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Frequency Analysis of an Arm of Macpherson Suspension on a Passenger Car

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***Abstract*. The smoothness, ride comfort, safety and handling of the car depends on the manner of suspension design and its corresponding details. One of main functions of the arm and the rubber bushings mounted on it is to reduce the vibrations and the noise transferred from the road to the passenger car components. This article presents the results of a frequency analysis of an arm of the MacPherson front independent suspension. For this purpose, three-dimensional geometric models of the arm and rubber bushings are created via finite element analysis (FEA) software SolidWorks, where Skoda Octavia passenger car was used as a prototype. The axial, radial and torsional stiffness of the rubber bushings were determined through analytical dependences and FEA. The obtained results had been used for calculation of the natural frequencies and mode shapes of the arm which were compared with experimentally obtained data.**

Keywords: natural frequency, mode shape, arm, suspension, rubber bushing, FEA, experimental study.

I. INTRODUCTION

Suspension reduces dynamic loads on the vehicle body and wheels by reducing the amount of force of shocks and vibrations. In some cases the suspension adjusts the position of the vehicle chassis.

During the suspension design the natural frequency and the vibration amplitude of the unspring components are one of the main focus elements.

It is well known that an optimal driving comfort is achieved when the natural frequency is in the range of 1-1,5 Hz. Increase of natural frequency above 1,5 Hz, leads to decrease of the comfort. In real life vehicle applications, the natural frequency changes between 0 to 20 Hz due to road imperfections [2]. The range of the variations of the natural frequency must be taken in consideration during the design process of the suspension and its various components which are exposed to dynamic stresses. That's why the determining of natural frequencies and mode shapes is crucial.

The control arm of the suspension were investigated by various studies [2,3,5,6,7,8,9,10, 11,12,13] by means of FEA software. Study [5,7] provides results regarding the static strength and frequency analysis of lower arm of double wishbone suspension. Results regarding static strength analysis of MacPherson independent suspension obtained by FEA are presented in [6,8]. Studies [5,6,9,10,11,12] show topology optimization of an arm while in [3,7,13] dynamic analysis of lower arm via numerical and experimental study is described.

The purpose of the study is to determine the natural frequencies and mode shapes of a front arm of a MacPherson type suspension of a passenger car Skoda Octavia. To achieve the goal a FEA and up-to-date software tools Simulation module of SolidWorks were used. The simulation results of the natural frequencies

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were confirmed by experimentally which is also part of the current study.

II. DETERMINATION OF THE SIFFNESS OF ARM RUBBER BUSHING

The MacPherson type is mainly used for front suspension in modern passenger cars, which stands out for its compactness, small mass and significantly less ball joints. The MacPherson type suspension includes the following main elements: arm, elastic element (spring) and shock absorber. Figure 1 shows a three-dimensional geometric (3D) model of a suspension, position 1 shows the arm, position 2 shows rubber bushing 1 and position 3 shows rubber bushing 1. The the axial, radial and torsional stiffness are presented in fig.1.

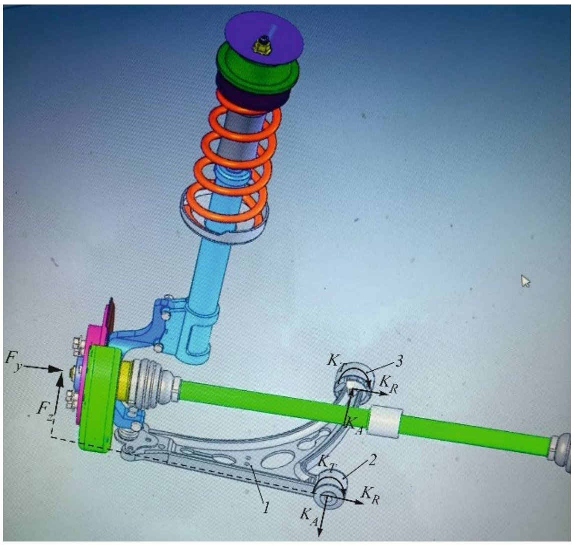


Fig. 1. Three-dimensional geometric model of a suspension and radial, axial and torsional stiffness in rubber bushing of an arm.

Natural frequencies and mode shapes of the arm are most often determined by performing a physical experiment, analytically determined or by using FEA.

The correct setting of the supports is critical in determining the natural frequencies and mode shapes of the suspension and the arm respectively.

