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DESIGN OF A LOW COST BRAKE MASTER CYLINDER FOR BAJA SAE VEHICLES

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Abstract. *This study presents the process of analyzing and sizing a brake master cylinder for use in Baja SAE vehicles, while aiming to reduce the cost of its application. To this end, a case study of the operating cycle of this component was conducted for a generic Baja model. This study, combined with structural analysis processes using the Finite Element Method (FEM), was able to provide the parameters that characterize the project. Whit these characteristics, a SAE J434 (D4018) geometry was built and refined, which makes use of components widely found on the market to reduce costs, capable of guaranteeing comfort and safety by simultaneously locking the axles of a prototype initially at 40 km/h in up to 6.4 meters with an effort of 400 N on the pedal, as it offers a deceleration of 0.99G, with a maximum efficiency loss of 0.808%.*

Keywords: *Baja SAE, Brake, Master Cylinder, FEM, Structural Analysis.*

Resumo. *Esse trabalho apresenta o processo de análise e dimensionamento de um cilindro mestre de freio voltado para aplicações em um veículo do tipo Baja SAE, enquanto se objetiva um menor custo para sua aplicação. Para tanto, foi conduzido um estudo de caso do ciclo de funcionamento desse componente para um modelo genérico de Baja, tal estudo, unido a processos de análise estrutural através do Método de Elementos Finitos (MEF), foi capaz de oferecer os parâmetros que caracterizam o projeto. Com essas características foi construída e refinada uma geometria em SAE J434 (D4018), que faz uso de componentes amplamente encontrados no mercado como meio de reduzir custos, capaz de garantir o conforto e segurança através do travamento simultâneo dos eixos de um protótipo inicialmente a 40 km/h em até 6,4 metros com um esforço de 400 N no pedal, isso pois, oferece uma desaceleração de 0,99 G, com perda máxima de eficiência de 0,808%.*

Palavras chave*: Baja SAE. Freio. Cilindro Mestre. MEF. Análise Estrutural.*

1. INTRODUCTION

Brake systems take advantage of the force of friction between two materials to promote the exchange of kinetic energy into thermal energy, as a form to generate deceleration of a usually circular displacement (Puhn, 1987 and Limpert, 2011). Over the centuries the brakes have evolved according to the power requirements of machinery, from wooden shoe brakes on carts to compressed air disc brakes on large trucks. Currently, hydraulic disc brakes, consisting of a master cylinder, brake calipers and brake discs, are the most widely used due to their efficiency (Puhn, 1987).

However, the sizing of its components is geared towards the consumer market for motorcycles and medium/large cars. One example is the master cylinder, components that undergo major changes in their dimensions according to the pressure they support and whether they are accompanied by auxiliary activation systems, such as ‘hydro boost’ capable of multiplying the forces acting on the master cylinder (Limpert, 2011).

From this, when it comes to Baja SAE vehicles - a type of off-road car used worldwide in university competitions promoted by the Society of Automotive Engineers (SAE) (SAE International, 2024) - which fit in as a middle ground between those mentioned due to their mass. Requiring more structural strength than those offered by motorcycles, but not being able to house auxiliary activation systems impacting on the excess force required by the driver if the component adopted is from passenger cars.

In this context, this study promotes the design of a brake master cylinder for a generic Baja SAE vehicle prototype, aiming for a low cost, so that the component is more adaptable and can be widely applied in the Baja projects.

2. METHODOLOGY

2.1. Defining the parameters

Machines are designed to work as a whole, while paying attention to the individual needs of the components. From this point, the project begins by identifying the problems and parameters that the target component will face (Norton, 2010). In order to include this process in the development of a brake master cylinder, it is proposed to apply it to a generic model of a Baja SAE vehicle, in addition, given its duty cycle, this component can be considered a pressure vessel, so its sizing must be consistent with this fact.

2.1.1. SIZING A GENERIC BAJA MODEL

Table 1 contains the static parameters that characterize the generic model, as well as dynamic parameters that are a consequence of the formulations contained in Milliken and Milliken (1995), Gillespie (1995) and Derek (2015). The braking system is designed to achieve simultaneous axle locking for both dirt and asphalt terrains, so that maximum use can be made of the static friction coefficient on the tires (Limpert, 2011).

