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Master Thesis

Design and implementation of an Emergency Brake System (EBS) for a Driverless Vehicle



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Abstract

Autonomous driving technology is gaining a lot of attention in the automotive industry. Since most of the accidents are caused by human related errors (94%), the widespread diffusion of autonomous vehicles is expected to reduce errors related to driver distraction, leading to less accidents and hence to an improved road safety. Nowadays, almost every vehicle is equipped with Advanced Driver Assistance Systems (ADAS) as adaptive cruise control, lane keeping, collision avoidance, parking assistance, and also includes Electronic Stability Program (ESP), Anti-locking Braking System (ABS) which are in charge of providing emergency assistance and reduce the workload of the driver.

In this context, the Formula Student Driverless competition provides a very stimulating and instructive environment, that helps students to gain knowledge by designing and implementing new technologies on real vehicles.

This thesis presents the design and implementation of an autonomous Emergency Brake System designed for a four-wheel drive electric vehicle to compete in the Formula Student Driverless competition. The EBS allows to bring the vehicle to a complete and safe stop within a space of 10 meters and with an actuation time less than 200ms. It can be actuated wirelessly using a Remote Emergency System (RES) or can be triggered whenever there is a fault in the vehicle by the opening of the Shut Down Circuit (SDC) from the Autonomous System.

The EBS is designed, referring to the competition rules and starting from the characteristics of the already existing vehicle braking system, as a hydropneumatic system able to autonomously pressurize the brake lines when triggered.

The system exploits a high-pressure canister filled with air in combination with a hydro-pneumatic intensifier, in order to obtain in the hydraulic lines enough pressure to actuate the brakes. Different solutions were evaluated basing on dynamic performances, overall weight and dimensions, and packaging.

SOLIDWORKS is the main CAD software used for components design, while analysis using HYPERWORKS are carried out on the parts to simulate the forces and stresses expected to be act on the system, verifying its structural compliance.

Later, EBS integration with other sub-systems part of the Autonomous System (Remote Emergency System, Low Voltage system, Shut Down Circuit) is considered to guarantee a correct and rule compliant system triggering.

Finally, a bench-test is set-up in order to test and validate the performances of the system with the actual components and to study its compatibility in prevision of a future in-vehicle mounting.

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Acronyms

ADAS - Advanced Driver Assistance Systems

ADS - Automated Driving System

AIR - Accumulator Isolation Relay

AS - Autonomous System

ASSI - Autonomous System State Indicator

ASMS - Autonomous System Master Switch

CAV - Connected and Autonomous Vehicles,

CV - Combustion engine Vehicle

DV - Driverless Vehicle

EBS - Emergency Brake System

ESP - Electronic stability programme

EV - Electric Vehicle

FBC - Front Braking Circuit

FS - Formula Student

HP - High Pressure

LVMS - Low voltage Master Switch

LVS - Low Voltage System

MC - Master Cylinder

RBC - Rear Braking Circuit

RES - Remote Emergency System

SA - Steering Actuator

SAE - Society of Automotive Engineers

SB - Service Brake

SDC - Shut Down Circuit

TS - Tractive System

TSMS - Tractive System Master Switch

Chapter 1

Introduction

Aim

The aim of this thesis project is to realise an Emergency Brake System (EBS) to be mounted in an electric driverless vehicle. The system is designed as part of the autonomous system of the SC19, a former electric class Formula Student Vehicle which has to be converted into a driverless vehicle by installing all the necessary systems and equipment to take part to the Formula Student Driverless competition. Therefore, the main design objective is to develop a fully functional system able to be compliant with the competition rules in terms of layout and performances, but at the same time capable to fit in the limited space available in the SC19 monocoque.

Thesis Outline

The entire thesis work is divided into four main chapters: in Chapter 1, an overview about the present situation of the automotive industry is given, underlining between the main emerging trends the importance and the increasing weight that autonomous vehicles are gaining in the current, and, in perspective, in the future society. Then, an introduction about the Formula Student competition (and in particular about the driverless competition in which the vehicle will have to race) is provided.

In Chapter 2, after an introduction on braking theories to explain and present the main used formulas, the braking system of the vehicle is presented giving also an overview about the functioning of its main components, to permit a more clear understanding of the system on which the EBS will have to be integrated.

The core of the work is presented in Chapter 3. Here, the complete EBS design is reported, from the evaluation of performances required to the system to the description of the components chosen for the final design, passing through main rules explanation, description and comparison of the different studied solutions, and

component functionality evaluation, underlining the main constructive choices and components selection criteria.

Finally, in Chapter 4, after an introduction about the different vehicle and EBS states which are necessary for the driverless competition, the EBS circuit integration with the vehicle systems in charge to control it, such as RES, SDC and LV, is presented. The mounting phase of the different components is also reported, as well as the complete bench testing of the system, presenting the test parameters which were used to verify the compliance with the design specifications.

1.1 Background

Automobiles made their appearance only on very recent times: since the first prototypes started to loudly and slowly move on the ground not even 150 years are passed, despite the multiple millenniums of humankind history. However, the evolution that this object has faced in this relatively short time is huge: today's cars are so different in every aspect from the ones of the beginning that placing an early model next to a new one, apart than for the 4 wheels it would be difficult to say that they belong to the same category.

Not only the automobile itself, but also the way it is perceived by people has dramatically changed during years: cars started their journey as a luxury product, so expensive and unreliable that only few people could afford to buy and maintain one. Then, slowly, due to technological progress in design and production methodologies (and, in some cases, due to political pressures) automobiles slowly became more affordable and popular, and after World War 2 (at least in Europe) the number of cars on the road started to increase exponentially.

Today, as vehicles are largely diffused, the way they are perceived is changing again; they are no longer seen as an object of desire, something that can substantially improve people's lives: people got used to own at least one car, and are starting to consider automobiles as a mere transport service. Especially in cities, that are getting bigger and bigger, vehicles are becoming a bargain because of taxes, increasing cost of fuel, parking, and traffic issues. In addition, there is an increasing attention and sensibility towards environmental issues; these are the main reasons why new and more eco-friendly service models for transport, as car sharing, Uber, or networked autonomous vehicles, are taking over the concept of private cars.

In 2010 the number of circulating vehicles hit 1 billion for the first time, and according to the World Health Organization, there will be 2 billion vehicles on the roads by 2030 [1]. This exponential increase will not only require new infrastructures but will also increase the probability of accidents and fatalities, the majority of which (around 94%) is caused by human errors and first of all by driver distraction [2]. Hence, to improve safety, a technology where cars can autonomously operate without the constant

attention of the driver and are able to communicate with each other and with the surrounding environment will have a fundamental role; that's why Connected Autonomous Vehicles (CAVs) are so important. Today, the most common and widely used technology used on these kind of vehicles to reach the goal of complete autonomy and communication is a cellular or Wi-Fi-based system that receives information on traffic conditions, but vehicles are increasingly being equipped with LIDARs and radar equipment [3].

In terms of automation capabilities, a paper presenting a classification system of six levels (from 0 to 5) was published in 2014 (and then updated in 2018) by SAE International. The different levels are the following [4]:

- **Level 0** - No driving automation: Only warnings or momentaneous interventions are possible, but no vehicle control by the automation system (if any).
- **Level 1** - Driver assistance: The Driving Automation System can control the longitudinal or the lateral vehicle motion. This is the category in which Cruise Control and Parking Assistance systems are included.
- **Level 2** - Partial driving automation: The Driving Automation System controls both the lateral and longitudinal vehicle motion; can be disabled by driver request, but it is still necessary to maintain the hands on the steering wheel.
- **Level 3** - Conditional Driving Automation. The ADS (while engaged) performs the entire DDT (Dynamic Driving Tasks), while the driver can turn its attention away but must be prepared to intervene in some situations when called by the vehicle to do so, out of the operational design domain (ODD).
- **Level 4** - High Driving Automation. When the ADS is engaged, the driver becomes basically a passenger, but self-driving is supported only in limited circumstances or spatial areas, outside which the vehicle must be able to safely stop if the driver doesn't retake control.
- **Level 5** - Full Driving Automation. No human intervention is required at all in every condition and place. The steering wheel can be removed from the vehicle.

A fully autonomous car is therefore a complex distributed system that integrates various computation, communication, and storage domains with a focus on intelligence and capability of decision-making. The “nuts and bolts” modules of the autonomous car include object detection, perception, learning, path planning, and execution [5].

Autonomous cars are not the future, but the present; to date, remarkable results have been achieved, and prototypes of level 5 autonomous cars have already travelled millions of miles in test drives. Anyway, since CAVs are a new thing, there isn't a complete legislation about them: up to date, each country has adopted different guidelines [3], but there are still some common open points, such as which additional safety regulations are needed both for testing and for the future widespread on-road diffusion of this kind of vehicles. Then, it is true that communication between vehicle and surrounding environment is crucial, but this means also that travel time, location and activity of users are constantly tracked, leading to privacy issues. Also, in case of accident of a self-driving vehicle, it is still unclear to whom the responsibility of the crash should be attributed (driver? Car manufacturer?). So, there is still some time before the complete adoption of this kind of vehicles, but the direction to take is clear, and some models on the market are already equipped with systems able to guarantee the highest levels of automation.

To conclude this part, it is possible to state that the adoption of CAVs constitutes a big technological transition, comparable to the one represented by the introduction of the internal combustion engine; the entire mobility as we know it today will change radically, leading (again) to new and huge societal changes.

1.2 Formula Student and Formula Student Driverless

The Formula Student is an international competition based on rules and guidelines derived from Formula SAE, a student design competition organized by SAE International (the acronym stands for Society of Automotive Engineers) and involving the engineering departments of different universities coming from multiple countries [6]. Started in 1980 at the University of Texas, Formula SAE currently counts more than 600 teams racing with their self-constructed cars in competitions taking place all over the world. First competition in Europe took place in 1998, while in 2010, following the increasing interest for electric vehicles, an electric class of competitions was started. The third and newest category to be opened is the Formula Student Driverless, for autonomous vehicles, which was raced for the first time in Germany in 2017 and subsequently introduced, starting from the following year, also by Formula SAE Italy, Formula Student UK and Formula East.



Figure 1.1 - Picture showing SC19, the Formula Student vehicle of Politecnico di Torino for which the actuator object of this thesis is designed, during racetrack tests.

The concept behind the competition can be easily explained: it is like if each student team would be contacted by a fictional company and asked to design and develop a small formula-style race car. So, basing on a set of rules given by SAE itself, the team designs, builds and tests a prototype that is subsequently evaluated under different aspects as a potential batch-production item: that means that not only performances,

but also construction methods (for example use of readily available standard components easy to replace) and financial planning are rated contributing to the final score of the team.

Therefore, the winner is not necessarily the team with the fastest car, but the one that at the end of the events has obtained the highest overall score from the different parts; after a series of technical inspections, aimed to check the different vehicle systems for safety and compliance with the rules, there are two main kinds of events, slightly different for the three main categories of the competition (Internal Combustion Engine Vehicle CV, Electric Vehicle EV or Driverless Vehicle DV):

- Static events
- Dynamic events

Considering the Driverless Vehicle (DV) category, that is the one in which the vehicle mounting the EBS system presented in this thesis will compete, the different sub-events and the maximum score assigned for each are reported in the following figure:

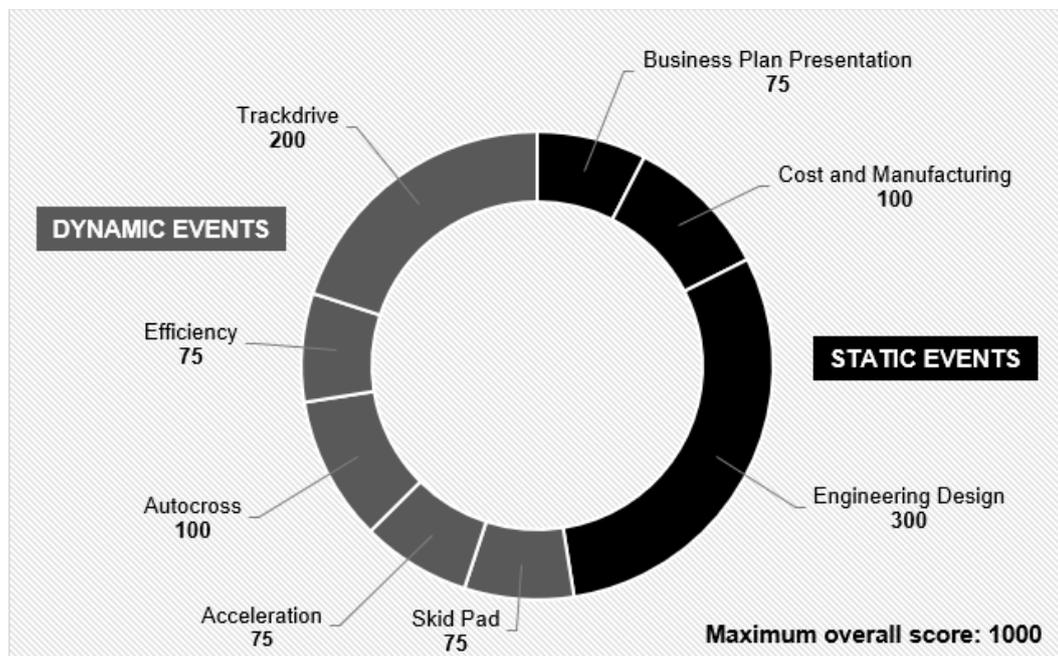


Figure 1.2 - Scheme of different sub-events and relative points

Teamwork experience, project and time management along with design, manufacturing and also business planning activities are elements giving to the participants of the competition a complete experience on the world of automotive industry, greatly improving the qualifications of young engineers.

Chapter 2

Braking system

In this chapter, an overview about braking systems is given. First, an introduction about the braking theory for ideal and actual conditions is given, explaining the main concepts and the related formulas. Then, since the EBS object of this thesis is to be designed and implemented in an already existing Formula Student vehicle, its braking system is presented, describing the main components and their basic working principles to give a clear view of the system with which the EBS will have to interface..

2.1 Braking theory

2.1.1 Braking in Ideal Conditions

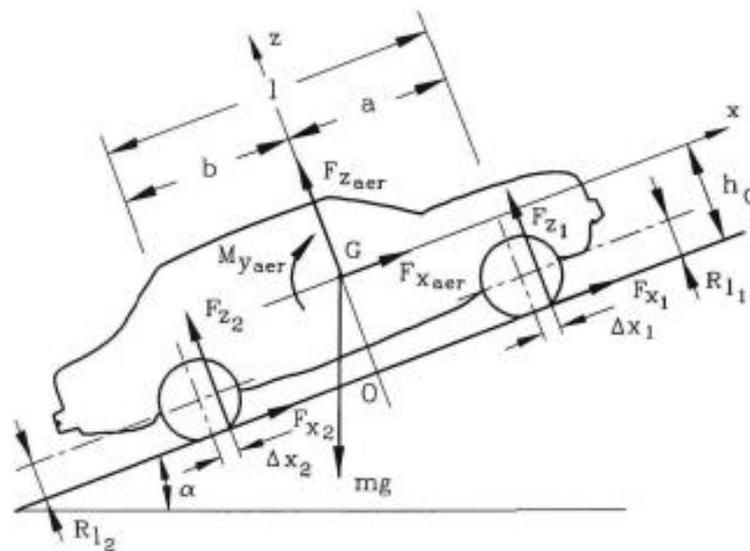


Figure 2.1 - Scheme of the main longitudinal forces and moments acting on a moving vehicle

As defined in [7], ideal braking can be defined as the condition in which all wheels brake with the same longitudinal force coefficient μ_x . The total braking force F_x can be therefore written as

$$F_x = \sum_{\forall i} \mu_x F_{z_i}$$

where the sum is extended to all wheels, and F_z is the vertical force acting on each wheel, that can be evaluated as

$$F_{z1} = mg \frac{(b - \Delta x_2) \cos(\alpha) - h_G \sin(\alpha) - K_1 V^2 - \frac{h_G}{g} \dot{V}}{l + \Delta x_1 - \Delta x_2}$$

$$F_{z2} = mg \frac{(a + \Delta x_1) \cos(\alpha) + h_G \sin(\alpha) - K_2 V^2 - \frac{h_G}{g} \dot{V}}{l + \Delta x_1 - \Delta x_2}$$

In the previous formulas, the Δx values (distance between the point of application of horizontal and vertical forces on the tyre and wheel centreline) are generally quite small (their difference, in particular, is almost zero) and they can be neglected. To make a simplified analysis, for two-axle vehicles with low aerodynamic vertical loading, the equations can be rewritten as:

$$F_{z1} = \frac{m}{l} \left[g b \cos(\alpha) - g h_G \sin(\alpha) - h_G \frac{dV}{dt} \right]$$

$$F_{z2} = \frac{m}{l} \left[g a \cos(\alpha) + g h_G \sin(\alpha) + h_G \frac{dV}{dt} \right]$$

Therefore, the longitudinal equation of motion of the vehicle represented in Figure 2.1, taking into account aerodynamic resistance, rolling resistance and tyre capability is

$$\frac{dV}{dt} = \frac{\sum_{\forall i} \mu_x F_{z_i} - \frac{1}{2} \rho V^2 S C_x - f \sum_{\forall i} F_{z_i} - m g \sin(\alpha)}{m}$$

The order of magnitude of aerodynamic drag and rolling resistance is generally much smaller than the one of braking force, and in addition rolling resistance can be considered more as causing a braking moment on the wheel than a braking force directly on the ground. Therefore, if considering a simplified braking study, these two components can be neglected. So, considering also a level road and no aerodynamic lift, the previous equation reduces to

$$\frac{dV}{dt} = \frac{\mu_x}{m} \left(\sum_{\forall i} F_{z_i} \right) = \mu_x g$$

and the maximum deceleration in ideal conditions can be obtained by inserting in it the maximum negative value of μ_x . If μ_x can be assumed to remain constant during braking, the motion of the vehicle occurs with constant acceleration, and the time and space to stop the vehicle from a given speed V are:

$$t_{stop} = \frac{V}{|\mu_x|g}$$

$$S_{stop} = \frac{V^2}{2|\mu_x|g}$$

Substituting 7 into 4 and 5 and keeping in mind that the values of μ_x are all equal in ideal braking, we obtain

$$F_{x_1} = \mu_x F_{z_1} = \mu_x \frac{mg}{l} [bcos(\alpha) - h_G \mu_x]$$

$$F_{x_2} = \mu_x F_{z_2} = \mu_x \frac{mg}{l} [acos(\alpha) + h_G \mu_x]$$

Using the previous equations, it can be easily obtained that

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

The equation above describes, in the plane $F_{x_1}F_{x_2}$, a parabola representing the locus of all the couples of values of F_{x_1} and F_{x_2} that lead to ideal braking conditions. Of the whole plot however, showed in the following figure, only a part is of our interest: the one with negative values of forces (so $-F_{x_1}$ and $-F_{x_2}$ positive, that means braking in forward motion) and with consistent values of μ_x (and therefore actually achievable braking forces).

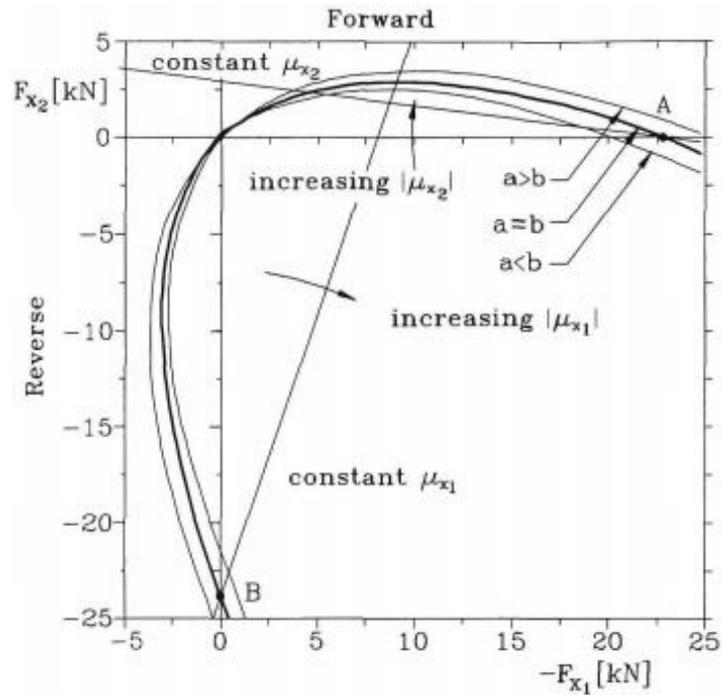


Figure 2.2 - Example of an "ideal braking" parabola plot. **NB** Forces are related to each axle and not to each wheel.

The braking moment instead, is equal to the braking force multiplied by the loaded radius of the wheel: if all the wheels have the same radius, the same plot is valid also if referred to braking torques. If, instead, the radii are different the plot is a bit distorted, but the overall shape remains essentially unchanged. The law linking F_{x1} to F_{x2} (and therefore M_{b1} to M_{b2}) to allow ideal braking, represented by the equation of the parabola reported above, depends on the mass and on the position of the centre of mass of the vehicle. So, for passenger vehicles generally only the lines for minimum and maximum load are plotted, assuming and that all intermediate conditions are included between them; for industrial vehicles instead, where the position of the centre of mass can vary to a larger extent, a higher set of loading conditions should be considered. To perform more precise computations, rolling resistance can be considered, and, more importantly, the torque for decelerating the rotating inertias should be added to consider, for example, the braking effect of the engine.

2.1.2 Braking in Actual Conditions

The ideal braking assumption is valid if the braking torque applied on each wheel is proportional to the forces F_z , if the radii of the wheels are all equal. This condition does not always occur, unless than a sophisticated control device is implemented trying to always allow ideal braking conditions.

In practice, the relationship between braking moments at front and rear wheels is different from the one following the ideal braking parabola reported above and is imposed by the parameters of the actual braking system of the vehicle. A ratio between the braking moments at the front and rear wheels can be defined as K_B

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

If all the wheels have the same radius, this value is equal to the ratio between braking forces (if the braking moment necessary to decelerate rotating parts is neglected).

The K_B value depends on the actual layout of the braking system: in the case of a hydraulic braking system, the braking torque is linked to the pressure in the hydraulic system with a relationship of the type

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

where ϵ_b , also referred to as efficiency of the brake, is the ratio between the braking torque and the force exerted on the braking elements (hence, it has the dimensions of a length), A is the area of the pistons, p is the pressure and Q_m is the restoring force due to the springs when they are present. The value of K_B is therefore

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

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

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

For disc brakes, that are the ones that will be considered in the following since they are mounted on the SC19 object of this thesis, ϵ_b can be considered almost constant and equal to the product of average brake radius, friction coefficient, and number of

braking elements acting on a single axle (as braking torques, as stated before, are referred to the whole axle). This means that if pressure acting on the front and rear wheels is the same, the K_B value is constant and depends only on geometrical parameters.

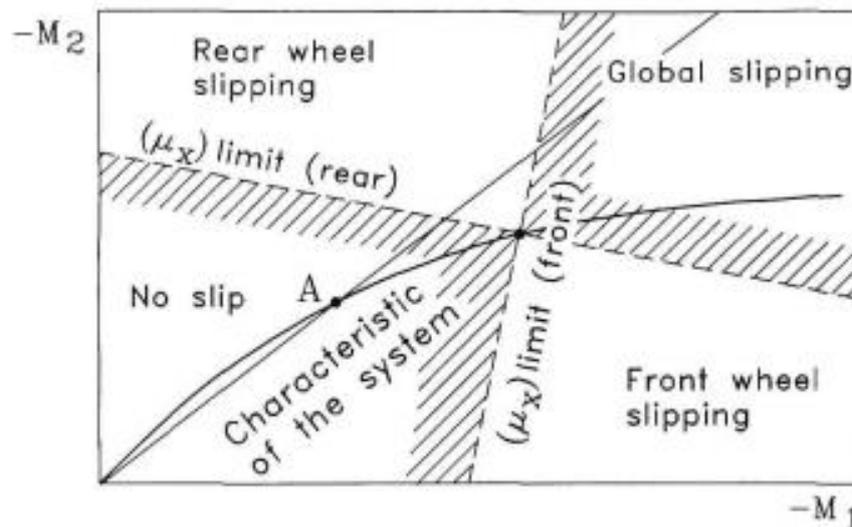


Figure 2.3 - Example of plotting of system characteristic line for a braking system with constant K_B , ideal braking curve and μ_x limit values for front and rear axles on the $M_{b1}M_{b2}$ plane. It can be useful to underline that in this case the μ_x limit values are high enough to obtain a working point beyond point A.

If K_B is constant, the characteristic line of the system on the plane $M_{b1}M_{b2}$ is a straight line passing through the origin. The Intersection of this characteristic line and the previously defined ideal braking curve defines the point in which the braking system is able to achieve the ideal braking condition.

On the left of this intersection point (named point A in the previous figure), i.e. for lower deceleration values, the rear wheels brake less than the required quantity and μ_{x2} is smaller than μ_{x1} . If the limit conditions are reached in this zone, as can happen for roads with poor traction, the front wheels lock first.

On the contrary, the working conditions beyond point A are characterized by the rear wheels braking more than required with μ_{x1} smaller than μ_{x2} . In this case when the limit conditions are reached, the rear wheels lock first, so the braking capacity of the front wheels is underexploited, as in the case reported in the Figure.

Considering vehicle handling, we would prefer to be in the first situation ($\mu_{x1} > \mu_{x2}$) as it would increase the stability of the vehicle; that's why the desired characteristics of a braking system should lie completely below the ideal braking line, while locking of rear wheels first would lead to directional instability, and should hence be avoided. In point A the ideal conditions are obtained: If the limit value of the longitudinal force coefficient occurs at that point, simultaneous locking of front and rear wheels occurs. The value of the ratio K_B for which this happens, at a given value of the longitudinal force coefficient μ_x^* can be easily computed as

$$K_B^* = \frac{b + h_G |\mu_x^*|}{a - h_G |\mu_x^*|}$$

2.2 Braking System Architecture of SC19

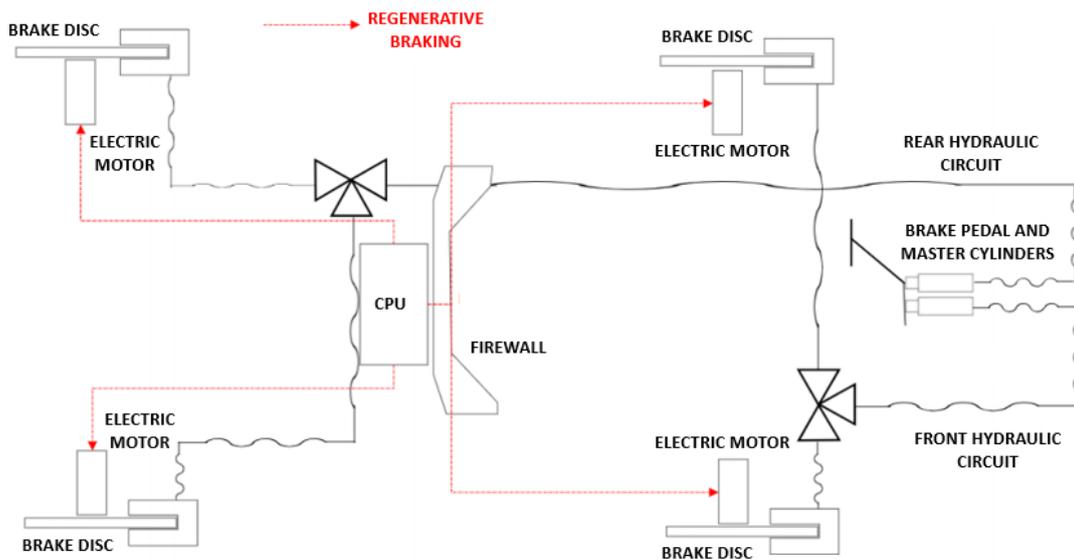


Figure 2.4 - Braking circuit layout of SC19

The braking system of SC19 (Figure 2.4) is a hydraulic braking system with regenerative braking capability. A regenerative braking system uses the electric motors of the vehicle, (in this case placed in the wheel hubs), to convert back the kinetic energy that would be wasted when the vehicle is braking (during deceleration or downhill running) into electrical energy which is generally stored in specific devices [8]. In the SC19, it is obtained with a strain gauge mounted on the rod end and with

springs, that have been chosen and preloaded so that the relation between applied force and braking torque is kept linear during the whole hydraulic and regenerative braking (Figure 2.5). The system is designed to completely exploit regenerative braking before starting the hydraulic phase, and hence have the maximum energy recovery. 90% of the brake pedal travel is exploited to have regenerative braking, while for the remaining last 10% of the travel braking is fully hydraulic.

