RESEARCH PAPER | Engineering | Volume : 4 | Issue : 11 | November 2014 | ISSN - 2249-555X

Theoretical and Experimental Studies on the Angular Deformation of The Rigid Cardanic Transmission

KEYWORDS: angular deformation, experimental stand, finite elements

Eugen Avrigean

Lucian Blaga University, Engineering Faculty, Emil Cioran Street, Sibiu, 550025

ABSTRACT *The present study refers to the analysis of the cardanic transmission of motor vehicles using analytical calculations strictly applied on this in order to determine the maximum deformation compared to the allowable limit. The cardanic transmission is then analyzed with finite elements with the help of the Solidworks and Cosmos programs, thus learning some new information, which, together with the experimental data, allow us to grasp the operating behavior of the cardanic transmission. The experimental stand we used permits the charging with allowable value loads as given in the literature in the field. The study takes into account the three aspects: analytical, finite elements and experimental, in order to correct the obtained values.*

INTRODUCTION

The cardanic transmissions of the motor vehicles and various industrial machines form the kinematic chain of transmitting the rotary motion from an engine to the drive wheels or the moving parts. [2].

The cardanic transmission refers to a set of machine parts (joints, shafts, intermediate bearings etc.) used for the remote transmission of the mechanical energy by rotation without increasing the torque gain between units, having a variable or invariable position in space. By judicious design of these machine parts and of the execution technology, it is ensured the increase of the operational reliability and a low metal usage[2].

ANALYTICAL CALCULATION OF THE CARDANIC TRANSMISSION

The calculation of the resistance of a cardanic transmission is performed for the main and the most used elements: the cardan shaft, the cardan joints and needle bearings, depending on the charging and operating regimes. [1]

1. Calculating the Torque Moment [2]

The torque of the cardanic transmission is calculated according to the vehicle type and the operating conditions, and for an one axle vehicle where the time of calculation of the cardanic transmission M_c is determined by considering the situation when the engine reaches the maximum torque M_M, and the gearbox is in stage 1, and it is resulted from the ratio:

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

where: M_M=300 N·m, and i_{cv1} is the transmission ratio of the first stage of the gearbox and it has the value 1.

$$M_c = 300 \cdot 1 = 300\, N \cdot m$$

2. Calculating the Shafts

- Calculating the Shaft in Torsion

The tensions due to torsion are determined in the following way:

a. For the pipe connecting these components: *the intermediate shaft* and *the flange* the tension can be determined with the following ratio

$$\tau_t = \frac{M_c}{w_t} = \frac{16 \cdot D \cdot M_c}{\pi \cdot (D^4 - d^4)} = \frac{16 \cdot 57 \cdot 300 \cdot 10^3}{\pi \cdot (57^4 - 53^4)} = 32{,}69\ MPa \quad (2)$$

b. The calculation of the torque for the intermediate shaft:

$$\tau_t = \frac{M_c}{w_t} = \frac{16 \cdot M_c}{\pi \cdot D^3} = \frac{16 \cdot 300 \cdot 10^3}{\pi \cdot 30^3} = 56{,}62\ MPa \quad (3)$$

c. The calculation of the torque for the hub fork:

$$\tau_t = \frac{M_c}{w_t} = \frac{16 \cdot D \cdot M_c}{\pi \cdot (D^4 - d^4)} = \frac{16 \cdot 45 \cdot 300 \cdot 10^3}{\pi \cdot (45^4 - 35^4)} = 26{,}46 MPa \quad (4)$$

The characteristics of the steels used in building the cardan shafts by the GWB Company were taken from the literature in the field.

