Modern Approaches in the Design of Measuring Equipment

Lukas Hruzik(✉), Jiri Struz, Jiří Havlík, and Květoslav Kaláb

Department of Machine Parts and Mechanism, Faculty of Mechanical Engineering, VSB – Technical university of Ostrava, 17. Listopadu 15/2172, CZ 708 33 Ostrava, Czech Republic

lukas.hruzik@vsb.cz

Abstract. This paper describes an advanced design procedure used in the design of two test devices. The paper describes the use of advanced numerical modelling methods applying FEM. Emphasis is placed on the accurate determination of the natural frequencies and the natural shapes of the test equipment frames. Also, the operating frequencies of the partial working mechanisms were evaluated and then compared with the natural frequencies of the designed frames. The paper further describes the use of the data obtained from the FEM in the design of the test equipment itself. The paper compares two approaches to frame design not only for test equipment.

Keywords: Measuring device type 1 · Measuring device type 2 · natural frequency · measurement area · tooth frequency

1 Introduction

When designing not only test equipment, two approaches can be applied:

- The drive part with the motor is a part of the frame;
- The drive part with the motor is not part of the frame of the test device, the measurement takes place outside the drive frame.

The second option is an interesting alternative to the first solution from several points of view.

- Possibility of different design of the drive frame and the measuring device frame with respect to different loads on both frames;
- Minimizing the negative effects of the drive on the measuring device (the devices are connected by a very flexible shaft);
- Economic aspects - two smaller frames are more advantageous in terms of price, especially in terms of production.

The undeniable disadvantages of this solution include the following challenges:

- Larger building dimensions;

J. Homišin et al. (Eds.): ICMD 2020, AHE 17, pp. 266–275, 2023.
https://doi.org/10.2991/978-94-6463-182-1_28

- The need for precise alignment of the two frames in relation to each other;
- The need to eliminate the transmission of negative effects of the propulsion device through the anchorage to the floor.

Despite the above disadvantages, it is the second type of design that is used in the construction of a new test device for measuring the engagement parameters of gears, as shown in Fig. 2. The design of the test device of the first type used to measure the effect of the length of the cardan shaft on the manifested vibrations, can be seen in Fig. 1.

From the point of view of measurement accuracy, it is very important that the frame does not resonate at the measurement frequency. The most important frequency is the engine RPM, so when measuring, it is advantageous to use the area below the first natural frequency of the test equipment frame. Hence there is a need to design the measuring device so that the first natural frequency is well above the frequencies at which the measurements are made. Another relatively significant frequency is the frequency at twice the engine RPM. To ensure that the results from these devices are not affected by other errors, it is also necessary to know the natural frequencies of the sub-mechanisms

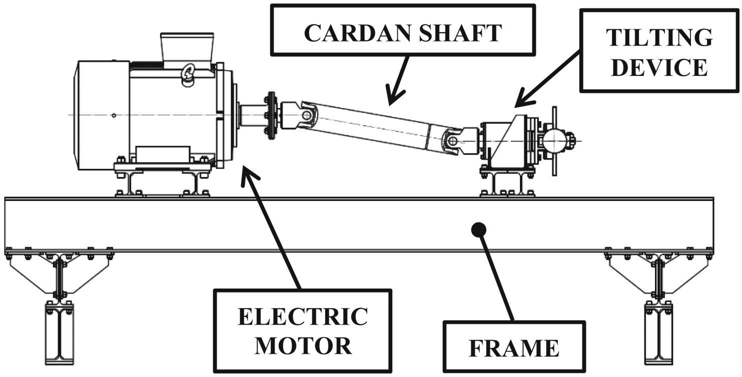

Fig. 1. Scheme of the measuring device of the first type

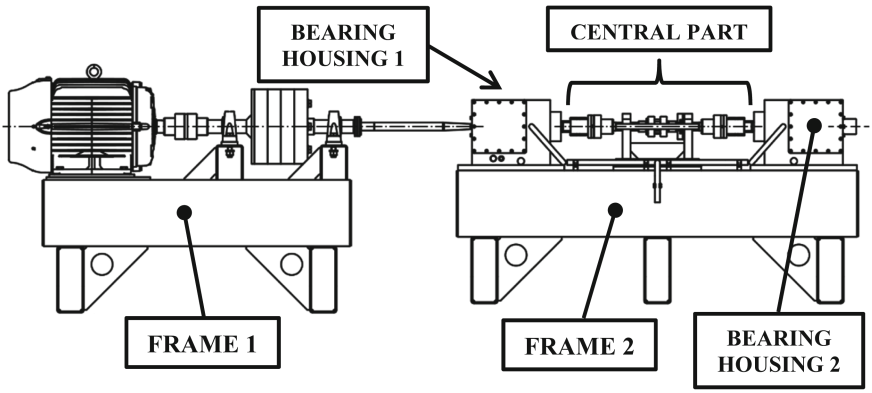

Fig. 2. Scheme of the measuring device of the second type.

