A TRANSZVERZÁLIS LAPRUGÓ NUMERIKUS MODELLEZÉSE ACÉL ANYAGGAL

NUMERICAL MODELLING OF TRANSVERSE SPRING LEAF WITH STEEL MATERIAL

Seif Eddine Habbachi [1], Attila Baksa [2], Marwen Habbachi [3]
[1]Habbachi.seifeddine@enib.u-carthage.tn, National Engineering School of Bizerte (ENIB), Tunisia
[2] attila.baksa@uni-miskolc.hu, Institute of Applied Mechanics, University of Miskolc, Hungary
[3] marwen.habbachi@student.uni-miskolc.hu, Institute of Applied Mechanics, University of Miskolc, Hungary

ABSTRACT

Automobile suspension systems (springs, shock absorbers, linkages) are becoming more important in today's world! They contribute to a vehicle's handling, braking, protect the vehicle itself, and protect occupants, cargo or luggage from damage and wear. Furthermore, suspension systems have many functions, like preventing the vehicle body and frame from road shocks, giving it stability, and providing comfort as well. In the meantime, suspension equipment needs to have lower weight, maximum deflection, low maintenance, and a low operating cost. During this work, we will focus only on the leaf spring suspension, more precisely, the transverse leaf spring. Our project is a numerical modelling of a transverse leaf spring using the finite element method. We will investigate the maximum deflection and stress distribution along a deformable body, with two-common vertical and anti-metrical stiffnesses configurations. We will compare the results with a theoretical model and with data provided from the Iveco company, to validate them. Finally, an important comparison will be presented between the numerical results and the provided data. In fact, the results will show that the structure under the applied force is safe.

***Keywords**: Transverse leaf spring; Numerical analysis; Vertical stiffness; bump rubber; hyper-elastic model; Mooney-Rivlin solid, Automotive*

1. Introduction

The importance of suspension systems in a commercial vehicle is one of the essential components in the automotive industry which needs continuous development and innovation to enhance productivity, improve existing properties, while minimizing energy (fuel) consumption. Replacing the conventional leaf spring with a transverse one is an objective for many automotive companies. In this case, the company, Iveco has proposed a new solution, to fabricate a leaf spring which can fulfill simultaneously, the absorption of the load (shock) and the role of the stabilizer bar. It is difficult to adopt an advanced leaf spring using a composite material
for a transverse leaf spring in order to increase the performance of the material and minimize weight. The main factor in designing a new leaf spring, is material strain energy. R.B. Charde and D.V. Bhope reported that spring deflection is a result of potential energy stored in the form of strain energy, due to loading and material elasticity. A leaf spring is usually made from carbon steel (plain), with a carbon percentage which varies between 0.9 and 1.0%. After the process of forming, it is followed by heat treatment of the material. This leads to better strength and greater material deflection. It enhances fatigue resistance that can cause material damage after a short cycle of work. The high cost of a vehicle (60 – 70%) is attributed to the material quality used and selected [1]. Results shows that composite leaf springs are better compared to the conventional leaf springs as reported in [2]. The authors also were able to use E-glass/Epoxy composite material to solve a major issue in the vehicle, providing lighter weight.
The aim of this research is to investigate a new product: A transverse leaf spring, for a specific client and compare the results obtained numerically, with analytic studies while verifying the resistance of the product under introduced loading.

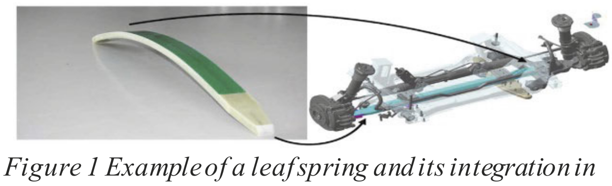

Figure 1 Example of a leaf spring and its integration in the suspension system

2. Modeling of transverse leaf spring

2.1. Material properties

- Leaf Spring

CATIA V5 was used to model the required parts, the leaf spring is made from high strength quench and tempering steel (51CrV4) which consists of carbon and other chemical elements as shown in Table 1 [3]. In order to increase its mechanical properties especially the tensile strength (Rm) and the yield strength ($Rp_{0.2}$), these leaves are subjected to heat treatment. The relationship between the strength and the true strain is illustrated as an elasto-plastic behavior obtained after fracture tensile tests were done on three specimens, seen in Figure 2. The average value of the mechanical properties is shown in Table 2.

