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MODAL DYNAMIC ANALYSIS FOR THE SHAFTS OF A GEARBOX

Abstract: *The main goal of this paper is to determinate the deformations of the shafts of a gearbox. A CAD model of a gearbox is designed in SolidWorks. The dynamic simulation of the gearbox is performed with ADAMS multibody software. In the first step of the simulation the parts are considered as rigid ones. The second step of the simulation proposes to consider the elasticity of the parts, especially of the gearbox shafts. For that, the shafts are meshed with tetrahedral finite elements in order to be considered as deformable solids. The results of the simulation are the shafts vibration modes and the deformations of the mass centers that are discussed in the paper.*

Key words: *computer graphics, shafts, modal dynamic analysis.*

1. INTRODUCTION

A composite mechanical transmission may consist of several partial mechanical transmissions (like sprockets, gears, belts, chain, etc.) connected in series or parallel, directly or through couplings.

When it comes to designing such complex mechanical transmission, in general, a few kinematic and dynamic characteristic parameters are known. The following kinematic and dynamic characteristic parameters are often known or can be deduced:

First design alternative:

- power on the output shaft of the transmission or the input power of the working machine to be coupled to the transmission to be designed, P_e, in $\left[kW\right]$;

- speed or angular speed on the transmission output shaft or input machine shaft n_e;

- operating mode of the transmission;

In this situation the designer must choose the transmission or component transmissions, the overall transmission ratio of the composite mechanical transmission, starting from the possible engine speeds.

Second design alternative:

- power on the output shaft of the transmission or power input of the working machine, P_e or P_i, in [kW];

- motor rpm n_s;

- total gear ratio transmission;

- operating mode of the drive train;

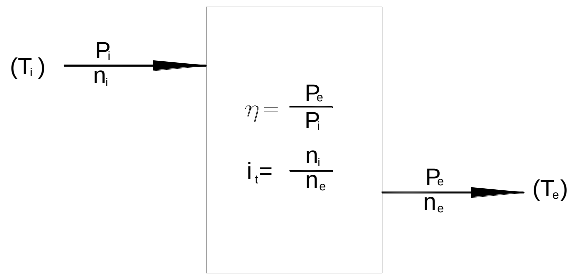

Fig. 1. The kinematic and dynamic parameters of a composite mechanical transmission.

The aspect of dynamic analysis is presented on many research studies. The aspect of dynamic analysis of the rotors using the principle of modal reduction is presented in [1].

Also, the modal analysis of rotating shafts is presented in [2]. The aspect of gear noise reduction for the automatic transmission, based on finite elements dynamic simulation, is presented in [3]. The non-linear dynamic behavior study of a flexible shaft is presented in [4].

In this paper we intend to design and study the dynamics of a gearbox.

With all component geometry data known, we have developed CAD models for all parts of the gearbox, except for standardized components such as bearings, which were chosen according to the standard dimensions of the Solid Works library, we made the final virtual assembly of the CAD model, presented in Fig. 2.

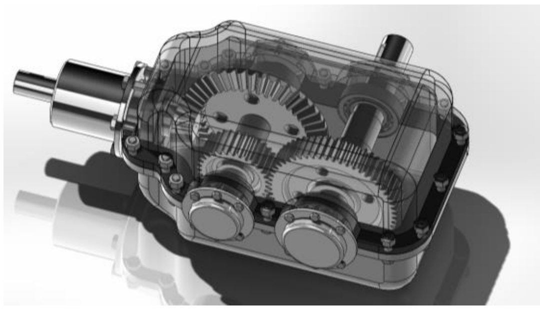

Fig.2. The assembled CAD model of the gearbox.

Below we present details of parts made according to the calculation of design in Solid Works.

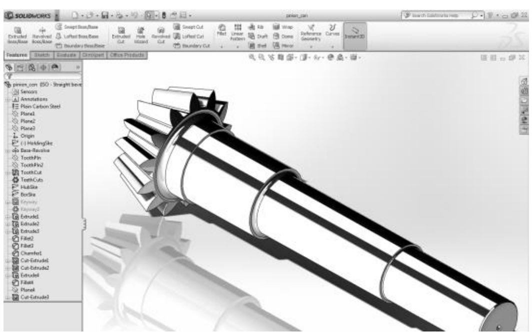

Fig. 3. CAD model of the bevel pinion gear.

