Design of double stage springs for Baja SAE type vehicles

Dimensionamento de molas de duplo estágio para veículos tipo Baja SAE[1]

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Resumo
Visando melhorar os parâmetros que influenciam no desempenho e conforto do protótipo da equipe Komiketo Baja UFSJ e atingir bons resultados nas competições Baja SAE, foi proposta uma nova

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metodologia para a substituição das molas lineares por molas de duplo estágio. A mola é um componente capaz de se deformar elasticamente e retornar à sua posição inicial, além de compor o sistema de suspensão automotiva e influenciar no desempenho dinâmico do veículo. A experimentação consiste na elaboração de uma tabela em Microsoft Excel e na simulação de um modelo do veículo percorrendo uma pista irregular, de alta e baixas frequências de oscilação da suspensão no software MSC Adams Car®. A simulação foi dividida em duas partes, sendo uma com o veículo operando com molas lineares e na outra, com as molas não lineares (duplo estágio). A fim de comparar o comportamento nas duas configurações, considerou-se a aceleração vertical como critério comparativo. Os resultados foram satisfatórios, apresentando melhorias, principalmente, no conforto de conforto em relação a mola linear anteriormente utilizada no veículo.
Palavras-chave: Molas, Suspensão, Duplo Estágio, BAJA SAE.

Abstract
Aiming to improve the parameters which influence the performance and ergonomics of the Komiketo Baja UFSJ prototype and to optimize its results during the Baja SAE competitions, a study proposed a new methodology to substitute linear springs for double stage spring. Springs are components with high elastic deformation's ability that compound the automotive suspension system and they influence the vehicle dynamic performance. The experimentation consists in using the Microsoft Excel software and making a simulation of a model going through an irregular trail presenting high and low oscillation frequencies of the suspension, using MSC Adams Car® software. The test was divided in two stages, one for the operation of the vehicle with linear springs and the other for non-linear springs (double-stage springs). Then, some graphics were presented with the necessary data, aiming to compare the behavior of both suspension conditions, where the vertical acceleration was considered as a comparative criterion. The results were promising presenting improvements as the ergonomics parameters, regarding the linear springs previously used in the vehicle.
Keywords: Springs, Suspension, Double Stage, BAJA SAE.

1. Introduction

Developed by Society of Automobile Engineers (SAE), Baja SAE project was created in the University of South Carolina, in the United States, and it had its first competition in 1976 under the command of Dr. John F. Stevens. The program aimed the construction of an off-road prototype by college students, promoting the practical application of its knowledge acquired in the classes and also aiming to prepare these students for the job market (Baja FATEC-SP, 2012). Therefore, the Komiketo Baja UFSJ was created to project, develop and build a reliable and technological off-road vehicle. Between the prototype subsystems, the suspension is responsible for the stability, the ergonomic and tire contact with the ground and it is composed by a group of pieces that absolve the vibrations imposed by the vehicle, besides connecting the wheels with the car bodywork. The suspension is associated with springs and shock absorbers that, without those, the discomforting would be greater and the lifespan of the vehicle's components would decrease.

Ride is about the influence in the vehicle comfort and the vertical dynamic. This analysis is subjective when there is different human perception for comfort. The study of this behavior is performed through the observation of vibrations transmission between ground and suspension system, from the chassis mounting points until the interface elements between vehicle and driver as seats, steering and pedals, performing in accordance to the ride compression of the vehicle (Gillespie, 2021).

One of the pieces that interfere in the comfort is the spring. This equipment operates in vibration isolation and it influences the vehicle performance. The team goals are to achieve good results in National and Regional competitions proposed by SAE and to improve the prototype dynamic and ergonomic behavior. Therefore, it was noted by the suspension sector the necessity to

change the used linear springs for double-stage springs, which will be designed for the vehicle project by Komiketo Baja UFSJ.

Once the present Komiketo Baja UFSJ prototype does not present good results in the vertical dynamic and comfort with its linear springs, a study was developed for the design of a double-stage spring of a vehicle mini Baja. In this article, analytical calculations and simulations were used with the support of MSC ADAMS Car® (Automatic Dynamics of Mechanical Systems) software, aiming to balance the vehicle's dynamic and ergonomic parameters.

2. Theoretical framework
This topic will approach the fundamental principles for the development of this study.

2.1. Suspension
Suspension is a mechanism that connects the wheels to the chassis, allowing the movement of these components so they absolve all the ground excitations. The subsystem history is ancient where the wagons and carriage already presented a type of suspension to improve the passenger comfort. According to Martins (2004), the main role of this system is to avoid the ground irregularities to be transmitted for the vehicle and therefore, for its passengers.

Suspension must present a good performance so the vehicle can have desirable vertical, longitudinal and lateral dynamics, maintaining the drivability when accelerating, braking and during curves. The requirements for this project varied according to the dynamic goals for the car.