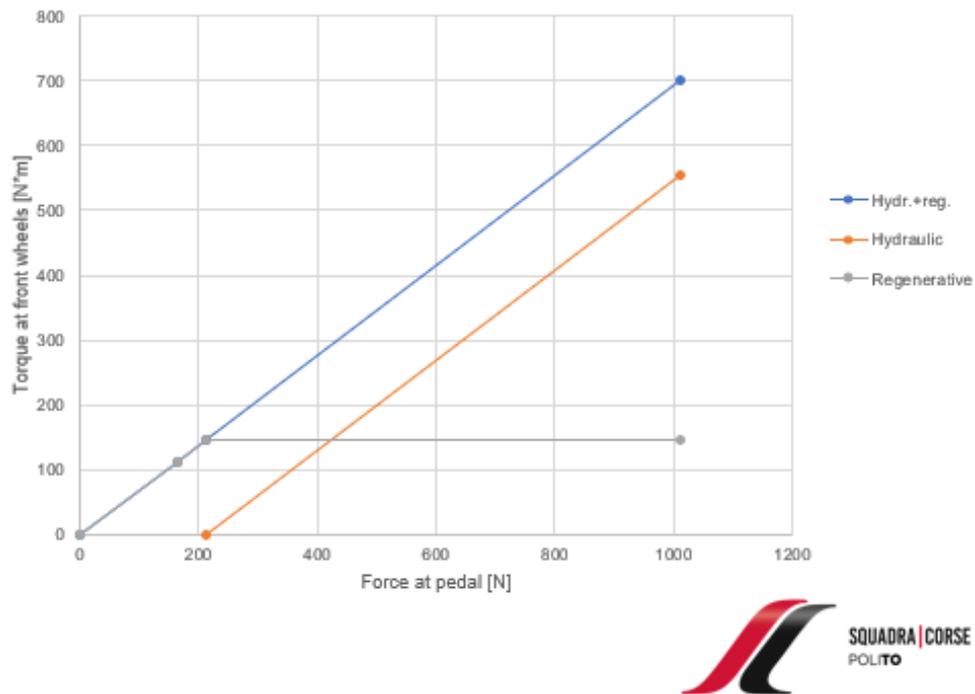


Figure 2.5 - Plot showing the linearity of the braking characteristic between regenerative and hydraulic braking. Image courtesy of Squadra Corse Polito.

2.2.1 SC19 Hydraulic Braking System

The EBS is not affecting the regenerative braking performances but will be implemented in the hydraulic circuit and will interact with the components already present in its layout. Therefore, the hydraulic braking system of SC19 is described in this section. Its main components, as reported in Figure 2.6, are:

- Brake pedal
- Master cylinders
- Fluid reservoirs
- Balance bar
- Hydraulic lines
- Brake callipers
- Brake rotors (brake discs)
- Analog pressure sensors, mounted on the hydraulic lines to provide a feedback if the pressure build-up is correct during brake actuation.

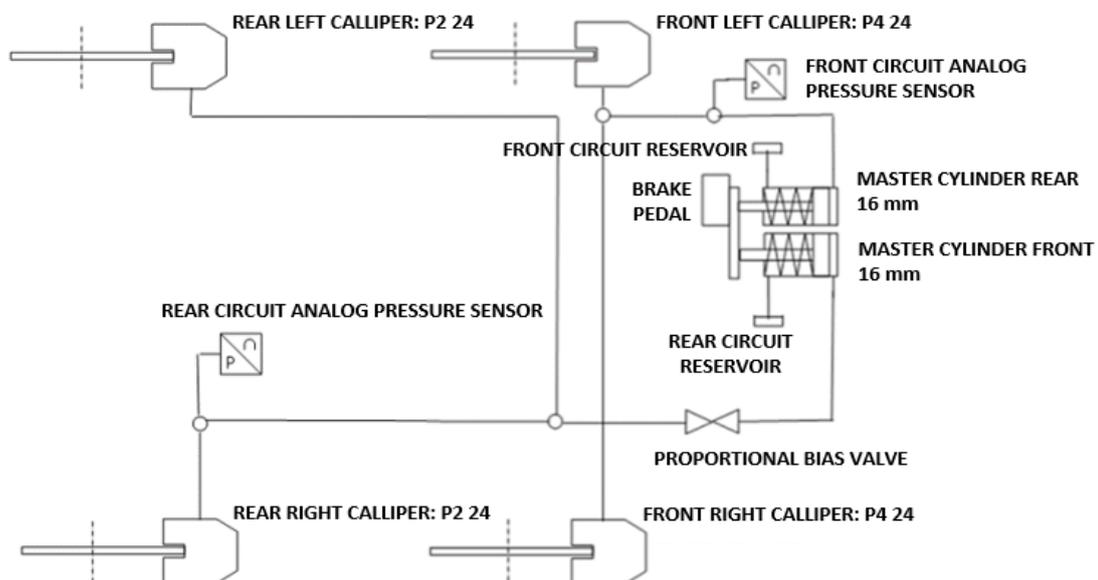


Figure 2.6 - Hydraulic circuit layout of SC19

Master cylinders

A master cylinder, or master brake cylinder, is a device able to convert the pressure acting on the brake pedal surface into hydraulic pressure, sending the pressurized brake fluid into the braking lines and hence to the brake callipers [9].

As the piston inside the master cylinder moves along the bore due to pedal pressure, its motion is transferred through the hydraulic fluid to the slave cylinders (i.e. the calliper pistons). Varying the ratio between surface areas of master cylinder and slave cylinder, the amount of force and displacement applied to each slave cylinder can be varied respect to the amount of force and displacement on the master cylinder.

In SC19 hydraulic circuit there are two master cylinders, one for pressurizing the front brake line (going to the front wheels callipers) and one for the rear one (going to rear wheels callipers). This configuration, also referred to as tandem master cylinders, comes to be very useful in the event of one braking circuit failure: in this case, the pressure build up on the other circuit would not be affected, and the vehicle would still have some braking capability.

The front and rear master cylinders are of the same dimensions (16mm diameter) to have an initial balance bar repartition at 50:50. An image showing the master cylinder mounted on the vehicle and its main dimensions is reported below.

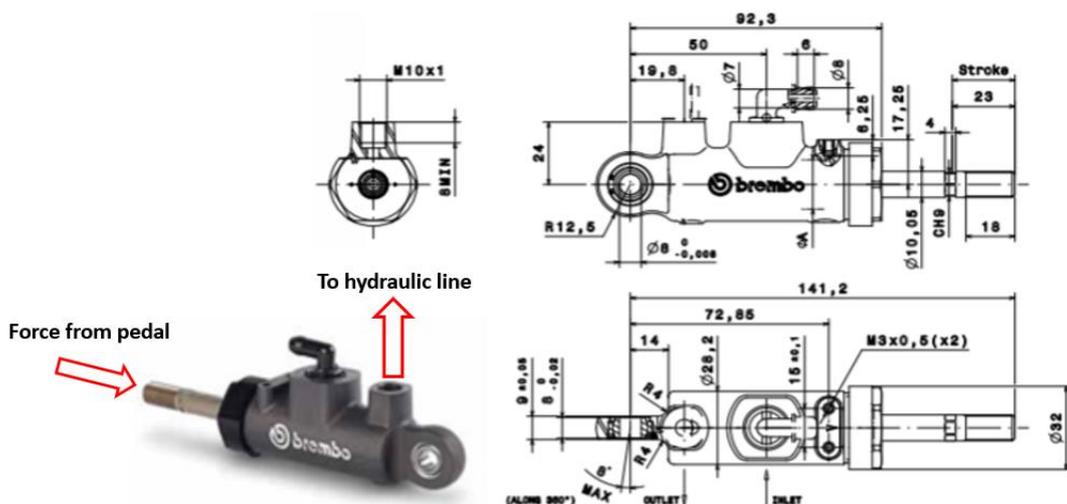


Figure 2.7 - Master cylinder of SC19 braking system: main dimensions and working principle.

Balance bar

A balance bar is a device designed to repartition between the two master cylinders (considering a dual master cylinder system) the force applied by the driver on the brake pedal. As evidenced in Figure 2.8, it is basically a rod which connects the two master cylinders, which are placed one on each end, and that has a pivot point that can be moved. Since the torque on one side of the pivot must balance the torque on the other side, the master cylinder that is closer to the pivot will receive a higher percentage of the total pedal force. [10] Therefore, the device can be very useful not only to equally split the pedal force between the master cylinders, but also to set a different repartition of the amount force which is provided to the different master cylinders.

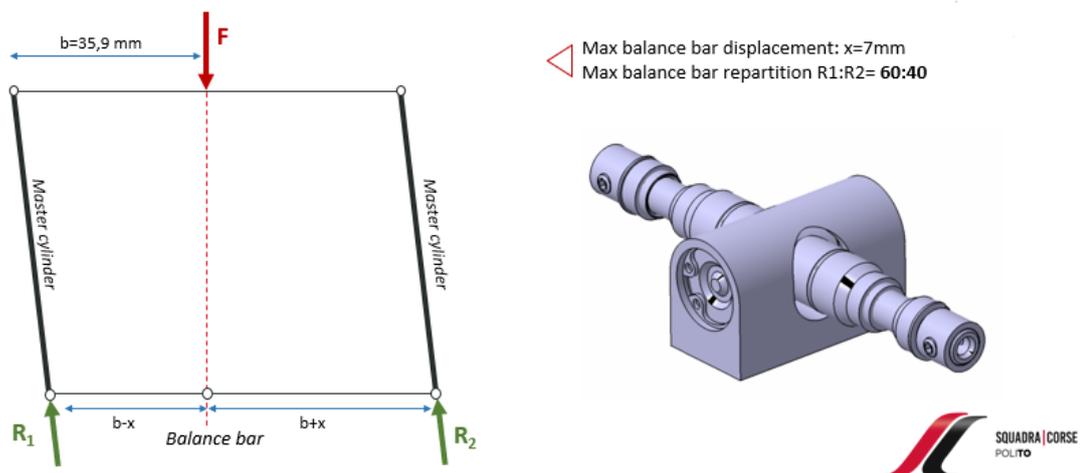


Figure 2.8 - SC19 balance bar geometry. Image courtesy of Squadra Corse Polito

In the SC19 case, the balance bar is used only to equally split the pedal force; it is set to have a 50:50 force repartition, (so, since the master cylinders have the same diameter, the force they generate is equal) but it can be calibrated up to a 60:40 front rear repartition in case of necessity.

Pedal assembly

The brake pedal of SC19 is reported in Figure 2.9. Its position has been set basing on ergonomic studies for the optimal position of the driver in the vehicle, and it has a

pedal ratio of 4.9. The pedal ratio is a parameter that indicates how much leverage is applied from the pedal to the master cylinders. It can be evaluated by dividing the vertical distance between force application point and pedal pivot point by the normal distance between pivot and master cylinder line of action (respectively indicated as **a** and **b**, with reference to the following Figure 2.9).

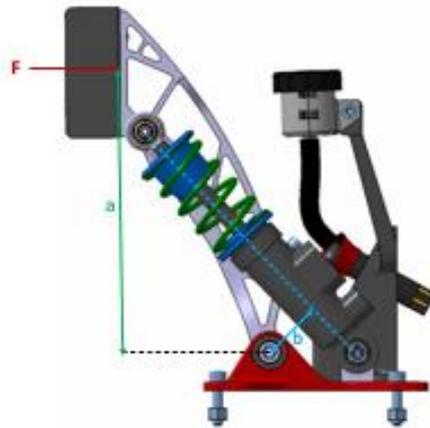


Figure 2.9 - Image showing brake pedal and the parameters used to evaluate the brake pedal ratio.

Generally, to guarantee to the driver a comfortable brake pedal operation, the pedal gain should be between 3 and 6. Higher the pedal gain, lower will be the force requested to the driver to operate the brake, but a higher pedal displacement will be also necessary. In the following Figure, the complete brake pedal subassembly is displaced, underlining its main components.

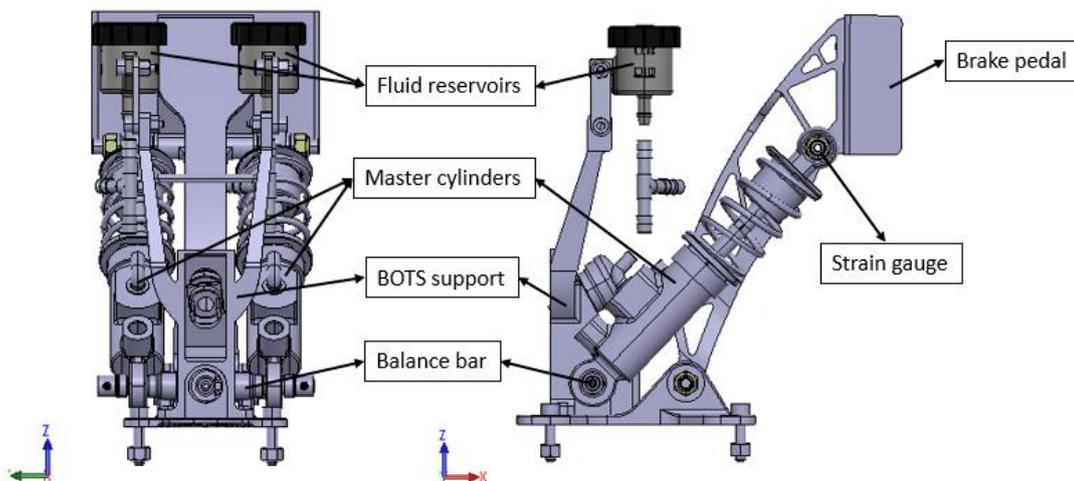


Figure 2.10 - Brake pedal subassembly, with main components evidenced.

Disc brakes

Both the front and rear wheels of SC19 are mounting disc brakes with floating disc and fixed calliper architecture. The main components of the brake subassembly are shown in the following Figure 2.11.

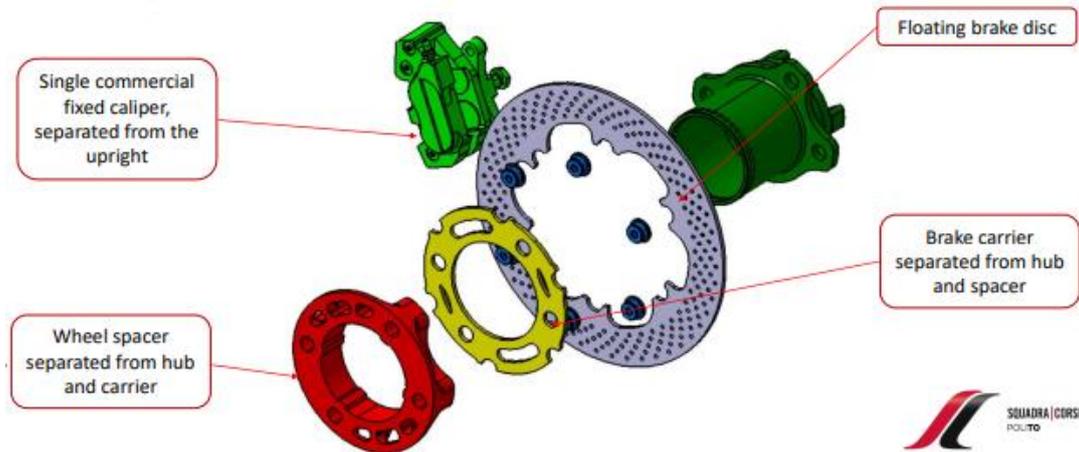


Figure 2.11 - Exploded view showing the main components of the SC19 disc brake subassembly.

Image courtesy of Squadra Corse Polito.

It is important to underline that even if the base architecture is the same, to obtain a different torque repartition between front and rear axle, different brake rotors and callipers are used between front and rear wheels, whose main characteristics are reported in the following table.

	Brake rotor radius	Calliper piston diameter	Pistons per calliper
Front brakes	94 mm	12 mm	4
Rear brakes	83 mm	12 mm	2

Table 2.1 - Table showing the main characteristics for front and rear brakes

In this way, considering as maximum braking condition a deceleration of 1.8g, the pressure between front and rear master cylinders is kept as close as possible (with the balance bar set at 50:50 repartition), while the brake torque repartition comes out to be around 65:35 front rear.

Hydraulic lines

Hydraulic lines are the pipes connecting the master cylinders with the front and rear brake callipers. In the hydraulic circuit of SC19 there are two main lines, one feeding the front and one feeding the rear brakes. As it is possible to notice in Figure 2.6, both these lines are divided into different parts, following the same scheme: each of them starts from its respective master cylinder and then goes up to a t-joint, where it separates into two sub-lines going on the left brake and on the right brake, respectively. The hose used for all these lines is the CARBOTECH 1/8", whose main characteristics are reported in the image below.

TUBO FRENO 1/8" CARBOTECH doppia trecciatura in fibra Carbotech 1/8" CARBOTECH BRAKE HOSE double Carbotech fibre braiding	
Diametro interno nominale Nominal internal Ø	3.2 mm
Min/Max diametro esterno Min/Max external Ø	6 / 6.5 mm
Pressione minima di scoppio Min. bursting pressure	1200 Bar
Pressione di prova Test pressure	600 Bar
Pressione di lavoro Working pressure	300 Bar
Peso specifico Weight	36.50 g/m
Raggio minimo di curvatura Min. radius of curvature	35 mm
Temperatura di utilizzo Working temperature	-40°C / +204°C (max.)
Anima Core	PTFE



Figure 2.12 - Image showing the main characteristics of the hose used for SC19 hydraulic braking lines

Chapter 3

EBS Design

In this chapter, the main design stages of the Emergency Brake System (EBS) for SC19 are presented, explaining the main design and constructive choices, as well as the main system evaluation criteria. After a short introduction on the competition rules, the performances which are required to the system and from which the design of the different layouts started are evaluated from the available vehicle data. Then, the main studied EBS solutions are presented, reporting for each of them the main functioning principle and the required calculations procedure. Each concept is critically evaluated in order to detect eventual faults, always considering the ease of integration with the already present components (described in the previous chapter) and the full rule compliance of the system. Finally, after the final solution is chosen, the component selection and in-vehicle positioning phases are reported.

3.1 Reference rules

In order to succeed at the competition technical inspections and to design a fully rule-compliant system it is necessary to start from the competition rules. Since the aim of the main project is to transform an already existing formula student vehicle that has raced in the Formula Student 2019 competition (in the Electric Vehicles category) and that was therefore already designed to be compliant with the general rules for EVs, the focus is now on rules for Driverless Vehicles (DVs) and more in specific, on rules for EBS. The main rules which will be mentioned in this chapter for EBS design and performance evaluation are presented in sections DV 3 and IN 6.3 of the Formula Student 2020 competition handbook [11], which are reported in the Appendix A of this document.

3.2 EBS required performances evaluation

The first consideration that has been done when approaching the design of the EBS system for SC19 was about how much force would be needed on the pedal to stop the vehicle under maximum deceleration conditions. To evaluate this force, it was necessary to start from the longitudinal forces $F_{xf_{wheel}}$ and $F_{xr_{wheel}}$ acting on each wheel on the front and on the rear axle, respectively. As previously reported in 2.3.1, where the hydraulic braking system of SC19 is presented, the vehicle was designed to have a braking torque repartition of 65:35 front rear. Therefore, it can be written that

$$\frac{M_{b_{front}}}{M_{b_{front}} + M_{b_{rear}}} \approx \frac{F_{x_{front}}}{F_{x_{front}} + F_{x_{rear}}} = 0.65$$

where $M_{b_{front}}$ and $M_{b_{rear}}$ are the braking torques for the whole axle, while $F_{x_{front}}$ and $F_{x_{rear}}$ (commonly referred to, in literature, also as F_{x1} and F_{x2}) are the braking forces referred to the whole axle. Considering the vehicle longitudinal dynamics equation and neglecting the aerodynamic force, it is also true that:

$$F_{x_{front}} + F_{x_{rear}} = ma_x = m\mu_x g$$

Starting from the previous two equations the forces for the whole axles can be computed, and from them, assuming that the vehicle is braking in a straight line and therefore no lateral weight transfer occurs (in compliance with rule DV 3.3.3), the forces acting on the single wheels are obtained as:

$$\begin{cases} F_{xf_{wheel}} = \frac{1}{2}F_{x_{front}} = \frac{1}{2}0.65 m\mu_x g & (front) \\ F_{xr_{wheel}} = \frac{1}{2}F_{x_{rear}} = \frac{1}{2}0.35 m\mu_x g & (rear) \end{cases}$$

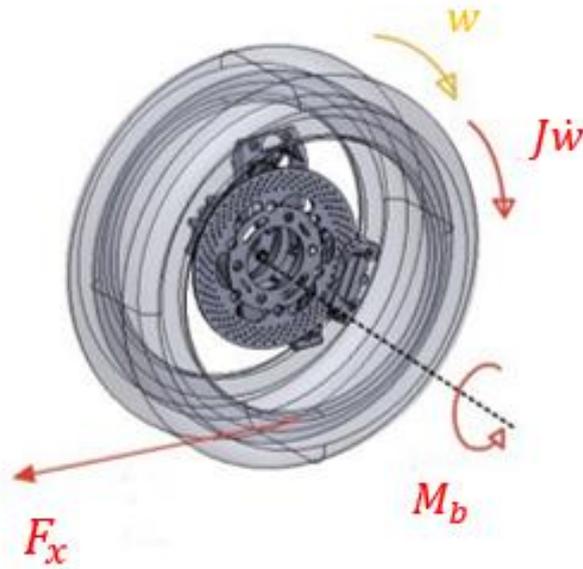


Figure 3.1 - Scheme of the main forces considered when writing the equilibrium equation of the wheel around its geometrical centre

Having these forces, writing an equilibrium equation on the decelerating wheel about its geometrical centre, it is possible to evaluate the braking moment for the single wheel $M_{bf_{wheel}}$ and $M_{br_{wheel}}$. With reference to Figure 3.1, where J_w is the wheel inertia (considered the same for all wheels), and \dot{w} is the wheel angular acceleration, that can be written as the ratio between a_x (again, equal to $\mu_x g$ for maximum deceleration conditions) and r_c (wheel loaded radius), it can be written that:

$$\begin{cases} M_{bf_{wheel}} = F_{x1_{wheel}} r_c + J_w \frac{a_x}{r_c} = F_{N1_{wheel}} \mu_{pad} r_{d_{front}} \\ M_{br_{wheel}} = F_{x2_{wheel}} r_c + J_w \frac{a_x}{r_c} = F_{N2_{wheel}} \mu_{pad} r_{d_{rear}} \end{cases}$$

Here, in the last equivalence, the relation between wheel braking torque and force acting on the wheel brake disc has been reported also: μ_{pad} is the friction coefficient of the disc, while r_d is the disc radius (which, as reported above, is different between front and rear brakes). In this way, it is possible to evaluate the normal force F_N required on each disc to generate the braking moment necessary to stop the vehicle guaranteeing maximum deceleration performances. Remembering that the number of pistons per calliper (and therefore, the calliper pushing area) is different between

front and rear brakes (as reported in 2.2.1), the pressure required on the brake calliper piston can be easily obtained from the normal force F_N as:

$$\begin{cases} p_{front} = \frac{F_{N_1}}{A_{calliper,front}} \\ p_{rear} = \frac{F_{N_2}}{A_{calliper,rear}} \end{cases}$$

Finally, the required force at master cylinders can be evaluated, and from that, the required force at the pedal. The formulas used for these calculations are reported below:

$$F_{MC} = A_{MC}(p_{front} + p_{rear})$$

$$F_{pedal} = F_S + \frac{F_{MC}}{pedal\ gain}$$

where F_S is the spring preload force, necessary to keep the pedal not moving in the first braking phase, when only regenerative braking phenomenon is exploited (first portion of Figure 2.5). In the following Tables 3.1 and 3.2 the main data used for calculations and the obtained numerical results are reported.

SC19 DATA		
Vehicle mass	m	190 kg
Hydraulic torque repartition	$[-]$	63/35
Wheel loaded radius	r_c	237.16 mm
Tyre friction coefficient	μ_x	1.8
Wheel inertia moment	J_w	0.27 kg m ²
Front disc brakes radius	$r_{d_{front}}$	94 mm
Rear disc brakes radius	$r_{d_{rear}}$	83 mm
Brake pad friction coefficient	μ_{pad}	0.4
Front calliper pistons area	$A_{calliper\ front}$	1810 mm ²
Rear calliper pistons area	$A_{calliper\ rear}$	905 mm ²
Spring preload	F_S	280 N
Master cylinder diameter	d_{MC}	16 mm
Pedal gain	$[-]$	4.9

Table 3.1 - Table showing the main data of SC19 used for calculations.

CALCULATION RESULTS		
Braking force at each front wheel	$F_{xf_{wheel}}$	1090.38 N
Braking force at each rear wheel	$F_{xr_{wheel}}$	587.13 N
Braking torque at each front wheel	$M_{bf_{wheel}}$	278.70 Nm
Braking moment at each rear wheel	$M_{br_{wheel}}$	159.35 Nm
Normal force acting on front callipers	F_{N_1}	7412.18 N
Normal force acting on rear callipers	F_{N_2}	4799.59 N
Hydraulic pressure at front brake line	p_{front}	40.97 bar
Hydraulic pressure at rear brake line	p_{rear}	53.03 bar
Force required at master cylinders	F_{MC}	1890.15 N
Force required at brake pedal	F_{pedal}	665.74 N

Table 3.2 - Table showing the numerical results obtained using the previously reported formulas

These are the parameters that the EBS should provide to the hydraulic system to guarantee braking with a deceleration of 1.8g (17.658 m/s², much higher than the minimum requirement of 8 m/s² requested by DV 3.3.2). Under this condition, starting from an initial velocity of 40 km/h (the minimum required for the brake test, as stated in IN 6.3.3), the vehicle should be able to arrive to a full stop within 3.50 meters, much less than the 10 meters regulation limit.

3.3 EBS functioning concept

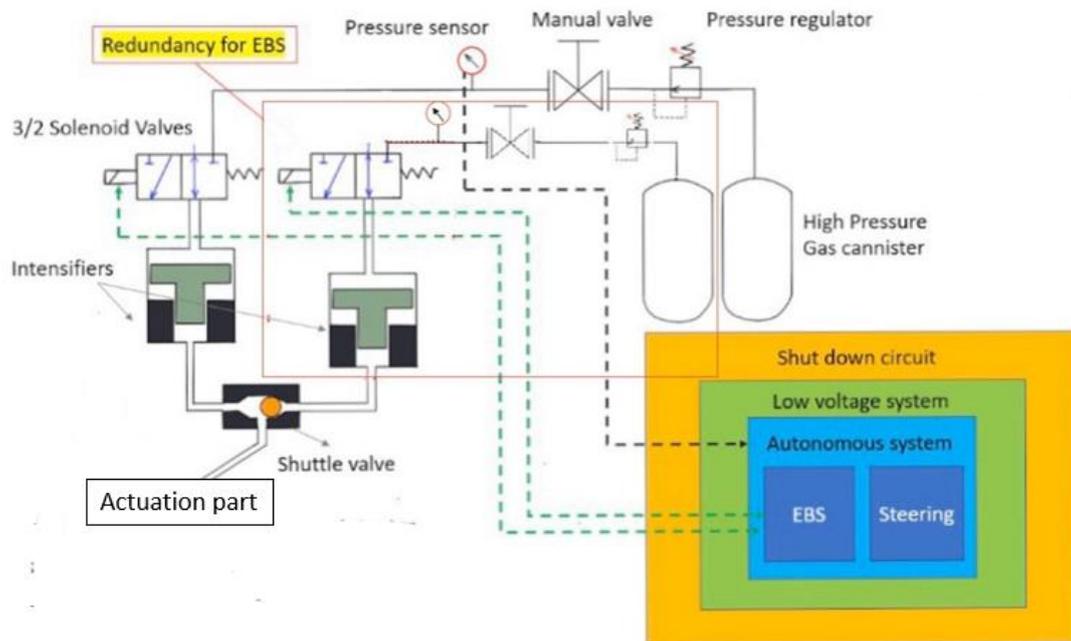


Figure 3.2 - Initial EBS concept layout.

