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Fakulta strojního inženýrství

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# ANALÝZA AKUSTICKÝCH VLASTNOSTÍ POHONNÝCH JEDNOTEK

# ANALYSIS OF POWERTRAIN ACOUSTIC PROPERTIES

Zkrácená verze PhD Thesis

Obor: Konstrukční a procesní inženýrství

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**Kľúčové slová:**
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# CONTENT

# 1 INTRODUCTION

The issue of computational modelling of noise and vibration of powertrain is the next logical step in the virtualization of the design process. Engine manufactures and also all producers not only in the automotive industry are trying to change the process of development in to the virtual design as much as possible. This is due to reducing design costs and shortening the time needed to put the product to customers.

Technical requirements for powertrains are now many times contradictory. It is expected: high performance, reliability, low power consumption, basic construction and low production costs as well as reduced emissions.

However, we can say that even if these properties are essential, ultimately the decision to buy a car is often overridden by the user comfort. As for the cars, is one of the parameters of comfort an acoustic convenience, sound while driving, which is generated e.g. from joints of plastic interior parts and also an aerodynamic noise. In the case of trucks and tractors is, in addition to the acoustic expressions, evaluated also the magnitude of the vibration. Certain maximum values prescribe various standards of hygiene. Also, impact on health and with this the corresponding quantity and quality of work done is a significant factor in choosing.

In the passenger cars is the primary source of noise and vibration the drive unit, but also it should not be forgotten the source from the wheels, transmissions, final drives and principally all of the subsystems that transmit power from the engine to the wheels.

Recently, it is more and more talking about alternative fuel vehicles, which should replace the conventional combustion engine and thereby potentially eliminate a key source of noise and vibration. It is difficult to predict the future in this field, as yet recently unimaginable car operated independently by a computer is now a reality. However, it is possible to say with certainty that if the source of energy for movement is the internal combustion engine, hydrogen engine, electric motor or other type,; there always will be the transfer of power to the wheels and thus always will occur larger or smaller vibrations or noise.

## 2 EVALUATION OF THE CURRENT STATUS

Baseline for determining the acoustic parameters of the drive unit, it is the determination of vibration of engine surface by which it can be further determined the acoustic pressure at a certain point in the area around the drive unit. Values of velocity at nodal points of finite element 3D model can be obtained from the results of simulations using the MBS system.

There are the methods in the field of vibration and noise, which enable to measure and locate their sources at a high level in terms of accuracy. These methods enable to verify the computational models or eventually suggest their modifications.

It should be realized that mentioned facts can be accurately applied to simple objects and structures. The combustion engine excluding substructures and their properties, which entering in to the MBS analysis also includes other elements that have a considerable effect on the vibration and associated noise.

The results of vibration of virtual engine surface are values that do not involve, for example, coolant, lubricating oil, effect of spreading of acoustic wave and their reflection inside the engine. Therefore, the deployment of computational methods for determining the emitted noise is technically possible, but the real engine will have different values of the studied parameters. Therefore, the values calculated in this way could be taken only as qualitative values, about which it is only possible to say that this version is better or worse and to determine the approximate difference. Comparing the accurate calculated values is from this perspective still the future.

