NUMERICAL AND EXPERIMENTAL MODAL ANALYSIS OF GEAR WHEEL

ANDREA CIZIKOVA[1], KATARINA MONKOVA[1], JAROMIR MARKOVIC[2], JAKUB CIENCALA[3],

[1]Technical University of Kosice
Faculty of Manufacturing Technologies
Presov, Slovak Republic
[2]Slovak Metrological Society, Banska Bystrica, Slovakia
[3]Technical university of Ostrava
Faculty of Engineering, Ostrava, Czech Republic

DOI : 10.17973/MMSJ.2016_11_201654

e-mail: katarina.monkova@tuke.sk

The paper deals with the theoretical and practical aspects that are necessary for identifying of modal parameters of the gear wheel as a component of the planetary gear mechanism, which are basic for the dynamic analysis. The paper presents thematically integrated part of the results of numerical and experimental modal analysis conducted on a particular gear wheel. The paper aimed to create experimental and computational model and subsequent verification of selected theories in defining the modal parameters, comparing results obtained numerical and experimental modal analysis and assessment of dynamic characteristics of the component with respect to its operating conditions. Numerical analysis was carried out in a computer CAD/CAM program PTC Creo, which works on the Finite Element Method. The results obtained by experimental modal analysis were evaluated by measuring system PULSE.Experimental measurements have shown that the values of natural frequencies along with the natural shapes are adequate and that no natural frequency is not the same or close to the gear frequency, so the component can be used in the real practice.

KEYWORDS

modal analysis, vibrations, modal parameters, excitation, frequency response function

1 INTRODUCTION

In the past two decades, modal analysis has become a major technology in the quest for determining, improving and optimizing dynamic characteristics of engineering structures. To appreciate its significance in the modern engineering arena and its potential for future science and technology, it is appropriate to capture some of the background facts which will help to underline this unique technology. Current modal analysis methods for identifying structures are usually based either on very simple and flexible techniques, or on very sophisticated computational ones.

Nowadays, the modal analysis is considered to be a modern study of the dynamics, which allows to determine the modal properties of the examined object. The modal properties are used to obtain the dynamic description of the mechanical system. In technical practice, most of the problems associated with excessive noise or mechanical vibration, caused by the characteristics of the mechanical system [Cacko 2014]. These properties are called "modal". It is obvious that thanks to the well-known modal parameter it is possible to prove and to predict the resulting properties of the mechanical system. The modal characteristics may be calculated and also evaluated under the test modal analysis[Baron 2016, Hloch 2008]. „To describe the modal parameters and behaviour of mechanical systems, there are used possibility of degradation of complex oscillatory plot into sub happening, each of which is characterized by natural frequencies and mode shapes its own.“ [Milacek 1992]

The resulting parameter of modal analysis belongs to the natural frequency of the system, custom shapes vibration and modal damping of the system. 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 verify the reliability of simplified mathematical models assembled in the so-called geometric coordinates, comparing the results of experimental measurements, the modification of mechanical systems connecting additional elements to such retune their out of band harmful effects. [Krolczyk 2014].

2 CONDITIONS OF THE EXPERIMENTS

The aim of experiment was to identify the modal parameters of the gear wheel as the component of planetary gear. Material of gear wheel was carbon steel C45. The PULSE analyser, model 2827 - 002 with its additional modules, was used for data processing. PULSE analyser consists of a measuring module type 3109 and communication module type 7533. The measuring module is used to create the geometry model, and constructing the measuring points. The communication module is used to export and import data, which is necessary for example at the base averaging FRF (Frequency Response Function) function.

As the vibration sensor was used piezoelectric accelerometer type 4374. When choosing an acceleration sensor, it is necessary to take into account its weight during measurement that is not a case of distortion of measurement data. For applications of the sensor of acceleration it is necessary to emphasize on its orientation. Correct orientation of the acceleration sensor is based on compliance of its coordinate system with coordinate measurement systems structure. Waking investigated of wheel was carried out using a modal hammer Brüel & Kjær, type 8203, which had a plastic tip. To amplify the sensor signal, the type of amplifier 2627–A was used, which could also be used as a simple converter in default mode with a sensitivity of 1 mV / pC. In the experiment, two amplifiers of the same type were used, one for amplifying the sensor signal and the second one for the response to the alarm signal amplification in modal hammer. The measuring set is shown in Figure 1.

