DESIGN, MODELING, SIMULATION, AND EXPERIMENTAL TESTS OF A QUARTER CAR SUSPESNSION PROTOTYPE

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Abstract. *In current scenario of automotive industry, simulation software and virtual prototypes have been used to considerably reduce time and cost associated with the development of new vehicles. The use of simulation software based on multibody and finite element methods helps the engineer in finding design solutions as well as evaluation under different conditions. However, virtual prototypes do not eliminate the experimental tests done in product development process. The tests are done for component parameters findings for model simulation and validation. In this context, this work presents a quarter car prototype model with double wishbones and experimental test for its validation. The prototype built uses Kart´s tire and customized spring/damper bicycle assembly. Both tire and suspension spring were experimented for force versus deflection curve measures. These curves were inserted as modeling parameters for computational model. The prototype was subjected to several tests with different suspension parameters set up. The experimental tests were reproduced in the multibody software ADAMS®. The difference between the first numerical model and the experimental counterpart was eliminated through a model updating process.*

Keywords*: simulation, multibody, experimental tests, suspension, validation*

1. INTRODUCTION

The arrival of computation tools incorporating CAD, CAE and CAM technologies allowed the automobile industry to considerably reduce the development time of its products. Engineers and designers, nowadays, can find a number of commercial software that allow the elaboration of computational models with a high degree of sophistication and detail richness for simulation of vehicles or their subsystems under several operation conditions. Commercial software, such as ADAMS, MEDYNA and AUTOSIM, are among the most used in the automotive industry to analyze the dynamic performance o vehicles, using a multibody approach. Other commercial packages, based on finite element methods, such as ANSYS, NASTRAN, ABAQUS and GENESIS, are widely used in the study and solving problems such as the determination of tensions and deformations in structural components subjected to static loads, fluid mechanics applied to aerodynamic studies and structural analysis of components subjected to impact loads. The use of these computational tools allowed a considerable reduction in the number of prototypes built and experimental trials done during the development of a vehicle.

Regardless of the elaboration of the computational model of a system, it cannot assume some peculiar aspects of a real system, such as slack, imperfections in materials and manufacture defects of mechanical components of the system under study. Thus, building a prototype, even in a reduced scale, the use of experimental trials is unavoidable. The prototype allows the complementation and validation of a computational model, since it can forecast with greater reliability the kinematical and dynamical aspects of a vehicle. Therefore, according to the methods now in use for the development of a new product the experimental and computational approaches are employed simultaneously. Recently Santos (1997) showed a suspension prototype for to validate a mathematical formulation of non linear dynamics active suspensions.

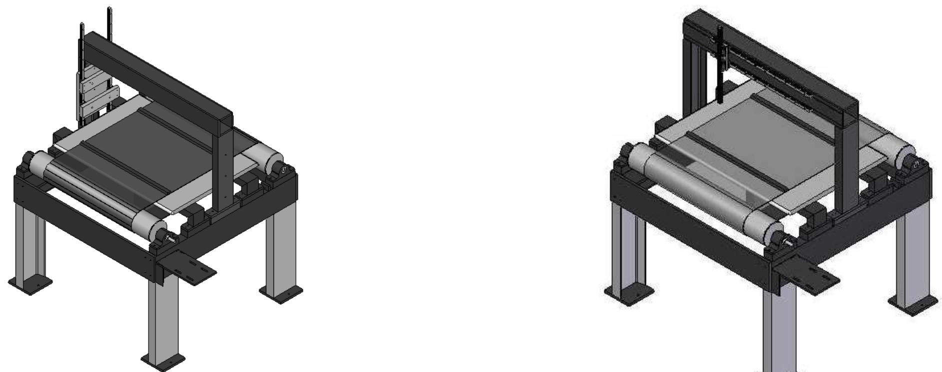
Within this context, this study focuses on the procedures done in the project of reduced scale prototype of a ¼ model of a vehicle equipped with a SLA or Double-wishbones suspension for a laboratory bench. Moreover, it analyses the elaboration of multibody model to represent the performance of this system with the greatest accuracy possible.

The prototype built will provide support for research projects related to the control of automotive suspensions, and can be used as an experimental apparatus for the development and test of active or semi-active control devices. Also, the prototype can be used for experimental trials to analyze the effect of parameters inherent to the suspension system on the kinematical and dynamic performance of the ¼ model vehicle.