In the following, an overview about the functioning concept of the EBS is given. As a matter of fact, even if the actuation part (and therefore the downstream end components of the circuit) will change between the different concepts presented in this chapter, the basic working principle of the upstream part of the system, the type of components and their role in the circuit will basically remain the same between all the proposed solutions. The main components of the system presented in Figure 3.2 are:

- High Pressure gas canisters
- Pressure regulators
- Manual valves
- Pressure sensors
- 3/2 normally open solenoid valves
- Intensifiers
- OR valve

The main design concept is to have a system which is capable of providing to the components in charge of brake actuation a pressure level sufficient to actuate the brakes guaranteeing the desired performances. The required hydraulic pressure is generated from a hydro-pneumatic pressure multiplier. This device provides pressurized oil at its output port, while it is fed with pressurized air which is coming from a high-pressure canister placed at the upstream end of the circuit. In order to be able to perform many actuations without having to refill the canister, the pressure at which it is charged should to be much higher than the one required for operation: hence, a pressure regulator is needed just on top of the canister, allowing to set the right value of air pressure at the beginning of the pneumatic line.

Between the pressure regulator and the intensifier, other components are present; first, a device able to efficiently trigger the system when necessary is needed. A suitable component for this task can be a 3/2 solenoid valve (3 ports, 2 different states, as reported in Figure 3.3). The three ports are connected to the high-pressure canister, to the pressure multiplier and to discharge, respectively. When current is flowing through the valve, the line going from hp canister to pressure multiplier is disconnected and the intensifier is communicating with the discharge port (Figure 3.3b), which is put at ambient pressure [12]. When instead current stops flowing through the valve, an electromechanical switch inside the valve puts in communication the pneumatic line and the pressure multiplier (Figure 3.3a), triggering the EBS.

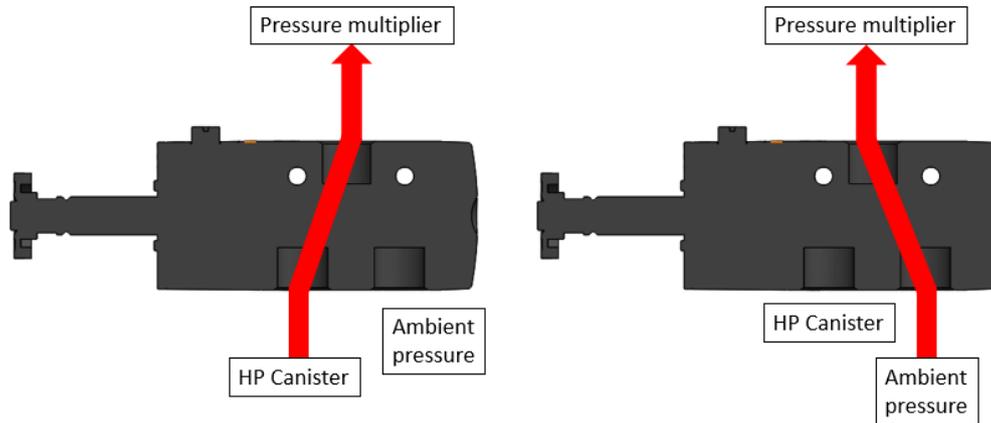


Figure 3.3 - Figure showing the 2 states of the 3/2 normally open Solenoid valves. On the right, the functioning of the valve when connected to the voltage source is shown (a), while on the left the operation of the valve when disconnected from alimentation (triggering EBS) is reported (b).

Then, a manual valve is placed between the canister's pressure regulator and the electrovalve, in order to be able to switch between the "EBS unavailable" state, in which the system cannot be activated, and the "EBS armed" state, in which the first part of the pneumatic line (up to the electrovalve) is pressurized and the EBS can be triggered by the 3/2 solenoid valve if necessary (more details about EBS and vehicle states will be given in Chapter 4). With reference to Figure 3.4, showing the different positions of the valve and the related active connections, three states are possible:

- OFF state, in which the canister pressure is discharged into ambient. This state should be actually avoided since it would lead to the canister pressure to be completely discharged into ambient.
- Intermediate state, in which both the output connections are closed. This state is necessary in manual driving mode, since the EBS is disarmed.
- ON state, in which the canister is connected to the pneumatic line going towards the solenoid valve. It is the only state in which the EBS can be actuated.

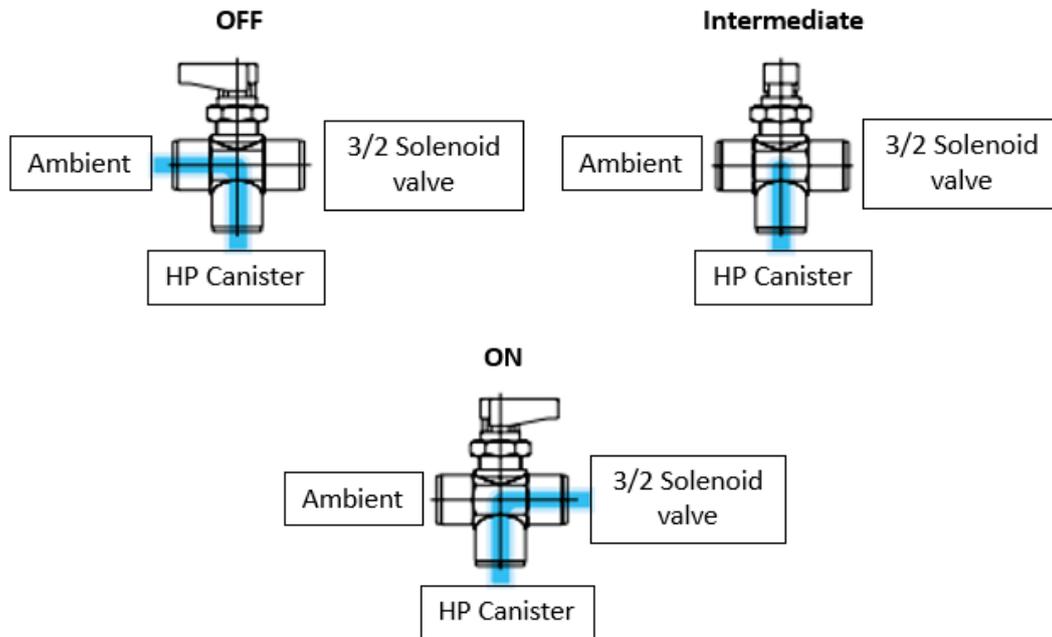


Figure 3.4 - Figure showing the different states of the manual valves each one with the related open and closed connections

As stated by rule DV 3.2.1, a fully redundant system, that must remain completely functional in case of a single failure mode, is required; that's why all the components previously described were actually doubled, leading to have a specular secondary circuit in parallel to the main one (underlined, in Figure 3.2, as "Redundancy for EBS") and equally capable of generating the required force to pull down the pedal, activating the brakes in case the main circuit has a failure.

The device in charge of establishing which one of the two circuits is connected to the component in charge of brake actuation is a shuttle valve (also called OR valve); it has a moving object (mostly, a sphere) inside it, and basing on the prevailing pressure that receives at the two input ports, will connect that port (and, therefore, one of the two twin-circuits) to the actuating device. The basic functioning of the component is reported in Figure 3.5. With reference to this figure, in normal operation the correct pressure build-up into the main circuit (indicated as Line 1) would ensure that this is the line connected with the actuator. Assuming instead that the main circuit would have a failure, it would not be able to build-up the expected pressure, so the pressure level at the first input of the OR valve will be lower than the one at the second input

port (connected to the secondary line, indicated as Line 2): in this case the sphere will move putting the latter in communication with the actuator, still ensuring a correct EBS actuation.

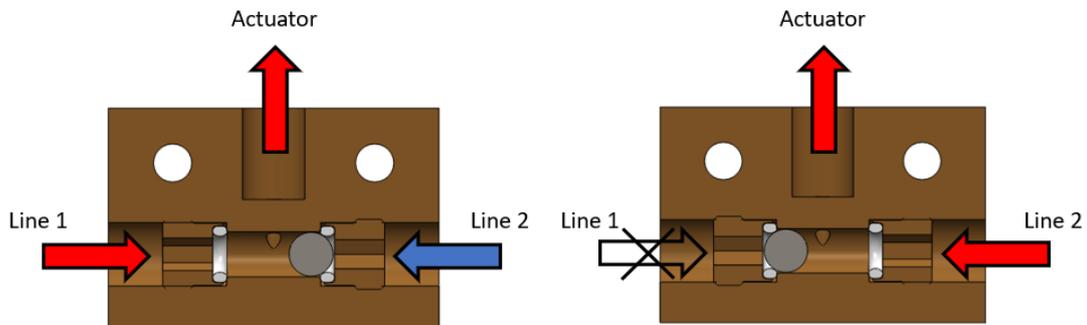


Figure 3.5 - Figure showing the functioning principle of the manual valve in normal operation and in case of main line failure.

3.4 Main evaluated solutions

In this paragraph, the main design solutions for the EBS are described. First, the initial concept, using a single hydraulic actuator for the actuation part is presented. Then, a second layout exploiting a two-actuators solution is described and finally, a third solution is presented, deploying a direct actuation on master cylinders, without the need of any actuator at all.

It is important to remark that all the concepts presented in this section are only the major milestones of a constant evolution of the system, as the result of a continuous critical review process that was carried on through the entire design stage in terms of rule compliance, system performances, components structural properties, overall weight and dimensions, and ease of access and mounting.

It must be underlined that, for the solutions requiring an actuator (first two concepts), the mounting in the vehicle of the brake pedal of SC18 (another Formula Student vehicle) was considered, due to packaging constraints. The SC18 pedal has a lower pedal gain (equal to 3, instead than the 4.9 of the SC19 pedal), and, even if this means that the pedal needs a bigger force to be actuated (as reported in 2.2.1), it requires also less pedal travel, therefore allowing to save space in case an actuator is to be placed behind the pedal. Therefore, for the first two concepts, the force required at the pedal

and consequently the force needed from the actuator, was evaluated with the same formulas reported in section 3.2, but considering a pedal ratio of 3 instead than 4.9.

3.4.1 Single actuator

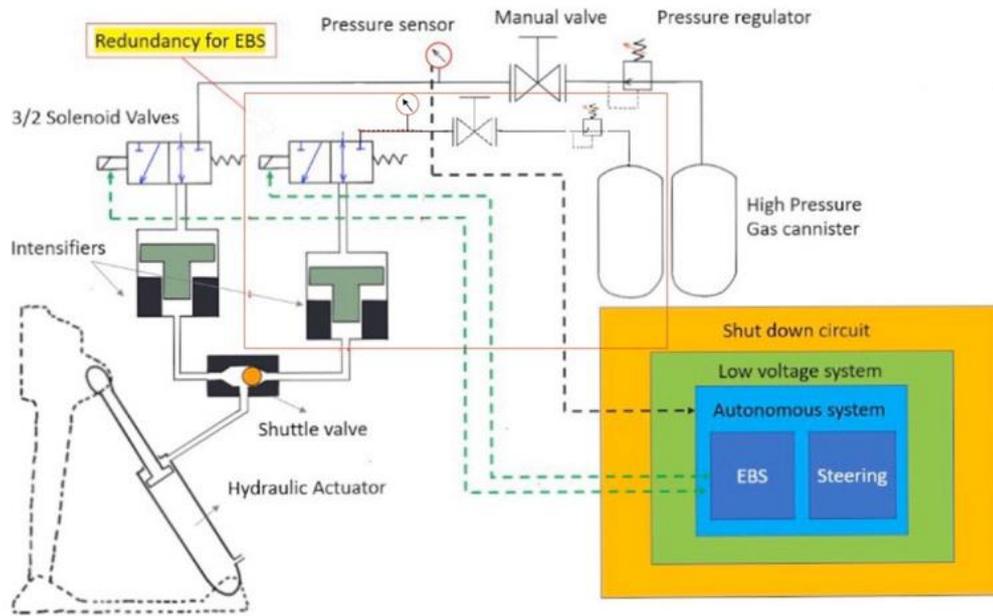


Figure 3.6 – EBS first concept layout, using a single hydraulic actuator to move the pedal

The first idea, suggested also from a previous thesis work that was done for the project [13], was to use a single hydraulic actuator able to generate the force necessary to pull down the brake pedal guaranteeing the required braking performances. The scheme of the system is reported in the previous Figure 3.6.

Hydraulic actuators are devices able to generate a mechanical force, that can be used to drive an output member, using the pressure of a liquid acting on a piston surface. They are suitable for applications where high actuation speeds and forces are required. Respect to pneumatic actuators, using air instead than liquid to generate pressure, hydraulic ones can guarantee higher actuation forces (up to 25 times greater considering cylinders of equal size) even using a piston of modest area [14]: this means that a high power output can be obtained using a small weight and size component. In addition, another important feature is the possibility of an hydraulic actuator to firmly hold its position, applying a constant force without the need of more fluid to be supplied; this is because the fluid inside the device (generally oil)

does not yield appreciably under stress, unlike in pneumatic solutions where the working fluid (air) is more elastic. More the fluid can be considered as non-compressible, more instantaneous is the power transmission.

The main issue of hydraulic actuators is that they have to be properly sealed, otherwise they can be subjected to fluid leaking, leading to system inefficiency and to potential damage of components.

System relevant calculations

To dimension the system components, it was necessary to first evaluate the force that the hydraulic actuator has to apply to pull the brake pedal. This force was calculated starting from the one which is required at the brake pedal to guarantee braking under maximum deceleration conditions (evaluated using the formulas reported in 3.2), as:

$$F_{actuator} = F_{pedal} SF \frac{Amplification\ factor}{Load\ factor}$$

SF is the considered safety factor, while an amplification factor (related to the pedal travel geometry) and a load factor related to the high required actuation rate (according to DV 3.3.1, the time between system triggering and start of deceleration must be less than 200 ms), were also taken into account [15].

Starting from this force and from an assumed initial value for the inner and outer diameters of the actuator piston (as well as of the piston stroke), the required hydraulic pressure needed on the actuator can be evaluated as

$$Hydraulic\ pressure = \frac{F_{actuator}}{A_{actuator\ pull\ side}}$$

$$A_{actuator\ pull\ side} = \frac{\pi(D^2 - d^2)}{4}$$

The starting values of the assumed parameters were chosen considering a trade-off between system compactness and required hydraulic pressure on the oil side. Then, the volume of fluid displaced by the actuator when pulling the pedal is

$$\Delta V_{actuator} = Stroke_{actuator} * A_{actuator\ pull\ side}$$

For the intensifiers, the oil side diameter is assumed around 70-80% of the outer diameter of the actuator.

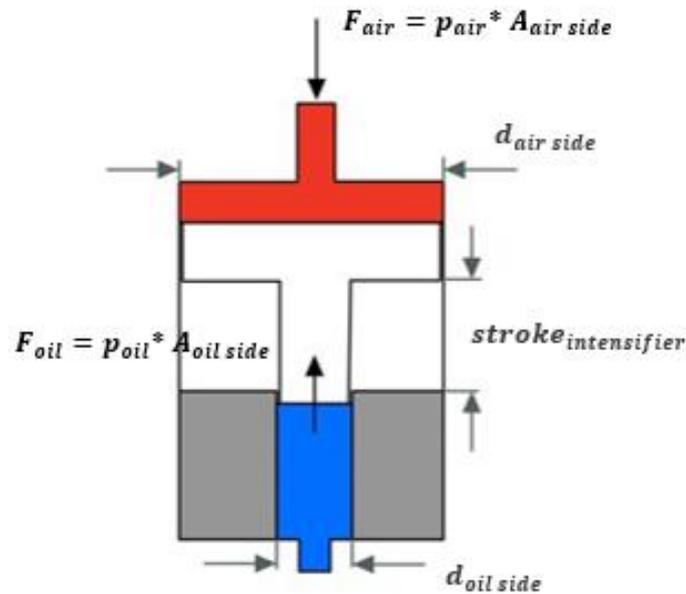


Figure 3.7 - Image showing the main dimensions and forces used for intensifiers calculations

Starting from this value, the oil side area of the intensifier is easily found from the equilibrium condition on the component ($F_{oil} = F_{air}$, with reference to Figure 3.7), and intensifier stroke and air side diameter can be subsequently evaluated as

$$stroke_{intensifier} = \frac{\Delta V_{actuator}}{A_{oil\ side}}$$

$$d_{air\ side} = \sqrt{\frac{Hydraulic\ pressure}{Pneumatic\ pressure}} * d_{oil\ side}$$

From here, using the formula reported below, the volume of air that must be displaced by the intensifier to obtain the desired displaced oil volume (indicated as $\Delta V_{actuator}$), is

$$\Delta V_{air} = stroke_{intensifier} * A_{air\ side}$$

For the canisters instead, calculations were done in order to set a volume and an internal pressure requirement for providing a number of EBS actuations sufficient to comply with the regulation. The component is then over-dimensioned respect to this requirement to guarantee a higher number of actuations respect to the minimum

necessary, considering a trade-off between the improved performances and the increased device weight and dimensions that this would lead.

To evaluate the number of possible actuations using a canister with a given volume and pressure, an iterative procedure has been carried on. Assuming constant the volume of gas in the tank, as well as the operating temperature, it is considered that for each actuation the system needs a certain ΔV_{air} , and therefore a certain mass of 10 bar air in the pneumatic line. This mass of air is to be subtracted from the canister, so it is initially at canister pressure.

$$\Delta m_{act,i} = \rho(CO_2, 10 \text{ bar}, 25^\circ C) * \Delta V_{air}(10 \text{ bar})$$

$$\frac{p_i}{m_i} = \frac{p_{i+1}}{m_{i+1}}$$

$$m_0 = \rho(CO_2, 130 \text{ bar}, 25^\circ C) * V_{canister}$$

So, for each actuation, the residual pressure level in the tank after this mass at canister pressure is subtracted is re-evaluated (p_{i+1} in the previous formulas), iterating the computation up to the point in which pressure in the device is not able to trigger the system anymore (i.e. less than the required 10 bar).

The relevant calculations for this first concept are presented in Table 3.3 of section 3.4.4.

3.4.2 Double actuator

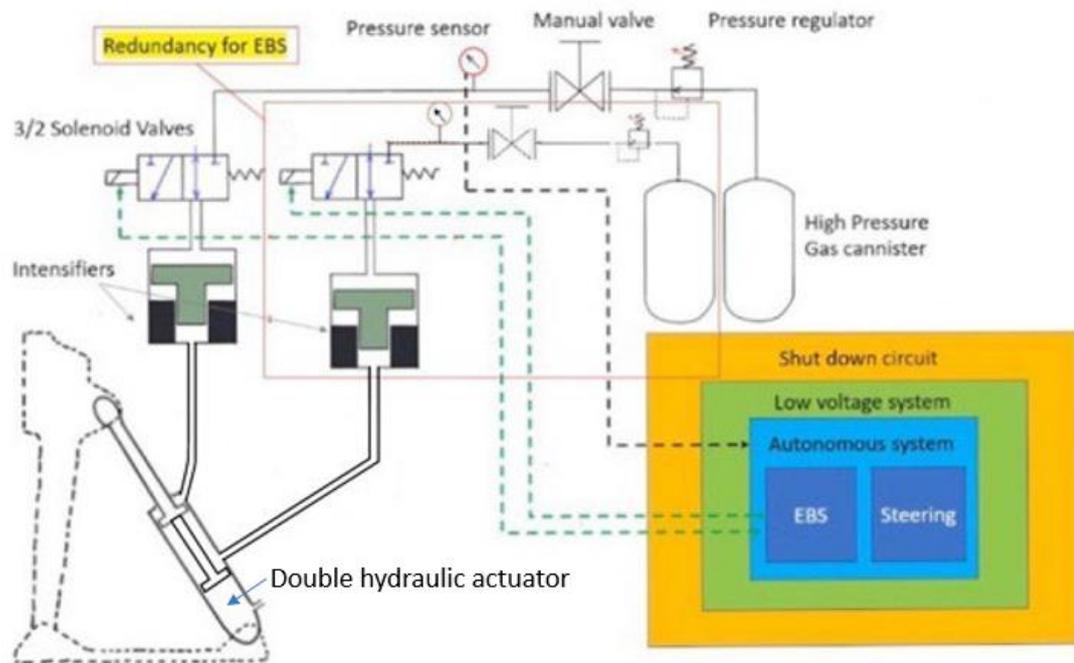


Figure 3.8 - Figure showing the circuit layout of the second concept, deploying a double hydraulic actuator

For the second proposed layout, the functioning of the EBS circuit remained basically unchanged. The main difference with the first concept is that a double actuator is used instead than a single one to generate the force required to pull down the brake pedal. A double hydraulic actuator (referred to also as tandem cylinder) is a system composed by two cylinders which are located into two separate chambers but driven from a common shaft, therefore designed as a single unit [16]. Since the fluid flow from and to the two chambers is provided by two different hydraulic systems, these types of components are very useful in applications requiring two independent circuits. Respect to a single hydraulic cylinder a tandem actuator is able to produce higher forces for the same operating pressure, with a smaller cylinder diameter. The main disadvantage can be represented, at least for the application described in this paragraph, from the fact that they generally require a substantial axial length. In this second system, whose layout is presented in Figure 3.8, each of the two parallel lines is acting on one piston of the tandem cylinder. Since the two multipliers are

therefore directly connected to the actuator, the OR valve that was previously necessary to establish which line had to control the piston is not needed anymore.

In the Figure 3.9 below, the functioning of the component is explained. When both lines are working, each will produce a certain force on the piston to which it is connected (indicated as F_1 and F_2 on Figure 3.9a), and the total force applied on the common shaft will be given by the sum of the single forces generated by both lines. Assuming instead that one line would have a failure, the force coming from that line will not be present, and therefore the resulting force on the piston will be equal to the force applied by the working line only (Figure 3.9b). This second condition, the most critical one, was the one considered for the actuator design. Calculations were done so that the force applied to pull the pedal in the case in which only one line is working would still be sufficient to generate a deceleration able to stop the vehicle within 10 meters starting from an initial speed of 40 km/h (as requested by rule IN 6.3.3).

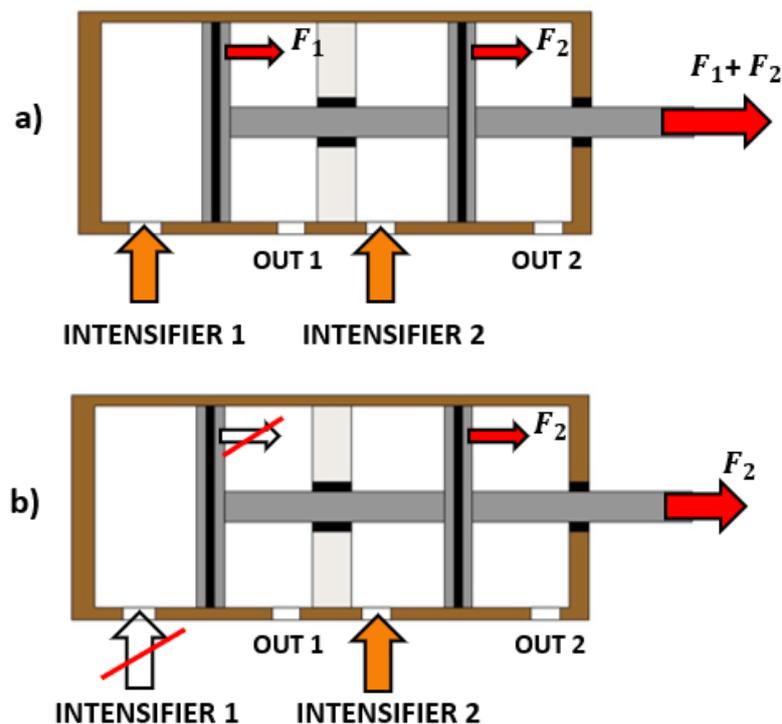


Figure 3.9 - Image showing the working principle of the tandem cylinder: a) during normal functioning, assuming both lines working; b) in case of one line failure.

Calculations procedure

Starting from the minimum deceleration value allowing to stop the vehicle within 10 meters (evaluated using the basic stopping distance formula reported in 2.1.1), the same procedure explained in 3.2 for pedal force evaluation and in 3.4.1 for the system components design was followed considering the case in which only one line is working. Once the main parameters (as intensifier stroke, air side diameter and oil side diameter) were identified, the force produced by the actuator was doubled, to simulate the case in which both lines are working and to evaluate how the system would behave. Then, the intensifier and actuator parameters were adjusted in order to reach a trade-off between the two possible conditions: to not generate a too high deceleration when both chambers are working, but at the same time to respect IN 6.3.3 if one line fails. The numerical results of these calculations are showed in Table 3.3 of section 3.4.4.

3.4.3 Acting directly on brake lines

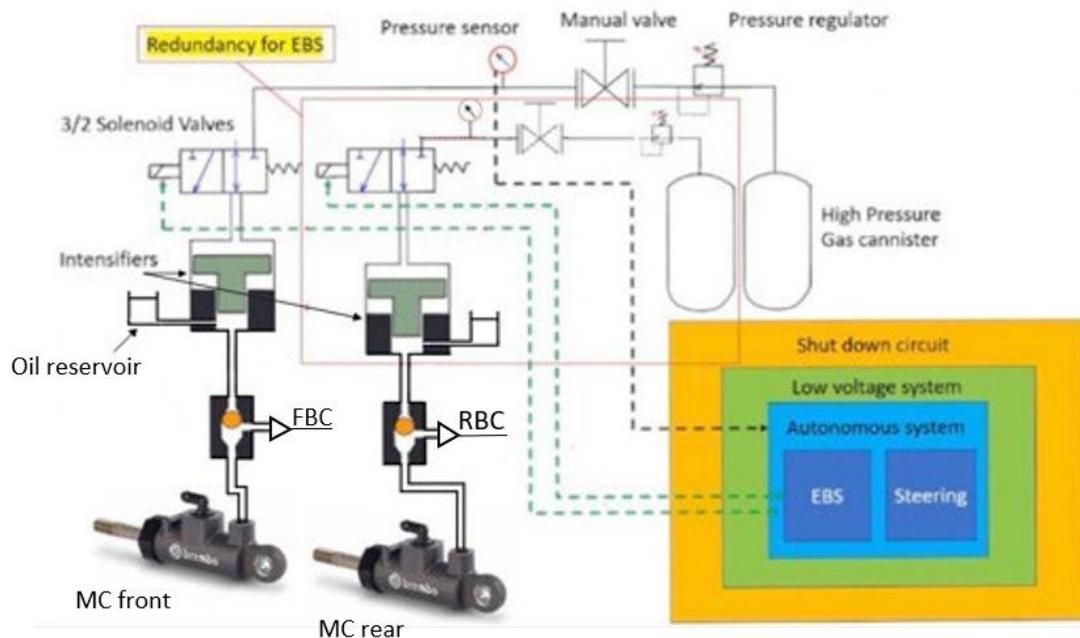


Figure 3.10 - Figure showing the circuit layout of the third concept, deploying direct actuation on brake lines

The third studied solution is presented in this section. It is characterised by the same basic circuit and components of the other two concepts described above up to the actuation part, which exploits a completely different principle. Instead than using a hydraulic actuator for generating a force to pull down the pedal, this system directly acts on the brake lines. The intensifiers are in fact directly connected, by means of OR valves, to the vehicle hydraulic braking lines (one to the front and one to the rear circuit, respectively indicated as FBC and RBC in Figure 3.10). Each OR valve is then connected on the other input port to the master cylinder of the corresponding line, and is therefore in charge of establishing, basing on which is the prevailing pressure between the two inputs, who is in charge of brakes actuation, if the intensifier or the master cylinder.

This configuration is designed in order to reduce the number of components and consequently the system weight and dimensions, but also to have a clearer distinction between manual driving and autonomous driving modes. To efficiently manage the passage between these two modes, different components have been considered: in

addition to OR valves, on/off manual valves and 3/2 solenoid valves were taken into account. Manual valves represented the simplest solution, but two valves would be required for each line (one placed between intensifier and hydraulic circuit and one placed between master cylinder and hydraulic circuit). In addition, they need to be manually operated every time that is necessary to switch from manual to autonomous driving and vice versa: this last point is not fully compliant with rule DV 3.1.6, since to access those components it would be necessary to dismount the entire front-end of the vehicle.

So, 3/2 solenoid valves were considered for this role: in this case, only two valves would be necessary (one for each line), but they are bigger and heavier, other than more complex to control.

