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VERTICAL VIBRATION ANALYSIS OF A FORMULA STUDENT VEHICLE USING DIFFERENT SUSPENSION SYSTEM CONFIGURATIONS

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Abstract: *The paper captures studies on the influence of vertical vibrations on the dynamic behavior of the racing vehicle of the ART TU Cluj-Napoca Formula Student Team. In order to evaluate the effects of mechanical vibrations, at different frequencies, on the vehicle during its movement on the circuit, respectively on its dynamics, two possible configurations of the suspension system, specific to Formula Student vehicles, were proposed - the configuration with spring, damper and anti-roll bar and the configuration with spring and damper. In this respect, based on the input data, and by developing two working models in MATLAB software - for a quarter vehicle and for a bicycle-type vehicle, comparative results were obtained, with graphical interpretation, which allow to identify the influences generated by vertical vibrations on the dynamic behavior of the considered vehicle.*
Key words: *racing vehicle, Formula Student, suspension system, vibrations, dynamic behavior*

1. INTRODUCTION

The wheel assembly, being considered the unsprung mass of the vehicle, is mechanically connected to the vehicle body, and is also called the suspended mass of the vehicle. The connection between the two masses is made via the suspension system, which has the purpose of allowing relative movements between these two structures. When the vehicle is travelling on a racetrack, the suspension system is necessary to ensure the comfort and stability of the vehicle. In other words, the suspension system improves comfort when the vehicle is travelling on the road as well as dynamic performance by protecting the occupants from the shocks and vibrations caused by the tires in contact with the road surface [1, 2, 5, 6, 10, 12-15, 18, 19, 21, 24-26].

Vehicle manufacturers use a wide range of options for suspension systems, which are essential for achieving both comfort and dynamic performance, depending on the type and destination of the vehicles. There are many alternatives available, and the choice of suspension system is essential when determining the distinctive characteristics of each vehicle [7].

Formula Student is an internationally known engineering competition founded by the Society of Automotive Engineers (SAE) that challenges students to design, build and race a Formula Student racing vehicle. The competition gives students a great opportunity to demonstrate their educational experience in real-world engineering projects, encouraging creativity, teamwork, and the development of practical skills. The objective of the participating teams is to build a prototype of a high-performance racing vehicle. Teams are evaluated on several criteria, including design, performance, cost, and commercial presentation [4, 15, 18, 22, 23, 28, 29].

One of the most important components that receives special attention during the FSAE vehicle design process is the

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suspension system. The suspension system must always maintain the largest possible contact area of the tires with the road surface during all accelerations, whether longitudinal or lateral. In order to keep the tire as perpendicular as possible to the road surface in all driving situations, to absorb shocks and to have rigid components that do not deform under the loads applied to them, the suspension must have an optimum kinematic structure [3, 11].

Due to its improved kinematic response compared to other available suspension system designs, the double A-arm suspension system (Fig. 1), also known as double wishbone suspension, is commonly found in high-performance vehicles. This type of suspension significantly improves the vehicle's handling and ride safety, while reducing the side slip angle, thus enhancing the vehicle's dynamic behavior. In addition, it seeks to provide good steering control during maneuvering, isolates high frequency vibrations caused by tire oscillations in response to the road profile, and helps the vehicle to respond to forces that occur while driving on the road [13, 16-18, 21, 23].

The double A-arm suspension system is also used in Formula 1 racing vehicles due to its simple construction and lightweight components. It has a short upper arm and a long lower arm (see Fig. 1) that allows the camber angle to be adjusted as the wheel moves vertically [13, 16, 22].

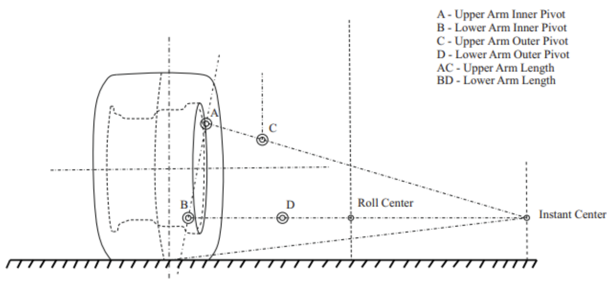

Fig. 1. Double A-arm suspension model [13].