Table 1. Generic Baja Model Parameters

Parameter	Unit	Value
Weight	[N]	3,000.000
Wheelbase	[mm]	1,500.000
Front Axle to CG Distance	[mm]	823.500
Front Tire Radius	[mm]	266.000
Rear Tire Radius	[mm]	266.000
CG Height	[mm]	560.000
CG Relative Height	-	0.373
Front Static Axle Load	[N]	1,353.000
Rear Static Axle Load	[N]	1,647.000

Therefore the braking system consists of a brake rotors made from AISI 1045 steel, decelerated by the activation of composite piston brake calipers (Yamaha® XTZ 250) fitted with brake pads from the same manufacturer composed of molded rigid asbestos, copper lines following the sealing system regulated by the SAE, a proportional brake valve (Allstar Performance® ALL48025) and a master cylinder in '*tandem'* configuration, enabling the assembly to be sized.

In sequence, to size the master cylinder cross-section, it is defined a maximum speed of 40 km/h for the model, a braking distance of 5 m, a pedal effort within the limits of maximum and minimum comfort gains (260 N - 470 N) defined by Limpert (2011). In order to calculate and define the required torque (Eq. 1) by the vehicle makes reference to two types of terrains: dirt - friction coefficient of 0.67 G - and asphalt - friction coefficient of 0.95 G -, where 'G' is the coefficient of gravity (Limpert, 2011). In addition, the torque supplied by the brake system was calculated (Eq. 2) for different values of the component's cross-section, with these being consistent with the piston-spring components found commercially, since in this way, despite limiting sizing, manufacturing costs can be reduced.

$$BT_R = W.(F_{Dz,i} - F_{z,i}\,.\,a)\,.BF\,.R_{T,i} \tag{1}$$

$$BT_S = p_l\,.\,A_{wc}\,.\,BF\,.\,R_{ef}\,.\,n_{c,i}\,.\,k \tag{2}$$

where, W = weight; $F_{Z,i}$= static load in 'i' axle; $F_{Dz,i}$= static load distance in 'i' axle; a = deceleration; BF = brake factor (0.8); $R_{T,i}$ = tire radius from 'i' axle; p_l = line pressure; A_{wc} = wheel cylinder area; R_{ef} = brake rotor effective radius; $n_{c,i}$ = efficiency in 'i' axle (0.98) and k = number of rotors by axle.

Finally, the torque values were compared for both terrains and the section diameter was chosen when it was clear that the supplied torque exceeded the required torque when the system activation occurs for the comfort conditions defined.

2.1.2. PRESSURE VESSEL SIZING

When analyzing a differential element in the wall of a pressure vessel, it is subject to three stresses: Tangential (Eq. 3) - causing shear parallel to the plane of the cross-section; Radial (Eq. 4) - acting normal to the surface of the vessel and causing compression/dilation of the cross-section; Longitudinal (Eq. 5) - acting in a direction parallel to the vessel surface and causing longitudinal compression/dilation (Budynas, 1998, Budynas and Nisbett, 2014, Riley, Sturges and Morris, 1998).

$$\sigma_t = \frac{p_i r_i^2 - p_0 r_0^2 - r_i^2 r_0^2 (p_0 - p_i)/r^2}{r_0^2 - r_i^2} \tag{3}$$

$$\sigma_t = \frac{p_i r_i^2 - p_0 r_0^2 + r_i^2 r_0^2 (p_0 - p_i)/r^2}{r_0^2 - r_i^2} \tag{4}$$

$$\sigma_l = \frac{p_i r_i^2}{r_0^2 - r_i^2} \tag{5}$$

where, p_i= internal pressure; p_0= external pressure; r_i= internal radius; r_0= external radius; r = differential element radius, as can be seen in Fig. 1.