occurring in the device, e.g. gears, unbalances, etc. According to [1–3] or [4] at engine speed frequency, the causes of vibration can usually be expected as mass unbalance, parallel misalignment, angular misalignment, bearing misalignment. At the second speed frequency vibrations are usually caused by parallel misalignment, angular misalignment, bearing misalignment. Vibrations caused by electrical faults can be expected at motor speeds or at twice the mains frequency. Electrical faults can include [1, 3].

- Broken rotor bar;
- rotor bar passing frequency;
- twice line frequency;
- static eccentricity;
- dynamic eccentricity;
- loose sheet.

The algorithms for calculating the tooth frequencies that were required in the design of the second type of frame are described in detail in [5, 6]. The procedures described in [5, 6] lead to calculations of tooth frequencies, but in any case, it is necessary to keep in mind the effects of the whole gearing, for example [7].

2 Materials and Method

The frame was subjected to modal analysis in FEM. For computational reasons, the frame was made up of shell elements with an assigned thickness. Electric motors, gearboxes and other working/measuring mechanisms were replaced by mass points defined in the element centres of gravity. In the first type of frame, the fixed bond was defined at the holes for mounting the frame to the floor; in the second type of frame, the fixed bond was defined at the surfaces of the footings that touch the floor. The analysis includes gravitational effects (Figs. 3 and 4).

The results of the modal analyses are presented in Table 1. In the case of the first type of frame, the eigenmodes on the first three frequencies are unsatisfactory. These

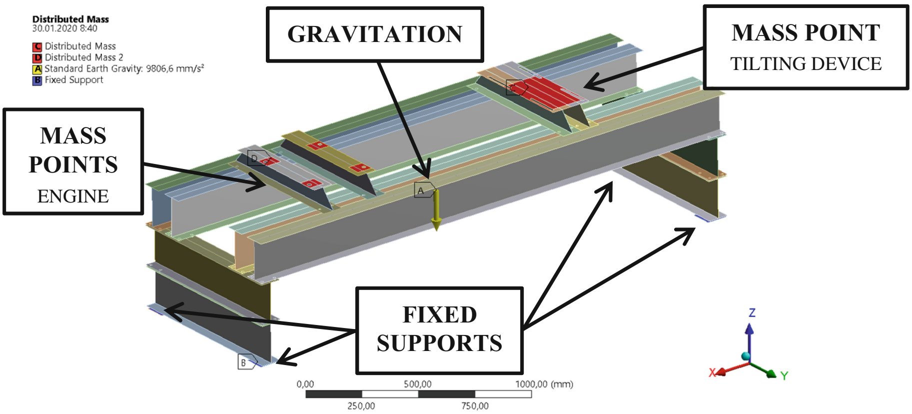

Fig. 3. FEM model of the measuring device of the first type.

frequencies interfere with the measured area and need to be shifted out of the measured area by further adjustment. In the case of the second type of frame, the values of natural frequencies are unsatisfactory, and the frame needs to be further modified.

Based on the results of the modal analyses, the frame designs of the measuring devices were modified. The frame of the first type was reinforced with steel welds and the distance between the support parts (legs) was changed. For the second type of frame, a different frame concept was adopted, mainly for economic reasons. The resulting frame is heavier, cheaper to manufacture and, surprisingly, has better natural frequency values than sheet metal frames.

The same procedures were used to perform the analysis as in the previous case. Only in the case of the second type of frame, the top plates were modelled as 3D components and bonded to the frame using bonded contact only at the edges of the plates to replace the corner weld as accurately as possible (Figs. 5 and 6).

The optimized solutions show better eigenfrequency values, the eigenfrequency values on first three frequencies are shown on Fig. 7.