Between suspension classifications, there are the dependent and independent ones. The dependent class, also called as rigid suspension, created a dependence between the wheels of the same axle, hence, when a wheel receives energy, the movement is transferred to the other wheel. This model is sturdy and it has low manufacturing cost and simple project. For the independent suspension, the wheels are articulated without interfering with the other one and it can be applied both front and rear of the car. This geometry presents low weight, better drivability and control at each wheel.

2.2. Double A
The Double A geometry (Double Wishbone) is a type of independent suspension composed by two harms mounted in overlapping planes, having a superior and an inferior, and it can have different or the same size. This model was created after World War II and, nowadays, it is commonly used in high performance or luxury cars, because it allows a good dynamic behavior and it can easily suit these vehicle configurations. According to Almeida (2012), the overlapping harms can present as advantage precise control of the camber angle, small gauge variation, vibration isolation, resistant components, extended useful course and possibility of an oversteer or understeer configuration.

However, the model has elevated cost when compared to other suspensions and it needs a criterial design for a better performance.

2.3. Basic Concepts
Next, there will be discussed some fundamental parameters for the suspension project and methodology used in the springs design posteriorly presented in the fourth topic.

2.3.1. Ride
According to Wong (2009), ride is associated with the passengers' sensations during the vehicle movements. Problems related to discomfort during a trip can occur mainly from oscillations of several sources, being those irregularities in the ground, aerodynamic forces, engine and transmission vibrations and non-uniformities in the wheel or tire.

The aim of the ride study is to control those excitations, ensuring the passenger's well-being, considering the individual response to the vibrations and the natural frequencies of the human body, as it can be observed in Figure 1.
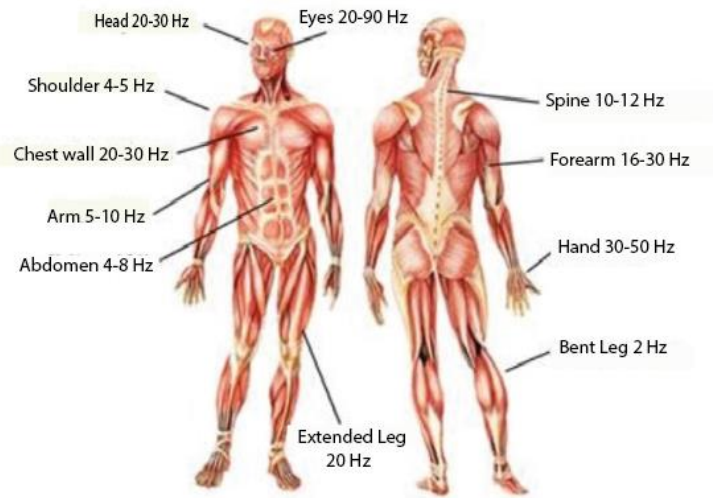

Figure 1. Natural frequencies of the human body.
Source: Pardal (2015)

2.3.2. Sprung and Unsprung Mass

Sprung mass corresponds to the car bodywork and all elements above suspension. However, the unsprung mass is related to all mass located below the suspension, as the suspension and some equipment connected to this subsystem (Diulgheroglo, 2018).

The unsprung weight in a wheel infers how it absorbs the ground vibrations and in the induction of forces to the suspended masses, once non-suspended ones respond to excitations with their own movements. High performance cars tend to have a lower unsprung weight due to a bigger wheel adherence to uneven tracks and greater predictability of the vehicle behavior, since the reaction forces acting against the floor will be smaller and there is more control of the wheel during braking and acceleration.

2.3.3. Ride Rate

The suspension system presents stiffness and damping properties. These characteristics infer the ability to isolate the vehicle's vibrations. Ride rate determines the shift of the sprung mass in a vertical displacement, considering sprung, shock absorber and tire rate. Data can be found by using Equation 1 (Gillespie, 2021).

$$RR = \frac{k_t \times k_c}{k_t + k_c} \tag{1}$$

Where k_t is suspension stiffness and k_c is spring stiffness.

2.3.4. Natural Frequency

Natural frequencies, according to Bolina *et al*. (2014), indicated a free oscillation rate of structure after the forces that caused its movements ceased, therefore, they represent a quantity of vibration of the structure when there is no force being applied. This number is related to the rate and it is the

inverse of structure mass, being a real positive number defined by Hz or rad.s[-1]. Gillespie (1992) affirms that street cars present natural frequencies of 1 to 1.5 Hz and performance ones have values of 2 to 2.5 Hz for a better dynamic behavior. For this study regarding the frequency and comfort, it was considered a rate of sensibility admissible for the human body and it can be calculated using Equation 2, in hertz, or Equation 3, in rad.s[-1].

$$f_n = \frac{1}{2\pi}\sqrt{\frac{RR}{M}} \tag{2}$$

$$\omega_n = \sqrt{\frac{RR}{M}} \tag{3}$$

Therefore, RR corresponds to Ride Rate and M to vehicle mass.