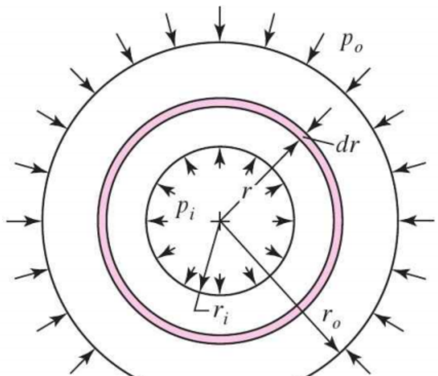

Figure 1. Cylinder Subjected to Internal and External Pressures (Budynas and Nisbett, 2014)

In addition, for the design of this type of structure there is a differentiation according to wall thickness, because for thick-walled vessels (with a thickness greater than 5% of their external radius), which is the case in this study, it is necessary to evaluate both radial and tangential stresses (Budynas, 1998, Budynas and Nisbett, 2014, Riley, Sturges and Morris, 1998).

In the case of the master cylinder, the internal pressure is, as described by Limpert (2011), dependent on the force applied to the pedal, as well as other constants such as efficiency, lever ratio and cross-section. Consequently, the next step is to define the maximum force applied to the pedal for a critical situation, so that the design is more reliable. As shown by Puhn (1987), in an emergency situation, the pedal will be subject to intense and rapid activation, so the speed of application multiplies the weight applied by the driver's leg on the component, reaching a state where the force felt by the pedal is twice the driver's weight. In addition, the male 99th percentile, which covers 98% of the world's population, indicates that a man's mass is 111.2 kg (Tiley, 2011), so the force used to calculate the pressure inside the master cylinder is 2224.0 N.

Finally, the design of the structural stresses felt is given according to the concepts developed by von-Mises (1913) and revised by Barsanescu and Comanici (2017), with thicknesses of 1.0, 1.5, 2.0, 2.5, 3.0 and 3.5 mm being evaluated, since the student license of the Ansys® Workbench 2023 R2 software has processing limitations.

2.2. Structural Analysis

Proof bodies were built using Solid Works® Student 2023 software to represent a differential element in the wall of a pressure vessel. It is worth noting that those bodies share an internal diameter and length, varying only in terms of wall thickness. The analysis were carried out using the Ansys® Student 2023 R2 software, using the 'Static Structural' input, with the mesh being defined in triangular format and refined by the convergence process, as described by Ho-Le (1988), Zienkiiewicz and Taylor (2000), Fish and Belytschko (2007).

With respect to this analysis, the boundary conditions applied are those consistent with the work cycle of the component, considering that it operates fixed to the chassis and protected by the fairing, with external and internal pressures acting on it, Fig. 2 shows the applied conditions and Tab. 2 characterizes them.

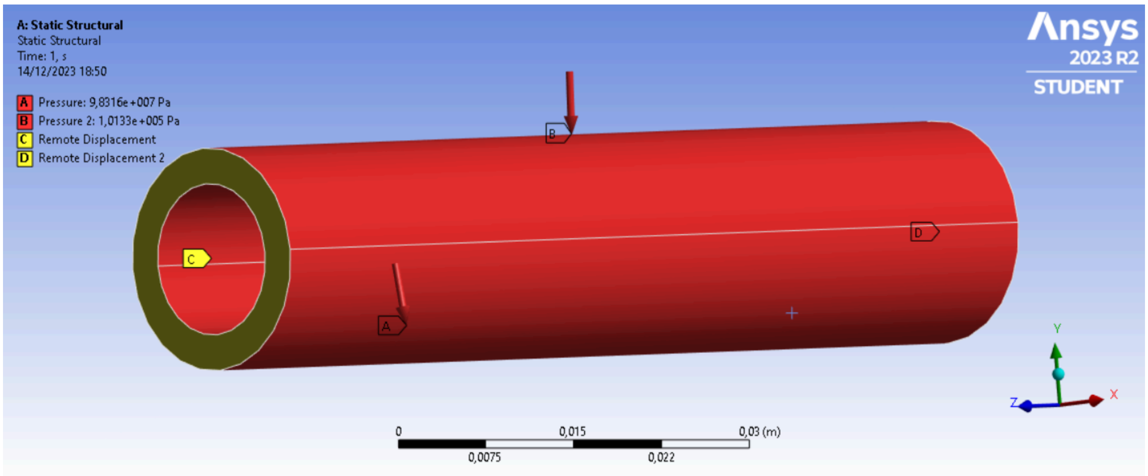

Figure 2. Boundary Conditions Applied to the Differential Element

Table 2. Boundary Conditions

Input	Referential	Application Point	Configuration
Pressure	A	Internal faces	Generated by maximum pedal force
	B	External Faces	Atmospheric pressure
Remote Displacement	C	Left side face	XY plane = 0; YZ plane = 0; ZX plane = 0; X Axis Rotation = Free; Y Axis Rotation = Free; Z Axis Rotation = 0;
	D	Right side face	XY plane = Free; YZ plane = Free; ZX plane = Free; X Axis Rotation = 0; Y Axis Rotation = 0; Z Axis Rotation = 0;

