v. 3, n. 5, 2023

Journal of
Engineering
Research

OPTIMIZATION METHODOLOGY OF OFF-ROAD VEHICLE'S LATERAL DYNAMICS DESIGN

Matheus de Souza Nascimento
Universidade Federal do Vale do São Francisco (UNIVASF) – Campus Juazeiro; Mechanical Engineering Course

José Bismark de Medeiros
Universidade Federal do Vale do São Francisco (UNIVASF) – Campus Juazeiro; Mechanical Engineering Course

Ramon Magalhães Brasileiro
Universidade Federal do Vale do São Francisco (UNIVASF) – Campus Juazeiro; Mechanical Engineering Course

Abstract: The growth of the *off - road vehicles market* is a direct result of the search for the practice of motorsports or, even, of the applicability in economic activities that need to be carried out in irregular terrains. This way, it is fundamental that the Lateral Dynamics project provides the intended behavior, in addition to allowing adequate interaction between the vehicle and the external forces acting. The present work is based on the development of a sequential methodology of dynamic-lateral design. For this, an algorithm is developed in the MATLAB® software, being integrated to the modeFRONTIER® software, in order to vary geometric and elastic parameters, intrinsic to the vehicle, obtaining optimized *setups*. This impacts on the reduction of time and costs in the conception of a vehicle, making the project more accurate, since fewer prototypes will be developed and tested. Based on this, several *setups* can be obtained and, subsequently, the one that best suits the previously defined objectives is chosen. Finally, the present work obtained a model that guarantees that 65% of the lateral load transfer is on the rear axle, promoting an *oversteer behavior*. Furthermore, with a *roll rate* below 5 °/g, it is possible to obtain a vehicle with satisfactory stability.
Keywords: Lateral Dynamics, Off-Road Vehicle, Optimization, Lateral Load Transfer, Roll Rate.

INTRODUCTION

The integration of vehicle projects aimed at greater performance is a factor present in the scope of the automobile industries, with increasingly competitive demands. This is due to the fact that such machines facilitate the performance of complex activities or, simply, the practice of *motorsports* that significantly move the *off-road vehicle market*. The MORDOR INTELLIGENCE (2020), states that this global market, by the year 2026, must reach 18 billion dollars, surpassing the investment of 14 billion dollars in the year 2021. This means that project methodologies must be increasingly more refined and need to rely on increasingly accurate tools, in order to meet the previously stipulated objectives and to obtain the target performance.

It is essential that *off-road vehicles* can overcome any type of obstacle or irregularity, always guaranteeing the safety and dynamic performance desired by the user. The design of the suspensions is based on the choice of the *setup* of geometric and elastic parameters, that is, those that denote how the forces and moments tend to act on the system. In addition, these are also essential for gains in acceleration during straight-line or cornering, since, primarily, they must guarantee the optimization of the operation of each tire. It is valid to state that the dynamics of a vehicle is basically based on the interaction between the tires and contact patch and on the spatial position in which the CG (Center of Gravity) is located, since this denotes how the forces occur and the acting moments, with an interdependence between both.

It is intuitive to see that there is a predominant analysis of how forces are induced during tire operation. Prescribe Milliken *et al.* (1995, p. 14): "The tires also provide the forces used to control and stabilize the vehicle and to resist external disturbances from the road and wind". However, there is a scarcity of technical information about tires, since they are complex elements. This is a factor of great negative weight, especially for *off-road companies* that have fewer resources, since the entire dynamic model could be more accurate through the use of such data. Anyway, Lateral Dynamics can be analyzed through simplified mathematical formulations, but which still provide a satisfactory representation of reality. Numerical optimization methods also allow such formulations to be analyzed simultaneously, in several domains, obtaining more efficient *setups*.

OBJECTIVES

- Elaborate a methodology for the Lateral Dynamics design of an *off-road vehicle*, based on a logical sequencing;
- To develop an algorithm, using the MATLAB® software, that allows to include the main mathematical formulations that describe the Lateral Dynamics of a vehicle;
- Adopt the integration of such formulations, to a numerical optimization software (modeFRONTIER®), also as a way to increase the domains of *inputs*;
- Analyze the *outputs* obtained and select certain optimized geometries, aiming to choose the one that best suits the fulfillment of the desired behavior.

THEORETICAL FOUNDATION

The dynamics of a vehicle is based on the freedom presented by its reference axes, accounting for translation or rotation movements, in which inertial forces tend to arise as a result of such movement conditions. The positioning of the coordinate system can be given with the origin located at the center of gravity of the vehicle. In such a way, as prescribed by Milliken *et al.* (1995, p.114): "The properties of inertia (moments and products of inertia) remain constant in relation to this set of axes, but must vary if referenced to a set of axes fixed to the ground, for example ".

Basically, such axes are based on the three dimensions of Euclidean three-dimensional space: longitudinal, which is located in the largest dimension of the vehicle and parallel to the ground; lateral (or transverse), being also parallel to the ground, but perpendicular to the longitudinal dimension; and the vertical, characterized as that which is perpendicular to the ground and orthogonal to the other coordinate axes. By analogy, rotational movements are also defined, called *pitch, roll* and *yaw,* corresponding to the rotations performed on round of the longitudinal, lateral and vertical axis, respectively. Figure 1 illustrates the positioning of the coordinate system and the respective senses and directions agreed for each axle, through SAE regulation.