2. DESCRIPTION OF THE AUTOMOTIVE BENCH

The laboratory bench, shown in Fig. 1(a), is propelled by a tri-phase induction electric engine with the following characteristics: approximate power 1.5 kW, operation frequency 60 Hz, maximum rotation 356.05 rad/s. The engine spins a 60-mm diameter pulley, which transmits its movement through a belt to a 120-mm diameter. This last pulley is mounted on an axle end of one of the cylinders in the bench making the cylinder spin in the bearings. The movement of this cylinder is transmitted to another one through a flat belt with the following dimensions: 800 mm width and 1.97 m length. The belt behaves as a mobile runway for the prototype, since it remains in rest because it is mounted on linear guides that allow only the vertical movement of the suspension.

Figure 1 (b) depicts the bench configuration that also allows the mounting of a whole vehicle prototype in reduced scale. Besides de vertical movement, as in ¼ vehicle model prototype, at the extreme of the vertical guide there is a spherical joint that is mounted in the whole vehicle model allowing isolated lateral movements or those combined with vertical ones.



(a) (b)

Figure 1. Laboratory bench configured for use: (a) ¼ vehicle prototype, and (b) whole vehicle model in reduced scale.

The linear speed control of the flat belt is done through a frequency inverter connected to the electric engine. This inverter allows the selection and control of the engine rotation. For a given engine rotation, the linear speed of the belt can be easily calculated as a function of the transmission relation between the pulleys, the cylinder radius and the belt thickness. However, due to energy losses to friction in the bearings of the cylinders and between the belt surface and the bench top, the linear speed of the belt at work will be lower than the calculated, thus requiring the use of a photoelectric sensor to measure the rotation speed of the cylinder being moved.

The excitement of the systems being tested on the bench is done with rubber obstacles fixed on the belt tensioned side.

3. QUARTER CAR PROTOTYPE PROJECT

The major suspension prototype requirement in the design of a quarter car is the system's geometry flexibility to allow the study of its dynamic behavior under several configurations. Among several suspension systems available in the market, meeting such a requirement, the one chosen was the SLA or Double Wishbones.

Next step was to establish a layout for the system and define the mechanisms that would allow, whenever possible, the variation of geometric or physic parameters of the suspension system, independently of all other parameters. Once defined a configuration that met this requirement, next step was to establish the dimensions of the mechanical components, such as control arms, vehicle's sprung mass, and choice of mechanical components, such as spherical joints, bearings, spring/shocks groups, wheel and tires. The project of the elements to be manufacture was done keeping as reference the information of a real vehicle suspension system using the same configuration as the prototype. Subsequently, next procedure was to establish a scale factor that would represent the reference vehicle suspension geometry and would allow the prototype to take the greatest volume of the available space for its use in the bench. However, the factor that determined the choice of the reduction scale was the tire. The chosen tire was a model for Karts, since this was the smallest size that maintained a functional similarity to a real vehicle tire. A comparison of the basic dimensions of the Kart tire with the reference vehicle (width of the rolling surface and diameter) yielded two scale factors, 1.5 and 2.5 respectively. The factor 1.5 was used in the determination of the dimensions of the system's geometry elements, while the value 2.5 applied in the determination of the prototype suspended mass. The dimensions of the suspension geometry elements were established dividing the dimensions of the geometric elements of the reference vehicle by 1.5. In the determination of the prototype's sprung mass, initially a steel rectangular block was used; with dimensions conferring the value of ¼ of the reference vehicle suspended mass, approximately 265 Kg. Since the density of a rectangular prism can be expressed as a function of its mass and volume, its mass could be calculated by Eq. (1).

$$M = d \times V = d \times a \times b \times c \tag{1}$$

Where *M* is the mass, *d* the rectangular block material density, *V* the volume, and *a*, *b* and *c* the edges dimensions. The prototype's suspended mass was determined dividing the values of the edges dimensions of the rectangular block by the scale factor 2.5 and applying the resulting values in Eq. (1), together with the density of the chosen material for the manufacture of this component (cast iron => 7150 Kg/m^3). The final result was a solid cast iron block with a mass of approximately 16.0 Kg. This value of sprung mass was distributed in a rectangular block, in the supports for the control arms and in the support for the group spring/shocks. The prototype's suspension geometry elements in their final configuration diverged a little from the adopted scale due to the need of incorporating modifications that would allow adjustments to change the system's geometry. The built and assembled prototype in its final configuration, Double Wishbones, is depicted in Fig 2.

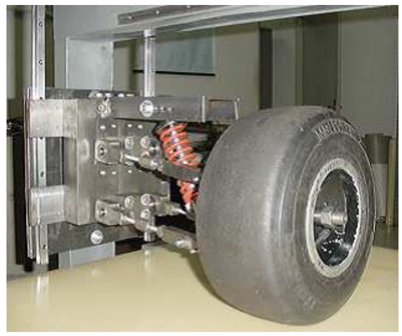


Figure 2. Prototype final configuration.