In this way, the 'C' referential limits the axial displacement of the face and the torsion of the body, allowing the analysis of the cross-section deformation. In addition, the 'D' referential, which limits rotations, allows analysis of the longitudinal deformation that the material undergoes as a result of the deformation of the section.

2.3. Safety Factor

As developed by Norton (2010), the safety factor varies according to the types of stress that the part undergoes, as well depending on the conditions on which the project is based, being judged through three groups: F1 - material data; F2 - conditions of use and F3 - precision of the analytical model, with the coefficient being chosen as the one with highest value among the three groups. In sequence, Tab. 3 describes the analysis carried out for each factor, demonstrating and justifying its choice.

Table 3. Safety Factor Assessment

Group	Criteria	Value
F1	Material information comes from catalogs of internationally recognized institutions that acquire their data through experimentation.	2.0
F2	The master cylinder is fixed to the prototype, in addition to being protected by its chassis and fairing, therefore, it is not subject to many boundary conditions other than those established in the laboratory.	2.0
F3	The analytical model adopted faithfully represents the component's use cycle.	2.0
Chosen Safety Factor		2.0

2.4. Geometry Confection

A brake master cylinder can be considered as a pressure vessel, therefore in its design the appropriate international standards are used, these being the ASME Section VIII - Boiler & Pressure Vessel Code (ASME 2019a, 2019b). However, given the proportions of the designed component, it is understood that the reproduction of all these parameters would result in oversing; thus some fractions of the aforementioned standards were not considered.

Furthermore, regarding the lower production cost, this study is limited to the design of the cylinder structure, so the other components will be taken off the market. Thereby, the spring-piston assemblies used in the project are the generic used in motorcycles (recommending those referring to the Honda® XRE 300), as well as the sealing system and fluid reservoir (recommending those referring to the Chevrolet Vectra 1998 - Bosch® 2262826).

In the first instance, the primitive geometry was developed in a rectangular format, focusing on the ability to encompass the aforesaid generic components and the positioning of the bores, dealing with the fluid inlets and outlets, as well as the bores dedicated to lubricating the pistons. It is worth noting that those for the fluid outlets are in line with SAE sealing system, with an M10x1.25 threaded hole, and the rest follow the patterns seen in master cylinders for both motorcycles and cars available on the market.

Once this geometry has been defined, it is proposed to carry out the same process mentioned in item 2.2, but sequentially taking it to the 'Structural Optimization' function. In this process, the faces that receive the boundary conditions and cover the fluid flow are restricted, admitting a percentage of 35% mass reduction for the generation of a new geometry; this geometry is then refined in the CAD environment to produce the final body.

2.5. Materials and Fabrication Method

Once with the maximum values for the stress that the component must be able to withstand, as well as being aware of the value of the safety factor established, it is used the Metals Handbook (1990a, 1990b), SAE International (1986, 2004), Bauccio, *et al.,* (1993) and Davis, *et al.,* (1996) bibliographies to carry out a comparative survey of the materials and their mechanical properties. After the survey, the calculation is made to define the safety factor for each material and thickness analyzed.

Observing the recurrence of a cylindrical geometry for bodies that perform the same function, it is anticipated that the project conceived by this study shares this characteristic. For this format and considering the complexity of its structure (with arrangement of holes and extrusions to support the fixture in chassis and the fluid reservoir) casting manufacturing processes are a viable option (Stefanescu, *et al.,* 1988).

It should also be noted that manufacturing processes and their different methods greatly influence the choice of material (Groover, 2020). Therefore, taking into account technical parameters, cost and accessibility the candidates for the project are defined, relating their mechanical properties and the thickness required for it to exceed the established safety standard.