By d'Alembert's Principle, the forces of inertia appear, in which they present the same denominations imposed to the respective coordinate axes: longitudinal, lateral and vertical forces. In short, longitudinal forces are associated with reactions between the ground and the tire's contact patch, arising from accelerating or decelerating the vehicle. Analogously, lateral forces also arise at such contact patch, but are linked to the realization of paths along a turn radius. In the case of vertical forces, the forces essentially come from irregularities that cause elastic deformations in the tires.

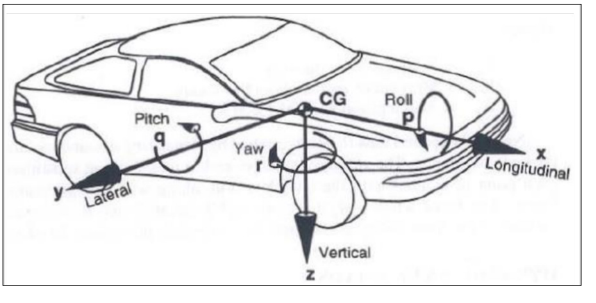

Figure 1. Standard coordinate axes for a vehicle, according to SAE, highlighting the respective rotations. Source: Gillespie (1992).

It is essential to use elastic elements, such as compression springs with a certain stiffness (*k*), to store kinetic energy and consequently control the relative displacement between the wheels and the chassis. The tire is also considered a type of elastic element, in which it can be treated as a set of springs and is endowed with an equivalent stiffness constant (*kt*). Besides these, the damping elements, denoted essentially by the viscous dampers (*c*), are components responsible for the dissipation of part of the energy, at each executed cycle. Tires, in practice, are equipped with such damping, but dynamic models generally neglect this parameter.

> The use of springs to allow the vertical movement of the wheels in relation to the chassis, on rough roads, is reminiscent of horse-drawn vehicles. This method of controlling displacement was adopted early in the history of the automobile. The damper was subsequently introduced between the sprung and unsprung masses to suppress oscillation and control the movement of the sprung mass due to longitudinal and lateral acceleration. (MILLIKEN *et al.*, 1995, p. 781).

It is common to subdivide the total mass of the vehicle into two categories: sprung mass, which includes the components that are above the suspension, such as the chassis and everything inside it; and the unsprung mass, which likewise comprises components below the suspension, such as those that are coupled to the wheel only. However, it is important to highlight that there are components that are partially located in both situations, such as the suspension control arms, the springs-shock absorbers, the axle shafts and any other element that is simultaneously connected to the two subdivisions. In Figure 2, a diagram of the arrangement of such masses and elasticity and damping elements can be seen.

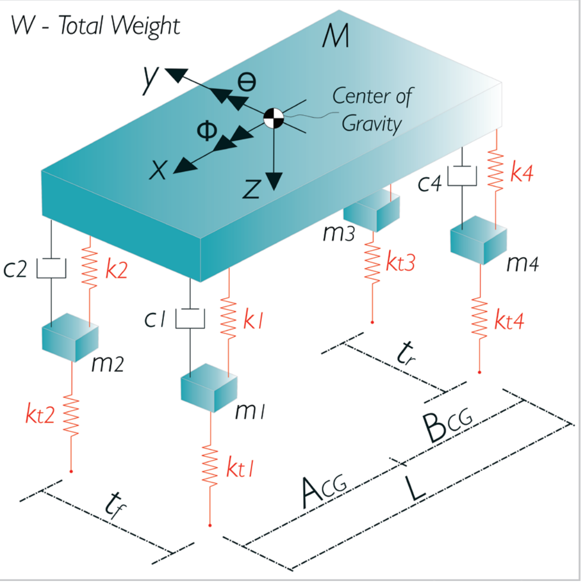

Figure 2. Representation of a vehicle model, with the respective axes, showing the components that influence its dynamics. Source: Adapted from Nicolazzi *et al.* (2012).

In addition to the elastic and damping elements mentioned above, the geometric parameters also influence the dynamics of the vehicle. This means that the moments that make the chassis roll are dependent on the distance between the wheels, either frontally or laterally, in addition to the positioning of the center of gravity. This distance is called track width, divided into front (t_f) and rear (t_r); and can be called wheelbase (L), when referring to the side view.

Another extremely important geometric parameter is called the installation ratio (IR). In order for the spring-shock absorber system to act collinear to the vertical axle, the stiffness constant is geometrically corrected by the installation ratio (IR), implying an equivalent stiffness, called *wheel center rate* (k_w) and which is calculated by Equation 1, as prescribed by MILLIKEN *et al.* (1995, p. 114). In practical terms, it is equivalent to the behavior of the spring-shock absorber system, performed on the vertical axle. The same applies to damping (c_w), given Equation 2.

Journal of Engineering Research ISSN 2764-1317 DOI 10.22533/at.ed.317352310022

$$k_w = k \cdot (IR)^2 \quad (1)$$

$$c_w = c \cdot (IR)^2 \quad (2)$$

LATERAL DYNAMICS

Carrying out movement along any axis of the vehicle is always accompanied by load transfer phenomena, that is, inertia causes certain tires to present reductions in the acting loads, so that others can receive such forces. Cornering movements are endowed with lateral load transfers, in which the tires external to the curves present significant increases in the acting vertical forces. At the given moment when the vehicle makes a curve, whatever its turn radius, friction forces appear at the contact patch of the tire with the ground, allowing the reduction of its slip and providing that the movement is carried out with a given level of security and stability. These friction reactions are called lateral forces and are associated with the mechanisms that generate forces on the part of the tyres: adhesion and/or hysteresis.

