OPTIMIZATION OF THE HUB FORK OF A CARDAN JOINT

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ABSTRACT: The present research focuses on the theoretical analysis of a cardanic transmission component, namely the hub fork, by means of the analytical calculation. It also approaches the matter from an experimental perspective, simulating the functioning of the part within the cardanic transmission assembly. We used the cardanic transmission of a Dacia vehicle and applied real-value stresses (like in practice).

KEY WORDS: static, cardanic transmission, stresses, optimization.

1 INTRODUCTION

In the technical field, a joint refers to a kinematic coupling, which is the mobile and direct link between two parts, which hinders the relative motion, but allows rotations.

The universal joints of the spherical quadrilateral are called cardan joints (to be more specific, monocardanic) and are asynchronous.

2 ANALYTICAL CALCULATION APPLIED TO THE HUB FORK OF THE CARDAN JOINT OF A DACIA 1307 VEHICLE.

When calculating the torque of the cardanic transmission we take into account the vehicle type and the operating conditions. In case of a vehicle with a single drive axle, the computing moment of the cardanic transmission M_c is determined by considering the point when the engine reaches maximum torque M_M, and the gearbox is in gear 1, and it is expressed by the relation:

$$M_c = M_M \cdot i_{cv_1}, \quad (1)$$

where: M_M=300 N·m, and i_{cv1} is the transmission ratio of the first gear and is below the value 1.

$$M_c = 300 \cdot 1 = 300\, N \cdot m. \quad (2)$$

Force F acts on the cardan fork (it is perpendicular to the fork - figure 1).

There is a dangerous section A-A which is influenced by bending and torsional stresses. The force F which strains each arm of the cardan hub fork is calculated with the relation:

$$F = \frac{M_c}{2 \cdot R} = \frac{300 \cdot 10^3}{2 \cdot 27{,}7} = 5{,}42\ kN, \quad (3)$$

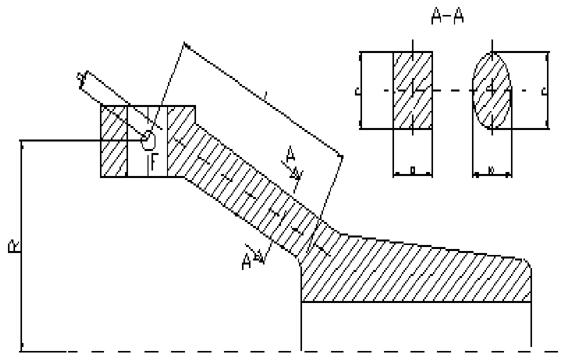

Figure 1. Scheme for calculating the cardan joint fork

where: M_c is the moment of calculation of the cardanic transmission; R – the average radius where the force F acts.

The normal bending stress in section A-A is:

$$\sigma_i = \frac{M_i}{w_i} = \frac{F \cdot l}{w_i} = \frac{5{,}42 \cdot 10^3 \cdot 32}{2312} = 75{,}02\ MPa \quad (4)$$

and for the elliptical section is: $w_i = \frac{b \cdot h^2}{10} = \frac{20 \cdot 34^2}{10} = 2312\, mm^3$.

The arm of the hub fork in section A-A influenced by force F is subject to the torsional stress:

$$\tau_t = \frac{M_t}{w_t} = \frac{5{,}42 \cdot 10^3 \cdot 7}{3360} = 11{,}28\ MPa \quad (5)$$

where $w_t = \frac{\pi b^2 h}{16} \approx 0{,}2 \cdot 20^2 \cdot 42 = 3360\ mm^3$ for the elliptical cross section.

The allowable bending stress of the materials used in the cardan forks is σ_{ai}=100-180 N/mm^2.

The cardan hub forks are made of a medium carbon steel, 0,35-0,45%, or of low alloy improvement steels. After it is quenched and tempered, the hardness of the fork ranges between 197 and 300 HB depending on the type of vehicle.