2.6. Performance Analysis

Any body made of a ductile material, when subjected to forces, tends to deform according to the region of incidence and intensity of the force (Budynas, 1998, Budynas and Nisbett, 2014, Riley, Sturges and Morris, 1998). In the case of a cylindrical structure, such as the one studied, the occurrence of an internal force generates deformation in such a way as to increase the cross-section diameter. This increase has a negative influence on the behavior of the component, since, as said by Limpert (2011), the braking forces are directly dependent on the cross-sectional area.

It is worth noting that thermal expansion, which results from the increased temperature of the brake fluid when pressurized, has a very low degree of magnitude and thus is disregarded in the analysis.

Therefore, reusing the same conditions and mesh proposed in item '2.2.', the 'Deformation (Total)' input is analyzed to note the deformation values and calculate their influence on the master cylinder's performance. This calculation was made by comparing the braking forces and deceleration generated by the system without deformation and whit maximum deformation.

3. RESULTS

3.1. Master Cylinder Parameters

As can be seen by the Tab. 4 and Tab. 5, for a master cylinder with a cross-sectional diameter of 12.7 mm the supplied torque by the braking system overcomes the torque required by the vehicle, while keeping within the standard of effort generated by the pilot previously established. It should also be noted that with this configuration the system provides 0.99G of deceleration, thus, fulfilling the objectives of comfort but stopping at a distance 28% higher (6.4 m).

Table 4. Required Torque for Different Velocity and Terrains

Ideal Displacement Characteristics				Required Torque [N.m]			
Velocity [km/h]	Velocity [m/s]	Deceleration [m/s^2]	Deceleration [G]	Required Torque - Dirt		Required Torque - Asfalt	
				Rear	Front	Rear	Front
30.00	8.33	6.94	0.71	152.228	382.432	215.846	542.254
35.00	9.72	9.45	0.96	101.203	433.457	143.497	614.603
40.00	11.11	12.35	1.26	42.328	492.332	60.017	698.083
45.00	12.50	15.63	1.59	-24.397	559.057	-34.593	792.693
50.00	13.89	19.29	1.97	-98.972	633.632	-140.334	898.434

Table 5. Supplied Torque and Case Study

Pedal Force [N]	Supplied Torque [N.m]				Case Study					
	Supplied Torque per Disc		Supplied Torque per Axle		Dirt			Asfalt		
	Rear	Front	Rear	Front	Rear	Front	Results	Rear	Front	Results
250.00	309.096	247.277	309.096	494.533	Approved	Approved	Valid	Approved	Reproved	Non Valid
300.00	370.915	296.732	370.915	593.464	Approved	Approved	Valid	Approved	Reproved	Non Valid
350.00	432.734	346.187	432.734	692.375	Approved	Approved	Valid	Approved	Reproved	Non Valid
400.00	494.553	395.643	494.553	791.285	Approved	Approved	Valid	Approved	Approved	Valid
450.00	556.373	445.098	556.373	890.196	Approved	Approved	Valid	Approved	Approved	Valid
500.00	618.192	494.533	618.192	989.107	Approved	Approved	Valid	Approved	Approved	Valid

Applying the load defined in section 2.1.2 to the component, an internal pressure of 98,316,291.065 Pa will be noted in the vessel. Using this value to the boundary conditions described in 2.2., the safety factor defined in item 2.3. and using the 'Equivalent Stress (von-Mises)' input the maximum expected stresses for each thickness is shown in Tab. 6.

Table 6. Expected von-Mises Equivalent Stress for Various Thickness with Safety Factor

t [m]	Von-Mises Equivalent Stress [Pa]
0.0010	647,500,000.00
0.0015	439,390,000.00
0.0020	336,710,000.00
0.0025	272,530,000.00
0.0030	227,800,000.00
0.0035	197,700,000.00

3.2. Geometry

In sequence are shown the primitive geometry (Fig. 3), the final geometry (Fig. 4) and the master cylinder assembly with the identification of the components, but with omission of the brake fluid reservoir (Fig. 5).

