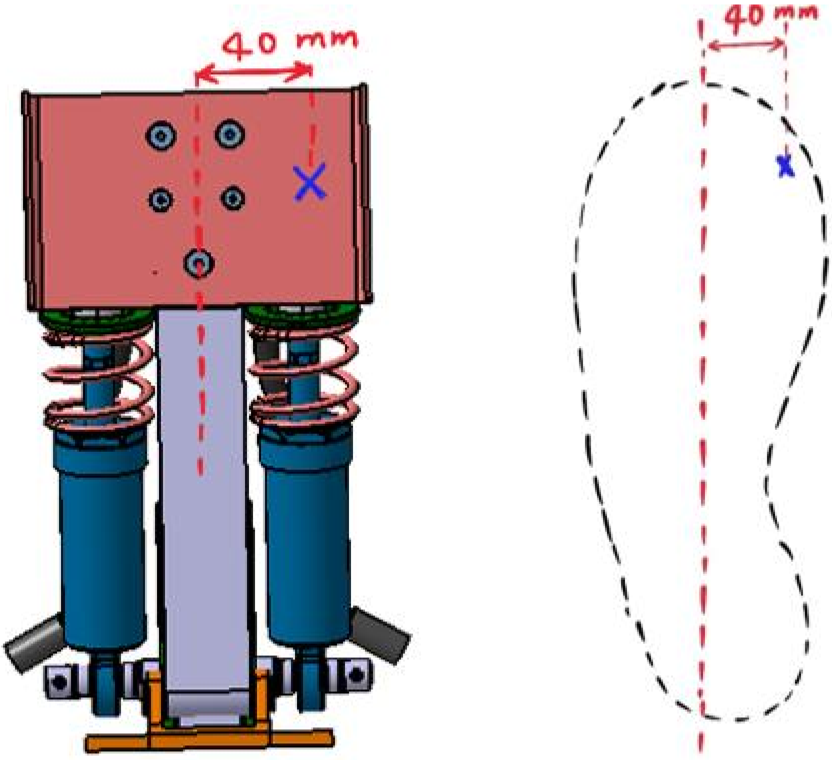


Figure 46 Force application on pedal in YZ plane for push condition

Solver Used: Optistruct

Element Size: 2mm

Material: Aluminium 7075

Material Properties

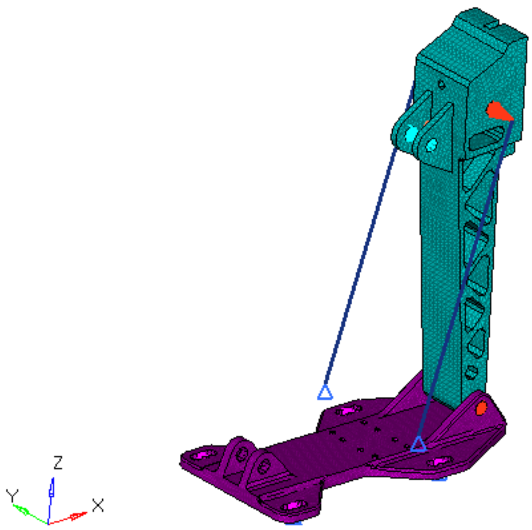
Elastic Modulus: 70000 N/mm2