The prototype was equipped with an integrated group of spring and shock absorber. The shock absorber, originally used in bicycles, was modified to meet the mounting requirements of the prototype suspension. An elongation of the spindle was made and Vaseline was introduced inside the cylinder to provide viscous damping. Also, tests were done with two springs, and the one that performed more adequately for the prototype was the one used in scooters.

4. ADJUSTMENTS ALLOWED FOR THE QUARTER CAR PROTOTYPE SUSPENSION

The adjustments allowed for the prototype are:

Relative Position between the control arms: There is a possibility to mount the supports of the control arms in different positions, allowing the introduction of an angle between the upper and lower control arms. This change in the relative inclination between these arms changes the position of the suspension rolling center in the frontal perspective.

Adjustment of the Control Arms Length: The control arms consist of three parts mounted next to each other through threaded connections. The adjustment system works in a manner similar to a steel cable extender, turning the intermediate part changes the length of the arm. The system allows an increase of up to 20 mm in the length of each arm. Thus, it is possible to obtain camber values of the wheels between 5° positive and negative and adjust the divergence or convergence of the wheels in up to 10.5°.

Adjustments on Assembly Spring/Shock absorber: The two parameters with possible adjustments are the elastic and the damping forces and the spring pre-load. The elastic and damping forces can be modified by changing the assembly angle in relation to the vertical. The possible adjustment angles for the assembly are 10°, 16°, 21° or 26°. According to Adams (1993) assembling angles greater than 30° force shock absorbers to work ineffectively. Also, it is possible to adjust the spring pre-load, through a gadget consisted of two bases, one fixed and the other one mobile, where the two extremes of the spring dressing the shock absorber are supported. The adjustment is done by moving the mobile base through a thread.

- Camber and Convergence/Divergence Adjustments: Adjustments can be made without changing the length of the control arms. These adjustments are done inserting small metal wedges between the parts that form the wheel knuckle. Another alternative for adjusting convergence/divergence is through a lock system inserted to avoid wheel steering.
- SLA Configuration: There is a possibility of assembling control arms with different lengths. This configurations, known as Short Log Arm (SLA) presents the upper control arm shorter than the lower one. For assembling the prototype in this configuration it was necessary to introduce a 108 x 94 x 38-mm cast iron block between the upper balance support and the block representing the vehicle's sprung mass. With the addition of this block to the system the sprung mass of the system changed from 16.0 Kg to 18.6 Kg.

5. EXPERIMENTAL TRIALS TO DETERMINE SPRING AND TIRE RADIAL STIFFNESS

This section describes the trials done to determine the prototype's spring and tire radial stiffness. This information was required since such parameters were part of the prototype's computational model.

The trials were done using MTS (Material Test System) equipment. This equipment has two actuators turned by a hydraulic system, capable of applying controlled displacement or load to the object under study.

Thus, to determine the helical spring stiffness curve used in the suspension, the set spring/shock absorber was assembled as shown in Fig. 3. After the spring was positioned between the actuators a displacement was imposed between them and the resulting force was measure with the load cell of the test equipment.

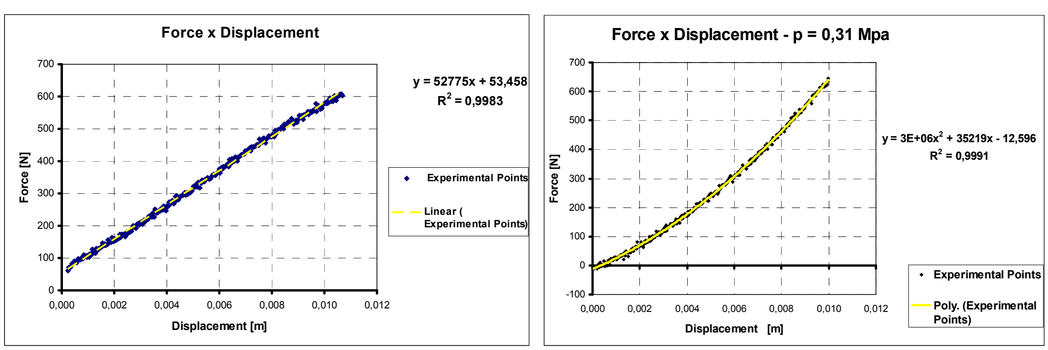
Figure 3. Experimental trial to determine the stiffness parameters of spring and tire, respectively.