Table 1 Chemical composition of the steel grade (51CrV4)

		C%	Si %	Mn%	P%	S%	Cr%	V%	Al
52CrV4 (EN10089:2002)	Min	0.47	-	0.70	-	-	0.90	0.100	-
	Max	0.55	0.40	1.10	0.025	0.025	1.20	0.250	-
Present heat	-	0.52	0.34	0.98	0.014	0.009	1.06	0.12	0.025

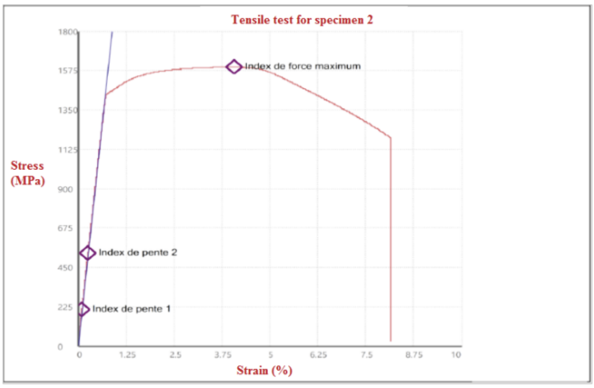

Figure 2 Tensile test curve of material (51CrV4)

Table 2 Mechanical properties of 51CrV4 material

Specimen	1	2	3	Average value
Ultimate tensile strength Rm (MPa)	1600	1595	1596	1597
Yield strength σ_y (MPa)	1472	1464	1466	1467
Elongation to fracture *(A %)*	12.43	12.50	11.93	12.29
Striction *(Z %)*	45.05	45.05	41.67	43.92

- Bump stop rubber buffer

Bump stops as shown in Figure 3, are vital and critical components of suspension systems in a vehicle, since they prevent its components from damaging over-compression, while eliminating harsh bottoming of the suspension, including the ones caused by vibration of the leaf springs.

In this instance they are modeled in Abaqus as a deformable body, while having a mechanical behavior different from the leaf spring. Classified as a rubber material, this material exhibits nonlinear mechanical properties and large deformations due to their complex molecular structure [4]. There are several laws such Neo-Hookean, Ogden, etc. But the Mooney-Rivlin solid law, [5] which is a special case of the generalized Rivlin model (called the polynomial hyper elastic model), illustrated in equation (1), was adopted to describe its behavior under loading. The strain energy for a two-parameter Mooney-Rivlin model of invariants scalar $\bar{I}_1$ and $\bar{I}_2$ of the left Cauchy-green deformation tensor $\boldsymbol{b}$ is written in equation below.

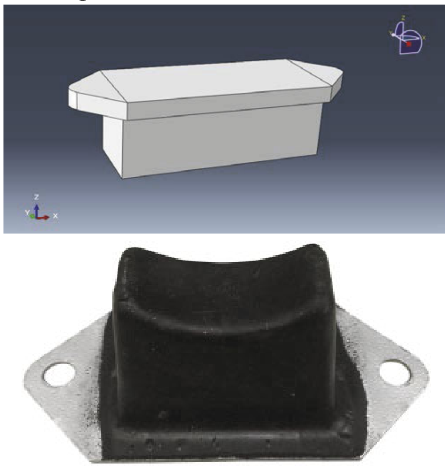

Figure 3 Elastomer bumper part

$$W = \sum_{p,q=0}^{N} C_{pq}(\bar{I}_1 - 3)^p(\bar{I}_2 - 3)^q + \sum_{m=1}^{M} \frac{1}{D_m}(J-1)^{2m} \quad (1)$$

Where $C_{00} = 0$, C_{pq} are material constants related to the distortional response and D_m are material constants related to the volumetric response. For a compressible Mooney–Rivlin model, we have $N = 1$, $C_{01} = C_2$, $C_{11} = 0$, $C_{10} = C_1$, $M = 1$. So, it can be simplified to the equation (4).

