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Original scientific paper

Investigation of Toothed Shaft from the View of Modal Parameters

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Abstract: The paper deals with the investigation of toothed shaft from the view of modal analysis, which is a base of dynamic analysis. The gear mechanism is newly designed based on new principle, at which the motion and power are transferred from one input shaft to two output shafts in one direction. Considering that structures vibrate in special shapes when excited at their resonant frequencies, by understanding the mode shapes, all the possible types of vibration can be predicted. The paper aimed to create experimental and computational model of the shaft with subsequent verification of selected theories, to compare results obtained by numerical and experimental modal analysis and to assess the dynamic characteristics of the component with respect to its operating conditions. There are described conditions of both numerical and experimental modal analysis in the paper. The measurements have shown that the values of natural frequencies along with the natural shapes are adequate and comparable within both of the investigation methods. Moreover, from the results it is clear that output shaft should not be subjected to resonance. On the other hand, by means of modal analysis, the number of teeth was specified, that can be risky at the dynamic load of the gear, if the gear ratio changes.

Keywords: analysis; frequency; gear; modal parameters; resonance; toothed shaft

1 INTRODUCTION

In engineering, the enormous commercial and safety aspects associated with redesigning a machine oblige the best possible understanding of dynamic properties of structures and the repercussion of any design changes. Modern structures must be light in weight and high in strength. The rapid development in the aeronautical and astronautical industries has challenged many disciplines of engineering with diverse technological difficulties. The dynamic behaviour of many types of structures have been a significant catalyst to the development of modal analysis. Aircraft and spacecraft structures impose stringent requirements on structural integrity and dynamic behaviour which are shadowed by rigorous endeavours to reduce weight. Modal analysis has also found increasing acceptance in the civil engineers' community where structural analysis has always been a critical area. The concern of dynamic behaviour of civil structures under seismic and wind loading warrants the application of modal analysis. Experimental modal analysis as a troubleshooting tool also plays a crucial role in the study of vehicle noise and vibration harshness. A simple modal analysis of a body-in-white or a subframe structure is a typical application. However, more sophisticated applications have been achieved such as those involved in vehicle fatigue life estimation, vehicle suspension with active vibration control mechanism and condition monitoring and diagnostic system for the vehicle engine. Another prominent application area for modal analysis is in the behaviour study of classical engineering structures (machines, mechanisms, etc.). [1]

Modal analysis is used to solve a number of technical problems such as the specification of modal frequency system which, in accordance with excitation frequencies can lead to resonance; the critical speed, etc., when verifying the reliability of simplified mathematical models assembled in so-called geometric coordinates, comparing the results of experimental measurements, modification of mechanical systems connecting additional elements to such retune them out of band harmful effects. [2]

The basic idea of modal analysis is to describe complex phenomena in the dynamic behaviour of components by simple features, i.e. by natural vibration modes. This area of modal analysis that was developed during the nineteenth century is based on mathematical knowledge related to the solving of partial differential equations that describe various continuous dynamic structures. An elegance of the solution is obvious, while the scope of solvable structures is limited. The concept of discretization of the object and the introduction of matrix analysis achieved the top level in the theoretical modal analysis at the beginning of the last century. The theory has been developed so that structural dynamic analysis of any system can be performed in the case when the mass and stiffness distributions of a structure in matrix forms are known. However, this theory could only be implemented after computers have been accessed. In this aspect, theoretical (or analytical) modal analysis is considered a numerical modal analysis. [3]

The theoretical basis of the technique is secured upon establishing the relationship between the vibration response at one location and excitation at the same or another location as a function of excitation frequency. For a harmonic force $f(t) = F(\omega)e^{i\omega t}$, the response of the system is another harmonic function $x(t) = X(\omega)e^{i\omega t}$, where F is a maximal force, X is a complex amplitude, ω is an angular velocity and t is time. The ratio of the displace response and the force input is often defined as the Frequency Response Function (FRF) of the system that is simply mathematically expressed by Eq. (1) [4, 5]

$$H_{(\omega)} = \frac{X_{(\omega)}}{F_{(\omega)}} \tag{1}$$

The final goal of the modal analysis is to achieve a mathematical model of the structure that matches as closely as possible the experimental results obtained in the tests.