As can be observed in Fig. 3, the procedure for the trial with the tire was similar to the one used for the spring. The trial parameters for each component are shown in Tab. 1.

Table 1: Trial parameters for spring and tire

Component	Number of trials	Displacement Applied [m]
Spring	5	0.015
Tire [0.31 Mpa]	3	0.010
Tire [0.25 Mpa]	3	0.010

The results of the trials were obtained as strength and displacement between the terminals. These parameters were used to make the characteristic curves of component stiffness, as shown in figures 4 (a) and 4 (b).



(a) (b)

Figure 4. Curve Force x Deflection: (a) spring and (b) tire (air pressure 0.31 Mpa).

The spring presents a linear performance in the analyzed range, as shown by the curve force x displacement in Fig. 4 (a). Thus, the value 52775 N/m was found for the spring stiffness coefficient, which corresponds to the value of the angular coefficient of curve force x displacement. However, in fact, the spring evaluated was excessively stiff. Therefore, another spring was tested, with a stiffness coefficient of 22730 N/m. As observed in Fig. 4 (b) the tire radial stiffness presented a non-linear behavior in the displacement range evaluated. This same behavior was observed in the trial done with 0.25 Mpa inflation pressure.

6. COMPUTACIONAL MODEL

In the search to represent as faithfully as possible the designed automotive suspension prototype, a computational model was elaborated using CAD and CAE tools. CAD's software used was Solid Edge V12, while the computational model was elaborated in the multibody analysis and simulation software ADAMS.

The geometry of a model has as major functions to represent graphically the object modeled and to serve as a reference for the addition of kinematical and force restrictions (Leal, 2002). The tridimensional drawing of each rigid body was generated in CAD environment. The assembly drawing was obtained integrating all individual components to generate the chosen project configuration for the suspension prototype. Also, a tridimensional drawing of the bench was generated, representing the computational model part known as ground (fixed part).

The computation of mass and inertia properties of each component was automatically done by CAD software, and the only parameter supplied was the material density. These properties were specified to each part of the model in the CAE environment.

The connections between the rigid bodies of the system were established in the modeling sequence. The connection among the components was done by kinematical restrictions such as translation and revolution joints that determine the relative movements allowed among the parts.

Since the spring used presents a linear performance, the introduction of its stiffness in the computational model is done by providing the proportionality constant value and the positioning points of its ends.

Because the MTS equipment does not allow adequate trial conditions for the determination of the characteristic curve for the shock absorber, i.e., as a function of speed, the determination of the working parameters of this component was done through experimental data obtained in trials with the prototype. The value of the damping coefficient was estimated using the methodology of logarithmic decreasing (Den Hartog, 1956), used in mechanical systems with viscous damping. Subsequently, this value was supplied into the model.

The tire was modeled using the methodology proposed by Blundell (2004), according to which the tire acts as a spring with free length equals to its non-deformed radius, positioned between its center and contact point with the ground. Thus, the tire is modeled working only with radial stiffness, which was measured experimentally. The characterization of this parameter was done supplying the curve obtained experimentally and shown in Fig. 4 (b).

The experimental trials were done with the prototype being excited by passing over obstacles at a given speed. The simulations of passing the model over the obstacle maintaining a stationary position with the tire supported on the base of the model. A vertical translation movement was imposed to the base in such a way that at every moment *t* of the simulation the surface will be at a height specified according to the obstacle profile and the passing speed. Figure 5 depicts the complete computational model.

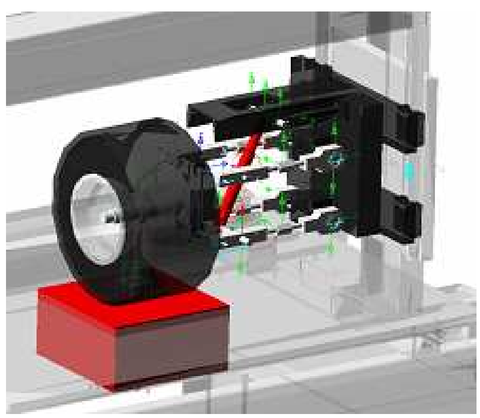


Figure 5. Complete computational model.

7. EXPERIMENTAL TRIALS

The validation of the computational model was done with experimental trials that consisted of subjecting the prototype to different operational conditions, varying the physical parameters of tire pressure, damping force and assembling angle of the shock absorber in relation to the vertical. The exciting conditions also varied through changing the speed passing over the obstacle as well as the obstacle profile. The profiles of the obstacles are shown in Fig. 6.