In terms of construction, the double A-arm suspension system has various basic components such as A-arms, push/pull rod, spring and damper system and additional anti-roll bars (or stabilizer bars). The anti-roll bar can be used in either the front or rear axle configuration of the vehicle to reduce overall roll during excessively tight turns at high travel velocities. In addition, FSAE vehicles are fitted with anti-roll bars to improve cornering performance, and the increased torsional stiffness will produce an understeer effect on the front axle and an oversteer effect for the rear axle. Pushrods and anti-roll bars are considered tuning components in the racing community as they can modify and improve the dynamic behavior of the vehicle [15].

Starting with the assumption that the surfaces of the roadways are far from flat, when driving on different types of roads, vehicles are subject to a variety of disturbances, including bumps, braking forces, propulsive forces, and centrifugal forces when cornering. All these variables make the driver uncomfortable and influence the handling of the vehicle. Limiting vertical acceleration fluctuations of the vehicle body will improve comfort. In addition, reducing the vertical force variations that each tire changes with the road are one way to ensure good vehicle handling. Suspension stiffness and vehicle weight distribution are the main factors influencing comfort and handling [9].

Frequencies between 0.25 and 25 Hz are taken into consideration for road vehicles, while higher frequencies are considered for racing vehicles. Due to their vertical elasticity and low mass, tires, among other characteristics, can absorb small road disturbances at high frequencies. However, tires have little effect and can be considered stiff at frequencies below 3 Hz [9].

When investigating the handling characteristics of the vehicle, it is also important to analyze the suspension geometry to assess the springs and dampers in order to establish criteria for choosing the appropriate stiffness and damping frequencies for each suspension according to the road surface and the intended use of the vehicle [8].

The aim of this work is to analyze the effects of vibrations at different frequencies on the ART TU Cluj-Napoca Formula Student Team's vehicle by simulating two suspension system configurations. The influence of vibrations during vehicle movement on the dynamic performance when running on the road circuit during SAE competitions could be observed graphically by means of numerical calculations performed using MATLAB software based on this data.

2. COMPUTERIZED SIMULATION

For the computer simulation, it is proposed to develop two working models to allow a detailed analysis of the vehicle's dynamic performance using two suspension system configurations, namely the spring, damper, and anti-roll bar configuration (hereafter referred to as configuration 1) and the spring and damper configuration (referred to as configuration 2). Thus, the model for a quarter vehicle and the model for a bicycle-type vehicle were used to analyze the influence of vibrations during vehicle movement on the SAE circuit during competitions.

2.1. Quarter vehicle model for a vehicle vibration

The most practical suspension system model is the quarter-vehicle model (Fig. 2), as it allows quick simulations of the suspension system. Thus, this model, comprising two solid masses, m_s and m_u, referred to as the suspended and unsprung mass of the vehicle, is used to calculate vertical vibrations. The unsprung vehicle mass, m_u, represents one wheel of the vehicle, consisting of the hub, spindle, rim, and tire, while the sprung vehicle mass, m_s, represents one quarter of the vehicle body. This analysis also considers the stiffness k_s, which represents the stiffness of the tire, and the spring stiffness k_u.

Vibration analysis for this type of model was carried out on a wheel on the front axle. Starting from the initial vehicle data specific to this configuration (Fig. 3), the vehicle suspended mass m_s, vehicle unsprung mass m_u and tire stiffness k_u, show constant values, while the spring stiffness k_s is variable depending on the type of vehicle suspension system configuration considered.