Hence, OR valves resulted to be the best choice: only one component per line is required and they are smaller, lighter, and with a simple control logic, allowing to passively switch from manual to autonomous without the need of any manual operation.

When in manual driving, the EBS system is disabled (manual valve at intermediate position), there is no pressure build-up in its circuit, and hence the master cylinders, commanded by the brake pedal, are in charge of pressurizing the brake lines. In autonomous driving mode instead, the manual valves are in the ON position, and the EBS is ready to start the emergency braking whenever triggered by the solenoid valves. When the system is actuated, it generates on the intensifiers output a hydraulic pressure that prevails over the one of the master cylinders side (since the brake pedal is not pressed), connecting the multipliers with the hydraulic brake lines. In this way, also DV 3.1.5 is fully respected.

In order to allow a correct system functioning the connection to an oil reservoir, as well as the presence of discharge and purge ports, are necessary on the intensifiers to permit the pressure release after the actuation is performed, in the brake releasing phase. To reduce the number of components in the system, the pressure multipliers can be connected to the oil reservoirs already present in the pedal assembly of the vehicle (described in 2.1.1) and used by the master cylinders with the same aim.

Calculations procedure

Removing the actuator, from a calculation point of view, implies that the actuator force, and consequently the force needed at the pedal (from which the calculations for the other systems started) are not necessary anymore. The starting point, since actuation takes place directly on the hydraulic lines, is now the pressure level required at front and rear lines to guarantee braking under maximum deceleration conditions, evaluated in section 3.2. The aim is to design the intensifiers as to simulate the two master cylinders of the system, but able to generate the required hydraulic pressure starting from pneumatic pressure instead than from an input force. Hence, the intensifiers oil side diameters, as well as their strokes, are initially assumed to be equal to the master cylinder ones. Then, starting from the hydraulic pressures for front and rear lines, the air side diameters are evaluated with the formula reported in 3.4.1. Finally, performances in terms of vehicle deceleration, stopping time and stopping distance are computed for each of the system different possible conditions (front line failure, rear line failure, or both lines working), to check also its rule compliance. Numerical results of the calculations are presented in Table 3.5 of the following section.

3.4.4 Comparison between the different solutions

The numerical results for the main calculations of the previously described solutions are reported in the tables below, to compare the three concepts from the point of view of rule compliance (in terms of system redundancy and performances), design and packaging. It should be remarked that the results concerning the canisters requirements are shown only for the first solution, since the specifications (in terms of volume and internal pressure) found for the component resulted to be compatible also for the other concepts.

SINGLE ACTUATOR CALCULATIONS		
Actuator piston outer diameter	D	25 mm
Actuator piston inner diameter (rod)	d	12 mm
Actuator stroke	$stroke_{actuator}$	30 mm
Safety factor	[-]	1.2
Load factor	[-]	0.6
Amplification factor	[-]	1.44
Deceleration	$-a$	17.66 m/s ²
Stopping distance	S_{stop}	3.50 m
Stopping time	t_{stop}	0.63 s
Pedal force*	F_{pedal}	910.05 N
Actuator force	$F_{actuator}$	2620.94 N
Pneumatic pressure	p_{air}	10 bar
Hydraulic pressure	p_{oil}	69.38 bar
Intensifier oil side diameter	d_{oil}	17.5 mm
Intensifier air side diameter	d_{air}	46 mm
Intensifier stroke	$stroke_{intensifier}$	47 mm
Oil volume displaced by actuator	$\Delta V_{actuator}$	1.13*10 ⁻⁵ m
Air volume displaced by intensifier	ΔV_{air}	7.86*10 ⁻⁵ m
Minimum n° of actuations considered	[-]	5

Canister pressure	$p_{canister}$	130 bar
Canister volume	$V_{canister}$	$75 \cdot 10^{-3}$ l
N° of actuations	[-]	11

*Pedal force evaluated using the pedal ratio of the SC18 brake pedal, see 3.4 introduction

Table 3.3 - Table reporting the main calculations results for the first system

DOUBLE ACTUATOR CALCULATIONS			
Safety factor		[-]	1.2
Load factor		[-]	0.6
Amplification factor		[-]	1.44
Actuator force	One chamber	$F_{actuator}$	878.01 N
	Both chambers		1756.03 N
Pedal force*	One chamber	F_{pedal}	508.11 N
	Both chambers		1016.22 N
Deceleration	One chamber	$-a$	6.47 m/s ²
	Both chambers		20.80 m/s ²
Stopping distance	One chamber	S_{stop}	9.53 m
	Both chambers		2.97 m
Stopping time	One chamber	t_{stop}	1.72 s
	Both chambers		0.53 s
Pneumatic pressure		p_{air}	10 bar
Hydraulic pressure		p_{oil}	43.67 bar
Intensifier oil side diameter		d_{oil}	14 mm
Intensifier air side diameter		d_{air}	30 mm
Intensifier stroke		$stroke_{intensifier}$	40 mm
Actuator piston outer diameter		D	20 mm
Actuator piston inner diameter		d	12 mm
Actuator stroke		$stroke_{actuator}$	30 mm

*Pedal force evaluated using the pedal ratio of the brake pedal of SC18, see 3.4 introduction

Table 3.4 - Table reporting the main calculations numerical results for the second system

BRAKE LINES ACTUATION CALCULATIONS			
Pneumatic pressure		p_{air}	10 bar
Hydraulic pressure at front brake line		p_{front}	40.97 bar
Hydraulic pressure at rear brake line		p_{rear}	53.03 bar
Intensifier oil side diameter		d_{oil}	16 mm
Intensifier stroke		$stroke_{intensifier}$	6.35 mm
Intensifier air side diameter (front circuit)		$d_{air\ front}$	32.39 mm
Intensifier air side diameter (rear circuit)		$d_{air\ rear}$	36.85 mm
Deceleration	Both lines working	$-a$	17.66 m/s ²
	Only front line		11.48 m/s ²
	Only rear line		6.18 m/s ²
Stopping distance	Both lines working	S_{stop}	3.50 m
	Only front line		5.38 m
	Only rear line		9.99 m
Stopping time	Both lines working	t_{stop}	0.63 s
	Only front line		0.96 s
	Only rear line		1.80 s

Table 3.5 - Table reporting the main calculations numerical results for the third system

Redundancy

As stated in DV 3.2.1 (reported in Appendix A and already cited in section 3.3), the system should remain fully functional in case a single failure happens. Hence, there is the need of a completely redundant system, able to guarantee the required performances even in case of one failure. So, the different presented solutions were critically analysed in order to detect the main issue when dealing with redundancy: the singularity of components. A fully redundant system is in fact, by definition, a system in which all the critical components are doubled, so that in the event of failure of one of them, the whole system functioning is not compromised.

With the implementation on each system of two twin circuits in parallel one to each other, the requested redundancy level is assured up to the intensifiers. So, the analysis is now focused on the actuation part, downstream of the pressure multipliers.

1. The single hydraulic actuator solution cannot be considered fully redundant, due to the fact that there is only one device (the OR valve) to connect the two intensifiers with the actuator: in case there is a failure in one line, the other one is still working and able to deliver pressure, but if the OR valve fails (in case of valve locking) the system would not be able to deliver any pressure to the actuator, invalidating the whole braking operation.
2. For the second solution, exploiting a hydraulic double-chamber tandem cylinder for actuation, this cannot happen, since each line is directly connected to an actuator chamber. So, there is not a single critical component which malfunctioning would compromise the whole system functioning as in the previous case, except for the actuator itself: if it somehow locks, the EBS actuation fails. A solution with a couple of actuators to be placed behind the brake pedal was also considered, to deal with this issue (which holds also for the first concept), but it would hugely increase the total weight and complexity of the system, so it was discarded.
3. The third system guarantees the highest redundancy level: acting directly on the brake lines there is no need of any actuator at all, and the two lines going from the hp canisters to the OR valves are completely specular, meaning that each component is doubled in its twin-circuit. This solution can be therefore considered as fully redundant.

Performances

From the point of view of performances, the relevant entities to be referred to, between all the values reported in tables 3.3, 3.4 and 3.5, in order to compare the different solutions, are: vehicle deceleration ($-a$), vehicle stopping time (t_{stop}) and vehicle stopping distance (S_{stop}). The last two parameters are evaluated from an

initial velocity of 40 km/h, which is the minimum velocity to be reached from the vehicle during the brake test, as stated by rule IN 6.3.3 (reported in Appendix A).

1. The first solution, being designed to achieve the vehicle maximum possible deceleration is clearly compliant with performance requirements, generating an deceleration of 17.66 m/s^2 (corresponding to 1.8 g), much higher than the minimum required value of 8 m/s^2 (DV 3.3.2) and allowing the vehicle to completely stop in only 3.50 m, much less than the 10 m requested by IN 6.3.3.
2. For the second solution, the design aim was to set the system specifications in order to allow the vehicle to be 100% rule compliant and pass the tests even in case of one line failure (i.e. when only one chamber is working). However, in this way the deceleration value that the vehicle would have to withstand in case both actuator chambers are working (so during normal operation) resulted to be very high. So, the system was adjusted performing a trade-off between the maximum decelerations for the two cases, to have a value which is sustainable from the vehicle when both lines are working, but that is not too low when only one line is generating pressure. The obtained results are reported in Table 3.4: in case of both lines working the computed values for deceleration and stopping distance are 20.80 m/s^2 and 2.97 m respectively, both compliant with rules. In case of only one chamber operation, the stopping distance requirement would be respected (9.53 m), while the deceleration value (6.47 m/s^2) would result lower than the required one.
3. For the third solution, three working modes are available, considering also the system possible malfunctions: both lines working, only front line working and only rear line working. For each of the three cases, the performance parameters (reported in Table 3.5) are evaluated, with the result that in the first two conditions the system would be fully rule compliant, while for the third one, in which only the rear brake line is working, only the requirement on the stopping distance would be respected (with 9.99 m over 10), while the deceleration vale, equal to 6.18 m/s^2 , would result to be less than the required 8 m/s^2 .

Packaging

The three different solutions are evaluated also from the point of view of system complexity, total weight and overall dimensions, taking into account also what emerged from the preliminary meetings with the company in charge to build the physical components.

1. The first solution would require two components for the actuation part (actuator and OR valve), and it is the system that also necessitates of the biggest intensifiers, having to generate a huge hydraulic pressure (69.38 bar).
2. For the second concept instead, only one component (the double-chamber actuator) is needed downstream of the pressure multipliers. Even if its diameters are reduced respect to the single-chamber actuator, the physical device resulted to be huge, with an overall length of approximately 300 mm. In addition, the system design would be more complex because of additional gaskets needed on cylinder rod and pistons due to the second chamber, and leading to higher frictions especially on the brake release phase (therefore, also the spring would have to be verified to guarantee the correct return of the pedal). Also, the correct cylinder filling operation, avoiding the formation of air bubbles, would be more critical. For what concerns the intensifiers, their dimensions are reduced respect to the first solution, as well as the required hydraulic pressure (43.67 bar).
3. The third solution necessitates of two components downstream of the intensifiers but being just two small OR valves the system weight and dimensions are much reduced respect to the other solutions. In addition, the third is the configuration requiring the least intensifier stroke: only 6.35 mm are needed, respect to the 47 and 40 mm required by first and second solution, respectively.

In the following table, the main advantages and disadvantages of the different solutions are summarized.

1. SINGLE HYDRAULIC ACTUATOR	
PROs	CONs
<ul style="list-style-type: none"> • Performance requirements are fully met • Low actuator dimensions 	<ul style="list-style-type: none"> • Low redundancy degree, not fully rule-compliant system
2. DOUBLE CHAMBER HYDRAULIC ACTUATOR	
PROs	CONs
<ul style="list-style-type: none"> • Higher redundancy degree respect to single actuator • Lower intensifier dimensions • Only one component needed downstream of the intensifiers 	<ul style="list-style-type: none"> • Huge actuator dimensions • More complex system design • Possible issues with the deceleration value when only one chamber is working • Still not complete redundancy considering the case in which the actuator locks
3. BRAKE LINES ACTUATION	
PROs	CONs
<ul style="list-style-type: none"> • Minimum system weight and dimensions • Fully redundant system (lines are completely specular) • Simplest system design • Performances requirements fully met in case of both lines or only front line working 	<ul style="list-style-type: none"> • Possible issues with deceleration value when only the rear line is working

Table 3.6 - Table summarizing the main characteristics of the different solutions in terms of redundancy, performances and packaging.

From what presented above and summarized in Table 3.6, the best concept resulted to be the direct actuation on brake lines. It is in fact the best solution in terms of packaging (smallest intensifiers, only two OR valves needed for the “actuation part”),

and guarantees a 100% degree of redundancy, since the lines are fully specular. Considering performances instead, the only issue could be that when only the rear line is working, the deceleration value could be too low to be compliant with DV 3.3.2. However, this problem can be overcome by regulating the pressure level on the pneumatic line connected to the rear hydraulic circuit to a value higher than the 10 bar considered for the above calculations, in order to increase the force acting on the brake discs and consequently the rear braking capability, hence generating a higher deceleration. The direct actuation on brake lines was therefore chosen as the final EBS design layout.

3.5 Components design, evaluation, and in-vehicle positioning

In this section, the components which are present on the EBS realised circuit are described. Some of them had to be designed (as intensifiers, supports, hydraulic lines) and then manufactured from the contacted companies, while others (as HP canisters, pressure regulators, pressure sensors, manual and solenoid valves) had to be selected and bought basing on their properties and on their compliance with the requirements of the designed system. Also, great importance was given to weight and dimensions of each component, to increase the vehicle mass of the least possible amount (to not affect performances) and to use in the best way the available space into the monocoque. For each presented component, the basic function, selection criteria, geometry description and positioning inside the SC19 monocoque are reported.

3.5.1 Intensifiers

Intensifiers (or pressure multipliers) are, as reported in section 3.3, the components in charge of building the hydraulic pressure required to actuate the brake lines, receiving pressurized air as input. The devices main design specifications are reported in Table 3.5 of the previous section, while the company contacted in order to manufacture them is Fluido Sistem, specialised in custom pneumatic and hydraulic systems. A weight reduction analysis was carried on the initial proposal for the device, focusing on the used materials. The drawing of the final device chosen for the system with main dimensions and ports, is shown in the following Figure 3.11.

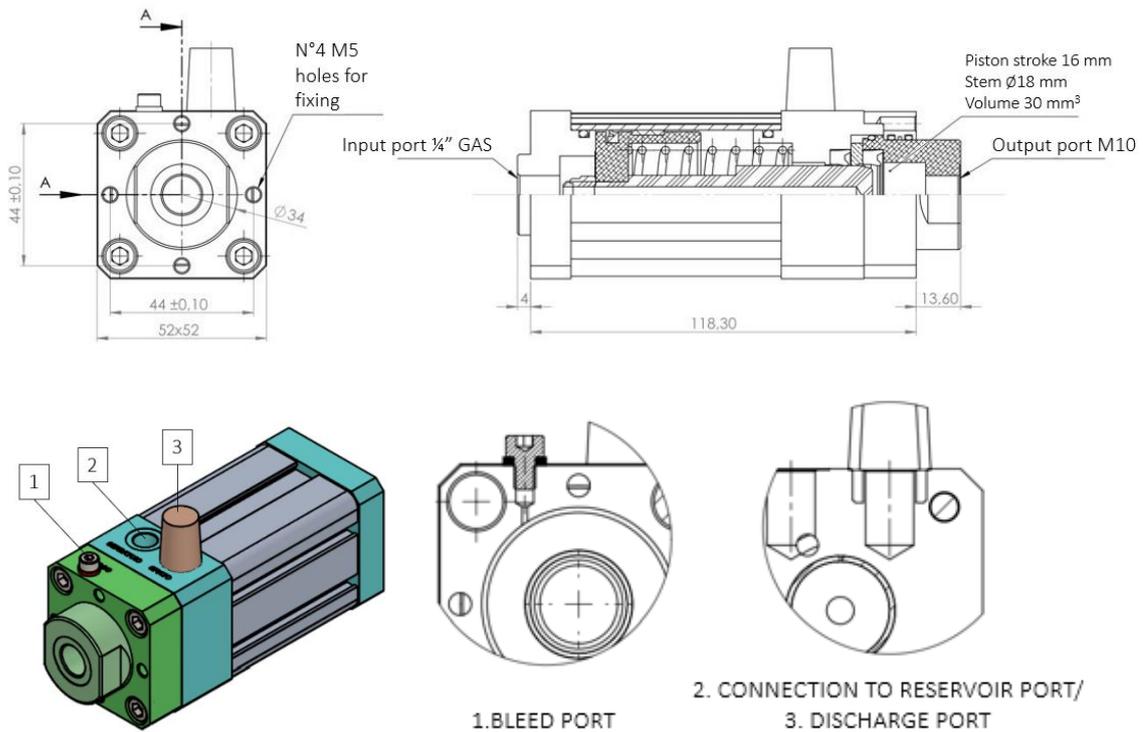


Figure 3.11 - Figure evidencing the main intensifier characteristics in terms of dimensions and connections

Except for the piston stem, which is made using C40 steel, intensifiers are completely realized in ergal (also known as 7075 aluminium alloy), which is an aluminium alloy characterized by excellent mechanical properties but that is at the same time lighter respect to steel. The adoption of this material allowed a weight reduction of about 33% (from 1138 to 760 g) respect to the initial design.

The air side and oil side diameters are slightly different from the ones reported in Table 3.5, as well as the piston stroke. In addition, the two intensifiers for front and rear lines are of the same dimensions, and not different as it was initially designed. These different specifications were concorded with the manufacturing company considering the differences between the designed “idealized” device and the actual realization of the physical component, needing additional parts (as a spring to ensure a correct return of the piston when the device is released, piston end-stops, gaskets, sealings, and the necessary bleed, discharge and connection ports shown in Figure 3.11), and also high strength to be able to sustain high forces application in a very reduced amount of time (the so-called water hammer).

A quick actuation capability is in fact required, with the start of deceleration that must take place in maximum 200 ms from the system triggering, as stated by rule DV 3.3.1. The main parameters of the actual intensifiers are reported in the following table.

INTENSIFIERS - FINAL SPECS	
Oil side diameter	18 mm
Air side diameter	40 mm
Piston stroke	16 mm

Table 3.7 - Table reporting the final parameters for the intensifiers.

With these specifications, it was guaranteed that the device would be able to reach an output hydraulic pressure of approximately 40 bar when receiving 10 bars of pressurized air at the input port. The pressure level would still be enough to ensure the required performances in cases of both lines and only front line working. In the event of only rear brake lines functioning instead, the input pressure for the intensifier acting on rear lines should be regulated to a value higher than 10 bars to allow the required EBS performances (as already expressed in section 3.4.4).

3.5.2 OR valves

OR valves (or shuttle valves), as reported in 3.4.3, are the components establishing which are the parts in charge of brake actuation, if the master cylinders (and therefore the brake pedal) or the pressure multipliers. Their working principle is explained in section 3.3. Different components were taken into account: one proposed by Fluido Sistem (the same company in charge to build the intensifiers) and another manufactured from The Lee Company, whose main characteristics are reported in Table 3.8 below.

	Fluido Sistem	The Lee Company
Cost (for each valve)	100 €	420 €
Dimensions (LxWxH)	73x15x25 mm	26,16x6,35x6,35 mm
Weight	152 g	4 g

Table 3.8 – Table reporting the main characteristics for the different considered OR valves.

Both solutions resulted to be well performant, able to work within the system range of pressures and with very low fluid leakages, but if the second one had the advantages of lower weight and dimensions, it was also much more expensive than the other one (420 € versus 100 € each). So, after doing a trade-off between costs and correspondent benefits (mainly, in terms of weight saving), the proposal from Fluido Sistem was chosen, also in the perspective of a possible weight reduction if changing the material from the initial brass to aluminium. A CAD image of the final component and its connections is shown in the next Figure 3.12.

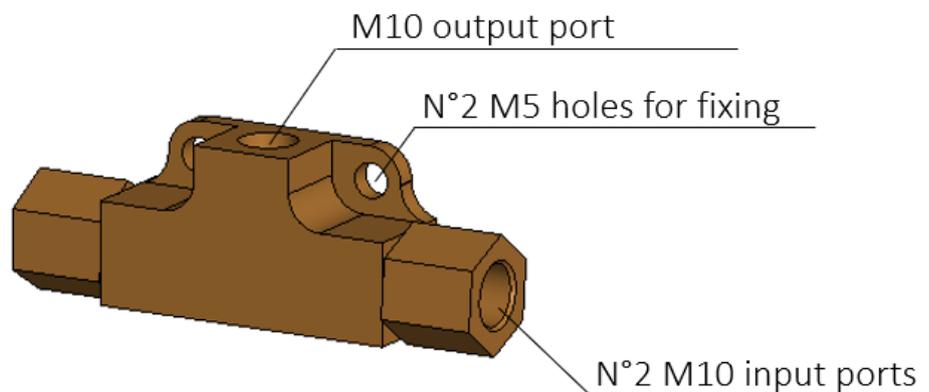


Figure 3.12 - Overview of the OR valve chosen for the system also showing the available connections.

3.5.3 Support for intensifiers and OR valves

For the OR valves, the most suitable location appeared to be the front end of the vehicle, near to the pedalbox assembly: in this way, they are close either to the brake master cylinders, as well as to the connections with the hydraulic braking lines. Since it would be better to place also the intensifiers close to these valves (in order to limit the length of the hydraulic lines, obtaining a more compact system), a single support was designed to integrate both the pressure multipliers and the shuttle valves, to make the best possible use of the limited space available inside the SC19 monocoque. Different layouts have been evaluated for the support, with the main design steps reported in Figure 3.13. The design was changed basing on the results of structural analyses that were done on the component to verify its strength while keeping a low weight. The aim was in fact to obtain a structure able to withstand without

deformations not only the forces generated by the components, but also the external forces to which it can be subjected to (as, for example, an unintentional collision with the driver foot).

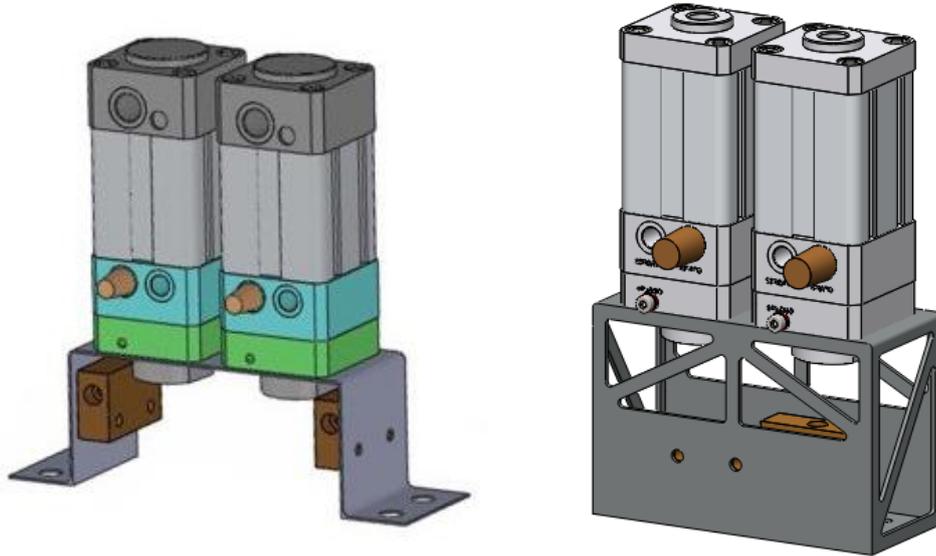


Figure 3.13 - Different solution evaluated for the intensifiers and OR valves support.

The first version (on the left part of the first figure), was a simple flange with intensifiers on top and OR valves positioned on the side surfaces. In its second version instead, the support was designed with a boxed-shape, open in the lower surface and with OR valves positioned on the inside surfaces of the front and rear sides. On the which are present on all the side surfaces (necessary to insert the OR valves connections), triangulations are realised to increase the component stiffness.

The structural analysis of the component was carried considering the application of a force of 50 kg at the top of the intensifier (to simulate a strong impact with the driver foot) both in the X and in the Y directions.

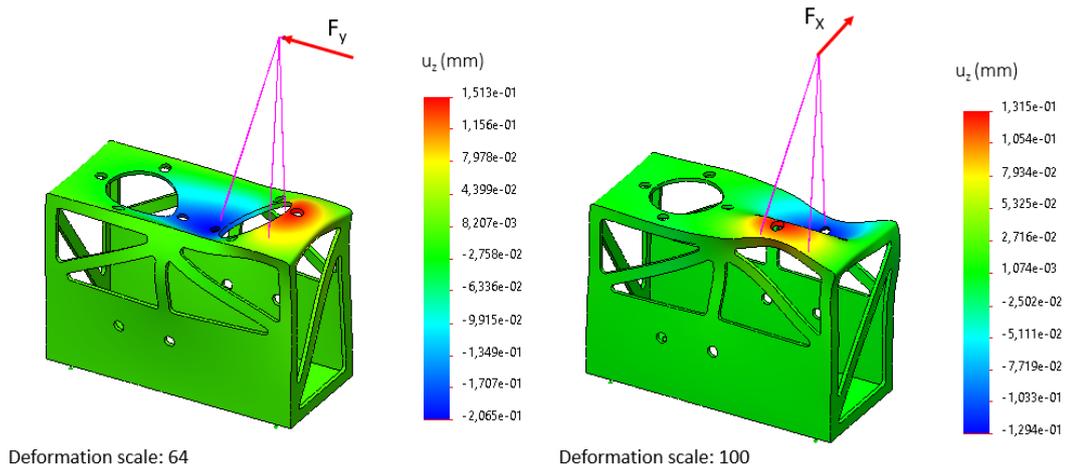


Figure 3.14 - Figure showing the deformed shape and the values of Z-displacement of the support when applying a force along the Y-axis (left) and on the X-axis (right)

In Figure 3.14, the obtained results in terms of deformation along the Z axis are displayed. It can be observed that, despite the applied force is quite high (almost 500 N), the component undergoes a very low deformation, reaching a maximum of 0.2 mm in the region in which it is compressed (coloured in blue, in the left image of Figure 3.14). The deformation scale is a parameter indicating how much the results are to be amplified to obtain the showed graphical representation of the deformation. Another important analysis that was done to validate the component design is about eventual collisions which may happen between the full subassembly and the moving parts placed in the mounting environment and, in particular, with the accelerator pedal: since the chosen location for the complete subassembly (made of intensifiers, support, and shuttle valves) is behind the accelerator pedal, it is important to avoid collisions when the pedal is pressed during manual driving, to not impair the system performances. So, from the available data about the maximum accelerator pedal travel, the correspondent pedal rotation was evaluated to verify that no collisions were happening between the components. A basic scheme showing this operation is reported in the following Figure 3.15.

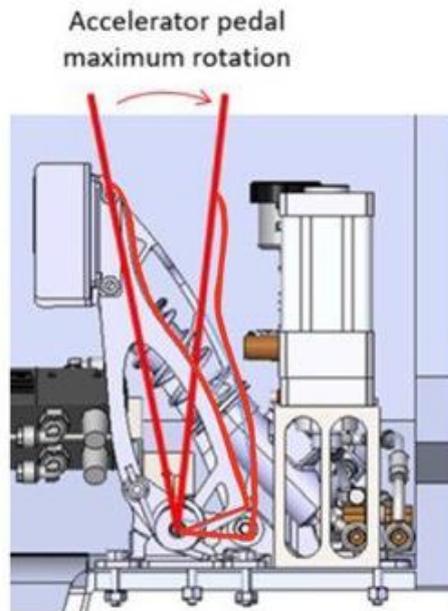
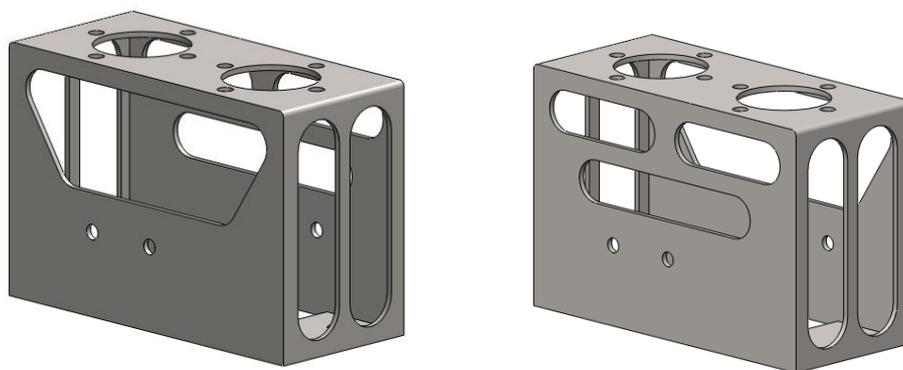


Figure 3.15 - Scheme evidencing the accelerator pedal rest position and maximum travel

Once having verified its structural performances and that no collisions were happening, the second version of the support was chosen to be realised, but during the production phase its shape was slightly changed, in agreement with the company in charge of the manufacturing operation. The changes did not affect in a significant way the component strength, that remained substantially the same (thanks mainly to the 2 mm thickness adopted for every surface), but allowed an easier positioning and connection of the hydraulic lines inside the component. The drawing of the final support, with its dimensions, is reported in Figure 3.16 below.