$$W = \sum_{p,q=0}^{N} C_{10}(\bar{I}_1 - 3) + C_{01}(\bar{I}_2 - 3) + \frac{1}{D_1}(J-1)^2 \quad (2)$$

from where C_{10}, C_{01} are material constants, $J = \frac{dV}{dV^\circ}$ is the Jacobian scalar (express the volume change in initial and current configuration of a body), $\bar{I}_1$ and $\bar{I}_2$ as described above, they are expressed as shown in equation (3) and (4) as a function of J and the first two scalar invariants I_1, and I_2 of the left Cauchy-Green deformation tensor $\boldsymbol{b}$

$$\bar{I}_1 = J^{-\frac{2}{3}} I_1 \quad (3)$$

$$\bar{I}_2 = J^{-\frac{2}{3}} I_2 \quad (4)$$

There are many ways that we can identify Mooney-Rivlin constants such a tensile test. Suppose that the elastomer bumper is subjected only to radial stiffness. Using finite element analysis, we can easily identify these parameters based on the experimental data, which contains the nominal stress and the nominal strain of the bushing material.

The Mooney-Rivlin Solid model parameters are shown in the Table 3.

Table 3 Mooney-Rivlin strain energy coefficients

C_{10}	C_{01}	D1
16.84	−8.42	0

2.2. Theoretical modeling

Analytically the leaf spring is modeled as a continuous beam with two supports [6]. Furthermore, the loading is applied symmetrically at both endpoints in a vertical stiffness configuration in figure 5 and anti-symmetrically in the anti-metrical stiffness case. Some hypotheses of beam theories must also be assumed:

- ✓ Euler-Bernoulli Hypothesis: The shape and geometry of cross-sections of a beam do not change in a significant manner under applied transverse loads. This means, that a cross-section can be assumed as a rigid surface during deformation and can only rotate.
- ✓ During deformation, the cross-section of the beam is assumed to remain planar and normal to the deformed axis of the beam.

❖ **Vertical stiffness configuration**

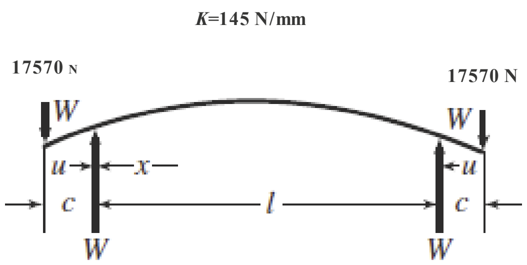

Figure 4 Vertical stiffness leaf spring (continuous beam)

Table 4 Results comparison in reference points

Point of application	Analytical deflection (mm)	Numerical deflection (mm)	Analytical stiffness (N/mm)	Numerical stiffness (N/mm)	Error (%)
RP-1	121 .04	118.938	145	145.02	0.013
RP-2	121 .04	119.04	145	144.89	0.075

To identify the equation of deflection on a leaf spring, we need to start with an equation (5). After integration of the equation and considering the initial conditions, we assume that the inertia is constant. We obtain the deflection on both the constant and the variable form section of the leaf spring, illustrated by equations (6) and (7), respectively. The numerical application value considers the applied force and the desired stiffness. The table 1 presents a summary and comparison of the introduced results of two reference points.

$$\frac{EI\ d^2y}{du^2} = M(x) \tag{5}$$

Deflection on constant section form:

$$\boldsymbol{y} = -\frac{Wcx}{2EI}(l-x) \tag{6}$$

$$A.N \quad y = -17570*\ 282.5\ \frac{800^2}{8*203000*28475}$$

$$= -68.694 \text{ mm}$$

Deflection on variable section form $\boldsymbol{y} = \frac{Wc^2(2c+3l)}{6EI}$ (7)

$$A.N \quad y = 17570*\ (282.5)^2\frac{(2*282.5+3*800)}{6*203000*28476)}$$

$$= 121.04 \text{ mm}$$

The stress acting on the leaf spring is a bending stress which is introduced by the equation below.

$$\sigma = \frac{M}{Z} = \frac{6Wl}{bt^2}$$

$$A.N \quad \sigma = \frac{6*\ 17570.*310}{90*16^2} = 1418.41 \text{ MPa} \tag{8}$$

❖ **Anti-metrical stiffness configuration**

The second configuration is known by its shape in the form of letter S, it occurs when a vehicle is under a specific condition, where the motion of the two gears will be antisymmetric. In this case, the leaf spring fulfills at the same time, two functions one as suspension system and another as a stabilizing bar. Figure 6 is a graphic illustration of the described case; directing attention to the applied static force $F_{ST} + \Delta F$, $F_{ST} - \Delta F$ and the required stiffness to be satisfied.

