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Study of the Cardan Cross Using the Experimental and Analytical Method

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Abstract

The work presents the approach of the finite element investigation of the cardanic transmission joint. We treat the problem of structural behaviour, namely analyses that target the strength-related behaviour. It is thus sought to determine certain parameters (nodal displacements, stresses, strains) in the conditions of applying various types of loads - forces, pressures or moments. For this, we used a finite element analysis software – Cosmos - a product integrated in the SolidWorks package.

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Keywords: cardan cross; compression; mechanical characteristics

1. Introduction

The cardanic transmissions of cars and of various industrial machines are part of the kinematic chain of transmitting the rotary motion from an engine to the drive wheels or to the moving subassemblies. The resistance calculus of the cardan transmission is done for the main elements, which are also the most stressed ones: the cardan shaft, the cardan joints and the needle roller bearings, depending on the loading and functioning regimes.

Following the market study, there has been noticed that within a cardanic transmission, the most loaded elements are the two cardan crosses, so the study presented here will focus on this type of element. The cardanic transmission of a Dacia 1307 vehicle [2,9] was chosen for this research.

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Fig. 1. Types of cardan transmissions made by the Eurocardan Company [9].

The classification of the cardan joints is the same as that of cardanic transmissions and therefore the mechanisms (spatial) which serve to transmit the rotary motion between two concurrent shafts, having a usually variable angle between axes and whose transmission ratio is a periodic quantity, with a mean value of one, are called **asynchronous universal joints**. These can be obtained by successive kinematic, constructive and structural transformations of the spherical quadrilateral. In the technical literature, all the constructive versions of the spherical quadrilateral mechanism with quadratic mobile sides (the corresponding center angles are right) and the hyper-quadrant base (the corresponding center angle is obtuse) are called **cardan joints**. Therefore, all the universal joints deriving from the spherical quadrilateral are called cardan joints (more precisely, monocardanic) and these joints are asynchronous.

Following the conducted market research on the types of faults and their frequency, it has been observed that most failures were detected in the bearings (41.38%), the cross (31.03%) and less in the fork and the hub (table 1). [7]

Table 1. Detecting defects of the transmission main components [7].

	all	most	few	very few	none	unanswered	Total
bearings	0	12	3	3	5	6	29
	0	41,38	10,34	10,34	17,24	20,69	100%
cardan cross	0	9	6	1	7	6	29
	0	31,03	20,69	3,45	24,14	20,69	100%
fork	0	3	5	5	10	6	29
	0	10,34	17,24	17,24	34,48	20,69	100%
hub	0	3	4	6	10	6	29
	0	10,34	13,79	20,69	34,48	20,69	100%
other flaws	1	0	1	6	15	6	29
	3,45	0	3,45	20,69	51,72	20,69	100%

In this context it is recommended to study the causes of these failures and find solutions to solve them, either by redesigning these areas or by choosing other materials or redesigning processes. In conclusion, we focus our research on the cardan cross. [7]

2. Calculation of the cardan cross [4, 5]

The cardan cross is subjected to bending, shearing and crushing stresses by the force F_1 (fig. 2).

The force F_1 is calculated with the relationship:

$$F_1 = \frac{F}{\cos\gamma} = \frac{5.42}{\cos 20} = 5.76 \text{ kN}, \tag{1}$$

where: γ is the angle between the shafts' axes.

$$F = \frac{M_c}{2 \cdot R} = \frac{300 \cdot 10^3}{2 \cdot 27.7} = 5.42 \text{ kN}. \tag{2}$$

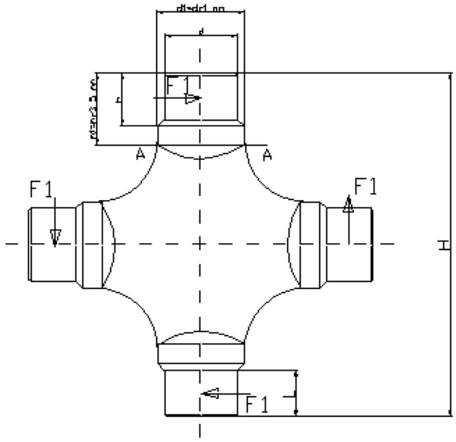

Fig. 2. Calculus scheme for the cardan cross.