The issue of modal analysis of a shaft was covered by several studies in recent years. Researchers Han and Zu [6] dealt with modal analysis of rotating shafts. A spinning Timoshenko beam, subjected to a constant moving load, was analysed in their study, at which a modal expansion technique was used. The formulation was based on a body-fixed axis reference system, and thus should be applicable to handle any general cross-sections. Using this alternative approach, dynamic quantities such as natural frequencies, mode shapes and system responses were easily computable. Mr. Chouksey with his co-authors [7] studied the influences of internal rotor material damping and the fluid film forces on the modal behaviour of a flexible rotor-shaft system. Experimental modal analysis was used in the study of modal properties of turbo-pump shaft, where an innovative suspending method has been proposed to reduce noise-to-signal ratio resulting from classic suspensions [8].

The possibilities to use the computer aid at numerical analysis of modal parameters were described in many investigations. In the study of researcher Ravikant [9], the model of drive shaft has been generated in SolidWorks and then imported into ANSYS workbench. Prof. Swapnil with his colleagues studied design and vibration of automobile gearbox element shaft with gear using Finite Element Analysis (FEA), at which software ANSYS was used for both 3D model creation and modal analysis [10]. The same software was used within the study [11] that dealt with modal parameters of shaft taken from the head stock of the lathe machine.

On the other hand, only a few researchers used two approaches to the modal parameters investigations. One of them is Mr. Nikhade with his colleagues [12], which dealt with the modal analysis of universal joint shaft for rolling mills. They compared results of natural frequency analysis obtained by means of software with the results from computational method performed based on Dunkerley's equation. Both experimental and FE modal analysis were applied at natural frequencies investigation of intermediate shaft of automobile gear box studied by prof. Dive [13].

The objective of presented research has been to identify the modal parameters of the toothed shaft as a component of special two-output gear mechanism developed at the authors´ workplace, at FMT TUKE. Two approaches, experimental and numerical, were used at the modal analysis for verifying the achieved results.

2 BASE CHARACTERISTICS OF THE GEAR

Investigated planetary gear mechanism consists of a two-stage transmission that is designed based on new principle, at which the motion and power are transferred from one input shaft to two output shafts in one direction. The module of every i^{th} gear wheel is $m_i = 1$. The kinematic diagram of entire gear mechanism is presented in Fig. 1 [14], while the 3D model is shown in Fig. 2. [15]

Toothed shaft, modal parameters of which are investigated, is labelled by number III. in Fig. 1 and the real view on shaft is shown in Fig. 3. It was produced from carbon steel C45, number of teeth is 72 and the module is 1 as it was mentioned above.

The restricted view on the gear ratios between the input shaft I. and the output shaft III. is in simplified form shown in Fig. 4. [15]

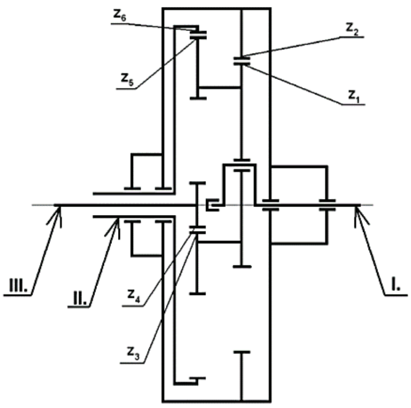


Figure 1 Planetary gear (I. – input shaft, II. and III. – output shafts, z_i – teeth number of individual gears: $z_1 = 90$, $z_2 = 99$, $z_3 = 81$, $z_4 = 72$, $z_5 = 101$, $z_6 = 110$)

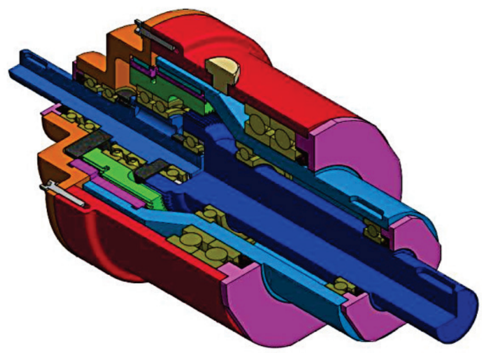


Figure 2 Virtual 3D model of designed gear mechanism [15]

Figure 3 Toothed shaft

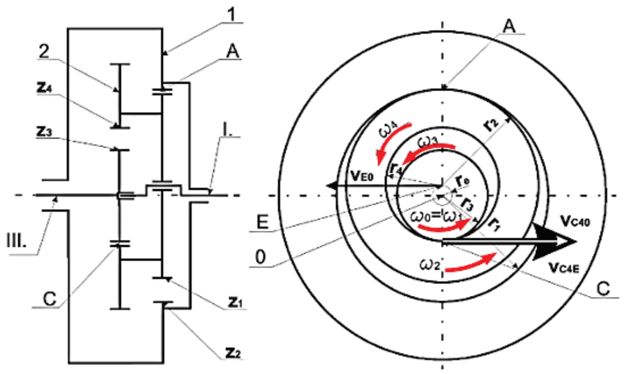


Figure 4 Restricted view on the gear ratios between the input shaft I. and the output shaft III. [15]