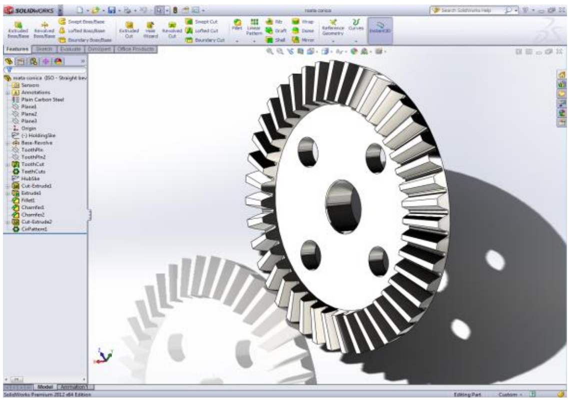

Fig.4. 3D model of bevel gear.

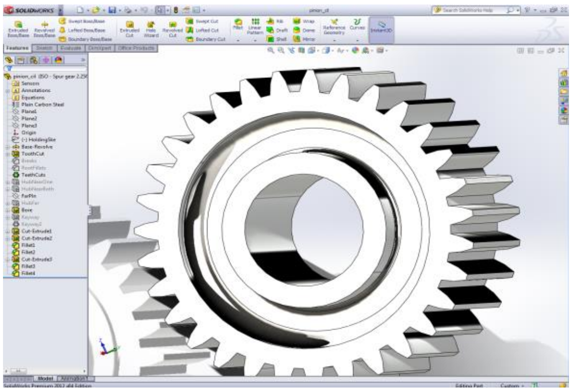

Fig.5. CAD model of cylindrical gear.

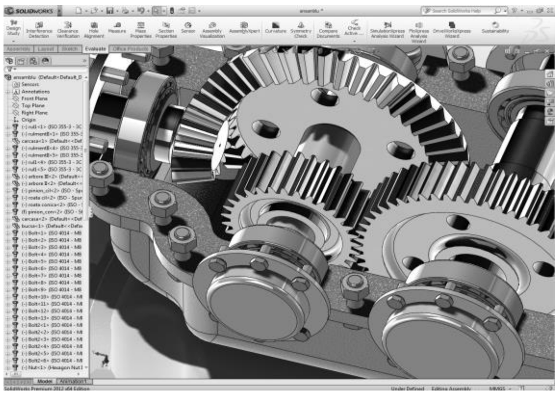

Fig.6. Detail of the gear unit assembly.

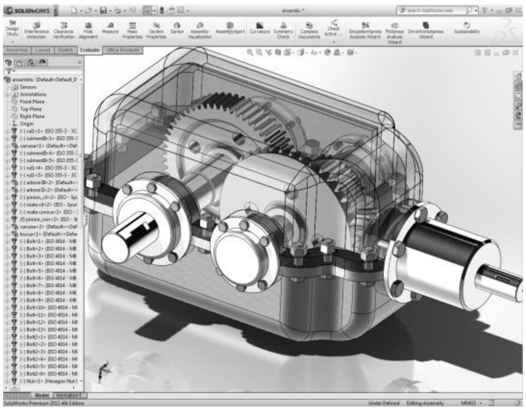

Fig.7. Details of the gearbox CAD assembly.

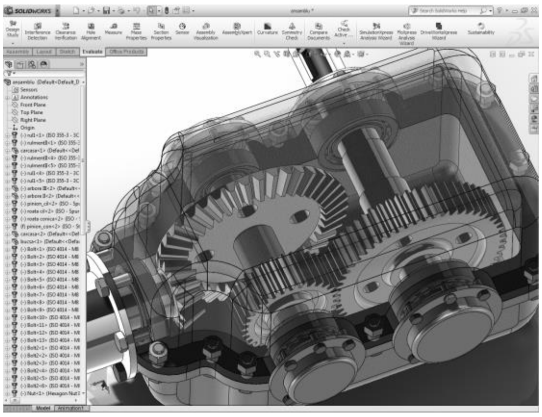

Fig.8. Details of the gearbox CAD assembly (top view).