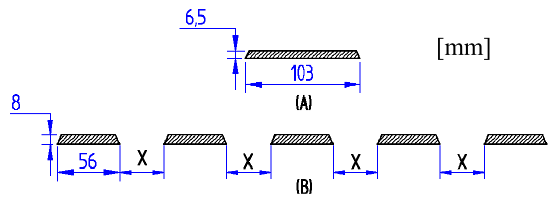


Figure 6. Used obstacle profiles: (a) single obstacle and (b) a series of 5 obstacles.

Table 2 presents the conditions used for testing the prototype.

Table 2: Experimental trials

Trial Number	**Passage speed [m/s]**	**Shock absorber assembling angle [°]**	**Tire Pressure [Mpa]**	**Obstacle Profile**
1	1.95	26	0.31	A
2	2.86			
3	3.79			
4	1.97			B (x = 0.10 m)
5	1.92			B (x = 0.15 m)
6	1.99			B (x = 0.20 m)
7	2.03	16		A
8	2.06	10		
9	1.92	26	0.25	A
10	2.90			
11	3.87			
14	2.01			B (x = 0.10 m)
15	2.01			B (x = 0.15 m)
16	2.01			B (x = 0.20 m)
12	2.03	16		A
13	1.97	10		

The prototype and the bench were equipped with the objective of collecting the following data: vibration signals of the sprung and unsprung masses and prototype's speed. The equipment used consisted of 2 piezoelectric accelerators, 2 signal conditioners, data acquisition board and a microcomputer. The calculation of the linear speed at the tire contact point with the rolling surface was done indirectly, by measuring the rotation of the cylinder with a photoelectric sensor.

The whole experimental apparatus and equipment used are shown in Fig. 7, where:

1 – Accelerator to measure sprung mass responses
2 – Accelerator to measure the unsprung mass responses
3 – Signal conditioner for accelerator 1
4 – Signal conditioner for accelerator 2
5 – Connector panel
6 – Microcomputer
7 – Sensor to measure rotation speed of the movement cylinder
8 – Power source for the rotation sensor
9 – Obstacle for exciting the vertical movements

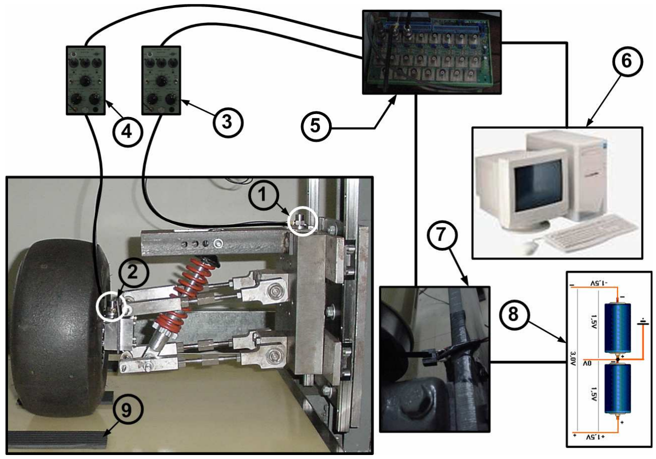


Fig. 7: Experimental apparatus and instrumentation used.

8. RESULTS

After the experimental trials were done, a small difference between the experimental results and those from the simulation was found, indicating the need to adjust the computational model. This can be observed in Figs. 8 (a) and 8 (b), where the results from the first trial are shown.

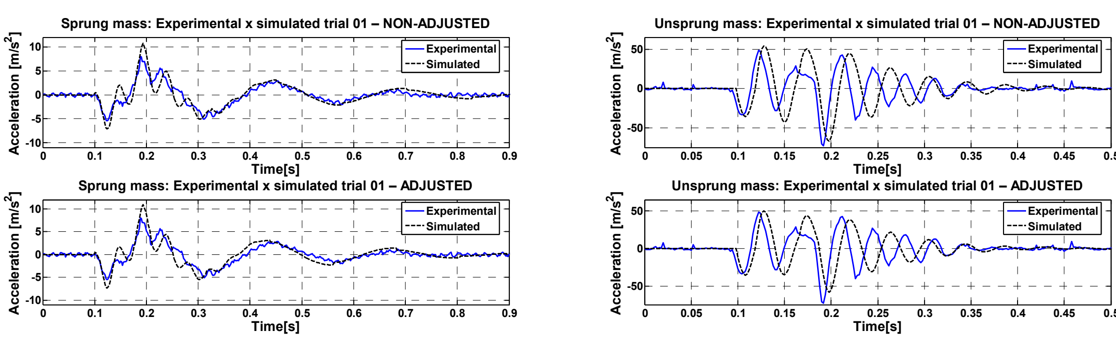
The adjustment was done by an optimization process to minimize the relative error existing between the experimental and the numerical simulation curves. The following objective function was used for the optimization.