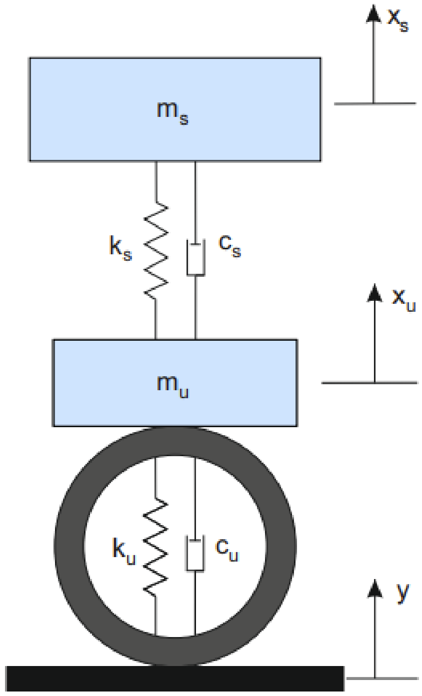

Fig. 2. A quarter vehicle model [12].

In the case of configuration 1, the coefficient k_{f1}, which represents the stiffness of the anti-roll bar on the front axle, shall be considered, and for the analysis of configuration 2, the coefficient k_{f2}, which represents the stiffness of the spring on the front axle, shall be considered.

Once the type of configuration on which the vibration analysis is performed has been determined, the natural frequencies of the system were determined using the relation [20]:

$$\omega_n = \sqrt{\frac{k}{m}}, \text{ in rad/s,} \tag{1}$$

where: k is the stiffness of the suspension system, in N/m; m - unsprung/sprung mass, in kg.

Knowing the values of the natural frequencies of both configurations of the suspension systems considered, the amplitude of transverse displacement for the situation where the vehicle is travelling at maximum speed was determined with the relation [27]:

$$A = \sqrt{x_0^2 + \frac{\dot{x_0}^2}{\omega_n^2}}, \text{ in mm,} \qquad (2)$$

in which: x_0 represents the displacement (x_0 = 0); $\dot{x_0}$ - vehicle travel velocity, in km/h; ω_n - natural frequency of the system, in rad/s.

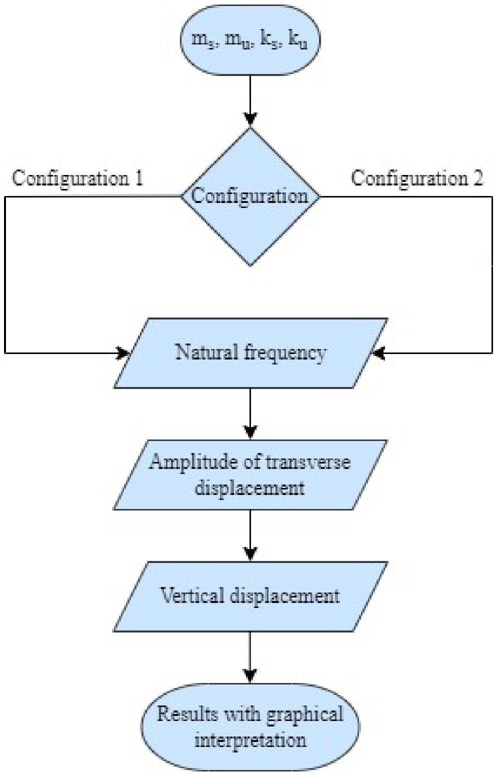

Fig. 3. Working methodology for a quarter vehicle model.

Using the natural frequency and the amplitude of the transverse displacement, the vertical displacement of the sprung mass and the unsprung mass is then determined by the relation [20]:

$$x(t) = A \cdot \sin{(\omega \cdot t)}, \text{ in mm,} \qquad (3)$$

where: A is the amplitude of the transverse displacement, in mm; ω - the natural frequency of the system, in rad/s; t - duration of the simulation, in s.

Thereafter, the vertical displacement of the sprung and unsprung mass can be plotted as a function of test duration, with a 60-second model.

2.1. Bicycle vehicle model for a vehicle vibration

Figure 4 shows the configuration of the vibration analysis system for a bicycle model vehicle. The vehicle body is assumed to be a rigid m-bar, representing half of the total vehicle mass. The moment of inertia of the lateral mass of the body I_y, represents half of the total moment of inertia. The front and rear axle wheels have masses m_1 and m_2 respectively and the stiffness of the tires is indicated by the parameters k_{t1} and k_{t2} [15].