2.3.5. Static Deflection

For Gillespie (2021), the static deflection, SD, is represented by the rate between the vehicle weight, w, and the suspension ride, RR, and it influences the car's natural frequency calculation. To obtain this parameter, Equation 4 can be used.

$$SD = \frac{w}{RR} \tag{4}$$

Regarding the static deflection's force, F_{SD}, of a spring, it can apply Equation 5 proposed by (Costa, 2006).

$$F_{SD} = \left(\frac{M_s}{2}\right) \times i_f\left(\delta_{y,est}\right) \tag{5}$$

where: M_s is the suspended mass of the vehicle and $i_f\left(\delta_{y,est}\right)$ is the force transfer coefficient.

2.4. Springs

Shingley *et al.* (2008) imply that a spring is a compound that presents flexibility in accordance with its designer goal and it allows application of force or torque. Its stiffness determines the car bodywork frequency and inclination during movements on the axes pitch, yaw and roll.

There are two common categories for car springs, the mechanics and the pneumatics types. Regarding the mechanics, there are bending, helical and torsion springs and they store the mechanical energy from deformation of the shock absorber. However, the pneumatic springs are filled with compressed air and have an easier regulation. In this study, it will be discussed about the helical springs.

The helical springs are common in ordinary vehicles and it can be also considered as spiral shape torsion springs. This model is compact, light and also easy to adapt in several suspension geometries. As the shock absorber, the helical springs also have compression stroke (bump) and distension (rebound). For its design, it is necessary to consider some variables such as spring free length, wire diameter, spring diameter and number of active spirals presenting a linear or non-linear behavior, as required by the designer.

2.4.1. Linear and Double-Stage Springs

According to Aranha *et al.* (2016), when a spring is evaluated as its behavior under loading, it can be classified as linear or non-linear. It is considered linear when the deformation is proportional to the submitted load, which can be described by Hooke's Law (Equation 6).

$$F = -K \times \Delta x \tag{6}$$

Where F is the force, K is the spring elastic constant and Δx is the difference of spring length as the deformation occurs.

Considered as non-linear, double-stage springs can present different values of K as the applied force changes. Figure 2 exhibits the behavior of both types of springs.
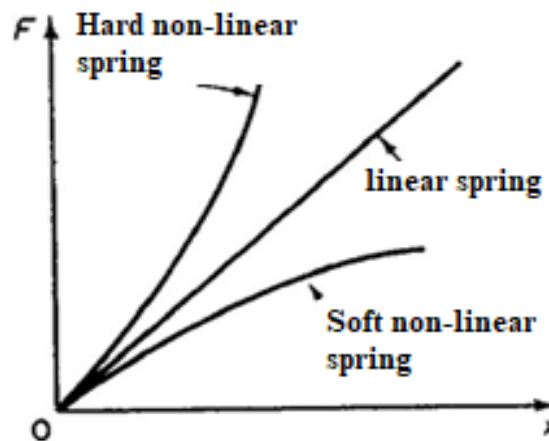
Figure 2. Linear, soft and hard non-linear spring chart.
Source: adapted Aranha *et al.* (2016)

2.4.2. Spring mass
As discussed by Costa (2006), the mass of the designed spring is found using Equation 7.

$$W = \frac{\pi^2 d^2 D N_t \rho}{4} \tag{7}$$

Where d is the wire diameter, D is the spring diameter, N_t is the total spirals number and ρ is the material density.

2.4.3. Spring Stiffness
As mentioned in the 2.3.2 topic, suspension stiffness is a factor that infers in vibration's isolation and, as consequence, in the vehicle comfort and dynamic behavior. To define spring stiffness is necessary to determine some parameters, such as the Ride Rate and identify the car performance during certain situations. For Menezes (2015), this quantity is regarding the spring and the Newton's force needed to deflect the spring in 1 mm. It is possible to obtain the data with the support of Equation 8.

$$k_c = \left(\frac{i_f M_s f_{dg}}{2\left(L_{e,c} + 0.5 L_{reg} + 5\right)} \right) \tag{8}$$

Where i_f is the force transfer coefficient, M_s is the prototype sprung mass, f_{dg} is the mass fraction on the evaluated axis, $L_{e,c}$ is the rebound and L_{reg} is the adjustment length.

2.4.4. Spring Natural Frequency

As previously cited, the natural frequency is the "frequency of a body in free vibration". However, the natural frequency of the suspension group and spring must not assume the same value due to resonance problems. According to Costa (2006), the spring natural frequency can be found using the calculation above, Equation 9.

$$f_n = 0.5 \sqrt{\frac{k_c \times g}{W}}$$
(9)

Where k_c is the spring ride, W is the spring weight and g is the gravity number.

3. Materials and Methods

For the experiment, it is necessary to define the spring material and input data. The values are pre-defined during the suspension project, and they depend on the goals predicted for the prototype dynamic behavior and the chosen material for spring. Initial parameters used in the experiment are presented in Table 1.