The static stiffness of the rubber bushings can be calculated by defined mathematical equations, by FEA or experimentally.

The radial stiffness of the rubber bushing can be determined by the dependence [1, 9]

$$K_R = \frac{7,5.\pi.L.G}{\ln\left(D/d\right)} k_l, \tag{1}$$

where k_l is the form factor and it is defined by the graphical dependence from [9];

G is the shear modulus, MPa, it is defined by Shore hardness Hs, $G = 0,117e^{0,034Hs}$ or through the graphical representation $G = f(Hs)$ [4];

D, d, L are the outer and inner radius respectively and the length of the rubber bushing, mm.

The dimension ratio η is defined as [9]

$$\eta = \frac{L}{0,5\left(D-d\right)}, \tag{2}$$

The axial stiffness and torsional stiffness of the rubber bushing can be determined by dependencies [1, 9]

$$K_A = \frac{2.\pi.L.G}{\ln\left(D/d\right)}, \tag{3}$$

$$K_T = \pi.L.G.10^{-3}\left(\frac{1}{d^2} - \frac{1}{D^2}\right)^{-1}. \tag{4}$$

III. METHODOLOGY OF STUDY

The object of the study is an arm of the MacPherson front independent suspension on passenger car Skoda Octavia. Figure 2 shows 3D model of an arm.

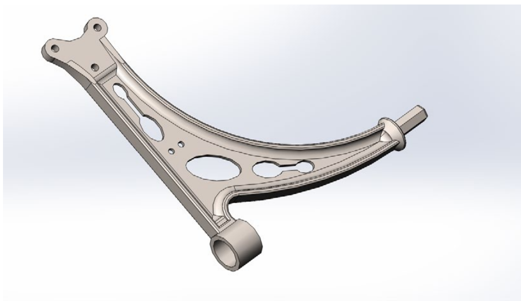
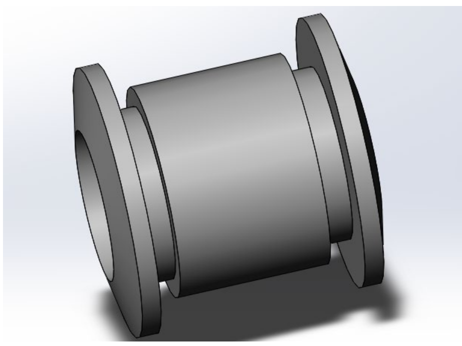


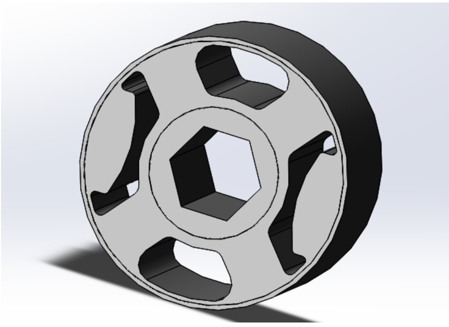
Fig. 2. Three-dimensional geometric model of an arm.

Rubber bushings were used as elastic supports of the arm therefore determining of stiffness was needed for correct definition of fixing.

Figure 3a and 3b shows 3D model of the rubber bushing 1 and 2, respectively.



a) rubber bushing 1



b) rubber bushing 2

Fig. 3. Geometric models of the rubber bushings.

The stiffness of the rubber bushings were determined via non-linear SolidWorks Simulation analysis. The elastic properties of the rubber bushings were estimated using the Mooney-Rivlin material model with five constants. The selected Shore hardness of the rubber is 70Hs. The experimental results of stress-strain relation in bi-axial tension of 70Hs from [4], had been used as an input for automatically calculation of the material constants conducted by SolidWorks. Rubber's poisson's ratio is 0,49 [1] while the density is 1130 kg/m^3 [1]. The metal parts of the rubber bushings are made of normalized steel 4340. A three-dimensional curvilinear finite element mesh was used for the rubber bushings.

Natural frequencies and mode shapes are determined by frequency analysis of SolidWorks Simulation. The arm is made of cast steel, according to EN10293. Its mechanical properties are shown in Table 1.

TABLE 1 MECHANICAL PROPERTIES OF THE ARM

Elastic modulus, Pa	**Poisson's ratio**	**Mass density, kg/m^3**
$2,1.10^{11}$	0,28	7800

Figure 4 shows arm fixation established by elastic supports on the mounting locations of the rubber bushings surfaces.