By D'Alembert's Principle, inertial forces must arise to provide dynamic equilibrium, in which, with the hypothetical situation of non-slip of the tires, there is equilibrium between lateral forces and inertial forces, with zero algebraic sums. This way, Lateral Dynamics arises, as a way of analyzing and equating the phenomena arising from the performance of cornering. As a result, vehicles tend to present a reduction in longitudinal acceleration and, consequently, an increase in lateral acceleration. It is intuitive to realize that at the apex of cornering, the vehicle presents only lateral acceleration and, after resuming straight-line movement, longitudinal acceleration increases again.

In this context, the tire undergoes lateral deformations quantified by *Slip Angle*, which are associated with the *grip generation,* that is, the ability to generate lateral force, during movement, based on the vertical force acting on the tire. It is known that the tire has a certain maximum capacity to generate *grip,* which is subsequently accompanied by a loss of *grip* and consequent slip of the tyres. This means that the vehicle tends to lose its stability and rotate in *yaw.* This is something similar to the *stall* that can occur in aircraft and these tend to lose lift. This way, high-performance vehicles always seek to exploit the maximum generation of lateral force, balancing the *trade-off* between performance gain and stability loss, acting in regions prior to the loss of tire *grip*. However, detailed analyzes involving the *Slip Angle* are more restricted to tire data which is generally not available.

The referred load transfer can be subdivided into two segments: the lateral load transfer of unsprung masses and the lateral transfer of sprung masses. The first is associated with components that are "below suspension", presenting lower inertia values and being endowed with a CG of the unsprung masses (Zu), where inertial forces act. The second refers to the load transferred by the chassis and by components that are associated with it. It can also be subdivided into two categories: geometric lateral transfer and elastic lateral transfer. In Figure 3, it is possible to visualize how the Lateral Dynamics are dealt with.

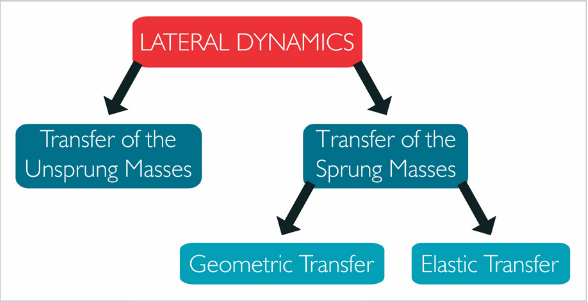

Figure 3. Subdivisions of the Lateral Dynamics.
Source: Own authorship (2022).

The transfer of unsprung masses does not depend on springs or shock absorbers, as seen in Equation 3, but only on certain geometric parameters: track width (t) and height of the CG of unsprung masses (Z_u); and of trivial inertial quantities: unsprung weight (W_u) and lateral acceleration of the vehicle (A_y). It is important to point out that such parameters must be treated together for each given vehicle axle, that is, separately for the front and rear. As for this referred CG, a good approximation can be made considering it at the height of the geometric center of the tire, that is, the value is approximated to the radius of the tire.

$$\Delta W_u = \frac{W_u\, A_y}{t}\, Z_u \tag{3}$$

As for the transfer of sprung masses (ΔW_{Sg}), Equation 4, the geometric component presents a direct relationship with the wheelbase and the distance that the given axle (x) holds in relation to the position of the vehicle's CG. This is denoted by a length parallel to the vehicle's longitudinal axis, as seen in Figure 2, with A_{CG}, relating to the front axle, and $B_{CG,}$ referring to the rear axle. In addition, consequently, is considered the sprung weight (W_S). Finally, there is also a significant height dependence on the suspension geometric roll center (Z_{rc}), that is, a geometric center where each suspension, front or rear, tends to rotate in *roll*.

$$\Delta W_{Sg} = \frac{W_S\, A_y}{t}\, \frac{L-x}{L}\, Z_{rc} \tag{4}$$

In this context, the line passing through the front and rear roll centers, as seen in Figure 4(*a*), is called the roll axle and denotes how the sprung mass rotates in *roll*. The distance between the CG of the sprung masses and such a roll axle provides the rise of the roll arm. With the action of inertial force, it is possible to have a moment of rotation (M_ϕ) proportional to this roll arm and that can also be increased with the action of the sprung weight force, as seen in Figure 4(*b*).

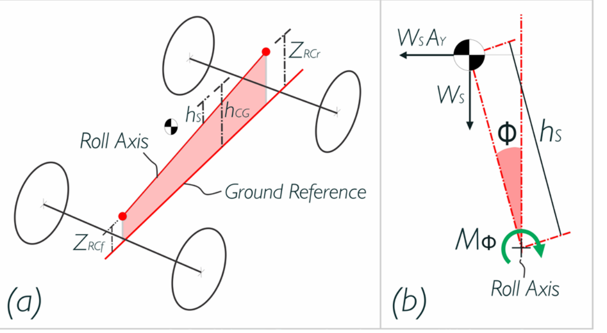

Figure 4. Geometric parameters, showing the roll centers. Source: Adapted from Gillespie (1992).