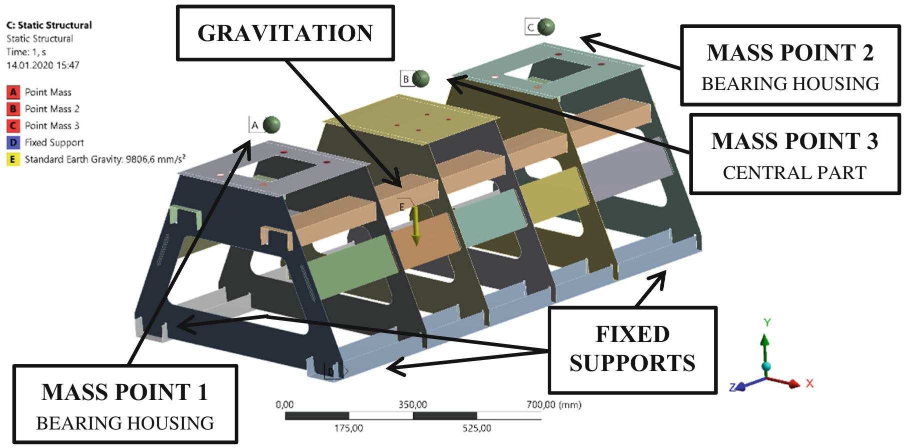

Fig. 4. FEM model of the measuring device of the second type.

Table 1. The natural frequency of the first version of the frames.

Natural frequency	Type 1 [Hz]	Type 2 [Hz]
1.	6,10	51,9
2.	21,8	65,8
3.	38,3	82,8
4.	40,4	101,9
5.	46,5	114,2
6.	62,0	120,5

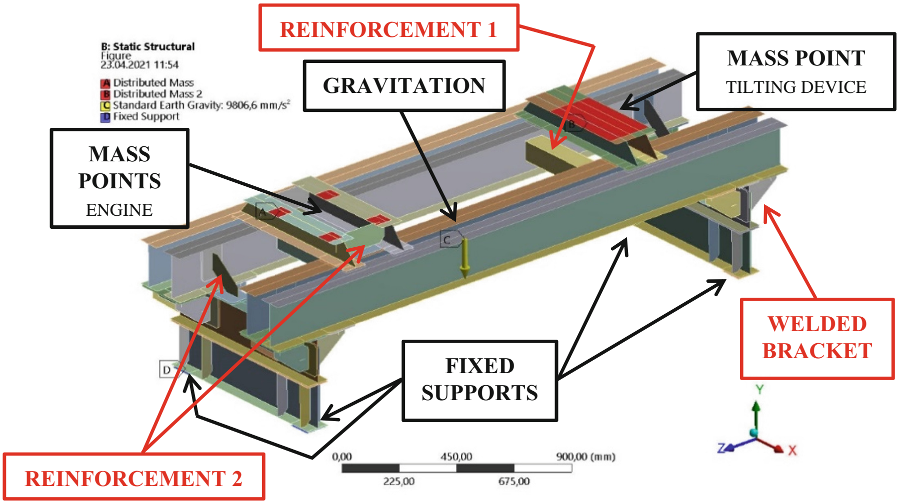

Fig. 5. FEM model of the new measuring device of the first type.

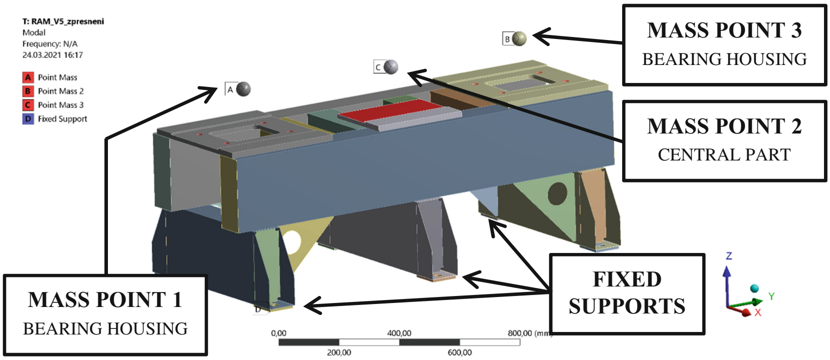

Fig. 6. FEM model of the new measuring device of the second type.

3 Result and Discussing

In the case of the first type of frame, the search area was ideal for measurement based on the modal analysis data. In essence, a graph was created, in which the natural frequencies determined by FEM were plotted and interleaved with the moment characteristics of the electric motor obtained from [8]. On Fig. 8 the following graph is indicated for the first type of frame before modification and reinforcement. The graph shows that the third, fourth and fifth natural frequencies of the frame are close together and measurements in this region are not an optimal. There is a relatively large measurement space between the first and second, respectively second and third natural frequencies, or even the fifth and sixth natural frequencies. The natural frequencies are spread over the entire speed spectrum of the electric motor, which is highly disadvantageous. The spectrum of the modified frame is indicated on Fig. 9.

