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Using frequency and modal analysis to attenuate low frequency waves

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ABSTRACT

The paper analyzes the response of structures and machinery to low frequency vibration (LFV) such as low frequency waves (industrial seismic waves) generated by different machines, technological processes, etc. The principles of the analysis can be applied for different kinds of excitation (stationary – deterministic, random and non stationary – continuous and variable). Some sources, as a result of human activity, for LFV and their subsequent low frequency waves (LFW) were investigated and analyzed. The goal was to determine the Eigen frequencies and Eigen modes of the measured structure and/or machinery not only by identifying sources of vibration but also applying operational modal analysis which is important in designing suitable modal damping. The operational frequency spectrum with Eigen modes of the structure and/or machinery is compared by means of frequency and modal analyses. Through modal damping, levels of low frequency noise may follow, and the transmitted LFV can be the reason why resonance or excitation of the Eigen modes occur. In the conclusions, the measured results of the LFW caused by a vehicles combustion engine is analysed as well as its effect on the cabin of the vehicle. Similarly electromotor resonance is investigated as well. The results of the frequency analysis, and the resulting measurements show the most effective way to attenuate high energy LFW.

Keywords: Seismic waves, Response, Attenuation I-INCE Classification of Subjects Number(s): 42, 49.2; 75.6

1. INTRODUCTION

In general when analysing increased dynamic loading of machinery, machine systems and structures, it is advantageous to apply frequency analysis and modal analysis which can help in diagnosing the effects causing increased dynamic loading (vibration and noise) (1, 3, 4). This is especially true when dealing with mechanical systems and structures in sensitive areas such as nuclear power plants. Most commonly they come from various support structures, mechanical systems, industrial structures and structural elements as well as other components that are effected by seismicity in a nuclear power plant. When analysing increased dynamic effects, vibro-acoustic diagnostic methods, based on processing the vibrating signal and noise emitted from mechanical, electromechanical systems and structures in and surrounding nuclear power plants (9, 14, 15, 16). Based on measuring principles, the signal is always measured at a defined point in the analysed mechanical system or structure which is exposed to vibration. However, this procedure does not always achieve the desired results (detection of maximum amplitudes in the mechanical system and or structure) and therefore modal analysis is essential to include when evaluating the safety of various components in nuclear power plants.

The frequency analysis processes the signal from one or more predefined measurement points of the investigated system or structure. Diagnosing structural nodes which are in close proximity to the measurement points returns information about the dynamic loading of that particular node, however it does not provide enough information about the system as a whole. However, when investigating the frequency spectrum at multiple measurement points may obtain more information, however the dependence between each isolated signal and their frequency parameters may not be clear. These dependencies are defined by determining the operational modes of the vibrating mechanical system. this method is based on multi-channel frequency analysis which uses a computer to animate the

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shape of the vibrating components. It also gives information about the frequency spectra, phase angle etc. at each measurement point from the measured signals (1, 6, 11).

The frequency analysis and operational mode shapes are mutually dependants, where the frequency analysis is used as a tool to determine the operational mode shape of the vibrating component. It is also the source from which the mode shapes are animated respecting mutual relations. It is necessary to comment that the operational mode shapes do not provide a thorough understanding about the dynamic response of the system to the excitation load. The animations are only indicative of the measured points at predefined frequencies. The reliability and effectiveness of the operational mode shapes and frequency analysis are a function of the operators experience.

Modal analysis observes three basic effects, to determine the natural frequencies, damping at these frequencies, and the mode shapes.

The application of mathematical relations to the time dependant signal of the dynamic event, vibration, impact, stress, deformation, etc. is a tool that the frequency analysis uses.

Modal analysis uses the frequency transfer functions (usually acceleration) on a pair of points in the given system. The frequency transfer function for a given part is constant and is given by the ratio between response and excitation. In order for modal analysis to fulfil its purpose it must use the spectra of input variables in complex form. Frequency analysis is therefore a tool that modal analysis can use to evaluate the response of the mechanical system or structure.

2. GOALS, INSTRUMENTATION AND METHODOLOGY

2.1 Goal of the Study

The goal of the study is to investigate and analyze the response of a deterministic excitation on a subtle (2D) structure and on a construction of a rotating machine (3D mechanical system) which is used in nuclear power station. The study shows that the low frequency excitation wave can cause resonance state at higher frequencies which can be dangerous for the dynamic stability of a mechanical and electro technical system, structure, etc and/or these systems can be very noisy in respect to their surroundings. Frequency spectrum of the measured signal and Eigen modes was compared and analysed for the investigated objects.