From the kinematic analysis of the mechanism results a gear ratio between the input shaft and the output shaft number III.:

$$p_{\text{I-III}} = \frac{\omega_{\text{I}}}{\omega_{\text{III}}} \tag{2}$$

$$p_{\text{I-III}} = \frac{z_4}{\left(\frac{z_2}{z_1} z_3 - z_4\right)} \tag{3}$$

where ω_I is input angular velocity of the shaft number I. and ω_{III} is output angular velocity of the shaft number III.

Considering that input angular velocity ω_{III} was 293,06 rads^{-1} and based on computed gear ratio $p_{I\text{-}III}$ = 4,21, it could be possible to specify output angular velocity and operating frequency of the shaft III. in the following way

$$\omega_{III} = \frac{\omega_I}{p_{I\text{-}III}} = \frac{293,06}{4,21} = 69,6 \ \text{rads}^{-1} \tag{4}$$

$$f_{III} = \frac{\omega_{III}}{2\pi} = \frac{69,6}{2\pi} = 11,08 \ \text{Hz} \tag{5}$$

Tooth frequency at the toothed shaft III. was computed by formula (6)

$$f_t = f_{III} \times z_4 = 11,08 \times 72 = 798 \ \text{Hz} \tag{6}$$

In the article two approaches to the modal analysis were presented focused on the numerical and experimental methods. Obtained data of modal analysis will be the base for the dynamic analysis and for the next experiments that authors are going to realize.

Modal analysis can be realized in two different ways, either in the theoretical plane as a calculation within the numerical analysis or in a practical level by performing experimental measurements on the real physical system.

The values of the modal parameters, which are acquired by computing methods are compared with the measured values acquired by experimental analysis. In the technical practice, these values coincide exactly only occasionally. The resonance frequencies can be identified by comparing results of modal analysis (natural frequencies) and excitation frequencies. The shaft operation can be dangerous when they are close to each other, so by means of modal analysis, a designer can avoid the gear failure already when designing a gear mechanism. If this situation occurs, the designer has to modify the stiffness of the component to change the value of the natural frequency. [16]

3 METHODS OF INVESTIGATION

3.1 Experimental Modal Analysis

Modal analysis is currently one of the fields of science that is developing rapidly. This is primarily due to the availability of new measuring and computing resources, without which it would be impossible to obtain, and then to process the measured data.

Along with the development of modern computer technology that enables to use finite elements for numerical computation, the experimental modal analysis (EMA) has become the main tool for solving complex structural vibration problems in real technical practice. [17]

Modal experimental testing is a technique used to derive the modal model of a linear time-invariant vibratory system. Test preparation involves selection of a structure's support, type of excitation force(s), location(s) of excitation, hardware to measure force(s) and responses; determination of a structural geometry model which consists of points of response to be measured; and identification of mechanisms which could lead to inaccurate measurement. [17, 18]

During measurements, the shaft was put on the foam rubber MOLITAN (as a soft pad) that simulates a freely supported part (without constraints). The PULSE analyser, model 2827 - 002 was used for data processing. It consists of the measuring module type 3109 and communication module type 7533. The measuring module was used for geometry model creation and for measuring points designing. The communication module was used for data exporting and importing.

Piezoelectric accelerometer type 4374 was used as the vibration sensor. During an acceleration sensor selecting, it is necessary to take into account its weight, because it can cause a distortion of measurement data. It is also very important to take into account the sensor orientation. Correct orientation of the acceleration sensor is based on compliance of its coordinate system with coordinate systems of measured component. Natural frequencies investigation of toothed shaft was carried out using a modal hammer Brüel&Kjær type 8203 with a plastic tip.

Within the experiments, two amplifiers of the same type 2627–A were used. One of them was used for amplifying sensor signal and the second one for amplifying excitation signal in modal hammer. The measuring set is shown in Fig. 5.