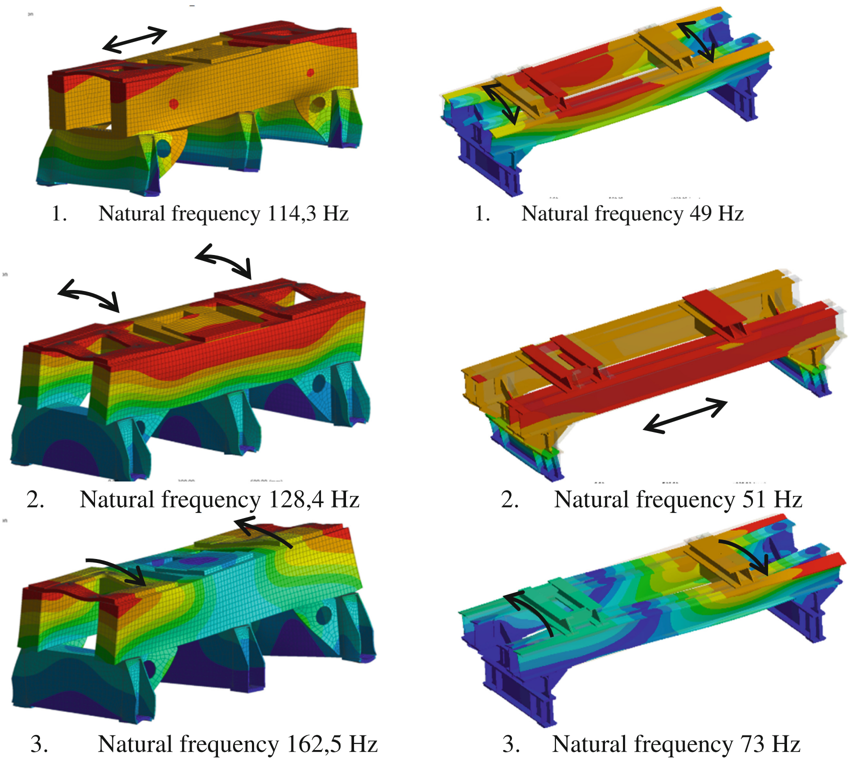

Fig. 7. Natural frequencies of modified measuring device frames.

In the case of the second type of frame the data obtained from the modal analyses were plotted together with the gear meshing frequencies. In addition, the harmonic frequencies of the engine RPM are plotted on Fig. 10. In the case of the second type of frame, a linear relationship between tooth frequencies and motor RPM was established. The graph shows that the first natural frequency of the frame is significantly higher than the engine RPM. In addition, the second harmonic RPM of the motor, which is located between the fifth and sixth natural frequencies of the frame, was checked. The graph also shows that when a third stage gearing is used, the gearing frequency will be the same as the third natural frequency of the frame. The advantage of the graphical representation is that the areas suitable for measurement can be immediately and relatively easily restored, thus minimizing the errors caused by the vibrations of the test equipment frames themselves.

From Fig. 9 by moving the first natural frequency above the speed spectrum as high as possible, the area suitable for measurement is increased a little influence on the measured results can be expected.

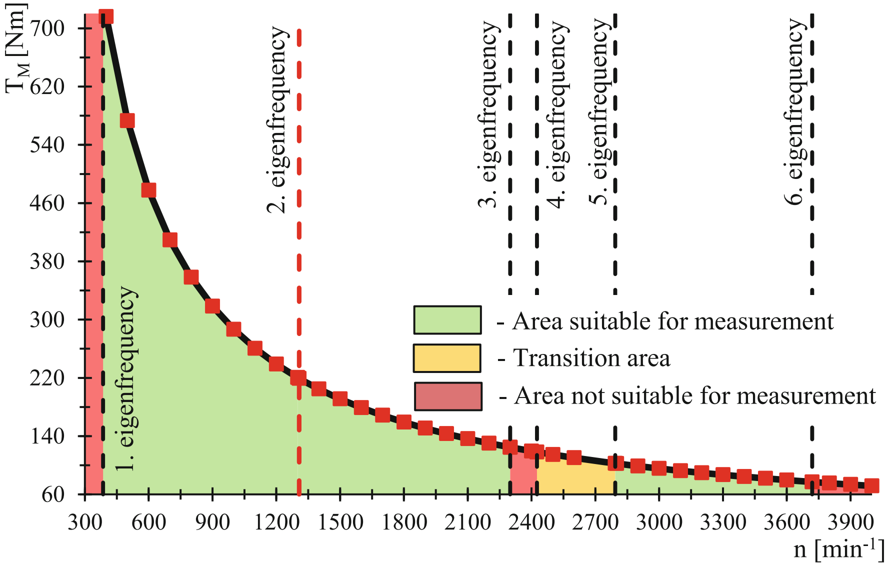

Fig. 8. Determination of the ideal area for measuring the second type of frame before modification.