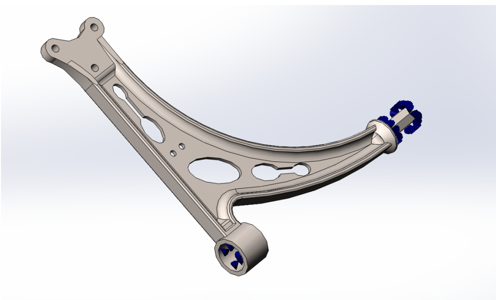


Fig. 4. Elastic supports.

A three-dimensional curvilinear mesh was generated (fig. 5). It includes 206 856 nodes and 129 636 elements.

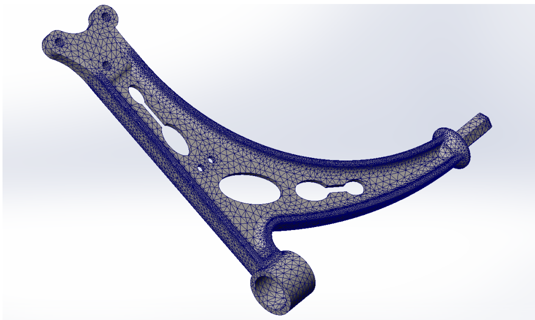


Fig. 5. FEA mesh.

The main goal of the experimental study was to determine the natural frequencies of the arm. It was carried by measuring and recording the accelerations along the three axes x, y and z at the same time. The measuring equipment includes: ADXL335 accelerometer module, power supply, multifunction I/O Device NI USB-6343, PC and a specialized software developed in LabView environment.

The accelerations on the three axes were registered by ADXL335 accelerometer. The ADXL335 is a small, thin, low power, complete 3-axis accelerometer with signal conditioned voltage outputs. The product measures acceleration with a minimum full-scale range of ±3 g. The power supply of the accelerometer is a constant stabilized voltage of 3,3 V. The information from the accelerometer is provided to the PC through the multifunction I/O Device NI USB-6343. The maximum data registration speed from the Device NI USB-6343 is 500 kSample/s, input range: ±0,2 V; ±1 V; ±5 V; ±10 V. For registration and processing of the accelerometer signals, specialized software in the environment LabView has been developed (Fig. 6).

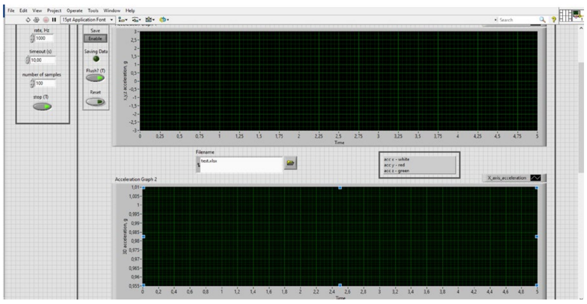


Fig. 6. Software front panel.

Figure 7 shows the object, the hammer and the measuring equipment for the experiment.

Fig. 7. Experimental determination of natural frequencies.

The accelerometer was attached to the arm. The mass of the accelerometer is 1,27 g and the mass of the arm was 3109 g. The total mass of the accelerometers of the suspension arm had negligible effects on the measurement. The FFT method in the LabView software was used, through which the data for the total acceleration in three- dimensional space were processed space. The accuracy of the method depends on the frequency resolution (1 Hz) of the measurement, since it determines the displayed peak amplitude on the PC screen that was measured.

IV. Results and Discussion

Table 2 presents the results of the axial, radial and torsional stiffness of the rubber bushings obtained by FEA. For rubber bushing 1, results obtained by dependencies (1), (3) and (4) are also presented.

TABLE 2 STATIC STIFFNESS OF THE RUBBER BUSHINGS

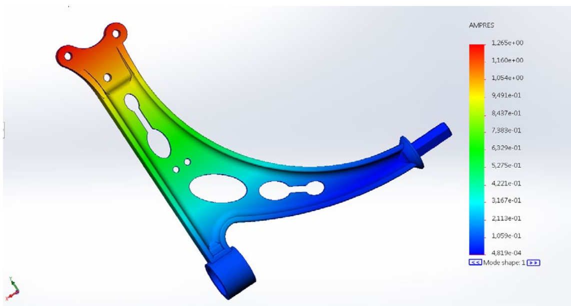
Stiffness	**Values**		
	Rubber bushing 1		***Rubber bushing 2***
	Formula	***FEA***	***FEA***
Axial stiffness (N/mm)	526	697	144
Radial stiffness (N/mm)	4143	5806	401
Torsional stiffness (Nm/deg)	2,5	2,55	1,26

From the results presented in table 2 regarding the stiffness of the rubber bushing 1, it is observed that there is a significant difference between the results obtained by FEA and by the dependencies. This is mainly due to the simplified geometric shape of the rubber bushings in the analytically calculation. In [9], it was found that the results for the stiffness obtained by FEA are insignificant different from the experimental results, as opposed to those get from the dependencies.