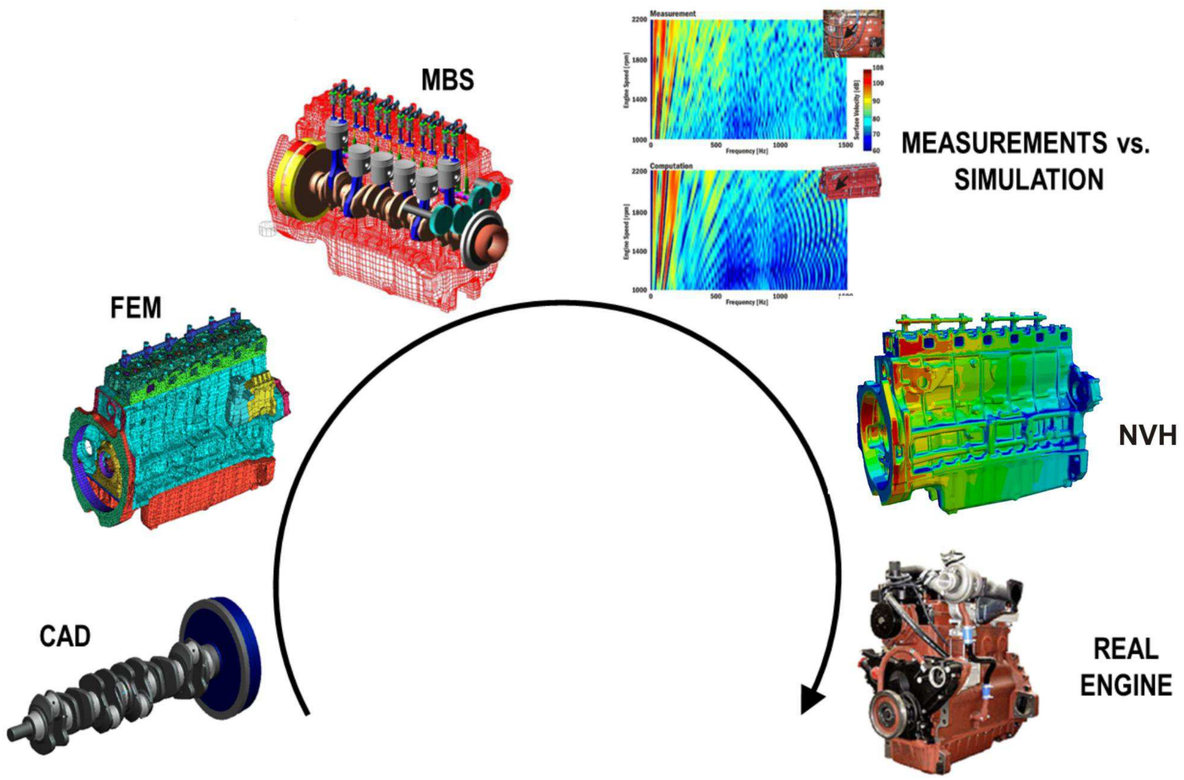


*Figure 1 Development phase of virtual engine*

As the acoustic pressure is directly dependent on the size of the velocity of engine surface vibration, it is faster and smarter way to determine this value as evaluating value. Comparison of the values for the different variants also gives only qualitative information and with a certain amount of uncertainty provides information about what is the difference between them.

By using computational methods based on BEM or IBEM, it is today possible to calculate the values of acoustic pressure emitted into the environment, on the base of the velocity of surface vibration. However, time-consuming of these methods is still relatively large and therefore their use in deciding on the design options at level of parts (block, power and bath) extends the time required for the development of the drive unit.

Also in this field there is fast and intense development and what it is now difficult or unrealistic it may be a common practice in a few years. The process of calculating the noise and vibration tends to that already in the development, it would be possible to hear from a computer what sound will affect the passenger in a car or what will hear a man standing on the street when a car goes around it. Estimation when this happens is pure lottery or shamanism.

# 3 AIMS OF THE THESIS

This dissertation thesis will focus on processing and visualization of the results obtained by using the virtual model. Input parameters are finite element mesh of engine and coordinated components of velocity (shift) at its nodal points.

Another aim is to propose a method how to evaluate various engineering designs and modifications in the area of the impact on vibration and thus the corresponding noise. The method will be fairly quickly and accurately compare the individual alternatives. By this method, it will be also possible to determine which variant is better or where it should come to edit.

From the last section follows, that it is preferable and more effective, as an evaluation criterion to choose the velocity of surface vibration. Therefore, in this work this criterion will be used and comparison will be on the basis of the speeds calculated on the normal components to the surface.

In order to verify the proposed method, it is necessary to take measurements. Measurements will be performed on basic models by using the contact and contactless sensors of velocity. One of the goals is a creation of a program that will process the measurement results and enable to view them by postprocessor and compare them with measurements.

Input data and computational part are implemented in Matlab program, or in a free program Scilab. Results will be presented also by the open-source postprocessor ParaView. This solution provides the ability easily develop and modify by anyone.

Another section of this thesis describes the individual steps just like they are carried out and as a logically follow. The final section includes samples of using the proposed method of evaluation of vibration respectively noise.