Regarding the material chosen for manufacturing, a benchmarking was performed with companies specialized in spring design for automotive suspensions.

To manufacture the equipment, the material must have a hardening steel with enough carbon so the quenching process can allow the spring's tenacious characteristic. The material can be chosen by its tensile strength.

Table 1. Project initial data.

Symbol	Name
a	Length of spring/damper assembly at maximum deflection
b	Length of spring/damper assembly at maximum deflection
c	Length between the joint of the tray and the lower fixing of the set
e	Length between the joint of the tray and the upper fixing of the assembly
Ms	Sprung mass
Lc	Bump
Le	Rebound
Ltot	Full course
Lreg	Regulation length
Kt	Tire stiffness

Source: Authors (2023)

Therefore, SAE 9254 steel is the most suitable material for the study. Steel is a carbon steel alloy with average hardenability and it is recommended for springs and torsion bars. It is indicated for springs that support shock loading and temperatures until 250°C. Information about this element is exhibited in Tables 2 and 3 (SAE 9254).

Table 2. Information of 9254 steel.

Variable	Value (unitary)
density	7.85 (g/cm³)
shear modulus	78 (GPa)
elasticity modulus	200 (GPa)

Source: MATWEB (2020)

Table 3. Composition of 9254 steel.

Element	Percentage (%)
C	0.51 -0.59
Cr	0.60 – 0.80
Fe	96.14 – 97.09
Mn	0,60 – 0.80
P	=0.035
Si	1.20 – 1.60
S	<=0.04

Source: MATWEB (2020)

After defining input data and material for the spring manufacturing, it was developed, with support of Microsoft Excel software, a sheet divided in three parts (Input Data, Material Properties and Spring Data) having the needed information for the spring design, based on Deodato (2019) methodology. Input data were obtained through the concepts and equations presented in the third topic. Excel spreadsheet has name, symbol, value, unit of measurement and it will be presented in the Results.

Then, a simulation started using the MSC ADAMS Car® software to analyze the comfort and dynamic performance. Therefore, there was developed the prototype and tracks with different obstacles, like different frequencies. High frequency route had obstacles of 300 mm x 400 mm and the low frequency track had a ramp with height of 1,5 m and inclination of 20° (Figures 3, 4 and 5). The experiment was performed in two stages. First, simulation of the car with linear spring and second using the double-stage spring. The modifications happened only in the car front axle. Aiming to compare both springs, the rear axle presented linear springs with stiffness of 20 N.mm⁻¹, besides having the same tire stiffness and damping constant.
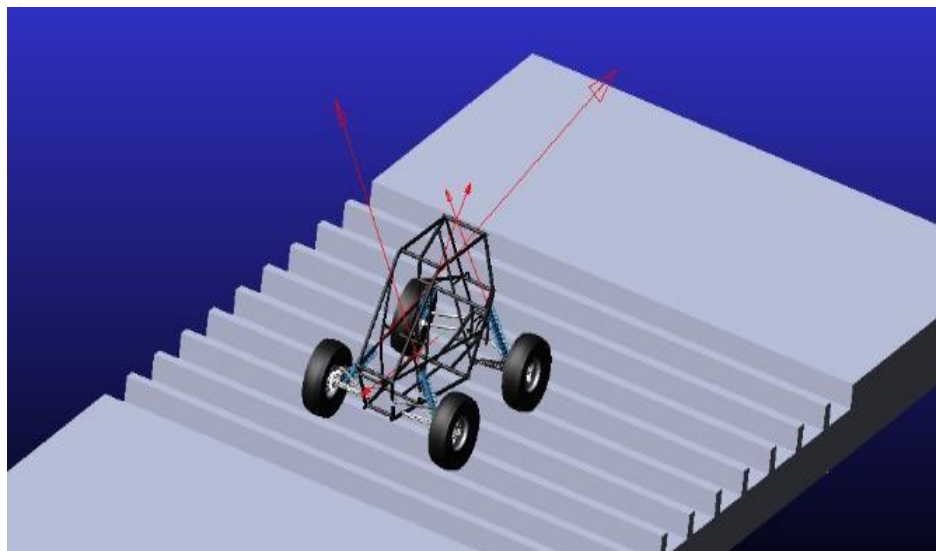

