

ASURT Formula Student Brake Design

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ABSTRACT

The Braking System is the most crucial part of the racing vehicle. There is no doubt, that if only one minority failure in the braking system took place, this would be more than enough reason to cause the racing team disqualification from the competition. Time is the main and the most important criteria for any racing competition; on the other hand the formula student "FS UK SAE" competition care the most about developing the automotive engineering sense in the students by putting them under strict rules normally taken from the original version "formula 1" to encourage their creativity to reach the optimum performance under these strict rules. One of the most important rules is "No Braking by wire", and the obvious consequences are more stopping distance and time. Braking distance is a critical facture in achieving racing success in a competitive domain. This report will cover using the bias bar, dynamic weight distribution "before and after braking" and carefully choosing the braking and suspension system components dimensions, in order to fulfill the main functions of "ABS and EBD" which are preventing the wheels from lock-up and preventing side skid of the vehicle during cornering in the different dynamic tests with full consideration of the maximum approachable deceleration of the vehicle without locking up without using any kind of electronic "actuators or control". Mathematical model "Matlab" and Physical model "AME SIM" will be used to support the report's results.

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INTRODUCTION

ASURT 14 is using 2 tandem master cylinders, 1 for the front axle and 1 for the rear axle, in order to be able to construct 4 hydraulic brake circuits to increase the safety measurements on the vehicle. In case of brake circuit failure the driver will be able to lock the wheels, moreover to avoid a huge problem in most of the racing vehicles, which is the weight distribution of the right and left hand side isn't symmetric because of the sprocket and differential placement, so we will be using adjustable modulating (bias) valve in the line to each caliper.

BASIC DESIGN

At first, in order to satisfy the basic design of the braking system components Newton and Pascal laws were used. A simple linear, lumped parameter model was created for initial system analysis. In addition some assumptions were taken into count.

These assumptions were only taken into count to be able to provide the base to stand on, but further detailed calculations and modeling for the previous stated assumption would take place in this report.

Table 1. Basic Design Assumptions.

Maximum Human force	455 N
Leverage ratio	5
Bias bar ratio	0
Stiffness of the components	neglected
Volumetric losses	neglected
μf between pad and rotor	Constant 0.4
μs adhesion limit	0.9
μd adhesion limit	0.7
Tire torsional stiffness	neglected

Human Force

For brakes without a booster, the brake system should be designed so that for a maximum pedal force of 445 N, a theoretical deceleration is of 0.9g is achieved.

$$F_{pedal} = F_{human} * L_p$$

(1)

The human force has been magnified through the pedal leverage ratio, and the pedal force has been applied to the master cylinder.

Braking System Components

1. Master Cylinder: - Support the brake system with the hydraulic line pressure based on the piston diameter.

$$Pl = \frac{F_{pedal}}{A_{mc}} \quad (2)$$

2. Caliper: - Convert the hydraulic line pressure into clamping force transformed to the pad based on the caliper piston diameter.

$$F_{clamping} = Pl * A_{calp} \quad (3)$$

3. Pad: - Convert Clamping force into friction force applied to the rotor, which is the main function of the braking system.

$$Ff = F_{clamping} * \mu_f * 2 \quad (4)$$

4. Rotor: - Transform the friction force to braking torque based on the effective radius of the rotor. This is a point at which the pad is assumed to be acting and for our purposes we assume it to be the midpoint of the inner and outer radii of the rotor.

$$Tb = Ff * R_{eff} \quad (5)$$

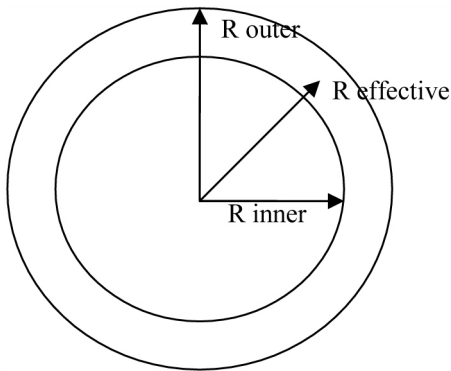


Figure 1. Radii Diagram

Braking Force and Tire

The braking force F_b is defined as the longitudinal force applied to the vehicle in order to stop the vehicle or slow it down. The braking force is based on the braking torque T_b and tire radius r_w .

$$F_b = T_b * r_w \quad (6)$$

The main problem of the braking force in the Formula student that any electronic actuators are prohibited since the braking force is limited by the normal force on the tire, therefore the highest achieved deceleration = 1g is only instantaneously. In figure (2) the deceleration is clearly limited by the weight distribution resulted during acceleration "before hitting the brake pedal", and the one resulted during deceleration. If the

forces are too high the coefficient of adhesion will change from static μ_s to dynamic μ_d which corresponds to a tire sliding in the real world. Therefore an iterative process compares the theoretical braking deceleration to the 'actual' deceleration the tires are capable of to determine at what point tires lockup and what the 'actual' deceleration of the car is.