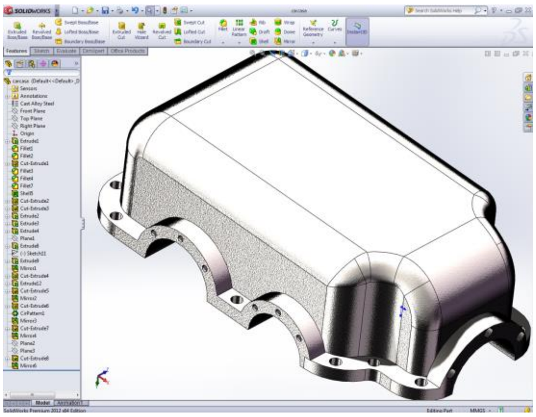

Fig.9. Detail of the gear housing.

Gear wheels CAD models were made using the GearTrax plug-in demo version, specifying input data, module, number of teeth, axle distance, reference rack [5, 6].

2. DYNAMIC MODEL OF THE GEARBOX IN ADAMS PROGRAM

In this paragraph we study the dynamic behavior of the gearbox shafts [7]. For this, the CAD model of gearbox was transferred as a Parasolid file to ADAMS program, where we intend to study the contact forces that occur during engagement, and other parameters kinematics and dynamics such shaft deformations.

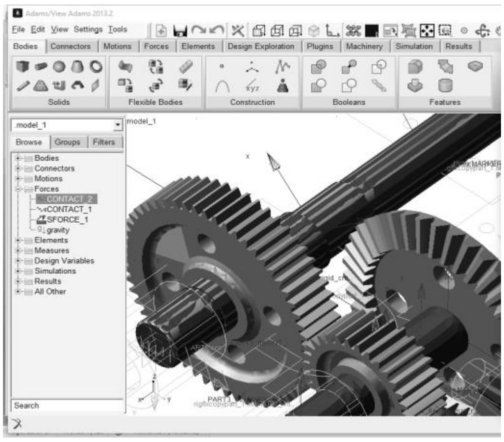

Fig.10. ADAMS dynamic model of bevel-cylindrical gear.

To define the conical and cylindrical gears, we defined the contact between the bodies of the two pairs of solids in the gear. Contact modeling was impact type, defining body stiffness, damping, and contact force exponent as shown in Fig. 11.

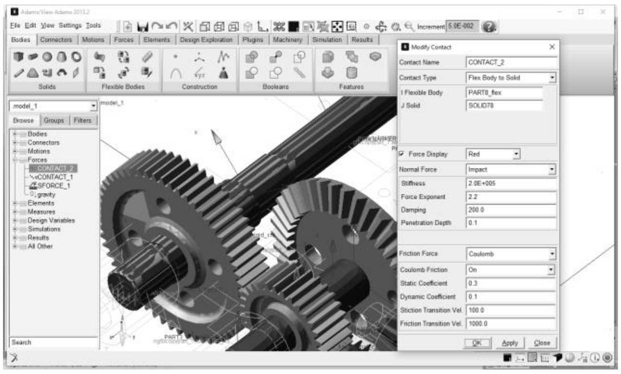

Fig.11. How to define the wheel teeth contact.

The input shaft motion gear in the gearbox was defined as an angular velocity of 157 rad / sec, as in Figure 12.

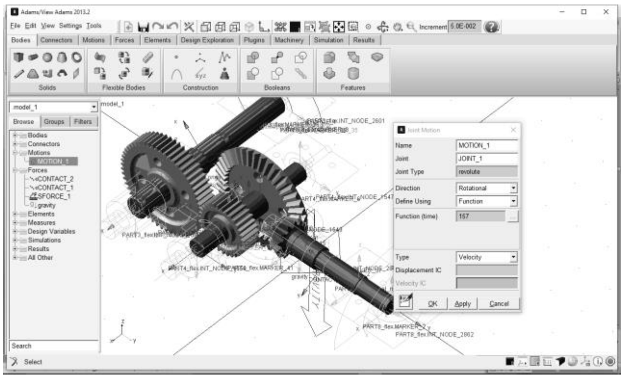

Fig.12. The law of motion of the input shaft.