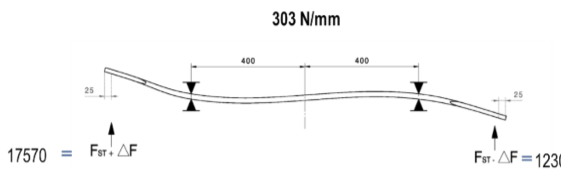

Figure 5 Anti-metrical Leaf spring (continuous beam)

2.3. Numerical modelling

To simulate how leaf springs behave under loading; it is an important to virtually simulate the problems of both configurations, transverse leaf springs which have Vertical Stiffness and Anti-metrical stiffness respectively. The model was carried out using the finite element software, Abaqus [7]. Some important assumptions were taken into account, such that the material is supposed to be isotropic and homogenous. Moreover, the leaf spring is considered to be under static loading. For the boundary condition in the first case, both limits of the surface of the elastomer bumper, were fixed as shown in Figure 7 ($Ux = Uy = Uz = 0\ ; Rx = Ry =$

$Rz = 0$) during the simulation. The applied force on both reference points RP-1 and RP-2, belong to the leaf spring and is equal to 17570 N.

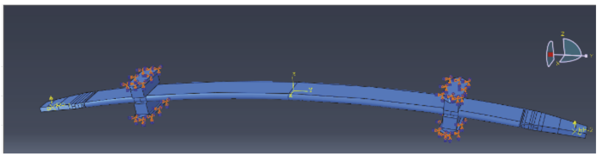

Figure 6 Vertical stiffness case Boundary conditions

The leaf spring is meshed using C3D8I with a solid continuous 3D element, with a linear approximation. However, the elastomer bumper was meshed using C3D8H element as a hyper-elastic material, as it was made of an elastomer material. Figure 8 shows the meshing for both parts.

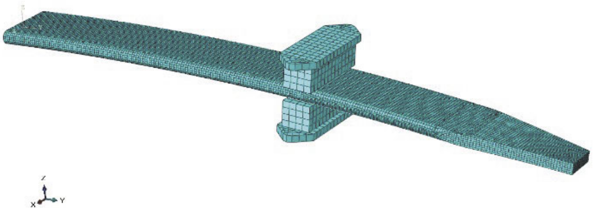

Figure 7 assembly mesh

2.4 Results

To proceed with the numerical results, we need to compare and validate the stiffness curve provided by Iveco, with the ones calculated on RP-1 and RP-2. As a result, a similarity between the two diagrams were found in Figure 9 and 10, and attains confidence on our subsequent results which provides a stiffness K=145 N/mm