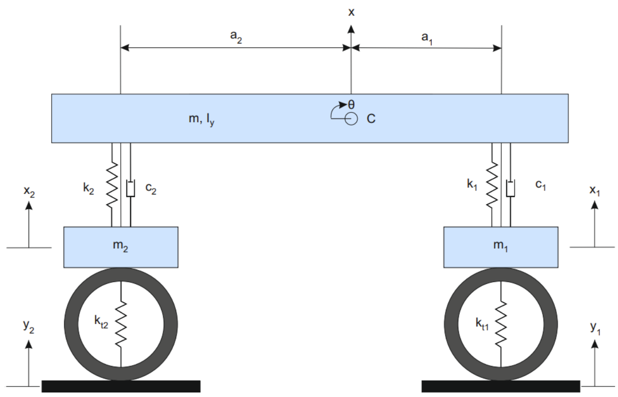

Fig. 4. A bicycle vehicle model [12].

In the vibration analysis for the bicycle vehicle model, the maximum displacements for each suspension system configuration are determined. Using the initial data specific to this model (Fig. 5), namely vehicle mass m, the frontal wheel mass m_1 and the rear wheel mass m_2, tire stiffness k_t, front axle spring stiffness k_1 and rear axle spring stiffness k_2, longitudinal distances between center of gravity and front axle a_1 and rear axle a_2, and the moment of inertia of the lateral body mass I_y, the natural frequencies of the vehicle during displacement are determined.

The type of configuration used is first determined by adopting the corresponding values of specific suspension stiffness on the front and rear axle. Therefore, the mass matrix [m], given by the relation [12]:

$$[m] = \begin{bmatrix} m & 0 & 0 & 0 \\ 0 & I_y & 0 & 0 \\ 0 & 0 & m_1 & 0 \\ 0 & 0 & 0 & m_2 \end{bmatrix}, \qquad (4)$$

is proposed to be solved.

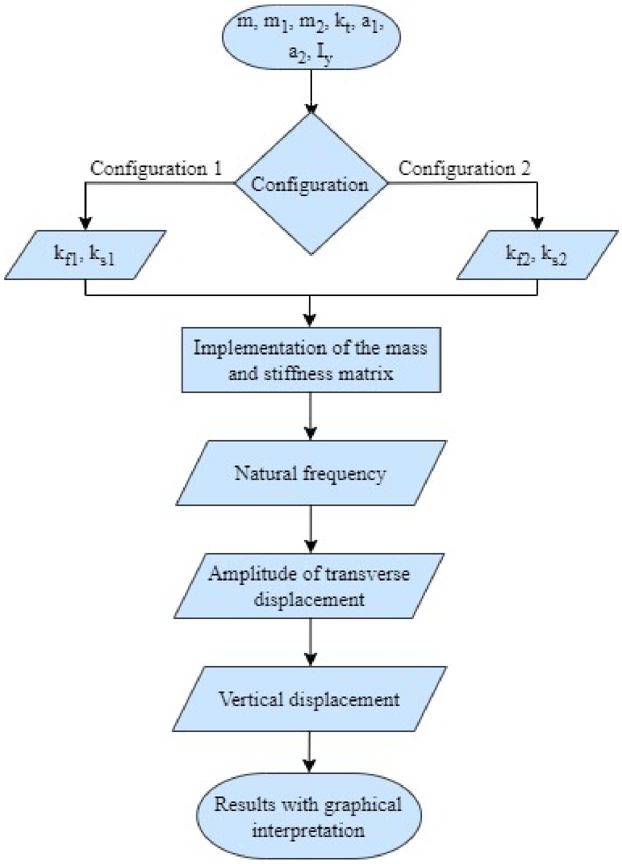

Fig. 5. Working methodology for a bicycle vehicle model.