2.2 Instrumentation and Methodology

Three deterministic excitation signals were measured. The sources where generated by an internal combustion engine, electromotor rotor and the consequent response on the electromotor framework was performed using the frequency analyzer PULSE, Dyn-X, FFT, M1 3560-B-X10 Bruel&Kjaer platform. This portable analyzer represents the system, which guarantees reliable measurement processes, analysis, and evaluation. The system consists of a piezoelectric accelerometer with a frequency range from 1 Hz to 10 kHz (amplitude ±10 %), hammer, display and memory module. To identify the energy dominant Eigen frequencies more precisely, the fast Fourier transform (FFT) analysis were carried out using the frequency analyzer PULSE. The methodology presented in the article can be applied also for other excitation sources of low frequency vibration. Sensor mounting on the structural elements of the investigated objects must coincide with the ISO 5348 for accelerometers and with respect to past experience (2). The goal is to ensure that the sensor correctly reproduces the motion of the analysed component without interfering with the response.

The measurement device and its technical parameters must be calibrated before measuring any seismic event. Other than the frequency range, for the type of signal, it is also very important to select the appropriate type of averaging as well and number of averages per unit time as well as a suitable time window (1, 3, 6, 11, 17). When performing the modal analysis the following time windows were used: averaging for ten values of the frequency spectrum, exponential for acceleration measurements, and transient for impact hammer. The frequency analysis for the deterministic signal utilized the Hanning time window. Definition of the measured parameter, such as the vibration acceleration, velocity (strength of vibration) and other performance parameters was performed. The duration of the seismic or low vibration signal is the basis for determining the strength of the vibration, since this energy parameter is used to evaluate the effect of the seismic or dynamic event on the mechanical/structural system. Before and after the measurement of seismic or dynamic events it is beneficial to measure any secondary vibrations from external and/or internal sources of dynamic loading (14, 15).

Throughout the measurements, other variables are recorded that are essential to the frequency analysis, such as the identification of measurement location (points) and their corresponding frequency spectra, and possible unique effects during the response measurements (random impacts and shocks caused by human activities in the surrounding areas). Such unique events, which can impact the correctness of the results, are an essential part of the seismic measurements. Using FFT analysis, the amplitude of the frequency components for the measured dynamic excitation and support structure was determined.

2.3 Use of Weightings

It is important to consider that using incorrect time windows can return incomplete results as can be seen from the Nyquist polar diagram and transfer function of a freely hanging beam (Fig. 1) (11). The freely hanging beam is excited by a modal hammer and therefore generates an impact signal. Utilizing Hanning`s weighting resulted in the destortion of the Nyquist polar diagram and transfer function (Fig. 1-right). The next step utilized exponential weighting, which represents the response character of the signal. The correct Nyquist polar diagram and transfer function are displayed in Figure 1-left. The frequencies of the eigen modes correlate. In order to obtain a non distorted result it is necessary to correctly classify the signal and establish the correct weighting for it (11).