Figure 1. The measuring set

3 METHODS USED AT THE EXPERIMENTS

Contemporary design of complex mechanical, aeronautical or civil structures requires them to become increasingly lighter, more flexible and yet strong. These stringent demands often made them more susceptible to unwanted vibrations. Where the vibration of a structure is of concern, the challenge lies on better understanding its dynamic properties using analytical, numerical or experimental means, or a combination of them. The process of determining the inherent dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes, and using them to formulate a mathematical model for its dynamic behaviour is called modal analysis. The formulated mathematical model is referred to as the modal model of the system and the information for the characteristics is known as its modal data.

The most used method to evaluate the numerical calculations is Finite Element Method - FEM. The numerical method used to simulate the stresses, deformations, natural frequencies, heat flow, electromagnetics, fluid flow, and so on. created the physical model. The principle of this numerical method is based on discretization of continuum to some number of finite elements, and these are being investigated parameters set out in the individual grid points.

The research method used at experimental method was the planned experiment. The most appropriate method to the processing of dynamic signals is a "Fourier Transform" (FFT - Fourier Transform). Fourier transform functions by converting the time dependence of the measured values in the frequency domain. It also compensates the primary function of harmonic sequence features that have different frequencies and phases. Compensation of the process is the need of making the product of simple waves approximately the shape of primary functions. Fourier transformation is defined as the sum of the substitution of any function $x(t)$ of harmonic functions [Malotova 2016].

Expression of Fourier transformation can be written in the following form [Bilosova 2011]:

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

where a_n and b_n are the Fourier coefficients of functions. Applying the Fourier transformation in digital form requires the creation of appropriate algorithms for processing discrete data, which is quite difficult. Currently optimized algorithm called "Fast Fourier transform". Experimental modal analysis to determine modal parameters examined in the housing with a frequency response, respectively frequency transfer function. FRF - Frequency Response Function has been implicated in response to the excitation system. It is essential to describe the relationship of linear mechanical systems. In general, the frequency transfer function can be defined as a measurement of the time course of the dynamic mechanical excitation system $f(t)$ and the corresponding time course of response to the mechanical system $x(t)$ in the frequency domain. The main reason for the frequency transfer function lies in its simple usage, in which it is possible to describe the response of the actual mechanical system [Trebuna 2012, Petru 2013].

Frequency transfer function is defined by the formula [Frankovsky 2011]:

$$H(\omega) = \frac{X(\omega)}{F(\omega)} \left[\frac{m}{N} \right].$$

2)

From the equation (2) expresses the frequency transfer function as the ratio of output and input to the system and also represents the property of representing the compliance of the analysed mechanical dynamic system [Hosnedl 2014]

4 EXPERIMENTAL AND NUMERICAL MODAL ANALYSIS

Modal analysis can be performed in two different ways either in the theoretical plane as a calculation or on 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 measured values acquired by the experimental analysis. In the technical practice these values coincide only occasionally. For the theoretical modal analysis, will draw up a motion equation and the resulting values of the modal parameters is obtained by applying the "modal transformation". Basis modal transformation consists in the replacement of the system with each other of the commitment of homogeneous differential equations by a system of independent, isolated to resolving, homogeneous differential equations. Complex calculations of the system of physical equations is in many cases necessary to undergo a needed simplification. The theoretical modal analysis becomes indispensable in the case of the absence of the actual mechanical system [Stojadinovic 2014, Panda 2014].

To implement the theoretical modal analysis can be summarised in the following steps, among which include [Bilosova 2011]:

1. compilation of the motion equation - physical model,
2. analysis of the free oscillation - modal model,
3. analysis of forced vibration for harmonic excitation - response model.

„In an experimental modal analysis will determine their own frequency, their own shapes vibration and modal damping of the mechanical system using the experimentally determined set of data frequency transfer function“. These functions are often referred to as "frequency characteristics". However, only the properly done experimental modal analysis determines what the most accurate real modal properties of a mechanical system. [Ungureanu 2011, Krehel 2016]

The opposite procedure is in the implementation of the experimental modal analysis, which includes the following steps [Metelski 2015]:

1. measuring appropriate sets of frequency response functions - response model,
2. the analysis of measured data - modal model,
3. the implementation of additional calculations - physical model.