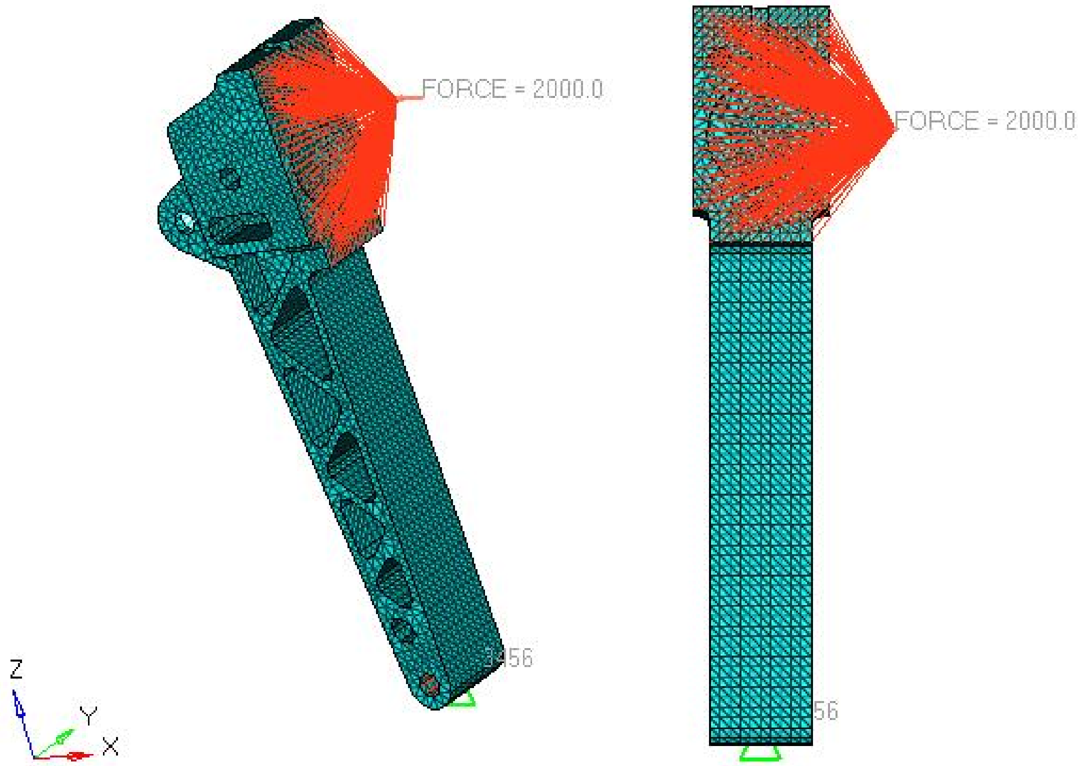
Density: 2.71e-006 kg/mm3

Poisson's ratio: 0.346

Element Type: Tetrahedral

Units used: N mm s





Displacement: pedal (push)

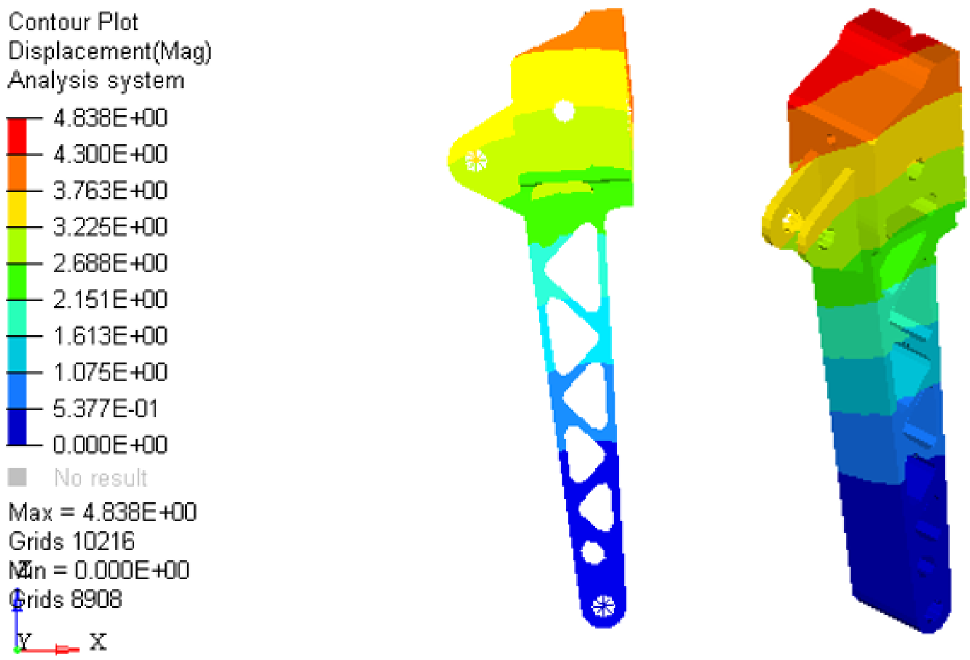


Figure 47 FEM – Displacement Pedal (Push force)