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3 STATIC ANALYSIS CONDUCTED ON THE CARDAN JOINT OF A DACIA 1307 VEHICLE.

Geometrical modeling can be presently approached by using computer aided design software or modules included in the finite element analysis programs for computer aided design. Cosmos is such a program for the finite element analysis, and it is incorporated into the Solidworks program, which, by the accuracy of the results, is often used in studying the static and dynamic behavior of the components of technological systems. In fact, Solidworks is used for developing the geometrical model of the cardanic transmissions components and for assembling them, and Cosmos, based on the geometry taken from Solidworks, helps generate the finite elements network, perfects the contacts between the component parts and allows the application of strains and stresses. The Cosmos program has modules dedicated to specific areas, such as: analysis of structures in general, fluid mechanics, thermal analysis. Each module, in its turn, possesses a complete set of analyses for linear or nonlinear, static or dynamic issues, which lead to the development of a complete research.

This program also contains a structural module which, due to its broad possibilities, has been chosen for the analysis of the static and dynamic behavior of the cardanic transmission assemblies. For the present research, we chose the cardanic transmission of the Dacia 1307, manufactured in Romania, later on transferring the conducted researches on other different size vehicles of the same class.

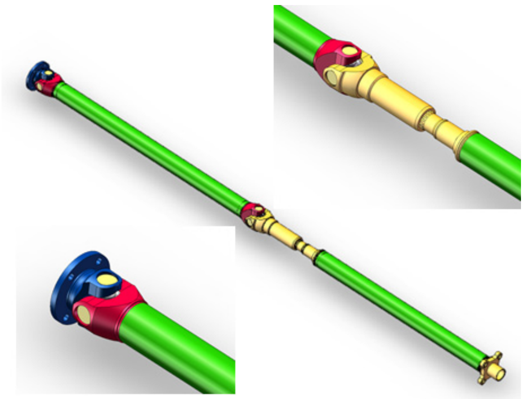


Figure 2. The cardanic transmission modeled in Solidworks (general view)

As mentioned before, the geometrical modeling was done by using Solidworks. We chose this solution because of the problems of transferring a model saved under IGS format from another computer aided program in the Cosmos program.

The cardanic transmission assembly of Dacia 1307 was modeled on the following components: the flange from the end of the drive, the Φ 630 tube, the splined arbor, the splined hub, the cardan cross 1, the hub fork 1, the Φ 730 tube, the hub fork 2, the cardan cross 2 and the flanged fork towards the differentiator and then the entire model was assembled. We also modeled the bearings mounted on the ends of the cardan crosses. Figure 2 presents the model of the cardanic transmission in closed loop variant.

We eliminated the insignificant details such as small fillet radii or niches and approximated the inhomogeneous areas on the structure with homogenous finite elements to obtain a model whose behavior will be as similar to the real one as possible and to get the shortest functioning time.

The purpose of the static analysis is to determine the strains and stresses that act on the model in a static condition. This analysis has been conducted on the parameterized model of the cardanic transmission assembly of a Dacia 1307 vehicle. We studied the static behavior of the cardanic transmission components and we compared the results with those of the finite elements analysis.

We analyzed the end towards the differential by applying strains on the flanged fork, annulling the freedom of movement, therefore fixing this end. At the end towards the drive, we apply a constant moment of 300 Nm. Figure 3 shows the position of the strains and stresses.