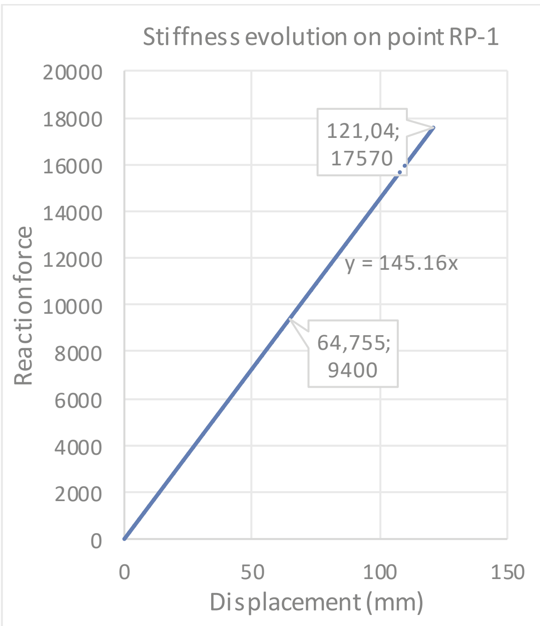

Figure 8 Numerical stiffness

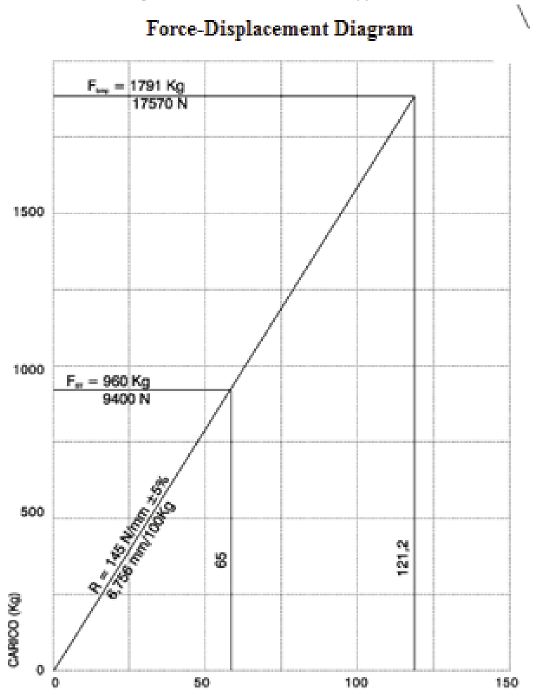

Figure 9 Experimental stiffness

To determine the developed stress, and the maximum deflection under the boundary conditions applied for both sets of configurations, vertical stiffness, and anti-metrical stiffness; results were shown below in Figure 11. As an interpretation for the first, we can say that the structure resists, since the numerical value of the maximum stress is below the limit stress that causes plastic deformation ($\sigma_{max} = 1351\text{MPa} \leq \ \sigma_y = 1467\text{MPa}$)

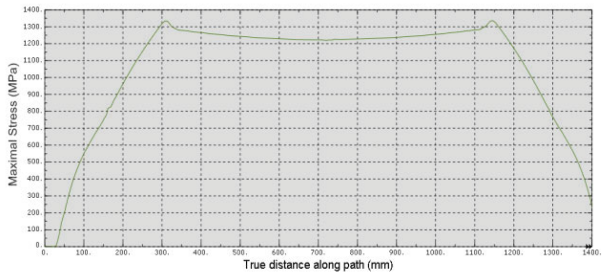

Figure 10 Stress distribution in Vertical stiffness case

The deflection curves provided with numerical analysis is very close to the one which was introduced before, by the equations (2) and (3). Figure 12 is an illustration of both results in the same graph.

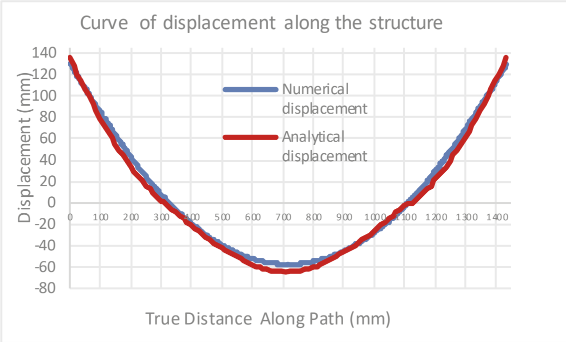

Figure 11 Analytical versus numerical deflection

As an interpretation for the second case, it is obvious that the maximum stress provided by the finite element method is set below the avoidance of the flowable stress, to cause plastic deformation. in other words, $\sigma_{max} = 1387$ MPa $\leq \sigma_y = 1467$Mpa. So, is the structure in this case as well:

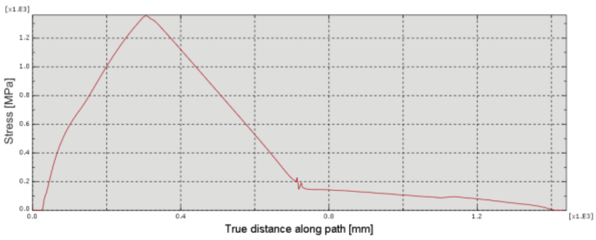

Figure 12 Stress distribution in anti-metrical stiffness case

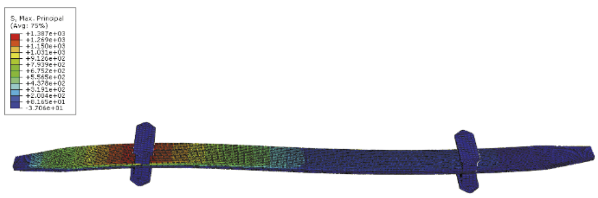

Figure 13 Graphic visualization of stress distribution

3. Conclusion and further research

The numerical analysis of a 3D transverse leaf spring, under static loading, was investigated and a comparison between numerical, experimental, and analytical results were approved. It is worth to also to mention that further studies are needed to consider the dynamic effect during loading. However, to decrease energy consumption, a possibility of fabricating this type of leaf spring using a composite material, such E-glass / Epoxy material has been shown.

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