Figure 3. Transposition of high frequency oscillation obstacles.
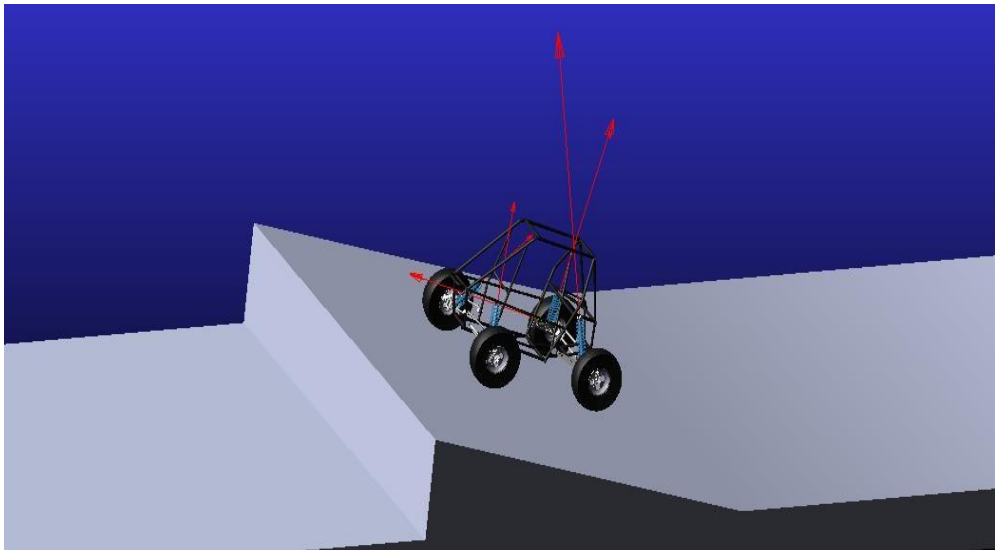
Source: Authors (2023)

Figure 4. Prototype performing ramp ascent.
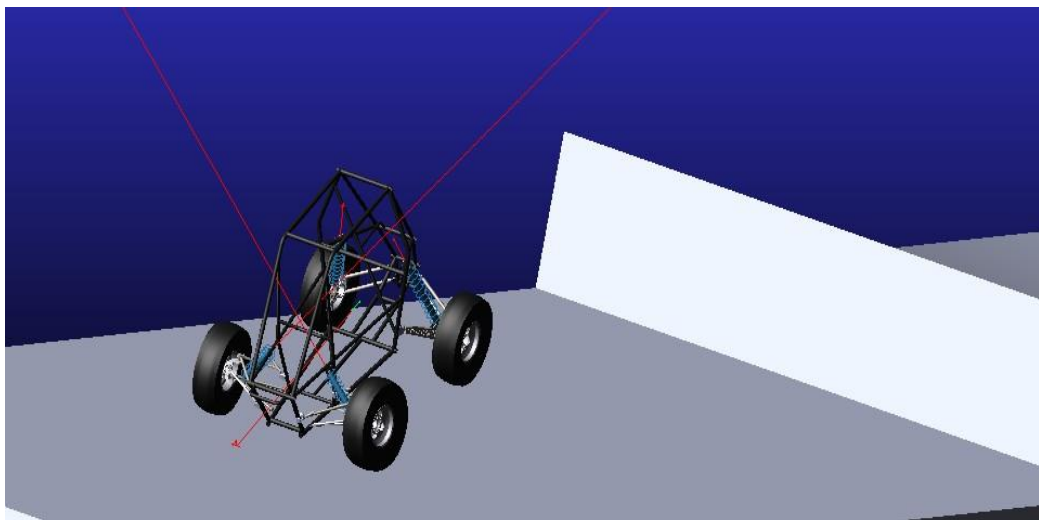Source: Authors (2023)

Figure 5. Prototype choking with the ground after the ramp ascent.
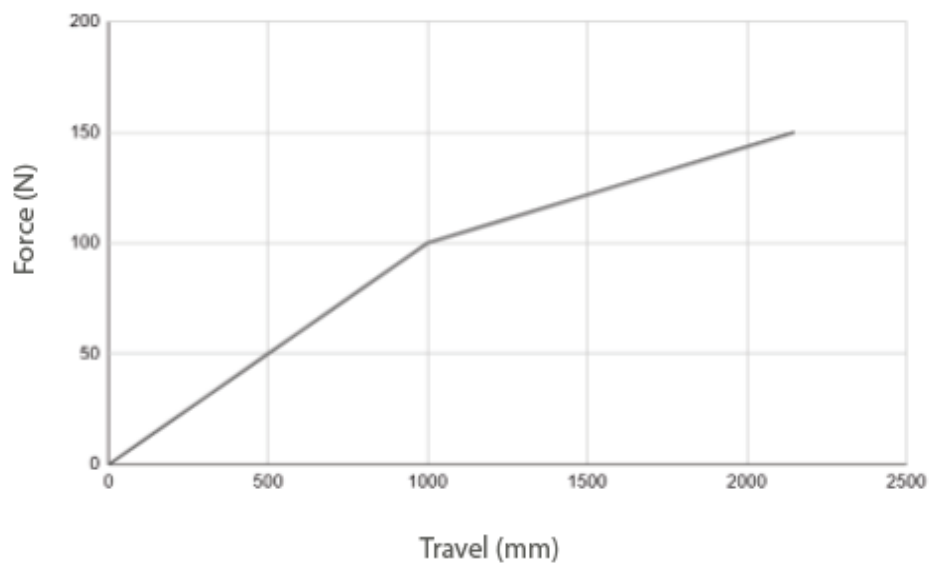Source: Authors (2023)

4. Results

Linear spring dimensions were determined with the support of a table, as can be observed below. Input data are the values defined during the project and spring data are for information resulting from the previously cited equations (Table 4).