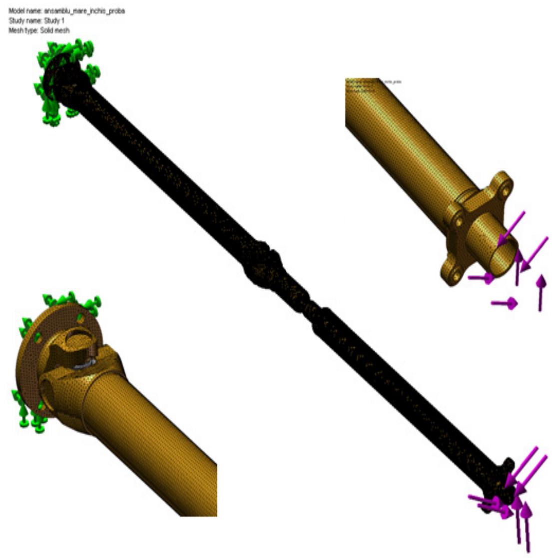


Figure 3. Meshing and application of stress and strain on the cardanic transmission (general view)

The cardanic transmission components are: 1- flanged fork, STAS 880 – 80; 2- hub fork, STAS 880 – 80; 3- hollow shaft encoder, STR 302-88; 4-

cardan cross, 18 Mn Cr 10 – STAS 791 – 88; 5- safety ring, 6- needle roller bearing, 7- boss fork, OLC 45, STAS 880 – 80; 8- intermediate shaft, 40 Cr 10 – STAS 791 – 88; 9- hollow shaft encoder, 10- flange, STAS 880 - 80.

The variation graphs of the von Mises stress and of the main strains of the cardanic transmission assembly are shown in figures 4 and 5.

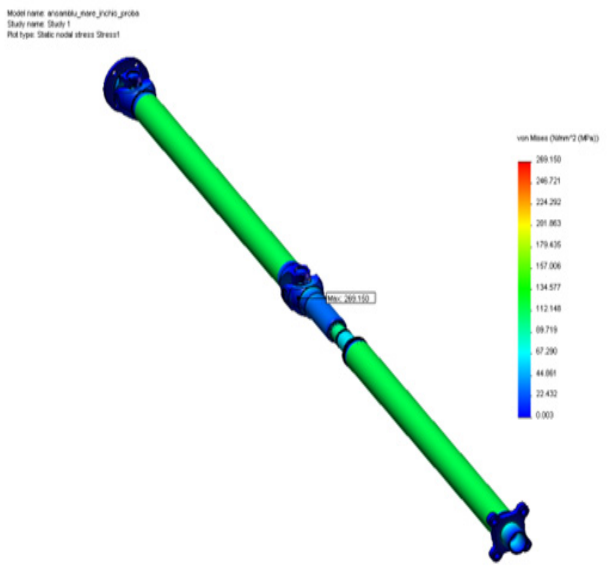


Figure 4. Variation graph of von Mises stress (σ_{VM}=269,15 MPa)

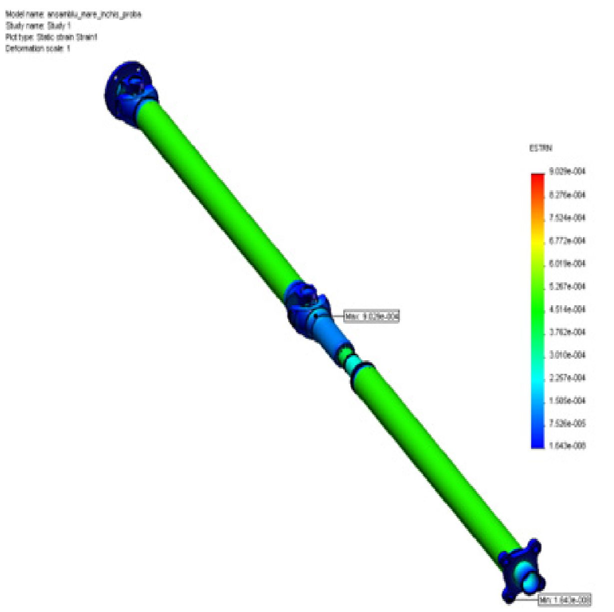


Figure5. Variation graph of the unit strain (ε_{VM}=9,03·10^{-4})

The analysis of the graphs showing the variation of the von Mises stress for the cardanic transmission assembly shows that its maximum value is 77.85 MPa, on the hub fork. The von Mises stress in the other areas of the hub fork is smaller than the material's allowable value (180 MPa).