The geometric roll center has a great influence on vehicle behavior, also because it limits how much a given tire tends to lift and, logically, lose contact with the ground, denoting the so-called Jacking Effect. How does prescribe NICOLAZZI *et al.* (2012, p. 217 and p. 220), the heigth of the CG of the sprung masses (h_{CGs}) and, consequently, the length of the roll arm (h_s), can be obtained by Equations 5 and 6, respectively.

$$h_{CGs} = \frac{W\, h_{CG} - (W_{uf} + W_{ur})\, r_d}{W_S} \tag{5}$$

$$h_S = h_{CGs} - \frac{Z_{RCf}\, A_{CG} + Z_{RCr}\, B_{CG}}{L} \tag{6}$$

The other component of the lateral load transfer of the sprung masses corresponds to the elastic transfer, that is, the one in which the elements endowed with stiffness act to increase the vertical force of the tires that are receiving the transferred loads. Therefore, one must associate the stiffness of the *wheel center rate* (k_w) and the tire (k_t), in order to find an equivalent magnitude that considers

Journal of Engineering Research ISSN 2764-1317 DOI 10.22533/at.ed.317352310022

the influence of both, resulting in the *ride rate,* according to Equation 7. In this context, the restoring forces arising from elasticity tend to provide a stiffness that acts in opposition to the movement of *roll* of the sprung mass, accumulating the energy that is transferred during the cornering moviment. Such a restorative parameter is called a *roll stiffness* (K_ϕ), being given by Equation 8, for the case of independent suspensions, where $K_{\phi AR}$ corresponds to the anti-roll bar (if any).

$$h_{rr} = \frac{k_w k_t}{k_w + k_t} \quad (7)$$

$$K_\phi = \frac{1}{2} k_{rr} t^2 + K_{\phi AR} \quad (8)$$

In addition to the lateral load transfer, the phenomenon of chassis roll also occurs, as seen in Figure 5. When acting on the roll arm, the inertial forces provide the *roll* movement. Therefore, one can analyze the *roll rate* ($\frac{\phi}{Ay}$), Equation 9, which relates the *roll angle* (ϕ) to the lateral acceleration of the vehicle, that is, for each value of this acceleration, there is a corresponding angle. This parameter is fundamental for choosing the *setup* of the spring and tire stiffness constants, providing better rotation control imposed on the vehicle chassis and impacting greater stability and comfort for users.

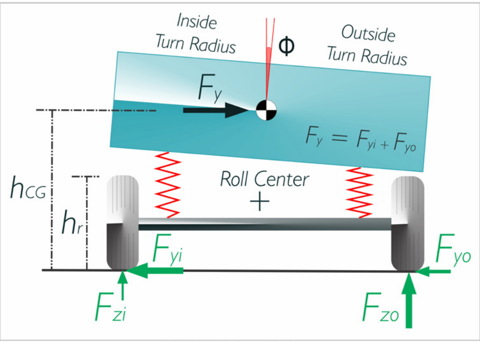

Figure 5. Representation of lateral load transfer and chassis roll, in front axle, during a cornering. Source: Adapted from Gillespie (1992).

In this context, the elastic lateral transfer (ΔW_{Ss}) is passive of a *setup* negotiation involving the stiffness constants, since its increase occurs through the increase of such constants, as can be seen in Equation 10. In question, the index "x" (in red), indicates the *roll stiffness* of the axle to be treated. It is well known that chassis roll and the load that is transferred, even though they are completely different phenomena, are functions of the roll arm. Therefore, there is a *trade-off* between these magnitudes, since the increase in the roll arm provides an increase in the transferred load, but there is also an increase in chassis roll and the consequent loss of stability and comfort.

> The *ride rate* and *roll rate* have a significant effect on performance during cornering, as they directly affect the distribution of normal forces on the tires and, consequently, the lateral forces, from the tires, that arise in each corner. Unfortunately, the car designer or chassis controller does not have direct control over *suspension travel* and *roll rates*, only the individual suspension elements: *links,* springs and anti-roll bars. (MILLIKEN *et al.*, 1995, p. 580).

$$\frac{\phi}{A_y} = \frac{W_s h_s}{K_{\phi f} + K_{\phi r} - W_s h_s} \quad (9)$$

$$\Delta W_{Ss} = \frac{W_s A_y}{t} h_s \left(\frac{K_{\phi x}}{K_{\phi f} + K_{\phi r}} \right) \quad (10)$$

All design involving Lateral Dynamics is responsible for predicting the behavior during cornering. This can denote two types of behavior: *understeer,* in which the vehicle tends to direct the rear axle towards the center of the curve and increase the design turn radius; or *oversteer,* in which the front axle is directed towards the center of the curve and there is a tendency to reduce this turn radius. Commonly, the first behavior is imposed on

Journal of Engineering Research ISSN 2764-1317 DOI 10.22533/at.ed.317352310022

passenger cars, leaving the second as more applicable to higher performance vehicles. As cast by NICOLAZZI *et al*. (2012), vehicles that have the CG closer to the front axle, have a greater tendency to *understeer behavior*, unlike the CG located closer to the rear, which causes a greater propensity to *oversteer*. This is purely logical, since such a location is essential to delimit the load transfer.

The lateral transfer of total load is always constant for a given *setup*, implying that each axle has a percentage of this parameter. This way, *oversteer* vehicles have CG located closer to the rear axle; higher roll center at the rear relative to the front axle; and higher *roll stiffness* values. The greater the load transferred, on the same given axle, the greater the tendency for the inner tire to slip into the curve, the one prone to greater *grip generation*, since they have a smaller turn radius. Therefore, the tires outside the curve present greater vertical loads, causing a reduction in the maximum generation of lateral forces by the inside tires, as seen in Figure 5.