Because the gearbox shafts are working at high revs, the highest being 157 rad / sec, it is necessary to do a study of the rigidity of the shaft, to study bending deformations. For this we considered the flexibility of shafts I, II and III (like deformable solid as it is in reality) and in the next paragraph we will perform a modal-dynamic study of these shafts.

3. DYNAMIC MODAL ANALYSIS WITH THE MSC.ADAMS SOFTWARE OF THE GEAR SHAFTS

The instant position of a marker that is attached to a node P on a flexible body B, is computed as the sum of three vectors (Fig. 13).

$$r_p = \vec{x} + \vec{s}_p + \vec{u}_p \qquad (1)$$

where: $\vec{x}$ represent the position vector from the fixed reference system origin to the origin of the local reference system, B attached to the flexible body.

$\vec{s}_p$ - represent position vector of the undeformed location of the point P, in relation to the local reference system attached to the body B;

$\vec{u}_p$ - is the translation of the point deformation vector P, the position vector from the undeformed point location to the deformed location;

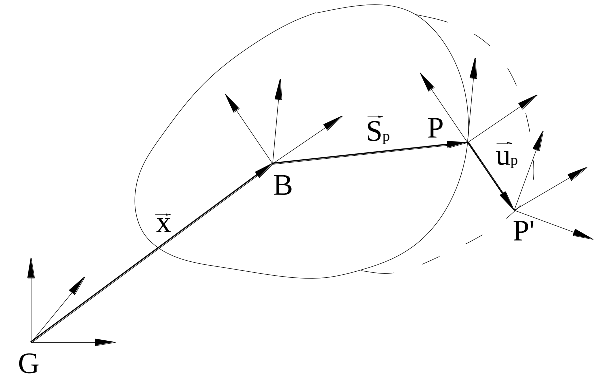

Fig. 13. Deformation point position vector P' of a flexible body relatively to a reference system attached to body B and base *G*.[8]

If we rewrite the equation (1) into a matrix form, expressed in relation to the basic coordinate system:

$$r_p = x + {}^{G}A^{B}\left(s_p + u_p\right) \qquad (2)$$

where: x is the position vector, from the origin of the fixed system, to the origin of the local reference system attached to the flexible body B, expressed in the fixed (base) coordinate system. The elements of vector $\vec{x}$, x, y and z, are the generalized coordinates of the flexible body.

In the previous paragraph, where we studied the kinematic and dynamic parameters of the mechanical transmission, the system components were considered rigid solids. In fact, during the operation of the gearbox assembly, the shafts get bending and torsional deformations due to the elasticity of the material. Because the forces that load the shafts vary after a symmetrically alternating cycle, and the moment after a pulsating cycle, the variation of these loads produces a variable load, and the fatigue of the material.

The overlapping frequency spectrum with the spectrum of the disturbing forces may lead to the natural resonance frequencies, so in this section we considered necessary, for the system, to consider deformability of the elements, in particular the shafts. It is characterized by large resonance amplitudes of movement in certain points or areas of the transmission, accompanied by high stresses or substantial relative movement, which can lead to fatigue breakage, malfunction, wear or sharp noises.

In order to consider the deformability of the shafts we made a modal-dynamic analysis, where we proceeded with the steps described below.

The shafts were meshed into tetrahedral-type finite elements, the master and slave nodes were defined according to the body links in the system, then the dynamics simulation was run, considering the flexibility of the shafts.

By post-processing of the results, as we will present the deformations, the speeds and accelerations of deformations for the marker corresponding to the center of mass of the shafts I, II and III.

4. DEFORMATIONS OF THE MASS CENTER FOR THE SHAFT II

These results are presented in Fig. 14 to Fig. 18.

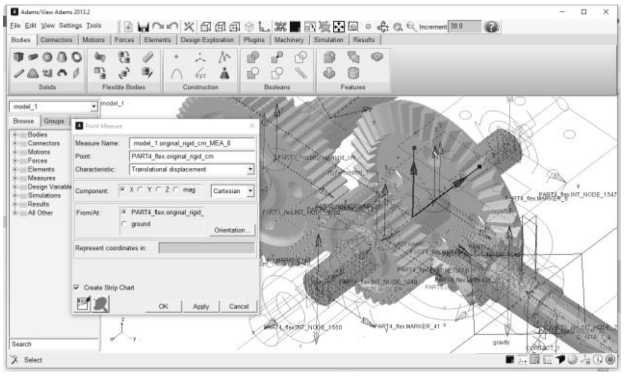

Fig. 14. Marker attached to the center of mass of the shaft.