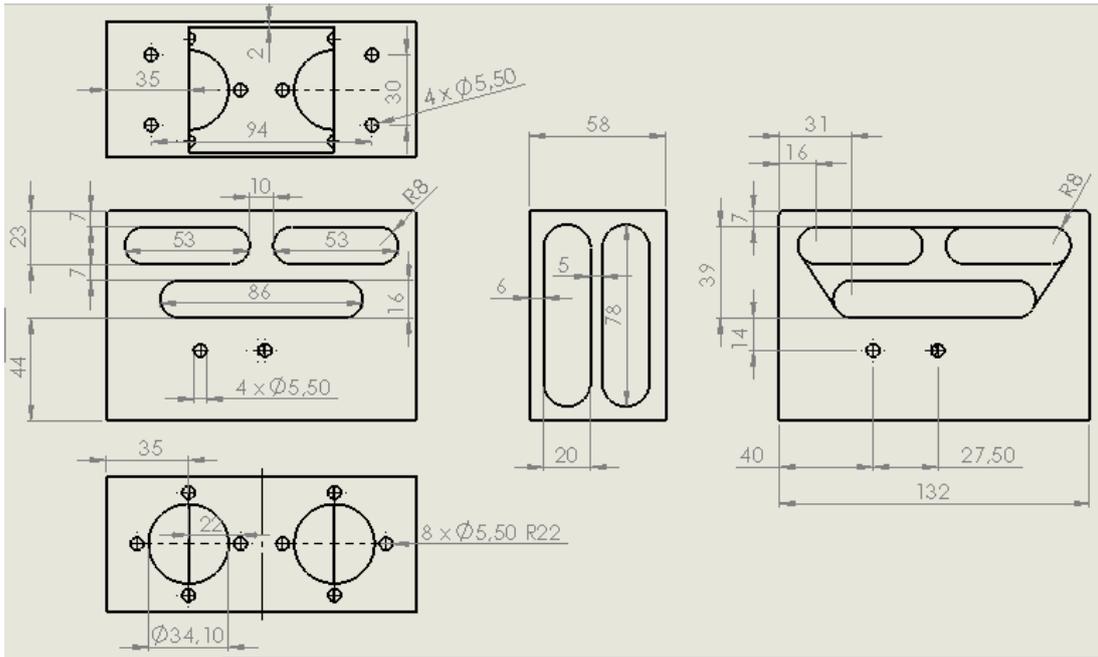
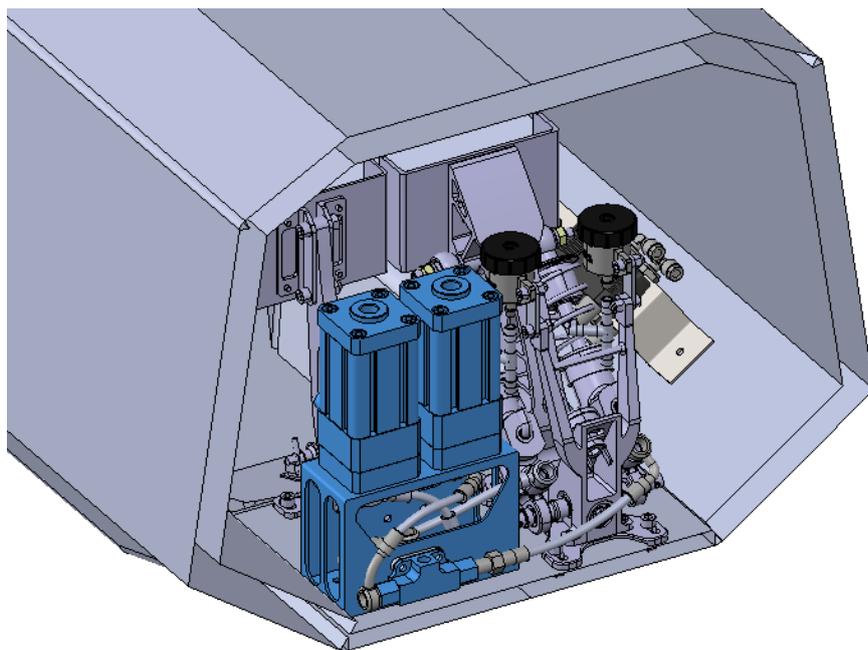


Figure 3.16 - Figure showing 3D CAD views from SolidWorks (above) and the technical drawing of the final version of the support (below).

The support is realised in ergal (7075 aluminium alloy) to maintain a low overall weight respect to steel. In the images below, a CAD view of the complete subassembly (including intensifiers, shuttle valves and support) positioned inside the vehicle monocoque, behind the accelerator pedal, is shown.



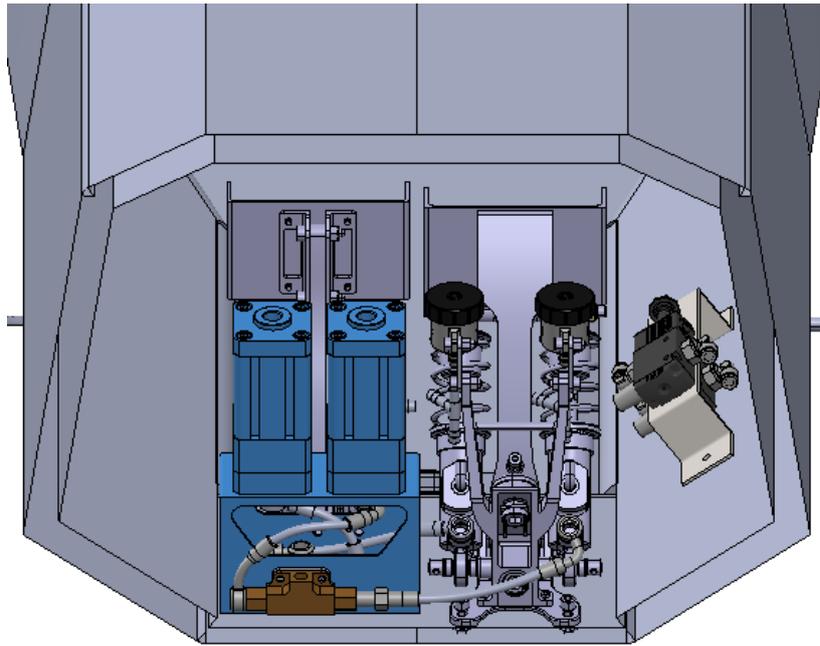


Figure 3.17 - CAD images, from SolidWorks, showing the positioning of the complete subassembly in the vehicle.

3.5.4 3/2 Solenoid valves

3/2 Solenoid valves have the fundamental role of triggering the EBS when needed. Their basic working principle is explained in section 3.3. The required component is an electrovalve which is normally open, meaning that at its rest position (i.e. when the electrical command is not provided) is open, connecting the hp canister with the intensifier and triggering the EBS, while it closes (connecting pressure multiplier to discharge port) when an electrical command is given. Also in this case, different components were considered, and the more convenient solution resulted to be a valve produced by AirComp: the EV8 1/4" 22 3 SL PM NO M. It is in fact a lightweight device (120 grams), with reduced dimensions and fully compliant with the system specifications: capability to work with a 12 V DC supply (that is the tension of the vehicle LV system, from which the electrovalve is powered), and within the system pressure range.

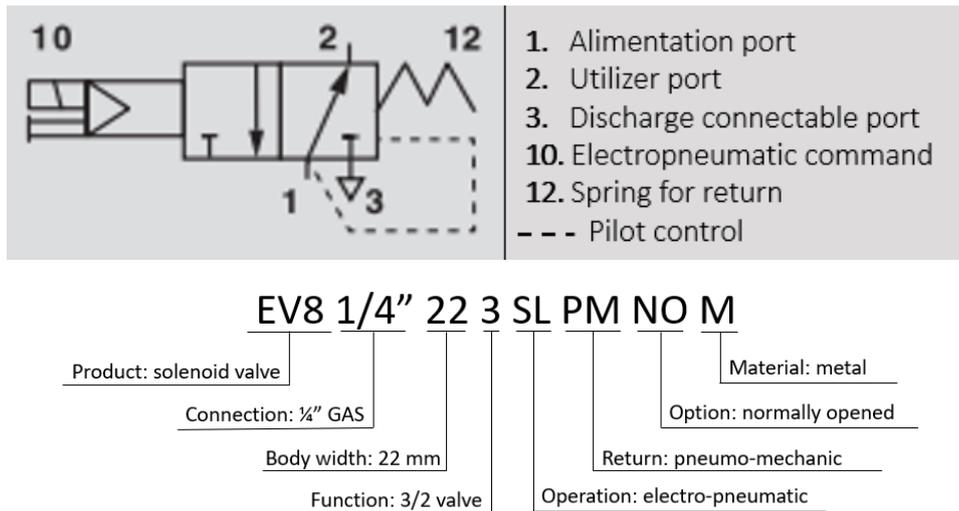


Figure 3.18 - Figure showing the pneumatic scheme for the chosen electrovalve, and its main characteristics.

As reported in Figure 3.18, showing the valve pneumatic scheme, the command of the device is electropneumatic [14]: therefore, to bring the valve to its closed position not only a voltage (which is provided through the coil), but also a certain pressure has to be applied to the alimentation port, to overcome the spring force and the pilot control. The materials used for the component realization are displayed in the following Figure 3.19, taken from the AirComp catalogue.

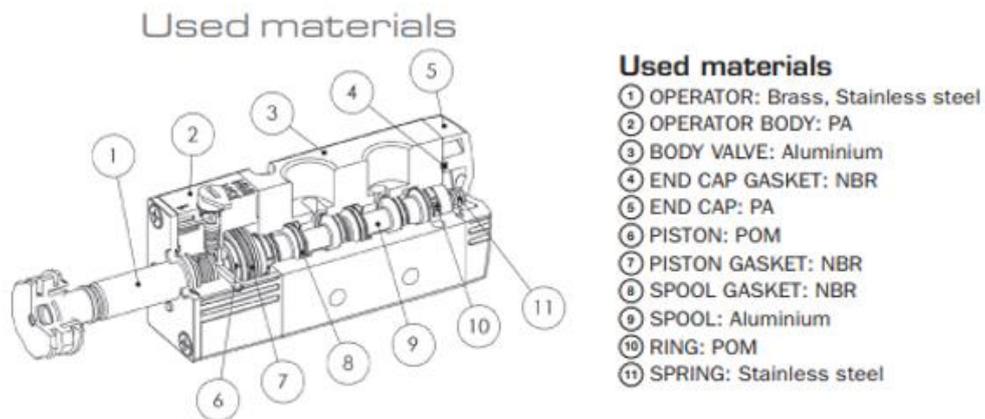


Figure 3.19 - Figure listing the materials utilized for the electrovalve.

The main dimensions of the component and of the connection ports are reported in the following Figure 3.20, where also the coil (in charge of providing the electric signal) and the related electrical connections are shown (in the last image).

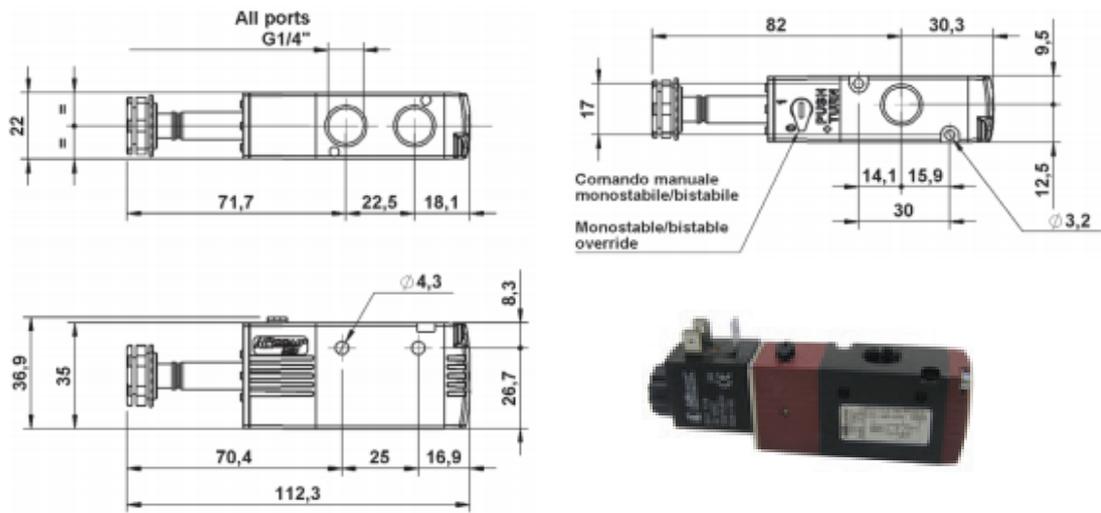


Figure 3.20 - Overview of the dimensions and ports for the 3/2 solenoid valve. In the last figure, also the coil mounting is shown.

3.5.5 Support for 3/2 Solenoid valves

Solenoid valves are located, in the EBS circuit, between manual valves and pressure multipliers. Therefore, they must be in a position which is easily reachable from these components, identified with the front part of the vehicle, on the left side of the driver. To have a compact, more ordered, and easy to mount system, a support was designed for these components. Its main dimensions are reported in Figure 3.21 below.

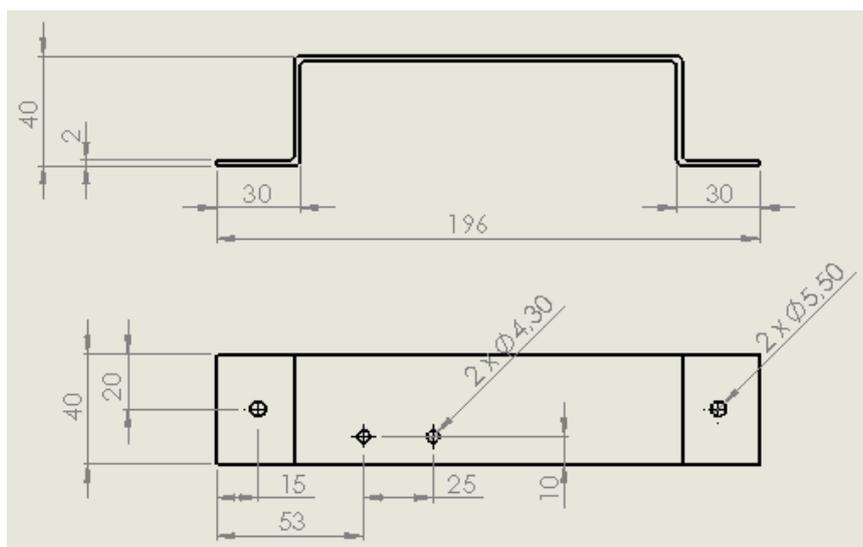


Figure 3.21 - Figure showing the main dimensions for the solenoid valve support.

The support is entirely manufactured in ergal (7075 aluminium alloy), with a thickness of 2 mm to be able to sustain stresses without deformations. In the image below, the 3D CAD model from SolidWorks is shown for the component alone and for the complete subassembly made up of the support, the two solenoid valves needed for the EBS system and the connections required at the ports. 90° fittings are used for connections with the pneumatic lines (from manual valves and to pressure multipliers, respectively) while silencers are placed at the discharge ports of the valves.

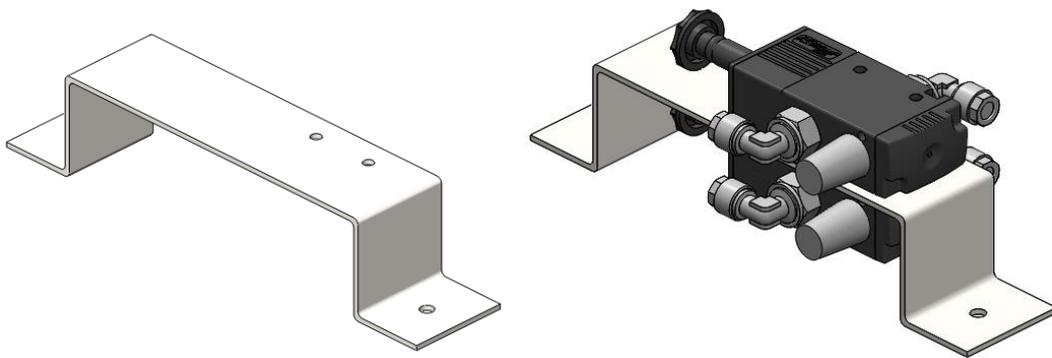


Figure 3.22 - CAD images showing the support for solenoid valves (left) and the complete subassembly (right).

Finally, in the figure below, a CAD image shows the positioning of the solenoid valves subassembly inside the vehicle.

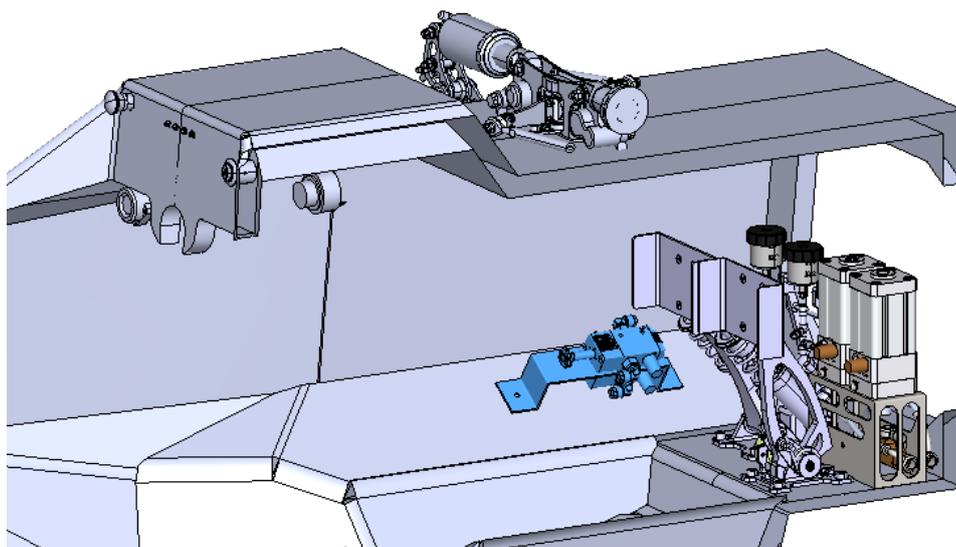


Figure 3.23 - CAD image showing the position of the solenoid valves subassembly inside the vehicle

3.5.6 Manual valves

Manual valves are in charge of pressurizing the whole pneumatic lines. As discussed in 3.3, basing on their position they can prevent the EBS from functioning (when, for instance, manual driving is required) or they can put the system into the “armed” state, allowing the 3/2 solenoid valves to trigger the emergency braking when needed. To choose the right component, different solutions were evaluated, taking into account of weight, dimensions, eventual limits on pressure at the ports and cost. A three ports-three positions valve resulted to be the best solution: compact, light, and cheaper than other alternatives (less than 10 € each), able to work within the required pressure levels, and presenting the possibility to work at an intermediate position (evidenced in Figure 3.24) in which the connection between the input and both output ports is closed. This is an important requirement, since it means that disconnecting the pneumatic circuit is not necessarily leading to the complete discharge of the canister pressure into ambient (as it would be in a three ports-two positions valve), but that the tank pressure can be maintained.

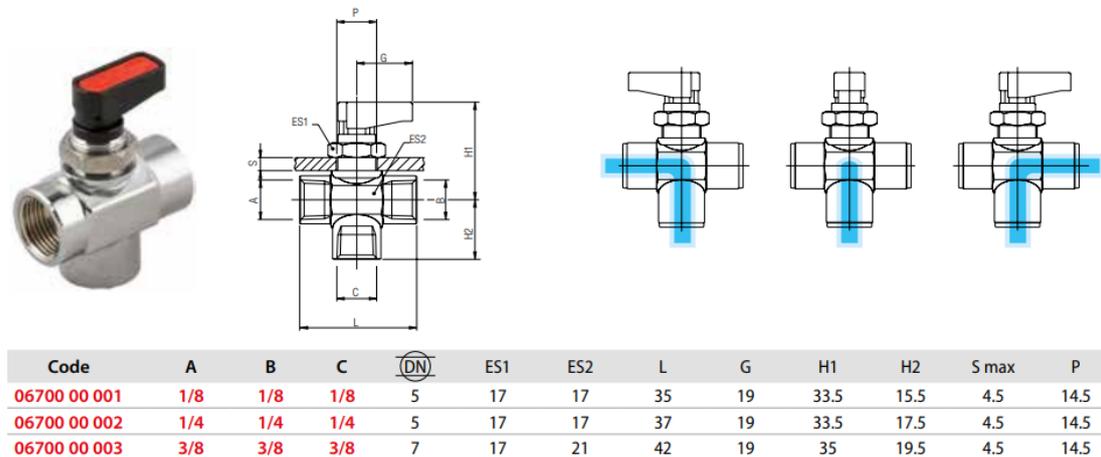


Figure 3.24 - Figure showing the main dimensions for the chosen manual valve. The purchased configuration is the 002, with all the ports of 1/4”.

3.5.7 Support for manual valves

As EBS deactivation points, manual valves are directly subjected to rule DV 3.1.6, and are therefore to be placed in a position in which they can easily be accessed and operated. A suitable location can be on the back on the driver seat, from where they can be easily connected also to the canisters, and from where the output pneumatic line can easily enter inside the monocoque (running at the left of the driver) to reach the solenoid valves in the front part of the monocoque. To integrate and to firmly hold the valves and their connections, a support was designed, to be then 3D-printed and glued to the monocoque external surface using a strong structural adhesive.

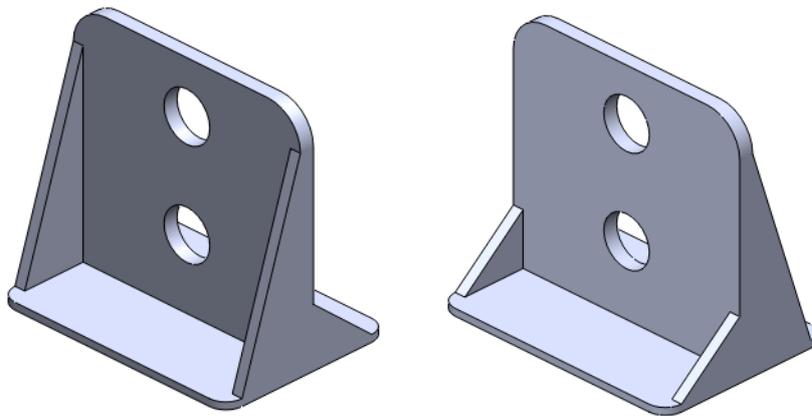


Figure 3.25 - CAD image of the manual valves support.

Figure 3.25 shows the CAD model of the support, while in Figure 3.26 the complete subassembly including also manual valves and connections is presented.

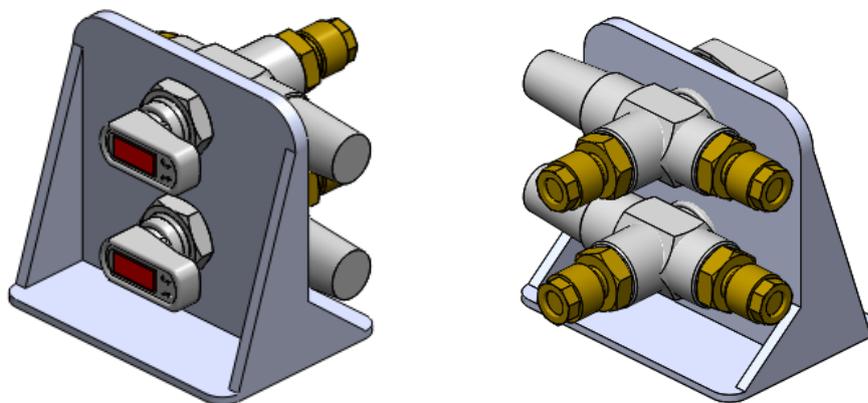


Figure 3.26 - CAD image of the complete manual subassembly

Straight fittings are used for both the input and output pneumatic connections, while a silencer is mounted to the output port which is working when the valve is in the OFF position, to allow an eventual canister discharge. The technical drawing showing the main dimensions of the component is reported below. The thickness of the vertical plate was increased respect to the one of other surfaces up to 4 mm, to obtain a better resistance in the zone in which the manual valves are to be tightened. The material used for the realization of the support is PLA, a polyester used for 3D printing which allows to obtain a very light component.

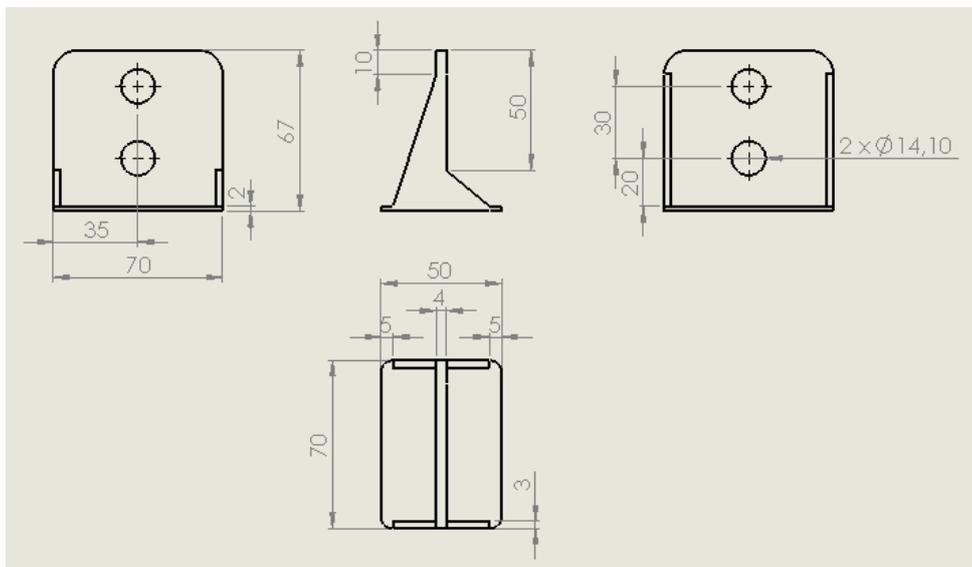


Figure 3.27 - Drawing showing the support dimensions.

The positioning of the manual valves subassembly in the vehicle is shown in the following Figure 3.28.

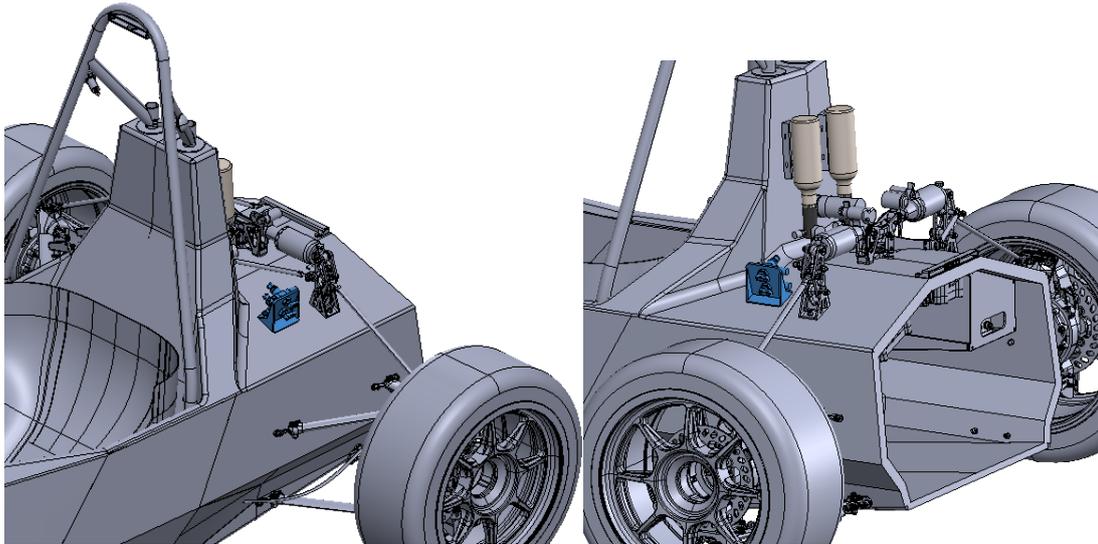


Figure 3.28 - CAD images showing the positioning of the manual valves subassembly in the vehicle.