In the context of experimental measurements was made modal analysis of the gear wheel by setting the analyser in the measuring system of the PULSE. In the measuring system, it was defined the apparatus such as the modal hammer and accelerometer with amplifiers with their basic parameters. The next step was the creation of the geometry in the software MTC – Hammer. On the model, 81 measuring points were highlighted. (Figure 2)

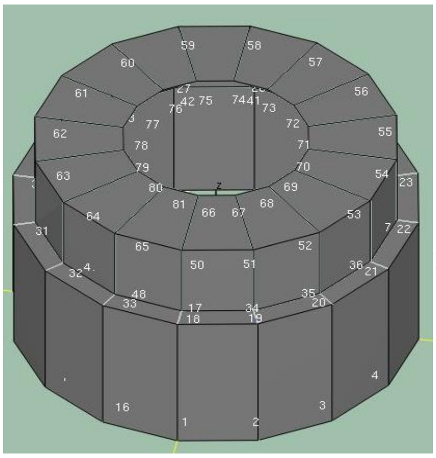


Figure 2. Geometry models created in the measuring system PULSE

After geometry creation, the next step was the location of the sensor, its orientation and the definition of excitation place. It was also important to define the reference point that was determined in excitation point with the number 10, shown in Figure 3.

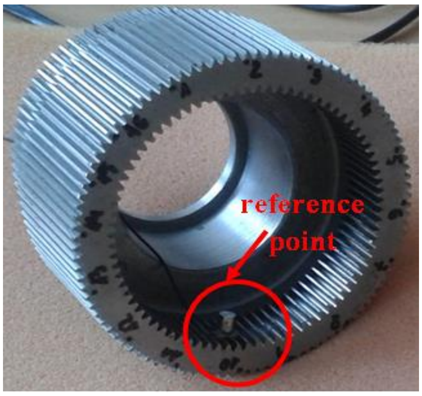


Figure 3. Location of the reference point on the gear wheel

When setting the parameters of the FFT analysis, it was necessary to define the number of discrete time data, which are used for the calculation of the spectrum. The set value for the number of discrete time data was 6 400 of spectral lines. Discrete time data are necessary when calculating the spectrum. Investigated the frequency has been set in the interval 0 - 25,6 kHz with a frequency resolution of 4 Hz. At the studied gear wheel, drew measuring points using the paper rulers.

The distance of the measuring points was determined by the software when creating the geometry of the model. An important part of the experiment was saving the reference element, that was the particular purpose for which the modal test shall be conducted. The gear wheel was studied positioned on a soft foam that was shown on Figure 1. The way of the excitation was elected by means of modal hammer. The choice of the method of the excitation was due to the small size of the wheel, the shape and stiffness. After setting all the necessary data, it was possible to realize the measurement of the modal data. The measurement of the modal data consisted of the sensor of the acceleration attached to the reference point using the beeswax, see Figure 3. After connection of sensor with reference point, the excitation of gear wheel by modal hammer followed. Excitation, the gear wheel on each of the measurement points took place on the stroke of modal hammer. Experimental modal analysis in the individual points of measurement carried out in two directions - radial and axial direction. A modal hammer to each of the measuring points struck five times in one direction, because of the averaging FF (Fast Fourier Transformation) function. This process is then repeated the experiment in making the second direction stroke of hammer. To view the various modes of vibrations was used module REFLEX PULSE system.

Parallel with the experimental modal analysis, the mathematical model of the gear wheel was prepared. The calculation was carried out by the method of finite elements in the software PTC Creo. It is presented in Figure 4.

Computing part of the modal test consists first in the creation of a model of the wheel and selecting the material - steel. After you create the model, define the material properties of the wheel. Among the basic material properties include the specific density of the material, whose value is 7,82708 10^{-6} $kgmm^{-3}$, the value of the Young's modulus is 2,1.10^{8} kPA, Poisson's value is 0,27. The value of the Young's module and Poisson's numbers correspond to the tabular value for steel. In the next, steel as the material was assigned to component. After this step, it was necessary to create a finite elements grid that has been defined with density of 5 mm. In relation to the fact that the soft foam (on which the gear was positioned during experimental analysis) simulates unconstrained body, the same conditions were setup at numerical analysis. Consequently, it was necessary to define the frequency range in which the analysis was carried out.

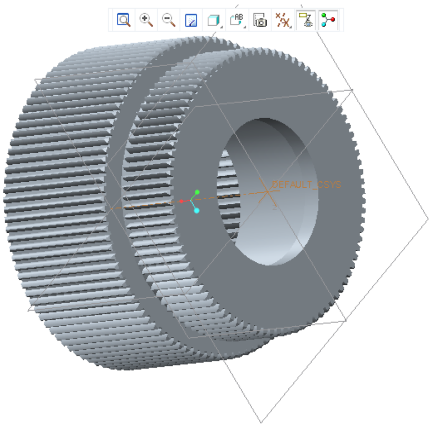


Figure 4. Virtual model for numerical analysis created in PTC Creo

5 RESULTS AND DISCUSSION

The sampling frequency was setup at 6400 Hz with the accuracy 2 Hz. Final FRF is presented in the Figure 5. The natural frequencies of the gear were defined by means of the RFP (Rational Fraction Polynomial) method on the base of the measured data. They are listed in the Table 1, where the values of frequencies achieved by means of FEM method in software PTC Creo are also presented.

