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EXPERIMENTAL STUDY OF A REDUCED MODEL OF ¼ VEHICLE SUSPENSION

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Abstract. *The study of vibratory systems in the current context of Automotive Engineering brought with it a series of technological innovations. The development of highly safe, comfortable and dynamically efficient vehicles is only one of the benefits of vibration theory in the progress of the automobile industry, especially in the design of vehicle suspensions. These are responsible for absorbing irregularities from the ground providing comfort as well as maintaining the stability of the vehicle. The aim of this paper is to build a test bench of a reduced ¼ vehicle suspension model, in which simulations of a suspension system characteristic of popular cars can be performed. The system consists of four parts: the sprung mass (relative to ¼ of the weight of the car), the unsprung mass (corresponding to the tire wheel assembly), the springs and the damper belonging to the suspension assembly. The analysis were performed using the MATLAB R2015a software, considering a scale of 1:10 between the test bench model and the suspension of a passenger car. The vertical responses of the sprung mass to the excitations of an harmonic terrain profile are evaluated.*

Keywords: *vehicle dynamics, experimental analysis, ¼ vehicle, vibrations, passive suspension.*

1. INTRODUCTION

The origin of the suspension systems dates back to the time when the main mode of transport was still the one of animal traction. The carriages that carried loads and people were stiff and transmitted all irregularities from the floor to the passengers. Then what was equivalent to the spring beam system was introduced. Only recently the suspension system began to be equipped with spring and damper. At the beginning of the twentieth century, low system-generated damping was common, but this mattered for low stability. What has become necessary with the evolution of the fastest and most powerful vehicles (Merling, 2007).

The dynamic behavior of a car is directly related to its suspension system. The spring - damper assembly, in addition to the other components of the suspension, have several functions when you are driving your vehicle. One of the main functions is the isolation of the passenger compartment against vibrations due to imperfections in the road. Another typical function of the system is to promote better grip. (FREITAS Jr, 2006)

Comfort and stability are the desired goals when designing a suspension. It is important that it is at the same time soft to provide comfort to the occupants and stiff enough to ensure a good grip on the ground. As they are conflicting parameters, it is a challenge to get an ideal fit. (Merling, 2007).

In the study carried out by Macorin (2006) it was proposed the building and study of a suspension bench for off-road vehicles. The model was proposed virtually for a first analysis and after that the tests happened to be realized in the experimental bench. The author concluded as valid the results obtained, stating that it is possible to study different suspension geometries through the experimental bench.

When comparing the results of simplified or classic ¼ vehicle models such as the models obtained by virtual prototype analysis, Freitas Jr (2006) concluded that it is feasible to use the simplified or classical model for analysis in the frequency domain. The data analyzed were obtained from the dynamic responses of the sprung and unsprung mass of a classic model of ¼ of vehicles and of the virtual prototype built with using ADAMS software.

Rezende and Borges (2003/2006) have two works in the area of suspension bench. In the first one it is exposed the entire process for the development and computational construction of a bench of a ¼ vehicle suspension model. In the most recent work, the authors focused on the approach and validation of the ¼ vehicle prototype computational model. In the work the procedures for the execution of such validation were explained. When comparing the results it was observed a difference between the computational and experimental results, being necessary the adjustment in the computational model for a correct representation of the real system, representation that was considered satisfactory by the authors.

Zago et al. (2010) presented in their work a study of an optimal control for the suspension of a quarter vehicle. In his work the ideal parameters for the construction of the bench for the experimental evaluation were shown, being based on the analysis carried out by the computational simulations. In their conclusions the authors stated that the bench can be used in the experimental validation of the one-quarter suspension model, helping to understand the literature about the content.

In this work the proposal is to analyze the operation of a suspension system that the present paper was elaborated. The objective is the construction of a test bench containing the reduced ¼ vehicle suspension model, where the responses to different types of excitations were analyzed.