3.5.8 HP canisters and pressure regulators

The high-pressure canisters are the components located at the upstream end of the EBS circuit, with the role of pressurizing the pneumatic lines. To be able to perform an entire mission without having to refill the tanks, it was evaluated a minimum number of 5 required actuations per canister:

- Two for the initial check-up sequence of EBS and of its redundancy, needed in order to pass in the AS Ready mode, as stated in DV 3.2.4. In this sequence, the twin circuits are tested firstly together and then singularly to verify if they are able to build the expected pressure value. More details about the check-up sequence will be provided in section 4.1.3.
- One to pass to the Ready to Drive mode (more exhaustively described in Chapter 4.1.3).
- One to end the mission and stop the vehicle.
- An additional pressure loss would take place when bringing the EBS in the “armed” state before the ASMS is on: in this case, being the ASMS initially open, there would be a pressure release from the canisters, and a consequent EBS actuation, until the ASMS is closed, bringing the solenoid valves in the closed position.

Starting from this number, the required specifications in terms of internal pressure and volume were set as reported in section 3.4. As for the other components, different solutions were considered and evaluated mainly basing on weight, dimensions and cost. Since most of the canisters working at around 130 bar were not able to reduce the output pressure up to the desired value (around 8-10 bar) with the built-in regulator, a separate additional pressure regulator was searched, obtaining a two-stages regulation. The chosen components and their main characteristics are displayed below.



X- SHORT PAINTBALL HP SYSTEM	
Internal filling volume	0.15 l
Filling pressure	200 bar
Length (with regulator)	215 mm
Built-in regulator minimum pressure	56 bar
Diameter	50 mm

MANCRAFT HRR REGULATOR	
Output pressure range	4.13 – 17.24 bar
Input connection	G 1/2" thread (compatible with canister)
Length	75 mm



Figure 3.29 - Main characteristics for the chosen system of hp canister (above) and pressure regulator (below).

Adopting these components and considering that the canisters are charged to the maximum filling pressure (200 bar) more than 30 actuations resulted to be possible (using the method presented in section 3.4.1 for calculations), much more than the 5 considered as necessary for a single mission.

3.5.9 Supports for canisters and pressure regulators

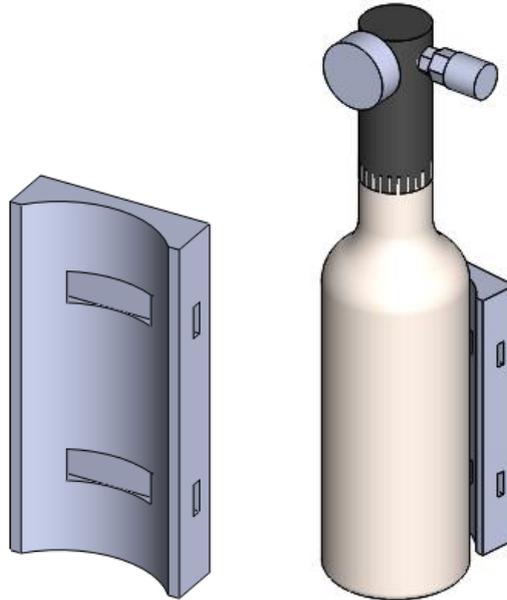


Figure 3.30 - 3D CAD image showing the support of the canister and the assembly of canister, support, and pressure regulator.

Considering the canister and regulator placement in the vehicle, a set of rules about compressed gas cylinders and lines (reported in section T9 of the Formula Student 2020 Rulebook) had to be considered. In particular, rule T 9.1.1 has to be considered, stating that any system on the vehicle that uses a compressed gas as an actuating medium must comply with the following requirements [11]:

- 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 T 1.1.14, 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.

Therefore, a suitable location for placing the components resulted to be the back side of the driver seat, in a position to be protected from rollover and collisions and mounted vertically to not point in the driver direction. In this way, it can be also easily connected to the manual valves subassembly, located in the same region. As in the case of manual valves, two supports (one for each canister) were designed to be 3D-printed in PLA and then glued in the desired position using a strong structural adhesive, able to ensure an adequate fixing. The components were designed basing on the canister dimensions and considering that the pressure regulator must be placed, as requested by rules, directly on top of the tank without intermediate stages. Passages are realised inside the supports to allow the fit of metallic cable ties, in charge of securing the canisters and regulators to the support itself. In Figure 3.30 CAD images showing the support alone and the subassembly made of canister, regulator and support are reported, while in the following figures the subassembly in-vehicle positioning is shown.

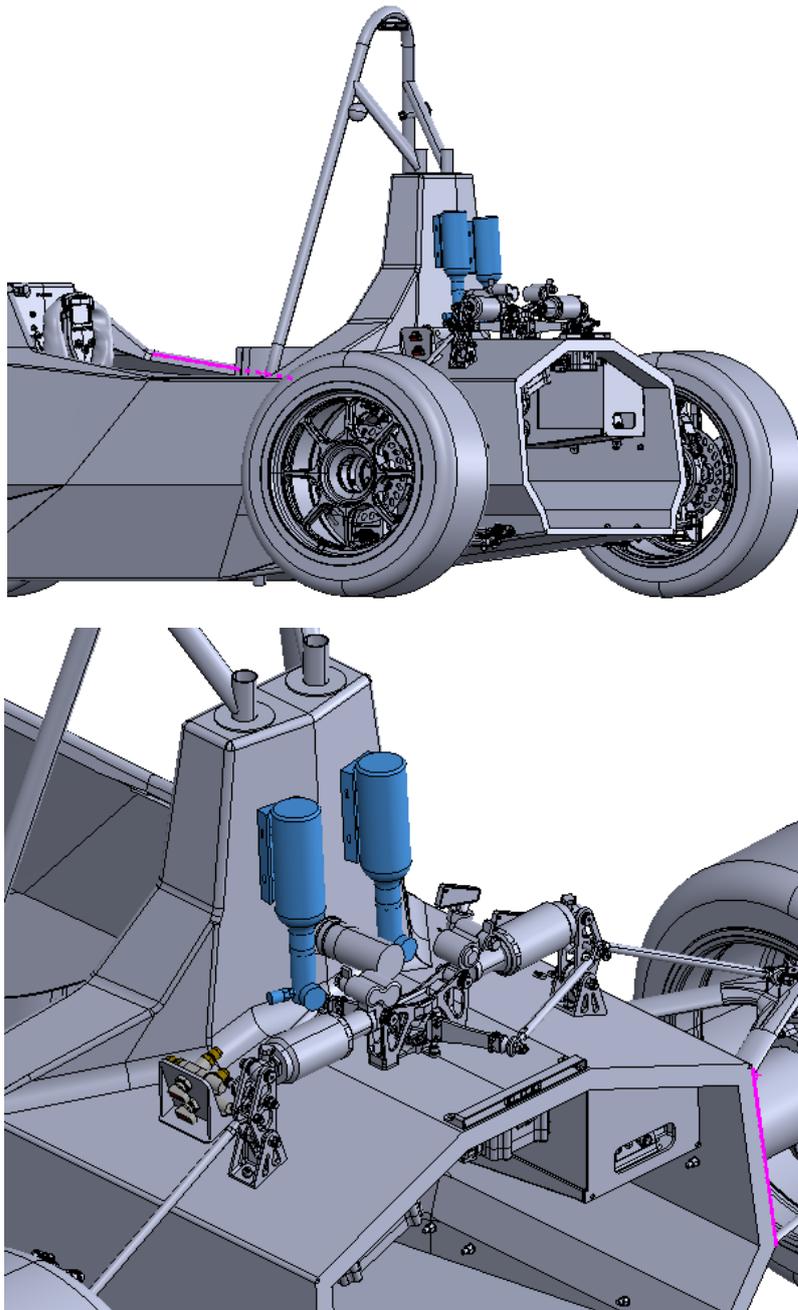


Figure 3.31 - CAD images showing the in-vehicle positioning of canisters, pressure regulators and their supports

3.5.10 Pressure sensors

In order to successfully complete the EBS transition into armed state (required for the vehicle to pass in the AS Ready mode), it is necessary to verify that the system is able to build up the required level of pressure. Therefore, in addition to the analog ones already present on the hydraulic lines (as reported in section 2.2.1), additional pressure sensors are placed in both the pneumatic circuits upstream of the 3/2 solenoid valves. The selected component and its principal characteristics are reported in Figure 3.32. It is a digital output pressure sensor: one end has to be connected to the pneumatic line, while the other one to a voltage supply (the LV system of the vehicle can be used) and to a digital output to read the signal. A threshold pressure can be set between 1 and 10 bar, and a LED light indicates if the value read from the line is higher or lower than this threshold.



Caratteristiche - Features

Fluido - Fluid	aria, gas no corrosivi e non infiammabili air, non corrosive gas
Campo di regolazione - Setting pressure range	0 - 10 bar
Pressione massima - Withstand pressure	15 bar
Tensione di alimentazione - Supply voltage	12 - 24V DC +/- 10%
Tempo di risposta - Response time	5 ms o inferiore / 5 ms or less
Tipo di uscita - Output method model	PNP 80 mA
Consumo corrente - Current consumption	21 mA max
Protezione IP - IP protection	IP40
Temperatura ambiente - Working temp range	0 - 50 °C (no condensation of freezing)
Umidità ambiente - Ambient humidity range	35 - 85% RH (no condensation)
Connessione - Port size	G 1/8 (M5 interno - M5 internal thread)
Lunghezza cavo - Cable length	3 m
Peso - Weight	50 g
Led	rosso - red

Figure 3.32 - Figure from the AirComp catalogue reporting the selected pressure sensor and its main specifications.

3.5.11 Pneumatic and Hydraulic lines

Pneumatic lines

Pneumatic lines have the role of putting in communication all the components of the EBS circuit positioned between the high pressure canisters and the intensifiers, to which they have to deliver the right value of pressure to allow the desired EBS performances. Both the twin lines of the system are realised using a polyamide tube with an internal and external diameter of 6 and 8 mm, respectively. A scheme of one of the two lines, with all the related components and the chosen connections is reported in the following Figure 3.33.

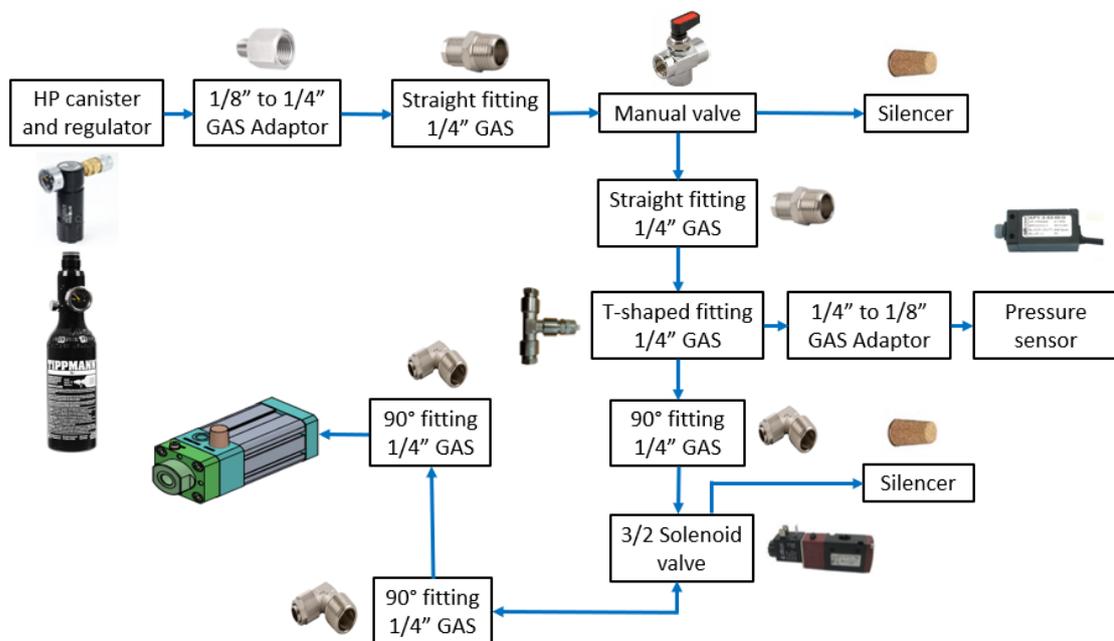


Figure 3.33 - Scheme of the pneumatic connections between the components.

Hydraulic lines

Hydraulic lines have the role of delivering the pressurized braking fluid from the pressure multipliers (or, depending on the position of the shuttle valve, from the master cylinders) to the brake callipers. In the following Figure 3.34, the hydraulic lines layout required for the EBS is reported.

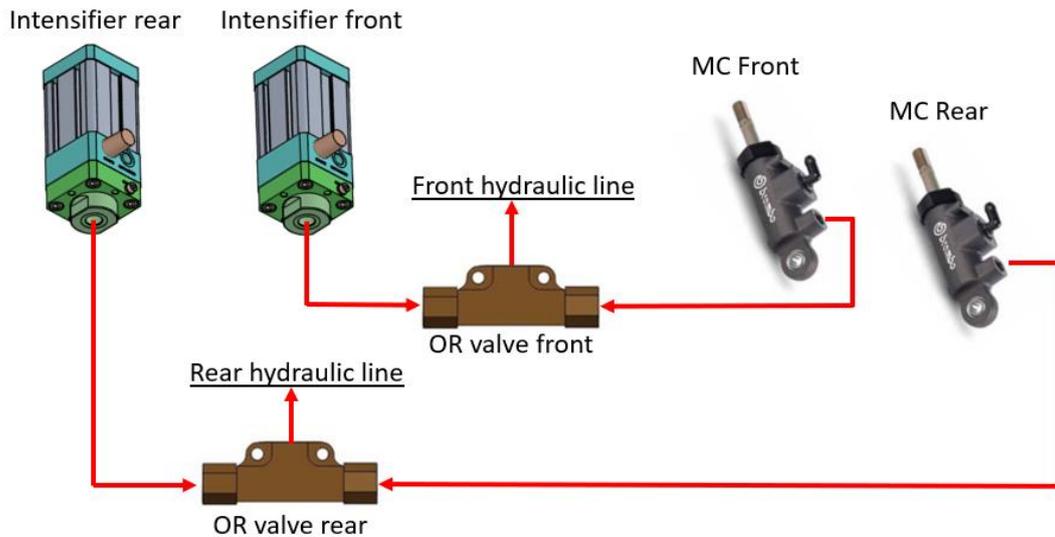


Figure 3.34 - Scheme of the system hydraulic connections

The total number of required lines is then six, three for each subcircuit. They are classified in the following Table.

FRONT SUBCIRCUIT	REAR SUBCIRCUIT
<ul style="list-style-type: none"> • Line 1, from intensifier front to OR valve front • Line 2, from master cylinder front to OR valve front • Line 3, from OR valve front to front t-joint 	<ul style="list-style-type: none"> • Line 4, from intensifier rear to OR valve rear • Line 5, from master cylinder rear to OR valve rear • Line 6, from OR valve rear to rear t-joint

Table 3.9 - Table reporting the hydraulic lines classification

As reported in section 2.2.1, hydraulic lines are already present in the vehicle. So, in order to add the least number of components, it was decided to reuse the existing lines whenever possible and to design only the new ones, which are necessary to connect intensifiers and master cylinders to the OR valves. So, lines going from the t-joints up to the brake callipers were kept, and a study was performed to check if the existing lines going from the master cylinders to the t-joints could be adapted in order to be positioned between the output ports of the OR valves and the t-joints. For the front line, this operation resulted successful: the banjo fitting can be removed from

the front master cylinder and mounted on the correspondent OR valve output without issues. For the rear line instead, due to the fact that the valve is placed further to the right (from the driver point of view) respect to the master cylinder, the existing line resulted to be too short to be connected. A different solution was evaluated, trying to prolong the line through the use of an extension, but it resulted not feasible because of the geometry of the mounted banjo fitting mounted on the hose. Therefore, with reference to Table 3.9, only Line 3 (and the portions of lines going from the t-joints to the brake callipers) could be maintained, while the other ones (1, 2, 4, 5, 6) were to be designed.

In order to design these lines, two important points have to be considered. First is that except for line 6, which is connecting components that are placed at a bigger distance between them, the lines are connecting components mounted in a very limited area with very narrow gaps and hence low possibility to make adjustments. In addition, while for pneumatic lines it was possible to cut the tubes and then mount the connections, obtaining a line of the desired length, hydraulic lines are to be ordered with already mounted fittings, so it is not possible to adjust their length in case it would be necessary. Therefore, each line must be carefully designed in order to avoid having a too long or too short hose that would not be able to fit in the required space. So, an initial study was performed to determine for each line which components were to be connected and where the connection would have to run.

To establish the main paths for the lines, the catalogue of the available fittings and connections was studied, considering their dimensions and all the different possible orientations. Then, each line was drawn in the SolidWorks environment of the system assembly and measured in order to obtain the parameters necessary to proceed with the order, as length and fittings orientation. The final design of the lines is shown in the images below.

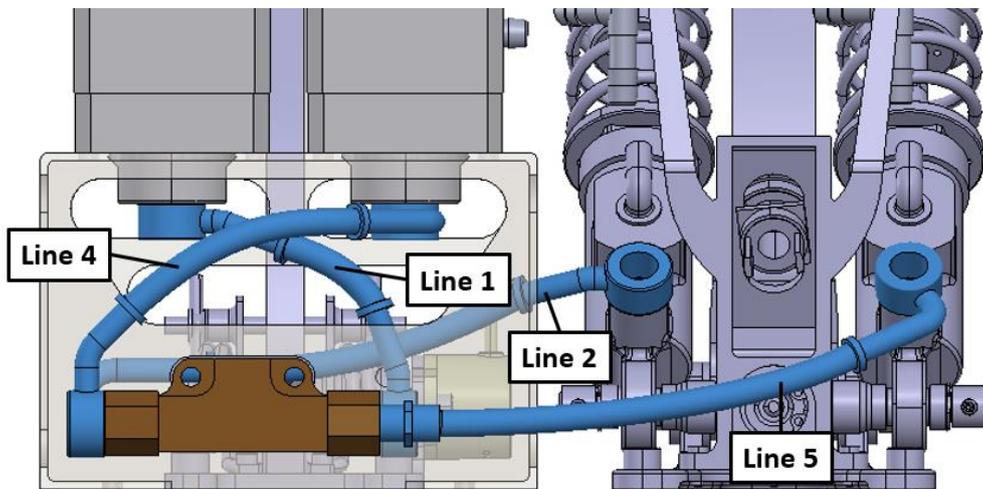
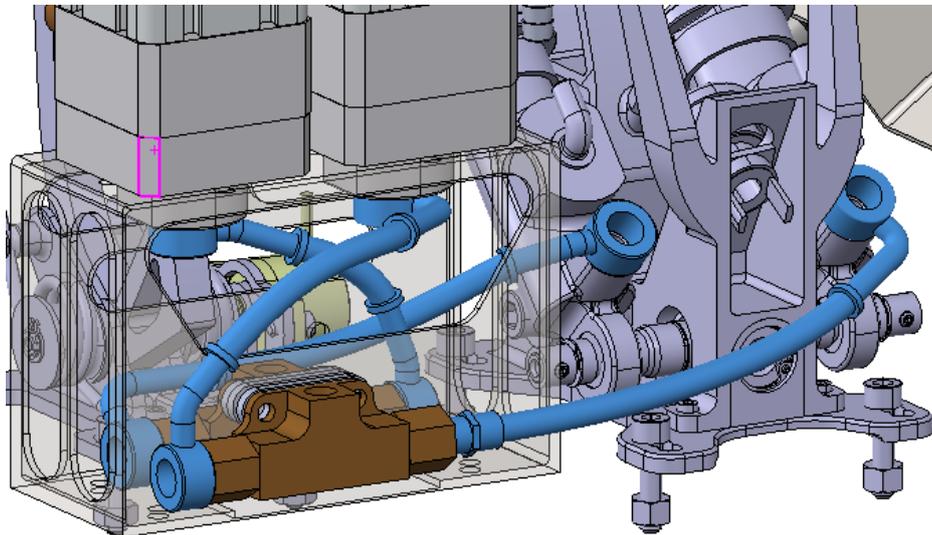
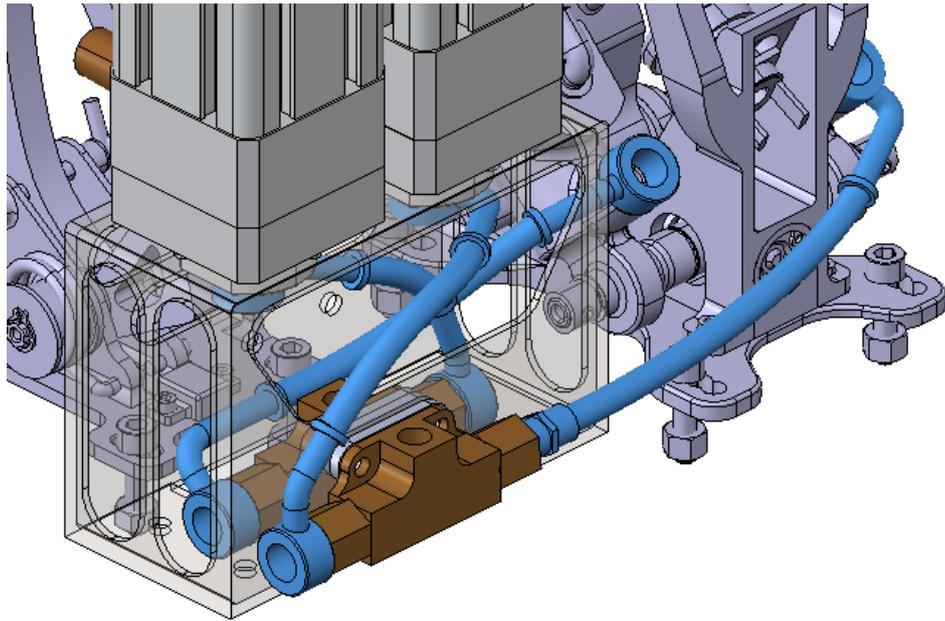


Figure 3.35 - CAD images evidencing the hydraulic lines and their position in the system

Line 6 is not represented in the previous images since its path was easier to design: it was sufficient to start from the existing line, which is ending on the rear master cylinder (the one on the right, with reference to the 3rd image of Figure 3.35), and extend it by measuring the distance between its end point and the OR valve output. Attention was given to choose an end fitting compatible with the t-joint input. The hoses are realised with an internal PTFE core and an external highly flexible metal braiding, guaranteeing high braking performances also at high temperature. The main characteristics of the component are reported in the following Table 3.10. Fittings are instead realised with a special aluminium alloy, called TITANAL, light and able to resist to the action of corrosive agents such as the oil used for brakes.

Size [in]	Internal diameter	External diameter	Minimum curvature radius	Maximum working pressure	Weight
1/8"	3.2 mm	7.3 mm	25.4 mm	310 bar	6.6 kg/100m

Table 3.10 - Table showing the main characteristics of the DASA hoses chosen for the hydraulic lines

The final specifications used for ordering the components are reported in Table 3.11 below, while Figure 3.36 shows the geometries of the used fittings. The lines were provided by Dasa.

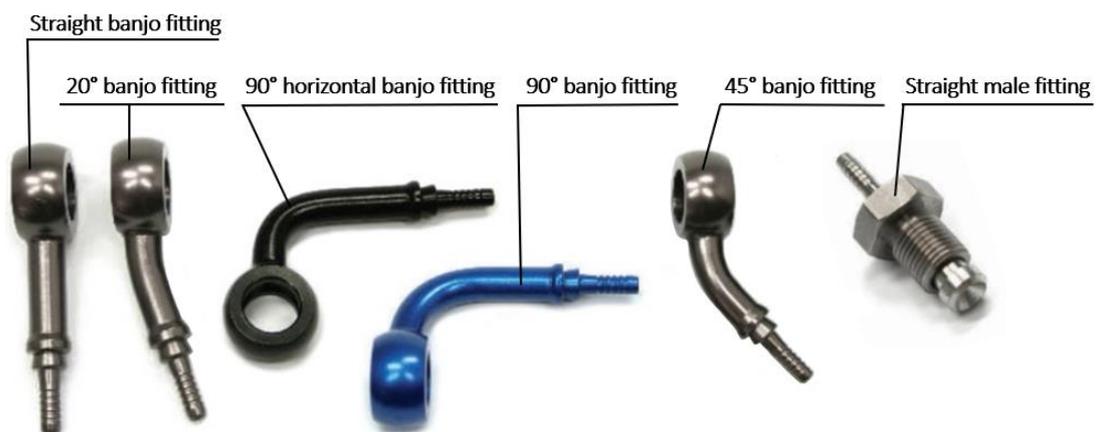


Figure 3.36 - Figure showing the fittings geometries chosen for the lines

LINE CODE	START FITTING	END FITTING	LENGTH
1	20° banjo fitting	20° banjo fitting	90 mm
2	20° banjo fitting	90° banjo fitting	160 mm
4	90° horizontal banjo fitting	45° banjo fitting	130 mm
5	Straight banjo fitting	Straight male fitting	1450 mm
6	90° horizontal banjo fitting	straight male fitting	180 mm

Table 3.11 - Table reporting the final specifications for each line to be ordered.

3.6 additional steps for in-vehicle mounting

To correctly integrate the EBS in the vehicle, some adjustments have to be done on the pedalbox layout: as stated in section 3.4.3, to save space and to keep low the number of components, the intensifiers have to use the same oil reservoirs used from the master cylinders. So, the reservoirs connections had to be changed in order to have two exit ports, so that the same component can feed both the master cylinders and the pressure multipliers. After having considered different solutions, the most convenient in terms of packaging (to not generate interferences between components and to guarantee an easy mounting operation), was to mount a t-joint below the reservoir. The reservoir height had to be increased to make the new connection fit in the system, so an extension was designed to be 3D printed and mounted between the support and the reservoir. However, it was verified that this increase in height was not generating interferences with other components (mainly, with the rotation of the accelerator pedal). Images showing the components inserted in the assembly are reported below in Figure 3.37.

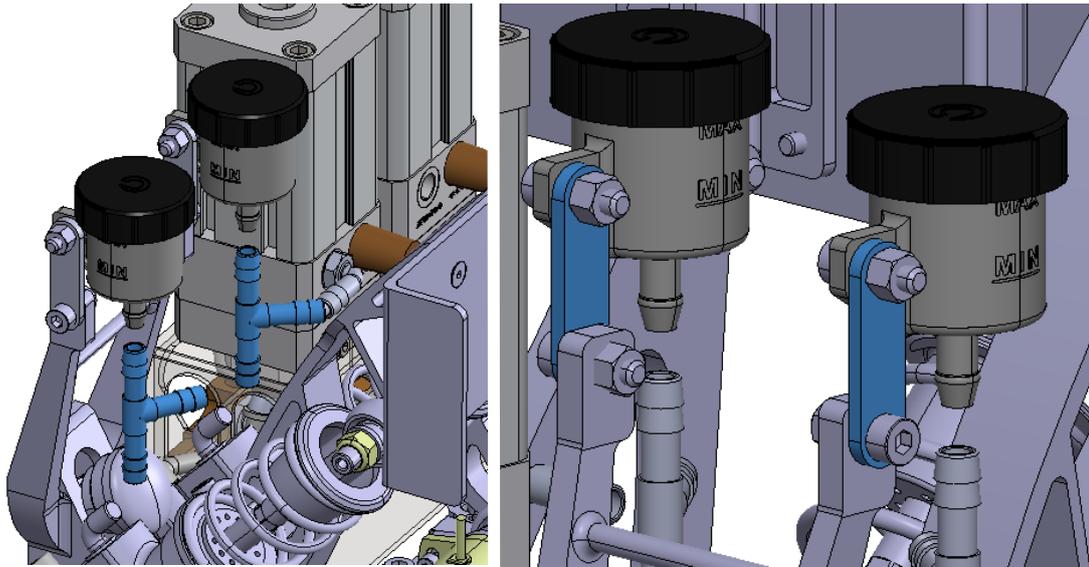


Figure 3.37 - Images underlining the position of the reservoir t-joints (on the left) and of the supports to increase the height (on the right)

Chapter 4

EBS testing and integration in the driverless vehicle

In this section, the phases of assembling, integration with other vehicle systems and functional testing of the designed EBS are presented. First, an overview about the different EBS states required by the rules is given, explaining which additional component are to be placed in the circuit and their role in the whole system functioning. Then, the different components of the system, presented in section 3.5 are mounted and bench-tested. Finally, the additional steps necessary in prevision of the future in-vehicle mounting of the system are described.