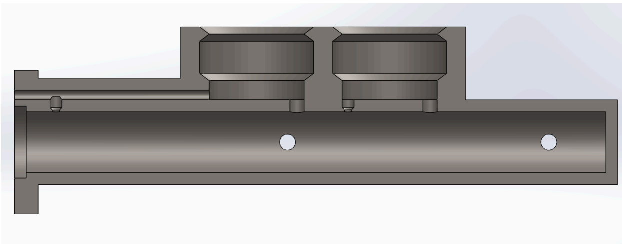

Figure 3. Primitive Body in Section (Right View)

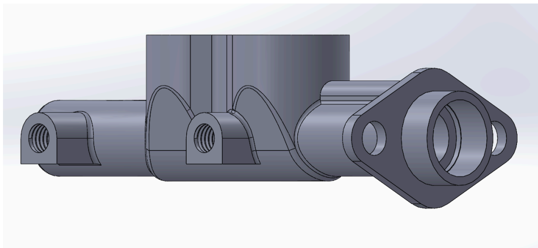

Figure 4. Final Body

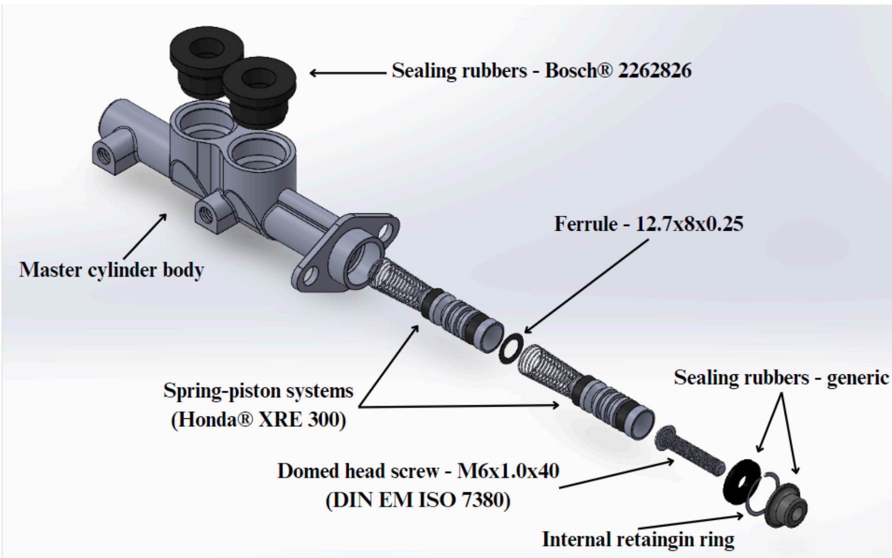

Figure 5. Master Cylinder Assembly

3.3. Fabrication Method and Materials

Among the various types of casting, the sand casting process stands out for its low cost and adaptability, being able to produce complex shapes, especially in terms of internal cavities (Stefanescu, *et al.,* 1988), offering what the geometry requires, therefore being adopted. In addition, amid the possible materials, the search is initially focused on cast irons, since they can be easily applied to sand casting, as said by Stefanescu, *et al.,* (1988) and are widely available. Among its types, SAE J434 (D4018) was chosen, which has a Yield Strength of 275 MPa (SAE International, 1986, 2004), thus manifesting a safety factor of 2.018 for 2.5 mm thickness, meeting the safety requirements.

3.4. Expected Performance

The SAE J434 (D4018) has a Modulus of Elasticity of 152 GPa (SAE International, 1986, 2004), so for the conditions of use there is a maximum deformation of the master cylinder cross-section of 0.51 mm. With the occurrence of such an expansion the system presents a loss of 0.808% in its braking force generation and, therefore, deceleration of the vehicle. Given the magnitude order of this loss, it can be inferred that the project performs satisfactorily.

4. CONCLUSION

The master cylinder presented shows itself to be an option capable of achieving the goals for safety and comfort while maintaining a low cost, therefore being viable for Baja SAE teams wishing to carry out a more assertive braking system design. However, the following are still recommended: a fluid dynamic analysis of the behavior of the fluid, so that it is possible to validate the laminar flow in the lubrication, inlets and outlets bores; a thermal expansion study to validate the disregard of this parameter; a study to understand which of the various components available on the market are best suited to this application and an experimental analysis to validate its performance.

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6. RESPONSIBILITY FOR INFORMATION

The author is solely responsible for the information included in this work.