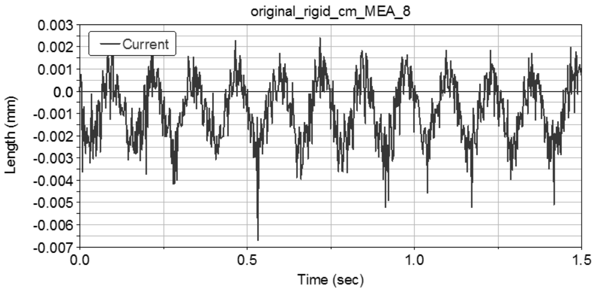

Fig.15. Motion of the shaft center, along the x-axis.

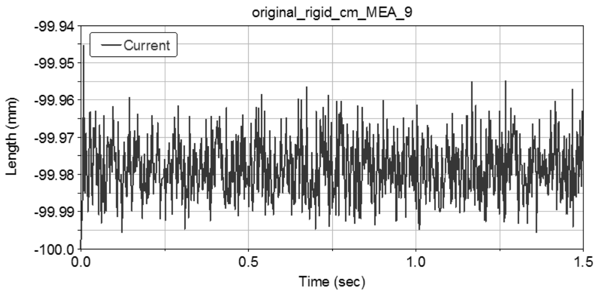

Fig.16. Motion of the shaft center on y-axis.

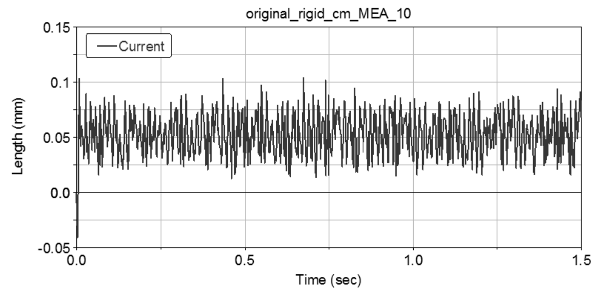

Fig.17. Motion of the shaft center, along the z axis.

The deformation of translation, of the marker attached to the second shaft centers of the mass is as follows.

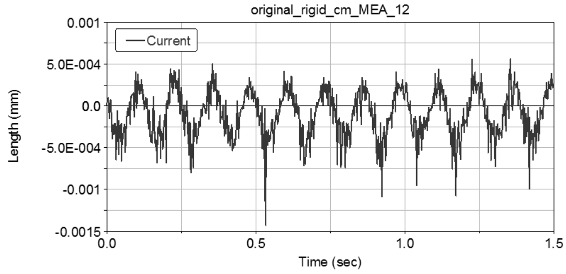

Fig.18. Deformation of the shaft center, along the x-axis.

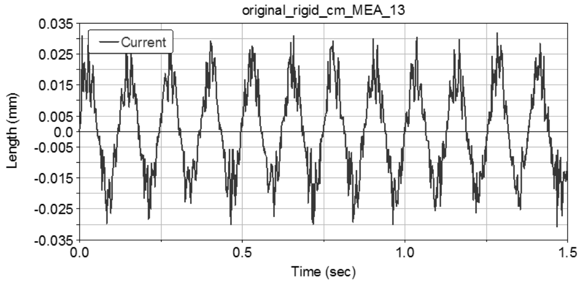

Fig.19. Deformation of the shaft center, along the y-axis.

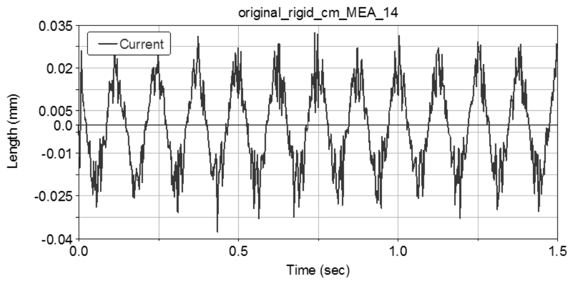

Fig.20. Deformation of the shaft center, along the z axis.