# 4 ELECTRODYNAMIC VIBRATION EXCITER

To verify the model and the chosen procedure, but also to obtain input parameters into the computational model it is necessary to take measurements. For fine-tuning, it will be the measurement at first realized on a single experimental object and then to the real parts. In the experiment, it is advisable to excite the structure. From the possible ways of excitation, it was chosen vibration exciter, as the most flexible.

One of the most common vibration exciter is an electromagnetic vibrator. It works on a similar principle as a speaker. In the center, it is usually placed permanent neodymium magnet (exceptionally, the permanent magnet is replaced by coil) around which is voice coil, through which is transmitted the alternating current. In the coil is induced alternating magnetic field, which makes it move. Force applied to the coil is then transmitted to the measured object. [8]

## 4.1 DESIGN OF VIBRATION EXCITER

The basis, in the engineering design of the vibration exciter, are available information about industrially produced exciters. In order to use the driver as flexible as possible, method of mounting of the cover must allow tilting about a horizontal axis to both sides of an angle greater than 180 °. The mount of the frame is provided with holes for fastening bolts with an eye in order to hang the exciter on flexible lines or firmly mount to the ground.

The main requirement is to ensure the position of the coil closest to the center of its stroke in any position of exciter. Also, that the coil will remain in coaxial relation to a permanent magnet, so there are not friction losses, which might tend to damage the drive. Of course, there should be the possibility of easy assembly and disassembly.

Axisymmetric of the coil was assured by the 3 guide rods and the central cross. In order to eliminate friction, it was used a special plastic. Backing the position in the middle of the stroke was carried out by a spring, which was fitted on the oscillating rod through the holder, which was supported against axial movement by the pin. On the upper lid was placed the second holder. The whole engineering solution of the first design is shown in Figure 2.

After assembly the first design and start testing, it was found that handles mounting which enables rotation, was after several attempts unable to hold the weight of cover with drive unit. This was due to rotation of support in inside the cover. Also, mounting of the coil, after a few minutes of operation and especially after loading at low frequencies but on high amplitude, stopped fulfil its purpose and the coil fell and touched the permanent magnet.

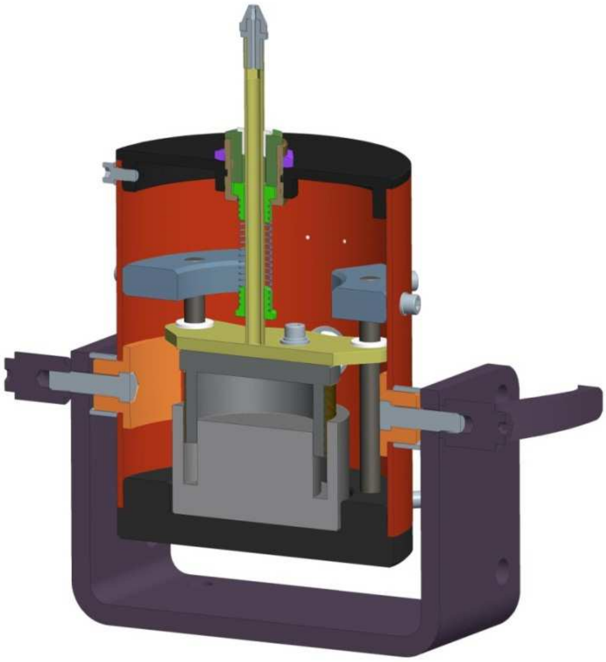

*Figure 2 First design of exciter*

For ensuring of axisymmetric and center position of voice coil, it was designed special spring (Fig. 3), where the axial stiffness in comparison with the radial is much smaller. Due to its shape it provides axisymmetric also in the deformation.