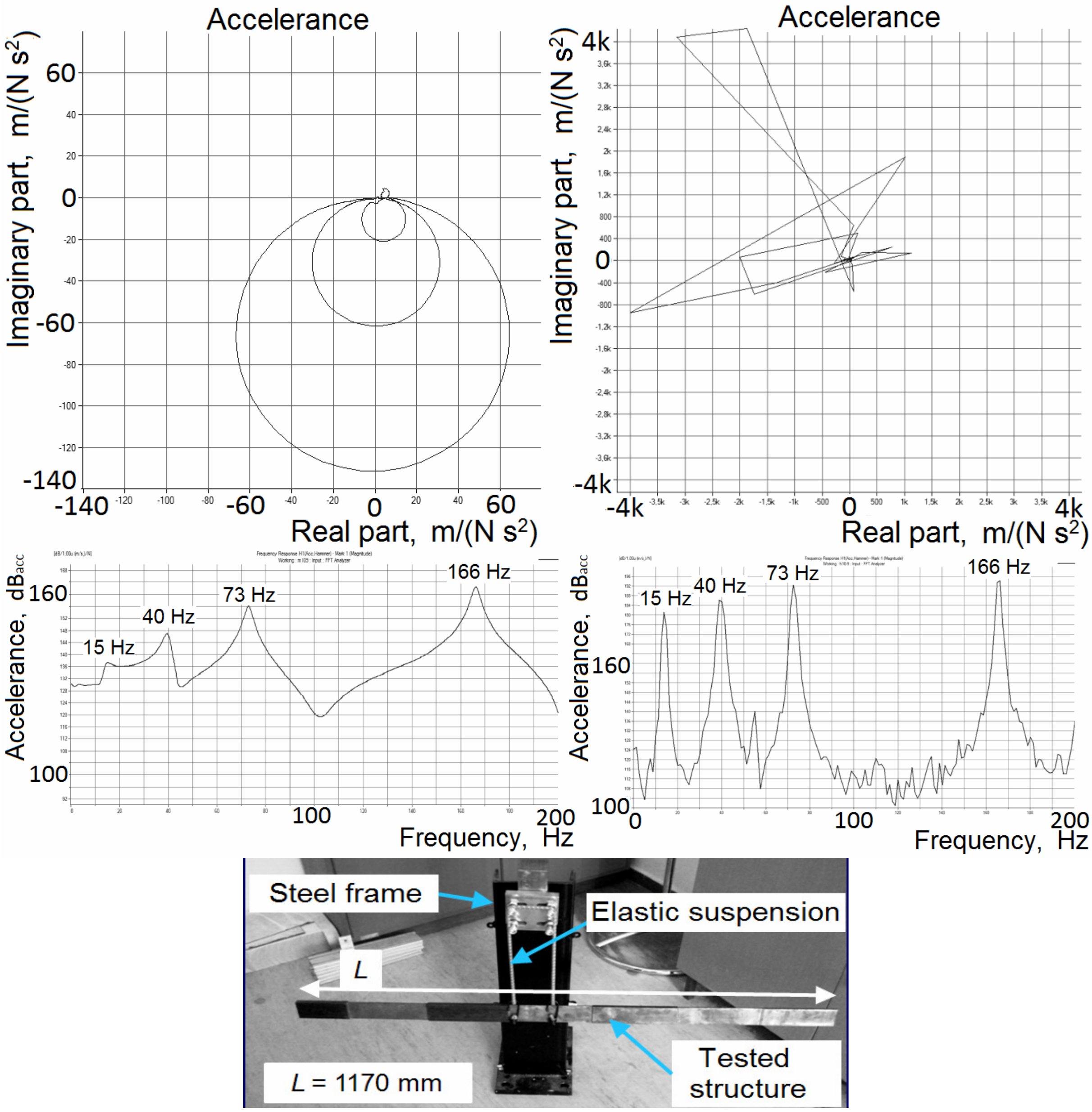


Figure 1 – Nyquist polar diagram and transmission function utilizing the exponential window (left) and Hanning window (right) for impact excitation of a suspended beam (down)

3. APPLICATION OF FREQUENCY AND MODAL ANALYSIS

3.1 Membrane (2D) Structures

Again, it is necessary to emphasize that the results of the experimental modal analysis are based on the mutual connections within the observed component where it is imperative that the linearity of the system be investigated. Linearity is evaluated by means of coherence, which is obtained through processing the measured spectra. It is necessary to emphasize the fact that experimental modal analysis is an incomplete evaluation without an accompanying numerical simulation, mainly in more complex structural systems (1). On the other hand, measurements can facilitate the identification of real structural parameters such as material properties, damping etc. that can be used in the numerical simulations. Therefore it is necessary to consider numerical analysis not as an alternative to measurements but rather as two mutually dependant methods that together provide a complete picture of the problem at hand (11).