2. MATHEMATICAL FORMULATION

The vehicle suspension can be modelled as a two degree of freedom system, shown in Fig. 1. It consists of four parts: the sprung mass (relative to ¼ of the weight of the car), the unsprung mass (corresponding to the tire wheel assembly), the springs and the damper belonging to the suspension assembly. The aim is to represent the system and to establish numerical analysis regarding to the vertical response of the sprung mass whem the system is submitted to excitations of a terrain profile. In this model we disregard the rolling and loading effects caused by the interaction of the other wheels with the track (GILLESPIE,1992) (SOUSA,2016).

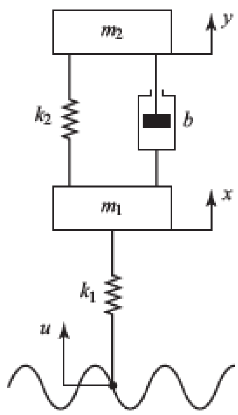

Figure 1. ¼ vehicle suspension model Source: Adapted Ogata (1998).

The equations of motion of the system are derived from the application of Newton's second law, and are given by

$$m_1\ddot{x} + b\dot{x} + (k_1 + k_2)x = b\dot{y} + k_2 y + k_1 u \quad (1)$$

$$m_2\ddot{y} + b\dot{y} + k_2 y = b\dot{x} + k_2 x \quad (2)$$

Where m_2 corresponds to the sprung mass, k_2 to the spring stiffness of the suspension, b to the damping coefficient, m_1 to the unsprung mass, k_1 to the equivalent stiffness of the tire, y to the vertical displacement of the sprung mass, x to the displacement of the unsprung mass and u to the displacement from the ground.

Modeling these equations in state space, we obtain:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ \frac{-(k1+k2)}{m1} & \frac{-b}{m1} & \frac{k2}{m1} & \frac{b}{m1} \\ 0 & 0 & 0 & 1 \\ \frac{k2}{m2} & \frac{b}{m2} & \frac{-k2}{m2} & \frac{-b}{m2} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{k1}{m1} \\ 0 \\ 0 \end{bmatrix} * u \quad (3)$$

$$\begin{bmatrix} y_1 \\ y_2 \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} \quad (4)$$

The simulation of the dynamic behavior of the test bench was carried out in MATLAB software from the numerical modeling of its components, according to the quoted governing equations extracted from the theory of mechanical vibrations. The use of MATLAB is justified due to the interest in obtaining a theoretical curve of the behavior of the bench.

3. 1/4 SUSPENSION BENCH DESIGN AND CONSTRUCTION

A CAD model of the test bench, shown in Fig. 2, was carried out through the software Catia V5R19 with the purpose of illustrating the design of the system as a whole, besides allowing the characterization of its measurements and making possible its construction for later experimental tests. The design in a 3D software is important because it allows the characterization of the dimensions of the bench before the final model building. It is possible to change and adapt the dimensions in a simple and practical way in the virtual environment. Achieved a model considered satisfactory begins the physical construction of the prototype.

Based on the ¼ vehicle model, and based on the desired dimensions the design was performed using the software Catia V5R19. After completing the CAD model (Fig. 2) and defining all measurements, the test bench construction was started. The components of the structure (springs and dampers) used correspond to those used in RC vehicles, which allow the modification of variables such as spring stiffness (elastic parameter) and fluid damping coefficient. The supports are made of expanded PVC 20 mm thick and the rods are made of 1045 steel with a diameter of 12 mm each.