Oversteer behaviors come from rear axles with greater lateral load transfers, as there must be greater *slip* of the rear tires, so that the front axle can be directed to the center of the curve. The analogue occurs for obtaining the *understeer behavior*. In general, it is important that the geometric parameters are perfectly contained in the longitudinal plane of the vehicle, to guarantee equivalent transfers to both sides. The lateral transfer of the total load for each axle (ΔW_x), can be given by Equation 11, in which there is an algebraic sum of all components for this given axle (x). Finally, we arrive at the percentage corresponding to each axle (%ΔW), Equation 12, in relation to the total value. In question, this is one of the most important parameters to analyze the behavior of the vehicle, on the cornering.

$$\Delta W = \Delta W_u + \Delta W_{Sg} + \Delta W_{Ss} \tag{11}$$

$$\%\Delta W = \frac{\Delta W_x}{\Delta W_f + \Delta W_r} \tag{12}$$

As the alteration of the geometric parameters, commonly, is restricted after the vehicle is manufactured, it is important to point out that the manipulation of the elastic component is the simplest factor for the alteration of the dynamic-lateral *setup*. This way, it is less complex to change the spring and tire constants, in addition to the possibility of installing anti-roll systems. One of these is the anti-roll bar, which has the functionality to increase the *roll stiffness* of a given axle, in addition to contributing to the reduction of chassis roll. SEWARD (2014, p. 111) recommends that the anti-roll bar not contribute more than 50% of the *roll stiffness*, in order to prevent independent suspensions from behaving like the rigid suspensions type.

METHODOLOGY

The present work is based on the elaboration of a methodology that allows the integration of the main parameters involved in the design of the vehicle dynamics applicable to the *off-road* scope. Such project sequence, part of the elaboration of an algorithm in MATLAB®, version R2021b, in which the mentioned equations are introduced. Subsequently, there is integration with modeFRONTIER®, version 2020 R3, in order to define *input ranges and* analyze the best *setups* obtained. The first software contains the mathematical equations to be worked on and the second has the objective of optimizing the *outputs*, such as load transfer, as seen in Figure 6. Thus, it is possible to add a parameter maximization or minimization command, in addition to constraint it from reaching certain values.

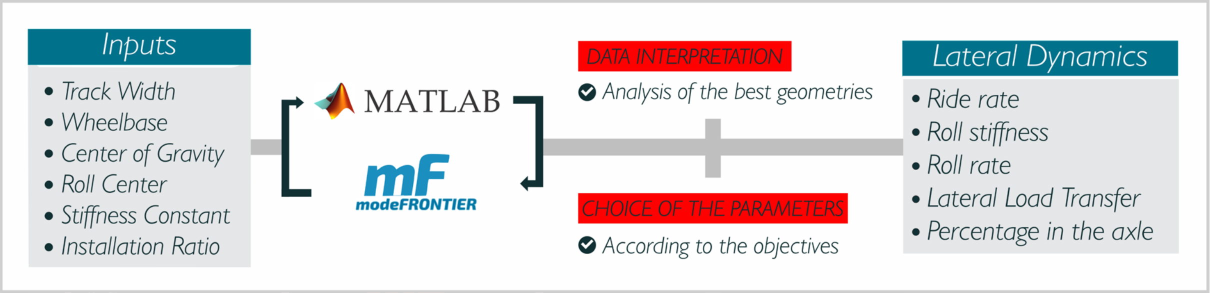

Figure 6. Flowchart of integration between the *software*, highlighting the main *inputs* and *outputs* analyzed. Source: Own authorship (2022).

CHOICE OF OPTIMIZATION ALGORITHM

The integration between the both softwares is fundamental to parameterize and optimize the dynamic-lateral behavior. This allows the reduction of design time; greater accuracy on the calculations performed; it allows showing the "sensitivity" that each *output* presents, in relation to the domain; in addition to being an application of knowledge management, for the evolution of subsequent prototypes. The optimization can be carried out by several algorithms available in modeFRONTIER®, in which MOGA-II - Adaptive Evolution was chosen. This is configured as a multiobjective genetic algorithm, that is, it is a class of evolutionary algorithm, which uses techniques based on evolutionary parameters, which seek to carry out optimizations based on "learnings" from previous iterations.

DEFINITION OF *INPUT RANGES*

Ranges applicable to the domains of each *input* parameter can be defined, using values previously analyzed or used, in addition to literature recommendations, valuing tangible values for the manufacture of the *off - road vehicle*. This way, the present work has as its starting point the use of values gradually prioritized along the 6 *off-road vehicles*, of the baja SAE type, conceived by the Baajatinga Baja SAE Team. The provision of such parameters is essential to have an even more effective starting point, as other geometries could be previously tested.

In Table 1, it is possible to visualize the main parameters used as *input* for the multiobjective analysis, highlighting the respective *ranges* and interval steps adopted for this purpose. All values are inserted in modeFRONTIER®, in which the calculations are performed by MATLAB®, based on the previously mentioned equations and which consider the vehicle in steady-state, without taking into account the non-trivial data of the tires. It is important to point out that we want to obtain a Lateral Dynamics that provides the *oversteer behavior*. Even so, in the present work, the use of the anti-roll bar was not considered.