Table 4. Linear spring design.

Front Entry Data		
	Value	Unit of Measurement
Total Travel AFCO	152	mm
Bump	101	mm
Rebound	51	mm
Sprung Mass (front)	34,54	kg
Unsprung Mass (front)	28,35	kg
Adjustable Length (Lreg)	100	mm
Tire Stiffness	43,4	N/mm
Spring Material: steel 9254		
Stiffness Modulus (G)	80000	Gpa
Material Density (p)	7,85	g/cm³
Spring Data		
	Value	Unit of Measurement
Wire diameter (d)	7	mm
Spring inner diameter (Di)	55	mm
Outside diameter spring (Do)	69	mm
Average spring diameter (D)	62	mm
Number of active turns (Na)	9,642	-
Total number of turns (Nt)	11,642	-
Spring stiffness (kc)	10,448	N/mm
Spring mass (w)	685,0539508	g
Natural frequency (Fn)	6,114832096	Hz

Source: Authors (2023)

As the linear spring values were defined, the data to design the double-stage springs were obtained. As the stiffness of linear springs presented good results, it was looked to maintain the value similar to double-stage spring, being 17 N.mm⁻¹ and 23 N.mm⁻¹. The respective curves can be observed in Figures 6 and 7. As it can be noted in Table 5, the linear spring will present only a suspension natural frequency value, as the non-linear one will present two values.

Figure 6. Linear spring stiffness.

Source: Authors (2023)
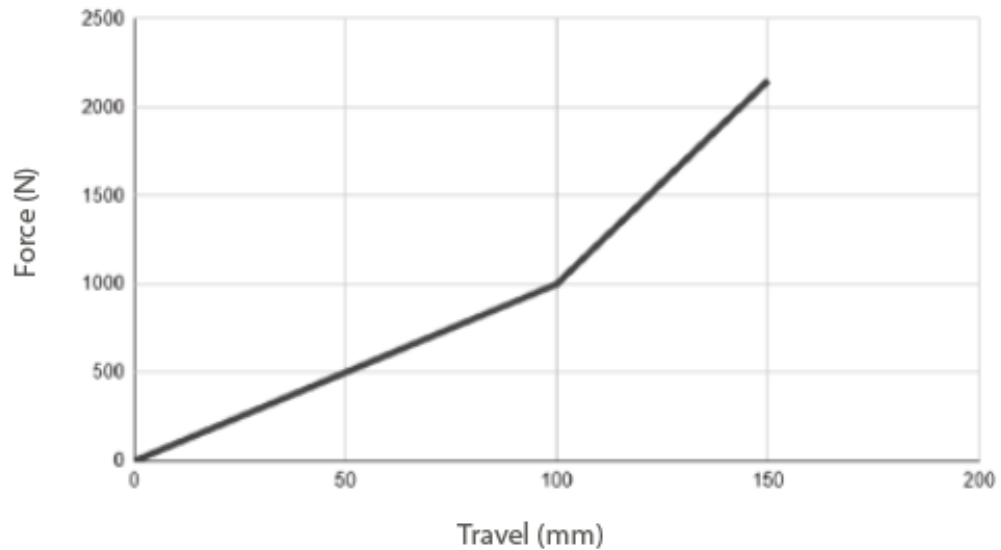
**Figure 7. Double-stage spring.**
Source: Authors (2023)

Table 4 represents the spring stiffness values. In this figure, it appears that natural frequency values are satisfactory for a car with a good performance in competition and, according to Milliken (1995), these values can reach 2.5 Hz, see Table 4.

Table 5. Springs stiffness values.

| Definition	Hardness (N/mm)	Natural frequency (Hz)
Linear spring	10	1.9
First stage spring (equivalent stiffness)	10	1.9
Second stage spring (fully compressed comfort spring)	23	2.3

Source: Authors (2023)

Aiming to set comfort spring height, so its maximum compression occurs during a certain period, it was performed a simulation having small high frequency obstacles to represent the average of excitations suffered by the car on the test track. The maximum displacement was analyzed to avoid spring lock during similar routes that require low compression values (Figure 8). Therefore, the compound was designed to have two-thirds of the total process value, thus the suspension would maintain the natural frequency at acceptable values for comfort, due to low equivalent stiffness in the springs, besides having greater stiffness for bigger obstacles. All simulations were stated through a maximum spring displacement to acquire more precise results.