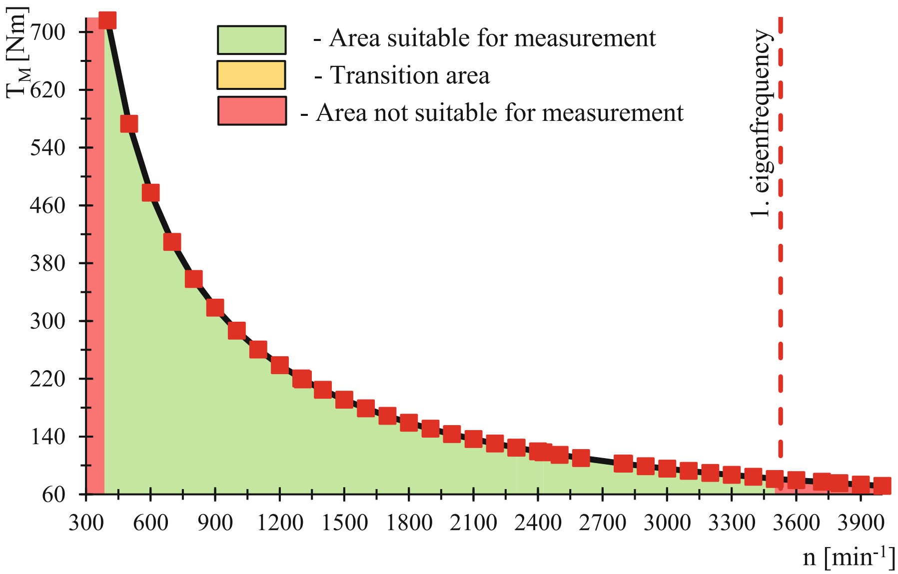

Fig. 9. Determination of the ideal area for measuring the second type of frame after adjustment.