4.1 EBS circuit integration

4.1.1 Introduction on EBS and vehicle states

With reference to rule DV 2.4.6 reported in the Formula Student 2020 rulebook [11], the EBS can have only three possible states:

- **Unavailable:** the actuator is disconnected from the system or the energy storage is de-energized, so the emergency brake manoeuvre is not possible. In the designed system, this state is realised turning the manual valves at intermediate position, so that there is no pressure in pneumatic lines and EBS cannot be activated.
- **Armed:** able to initiate the emergency brake manoeuvre immediately if the SDC is opened or the LVS is interrupted. In the designed system this condition can be obtained turning the manual valve in the ON position: in this way, the EBS circuit is pressurized up to the solenoid valve, that can trigger the emergency brake whenever the tension supplied is interrupted.
- **Activated:** brakes are closed and power to EBS is cut. Brakes may only be released after performing manual steps. In the designed system, this state is realised pressurizing the hydraulic brake lines through the pressure

multipliers (commanded by the solenoid valves). After the actuation, brakes can be released by manually re-activating the solenoid valves (for example acting on the RES) and turning again the manual valves, allowing pressure discharge.

The different EBS states, as well as their correct identification, are fundamental for realising the transitions between the different vehicle states. In the following Figure 4.1 the different vehicle states and the operations that must be performed to allow the transitions between them, are reported.

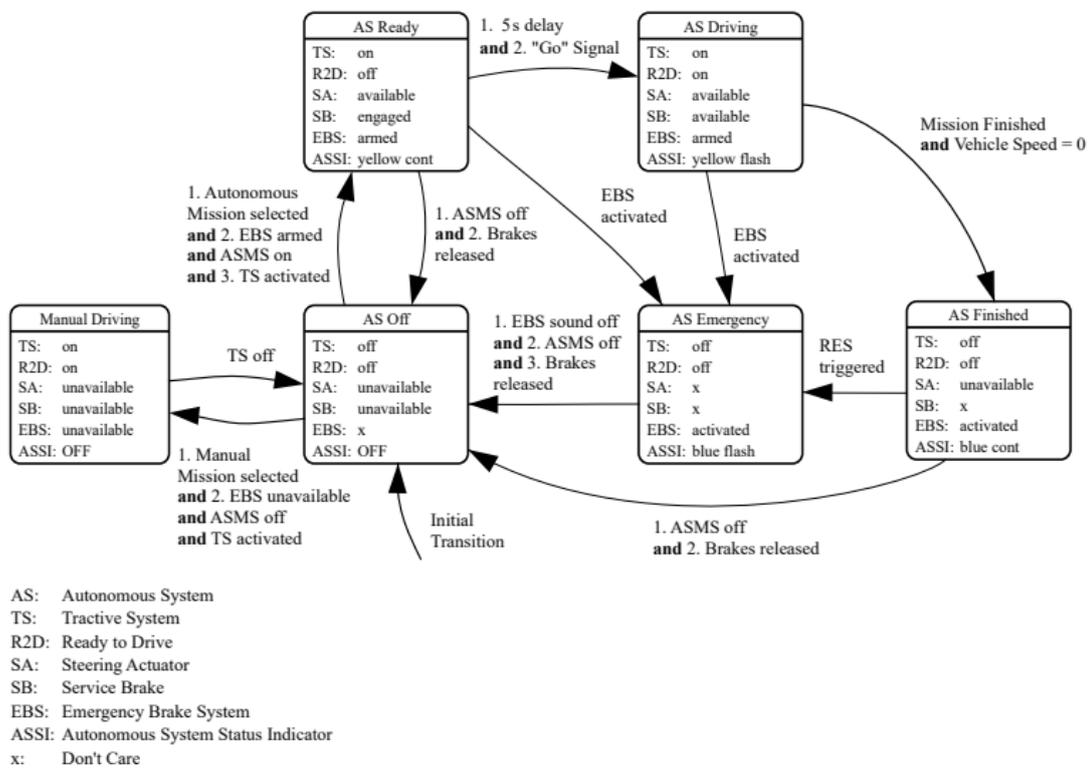


Figure 4.1 - Figure showing the different vehicle states and the conditions necessary to transition from one to another.

4.1.2 EBS circuit integration

As stated by rule DV 3.1.2, reported in section 3.1, the vehicle must be equipped with an EBS and with a EBS relay. The EBS can be triggered by the opening of the LVMS or of the ASMS, but also of the SDC. The latter, is a circuit that has to be closed in cases of manual driving (having verified that the autonomous system is OFF) and in autonomous driving when the autonomous mission is selected and there is sufficient brake pressure build up. It can be opened from the AS or from the RES, bringing the vehicle into the “AS Emergency” state and starting the emergency braking operation. A scheme of the SDC, taken from the rulebook, is shown in the figure below.

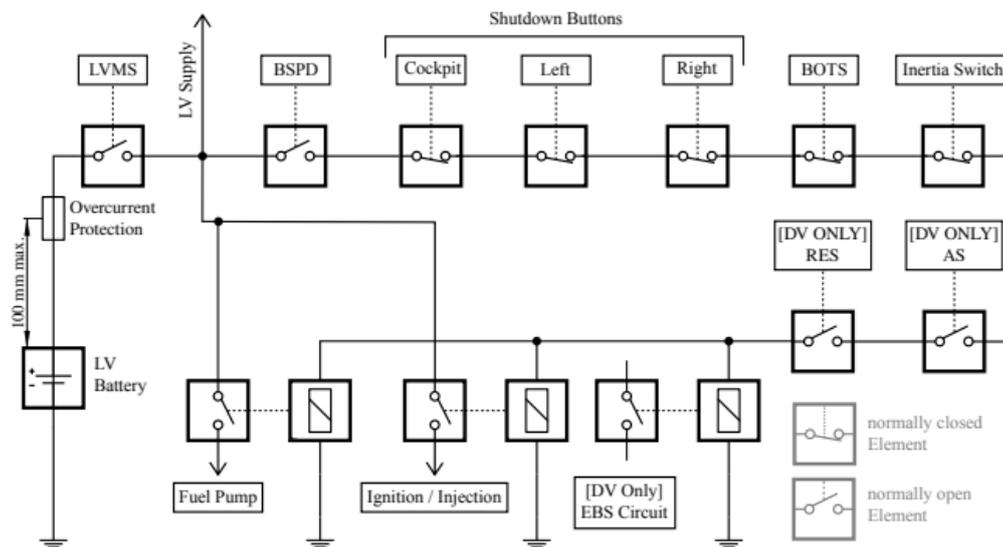


Figure 4.2 - Image showing the shut-down circuit scheme and its main components. *NB* As requested by the rules, all the circuits that are part of the SDC must be designed such that in the de-energized/disconnected state they open the shutdown circuit.

The EBS relay is instead to be supplied by the SDC but should act on the LV circuit. It is important that the system is designed to have the relays placed in parallel to the AIRs, so that when the SDC opens, the AIRs opening delay (requested by the rules) does not affect the relay operation [15].

In the designed system, the relays are used to pilot the 3/2 solenoid valves actuation in a passive way: the relay alimantation connections can be connected to the SDC of the vehicle, while the switching part to the LV circuit, just upstream the solenoid

valve. In this way, when the SDC is closed, the relay switch is also closed, guaranteeing the tension necessary to the electrovalve to remain closed. When instead the SDC is opened, it opens the switch in the relay, which in turn opens the solenoid valve triggering the EBS. Figure below shows the Finder relay selected to be mounted on the system, its basic electric scheme and main electrical parameters.

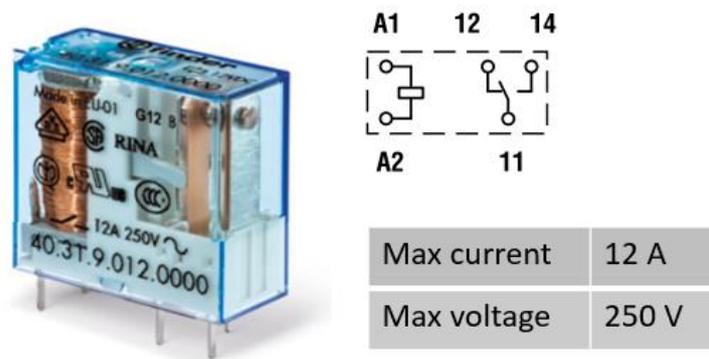


Figure 4.3 - Relay scheme and parameters

When considering the EBS integration with other vehicle systems, also rule DV 3.2.4 (reported in section 3.1) has to be taken into account. To allow the transition to the “AS Ready” state in fact, an initial check-up must be performed to ensure that the EBS and its redundancy are able to build the expected brake pressure. This means that the two circuits are to be tested not only together, but also singularly. To individually test the actuation paths and fulfil the rule, transistors like MOSFETs have to be inserted in the circuit.

MOSFETs (term standing for Metal Oxide Field Effect Transistor) are devices with three terminals: drain, source, and gate. They can be classified into Enhancement mode (E-MOSFET) or Depletion mode (D-MOSFET) basing on the construction characteristics. The main difference between these two types is that in E-MOSFETs the terminals are physically separated, while in D-MOSFETs they are connected [16]. A further distinction can then be done between N-channel and P-channel MOSFETs. The difference, in this case, is in the functioning logic: an N-channel MOSFET is open until the voltage provided at the gate is below the threshold value, while it closes when the gate voltage is higher than the threshold (which is a constructive characteristic of the device). In a P-channel instead, the functioning is the opposite:

the MOSFET is closed for low voltages and open for voltages above the threshold. For the designed system, N-channel enhancement MOSFETs were chosen to be implemented. Their role is to command each of the solenoid valves of the EBS circuit singularly, independently from the opening of the SDC, AS and LV system, but with an input coming from DSPACE, which is a control unit able to set different vehicle parameters also from remote. The kind of input provided by the DSPACE is a digital voltage signal, typically around the value of 5 V. For the majority of common MOSFETs, this value would not be sufficient to open the device, since it would result lower than the threshold voltage required by component. Two alternative strategies can be considered to deal with this issue:

- Using a driver to increase the voltage provided by the digital output of the controller [20].
- Using a logic level MOSFET. Logic level MOSFETs are designed to be able to fully turn on from the logic level of a microprocessor (hence, with a very low threshold voltage value).

Using a logic level MOSFET, the switching time of the device is slightly increased (due to the smaller currents flowing in the gate to charge the gate capacitance during the switching transients). However, since the system is not requiring a high actuation frequency the second solution was chosen, to have less components in the system: in this way, only two transistors are sufficient to drive the valves.

ZXMS6005DG 60V ENHANCEMENT MODE N-CHANNEL INTELLIFET™ MOSFET

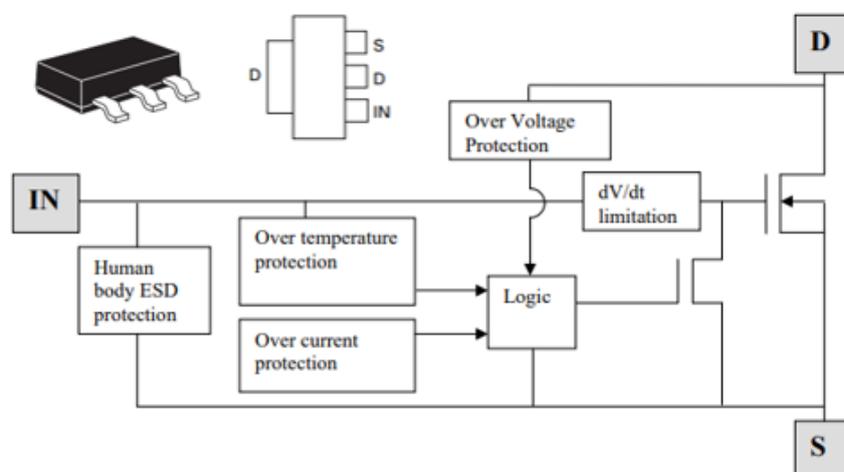


Figure 4.4 - Image showing the chosen MOSFET and its internal circuit

The chosen component to be placed in the circuit is shown in the figure above with its internal circuit. It is specifically designed to be driven from a microcontroller digital output and integrates over-temperature, over-current, over-voltage and electrostatic discharge (ESD) protections. Even if it was clearly specified in the component description, the MOSFET datasheet was considered to verify its ability to be driven from a digital controller output. In particular, two parameters are useful to understand if it is a logical level device: value of resistance when the component is ON (indicated as $R_{DS(ON)}$) and threshold voltage V_{TH} . The resistance parameter is in fact specified for low voltage values (generally for $V_{IN} = 5\text{ V}$, and in this case also for $V_{IN} = 3\text{ V}$), while the threshold voltage for device switching is very low, between 0.7 and 1.5 V, indicating that is effectively a logic level MOSFET. The complete datasheet of the device is reported in the figure below.

ELECTRICAL CHARACTERISTICS (at $T_{amb} = 25^{\circ}\text{C}$ unless otherwise stated).

PARAMETER	SYMBOL	MIN	TYP	MAX	UNIT	CONDITIONS
Static Characteristics						
Drain-Source Clamp Voltage	$V_{DS(AZ)}$	60	65	70	V	$I_D=10\text{mA}$
Off state Drain Current	I_{DSS}			1	μA	$V_{DS}=12\text{V}, V_{IN}=0\text{V}$
Off state Drain Current	I_{DSS}			2	μA	$V_{DS}=36\text{V}, V_{IN}=0\text{V}$
Input Threshold Voltage	$V_{IN(th)}$	0.7	1	1.5	V	$V_{DS}=V_{GS}, I_D=1\text{mA}$
Input Current	I_{IN}		60	100	μA	$V_{IN}=+3\text{V}$
Input Current	I_{IN}		120	200	μA	$V_{IN}=+5\text{V}$
Input Current while over temperature active				300	μA	$V_{IN}=+5\text{V}$
Static Drain-Source On-State Resistance	$R_{DS(on)}$		170	250	$\text{m}\Omega$	$V_{IN}=+3\text{V}, I_D=1\text{A}$
Static Drain-Source On-State Resistance	$R_{DS(on)}$		150	200	$\text{m}\Omega$	$V_{IN}=+5\text{V}, I_D=1\text{A}$
Continuous Drain Current (a)	I_D	1.4			A	$V_{IN}=3\text{V}; T_A=25^{\circ}\text{C}$
Continuous Drain Current (a)	I_D	1.6			A	$V_{IN}=5\text{V}; T_A=25^{\circ}\text{C}$
Continuous Drain Current (b)	I_D	1.9			A	$V_{IN}=3\text{V}; T_A=25^{\circ}\text{C}$
Continuous Drain Current (b)	I_D	2.0			A	$V_{IN}=5\text{V}; T_A=25^{\circ}\text{C}$
Current Limit (d)	$I_{D(LIM)}$	2.2	5		A	$V_{IN}=+3\text{V}$
Current Limit (d)	$I_{D(LIM)}$	3.3	7		A	$V_{IN}=+5\text{V}$
Dynamic Characteristics						
Turn On Delay Time	$t_{d(on)}$		6		μs	$V_{DD}=12\text{V}, I_D=1\text{A}, V_{GS}=5\text{V}$
Rise time	t_r		14		μs	
Turn Off Delay Time	$t_{d(off)}$		34		μs	
Fall Time	t_f		19		μs	

Table 4.1 - Datasheet showing the electrical characteristics of the MOSFET.

From the datasheet, it can be noticed that the maximum Drain-Source Voltage V_{DS} that the device can reach is much higher than the 12 V source at which the load (and therefore the drain port of the MOSFET) will be connected. Also, the drain current limit $I_{D(LIM)}$ is compatible with the solenoid valve operation: since the valve requires a power of 3 W and a tension of 12 V to work, the necessary current was evaluated (as the ratio between power and voltage) to be 0.250 A, much lower than the limit.

In the following Figure 4.5, the driving circuit to be realised for the device is shown. IN is the digital input coming from DSPACE that is driving the MOSFET gate port, while the load to be driven is the 3/2 solenoid valve, which is connected to drain port

of the device and to the LV system battery (hence to a 12 V DC source). The source pin is instead to be grounded.

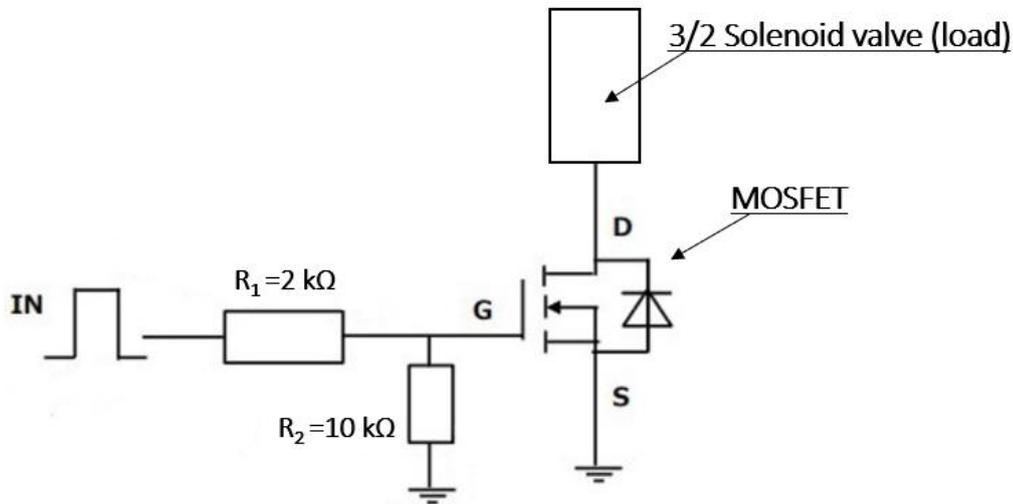


Figure 4.5 -Image showing the driving circuit of the MOSFET.

In addition, two resistances are placed in the circuit: a gate resistance R_1 and a pull-down resistance R_2 . The gate resistance has the main purpose of limiting the current peaks that the DSPACE output has to supply to the gate. The approximation that the gate current of a MOSFET can be considered approximately zero is valid only when the device is in a non-transient phase: in fact, when the MOSFET is switching, a current is needed to charge (or discharge) the gate capacitance, and only when this transient phase is completed the gate current reaches zero. Larger is the gate current, faster is the voltage change and therefore faster will be the device switching. Since the digital output of DSPACE can provide only a limited current value (around 5 mA), the gate resistance can be useful to guarantee that the gate current is kept to a value low and sustainable from the digital output. Its value was evaluated starting from the voltage provided by the DSPACE (5 V) and the value of maximum current that it was established to not overcome (2.5 mA out of 5), as

$$R_1 = \frac{V_{high}}{I_{max}} = \frac{5 V}{2.5 mA} = 2 k\Omega$$

The disadvantage of this operation can be related to the higher switching times that inserting this resistance is involving, but unless the device needs to be operated with

very high switching frequencies (and this, as stated before, is not the case), the magnitude of this time increase is not affecting the performances of the system.

Then, since the threshold voltage is very low, it could happen that even small tensions are sufficient to accidentally trigger the device (and therefore the EBS system). In addition, because of its internal oxide layer, the MOSFET is characterized by an internal capacitance that can oppose to the switch off, that could be problematic and could take time. Therefore, to avoid unintended system actuations and to quickly drain the residual internal capacitance when the device is switched off, a pull-down resistor R_2 has to be placed between the gate port and ground. A value of 10 k Ω was considered appropriate for this resistance. The two resistances are then placed as shown in Figure 4.5 to avoid having a voltage divider before the gate.

In Figure 4.6, an example taken from the FSG EBS reference guide [15] is presented, showing how the components necessary to integrate the EBS with the vehicle subsystems are placed.

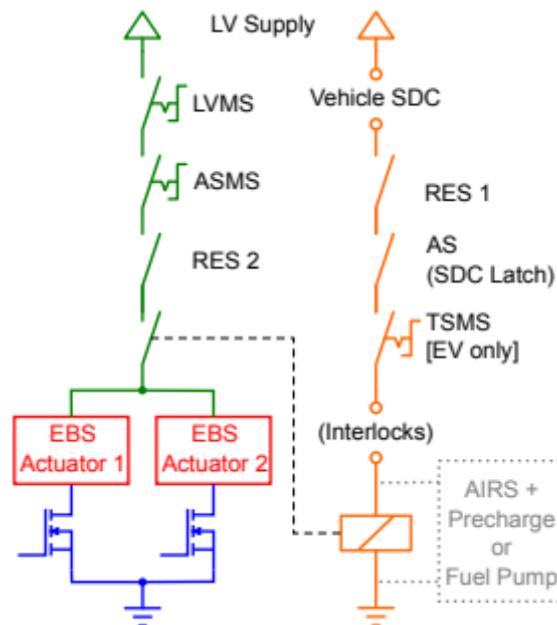


Figure 4.6 - Image showing an example of components integration.

The EBS relay, placed in parallel to the AIRs, can be noticed: it is connected to the SDC (in orange), but the switching part is acting on the LV circuit (in green) just upstream the EBS actuators (which, in the logic of the designed system, can be considered as the solenoid valves). It is basically the same application in the realised

system, with the only difference that two relays are present, one for each actuator. In blue, also the MOSFETs are shown, one for each actuator, used to independently control them. In the figure below an overview of the complete system is shown, reporting also the main control paths.

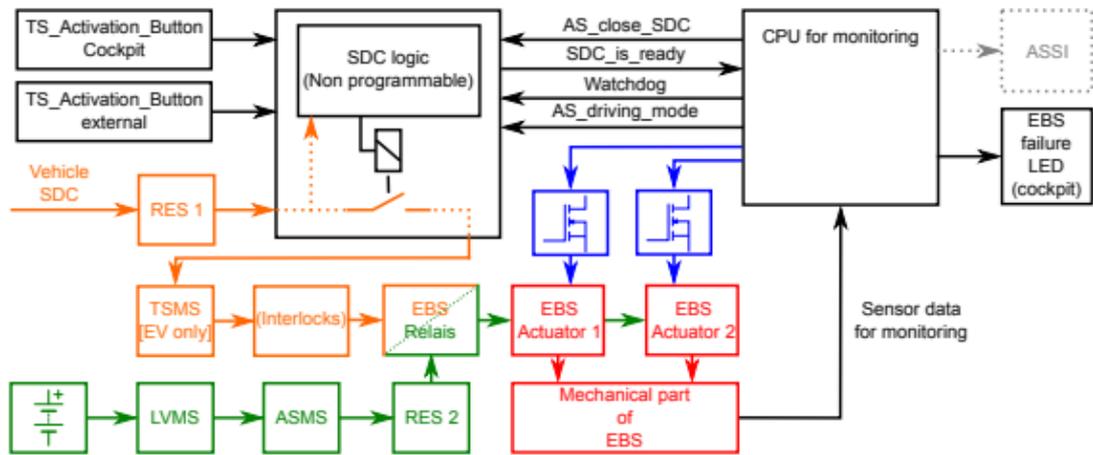


Figure 4.7 - Image showing the main system control paths.

4.1.3 EBS check-up sequence

In order to correctly perform the above mentioned initial EBS check-up sequence, a MATLAB code was written to be implemented into the complete state machine environment. The state machine is a system realised in the MATLAB Stateflow workspace that simulates all the vehicle states (reported in Figure 4.1) and in which all the necessary functions, parameters and control logics are implemented to efficiently manage the different states and the passages between them.

The check-up sequence must take place anytime that the system has to pass from the “AS OFF” to the “AS Ready” state (so, at the beginning of a mission), and has to be performed with a sequence of operations, which are described below:

1. **Both lines check:** both the MOSFETs are switched OFF (providing a low signal from the DSPACE output) and the pressure from the analog sensors placed in the hydraulic lines is evaluated. The check is considered successful if the average pressure coming from the two sensors is above 38 bar (meaning that both intensifiers are functioning providing around 38 bar each)
2. **Front line check:** only the MOSFET in charge to command the rear line is turned ON (closing the electrovalve), while the front one is left OFF. In this case, the check is considered successful if the pressure value read from the sensor in the front line is above 38 bar.
3. **Rear line check:** the MOSFET commanding the front line is switched ON, while the one controlling the rear EBS line is turned OFF. Similarly to the previous phase, the check is successful if the pressure measured from the analog sensor in the rear hydraulic lines is above 38 bar.

The operations reported above are to be performed in sequence, so if one of them is not successful, the sequence is interrupted and the whole passage to the “AS Ready” state is failed. Basing on this logic, the check-up function was written and implemented in the state machine environment. The obtained results, in term of average brake pressure (evaluated simply by summing of front and rear pressure and dividing by 2) are reported in the Figure 4.8 below.

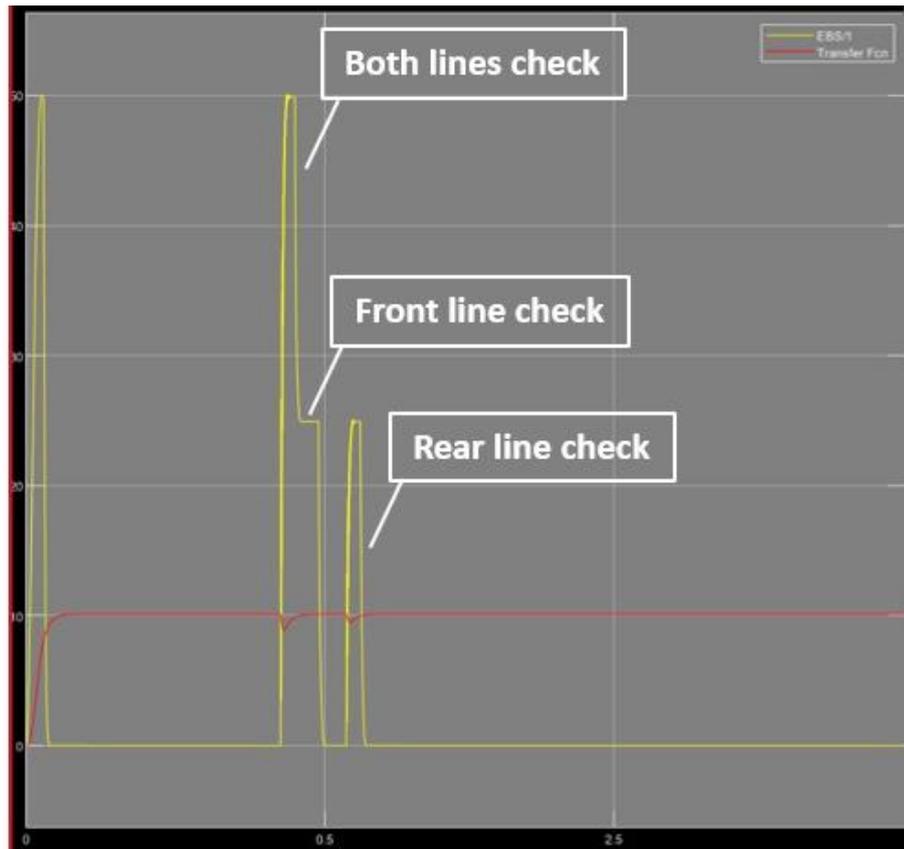


Figure 4.8 - Plot showing the results of the EBS check in terms of average brake pressure

In yellow, the average brake pressure is reported. The check-up sequence resulted to be successful: when both EBS lines are working, a pressure of around 40 bar is reached in each line. When instead only one line is triggered, the average pressure resulted to be slightly above 20 bar (meaning that the single line is able to reach a 40 bar pressure as expected). The initial pressure spike that appears on the graph is due to the fact that in the initial instants the manual valves are in the ON position but the ASMS is still open, so for some milliseconds the EBS is actuated. When the ASMS is closed the solenoid valves switch to the closed position and the brakes are released (average pressure is equal to zero). Then, when also the HV system is activated, the check-up sequence described above can start. In red, the air pressure in the hydraulic line is reported.