-Checking the Torsional Deformation

a. The torsion specific to the first pipe which connects the *intermediate shaft* to *the flange* is calculated with the ratio:

$$\varphi_{\circ} = \frac{c_d \cdot M_c \cdot L}{G \cdot I_o} \cdot \frac{180}{\pi} = \frac{2 \cdot 300 \cdot 10^3 \cdot 760}{81 \cdot 10^3 \cdot 26{,}15 \cdot 10^4} \cdot \frac{180}{\pi} = 1{,}24^0 \quad (5)$$

where M_c is the time calculation – 300 N·m ; L – the length of the cardan shaft - 760 mm; c_d- the dynamic coefficient (c_d=2 – for motor vehicles); G- the transverse elastic modulus; I_p- the polar moment of inertia of the shaft;

b. The torsion specific to the second pipe which connects *the hub fork* to *the hub fork* is calculated with the ratio:

$$\varphi_{\circ} = \frac{c_d \cdot M_c \cdot L}{G \cdot I_p} \cdot \frac{180}{\pi} = \frac{3 \cdot 300 \cdot 10^3 \cdot 630}{81 \cdot 10^3 \cdot 26{,}15 \cdot 10^4} \cdot \frac{180}{\pi} = 1{,}02^0; \quad (6)$$

c. The torsion specific to *the intermediate shaft* is calculated with the ratio:

$$\varphi_{\circ} = \frac{c_d \cdot M_c \cdot L}{G \cdot I_p} \cdot \frac{180}{\pi} = \frac{3 \cdot 300 \cdot 10^3 \cdot 98}{81 \cdot 10^3 \cdot 25{,}51 \cdot 10^4} \cdot \frac{180}{\pi} = 0{,}24^0; \quad (7)$$

d. The torsion specific to *the hub fork* is calculated with the ratio:

$$\varphi_{\circ} = \frac{c_d \cdot M_c \cdot L}{G \cdot I_p} \cdot \frac{180}{\pi} = \frac{3 \cdot 300 \cdot 10^3 \cdot 40}{81 \cdot 10^3 \cdot 260 \cdot 10^4} \cdot \frac{180}{\pi} = 0{,}01^0. \quad (8)$$

The value of the torque for the considered transmission is 300 Nm, which is the normal one while functioning, and the maximum allowable before breaking is 800 Nm. The value calculated for 300 Nm is 2,51^0. The maximum allowable torsional deformation is φ=7-8^0.[3]

THE FINITE ELEMENTS METHOD STUDY OF THE CARDANIC TRANSMISSION [4]

The CAE Module (Computer Aided Engineering) has been in-

Volume : 4 | Issue : 11 | November 2014 | ISSN - 2249-555X

troduced in the makeup of CIM (Computer Integrated Manufacturing) after the appearance of the CAD module (Computer Aided Design). It actually appeared at the same time with the finite elements method. The method was initially used to mechanically calculate the aircraft structures but it has later expanded much at all the material continuum problems. These problems are intended to determine, in a researched field, the values of one or more unknown functions such as: displacements, speeds, temperatures, tensions, specific deformations, etc., depending on the nature of the considered problem. In the chapter dealing with the finite element numerical simulations of the behavior of the cardanic transmissions, we will refer only to those problems approaching the structural behavior, i.e. analyses on the behavior in terms of resistance. These tests are intended to determine certain measurements (nodal displacements, stresses, deformations) under the conditions of applying different types of charges. The charges that may apply are forces, pressures and moments.

DETERMINING THE ANGULAR DEFORMATION BY APLLYING VARIABLE LOAD

The problem of the geometric modeling can be currently approached by using CAD software packages or modules incorporated into the finite element analysis programs for aided design. Such a program of finite element method analysis is the Cosmos program, a product incorporated in the Solidworks program, which, through the convenience and accuracy of the results it generates, is often used in researching the static and dynamic behavior of the components of the technological systems. Actually, the Solidworks program is used to geometrically model the components of the cardanic transmission and to assemble them, and the Cosmos program, based on the geometry it took from Solidworks, generates the finite elements network, improves the contacts between the components and allows the application of strains and charges.

For approaching the present research, we chose the cardanic transmission of the Romanian manufactured Dacia vehicle, being able to apply easily the conducted research to other types of vehicles of the same class.