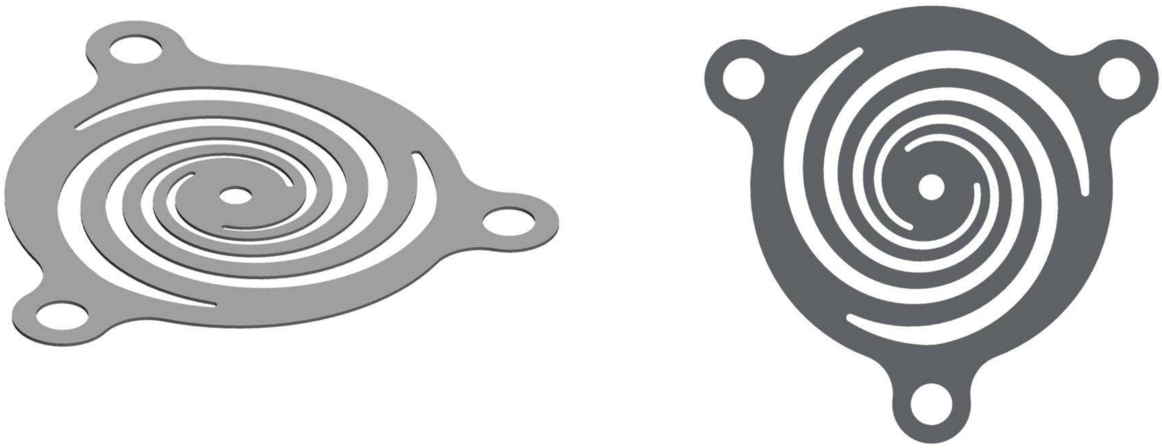

*Figure 3 Special spring of exciter*

This solution allows for easy adjustment of ensuring the center position. First by using FEM it has been designed shape and thickness of spring. The design was based on the weight of moving parts, maximum stroke and expected maximum power in the extreme positions

In the second design, it was also ensured the fixation in a certain position, and it was prevented rotation of the cover of exciter. The solution was based on rotation of support by 90°, what caused contact with the lower lid, and thus assurance against rotation.

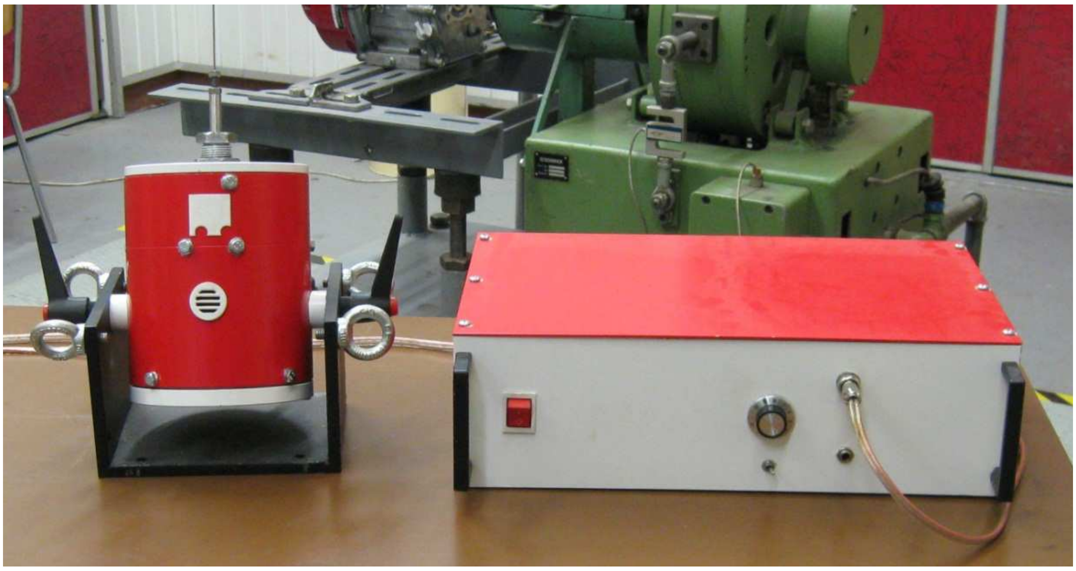

*Figure 4 Electrodynamic exciter and amplitude amplifier*

## 4.2 THE CONTROL SOFTWARE

The primary source of signals, for which it is the system designed, is the PC or laptop sound card. In order to generate the signal in real time, it was created a specific program.

The program was written by using Matlab and converted, so it can run on any PC with Microsoft Windows operating system. Program, in addition to generating the signal, allows a saving in to file *.wav. This type of file can be run in different programs and operating systems.

The program also allows to give as the input any *.txt or *.dat file with a record of the measurements and then to play or create an output file.