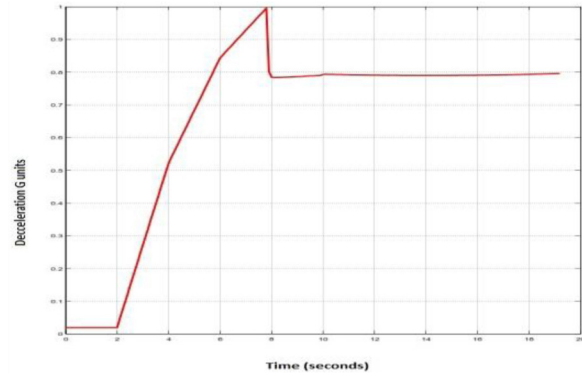


Figure 2. Basic Design Response

MAIN BRAKE SYSTEM DESIGN

Real brake system design is more complicated than stated before, as the stiffness of the brake system components, the volumetric analysis, compressibility of the brake fluid and pad and the torsional stiffness of tire have a huge impact on the brake system performance. Before any optimization steps could be taken, the stated parameters must be taken into consideration.

Brake System Components Stiffness

A schematic of the system can be seen in figure (3) with components and degrees of freedom labeled. The masses include: The master brake cylinder M_{mc} , the brake fluid M_{fl} , the caliper M_{calp} , the pistons M_{pist} , the caliper side brake pad M_{Cpad} , the piston side brake pad M_{Ppad} , and the rotor M_{rot} . The corresponding degrees of freedom include: The master brake cylinder x_{mc} , the brake fluid x_{fl} , the caliper x_{calp} , the pistons x_{pist} , the caliper side brake pad x_{Cpad} , the piston side brake pad x_{Ppad} , the caliper side rotor surface x_{Crot} , and the piston side rotor surface x_{Prot} .

The stiffness's include: The master brake cylinder k_{mc} , the brake fluid k_{fl} , the high pressure seal k_{seal} , the caliper k_{calp} , the pistons k_{pist} , and the brake pads k_{pad} .

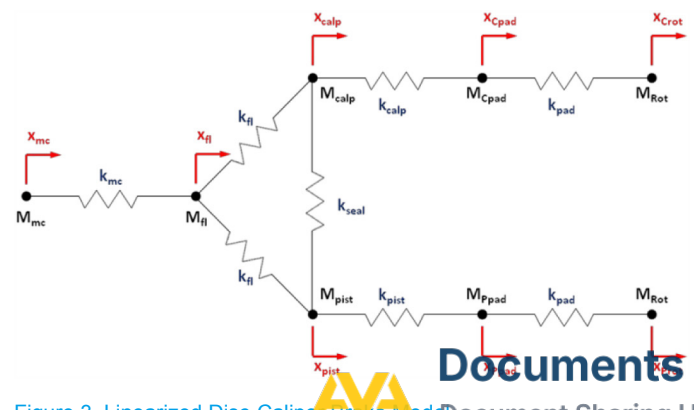


Figure 3. Linearized Disc-Caliper Brake Model

At this point AMESIM program is going to be used to provide the physical model needed for the Disc-Caliper Brake Model.

AMESIM Model

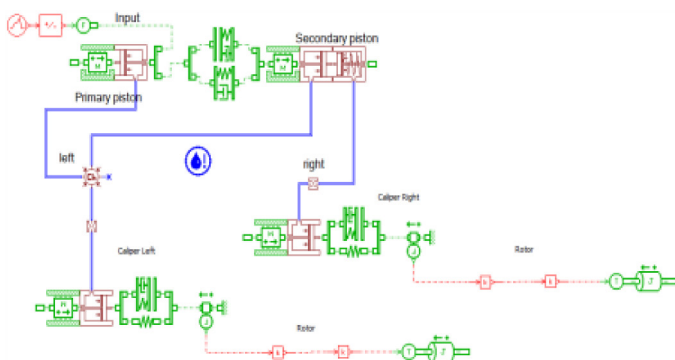


Figure 4. AMESIM Model

AMESIM provide the user with multi-level modeling, which is very useful to our case. The model assumes that the coefficient of friction between pad and rotor μ_f and the effective radius R_{eff} are constant, the two values could vary throughout a given braking event; Moreover The inertia I is used to calculate the angular acceleration $\alpha(t)$ of the rotor disc via equation (7), which is turn used to calculate the angular velocity $\Omega(t)$ of the rotor disc via equation (8)

$$\alpha(t) = \frac{-Tb(t)}{I} \quad (7)$$

$$\Omega(t) = \int \alpha(t) dt \quad (8)$$

The human force is assumed to be a step input into the model and the effect of the geometry of pressing the brake pedal was neglected for simplification of the model.