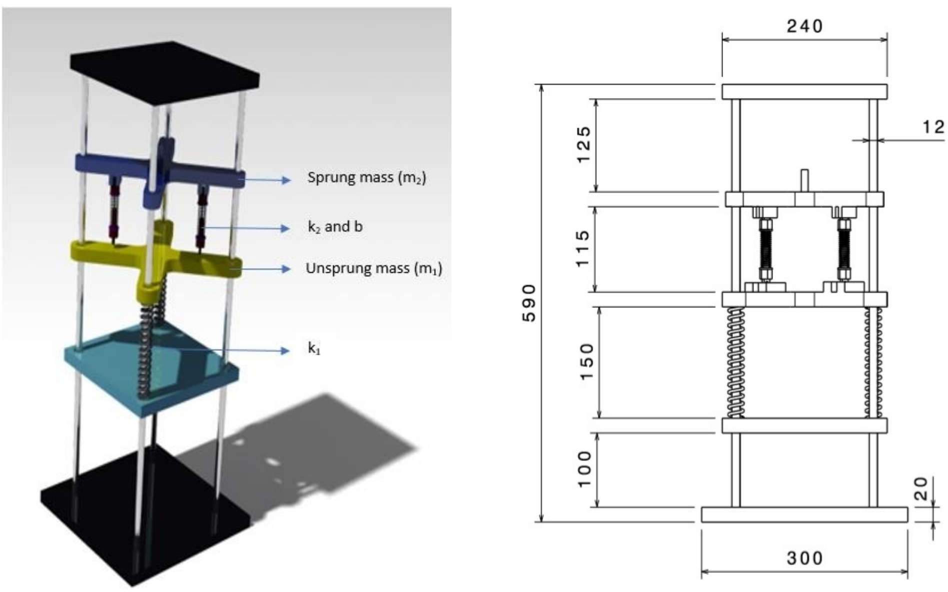

Figure 2. CAD model of the 1/4 test bench.

4. EXPERIMENTAL METHODOLOGY

The experimental tests, performed after the construction of the bench, were made under two conditions. First, a forced vibration condition was imposed, where the bench was excited harmonically by the rotation of a camshaft coupled to a three-phase motor with 1.5hp CFW-11W. The excitation conditions were modified by varying the motor rotation frequency. In the second set of tests, a free vibration condition was imposed, in which the bench was suspended until a certain height (initial condition) and then released for free vibration.

The engine promoted vibration of the test bench by rotating an axle containing a cam, moving the sprung mass and unsprung mass vertically above it.

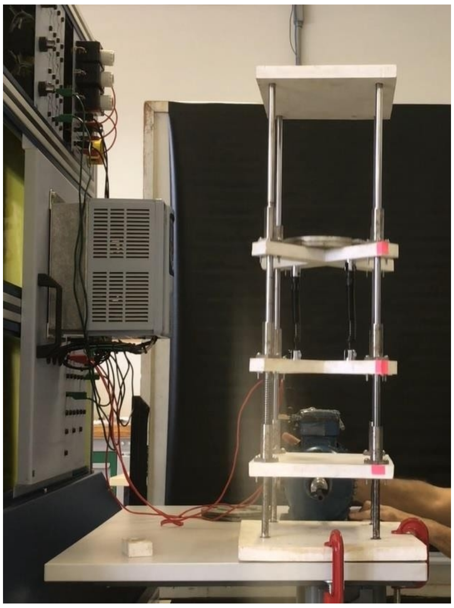

Figure 3. Test bench constructed and excited harmonically by the rotation of a camshaft coupled to a three-phase motor.

In each test the system was excited by the engine set at different frequencies. The test bench in motion was recorded in video at 60 frames per second with an iPhone 7, acquiring the displacements of the system from the program Tracker, which allows plot the temporal graphs of the vertical displacements of the bench from the analysis of the videos, recording the position of the masses sprung and unsprung (ie points of interest chosen).

With the test bench built (Fig. 3), the tests were performed for data acquisition. The tests were done by varying the motor rotation and the load of the sprung mass of the bench. The values used of load were 1 kg, 2 kg, and 3 kg; and for each mass value the tests were performed at rotations of 90, 100, 110 and 120 rpm.

5. RESULTS AND DISCUSSION

In order to carry out the theoretical simulation of the behavior of the test bench the values of the sprung mass, unsprung mass, spring and tire stiffness and the damping coefficient of the assembly were defined. The comfort parameters usually employed in the design of automobiles were also considered in the definition of such values, so that the natural vibration frequency of the system in the simulations was observed to remain close to 1 Hz, considered as a great design value for urban vehicles (GILLESPIE, 1992).