Figures 6 and 7 show the variation graphs of the von Mises stress and the unit strain.

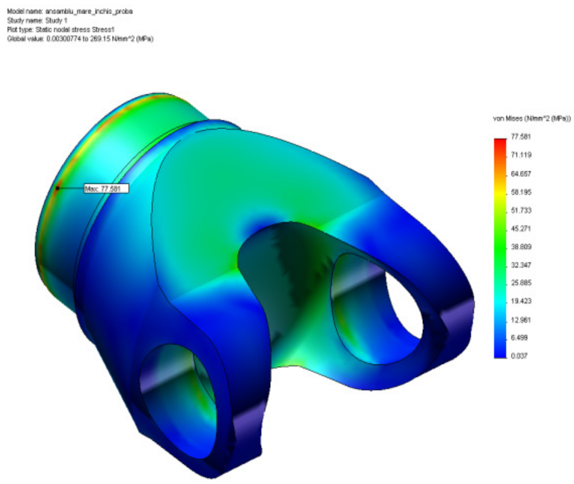


Figure 6. Variation graph of von Mises stress (σ_{VM}=77,58 MPa)

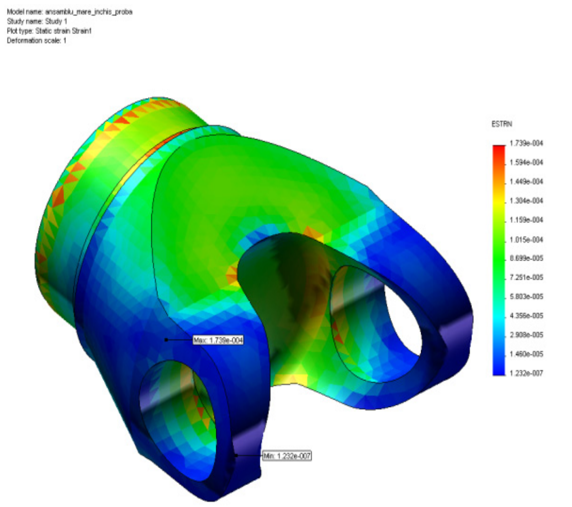
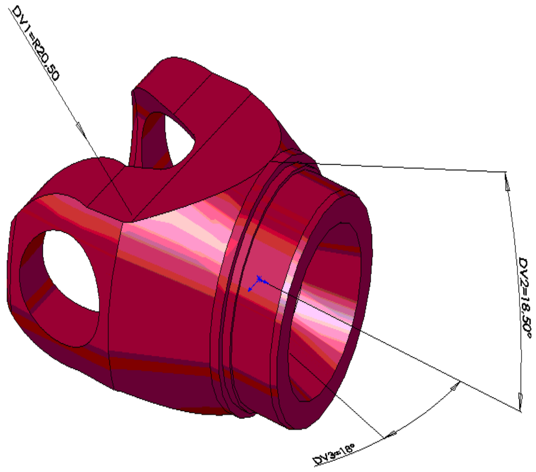


Figure 7. Variation graph of unit strain (ε_{VM}=1,739·10^{-4})

4 OPTIMIZATION OF THE HUB FORK

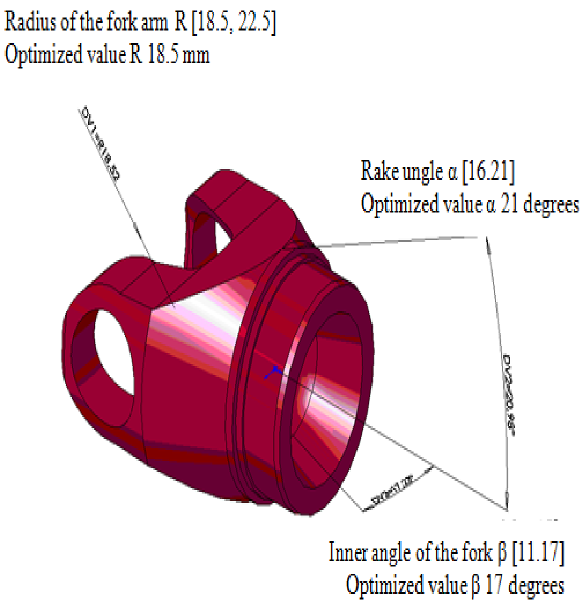
The optimization of the hub fork in figure 9 is performed so as its volume is minimum.