From the interpretation of the variation graphs of the deformations, it is observed that the shaft has bending deformations whose maximum value is 0.03 mm. Bending deformation is within acceptable tolerances, the permissible arrow for shafts supporting toothed wheels being given by the relationship:

$$f_{\max} \le f_{adm} = (0,01...0,03)\,m \qquad (3)$$

m is the module of the gear wheel. In our case, the module being 2.25 mm, the allowable arrow is within the limits: 0.0225...0.0675, so the maximum arrow of 0.03 mm is not greater than the allowable arrow.

5. SHAFT VIBRATION MODES

In this paragraph we will represent the deformed shape of the gear shaft II, corresponding to its own vibration modes.

Each mode of vibration is characterized by its own frequency, as well as by the mode of deformation.

The deformed representation of shaft II for vibration mode 7, with its own frequency of 3928.3 Hz, is shown in Fig. 21.

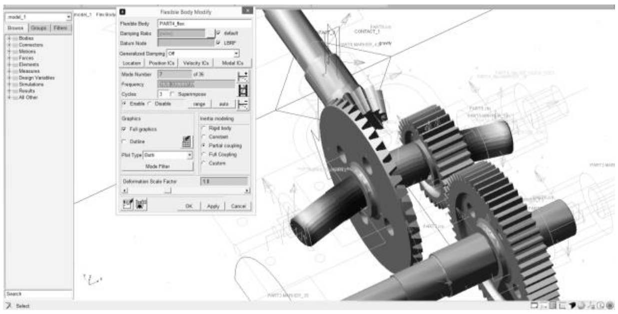

Fig. 21. Deformed representation of shaft II for vibration mode 7.

The deformed representation of vibration mode 8 of the shaft, with its own frequency of 3946.3 Hz, is shown in Fig. 22.

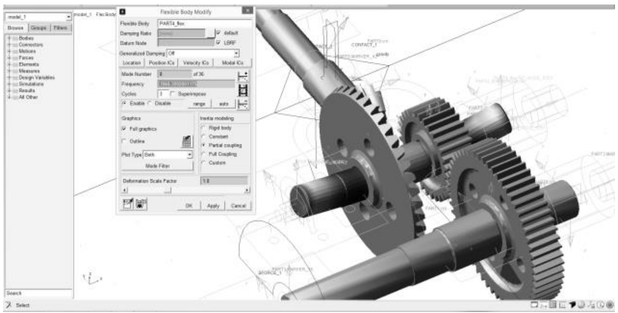

Fig. 22. Deformed representation of shaft II for vibration mode 8.

The deformed representation of the vibration mode 9 of the shaft, with its own frequency of 9312.4 Hz, is shown in Fig. 23.

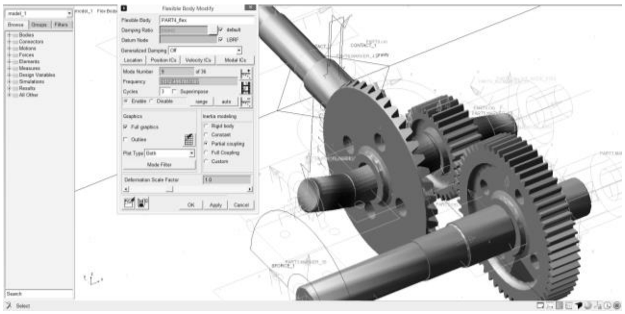

Fig. 23. Deformed representation of shaft II for vibration mode 9.

The deformed representation of vibration mode 10 of the shaft, with its own frequency of 9337.8 Hz, is shown in Fig. 24.

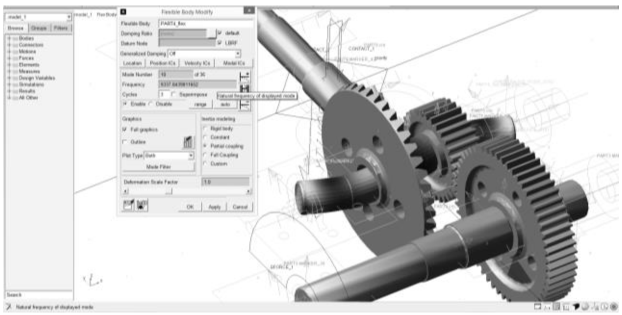

Fig. 24. Deformed representation of shaft II for vibration mode 10.