The geometric modeling was done, as we mentioned before, by using the Solidworks program. This solution was chosen because of the problems that arise when transferring a model saved in IGS format from another CAD software program into the Cosmos.

The geometric model used for the static analysis was developed in two versions: one with open loop transmission and one with closed loop transmission. The cardanic transmission assembly of the Dacia vehicle was modeled on the components, namely: the flange from the end of the drive, the Φ630 pipe, the splined arbor, the spline hub, the first cardanic cross, the first neck fork, the Φ730 pipe, the second neck fork, the second cardanic cross and the flanged fork towards the differentiator and then the entire model was assembled. The bearings which are mounted on the ends of the cardanic crosses were also modeled. Figure 1 present the model of the transmission in the closed version.

In order to obtain a model which would behave in a manner as close to reality as possible and trying to reach the shortest functioning time, the insignificant details were eliminated (small ray connections or niches) and the inhomogeneous areas from the structure were approximated with homogenous finite elements.

Thus we obtained a unitary network of finite elements, which is shown in the sequent figure 1 (the closed loop version).

Figure 1. The cardanic transmission assembly modeled with Solidworks (general view)

The static analysis aims to determine the state of tensions and deformations at charging the model in static circumstances. This analysis was applied to the parameterized model of the cardanic transmission in a Dacia 1307 pick-up, in the two versions of construction: with a closed loop transmission and with an open loop transmission. The static behavior was studied in both cases, in order to compare the results of the finite elements analysis of the structural components of the cardanic transmission.

Figure 2. The meshed cardanic transmission model (general view)

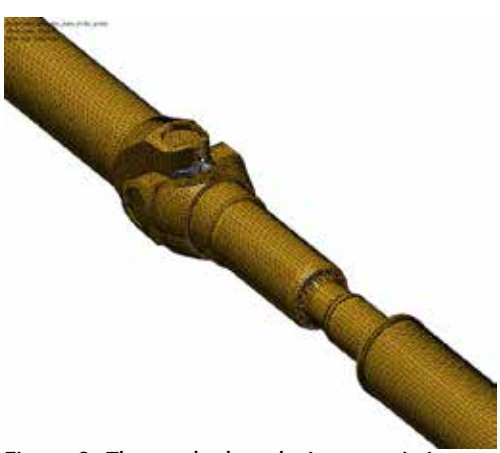

Figure 3. The meshed cardanic transmission model (detail)

Volume : 4 | Issue : 11 | November 2014 | ISSN - 2249-555X

In order to carry out the static analysis on the end towards the differential, we apply strains on the flanged fork, restraining all the degrees of freedom so that this end is fixed. At the end located on the drive, we apply a constant moment of 300 Nm. The position of the strains and charges can be seen in figure 4. The materials of the cardanic transmission components are: 1-flanged fork, STAS 880 – 80; 2- neck fork, STAS 880 – 80; 3- hollow shaft encoder, STR 302-88; 4- cardanic cross, 18 Mn Cr 10 – STAS 791 – 88; 5- safety ring, 6- needle roller bearings, 7- hub fork, OLC 45, STAS 880 – 80; 8- intermediate shaft, 40 Cr 10 – STAS 791 – 88; 9- hollow shaft encoder, 10- flange, STAS 880 - 80.

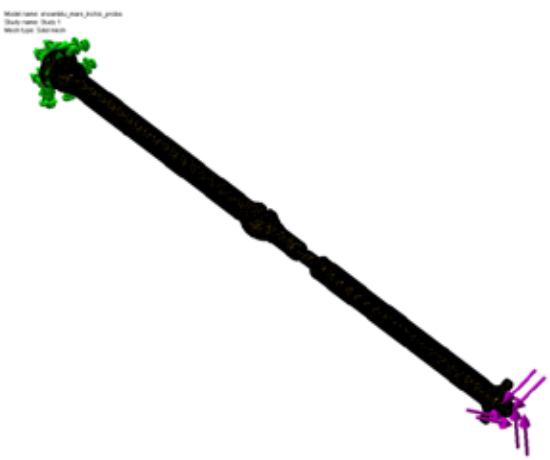

Figure 4. The application of charges and strains on the cardanic transmission (general view)