1. Master Cylinder

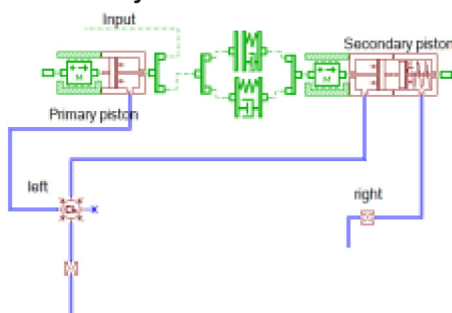


Figure 5. Master Cylinder Model

For the master cylinder in figure (5), the main function modeled is the piston effect within its two chambers. The return parts (equivalent relief valves) to avoid cavitations in the chambers are not considered in the master cylinder model. The master cylinder model only takes into account the restrictions on the

output of the front chamber, the rear chamber has restrictions but with a big hole diameter. The friction force function of the pressure is also not taken into account.

2. Hydraulic Fluid and Line Fittings

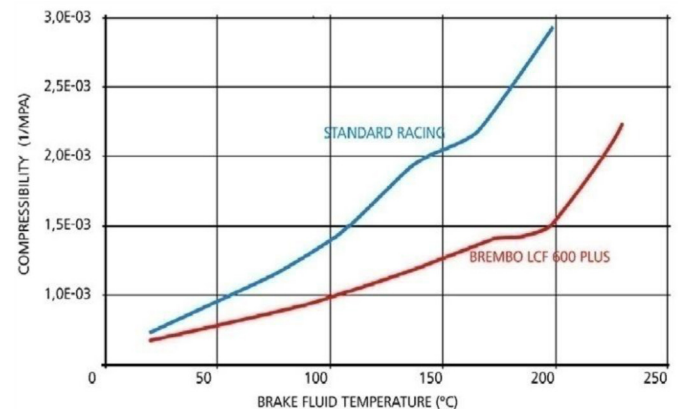


Figure 6. Brembo LCF 600 PLUS Compressibility

AMESIM allows the user to enter the specifications of the hydraulic fluid used in the model. ASURT is using "Brembo LCF 600 PLUS" brake fluid, in which its compressibility curve is shown in figure (6). The volumetric losses and efficiency of the hydraulic circuit will be discussed later in this report, however the friction of the hoses and line fittings of (AN-3) are neglected.

3. Caliper/Wheel Cylinder

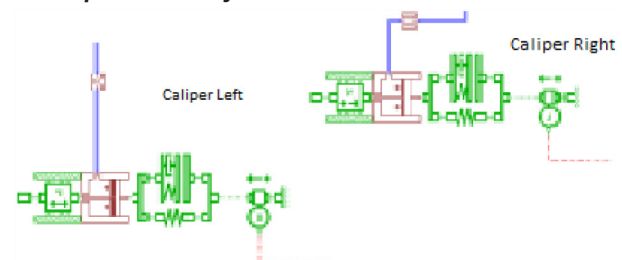


Figure 7. Caliper/ Wheel Cylinder Model

For the caliper in figure (7), the main function modeled is the piston stiffness effect and the friction force of the piston on the hydraulic pressure. Each piston is modeled independently with representative lumped stiffness and mass elements. The parameters are the same for both pistons. The viscosity damping effect of the 2 guide pins attached to the caliper was neglected due to the difficulty of measuring its value in a lab.

4. Response

In figure (8) the curve declares the difference in the hydraulic line pressure by value of 0.057 MPa between two wheels on the front axle. The difference in the hydraulic line pressure is because of the mechanism principle of the tandem master cylinder, as when the pushrod piston moved toward the floating piston, the opening connecting the first chamber with the reservoir is closed. As the pedal travel continues the resulting build up pressure is transmitted by the floating piston to the

next chamber of the master cylinder. The floating piston move forward and the opening connecting next chamber with the reservoir close, and now the whole master cylinder is building up pressure.