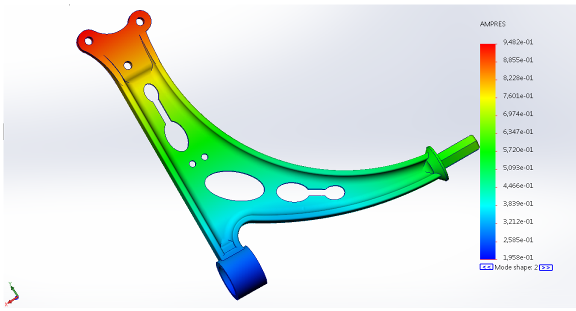
Table 3 presents the obtained results of the six natural frequencies of the arm. The first, second and third mode shapes are shown on Fig. 8 a, b and c, respectively.

TABLE 3 NATURAL FREQUENCY FROM FEA

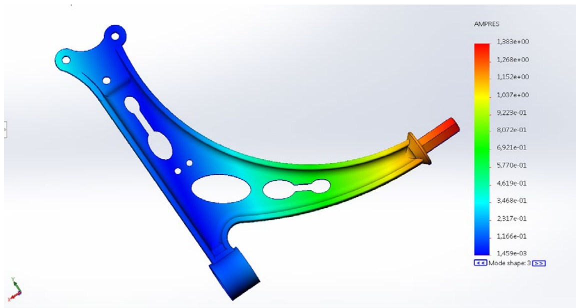
Mode number	**Natural Frequency, Hz**
1	10,03
2	60,52
3	112,01
4	114,13
5	188,51
6	305,7



a) first mode shape



b) second mode shape



c) third mode shape

Fig. 8. Mode shapes.

Figure 9 presents the experimental results.

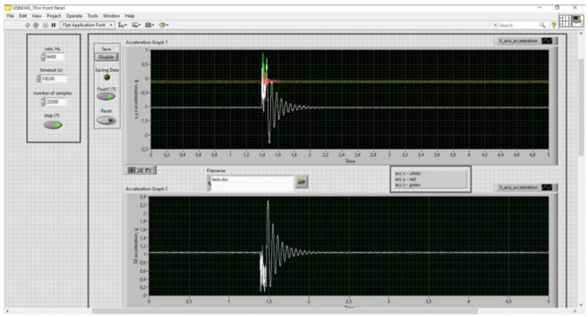


Fig. 9. Experimental acceleration results.

Table 4 presents the results regarding the natural frequencies obtained by FEA and results obtained experimentally.

TABLE 4 NATURAL FREQUENCY

Mode number	**FEA Natural frequency, Hz**	**Experimental Natural frequency, Hz**
1	10,03	16
2	60,52	75,6
3	112,01	136
4	114,13	160
5	188,51	250
6	305,7	350

The results obtained of natural frequencies by FEA are comparable to those obtained experimentally.

V. Conclusions

Based on the performed study the following conclusions are made:

It is preferable to use the FEA to determine the stiffness of rubber bushings with a complex geometry.

The results shows the lowest value of the natural frequencies of the arm is approximately 10 Hz. The next value of natural frequencies is significantly greater than the frequency of the excitation forces generated by the road surface irregularities (from 0 to 20 Hz) [2]. This means that when the car is in motion, no significant vibrations will occur in the arm, which would lead to a deterioration in the comfort of the passengers.

An experimental frequency analysis was performed to validate the FEA model developed of a front arm of a MacPherson type suspension. The results obtained by FEA are close to the results obtained experimentally.

The obtained results of the natural frequencies of the arm with fixation established by elastic supports are different from the results for the frequencies in [13], because the arm is attached with fixed supports and the geometric model is not exactly the same.

The designed suspension model, the study methodology and the obtained results can be used to topology optimization and fatigue analysis.

ACKNOWLEDGMENTS

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