The normal bending stress in cross-section A-A is calculated with the relation (3):

$$\sigma_i = \frac{M_i}{W_i} = \frac{F_1 \cdot \left(h_1 - \frac{L}{2}\right)}{0.1 \cdot d_1^3}, \frac{15.37 \cdot 10^3 \cdot \left(15 - \frac{10.5}{2}\right)}{0.1 \cdot 18.65^3} = 231 \text{ MPa} \le 750\text{MPa}. \tag{3}$$

The shearing load at the spindle's base is determined using the relation:

$$\tau_f = \frac{4F'}{3 \cdot A} = \frac{4 \cdot 7.19 \cdot 10^3}{3 \cdot 262.32} = 36.57 \text{ MPa}, \tag{4}$$

where: the force F’ is calculated with (5):

$$F' = \frac{M_c}{2(R - 0{,}5h) \cdot \cos\gamma} = \frac{300 \cdot 10^3}{2(27{,}7 - 0{,}5 \cdot 11) \cdot \cos 20} = 7{,}19\ kN \tag{5}$$

It is recommended that τ_{af} = 80-120 N/mm^2.

The verification at crushing is done by determining the specific pressure on the cross spindle, when subjected to the force F_1, with the relation:

$$\sigma_s = \frac{F_1}{d \cdot h} = \frac{5.76 \cdot 10^3}{18.28 \cdot 11.08} = 28.44 \text{ MPa} \leq 100\text{MPa}. \quad (6)$$

3. Study of the cardan cross using the finite element method

The software employed for the analysis using the finite element method is Cosmos, a product incorporated into the SolidWorks software package. The following figures show the variation graphs for the von Mises stress (fig. 3) and the variation graphs for the unit strain (fig. 4).

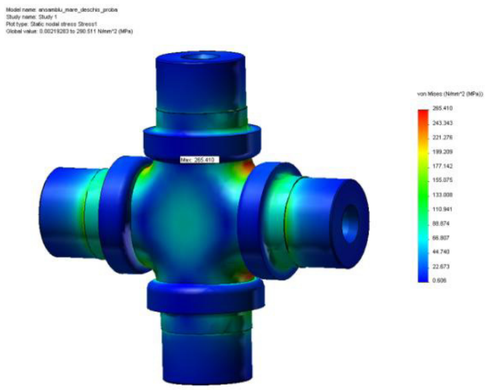

Fig. 3. Variation graphs for the von Mises stress – cardan cross (σ_{VM}=255.11 MPa).

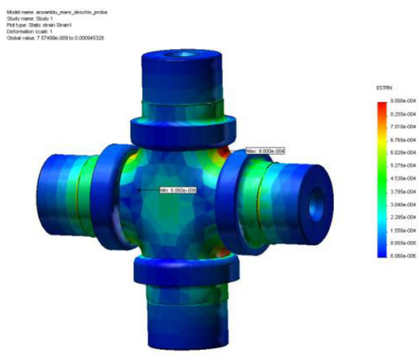

Fig. 4. Variation graphs for the unit strain - cardan cross (ε_{VM}=$9 \cdot 10^{-4}$).