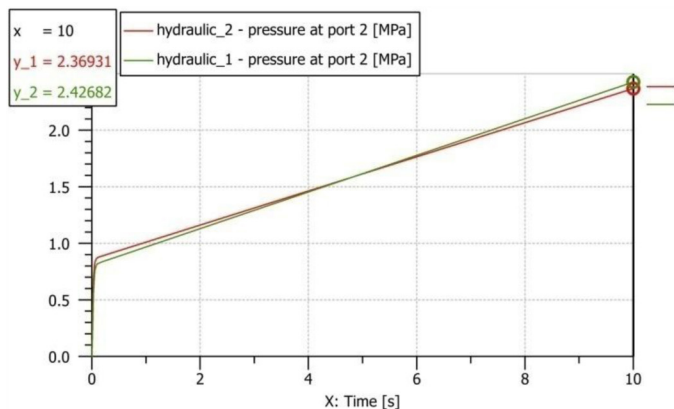


Figure 8. Front axle hydraulic line pressures on two wheels

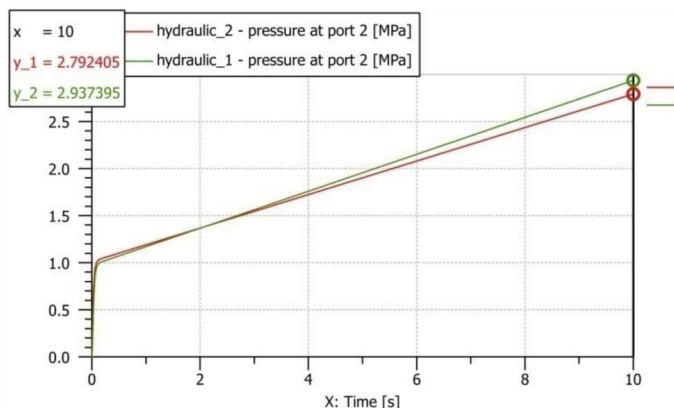


Figure 9. Rear axle hydraulic line pressures on two wheels

In figure (9) the curve also declares the hydraulic line pressure by value of 0.14 MPa between two wheels on the rear axle. The difference in the hydraulic line pressure is because of the build up pressure time delay between one hydraulic circuit and the other as stated before.

Based on the results in figure (8), figure (9) the AMESIM model could be used to adjust the modulating (bias) valve in the hydraulic line to each caliper, in order to reach the target performance, and this would be much clear later in this report.

Brake System Volumetric Analysis

In the brake fluid volume analysis the minimum amount of brake fluid volume that must be delivered by the master cylinder is determined, so that all fluid volume absorbing brake system components can function properly, and the brake pedal travel doesn't exceed its maximum value.

Table 2. Volumetric Analysis data and assumptions.

M _{mc} front bore diameter	23.8, mm
M _{mc} rear bore diameter	19.05, mm
K _{mef} (reference 1)	190 * 10 ⁻⁶ , cm ³ /N/cm ²
K _{mcr} (reference 1)	150 * 10 ⁻⁶ , cm ³ /N/cm ²
Brake Fluid	Figure (6)
Brake line material	Steel
Brake line outer diameter	0.475,cm
Brake line thickness	0.0686, cm
Brake line length (front/rear)	(200/250), cm
Brake hose material	Stainless Teflon
Brake hose outer diameter	1.27,cm
Brake hose thickness	0.794,cm
Brake hose length (front/rear)	(80/60), cm
wf	0.26, cm
wr	0.38, cm
Cs (reference 1)	18.6 * 10 ⁻⁶ , cm/(N/cm ²)
M _{calp} front bore diameter	Fixed 4 piston 28, mm
M _{calp} rear bore diameter	Floating 2 piston 25, mm

In the upcoming calculations based on the data stated in table (2), the maximum brake pedal travel required will be calculated taking into consideration that the braking system should lock at 70% of the maximum brake travel. Second and most important is the volumetric loss in each braking circuit, in order to be able to adjust the brake system for the target performance.

Volumetric Analysis Model

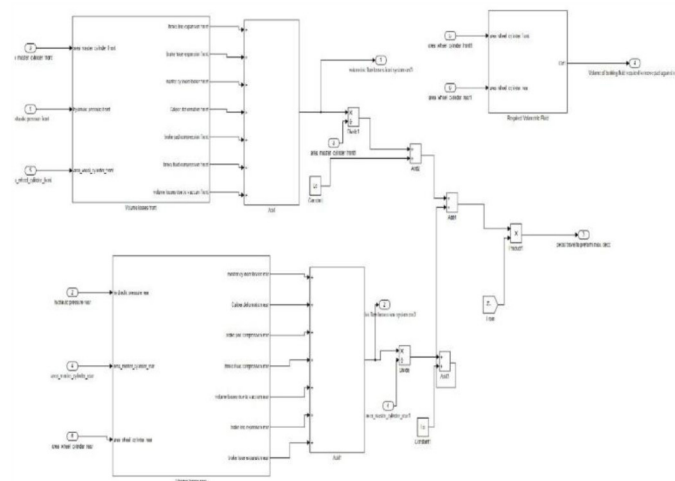


Figure 10. Volumetric Analysis Model

The volumetric analysis in figure (10) and the upcoming volumetric analysis for each component are for the ASURT 14 with brake pedal travel 9 cm, in which the most critical point of interest is at the wheel lock up.