We took into consideration the radius of the hub fork arm, the rake angle of the cardan fork and the inner angle of the hub fork.



a.

Figure 9.a. The part undergoing structural optimization analysis (a.) and the result of the optimization (b.)



b.

Figure 9.b. The part undergoing structural optimization analysis (a.) and the result of the optimization (b.)

The von Mises stresses in the hub fork were analyzed at the following intervals: **the radius of the fork arm DV1 between 18.5 mm and 22.5 mm, the rake angle of the cardan fork DV2 between 16 and 21 mm** and **the inner angle of the fork between 11^0 and 17^0**, respectively: *SIGMA* ≤ 260 MPa.

This led to the dependency graphs shown in Figures 10 to 17.

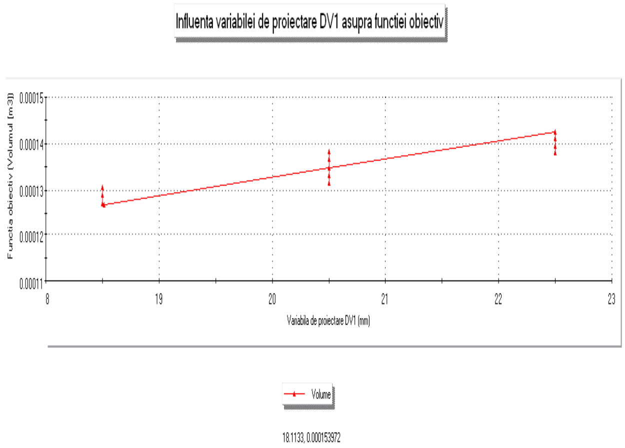


Figure 10. Influence of the hub fork arm radius on the volume

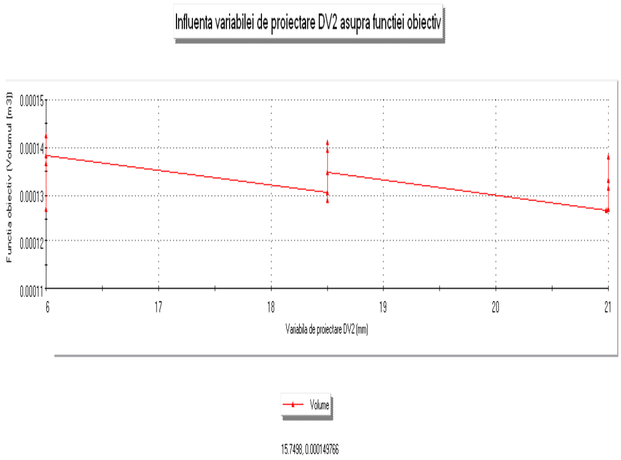


Figure 11. Influence of the rake angle of the cardan fork on the volume

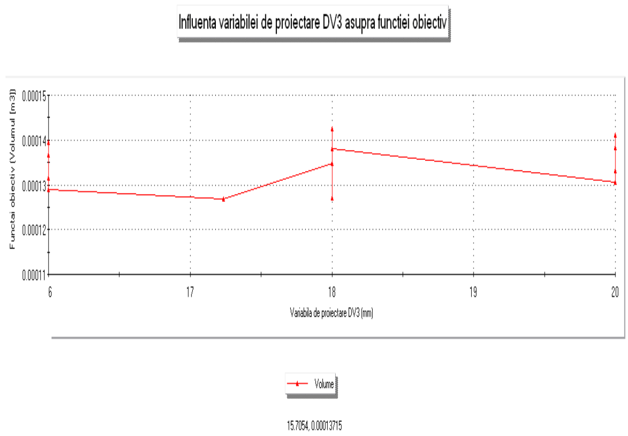


Figure 12. Influence of the inner angle of the hub fork on the volume

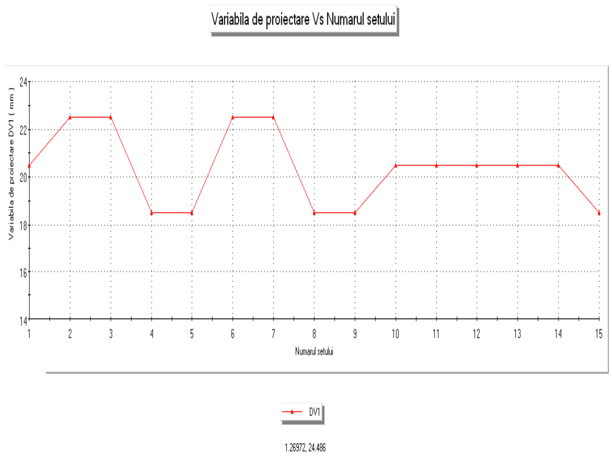


Figure 13. Variation of the hub fork arm radius during optimization

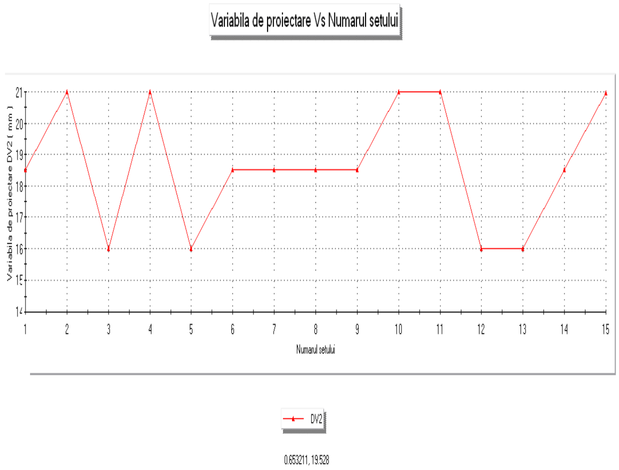


Figure 14. Variation of the rake angle of the cardan fork during optimization

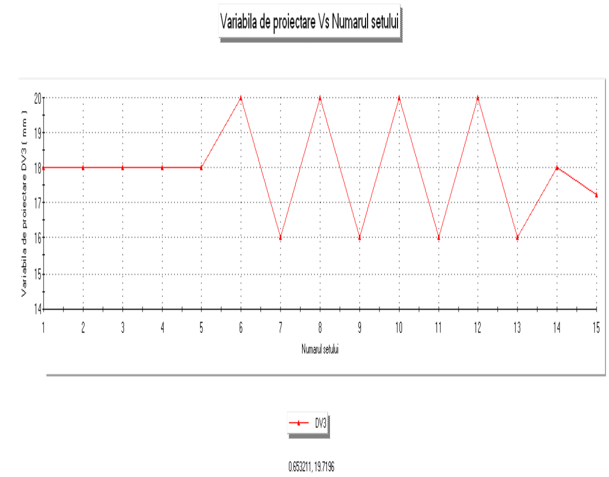


Figure 15. Variation of the inner angle of the hub fork during optimization

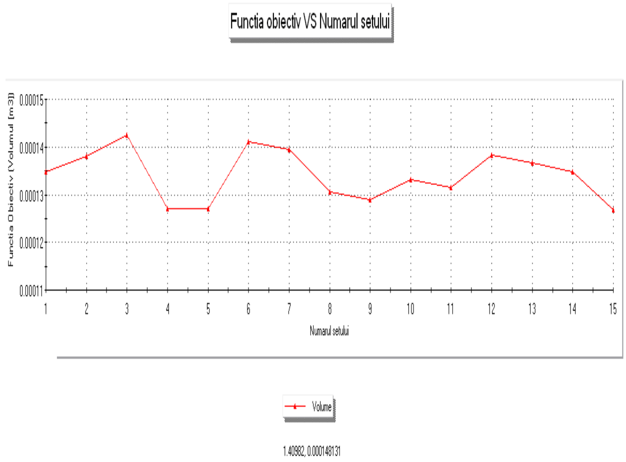


Figure 16. Variation of the hub fork volume during optimization