Stresses: pedal (push)

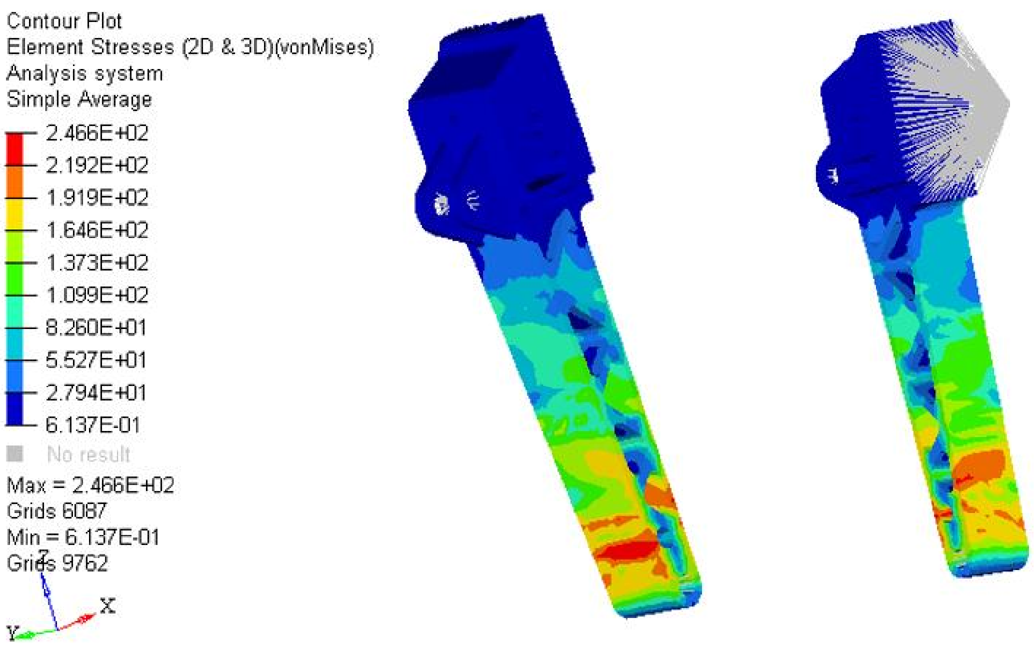


Figure 48 FEM – Stresses Pedal (Push force)

Displacement: base support (push)