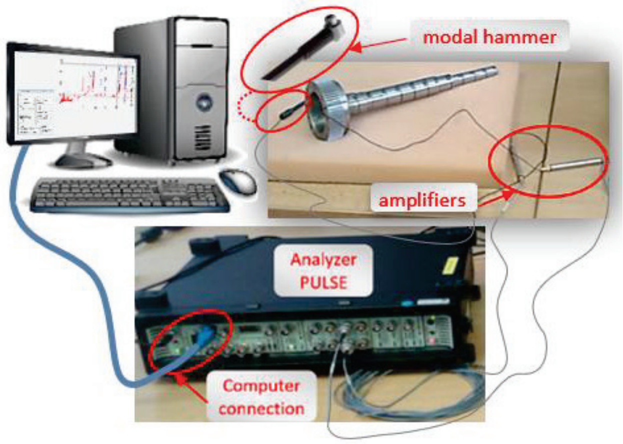


Figure 5 The measuring set

In the software MTC – Hammer simplified geometry of toothed shaft was created, where also measuring points for response were designed. They are highlighted in Fig. 6. Point number 5 was selected as the reference point (see Fig. 6). The distance between measuring points was specified by software MTC Hammer.

Excitation of individual points was carried out in two radial directions x and y (axis z was coincident with axis of the shaft) and it was done five times at every point in both directions because of averaging of FFT.

For dynamic signals processing the Fourier Transformation (FT) is usually used. It enables to convert a time dependence of the measured values onto the frequency domain. Expression of Fourier transformation is given by formula: [19]

$$x(t) = \frac{a_0}{2} + \sum_{n=1}^{\infty}\left[a_n \cos\left(\frac{2\pi n t}{T}\right) + b_n \sin\left(\frac{2\pi n t}{T}\right)\right], \tag{7}$$

where a_n and b_n are the Fourier coefficients of the function.

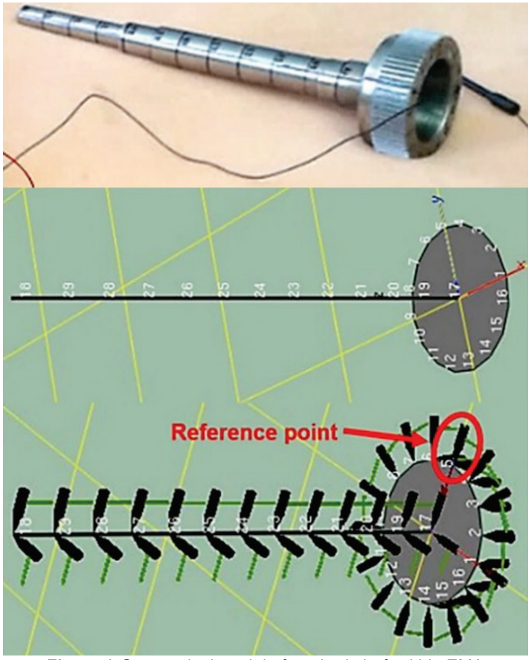


Figure 6 Geometrical model of toothed shaft within EMA

Applying the Fourier transformation in digital form requires the creation of appropriate algorithms for processing discrete data, which is quite difficult. It is a reason why optimized algorithm called "Fast Fourier transform" is currently used and it was also used at these experiments. FFT is a computationally efficient method of generating a Fourier transform. The main advantage of an FFT is speed given by decreasing the number of calculations needed to analyse a waveform. Final frequency response function is presented in Fig. 7.

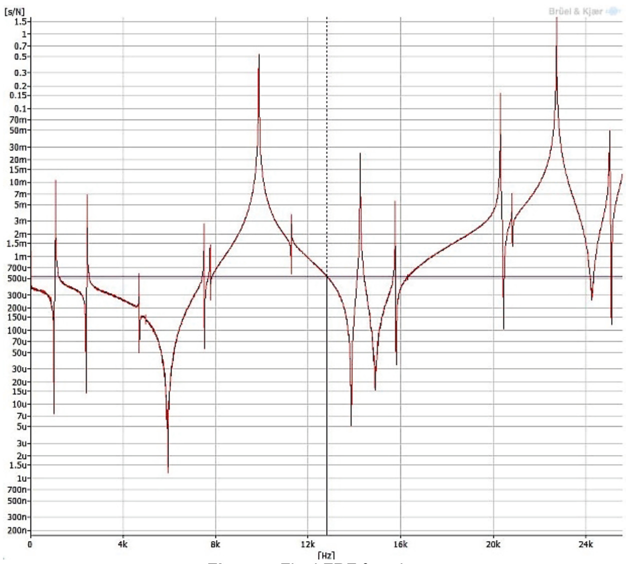


Figure 7 Final FRF function

3.2 Numerical Modal Analysis

The use of numerical techniques in combination with the development of measurement techniques and the use of modern hardware provide opportunities to analyse more and more complex physical phenomena. Of special importance here is the progress in techniques used to create virtual models of real structures which model their behaviour in the best possible way. [20]