Also, it is highlighted that the values of the stiffness constants of springs, analyzed, are still subject to *wheel center rate* correction. Therefore, constant values were defined for the installation ratios, as a way of simplifying the analysis and enabling a better integrated visualization of the considered parameters. Even with a larger sample of parameters to be varied, the final analysis can become somewhat complex and more difficult, for the selection of a geometry that meets the previously defined objectives.

Input parameter	Acronym adopted	*Range* adopted	Adopted *step*
Vehicle's CG height	H_cg	500 – 600 mm	5 mm
Fraction, from the wheelbase, of the longitudinal distance between the front axle and the vehicle's CG	P_r	0.52 - 0.60	0.01
Wheelbase	l	1300 – 1500 mm	10 mm
Front track width	t_f	1200 – 1400 mm	10 mm
Rear track width	t_r	1100 – 1300 mm	10 mm
Front roll center height	Zrc_f	150 – 300 mm	5 mm
Rear roll center height	Zrc_r	200 – 350 mm	5 mm
Front springs stiffness	Ks_f	20000 - 30000 N/m	200 N/m
Rear springs stiffness	Ks_r	25000 - 35000 N/m	200 N/m
Front tires stiffness	Kt_f	45000 - 65000 N/m	500 N/m
Rear tires stiffness	Kt_r	55000 - 75000 N/m	500 N/m

Table 1. *Input* parameters for numerical optimization, looking for the *oversteer behavior,* showing its adopted *ranges* and interval *steps*. Source: Own authorship (2022).

SETUP OF MODEFRONTIER®

1000 generations were adopted, that is, this value corresponds to the number of iterations performed with "learnings", contained in the *ranges* specified in Table 1. Thus, it is possible to notice that a given value, for a given parameter, can be repeated more than once turn. Even with this factor, it is fundamental that this value be considerable, in order to increase the probability of a certain domain term being used as *input* and, possibly, being available as a possible parameter candidate for the vehicle. Therefore, the use of smaller steps, along the *range,* is also a way to increase this probability. Finally, some geometries that reach the previously defined goals can be chosen and, later, the best one can be selected.

DEFINITION OF *OUTPUTS* AND THEIR RESPECTIVE OPTIMIZATIONS

Output selection stage is based on fundamental parameters to analyze the Lateral Dynamics of the vehicle and, specifically, to guarantee the desired behavior. In this context, Table 2 highlights these parameters and expresses the type of optimization applied, in which one can visualize minimization trends for the front axle and maximization trends for the rear axle. This is exemplified by the objective of obtaining an *off-road vehicle* with *oversteer behavior,* increasing lateral transfer on the rear axle, since many of these are not equipped with the differential mechanical component and require greater maneuverability during cornering by making curves with smaller turn radius.

Output parameter	Acronym adopted	Optimization
Roll stiffness	Kfi_f	Minimization
	Kfi_r	Maximization
Lateral load transfer of sprung masses: Elastic component	X1_f	Minimization
	X1_r	Maximization
Lateral load transfer of sprung masses: Geometric component	X2_f	Minimization
	X2_r	Maximization
Lateral load transfer of unsprung masses	X3_f	Minimization
	X3_r	Maximization
Percentage of lateral load transfer on rear axle	DeltaW_r_percent	Maximization
Roll rate (*roll gradient*)	Kgr	Minimization

Table 2. Main optimized *output* parameters, highlighting the respective optimizations. Source: Own authorship (2022).

RESULTS AND DISCUSSIONS

The analyzes took place without the occurrence of errors, with a duration of 98 minutes. This way, it was possible to obtain 1000 different configurations, constituted by the set of 11 parameters mutually analyzed, that is, each configuration presents a given value of each considered domain. Thus, even because the present project deals with a dynamic-lateral design methodology, one can aim to choose a *setup* that provides a percentage of lateral load transfer, on the rear axle, of around 65%. In addition, the use of a *roll rate* of up to 5.0 °/g is also considered, following MILLIKEN *et al.* (1995, p. 584), which characterizes it as a parameter of a "firm" and sporty vehicle. It is important to point out that these parameters can be found in *off-road vehicles* and tend to guarantee the referred *oversteer behavior.*

In this context, such objectives outlined for the present work can be better achieved by maximizing the lateral load transfer present on the rear axle and by minimizing the *roll rate.* In this case, the increase in *roll stiffness* and the rear roll center height are factors that contribute both to the *roll rate* reduction and to the increase in *oversteer behavior.* Figure 7 shows the dispersions obtained for the elastic parameters, showing the achievement of the desired percentage.