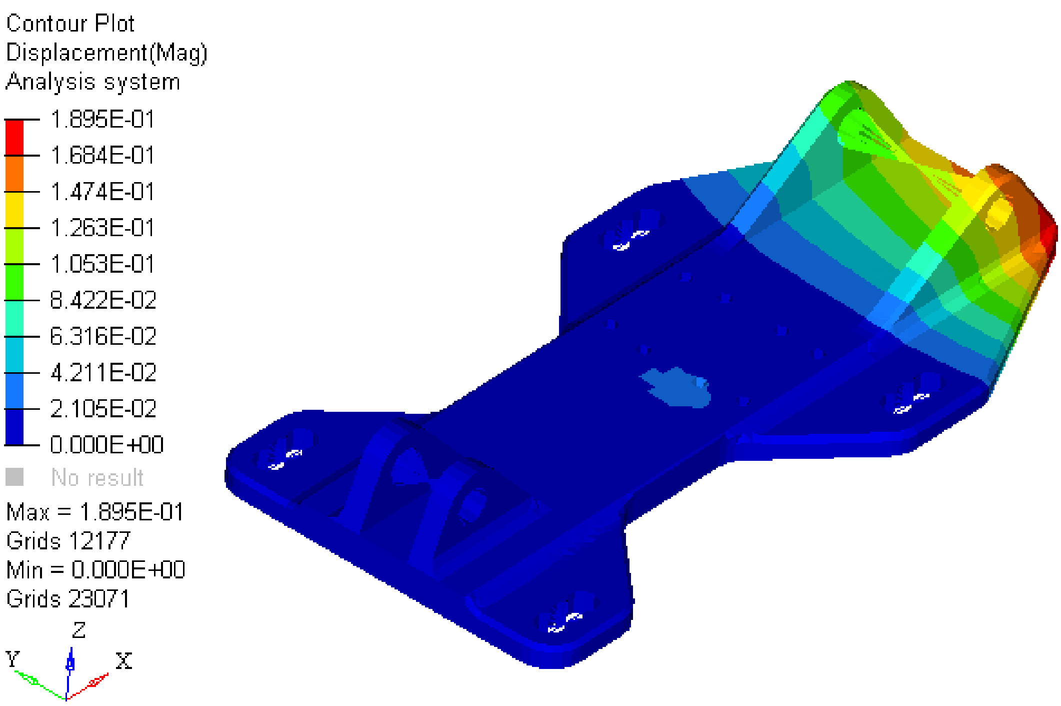


Figure 49 FEM – Displacement Base bracket (Push force)

Stresses: Base support (push)

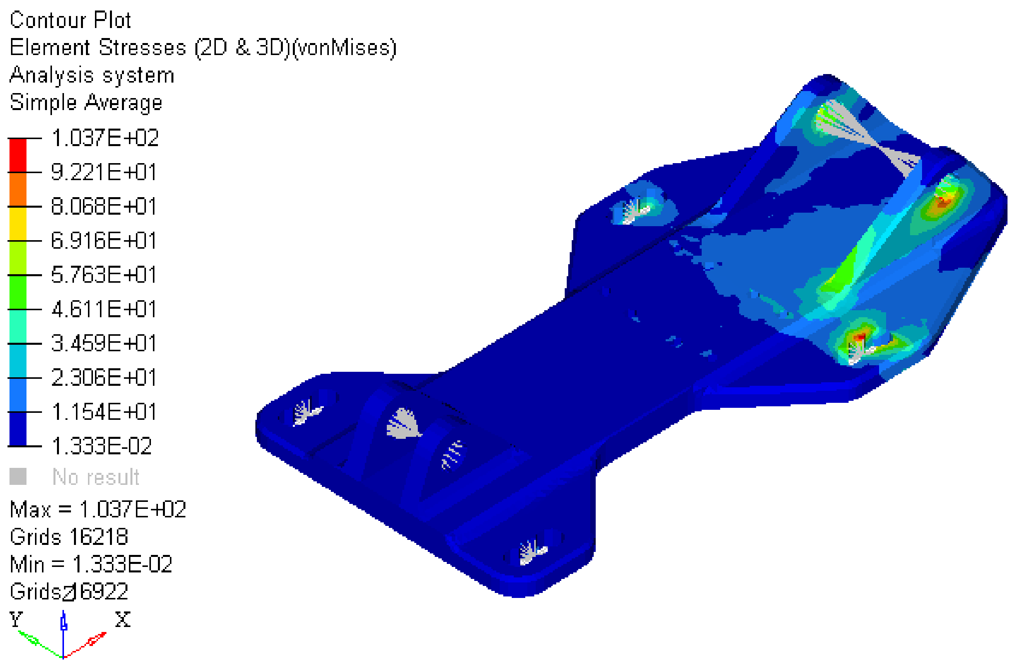


Figure 50 FEM – Stresses Pedal (Push force)

6.2.2 Pull condition

Displacement: pedal (pull)