Basically, finite element modelling subdivides a structure into very small regions, called finite elements (FE), where a displacement of the element is analytically obtainable. The selection of element types is determined by basic theory of elasticity and strength of materials. [21] Nodes are used to define each region of element. All of the finite elements are assembled into one large model taking care of the balancing of forces and compatibility at each interface. The boundary conditions are applied and this model with a large set of simultaneous equations is solved using numerical procedures. The knowledge or skills of defining nodes, elements, boundary conditions and solving equations are needed in order to model correctly. Nodes are defined according to the geometry of a structure, anticipated mode shapes and type of elements. Elements are selected based on the type of characteristic deformations anticipated which gives some ideas of mode shape patterns that are expected. Thus, the knowledge of expected mode shapes is important for identifying the type of elements and node density.

The boundary conditions are defined in the model to reflect the appropriate support conditions on a real structure. The physical restraint of the support needs to be known so that the actual boundary conditions can be modelled. In order to perform a correct analysis, types of solution schemes should be known which must adhere to the required analysis. [22]

Parallel with the experimental modal analysis, the numerical model of the toothed shaft was prepared in software PTC Creo. It is presented in Fig. 8. The 3D model was freely supported in the virtual environment; steel was selected as the material with the following characteristics: specific density 7,82708E+03 kgm^{-3}, Young's modulus 210 GPa, Poisson's value 0,27. The mesh was generated with 12782 finite elements of tetrahedral type.

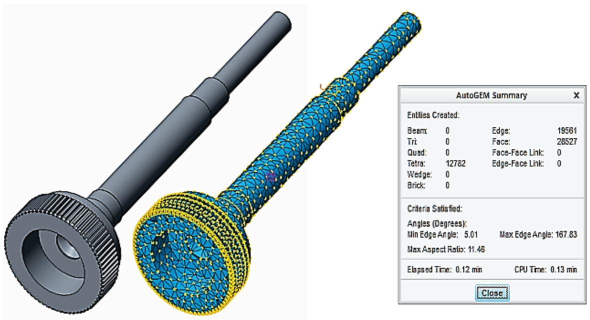


Figure 8 Virtual model for numerical analysis created in PTC Creo software

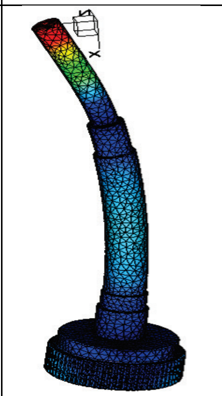
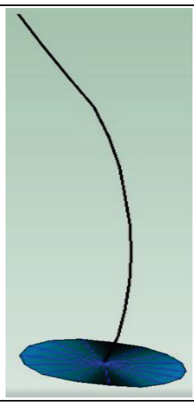
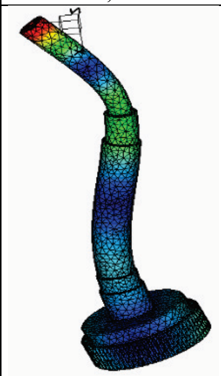
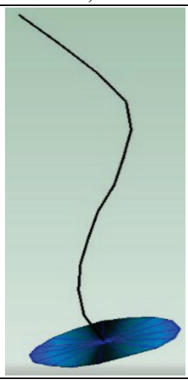
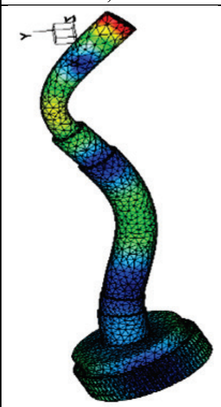
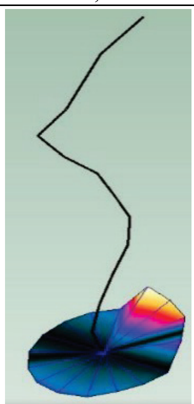
4 RESULTS AND DISCUSIONS

The first three natural frequencies obtained that were achieved by means of both experimental modal analysis and by Finite Elements Method (FEM) are listed in Tab. 1. Moreover, the first three mode shapes, which correspond to the first three natural frequencies, were evaluated within both methods of modal analysis. They are shown in the same Tab. 1.