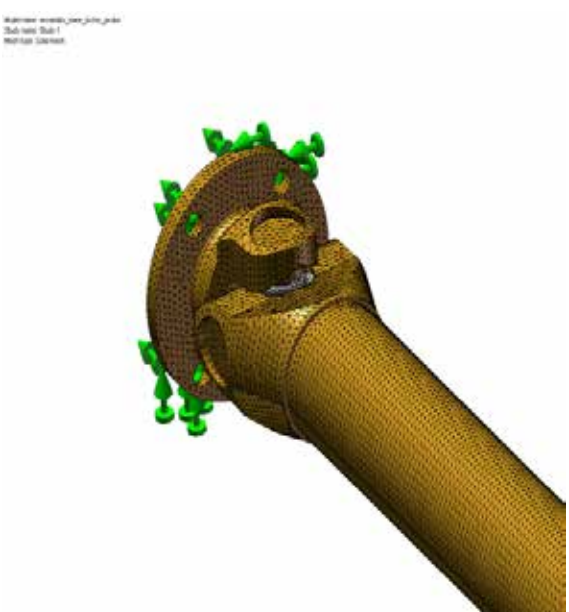

Figure 5. The application of strains on the flanged fork of the cardanic transmission (detail)

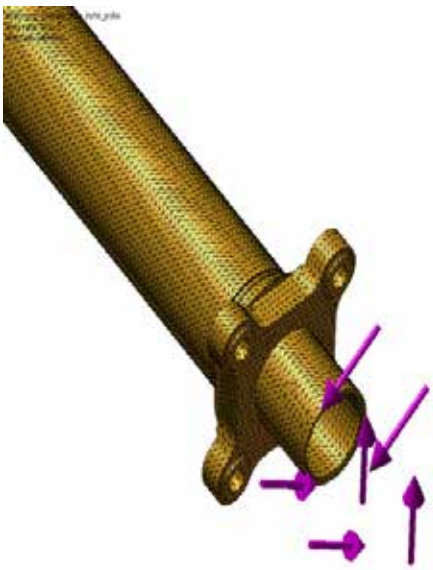

Figure 6. The application of charges on the neck fork of the cardanic transmission (detail)

We established that for a charge from 0 to the maximum allowable value (800 Nm) the maximum deformation is 7,4 degrees.

CHARGING THE CARDANIC TRANSMISSION WITH TORQUES SIMILAR TO THOSE IN PRACTICE

The experimental model contains the following main subassemblies:

1. the full cardanic transmission from where the flexible coupling was removed in order to allow the measurement of only the angular deformation of the transmission;
2. the basic framework, made of a rigid profile, fixed to the workbench;

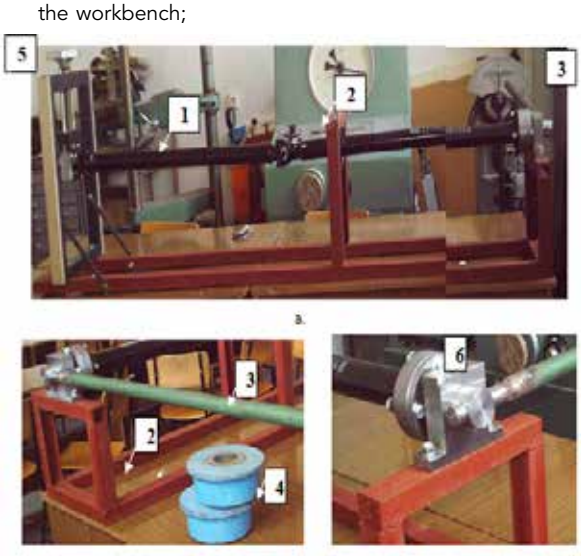

Figure 7. Experimental Stand

3. the support rod, where the weights will be mounted, measured according to table 1;
4. the weights needed for achieving the torque;
5. the device for angular variation of the cardanic transmission;
6. the angular deformation reading system.