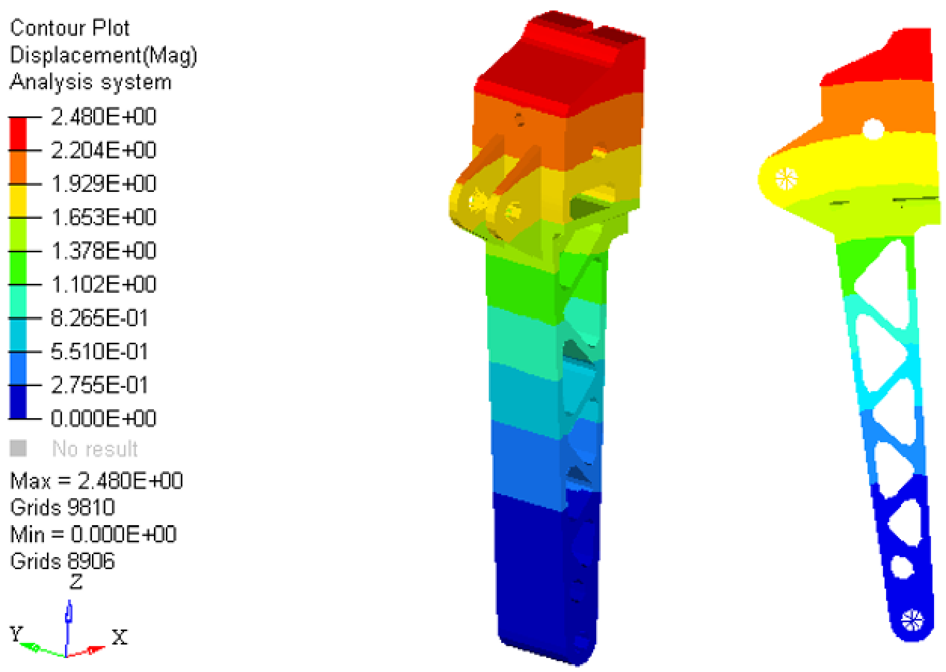


Figure 51 FEM – Displacement Pedal (Pull force)

Stresses: pedal (pull)

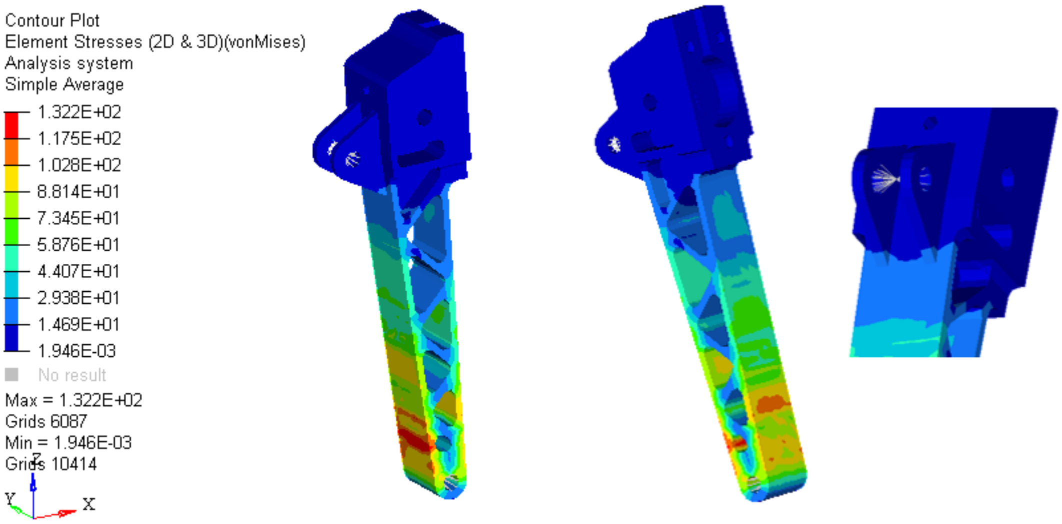


Figure 52 FEM – Stresses Pedal (Pull force)

Displacement: Base support (pull)