Having validated the EBS check-up sequence, another test was conducted connecting also the RES to the state machine, to check if the system performances are in line with the design and with the rules when simulating an entire driving mission. The plot

showing the behaviours of the average brake pressure in the hydraulic lines and of the air pressure in the pneumatic lines during the full mission is reported below.

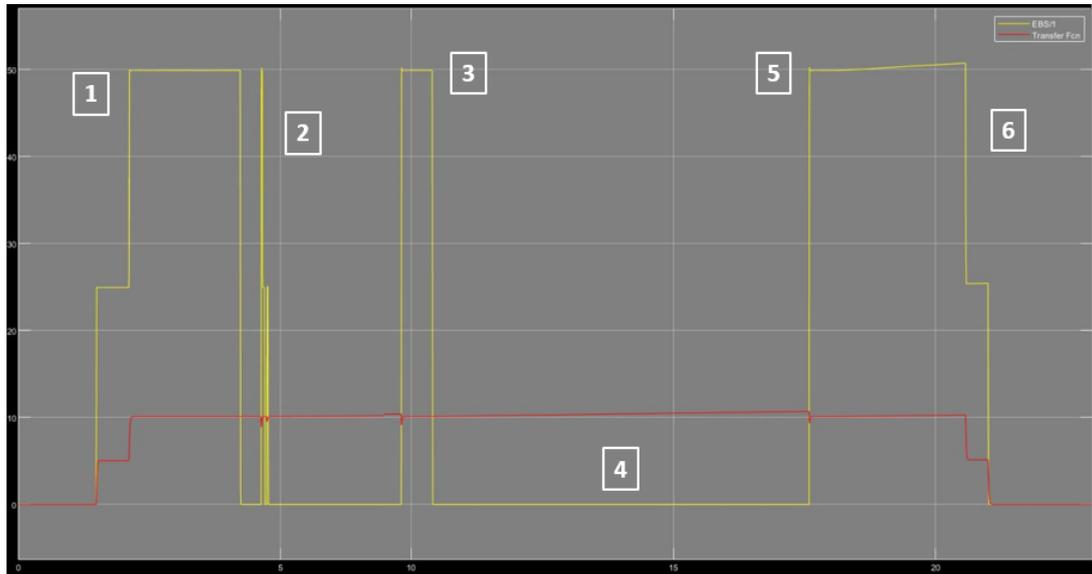


Figure 4.9 - Plot showing the behaviour of the brake and air pressures over the entire simulation.

With reference to the previous Figure 4.9, different phases can be underlined:

1. **EBS arming.** In this phase, the manual valves are turned in the ON position, arming the EBS. The brake pressure increases because the ASMS of the vehicle is open, so the EBS is actuated. When the ASMS is closed, as mentioned above, the EBS is released and the pressure drops to zero.
2. **EBS check-up sequence.** Once also the TSMS is closed, the check up sequence, described above, can start to check if the EBS and its redundancy are able to build the expected brake pressure (as stated in rule DV 3.2.4.). Once the test is considered successful, the vehicle can complete the transition into the AS Ready" state.
3. **Ready to Drive (R2D) mode.** To successfully complete the transition from "AS Ready" to "AS Driving" state, the R2D (Ready to Drive) mode must be on. As specified by rule EV 4.11.6, the transition to the ready-to-drive mode is possible only if mechanical brakes are actuated. Therefore, the EBS has to be triggered, and the pressure in the lines raises again.

4. **Autonomous mission.** During the autonomous mission, brake pressure is zero and the EBS is in the armed state, ready to pressurize the hydraulic lines if triggered from the solenoid valves.
5. **EBS actuation.** At the end of the autonomous mission, or if during the mission a problem is encountered, the EBS has to be activated, bringing the vehicle to the “AS Finished” and “AS Emergency” states, respectively.
6. **EBS Release.** After EBS actuation, manual operations are to be performed to release the brake pressure. In this case, as it is possible to see from the steps in the pressure decrease plots of both brake pressure and air pressure, the release phase ended with the opening of the manual valves, discharging pressure into ambient.

Considering what is stated above, the simulation can be considered as successful, with all the system components performing in the correct way.

4.2 System assembly and bench testing

Before installing the system into the vehicle, tests were carried both on single components and on the assembled system, in order to verify the functionality of the designed parts, lines, and connections. The first step was to mount the components to form the main system subassemblies, keeping in mind also how these units are to be connected and making, when necessary, changes and adjustments in geometry and in the positioning of some parts to facilitate the mounting operations.



Figure 4.10 - Images showing the mounting of OR valves on the support

In the previous images, the mounting of OR valves on their support is shown. In this case, some changes were necessary respect to initial design in order to allow an easier system mounting and a better functionality. First, one valve had to be moved (respect to its initial position shown in Figure 3.13b) on the outside of the same surface on which the other valve is mounted. So, the two valves are now placed on the inner and outer sides respectively of the front surface of the support (the one pointing towards the front of the vehicle), exploiting the same holes for mounting. This was done to allow an easier design of the hydraulic lines, that in this way can reach the valve following a less intricated path. In addition, to permit the mounting of the banjo fittings of the hydraulic lines (having an external diameter of 18 mm) on the valves ports, additional small supports were necessary to increase the space between the valves and the surface on which they are mounted. For the valve placed on the outer side, an additional thickness of 3 mm was sufficient, while for the one placed on the inner side, 6 mm were necessary to avoid interferences between hydraulic lines and the side surface. A detail of how these additional thicknesses (which were 3D printed in PLA) are placed is shown in the 4th image of Figure 4.10. For the mounting of the other components instead, no huge changes were necessary respect to the initial design. In the following images, pictures of the mounted system subassemblies are shown.

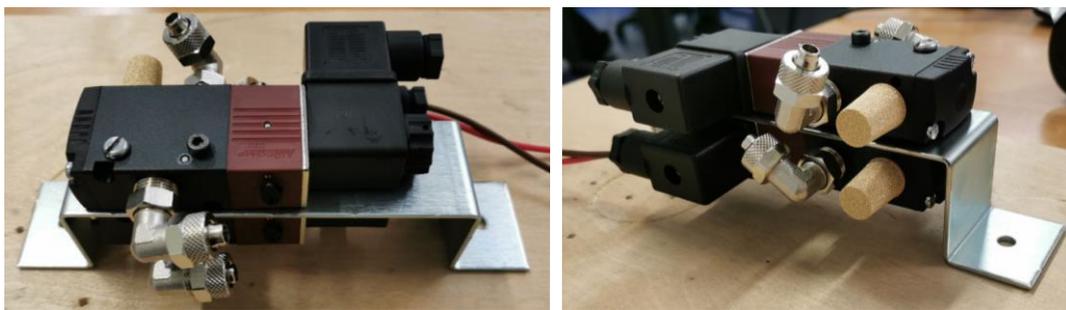


Figure 4.11 - Figures showing the 3/2 solenoid valves subassembly.



Figure 4.12 - Figures showing the manual valves subassembly (on the left) and the canister subassembly (on the right)

The electrovalves subassembly, and the unit mounting intensifiers and OR valves are then fixed to the support simulating the SC19 pedalbox environment and connected between them (and with the other system subassemblies) through the pneumatic lines, in order to be tested. Intensifiers were previously singularly tested by the company that manufactured them, hence their capability to build up a hydraulic pressure of 40 bar when receiving an input of 10 bar pressurized air was already successfully verified. The aim of the test is therefore to verify the functionality of the pneumatic lines and of all the related components, from the HP canisters up to the pressure multipliers. The actuation logic has also to be tested inserting in the circuit the EBS relays and connecting the system to the RES through a power supply, that simulates the LV battery of the vehicle. The layout used for testing is showed in the following images. Since, at this stage, intensifiers are still not connected to the hydraulic lines, a screw was mounted on the output port of the component to block eventual leakages.

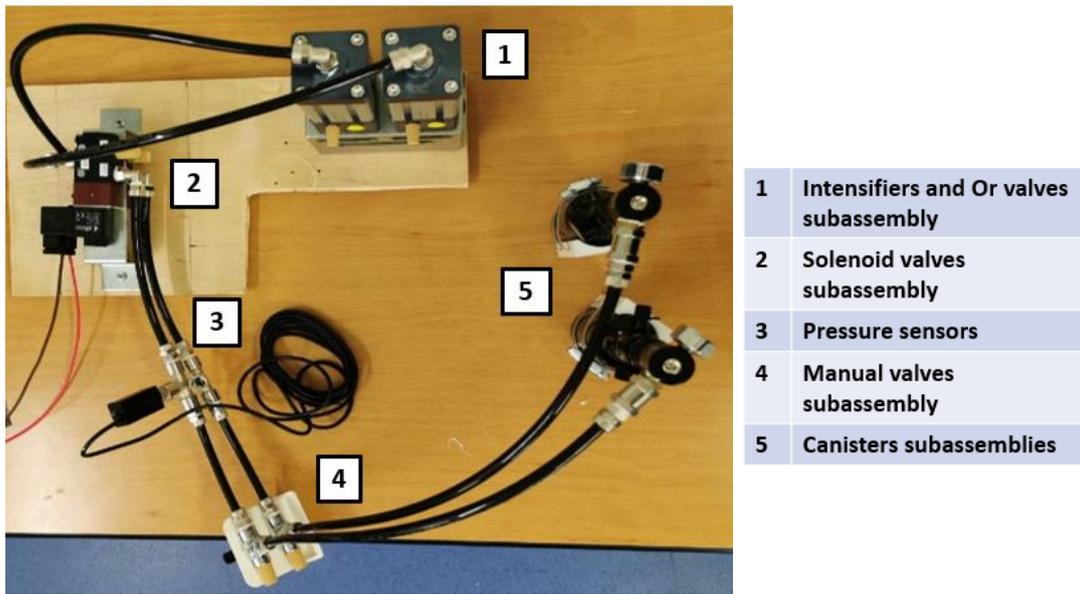


Figure 4.13 - Image showing the tested layout, with main components underlined

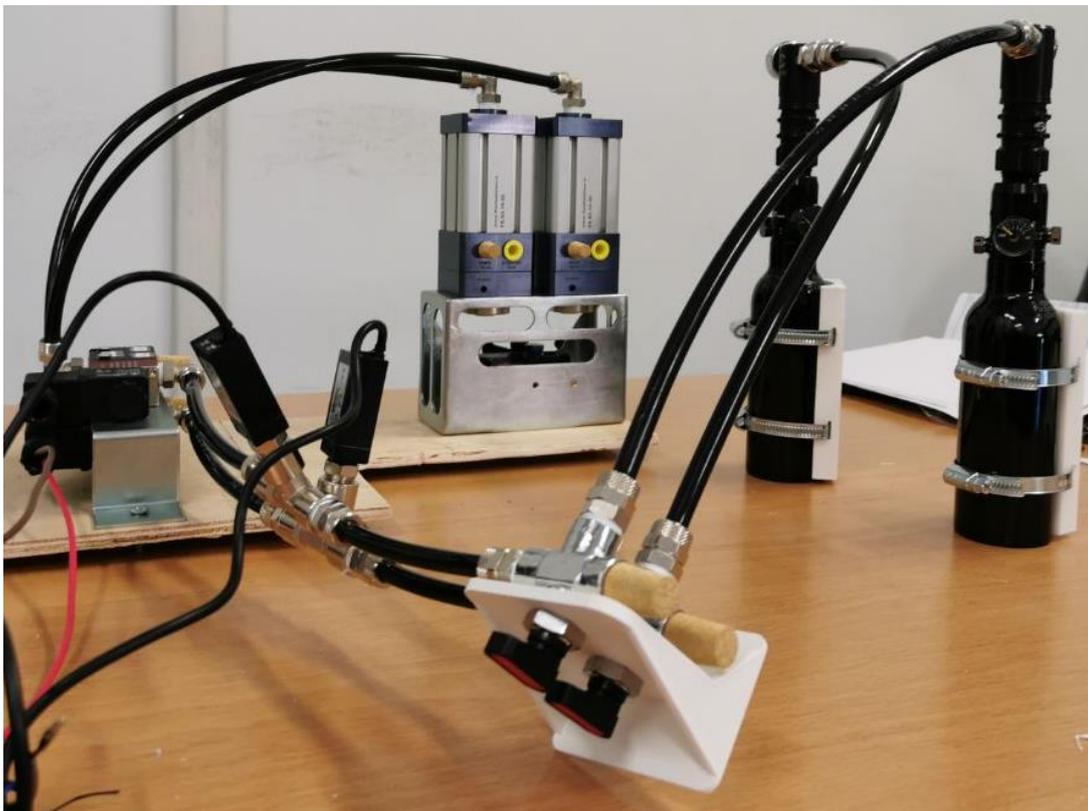


Figure 4.14a - Image showing the full tested system layout

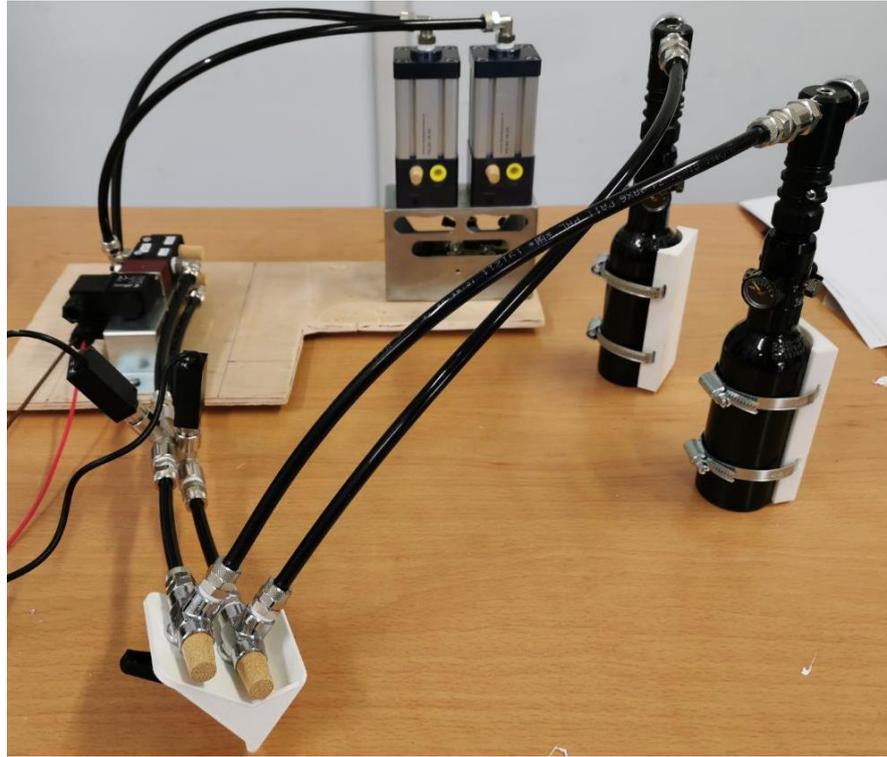


Figure 4.14b - Image showing the full tested system layout

Once having mounted the system, it was necessary to connect it to the power supply and to the RES to verify its correct functioning. The scheme of the electrical connections is reported in Figure 4.15 below. The Remote Emergency System, as mentioned in section 4.1.2, has the role of opening the vehicle SDC. In order to connect it to the system, four ports have to be considered: two are for alimentation and have to be connected to the poles of the power supply. The other two instead, can be seen as if connected through an internal commanded switch able to cut the tension across the EBS relay (to which it is connected), generating the opening of its switching part and consequently opening the electrovalve, triggering the EBS.

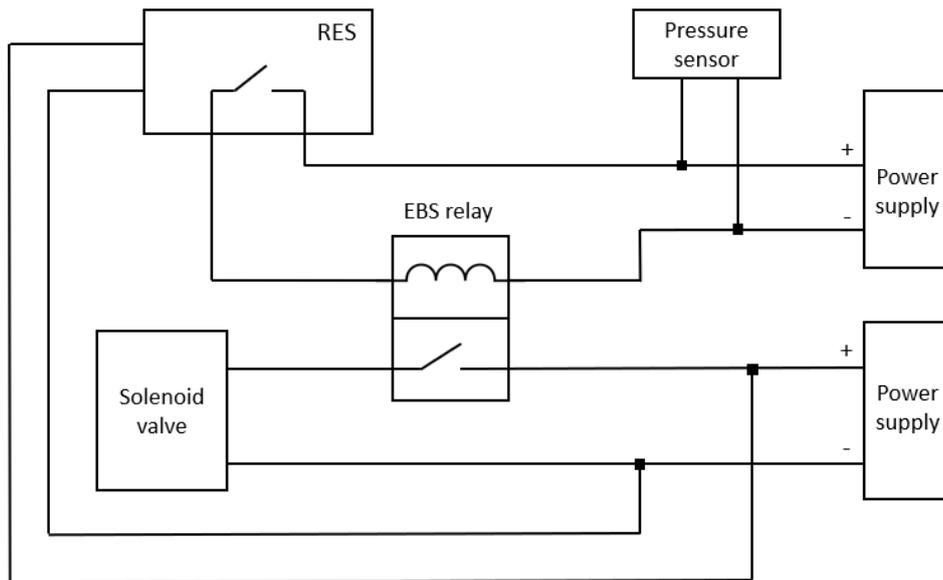


Figure 4.15 - Scheme of the electrical connections realised for the system testing

The pressure sensor is also mounted on the circuit and connected to the power supply in parallel to the EBS relay alimentation part. Even if the pressure sensor has a LED indicator, which lights up if the pressure is above the threshold value (that has to be set manually), the digital output port of the device was connected to an oscilloscope to verify if the output signal (provided in terms of voltage) is high or low: 0 V stands for low signal, indicating that the pressure value read from the line is below the threshold, while 5 V are provided as high signal, meaning that the pressure in the line is above the set threshold. The threshold was set directly connecting the component to a pressure source of known level, and then adjusting the regulation screw until the LED indicator switched off (or, alternatively, when the voltage reported on the oscilloscope dropped to zero): at this point, the threshold is set at the pressure level of the tank.

Once that also the electrical connections were set, the power supply is turned on and the system is tested. At the beginning, the EBS is in the “unavailable” state, in the condition for manual driving, with manual valve at intermediate position and pressure sensor detecting no pressure in the line (hence, with LED indicator OFF and sending a low output signal). It must be remarked that at this moment, the electrovalve is switched open, since the electric connection alone is not sufficient to bring the component to its closed position, but also a certain pressure level is needed

(as reported in section 3.5.4). First, the manual valves are moved from the intermediate to the ON position, putting the EBS into the “armed” state. In this way, they are pressurizing the pneumatic circuit up to the solenoid valves, that, receiving a sufficient pressure input and being connected to the power supply, immediately switch to the closed position, blocking the passage of air. The correct pressure build up in the line can be checked from the pressure sensor, that at this point has to light on the LED indicator and send to the oscilloscope the high signal value, meaning that pressure is above the set threshold. Then, the system actuation is simulated: the RES button is pressed, opening the connection between the “actuation ports” (with reference to figure 4.15 and starting the sequence of actions leading to EBS actuation (opening of the relay, opening of the valve, intensifiers actuation).

After the system actuation, the RES button was released, restoring the connections and therefore closing the valve, simulating the release phase.

The system fully behaved as expected: the functioning of the EBS relay, of the pressure sensor and of the 3/2 solenoid valve was in line with the designed logic and also the intensifier proved to be functional, emitting a clear actuation noise. The entire sequence was then repeated multiple times to validate the test, and after having verified that the performance of the system was the same, the test was considered successful.

Conclusions

Considering the results obtained from bench-testing, the system appeared to be fully functional, compliant with the rules and in line with the design requirements. The whole EBS design can be therefore considered as successful. To fully validate it, however, some additional steps are needed: first, it is necessary to complete the system assembly inserting the hydraulic lines. Then, the EBS must be mounted on the vehicle and connected with the pedal assembly, and an integration between the designed hydraulic lines and the ones which are already present has to be performed. Having already assessed the capability of the system to build the designed brake pressure and to respect the actuation logic as requested by the rules, components in charge of controlling the system (solenoid valves, relays, MOSFETs, RES) must be now be properly connected to the LV circuit and to the SDC of the vehicle so that they are able to efficiently interact between them and with the other units and systems mounted on the vehicle. Tests are then to be done on the full vehicle in order to verify that all the components are efficiently integrated and that EBS is able to actuate the brakes as expected.

References

1. *Road traffic death rate by WHO region and income level*, Geneva, Switzerland: World Health Org. Available at:
http://www.who.int/gho/road_safety/mortality/traffic_deaths_rates/en, 2013.
2. Young K., Regan M., Hammer M, *Driver distraction: a review of the literature*, Australasian College of Road Safety, Sydney, 2007.
3. Dasom Lee, David J. Hess, *Regulations for on-road testing of connected and automated vehicles: Assessing the potential for global safety harmonization*, in *Transportation Research Part A: Policy and Practice*, Elsevier, 2020. Available at <https://www.sciencedirect.com/science/article/pii/S0965856419308006>.
4. J3016, *Taxonomy and Definitions for Terms Related to Driving Automation Systems for On-Road Motor Vehicles*, SAE International, 2018.
5. Tien James M., *The Sputnik of servogoods: Autonomous vehicles*, *Journal of Systems Science and Systems Engineering*, 2017. Available at https://www.researchgate.net/publication/312098071_The_Sputnik_of_servogoods_Autonomous_vehicles.
6. Formula Student Germany website, online source, available at <https://www.formulastudent.de/about/concept/>.
7. Genta G., Morello L., *The Automotive Chassis Vol.2*, Springer, 2009.
8. Chau K.T., *Alternative Fuels and Advanced Vehicle Technologies for Improved Environmental Performance*, Woodhead Publishing Limited, 2014.
9. Motes J., *Functions of a Master Cylinder*, online source, available at <https://itstillruns.com/functions-master-cylinder-7491649.html>.
10. Subhash M., Sagar J., *Balance Bar Design and Motion Analysis of Pushrod*, in *Global Journal of Researches in Engineering*, Vol.14, Issue 1, 2014.
11. *Formula Student Rules 2020*, online source. Available at https://www.formulastudent.de/fileadmin/user_upload/all/2020/rules/FS-Rules_2020_V1.0.pdf.

12. Online source, available at [https://tameson.com/32-way-pneumatic-valve.html#:~:text=The%20two%20states%20of%20the,exhaust%20\(R%2C%2003\).&text=Bi%2Dstable%20pneumatic%20solenoid%20valves,position%20and%20are%20pulse%20operated.](https://tameson.com/32-way-pneumatic-valve.html#:~:text=The%20two%20states%20of%20the,exhaust%20(R%2C%2003).&text=Bi%2Dstable%20pneumatic%20solenoid%20valves,position%20and%20are%20pulse%20operated.)
13. Zubair M., *Design of Emergency Brake System for Formula student Driverless car*, Master thesis, 2019.
14. Online source, available at <https://instrumentationtools.com/hydraulic-actuators/#:~:text=Hydraulic%20actuators%20use%20liquid%20pressure,fluid%20pressure%20into%20mechanical%20force.>
15. R.K. Bansal, *A Textbook of Fluid Mechanics and Hydraulic Machines*, Laxmi Publications, 2004.
16. Online source, available at <http://enginemechanics.tpub.com/14105/css/Tandem-Cylinders-157.htm>
17. Belforte G., *Pneumatica, principi di base, strumentazione, componenti, tecniche digitali e circuiti, impianti, applicazioni*, Tecniche Nuove, 1987.
18. Stollberger M., Gebhardt M., *EBS Reference Guide 2019*, Formula Student Germany, 2018.
19. Online source, available at <https://www.amu.ac.in/emp/studym/100007851.pdf>
20. *MOSFET Gate Drive Circuit*, Application note, Toshiba Electronic Devices, 2018. Available at <https://toshiba.semicon-storage.com/info/docget.jsp?did=59460#:~:text=The%20basic%20requirements%20for%20a,for%20an%20N%2Dchannel%20MOSFET.>

Appendix A - Reference Rules

In this first appendix, the sections of the Formula Student 2020 Rulebook concerning the Emergency Brake System and the related inspection tests are reported.

DV 3 EMERGENCY BRAKE SYSTEM (EBS)

DV 3.1 Technical Requirements

- DV 3.1.1. All specifications of the brake system from T 6 remain valid.
- DV 3.1.2. 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).
- DV 3.1.3. The EBS must only use passive systems with mechanical energy storage. Electrical power loss at EBS must lead to a direct emergency brake maneuver (keep in mind T 11.3.1!).
- DV 3.1.4. The EBS may be part of the hydraulic brake system. For all components of pneumatic and hydraulic EBS actuation not covered by T 6, T 9 is applied.
- DV 3.1.5. When the EBS is part of the hydraulic brake system, the manual brake actuation (by brake pedal) may be deactivated for autonomous driving.
- DV 3.1.6. 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.
- DV 3.1.7. 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 100 mm length (shaft width of 20 mm) with "EBS release" in white letters on it.

- DV 3.1.8. 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.

DV 3.2 Functional Safety

- DV 3.2.1. 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.
- DV 3.2.2. The safe state is the vehicle at a standstill, brakes engaged to prevent the vehicle from rolling, and an open SDC.
- DV 3.2.3. To get to the safe state, the vehicle must perform an autonomous brake manoeuvre described in section DV 3.3 and IN 6.3.
- DV 3.2.4. An initial check has to be performed to ensure that EBS and its redundancy is able to build up brake pressure as expected, before AS transitions to "AS Ready".
- DV 3.2.5. The tractive system is not considered to be a brake system.
- DV 3.2.6. The service brake system may be used as redundancy if two-way monitoring is ensured.
- DV 3.2.7. 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.

DV 3.3 EBS Performance

- DV 3.3.1. The system reaction time (the time between entering the triggered state and the start of the deceleration) must not exceed 200 ms.
- DV 3.3.2. The average deceleration must be greater than 8 m/s^2 under dry track conditions.
- DV 3.3.3. 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.

- DV 3.3.4. The performance of the system will be tested at technical inspection, see IN 6.3.

IN 6.3 Driverless Inspection EBS Test

- IN 6.3.1. The EBS performance will be tested dynamically and must demonstrate the performance described in DV 3.3.
- IN 6.3.2. The test will be performed in a straight line marked with cones similar to acceleration.
- IN 6.3.3. During the brake test, the vehicle must accelerate in autonomous mode up to at least 40 km/h within 20 m. From the point where the RES is triggered, the vehicle must come to a safe stop within a maximum distance of 10 m.
- IN 6.3.4. In case of wet track conditions, the stopping distance will be scaled by the officials dependent on the friction level of the track.

Appendix B - MATLAB script for EBS check-up sequence

In this appendix, the MATLAB code mentioned in section 4.1.3, written in order to perform the EBS check-up sequence and to be implemented in the full state machine model, is reported. It was written using the persistent ack variable: basing on the ack value the different checks (for both lines, only front line and only rear line) can be performed in sequence, and if one of them fails, the sequence is interrupted, resulting in the failure of the transition between AS OFF and AS Ready modes.

```
function [EBS_result , MOSfront_comm , MOSrear_comm]=
fcn(EBScheck , p_brakes_front , p_brakes_rear)

persistent ack
if isempty(ack)
    ack = 0;
end

if EBScheck == 1
    MOSfront_comm=1;
    MOSrear_comm=1;
else
    MOSfront_comm=0;
    MOSrear_comm=0;
    EBS_result=0;
end

if (p_brakes_front + p_brakes_rear) >= 80 && EBScheck == 1
    ack = 1;
end

if ack == 1
    MOSfront_comm=1;
    MOSrear_comm=0;
end

if p_brakes_front >= 40 && p_brakes_rear <= 1.5 && EBScheck ==
1
    ack = 2;
end

if ack == 2
    MOSfront_comm=0;
    MOSrear_comm=1;
end

if p_brakes_rear >= 40 && p_brakes_front <= 1.5 && EBScheck ==
1
    ack = 3;
end
```

```
if ack ==3
    MOSfront_comm=0;
    MOSrear_comm=0;
    EBS_result=1;
else
    EBS_result=0;
end
end
```