The numerical modeling of the vibratory behavior of the experimental bench was performed using as parameters the values shown in Table 1, among which only the stiffness value of the suspension k_2 varied to 400 N/m. These values were defined after measurements in the constructed real bench, in order to simulate its theoretical behavior according to the system movement equations before the free vibration experiments.

Table 1. Properties for the ¼ vehicle suspension model.

Property	Symbol	Value
Sprung mass (kg)	**m2**	1.6
Unsprung mass (kg)	**m1**	0.6
Spring stiffness (N/m)	**k2**	200
Tire Stiffness (N/m)	**k1**	10000
Damping constant (Ns/m)	**b**	8

From the values initially defined, the numerical simulations were performed in the software MATLAB R2015a, in which the free vibration response of the sprung mass was verified according to the properties described in Table 1 In this way, in order to obtain the time history response, a step excitation of 0.55 m to 0.37 m was imposed on the system, so that it represents the initial height (initial condition) and equilibrium position of the sprung mass in the actual experimental tests carried out. The dynamic response obtained in the time and frequency domain is shown in Fig. 4 and 5.

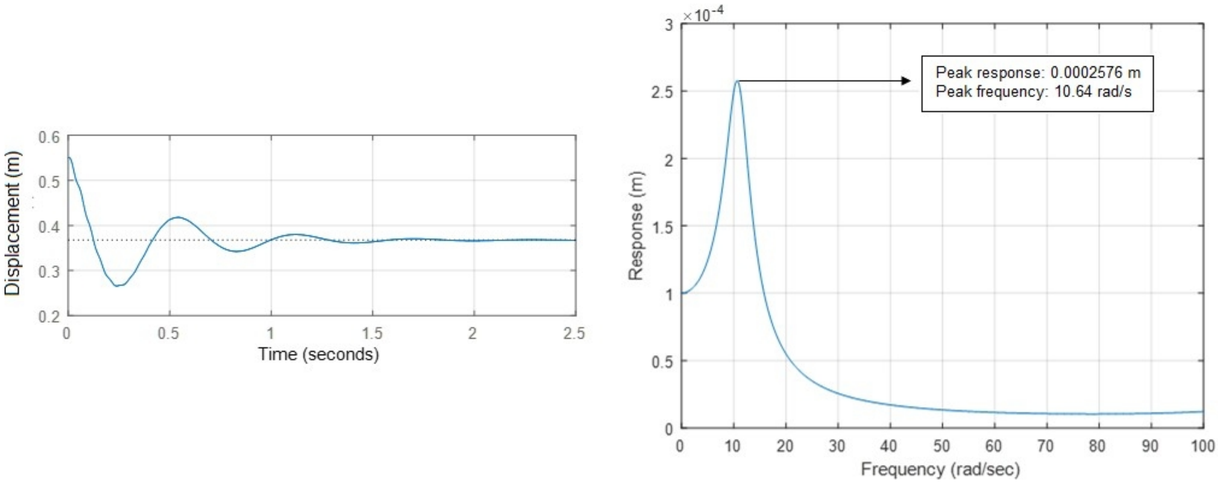

Figure 4. Response in time (on the left) and frequency (on the right) of the displacement of the sprung mass for the properties of Table 1.