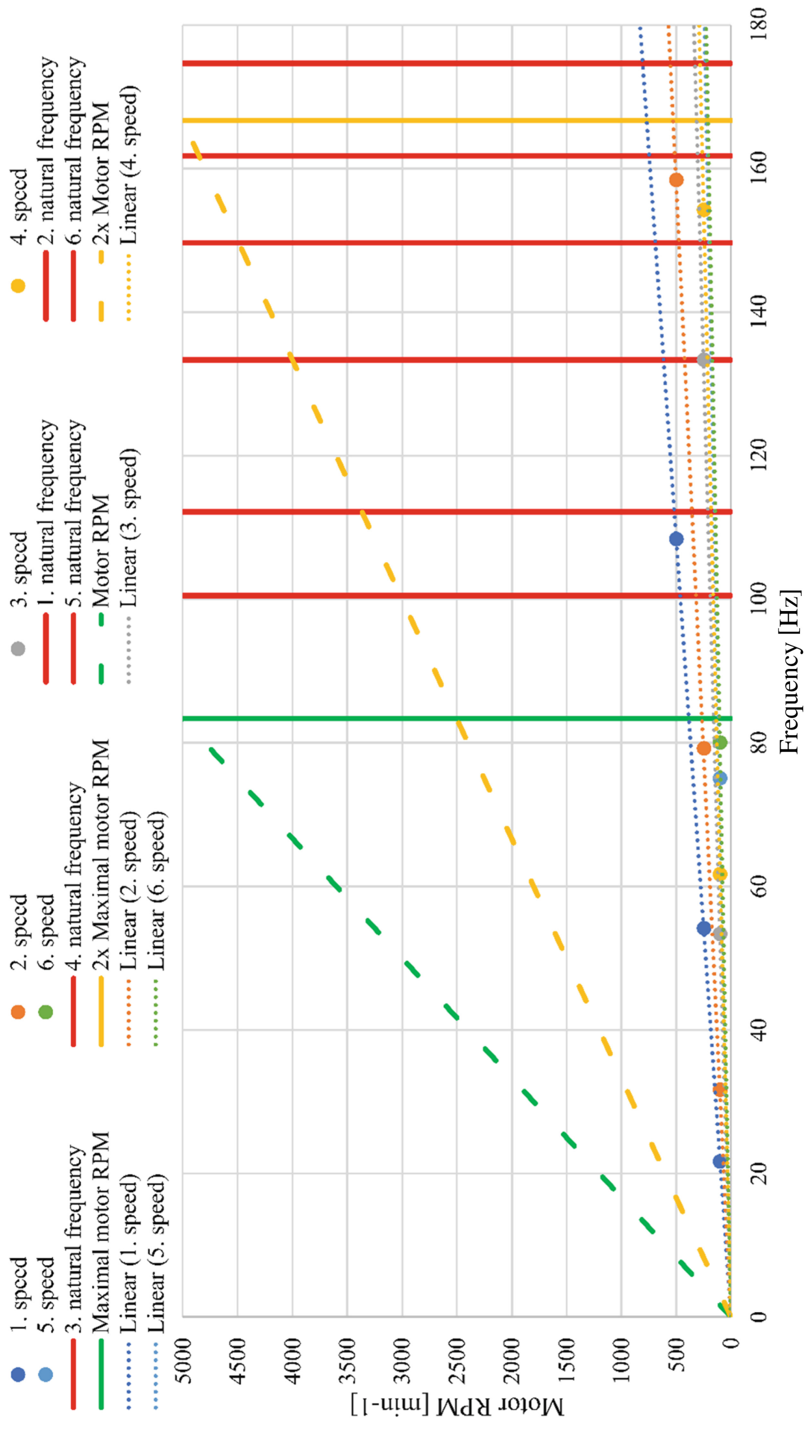

Fig. 10. Measurement areas approximated, for tooth frequency

4 Conclusion

The methodology described in this paper uses a modern approach in the design of the measurement frame, modal analyses and work with the acquired data. This method is often neglected in practice because of the mistaken belief that the computational procedure is time-consuming. However, thanks to newly advanced computational methods and tools, the necessary data can be obtained in a relatively short time with satisfactory predictive value. The paper continues to describe a frame modification procedure that cannot be generalized to all possible cases, but the procedure implies the necessity of shifting the natural frequencies out of the working frequencies. Additionally, two frame design approaches are compared, where in the first type of frame the drive motor and measuring system are mounted on one frame, and in the second type of frame the motor and measuring system are mounted on two individual frames.

Acknowledgement. The research presented in this paper is an outcome of the project SP2021/31 "Experimental and computational methods for dimensioning of machine components 2021" funded by the Czech Ministry of Education, Youth and Sports.

References

1. BILOŠ, Jan a Alena BILOŠOVÁ. Aplikovaný mechanik jako součást týmů konstruktérů a vývojářů: studijní opora. Ostrava: Vysoká škola báňská - Technická univerzita Ostrava, Fakulta strojní, 2012. ISBN 978–80–248–2755–1.
2. Berry, M., SKF cz a.s.: Ilustrovaná tabulka pro vibrodiagnostiku dle Berryho. Praha: www.skf.cz
3. ÁGOSTON, Katalin. Fault Detection of the Electrical Motors Based on Vibration Analysis. Procedia Technology. 2015, (19), 547–553. Dostupné z: 10.1016/j. protcy.2015.02.078
4. KILIÇ, Recep. Determination of Imbalance Problem in Electric Motor and Centrifugal Pump by Vibration Analysis. Acta Physica Polonica Series A. 2016, (130), 487–491. Dostupné z: https://doi.org/10.12693/APhysPolA.130.487
5. TŮMA, Jiří. Zpracování signálů získaných z mechanických systémů užitím FFT. Praha: Sdělovací technika, 1997. ISBN 80–901936–1–7.
6. TŮMA, Jiří. Vehicle Gearbox Noise and Vibration. První. Chichester, United Kingdom: WILEY, 2014. ISBN 978–1–118–35941–9
7. ULLAH, Najeeb, C. XI, T. CONG a H. YUCHENG. Rotordynamics analysis of asingle helical gear transmission system for high speed applications. JOURNAL OF MECHANICAL ENGINEERING AND SCIENCES (JMES). 2020, **14**(3), 7040–7048. ISSN 2289–4659. Dostupné z: https://doi.org/10.15282/jmes.14.3.2020.06.0551
8. HAVLÍK, Jiří. Analysis of machine parts in critical stresses. SP2017/39. Technical University of Ostrava, 2017. [04–10–2021].