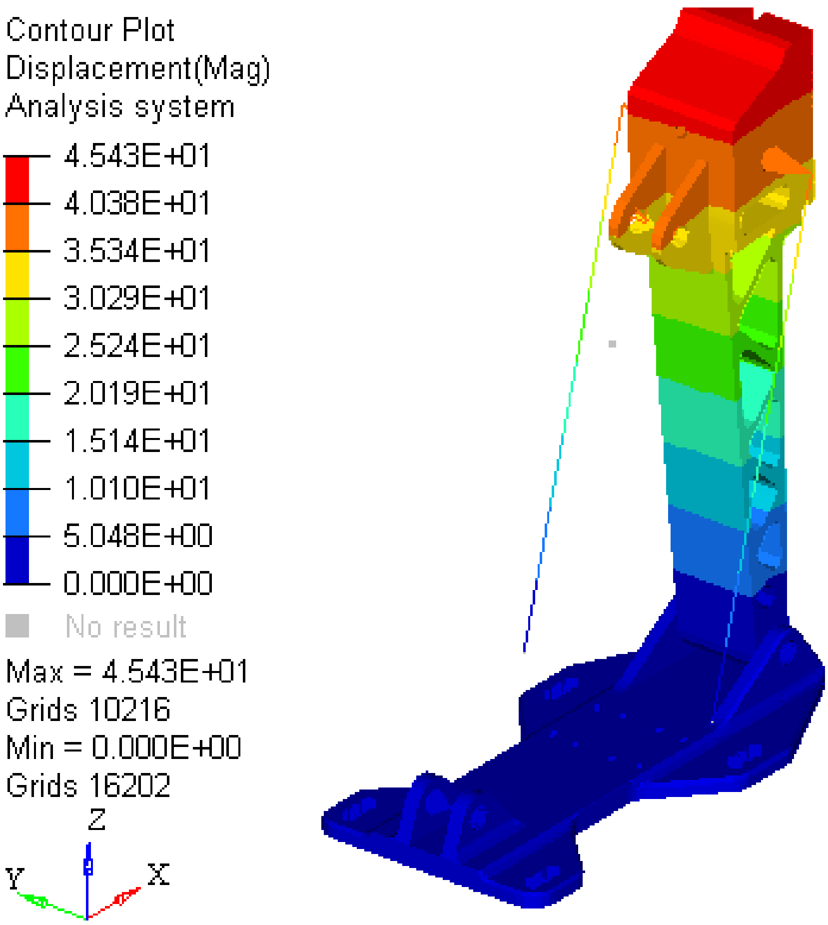


Figure 53 FEM – Displacement Base support (Pull force)

Stresses: Base support (pull)