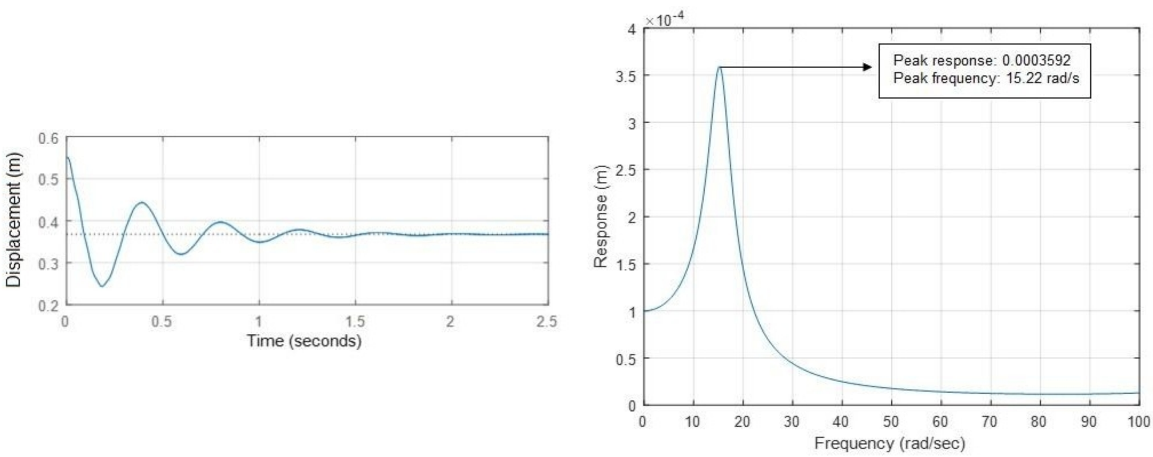

Figure 5. Response in time (on the left) and frequency (on the right) of the displacement of the sprung mass for the properties of Table 1 and $k_2 = 400$ N/m.

Figures 4 and 5 show that the free vibration behavior of the sprung mass for both stiffness values of k_2 can be classified as underdamped, since oscillation occurs around its equilibrium position (RAO, 2008). For k_2 being 200 N / m, the fundamental frequency obtained was 1.76 Hz, while for a k_2 value of 400 N / m the calculated fundamental frequency was equal to 2.47 Hz.

With the theoretical data obtained numerically, the experimental tests were conducted with the experimental bench. It was subjected to harmonic excitation by the action of a three-phase motor, whose oscillation frequency parameters could be controlled from a variable frequency drive added to it. Fig. 6 and 7 show two of the results obtained for the displacement time history of the sprung mass recorded with the use of video images at 60 frames per second processed in the Tracker program, which indicates the position of a point of interest in each frame and plots your data in the time domain. As in the initial numerical modeling, Tables 3 and 4 indicate the parameter values adopted prior to the execution of the tests.

Table 3. Properties defined for the ¼ vehicle test bench during one of the tests.

TEST 01			
Engine parameters		**Test bench parameters**	
Excitation frequency	2,9 Hz	**Sprung mass**	1,6 kg
		Unsprung mass	0,6 kg
Revolutions per minut	90 RPM	**Spring stiffness**	200 N/m
		Damping constant	8 N.s/m

Table 4. Properties defined for the ¼ vehicle test bench during one of the tests.

TEST 02			
Engine parameters		**Test bench parameters**	
Excitation frequency	3,9 Hz	**Sprung mass**	1,6 kg
		Unsprung mass	0,6 kg
Revolutions per minute	120 RPM	**Spring stiffness**	200 N/m
		Damping constant	8 N.s/m