In the following, the matrix for stiffness [k] is solved by the relation [12]:

$$[k] = \begin{bmatrix} k_1 + k_2 & a_2k_2 - a_1k_1 & -k_1 & -k_2 \\ a_2k_2 - a_1k_1 & k_1a_1^2 + k_2a_2^2 & a_1k_1 & -a_2k_2 \\ -k_1 & a_1k_1 & k_1 + k_{t_1} & 0 \\ -k_2 & -a_2k_2 & 0 & k_2 + k_{t_2} \end{bmatrix}. \quad (5)$$

Once relations (4) and (5) are solved, the natural frequencies need to be determined. For this purpose, the characteristic equation for undamped, free-moving vibrations is given by the relation [12]:

$$D = det\big[[k] - \omega^2[m]\big], \quad (6)$$

where: [k] is the stiffness matrix; ω - the natural frequencies of the system, [m] - the mass matrix.

In relation 6, the natural frequency ω is an unknown parameter, which belongs to the numerical computation. The natural frequency is determined by creating a symbolic scalar variable using the *syms* function [27], specific to MATLAB software. After performing this step, an 8th degree equation is obtained, while to obtain the natural frequency values, a polynomial is created to determine its roots using the MATLAB software *roots* function [30], where the roots of the polynomial represent the natural frequency values.

Knowing the values of the natural frequencies, it is possible to determine the amplitude of the transverse displacement for the situation where the vehicle is travelling at maximum speed, using relation 2. Next, the vertical displacement of the mass is determined using relation 3. Then, the vertical displacement of the suspended mass is plotted against the duration of the test with the 60-second model.

3. OBTAINED RESULTS

Following the MATLAB simulation, the results were obtained in graphical form, representing the vertical displacement of the suspended mass and the unsprung mass of the vehicle as a function of the duration of the model test, considering the adopted configuration of the suspension system.

Vertical vibrations analysis by using the quarter vehicle model. In the case of suspension system configuration 1, it can be seen in figure 6 that the high value of the stiffness of the suspension system causes a reduced vertical displacement of the suspended mass compared to the vertical displacement of the suspended mass in configuration 2 (Fig. 7). Therefore, the maximum value of the vertical displacement of the suspended mass is $1.86 \cdot 10^{-4}$ mm for configuration 1 and $4.43 \cdot 10^{-4}$ mm for configuration 2. From the point of view of the unsprung mass stiffness value, because the value of the tire stiffness coefficient and the mass value is constant for both configurations, the maximum unsprung mass displacement of $4.4 \cdot 10^{-4}$ mm.

Vertical vibration analysis by using bicycle vehicle model. Figure 8 captures the vertical displacement of the suspended mass as a function of model test duration for configuration 1. A specific aspect of this configuration is that the amount of vertical displacement is reduced due to the high stiffness of the suspension system. At the same time, the graphical representation shows that the lowest vertical mass displacements are associated with high system frequencies while high displacements are due to low frequencies. Thus, the maximum mass displacement found at 49.2 Hz has a value of $1.01 \cdot 10^{-4}$ mm and $1.01 \cdot 10^{-3}$ mm at 5 Hz. Regarding the vertical displacement of the suspended mass specific to configuration 2 (Fig. 9), it is evident that at a frequency of 24.6 Hz a maximum displacement of the suspended mass of $2.04 \cdot 10^{-4}$ mm occurs and $1.1 \cdot 10^{-3}$ mm at a frequency of 4.6 Hz.

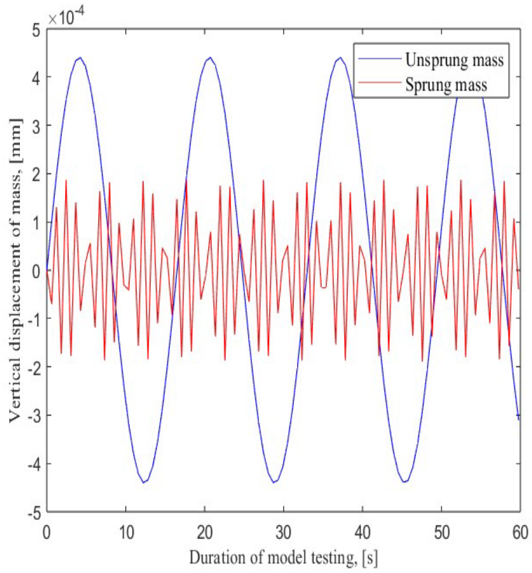

Fig. 6. Vertical displacement of mass as a function of model test duration, for the quarter vehicle model, configuration 1.