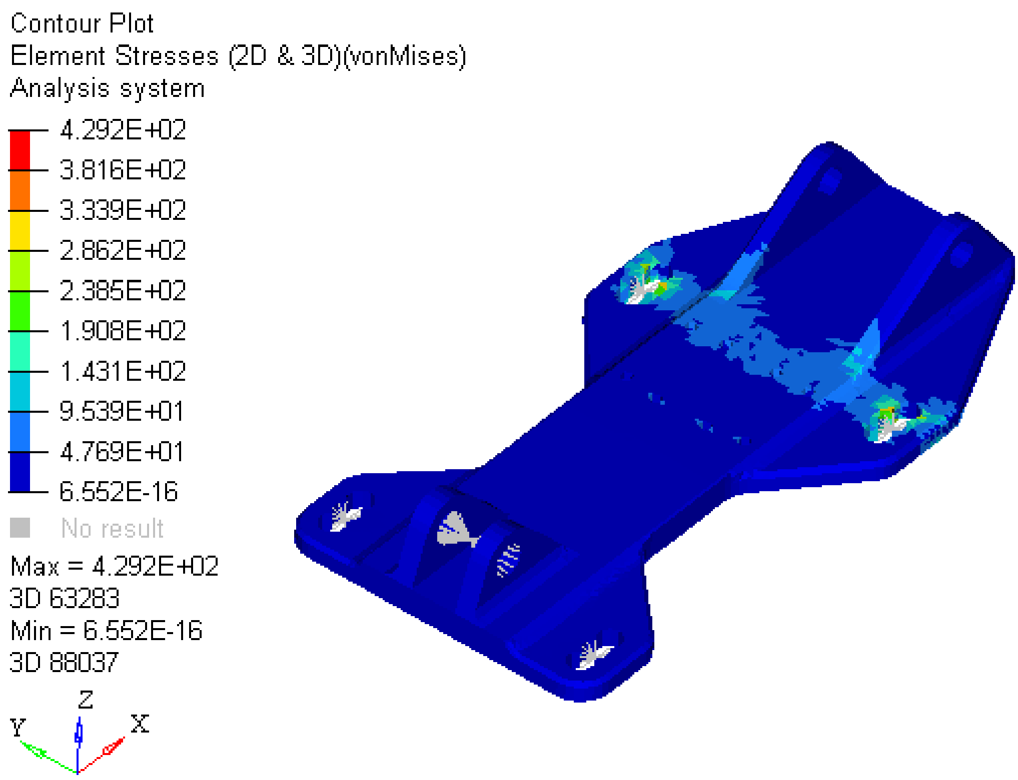


Figure 54 FEM – Stresses Base Support (Pull force) at pedal hinge

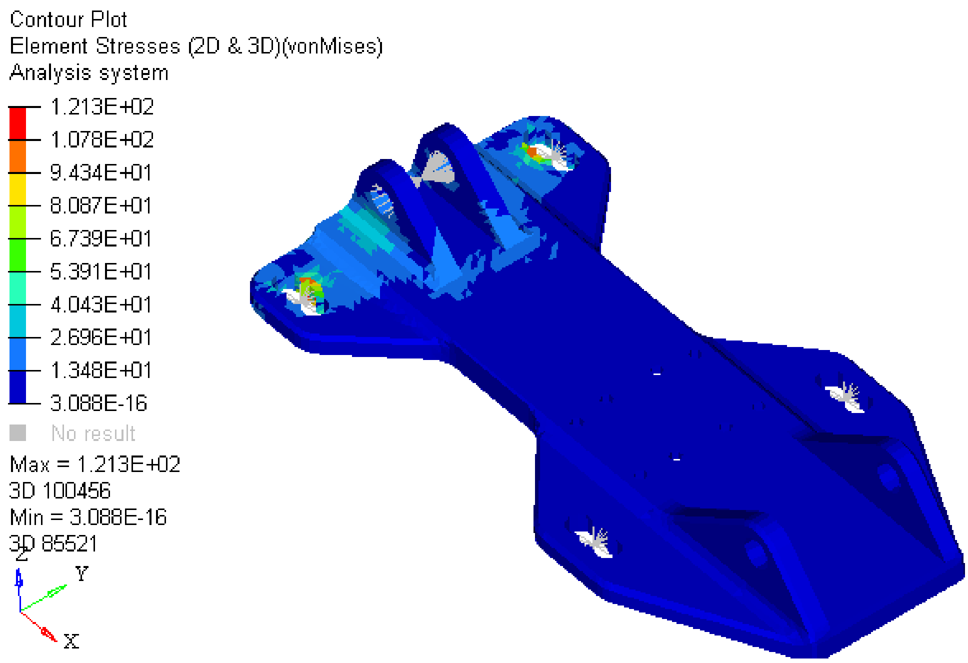


Figure 55 FEM – Stresses Base support (Pull force) at actuator mounting

Conclusion

From this thesis, a functional design solution for EBS has been developed and the feasibility analysis for its proper functioning has been performed. Most of all the important criteria of having a fail-safe EBS operation has also been developed by incorporating an algorithm which uses a redundant secondary system in the event of primary system failure.

As per FSG 2020 regulations, the EBS must be a passive system and it is optional to use the service brakes for EBS. Since there exists a hydraulic brake system in the car already and will not be used for primary braking in the developed Driverless car, we have incorporated it as part of our EBS. This has made the overall system less complex and with lesser components considering the space inside the cockpit is very limited.

The pedal box used is from last year's FS car and it was chosen based on its simple design and smaller X-direction encumbrance suiting the likes of the cockpit space availability in adding the EBS components. As a way forward, the design prototype will be fabricated and installed in the car for testing and evaluation of the EBS performance. The results obtained will be analysed for compliance to the requirements and improvised accordingly. Further topological optimisations can be performed on the pedal box to make it simpler and more compatible for EBS operation as well as in minimising the overall weight of the system.

The car which shall be developed for the next FSD competition can be designed based on the constraints faced in the current development of EBS for SC19. This gives an opportunity to develop a better and less complex EBS solution for the driverless car.

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