4. Checking the specific strains through Electric Resistive Tensometry

The investigation method used for the experimental research was the electric resistive tensiometry method. The tests were carried out in the laboratories of the Faculty of Engineering of Sibiu, Romania, using a testing set installed on the tensile, compression and buckling testing machine INSTRON 4303 (fig. 5). The elaborated experimental research methodology aimed at validating the theoretical results obtained using the numerical analysis with FEM, both for the results related to the static analysis and for the dynamic one. The testing set used for unfolding the experimental researches through the tensometry method consists of the testing device, the force input system, measurement transducers and the data acquisition system (fig. 5). [7]

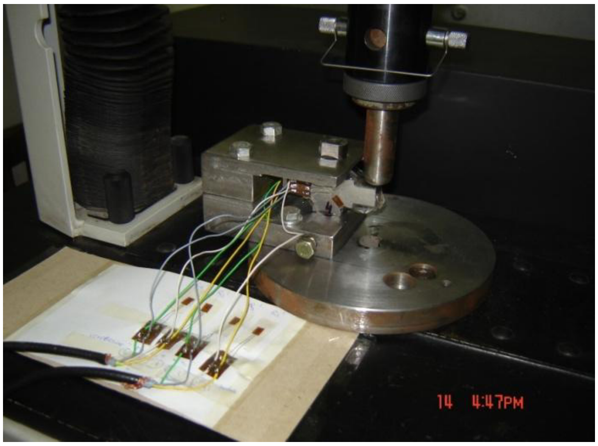

Fig. 5. Detail of the testing set installed on the INSTRON machine.

The data acquisition system consists of four modules: the transducers installed on the analysed directions of the cardan cross, the signal conditioning module (MB-38, Keithley Instruments Inc.), the analog–digital converter board (KPCI 3108, Keithley Instruments Inc.) and a software package controlling the acquisition system and processing the collected data. The experimental determination of the unit strain ε, on the two directions was done using the relation [8]:

$$\varepsilon = \frac{-4V_r}{GF[(1+\upsilon)-2V_r(\upsilon-1)]}\cdot\left(1+\frac{R_L}{R_g}\right); \tag{7}$$

The signal acquired from the analog-digital converter board KPCI 3108 is processed, filtered and saved by means of two virtual instruments created by the author in the TestPoint software package that accompanies the data acquisition device and is dedicated for data acquisitions (fig.6). The instruments contain blocks offered by the software, that allow the modification of the number of channels on which the acquisition is made, of the acquisition rate, of the total duration of the acquisition, as well as the filtering of data for eliminating the „noise" inherent to any acquisition and the saving of data as text files.

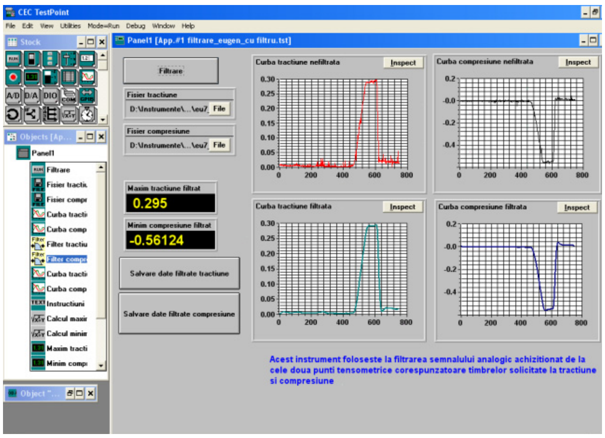

Fig. 6. The virtual tool used in the acquisition of unit strains on two directions and the filtering of the electric signal.

Conclusions

The data resulting from the market research made us focus our research on the part which seems to fail most often.

We have presented a computed model of a cardanic transmission applied on a Dacia vehicle in order to obtain stresses in calculations (231 MPa).

After the finite element analysis, the resulting stress was 255 MPa, close to that resulting from the calculation. The variation graphs of the strains on the two directions show that, on the one hand, the experimental results respect the linearity and, on the other hand, there is a good concordance with the results of the simulations using the finite elements method. In the most unfavourable case, the error percentage compared to the numerical simulations is around 9 %.

The unit strain experimentally obtained following the application of a torque of 300 Nm is 0.0066 mm, while the one obtained with FEM in the same field of study is of 0.0073 mm.

The present research analyses the cardan cross from an analytical and experimental point of view, aiming to verify the stresses and strains in regular usage (300 Nm).

A dynamic analysis will follow, as well as a study on the increase of the random strain (800 Nm) in order to observe the resulting phenomena.

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