$$FO = \sum((mse - mss)^2) + \sum((mnse - mnss)^2) \tag{2}$$

where:

- *mse* – vibration signal of the sprung mass obtained experimentally;
- *mss* – vibration signal of the sprung mass obtained numerically;
- *mnse* – vibration signal of the unsprung mass obtained experimentally
- *mnss* – vibration signal of the unsprung mass obtained numerically.

A first adjusting strategy was selecting the design variables, shock absorber coefficient, spring stiffness and the coefficients of the tire force deflection curve. No restriction function was used, with only the design variables being restricted to a range of 10% around their initial value. The optimizer used was DOT1, containing the optimization algorithm Broydon-Fletcher-Goldfarb-Shanno (BFGS). The values of the design variables were scaled to vary in the range -1 to +1 due to the different magnitude orders of the parameters involved. The optimization was done in 4 iterations, allowing a reduction of 31.4% in the value of the objective function. Figures 8 (a) and 8 (b) also show the results of the experimental trial 01 compared to their respective simulation done with adjusted computational model.

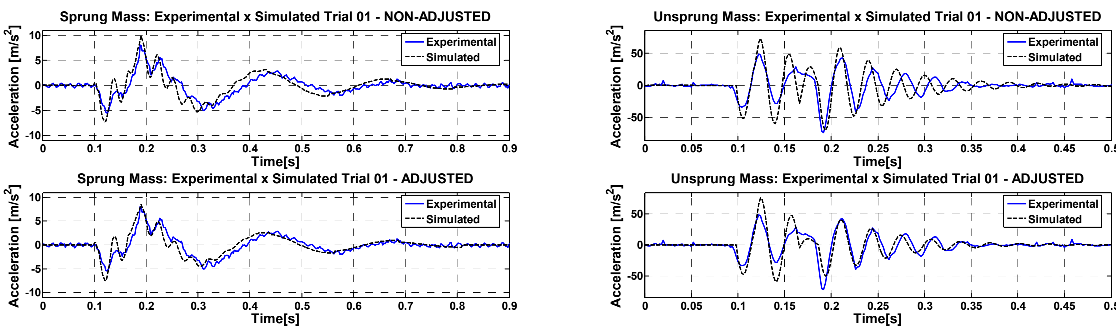


(a) (b)

Figure 8. Experimental x Simulation (adjusted x non adjusted model) – (a) sprung mass and (b) unsprung mass.

The analysis of figures 8 (a) and 8 (b) shows that the adjusted model adequately represented only the dynamic performance of the sprung mass. The same behavior was observed in the other trial conditions tested and reproduced numerically. Therefore, a new adjusting strategy was used. This second adjusting strategy was implemented using fewer design variables, associated to a change in the tire modeling procedure. In this case a hypothesis of tire constant stiffness was considered for the operational condition, although the experimental results indicated that this was not a linear relation. The linearity hypothesis was found reasonable considering that the operating tire (rolling) differs from the experimental test condition, where the tires had to accommodate the rolling surface in the supports between the terminals of the test equipment. Thus, the initial part of the force x deflection curves was discarded, more precisely the force values measured until 2 mm of deflection. From there on, the linear adjustments were done the other experimental points in all experimental curves obtained, supplying a radial stiffness of 138468 N/m for the tire with a pressure of 0.31 Mpa and 125800 N/m for a tire pressure of 0.25 Mpa. Thus, the adjustment was made with the design variables spring stiffness, damping coefficient and tire radial stiffness. These variables were also restricted to the ±10% range of their initial values. Once again the optimizer DOT1 was used and the design variables scaled as done in the first strategy. The optimization was done in 6 iterations, giving a 65.5% reduction in the objective function value.