Dynamic modal analysis was also carried out for the gearbox input shaft. The deformed representation of the shaft, corresponding to its own vibration modes, is shown below.

The deformed representation of shaft I for vibration mode 7, with its own frequency of 3343.44Hz, is shown in Fig. 25.

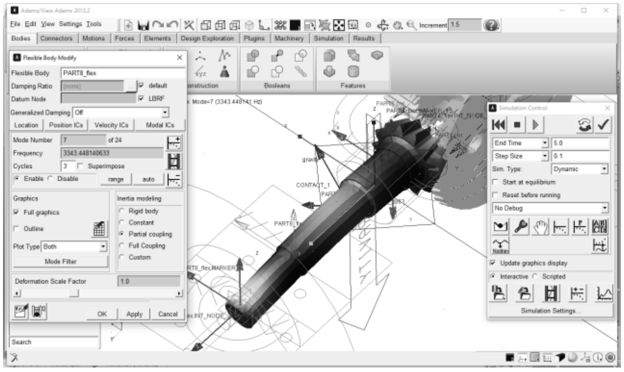

Fig. 25. Deformed representation of shaft I for vibration mode 7.

The deformed representation of shaft I for vibration mode 8, with its own frequency of 3359.3 Hz, is shown in Fig. 26.

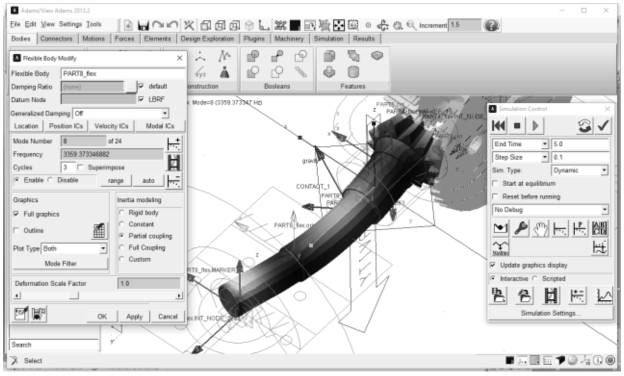

Fig. 26. Deformed representation of shaft I for vibration mode 8.

The deformed representation of shaft I for vibration mode 9, with its own frequency of 8390.89 Hz, is shown in Fig. 27.

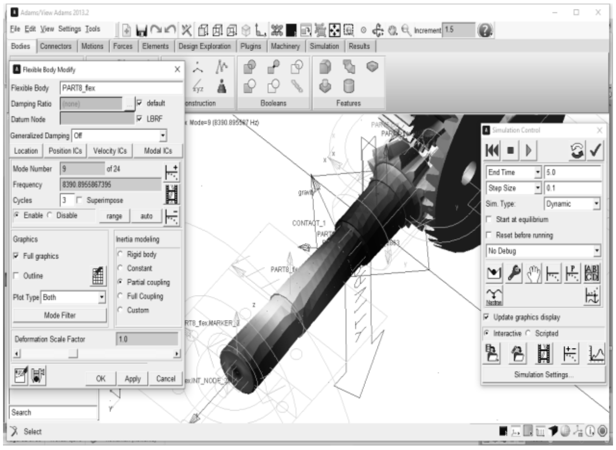

Fig. 27. Deformed representation of shaft I for vibration mode 9.

The deformed representation of shaft I for vibration mode 10, with its own frequency of 8769.57 Hz, is shown in Fig. 28.

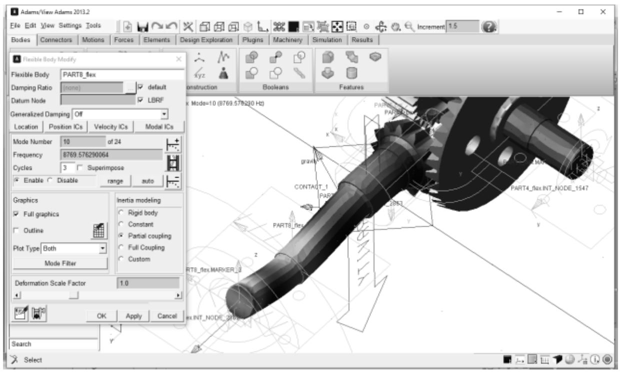

Fig. 28. Deformed representation of shaft I for vibration mode 10.