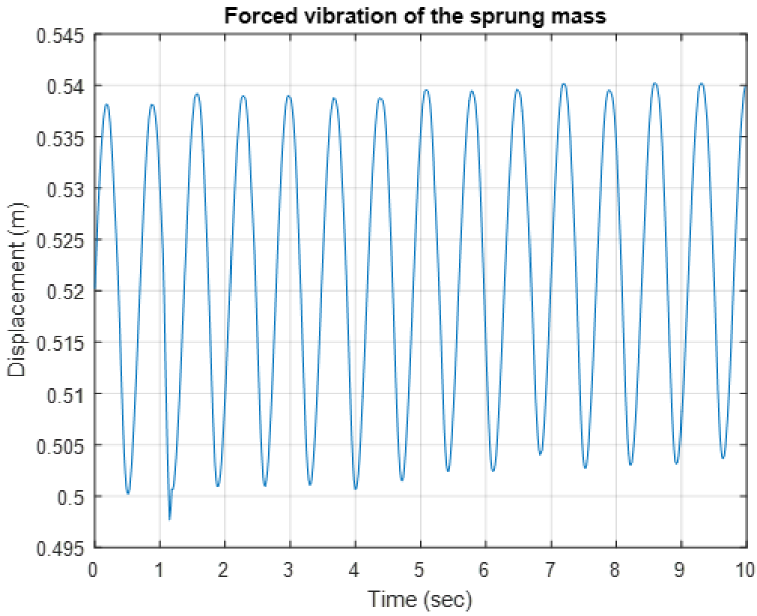

Figure 6. Time history displacement of the sprung mass (parameters Table 3).

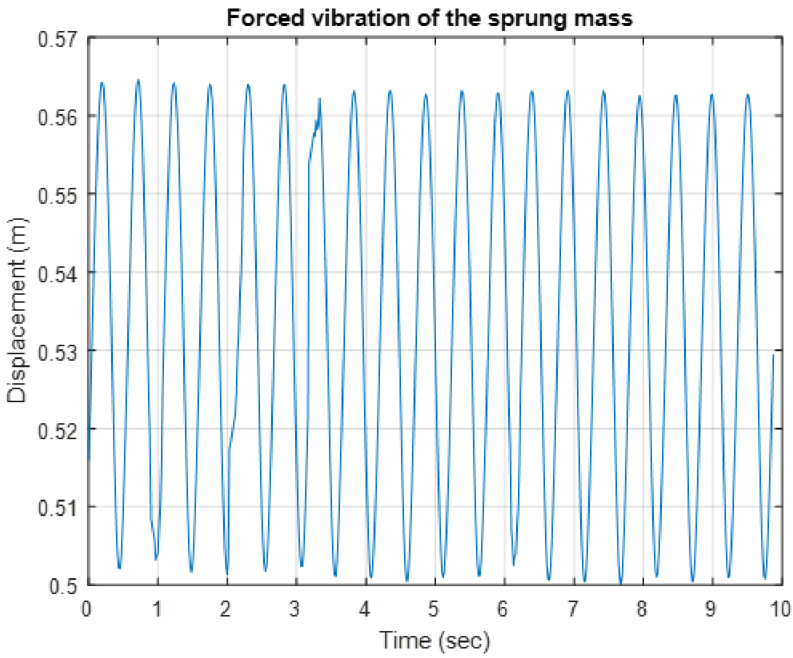

Figure 7. Time history displacement of the sprung mass (parameters Table 4).

The free vibration tests conducted on the bench were performed according to the following procedures: an initial displacement condition was applied to the system. Since it moves only vertically, the imposed displacement consisted of an initial height of 0,55 m above the ground, from which the system was released. Two springs were tested in the suspension with different stiffness values (k_2), to compare results. As shown in Fig. 2 and 3, the springs corresponding to the suspension and the tires are positioned in parallel, so that four springs were used in the entire bench, two at each level (sprung and unsprung). The first test was performed with two 100 N / m springs, and the second test with two 200 N / m springs. These values, in turn, are summed to obtain the equivalent stiffness k_2 (RAO, 2008), since the first test was performed with a 200 N / m and the second with a 400 N / m. In both cases the stiffness k_t remained constant at 10000 N / m, from two springs of 5000 N / m in parallel. The results of these tests are shown in Fig. 8, where a time history position data sprung mass is shown.