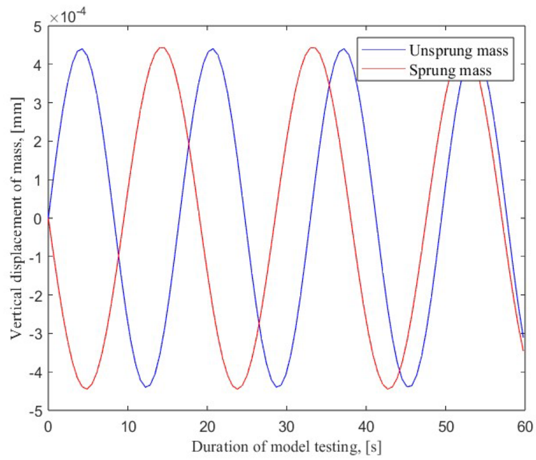

Fig. 7. Vertical displacement of mass as a function of model test duration, for the quarter vehicle model, configuration 2.

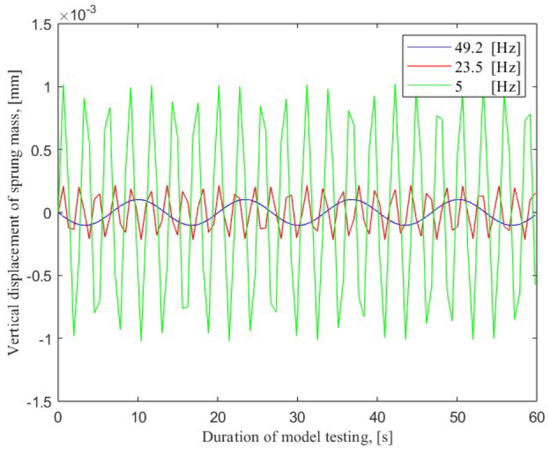

Fig. 8. Vertical displacement of mass as a function of model test duration, for the bicycle vehicle model, front axle, configuration 1.

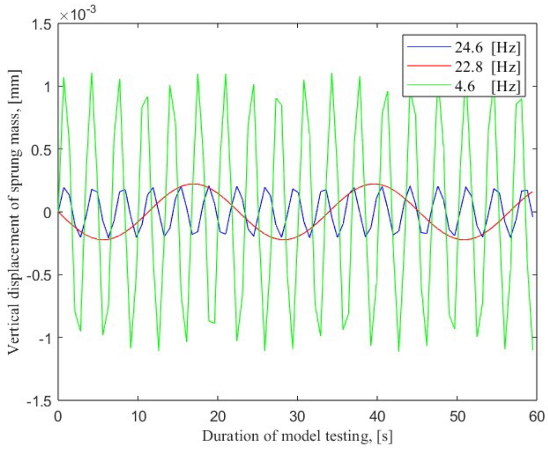

Fig. 9. Vertical displacement of mass as a function of model test duration, for the bicycle vehicle model, front axle, configuration 2.

4. CONCLUSIONS

In accordance with the proposed objective, an analysis of the vibrations existing during the track driving of a Formula Student vehicle has been carried out to investigate their influence on the vehicle. For this purpose, each type of vehicle vibration analysis model was run. At the same time, a comparison of the results obtained was carried out and two types of suspension system configurations specific to Formula Student vehicles were proposed for analysis, namely the spring, damper, and anti-roll bar configuration (configuration 1), respectively, the spring and damper configuration (configuration 2).