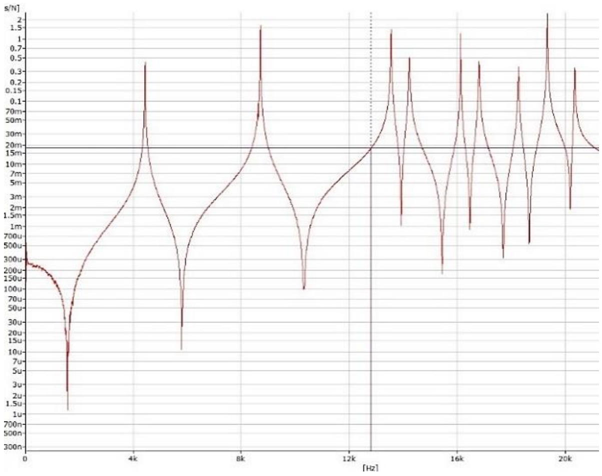


Figure 5. Final FRF function

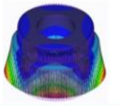
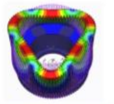
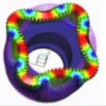
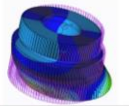
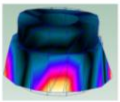
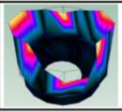
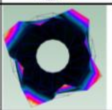
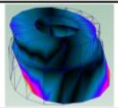
Table 1. Acquired values of the natural frequencies

Natural frequencies	*f*1 [Hz]	*f*2 [Hz]	*f*3 [Hz]	*f*4 [Hz]
Numerical method	4207,6	8 109,0	12535,3	13640,4
Experimental method	4 289,3	8 212,2	12 751,4	13 861,3

Moreover, the first four mode shapes, that correspond to the natural first four frequencies, were evaluated within the both methods of modal analysis. They are shown in the

Table 2.

Table 2. The first four natural shapes of gear wheel obtained by numerical and experimental modal analysis

Natural shape	1st	2nd	3rd	4th
Numerical method				
Experimental method				

From the Table 1 and Table 2 it is clear that the data is comparable, so the boundary conditions and other settings were defined correctly. The differences in frequencies between the values 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. While at the evaluation of experimental analysis, the grid of finite elements was row; it means that the elements were several times greater in comparison with numerical modal analysis in software PTC Creo.

The conformity of the achieved results also presents the Figure 6, where the first eleven natural frequencies obtained by both method are compared.

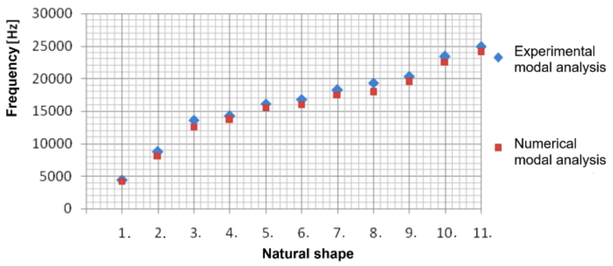


Figure 6. Comparison of the dependencies of natural frequencies measured at individual natural modes

Natural frequencies were also compared with the teeth frequencies of the gear *fg*. There are three various numbers of teeth at the gear, so the teeth frequencies fg_2, fg_3, fg_4 were computed on the base of rotational frequency of shaft *fs* and number of teeth of the gear n_2, n_3, n_4:

$$fg_2 = fs \ x \ n_2 = 17{,}08\,Hz\,x\,72 = 1229{,}76\,Hz \tag{3}$$

$$fg_3 = fs \ x \ n_3 = 17{,}08\,Hz\,x\,100 = 1708\,Hz \tag{4}$$

$$fg_4 = fs \ x \ n_4 = 17{,}08\,Hz\,x\,86 = 1468{,}88\,Hz \tag{5}$$

It is clear that the highest calculated teeth frequency is 1708 Hz and it is sufficiently away from lowest measured natural frequency 4 207,6 Hz.

6 CONCLUSION

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. Today there are many scientific papers, studies and publications on the use of modal analysis and also thanks to the possibility of purchasing the necessary software with devices and undoubtedly good facilities of laboratories.