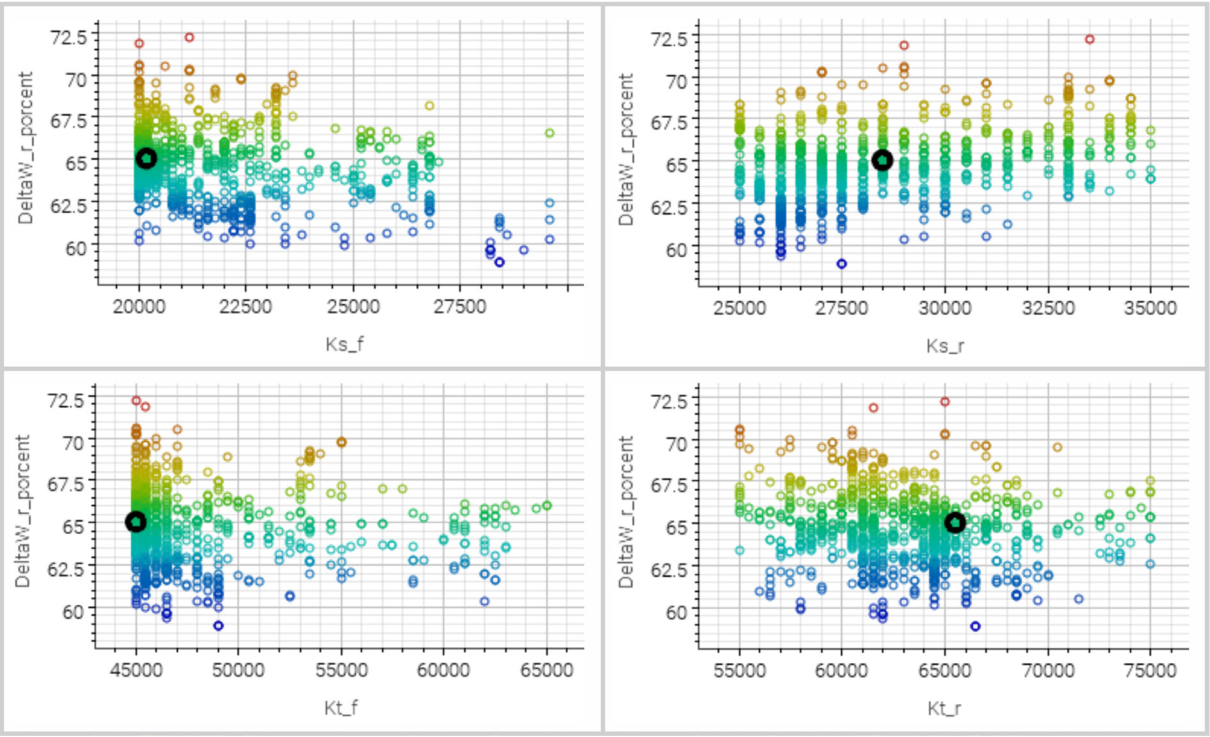

Figure 7. Dispersions of the used elastic *inputs*. Source: Own authorship (2022).

It is possible to notice the trend of greater concentration of *inputs* on the left side of the graph, for the elastic parameters of the front axle, and a greater concentration in the center of the graph, for the rear axle. This is explained by the minimization imposed on the front axle *outputs* and, similarly, by the maximization imposed on the rear axle. As for *inputs* of a geometric nature, the values can be seen in Figure 8.

It is possible to notice a greater concentration of geometric parameters in the central regions of each graph. This is of paramount importance to ensure greater vehicle stability, avoiding considerable chassis roll, concomitantly with the desired load transfer. With the use of a 100 millimeters larger front track width and a 10 millimeters smaller roll center height, the front axle is strictly designed to guarantee a smaller lateral load transfer and, consequently, a greater *grip generation* in the inner tire at the curve. It is essential to highlight that the choice of the rear track width and roll center also sought to promote the reduction of the Jacking Effect.

The large concentration of *inputs* in the highest height values of the front roll center can be explained by the tendency to reduce the roll arm, since this impacts on the minimization of the *roll rate,* as previously programmed. Another important point to highlight is the spatial position of the CG, as it is significantly closer to the rear axle, around 45% of the wheelbase value, aiming at *oversteer behavior*. In addition, it is common for *off-road vehicles* to have higher CG heights, due to the fact that they need greater ride height, in relation to the ground, that is, to avoid collisions with obstacles or irregularities present along the terrain. As for the most variable points, it is worth noting that they do not come close to the stipulated objectives, ending up prioritizing only certain *outputs.* Table 3 shows the *inputs* present in the adopted *setup.*