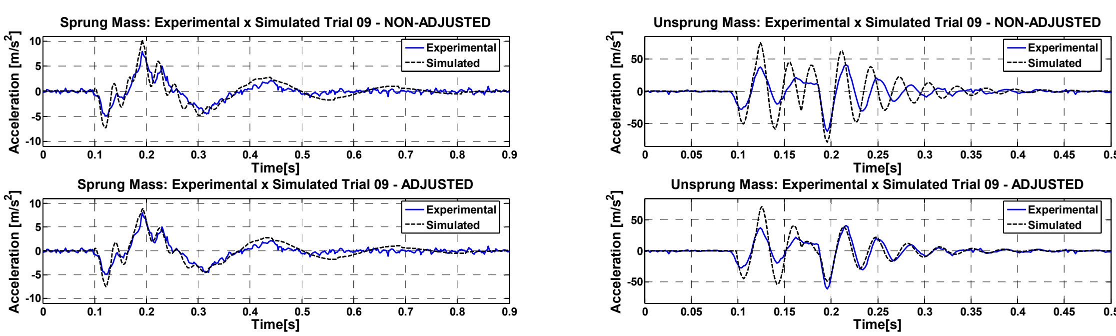
Figures 9 (a) and 9 (b) show the experimental results obtained for trial 01, compared with the respective simulations done with the adjusted computational model. The computational model adequately represented the dynamic performance of both sprung mass and unsprung mass. Although the results are not shown here, similar performance was observed for all other trials done with tire pressure at 0.31 Mpa.



(a) (b)

Figure 9. Experimental x Simulation (adjusted x non adjusted model) – (a) sprung mass and (b) unsprung mass.

A virtual reproduction of trials 9 to 16, in which the prototype was tested with a tire inflation of 0.25 Mpa required another model adjustment. The same optimization strategy used previously was used in this situation. However, only the tire vertical stiffness was used as design variable since the optimum damping values for the shock absorber and spring stiffness had already been found. As previously, DOT1 optimizer was used, and the optimization was done in 3 iterations and gave an objective function value reduction of 52.0%. Figures 10 (a) and 10 (b) illustrate the comparison of experimental and numerical results for trial 9, obtained with tire pressure of 0.25 Mpa.



(a) (b)

Figure 10. Experimental x Simulation (adjusted x non adjusted model) – (a) sprung mass and (b) unsprung mass.

The analysis of figures 10 (a) and 10 (b) shows that the computational model adequately represented the dynamic performance of both sprung and unsprung masses. Similar performance was observed for all other trials done with tire pressure 0.25 Mpa.

9. CONCLUSIONS

Some of the conclusions, not only in relation to the results obtained, but also from the procedures of the prototype design and its assembly on the bench, can be drawn.

In relation to the adaptations done on the bench to hold the prototype, the first major conclusion is that the adaptations successfully allowed the accomplishment of the experimental trials. However, such alterations were not sufficient for make the experimental trial a practical task. Many difficulties were found during the experimental trials.

The first difficulty found was the high friction due to the pre-load existing in the assembly of the linear guides responsible for transmitting the oscillating movement of the sprung mass in the vertical direction. The first time the prototype was assembled in the bench, the friction on the guides and the existing pre-load did not allow the prototype's sprung mass to do any oscillating vertical movement. This problem was solved removing a set of recirculating spheres in the guide assembly. This procedure considerably reduced friction and pre-load, allowing the vertical oscillating movement of the sprung mass. This change in the guide assembly suggests that, for the sprung mass used in this study was not the most adequate.

Another problem faced during the experimental trials was the great difficulty in maintaining the bench pulley aligned. A time consuming procedure for positioning and aligning the pulley during the intervals from one trial to the other was required. This difficulty was an indicative that the type of pulley used in the bench was not the most adequate. The use of this pulley is not indicated for the study of tire lateral properties when the prototype receives the steering system.

In relation to the 1.5 kW motor used to move the bench, its limitation was found on the use for speeds below 3.87 m/s, even with the use of a transmission of 2:1. At this maximum speed suggested the motor presented considerable heating after some working time. The overload in the motor could be identified from this excessive heating and its inability to maintain a constant rotation speed. In some cases the motor was not able to maintain the bench cylinders rolling for the desired time.

This performance limitation of the motor could be attributed to the high load represented by the prototype's sprung mass, the need for reasonable amount of available energy to overcome the bench cylinders inertia, energy availability to bend the belt on the cylinders, camber and the friction between the lower surface of the pulley and the metal support on the bench structure.

An analysis of the prototype design leads to two conclusions. The first one relates to the sprung mass, in which the value used in the project was high for the bench structure and available equipment. If the design were done with less sprung mass, certainly the problems with the motor would be minimized or eliminated. From this hypothesis, it also is supposed that the dynamic performance of the prototype could have been evaluated at higher speeds than those tested. The second conclusion is related to the working performance of the shock absorber, which did not present a viscous behavior in the prototype suspension.