Along with the development of modern computer technology, experimental modal analysis has become the main tool for solving complex structural vibration problems. For an existing engineering structure, it provides vital information on its dynamic behaviour, thus permitting intelligent solutions to vibration problems the structure may be experiencing. 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 perform.

ACKNOWLEDGMENT

The paper was published thanks to the grants VEGA 1/0614/15; KEGA 087TUKE-4/2015; KEGA 013TUKE-4/2014 with direct support of Ministry of Education of the Slovak Republic.

REFERENCES

[Baron 2016] Baron, P. et al. Proposal of the knowledge application environment of calculating operational parameters for conventional machining technology, Key Engineering Materials, Volume 669: (2016) pp. 95-102

[Bilosova 2011] Bilosova, A. Experimental modal analysis. VSB TU Ostrava, 2011. (in Czech)

[Cacko 2014] Cacko, P., Krenicky, T., Dobransky, J. Impact of an excessive wear of bearing on the mechatronic devices. In: Applied Mechanics and Materials. Vol. 460 (2014), pp. 99-106. ISSN 1660-9336

[Frankovsky 2011] Frankovsky, P. et al. Use of modal analysis in the diagnosis of the vibration of machinery, 2011 (in Slovak)

[Hloch 2008] Hloch, S. et al. Experimental study of surface topography created by abrasive waterjet cutting, Strojarstvo. Volume 49(4): (2008) pp. 303-309

[Hosnedl 2014] Hosnedl S. Theory based management and software support of property driven designing of technical products, MM Science Journal, Vol. 10, (2014) pp. 496 – 502, DOI: 10.17973/MMSJ.2014_10_201413

[Krehel 2016] Krehel, R., et al. Diagnostic analysis of cutting tools using a temperature sensor. Key Engineering Materials, Volume 669: (2016) pp. 382-390

[Krolczyk 2014] Krolczyk, G. M. et al. Influence of technological cutting parameters on surface texture of austenitic stainless steel, Applied Mechanics and Materials, Volume 693: (2014) pp. 430 - 435

[Malotova 2016] Malotova, S., Cep, R., Cepova, L. et al. Roughness Evaluation of the Machined Surface at Interrupted Cutting Process In: Manufacturing Technology, Volume 16, Issue 1, (2016) pp. 168-173

[Metelski 2015] Metelski, A. et al. Taguchi design of experiment versus dynamic programming approach in the optimization of turning process, Applied Mechanics and Materials, ISSN 1662-7482, Vol. 808, (2015) pp 66-71

[Milacek 1992] Milacek, S. Modal analysis of mechanical vibrations.CVUT Praha, Czech Republic, (1992) ISBN 80-0100-872X

[Panda 2014] Panda, A. et al. Progressive technology diagnostics and factors affecting machinability, Applied Mechanics and Materials, Volume 616: (2013) pp. 183-190

[Petru 2013] Petru, J. et al. Influence of cutting parameters on heat-affected zone In: Technical Gazette Vol. 20/ 2, pp. 225-230

[Stojadinovic 2014] Stojadinovic, S. M., Majstorovic, V. D. Developing engineering ontology for domain coordinate metrology, In. FME Transactions, vol. 42(3), (2014) pp. 249-255

[Trebuna 2012] Trebuna, F., Simcak, F., Hunady, R. Vibration and modal analysis of mechanical systems, Kosice, (2012), ISBN 978-80-442-1206-4 (in Slovak)

[Ungureanu 2016] Ungureanu, M. et al. Innovation and technology transfer for business development, Procedia Engineering, Volume 149: (2016) pp. 495-500

CONTACTS:

Eng. MSc. Andrea Cizikova, PhD.
Assoc. Prof. Eng. MSc. Katarina Monkova, PhD.
Technical University of Kosice
Faculty of Manufacturing Technologies
Department of Technological Systems Design
Sturova 31, 080 01, Presov, Slovak Republic
e-mail: a.cizikova87@gmail.com
e-mail: katarina.monkova@tuke.sk

Eng. MSc. Jaromir Markovic, PhD.
Slovak Metrological Society
Hviezdoslavova 1124/31
97401 Banska Bystrica, Slovak Republic
e-mail: slm@slm.sk

Eng. MSc. Jakub Cienciala
Technical University of Ostrava
Faculty of Mechanical Engineering
Department Applied Mechanics
17. listopadu 15, 708 55 Ostrava - Poruba, Czech Republic
e-mail: jakub.cienciala@vsb.cz