In the case of each of the two-suspension system optimization considered for the vehicle vibration analysis, the following conclusions are noted:

- the spring, damper and anti-roll bar suspension system configuration ensures a high stiffness of the suspension system, leading to high natural frequencies, but results in very little vertical mass displacement when the vehicle is in motion;
- the spring and damper suspension system configuration results in a significant decrease in natural frequencies occurred during track travel, but this leads to an increase in the vertical displacement of the suspended mass;
- for both suspension system configurations, it is noted that there is a high vertical mass displacement at low natural frequencies and insignificant vertical mass displacements at high natural frequencies;

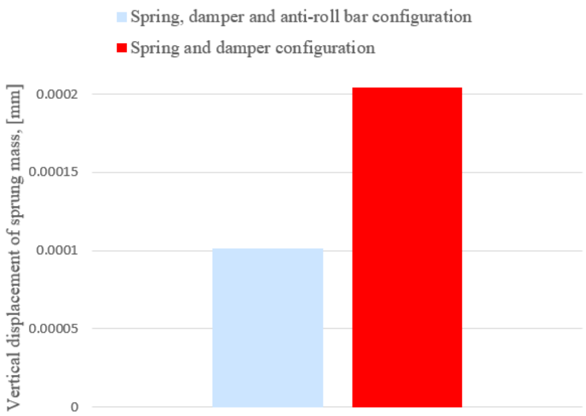

Fig. 10. Comparative analysis of the vertical displacement of the suspended mass in the model for a bicycle vehicle.

- in a comparative study at the front axle level of the vehicle on the vibration analysis model for a bicycle vehicle model, it was found that the spring, damper and anti-roll bar suspension system configuration provides a maximum vertical displacement of the suspended mass that is approximately 102% less than that generated by the spring and damper configuration (Fig. 10), and the maximum natural frequency generated by the configuration of the spring, damper and anti-roll bar suspension system is about 50% higher than that obtained with the spring and damper configuration (Fig. 11).

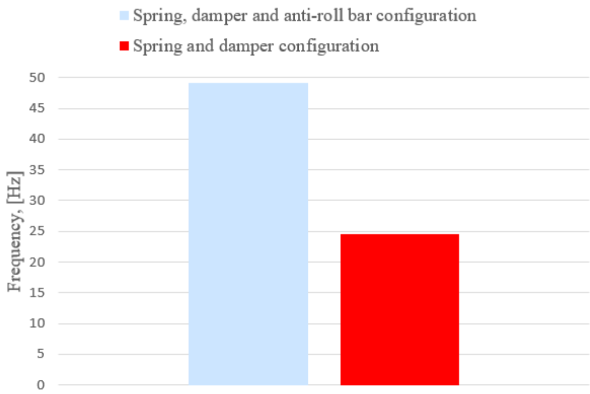

Fig. 11. Comparative frequency analysis within the model for a bicycle vehicle.

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Analiza vibrațiilor verticale ale unui autovehicul pentru Formula Student, utilizând diferite configurații ale sistemului de suspensie

Rezumat: În lucrare sunt surprinse studii privind influența vibrațiilor verticale asupra comportamentului dinamic al autovehiculului de curse din cadrul echipei studențești de Formula Student, ART TU Cluj-Napoca. Pentru evaluarea efectelor dezvoltate de vibrațiile mecanice, la diferite frecvențe, asupra autovehiculului în timpul deplasării acestuia pe circuit, respectiv asupra dinamicii lui, s-au propus două configurații posibile ale sistemului de suspensie, specifice autovehiculelor de Formula Student - configurația cu arc, amortizor și bară antiruliu și configurația cu arcuri și amortizoare.

În acest sens, pe baza datelor de intrare, și prin dezvoltarea a două modele de lucru în programul MATLAB - pentru un sfert de autovehicul și pentru un autovehicul de tip bicicletă, s-au obținut rezultate comparative, cu interpretare grafică, care permit identificarea influențelor generate de către vibrațiile verticale asupra comportamentului dinamic al autovehiculului considerat.

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