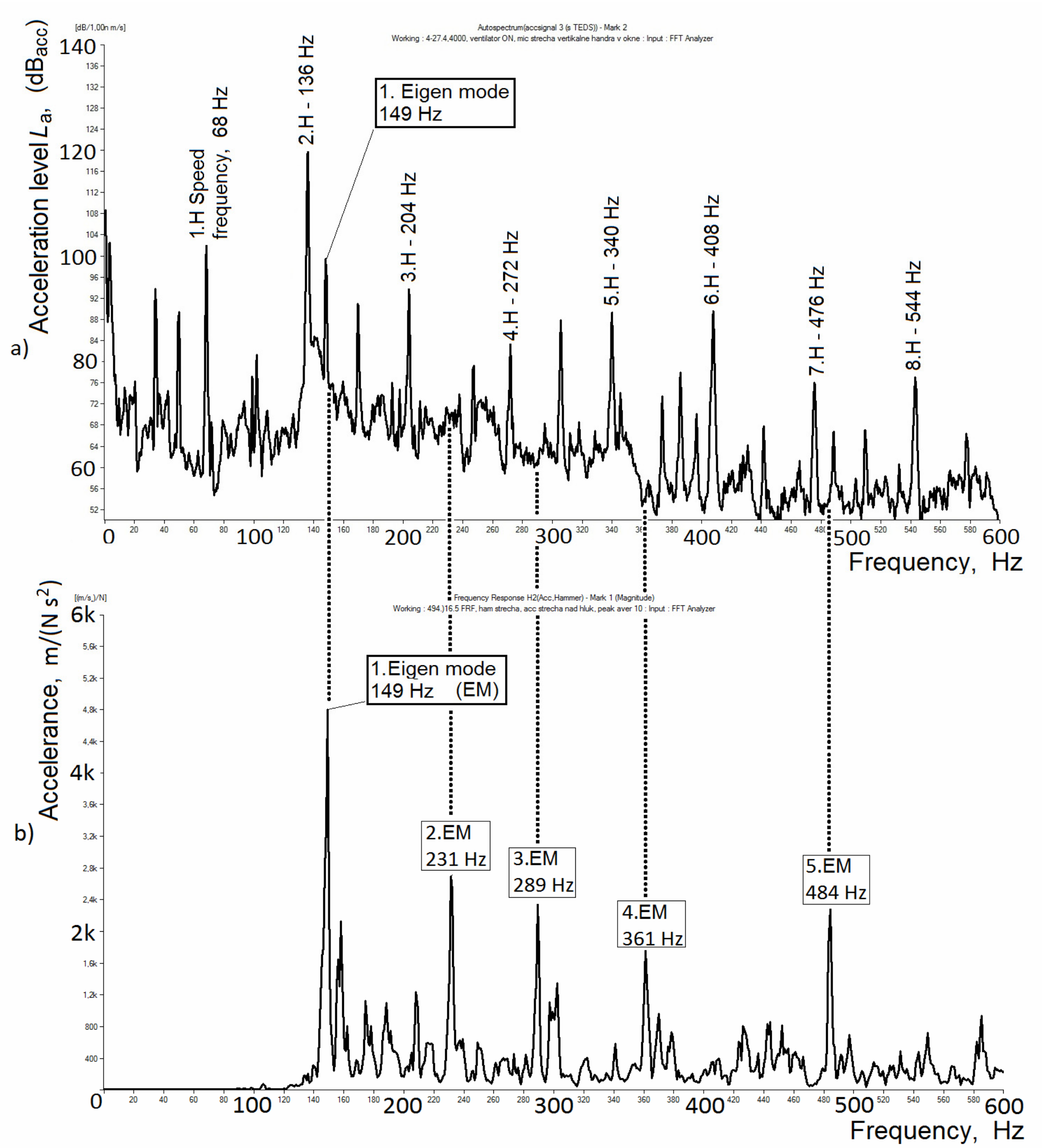


Figure 2 –a) FFT analysis of a subtle structure (car body) at a given rpm of the combustion engine; b) Eigen frequencies the modal analysis of the same subtle structure

Figure 2a shows the frequency analysis of an internal combustion motor and its effect on the cabin of the vehicle. The measurement was performed at defined operational conditions with the goal to determine the response and noise transmitted from the motor through surrounding components (doors, roof, floor, etc.) (3, 4). The motor was a gasoline powered internal combustion engine running at 4000 rpm. As a result, the chassis of the vehicle was excited by a deterministic signal for a defined interval of time. From the analysis of the frequency spectra, a strong oscillating frequency was observed, which generated low frequency acoustic waves within the cabin. In terms of an internal acoustic environment (10, 12) it is desirable to know the relationships between the excitation frequency and the revolutions of the motor and the Eigen frequencies of each individual chassis component. Using a modal hammer for modal analysis obtains the Eigen modes for the roof of the vehicle, whose first five Eigen (natural) frequencies can be seen in Figure 2b. Comparing the excitation frequency spectra of the roof with that of the vehicles motor returns information about any similarities in excitation frequency. In this case, for the defined operational conditions, such similarity doesn't occur which is a positive result in terms of generated acoustic energy within the cabin of the vehicle. If the frequencies where to coincide, unwanted resonance would occur, increasing noise in the cabin of the vehicle. From the frequency spectra, where the basic rotational and its harmonic frequencies change with respect to speeds, assuming that the natural frequencies remain constant, it is possible to determine at which frequencies resonance will occur for each vehicle component. If coinciding frequencies exist in the operational frequencies of the vehicle, it is possible to modify each individual component in order to tune the Eigen frequencies during operation and minimize vibro-accoustic energy within the cabin (9, 13). The discussed methodology can be applied in nuclear power plants provided that certain safety precautions are obeyed (5, 7, 14, 15, 16).

3.2 Rotating (3D) Machine

In practice, high level of dynamic loading are important to investigate during the startup and shutdown phase of any operational machine since they can pass through a resonant state. These undesirable effects can generate low frequency (seismic) waves. One example of such a dynamic effect can be observed on a three phase electromotor during braking where undesirable dynamic loading is generated and the severity of vibration increases as well as noise in the whole system. The measured electromotor displayed unacceptably high levels of vibration at about 540 Hz during shutdown which manifested itself in high levels of noise (Fig. 3). Undesirable increase in amplitude of the rotor occurred at a frequency of 135 Hz (Fig. 3a) (8). The goal of this experiment was to determine the cause of this increased dynamic load (vibration and noise) in the machines structure.