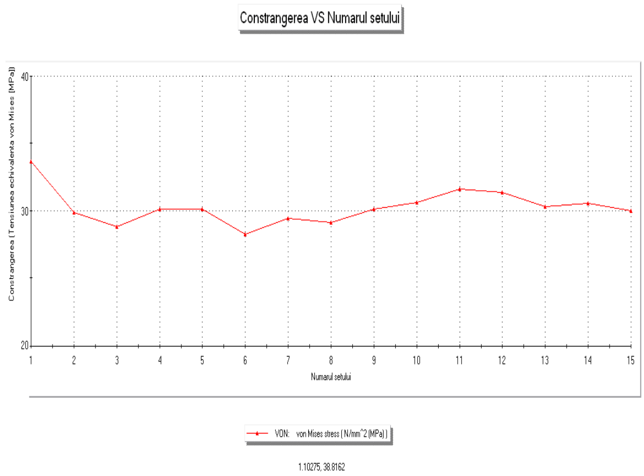


Figure 17. Variation of the von Mises stress in the hub fork during optimization

By analyzing these graphs we reach the conclusion that the parameters *thickness g, angle* and *radius r,* to be in accordance with the values set for the specified stress and the minimum volume, will be during the optimization: **the hub fork arm radius DV1: 18.5 mm, the rake angle of the cardan fork DV2: 21^0** and **the inner angle of the hub fork: 17^0**.

This model will be tested for resistance by means of the file of the optimization analysis, where the input data will be the results of the optimization. The same results are obtained, observing that the maximum value of the von Mises stress is smaller by about 8% than the one of the pre-optimization model.

5 CONCLUSIONS

It can be noted by observing the graphs of the variation of the strains in the two directions that first, it is preserved the linearity of the experimental results and, second, that the results are very much similar to those of the finite element simulation. Therefore, the worst error percentage will be about 9% compared to the numerical simulations.

After applying a torque of 300 Nm we experimentally obtained a von Mises stress with the value of 79.64 MPa, and that obtained with the finite element method in the same area of study has the value of 77.58 MPa, whereas the analytical calculation led to a result of 75.02 MPa. The obtained stresses are below the limit of the allowable values in normal functioning conditions, and it remains to be studied what happens in case of a choke start.

By applying a torque of 300 Nm we experimentally obtained the unit strain of 0.0066 mm, whereas by using the finite element method in the same area of study we obtained the unit strain of 0.0073 mm.

The model we discussed in this paper may be given as an example for other cardanic transmission components as well. We shall consider this aspect in the future.

- We presented a number of general notions of the cardanic transmissions;
- We chose the cardanic transmission of a Dacia 1304/1307 vehicle to demonstrate our theoretical and experimental research on a real model;
- By optimizing a cardanic transmission component, in this case the hub fork, we develop optimal parts, with relatively small differences in what the newly obtained stresses are concerned (8%).
- The hub fork of the cardanic transmission was studied with the finite element method, and the variation graphs of the deformations on the two directions show that the experimental results are linear, and that the results are very similar to the results obtained from the finite element method simulation. As a conclusion, the worst error percentage is about 7% as compared to the numerical simulations. In order to show the results obtained by the finite element method, we chose the rapid prototyping method being thus able to create the optimal full scale parts in the quickest time possible.

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