The deformed representation of shaft I for vibration mode 11, with its own frequency of 8791.24 Hz, is shown in Fig. 29.

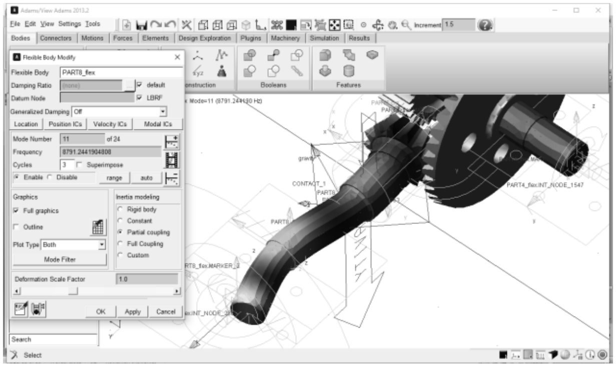

Fig. 29. Deformed representation of shaft I for vibration mode 11.

The deformed representation of shaft I for vibration mode 14, with its own frequency of 2,185E + 004Hz, is shown in Fig. 30.

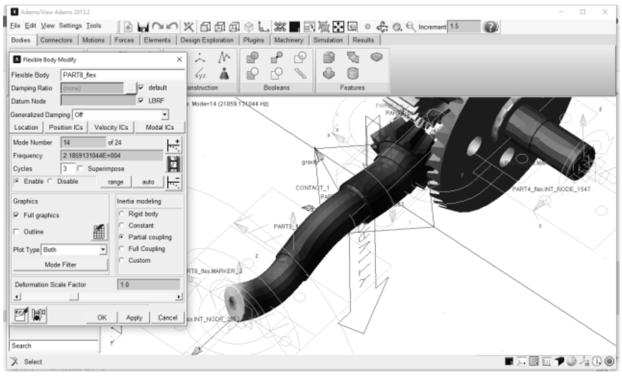

Fig. 30. Deformed representation of shaft I for vibration mode 14.

6. CONCLUSION

In this paper are presented some results concerning the modal dynamic analysis for the shafts of a gearbox. The ADAMS software allows the analysis of the gearbox motion considering the shafts flexibility.

REFERENCES

[1] Khulief, Y. A., & Mohiuddin, M. A. (1997). *On the dynamic analysis of rotors using modal reduction*. Finite Elements in Analysis and Design, 26(1), 41-55, ISSN: 0168-874X.

[2] Han, R. P., & Zu, J. Z. (1992). *Modal analysis of rotating shafts: a body-fixed axis formulation approach*. Journal of Sound and Vibration, 156(1), 1-16, ISSN: 0022-460X.

[3] Campbell, B., Stokes, W., Steyer, G., Clapper, M., Krishnaswami, R., & Gagnon, N. (1997). *Gear noise reduction of an automatic transmission through finite element dynamic simulation* (No. 971966). SAE Technical Paper.

[4] Berger, S., Bonneau, O., & Frene, J. (2000). *Influence of axial thrust bearing defects on the dynamic behavior of an elastic shaft*. Tribology International, 33(3-4), 153-160, ISSN: 0301-679X.

[5] Catrina, Gh., *Organe de masini*, vol. I, Reprografia Universitatii din Craiova, 1997, ISBN: 978-606-510-004-6.

[6] Catrina, Gh., *Organe de masini*, vol. II, Editura Universitaria, Craiova, 2002, ISBN: 978-606-510-897-4.

[7] Dumitru N., Margine A., *Bazele modelarii in ingineria mecanica*. Editura Universitaria Craiova, 2002, ISBN: 973-8043-68-7.

[8] Geonea I., *Calculul si proiectarea reductoarelor*. Universitatea din Craiova, 2018 (format electronic).

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