It is clear from Tab. 1 that results of both experimental and numerical modal analysis are comparable. The differences between the values of frequencies achieved by numerical and experimental analysis were probably caused by the fact that real body was not totally unconstrained (it was positioned on the soft foam) and also by unequal numbers of finite elements. At the experimental analysis, the grid of finite elements was less dense (this means that the number of elements was several times smaller as it was at the numerical modal analysis in PTC Creo software. It was due to software Lab-Shop and MTC Hammer of measuring system PULSE that processed EMA results).

From the results it is also clear that already the first natural frequency (as minimal natural frequency) 1072,6 Hz is higher than calculated teeth frequency 798 Hz. It means that every higher measured natural frequency will differ from the tooth frequency, so the output shaft should not be subjected to resonance.

Table 1 The first three natural frequencies and modes of toothed shaft (number of teeth 72) obtained by both numerical and experimental modal analysis

Natural modes / frequencies	Numerical method	Experimental method
The 1st natural mode / frequency f_1	 1072, 6 Hz	 1080,12 Hz
The 2nd natural mode / frequency f_2	 2 411,55 Hz	 2 423,34 Hz
The 3rd natural mode / frequency f_3	 4 639,47 Hz	 4 656,12 Hz

However, the gear has been designed so as to be possible to vary the number of teeth. Numerical analysis showed that the number of teeth in usable range affects the natural frequencies only in a very slight way. On the other hand, at the same angular velocity of the shaft, the number of teeth significantly affects tooth frequency that can be considered as excitation frequency. Therefore, the problem with resonance could occur, if the number of teeth of the output shaft will increase to the values of 98 - 100. In this case the excitation (tooth) frequency will be 1085,84 - 1108 Hz, which is close to the first natural frequency of the shaft. Fig. 9 shows the first mode of the toothed shaft with 98 teeth at which the natural frequency is 1076,1 Hz.

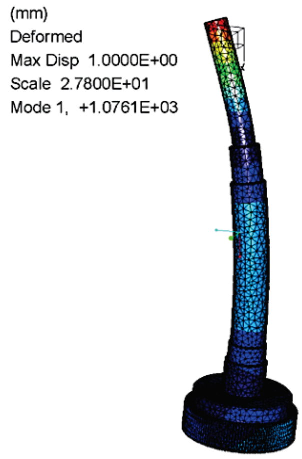


Figure 9 The first mode of shaft with the teeth number of 98

5 CONCLUSIONS

Modal data are extremely useful information that can assist in the design of almost any structure. It helps to identify weakness in the design and areas where improvement is needed. Usually reliable operation of the machine can be obtained by ensuring that the highest operational speed is below the first natural frequency of the shaft. [22]

The goal of the research was to find out if the toothed shaft of newly intended gear mechanism is well designed from the view of the stiffness and resonance frequency, because it is relatively thin with respect to its length. The excitation frequency was specified as the tooth frequency, so it depends not only on the angular velocity of the shaft, but also on the number of teeth of the gear. Modal analysis, realized by two approaches (experimental and numerical), showed comparable results with differences (in maximum 16,65 Hz at the 3rd natural mode) caused probably by both of the following reasons: different number of elements and different constrains, in spite of the fact that authors have tried to unify conditions as much as possible.

The results confirmed that the investigated output shaft should not be subjected to the resonance. From the results it is also clear that already the first natural frequency (as minimal natural frequency) 1072,6 Hz is higher than the calculated teeth frequency 798 Hz. It means that every higher measured natural frequency will differ from the tooth frequency, so the output shaft should not be subjected to the resonance.

On the other hand, by means of modal analysis, the number of teeth was specified that can be risky at the dynamic load of the gear due to a possibility of the resonance oscillations. It has been reported that the resonance problem may occur, if the number of teeth of the output shaft increases to a value in the range of 98 - 100. In this case the excitation frequency will be 1085,84 – 1108 Hz, which is close to the first natural frequency of the investigated shaft.

In the near future, next components of the gear mechanism will be subjected to the modal analysis, and to other tests to verify the functionality and reliability of the gear mechanism.

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Nomenclatures

m_i	module of i^{th} gear
z_i	number of teeth of i^{th} gear
ω_i	angular velocity in i^{th} shaft, rads^{-1}
F	force, N
X	amplitude, m
t	time, s
f	frequency, Hz
FEM	Finite Elements Method
FRF	Frequency Response Function
FFT	Fast Fourier Transformation
EMA	Experimental Modal Analysis

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