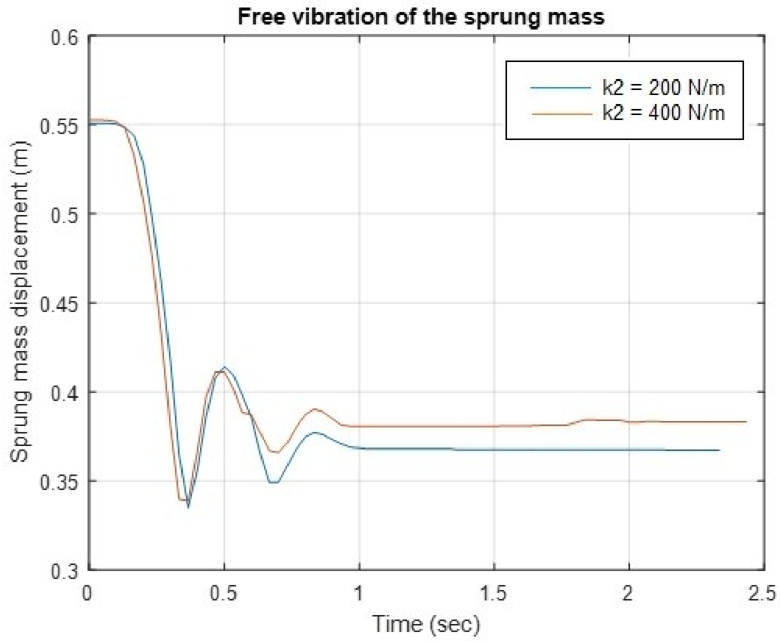

Figure 8. Free vibration curves obtained for the two considered stiffness $k_2$values.

Figure 8 shows that the experimental results obtained for the free vibration test can also be classified as underdamped, such as the numerical responses shown lately. For a variation of 200 N / m in k_2, no significant differences were observed in terms of natural frequency, which remained close to 1.7 Hz in the two tests. This effect is due to the influence of the friction on the workbench, which occurs between linear bearings and steel rods 1045 (Fig. 2 and 3) and ends up reducing the effectiveness of the springs in promoting significant changes in their behavior. The result closest to that obtained with numerical simulations (Fig. 4 and 5) occurred with k_2 equal to 200 N / m, producing similar response curves in the time domain in relation to the corresponding theoretical graph (Fig. 4 on the left).

The result obtained with the free vibration tests also showed that the natural frequency of the system differs from the ideal design frequency of 1 Hz by about 0.7 Hz, for both equivalent 200 and 400 N / m. This is due to the imperfections caused by the manufacture of the experimental bench, such as misalignment of steel rods, machining of PVC plates and fixing holes, as well as possible friction during linear bearings. It is therefore necessary to optimize the bench to meet the optimum design specifications for vehicle suspensions. However, the bench can still be used as an important educational tool in the characterization of vibration systems of two degrees of freedom, especially with regard to the underdamped vibration behavior.

6. CONCLUSION

The present work presented the design and construction a test bench based on the reduced model of ¼ vehicle. In view of this proposal, numerical simulations and experimental tests were performed comparing the results obtained with the optimum suspension design patterns prescribed by the literature in vehicle dynamics. The data extracted allowed to evaluate the parameters of mass, stiffness and damping of the suspension assembly, unsprung mass and tire stiffness before subjecting it to different harmonic excitation profiles.

The test results showed a similarity of the actual behavior of the bench with the theoretical one with an equivalent stiffness value of 200 N / m, resulting in an underdamped vibration natural frequency of the sprung mass of 1.7 Hz. This value differs from the ideal frequency of 1 Hz suggested by Gillespie (1992), and therefore shows that bench settings need to be optimized it in this regard. For the equivalent value of 400 N / m, no significant differences were observed in terms of natural frequency and time response in relation to the first value of k_2 equal to 200 N / m., Differences were perceived when comparing to the corresponding theoretical result. This fact evidences the existence of imperfections in the bench regarding to the theoretical model as described by Fig. 1, especially for increasing values of suspension stiffness.

The imperfections that cause this difference essentially result from bench building processes, such as machining of expanded PVC plates and fastening holes, misalignment of 1045 steel rods and friction between linear bearings and rods in the vertical motion of the sprung and unsprung masses (Fig. 2).

Although the bench has not presented a fundamental frequency of 1 Hz representing the ideal case for the suspension system, this bench can still be used as a teaching tool in vehicle dynamics and vibration courses.

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