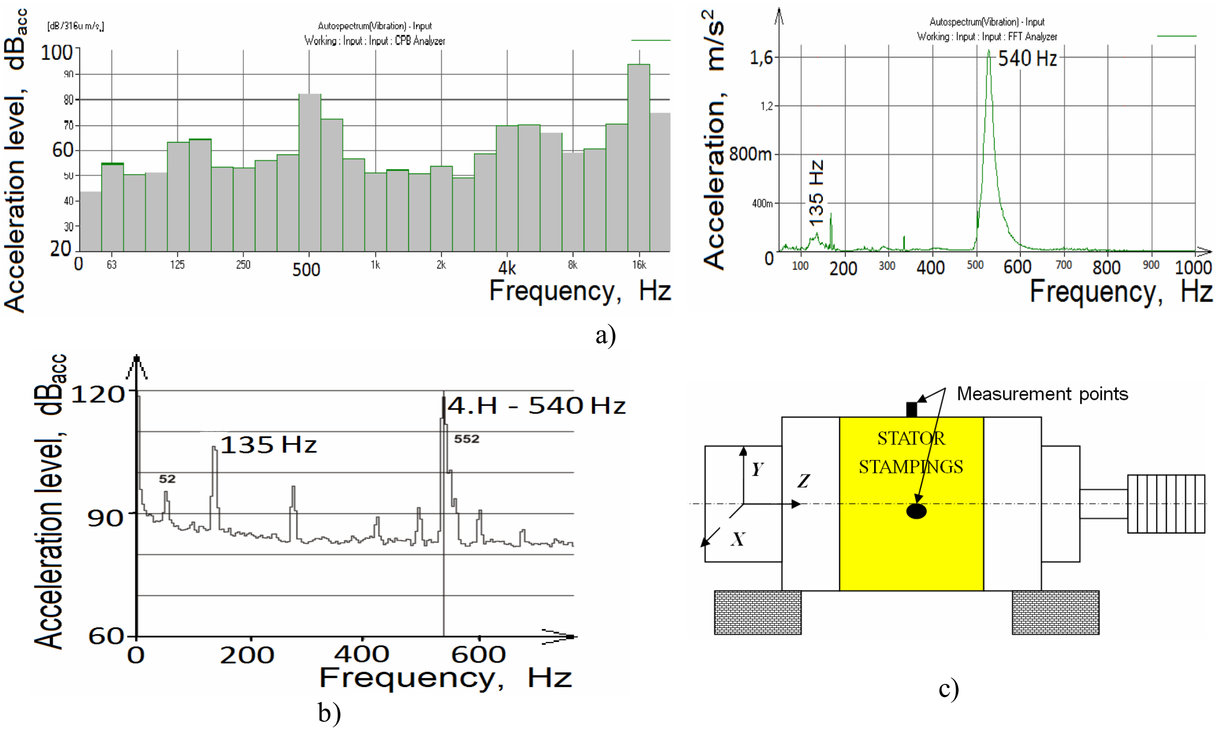


Figure 3 – a) Frequency spectrum and Eigen frequency (right) during freewheeling at specific electromotor revolutions; b) Measured frequency spectrum; c) Scheme of the electromotor

When determining the measurement points on the electromotor it was necessary to consider its structure. The sensor was mounted within the radial plane *X-Y* of the stator (Fig. 3c). After a number of measurements it was clear that for the given configuration, it was enough to measure in the vertical direction of the radial plane. In order to determine the causes of the high dynamic loading during freewheeling, change from 15000 rpm to 5100 rpm it is necessary to detect frequencies response within this range. Primarily, frequencies caused by imbalances in the system, clearances, damaged bearings, rotor or stator damage, etc. as well as the critical frequencies of the rotor. Therefore it is necessary to perform vibrodiagnostics on the electromotor and systematically eliminate possible causes for the increase dynamic loading.

The electromotor itself generated relatively significant periodic vibrations within its operational range, which was found to be caused by some form of imbalance in the rotor (8). Imbalances in the rotor can obviously effects the transfer states such as start-up and shutdown.

Based on the frequency analysis performed in laboratory conditions and at varying operational frequencies, undesirable dynamic effects occurred at 135 Hz within the motor and its support structure (Fig. 3a, b). the analysis showed that the fourth harmonic frequency (4.H) at 540 Hz displayed an unmistakably high amplitude which was confirmed with quality control measurements (see Fig. 3a, b). From the frequency analysis it can be concluded that the vibrational amplitude of the motor as a whole (with a maximum at 540 Hz) decreases as the operational speed of the rotor was increased or decreased from this frequency.