Figure 8. Graph of spring displacement by time.
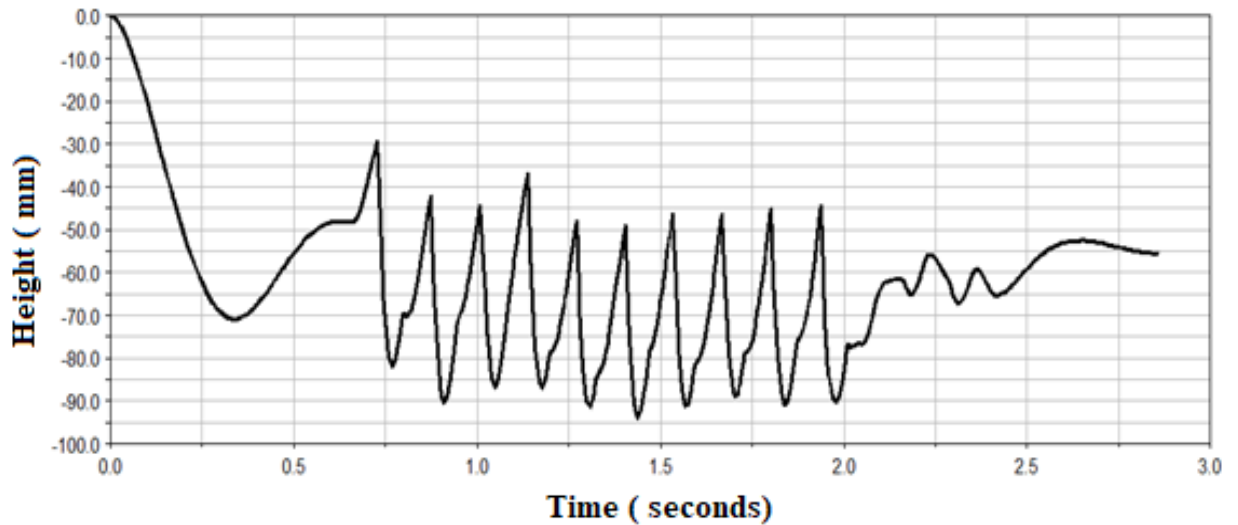
Source: Authors (2023)

To compare the linear spring and the new non-linear spring, acceleration of sprung masses was used as an evaluation criterion in both situations. According to the intensity of these parameters, wear in the suspension elements may occur and a pilot's discomfort (Figure 9). It happens due to low energy absorption of spring in high magnitude obstacles. Therefore, to reach the suspension end, flap absolves all energy, creating bigger accelerations to the sprung masses.
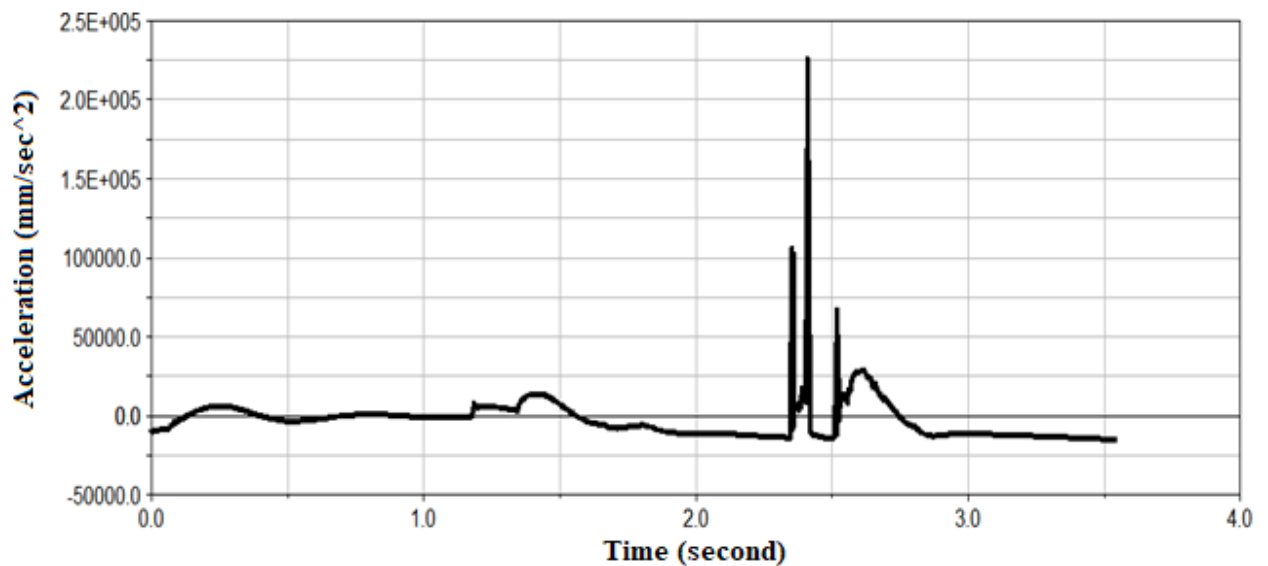
Figure 9. Graph of the prototype's vertical acceleration using linear spring.
Source: Authors (2023)