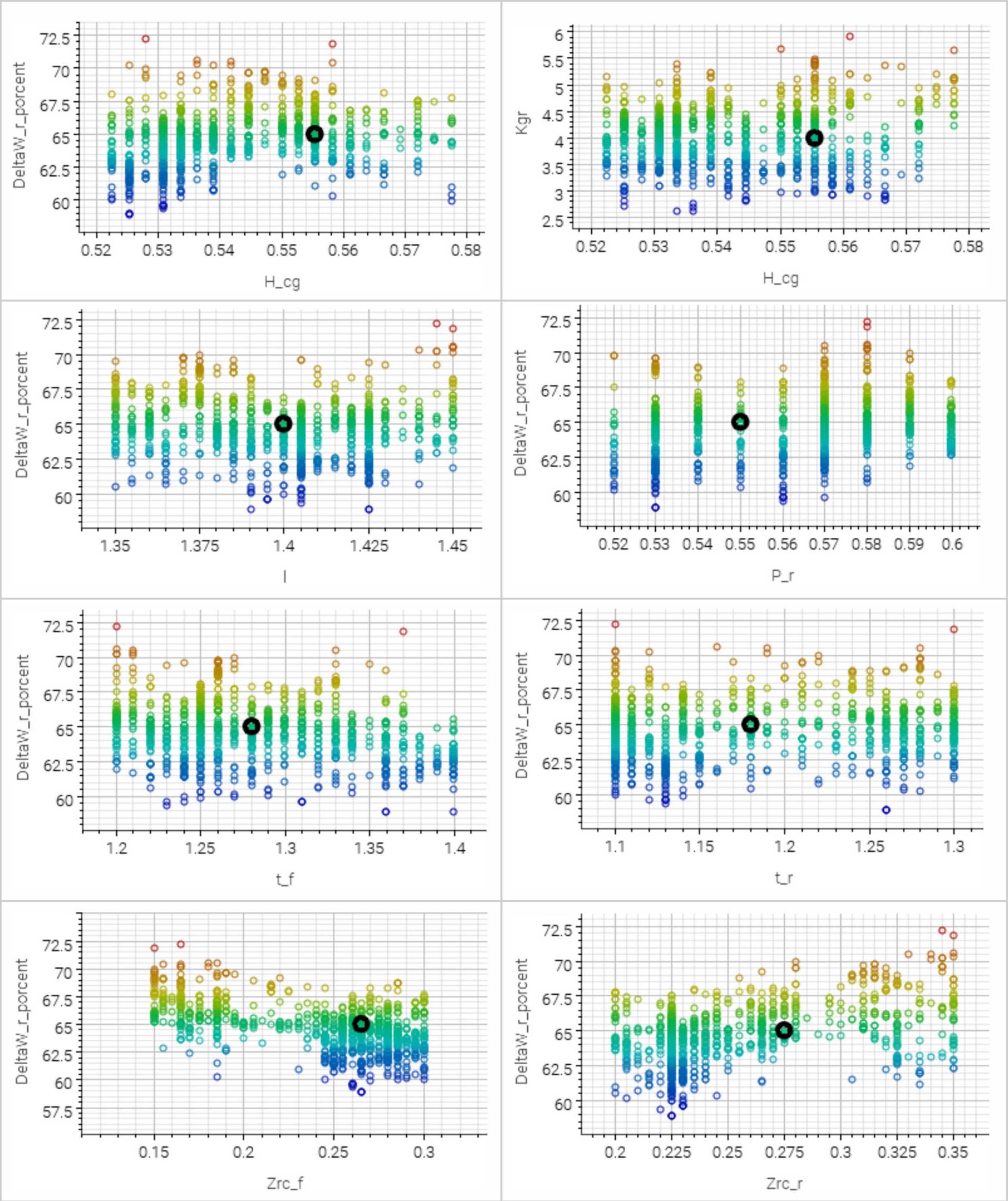

Figure 8. Dispersions of the used geometric *inputs.* Source: Own authorship (2022).

Journal of Engineering Research ISSN 2764-1317 DOI 10.22533/at.ed.317352310022

Input parameter	Acronym adopted	Adopted *setup* value
Vehicle CG height	H_cg	55.60 mm
Fraction, from the wheelbase, of the longitudinal distance between the front axle and the vehicle's CG	P_r	0.55
Wheelbase	l	1400 mm
Front track width	t_f	1280 mm
Rear track width	t_r	1180 mm
Front roll center height	Zrc_f	265 mm
Rear roll center height	Zrc_r	275 mm
Front springs stiffness	Ks_f	20200 N/m
Rear springs stiffness	Ks_r	28500 N/m
Front tires stiffness	Kt_f	45000 N/m
Rear tires stiffness	Kt_r	65500 N/m

Table 3. Adopted *inputs*, after choosing the optimized geometry. Source: Own authorship (2022).

CONCLUSIONS

It is possible to conclude that the present work achieved all the previously stipulated objectives, since an effective methodology was elaborated for the integrated and optimized development of the lateral dynamic parameters of an *off - road vehicle.* Based on the use of MATLAB®, to elaborate the algorithm containing the design equations of the Lateral Dynamics, based on the stationary regime, modeFRONTIER® allowed the elaboration of the analyzed domains, in addition to the optimization of the obtained *outputs.* This methodology is essential to obtain greater accuracy and practicality during the design phase, in addition to constituting an important knowledge management tool for the development of subsequent prototypes.

This way, several *setups* could be generated and mutually analyzed, promoting the choice of the one that best suited the concentration of 65% of the lateral transfer on the rear axle, in addition to a *roll rate* below 5°/g. This is an indication that future improvements, applied to the present methodology, may present metrics for the prior selection of a given number of *setups.* This, in order to select, later, the best or the one that best suits the objective. Furthermore, the stiffness constants of tires, or of air springs, can be analyzed based on the pressures used in the given elements. Therefore, it is concluded that the present work constitutes an important tool for the sequential and integrated design of the parameters involving the Lateral Dynamics, obtaining tangible *inputs* for the manufacture of the vehicle.

REFERENCES

GILLESPIE, Thomas D. Fundamentals of Vehicle Dynamics. Society of Automotive Engineers, Inc., Warrendale, USA, 470 p., 1992.

MILLIKEN, WF; MILLIKEN, DL Race Car Vehicle Dynamics. 1st ed. SAE International, Pennsylvania, USA, 890 p., 1995.

MODEFRONTIER. Algorithm Guide: Multi-Objective genetic Algorithm II. Version 2020R3. [S.1.]: ESTECO, 2020.

MORDOR INTELLIGENCE LLP. Off-road Vehicle Market. Available at: < https://www.reportlinker.com/p06079797/Off-road-Vehicle-Market-Growth-Trends-COVID-19 Impact-and-Forecasts.html?utm_source =GNW >. Accessed on: Aug. 2021.

NICOLAZZI, LC; PINK AND.; LEAL, LCM An Introduction to Quasi-Static Modeling of Motor Vehicles. Internal publication of the Department of Mechanical Engineering at UFSC, Florianópolis, 347 p., 2012.

SEWARD, D. Race Car Design. Palgrave Macmillan, London, United Kingdom, 276 p., 2014.