When calculating the critical revolutions of the electromotor rotor (Fig. 4a), the rotor is considered a thin disk the following relation can be expressed (8, 11).

$$n_k = \frac{1}{2\pi}\sqrt{\frac{3EJl}{m\,a_1^2 a_2^2}} \tag{1}$$

where E is the modulus of elasticity (Pa);

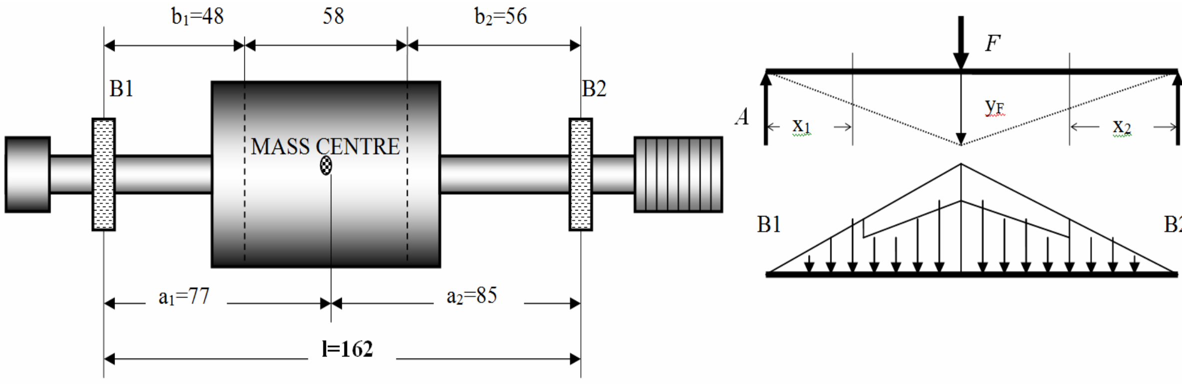
$J = \dfrac{\pi d^4}{64}$ – second axial moment of area (m^4), d – shaft diameter;

m – rotor mass (kg);

a_1, a_2 – distance of the mass centre from the bearings (supports) (m);

l – distance of the bearings (m).

After substituting the given values into the relation, the critical revolutions at the given conditions can be approximated (bending stiffness of the shaft is considered to be constant over its length) to be 20710 rpm. The rotor in question is very stiff amplified by its mounting on the electromotor (see Fig. 4a).



a) b)

Figure 4 – a) Schema of the electromotor rotor; b) Schema for the calculation of shaft stiffness along the axis of rotation

If varying stiffness is considered along the length of the shaft, its stiffness must be calculated utilizing Castigliano and Mohr methods (11). Mohr's method is based on moment (Fig. 4b). deflection be means of Castigliano for the problem in Figure 4b can be expressed as:

$$y_F \equiv v_F = \frac{\partial A}{\partial F} = \frac{F}{3E(a_1 + a_2)^2}\left[\frac{a_1^2 b_2^3 + a_2^2 b_1^3}{J_1} + \frac{a_1^2\left(a_2^3 - b_2^3\right) + a_2^2\left(a_1^3 - b_1^3\right)}{J_2}\right] \quad (2)$$

After inserting given parameters into relation (2), the bending stiffness of the rotor is calculated and substituted into the following

$$\omega_k = \omega_n = \sqrt{k/m} \quad (3)$$

which gives the critical angular velocity (critical revolutions) of the rotor, which was calculated to be approximately 41000 rpm. After comparing the critical revolutions of the rotor, it can be seen that the rotor plates have a significant effect on the stiffness of the shaft, therefor they must be considered when calculating the stiffness and deflection.

The fifth subharmonic critical revolutions are obtained at 8200 rpm and these subharmonic can excite a resonance state in the mechanical system. In the given case, this manifested itself in increased dynamic effects at approximately 8100 rpm (8). This amplitude is independent to the critical revolutions for which an increased dynamic load was measured. This indicated that one possible cause for the increased dynamic loading was the combined imbalances in the rotor, high tolerances or unsuitable mounting of the motor which causes resonance. This is verified by the frequency spectra from the modal analysis (Fig. 5).