As discussed in the theoretical framework topic, the analysis of the collected data during the vehicle's simulation, using a double-stage spring and performing a low frequency obstacle, presented excellent results. Figure 10 represents a considerable decrease of the sprung masses' acceleration. This is due to the second stage start that can absorb a higher quantity of energy because of its bigger stiffness. Therefore, a smaller quantity of energy is absorbed by the flap and thus, the decrease in acceleration forced on sprung masses occurs.
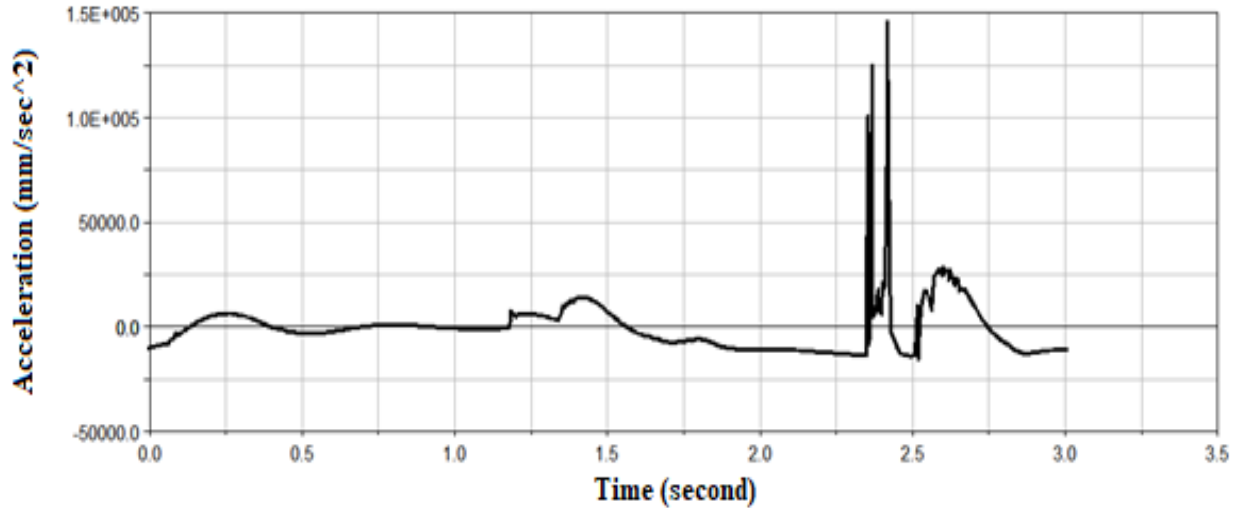
Figure 10. Graph of the prototype's vertical acceleration using double-stage spring.
Source: Authors (2023)

During the simulated route, it was noted the variation of displacement by time through the difference between both curves (Figure 11). It happens after the comfort spring lock and beginning of the second stage with only the resistance spring stiffness.
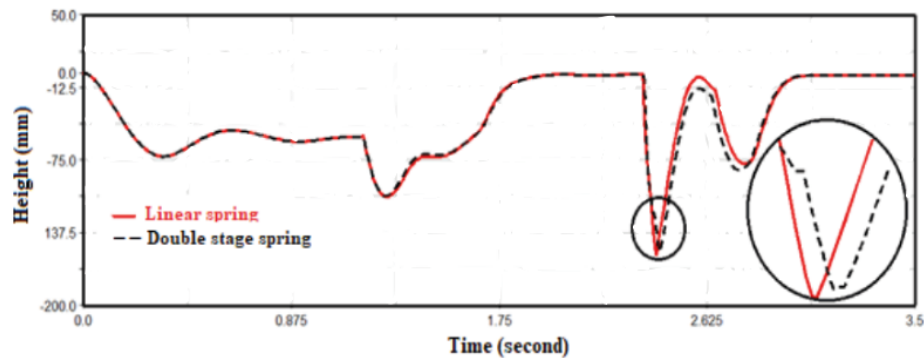

Figure 11. Comparative graph between springs displacement by time.
Source: Authors (2023)

5. Conclusion

According to the results of the double-stage design study for the Baja prototype and with linear spring comparison, it is concluded that the spring manufacturing material choice must be done thoroughly, once some properties can infer different parameters as elastic limit, fatigue, tensile strength. Besides, it can directly infer manufacturing, performance and durability of the equipment.

Besides, Greater values for the spring stiffness can allow higher energy absorption and avoid the vehicle suspension limit travel. However, it may occur an increase of natural frequencies, creating a discomfort for the pilot.

Then, double-stage springs need longer suspension travel and they are used in off-road routes, where they suffer excitations of different magnitudes, beyond having a better ability to absorb energy when submitted to bigger efforts. Therefore, when there is a decrease of velocity, there is more comfort and less wear on the vehicle's equipment.

The springs used in the vehicle's comfort (first stage) must be designed for a larger route regarding the total displacement. Hence, as the natural frequency stiffness is smaller, the prototype will present more comfort and performance to go through obstacles. The resistance springs (second

stage) need to perform a smaller displacement, or a significant increase of the natural frequency will occur during small obstacles, compromising the driving.

Acknowledgements

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