Volume : 4 | Issue : 11 | November 2014 | ISSN - 2249-555X

The test consists in applying to one end of the cardanic transmission of a bending moment (in the form of a weight at a distance of 1.2 m) starting from a minimum value (100 Nm) up to the maximum (800 Nm) that the cardanic transmission can tolerate. The other end of the cardanic transmission is bolted to the support of the stand.

The values of the angular deformationobtained from the three conducted studies (A. analytical, B. using finite elements FEM, and C. experimental) are presented in the following table (table 1):

TABLE – 1
The Method of Charging the Cardanic Transmission and the Obtained Deformations

Nr. crt.	Charge Value [Nm]	Analytical Angular Deformation [degrees]	MEF Angular Deformation [degrees]	Experimental Angular Deformation [degrees]
	100	0,84	0,89	0,93
	200	1,67	1,78	1,37
	300	2,51	2,67	2,66
	400	3,35	3,56	3,33
	500	4,18	4,45	3,97
	600	5,02	5,34	4,63
	700	5,86	6,23	5,20
	800	6,69	7,14	5,67

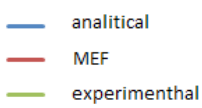

Figure 8. Variation of the specific deformation according to the charge

Based on these results we developed the variation graphs for the observed results in order to highlight the difference between the three examined aspects.

It can be seen that the three methods of analysis are similar and all fall below the maximum allowable deformation of 7.8 degrees stipulated in the literature.

CONCLUSIONS

In this paper we presented some general notions about the functioning of the cardanic transmissions, then we presented the calculation method applied to a physical cardanic transmission, according to the literature. This calculation was made based on the mathematical rations which apply to the components of the cardanic transmission.

Further on, the research highlighted certain general information about the finite elements and the means of applying them, and also models and analyzes the components of the cardanic transmission; the studies were done with the help of the computer aided design program Solidworks, whereas for analyzing the finite elements we used the Cosmos program. The working conditions of the cardanic transmission of a Dacia vehicle were simulated, the model being charged with values from zero to 800 Nm, and noting down the information.

For a correct analysis, we designed a stand that allows the charging from minimum to maximum (0-800 Nm), and after the reading, the data were written in tables in order to emphasize the differences between the methods of calculation.

After achieving the results, we observed that the values are between comparable limits and do not exceed the maximum allowable values of 7.8 degrees, as prescribed in the literature.

CONCLUSIONS

The present study can be taken as a model for other types of cardanic transmissions as well, with redesigning the stand for larger cardanic transmissions and using the finite elements programs with parameterized models.

REFERENCE [1] Cristescu, D. Ş.a. Automobilul. Construcţie. Funcţionare. Depanare. Technical Publishing House, Bucharest, 1986. | [2] Dudita, Fl. Transmisii cardanice. Technical Publishing House, Bucharest, 1966. | [3] Sofonea, G., Fratila M., Vasiloaica, C., Pascu, A., Avrigean, E. Rezistenta materialelor - Indrumar de laborator. "Lucian Blaga" University of Sibiu Publishing House, 2000.73 | [4]Radu, GH. N., Munteanu, M. GH., Bit, C. S., Rezistenţa materialelor şi teoria elasticităţii, Vol. III. Macarie Publishing House, Targoviste, 2000. | [5] Untaru, M. Ş.a. Calculul şi construcţia automobilelor. Didactical and Pedagogical Publishing House. Bucharest, 1982. | [6] * * *, Internal research conducted by the personnel of SC COMPA SA on the functioning of the cardanic transmissions. |