In relation to the manufacturing steps, simulation and model adjustment, the computation tools applied effectively fulfilled their roles. CAD software used to represent the three-dimensional models of the prototype's components and practically and effectively obtained the mass and inertial properties. The simulation software allowed to practically obtaining a prototype multibody model with the required detail level.

Two distinct conclusions can be drawn from the experimental trials for the characterization of the tire and spring. The trial methodology to obtain the spring's stiffness coefficient was adequate, as observed in other studies (Leal, 2007). The methodology used for the tire, was successfully used in other studies (Leal, 2007), but was not as effective in the present work. This conclusion is based on the fact that the prototype mathematical model that obtained the greatest representation in different operation conditions was the one that considered the relation force as function of tire deflection linear, although the experimental measurements indicated a non-linear relation. This lack of linearity can be attributed to the fact that the tire's rolling surface, at the first trial stage, passes through an accommodation period in trial equipment supports. Studies where this methodology was successfully used also report this effect; however, due to tested tire dimensions, it appears to be negligible. In the case of the prototype's tire, due to its reduced dimensions and the fact that it was a competition tire (high adherence), the effect of lateral grip to the trial bench support during the trials' initial measuring stages affected the results. An analysis of model fitting procedures indicated that for the rolling tire (a condition different from that in which the stiffness was measured) the linear relation is the one that best reflects the tire behavior.

The experimental trials to evaluate the vertical dynamic performance of the prototype were adequate for the model fitting procedure.

In general, the adjusted model was robust, forecasting the performance of the prototype satisfactorily, in practically all situations tested. As far as subjecting the prototype to speed variations passing over a single obstacle, it was observed the lowest and the highest speeds used in the adjusted model had greatest representation. At the intermediate speeds, the adjusted model was not representative. This unexpected fact indicated the need to study what happens in that speed range, and to repeat the trials under those conditions to detect if there were any procedure flaws that affected the quality of the experimental results.

In trials where regularly spaced obstacles were used, the adjusted model better represented the situations when this distance was smaller.

The methodology used for model fitting was valid. The adjusted models successfully represented the real prototype in most of the analyzed situations. The difficulties found to obtain a more precise and robust model could be associated to the difficulty in characterizing the prototype shock absorber, as well as the friction and the pre-load existing in the bench linear guides.

The following suggestions for advancing this study are given:

- Substitution of the linear guide assembly for another one with lower friction and pre-load, in such a way that lower resistance to the vertical movement of the prototype's sprung mass would be imposed.
- Substitution of the pulley for a more adequate one in relation to its alignment during bench operation, giving lower friction coefficient on the metal base and consuming less energy on cambering the action cylinders.
- Re-evaluation of the prototype's similarity relation to reduce the sprung mass in such a way that will not overload the bench driving motor.
- Manufacturing a specific spring/shock absorber assembly for the prototype. The shock absorber should be viscous. The assembly should be designed to allow the implementation of active and semi-active control gadgets.
- New experimental trials should be done to analyze the effect of the prototype's rolling speed over the obstacles in relation to the adjustment quality of the model obtained.
- Assembling a steering system that would allow turning the tire to analyze the tire lateral properties.
- Trial to explore other resources in the prototype, such as camber adjustment, variation in the suspension arms length, and convergence angle of the wheel-tire assembly.

10. ACKNOWLEDGEMENTS

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11. REFERENCES

Adams, H. **Chassis Engineering**. N.Y: HP Books, 1993. 133 p. ISBN 1-55788-055-7.

Blundell, M. V.; Harty D. **The Multibody Systems Approach to Vehicle Dynamics.** Warrendale: Society Automotive Engineers, Inc, 2004. 516p. ISBN 0 – 7680 – 1496 –9.

Den Hartog, J. P. **Mechanical Vibrations**. 4th ed. MacGraw – Hill, 1956. 436p.

Leal, M. F. **Desenvolvimento de uma Metodologia Moderna de Projeto de Veículos Visando Fabricação em Pequena Escala.** 2007. 169 f. Tese de Doutorado – Universidade Federal de Uberlândia.

Leal, M. F., 2002, "Otimização do Comportamento Dinâmico de Veículos Utilizando Superfícies de Resposta", Tese de Mestrado.

Milliken, W.F. and Milliken, D.L., "Race Car Vehicle Dynamics" , SAE International ,1995.

Santos, I. F. **Dinâmica não Linear de Suspensões Ativas**. In: XIV COBEM - Cong. Bras. de Engenharia Mecanica, 1997. Bauru, Sao Paulo, 1997. p. 1-8.

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