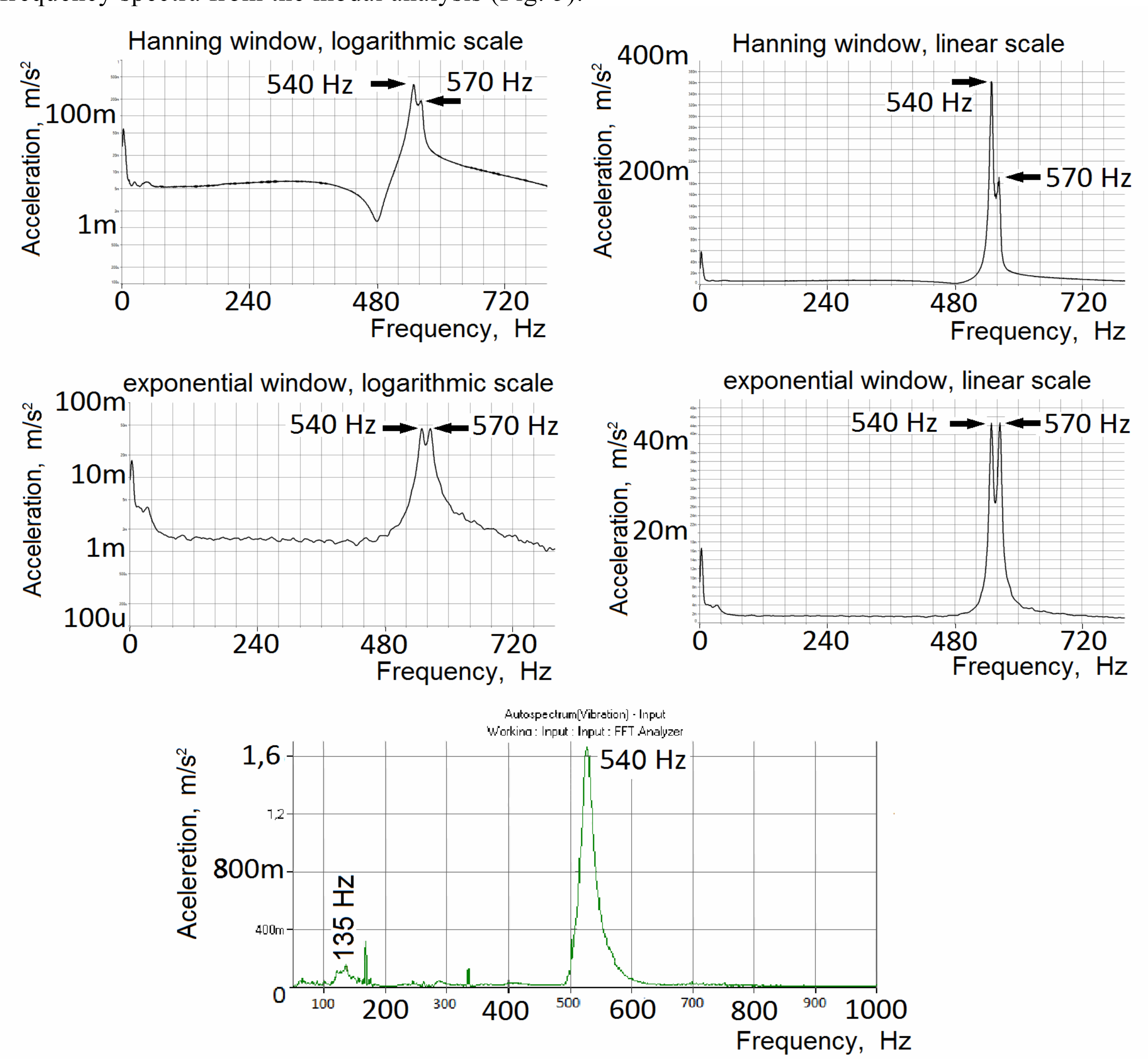


Figure 5 – Eigen frequencies from the modal analysis of the electromotor by using Hanning and exponential window for scale logarithmic (left) and linear (right). Frequency analysis of the electromotor in linear scale (bottom)

Modal analysis was performed with periodic impact excitation on the electromotor using Hanning and exponential windows expressed in logarithmic and linear form. The best results were returned by using the Hanning window (Fig. 5-top). This depends on the character of the generated signal. The figures expresses the frequency correlation between modal and frequency analysis, where the modal analysis simplifies the diagnosis of undesirable vibrations and noise.

It is possible to confirm the aforementioned statements, amplitude of vibration around 540 Hz significantly change their magnitude, which is characteristic for resonance state of a mechanical system. However it is important to emphasize that the resonance of the electromotor and its support structure was excited by low frequency waves. Some more detailed effects on the electromotor are explained in (8).

4. METHODS TO REDUCE OF LOW FREQUENCY WAVES

4.1 Resilient materials

Decreasing the low frequency vibrations, and in turn emitted low frequency noise, of thin (subtle) plates is possible by utilizing vibration resistant layers or composites which increase the damping factor of the plate. Vibration resistant layers significantly reduce the amplitudes of vibration within the resonant frequencies of the structure and may even push them to another frequency Figure 6 (13). Since this method increases the weight of the system, it is preferred to apply them on stationary equipment.

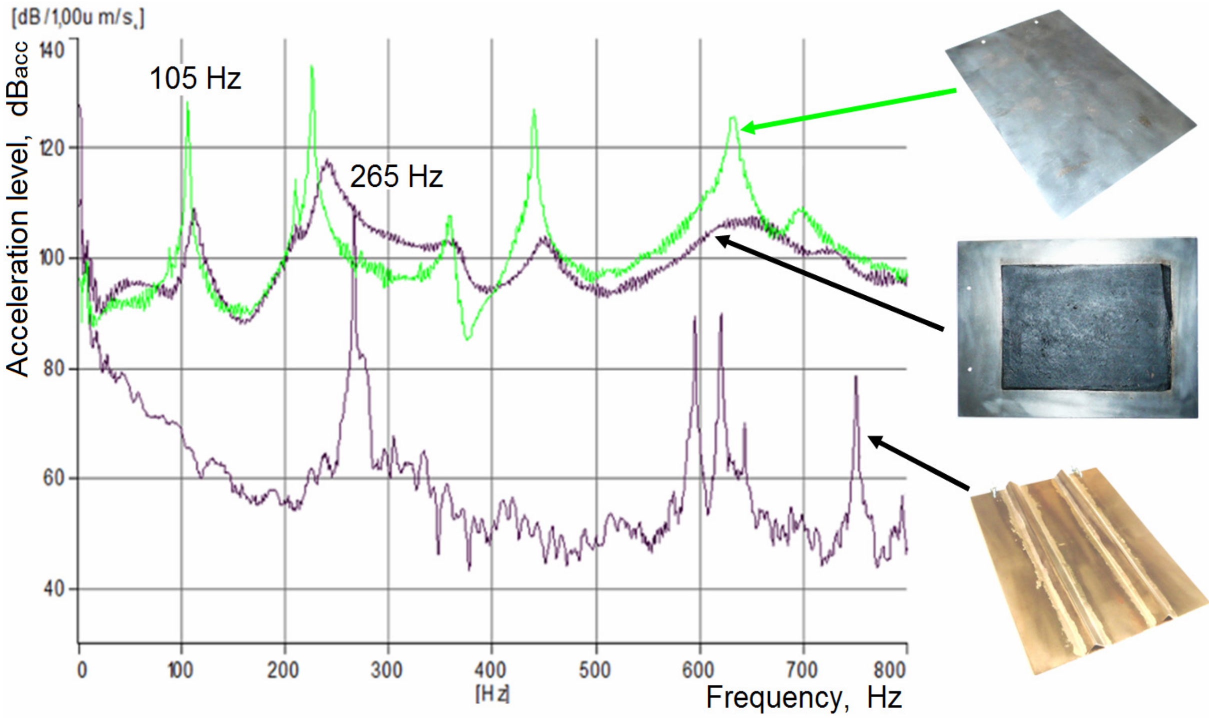


Figure 6 – Comparing the frequency spectrums of a non-damped subtle structure with a damped subtle structure utilizing resilient material (VIBROFOL) and by tuning of the structure

4.2 Structure tuning

Decreasing the vibroacoustic energy of a structure can also be performed by adding stiffening elements. These can sometimes be more difficult to implement but they can contribute to improving the resistance to vibrational effects such as impact that can improve the safety of a structure. For example, in a vehicle crash. In contrast to using vibration resistant layers, this method significantly shifts the Eigen frequencies to higher frequencies that are consequently of lower energy and less destructive character (Fig. 6). Effective elimination of noise, within a vehicle cabin for example, requires the analysis of individual components connected in the system to identify the greatest contribution to low frequency waves.

5. CONCLUSIONS

Utilizing modal analysis allows for relatively quick identification of a structure Eigen (natural) frequencies. In terms of low frequency waves, the first Eigen frequencies are critical when analysing subtle structures. The frequencies of the mode shapes for most machines and structures are generally high and energetically low. However, even these resonances can negatively influence surrounding systems such as electronics, measurement circuits, pipe systems whose safe operation is essential to safety in nuclear power plants. Identifying the mode shapes of a given structure, effective intervention can be applied to reduce the amplitudes of vibration as well as noise. For machines, inadequate mounting or implementation into the system can also lead to the generation of high dynamic loads. Tuning existing systems to push these frequencies reduces emitted vibroacoustic energy (12).

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