The solution designed in this way has several advantages. One of the most important is that the generator is not bound to a specific PC by the license. It allows creating a database of tested signals with different characteristics. Because the input to the amplifier is the output from the sound card and thus 3,5 mm termination, it is also possible running by using smart phones, what increases flexibility and mobility of device.

# 5 EXPERIMENTAL MODAL ANALYSIS

This chapter will discuss the experimental modal analysis, which is one of the methods of measurement and detection of dynamic properties of structures or systems.

## 5.1 PRINCIPLE OF THE METHOD

In the theoretical vibrational analysis, it is progressing towards from the physical model to the response model. In the EMA there it is the opposite way. Frequency response function (FRF), which forms the basis of the model it is expressed as follows [8]:

$$H = \frac{Output}{Input} = \frac{Motion}{Force} = \frac{Response}{Excitation}$$

The principle of modal testing is that the measured structure it is excited by the strength, which waveform it is measured and simultaneously it is measured waveform of the vibration response of the system on this force. For measuring of the response, it is usually used the sensor of acceleration. If this measurement it is made on of a sufficient number of points, the result is the response model.

The exact definition of a matrix element FRF is [8]:

$$\mathrm{a}_{jk}(\omega) = \frac{x_j}{F_k} = \sum_{r=1}^{N} \frac{\boldsymbol{\phi}_j^r \boldsymbol{\phi}_k^r}{\lambda_r^2 - \omega^2} \tag{1}$$

where $\lambda$ is a value of r-th mode, $\phi$ is the j-th element of the r-th vector of eigen forms ($\phi$), i.e. the relative displacement in the j-th point of oscillation on the r-th form and N is the number of modes.

If we use the relation 1, it is possible to show that an appropriate set of measured receptances must contain only one row or a one column of mobility matrix. This means that either the system is excited at one point and responses are measured in all other points, or the response is measured at one point, and the system is excited in all other points.

For a description of the interrelations of these signals in the frequency domain, it is used the FRF $H$, and in the time domain, it is used the impulse response function (IRF). These are so called the descriptors of the system and are independent on the signal [8].

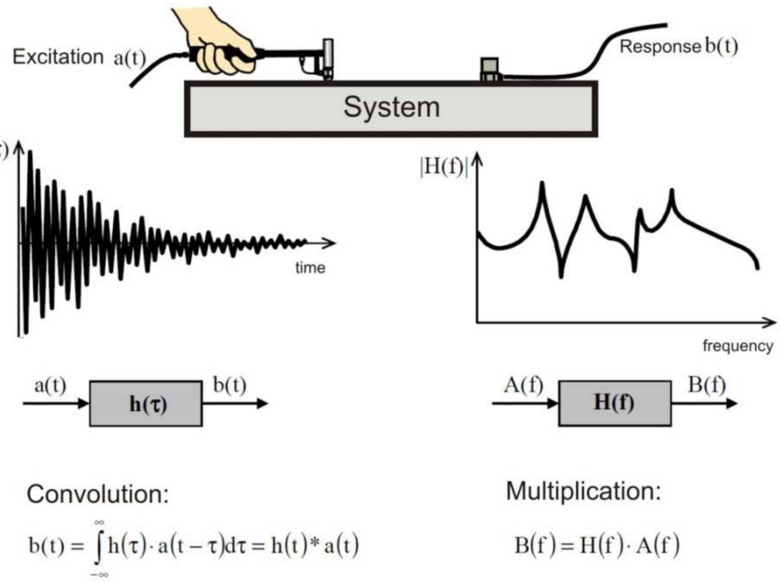


*Figure 5 Frequency response function and impulse response function [24]*

The proportion of the complex amplitude of the displacement and the excitation force is a function of receptance that is FRF but no longer dimensionless. For a system with 1° of freedom it looks as follows:

$$H(i\omega) = \frac{X(j\omega)}{F(j\omega)} = \frac{1}{k - m\omega^2 + j\omega b} = \frac{R}{j\omega - (-\delta + j\Omega)} + \frac{R*}{j\omega - (-\delta - j\Omega)}, \qquad (2)$$

where R is rezidum, Ω is natural frequency of damped oscillation, δ decay constant.

Because this is a complex function, it is not possible to represent all the information by 2D chart. The most commonly used it is displaying of the amplitude - frequency characteristics. All methods of displaying of FRF are shown in Fig. 6.

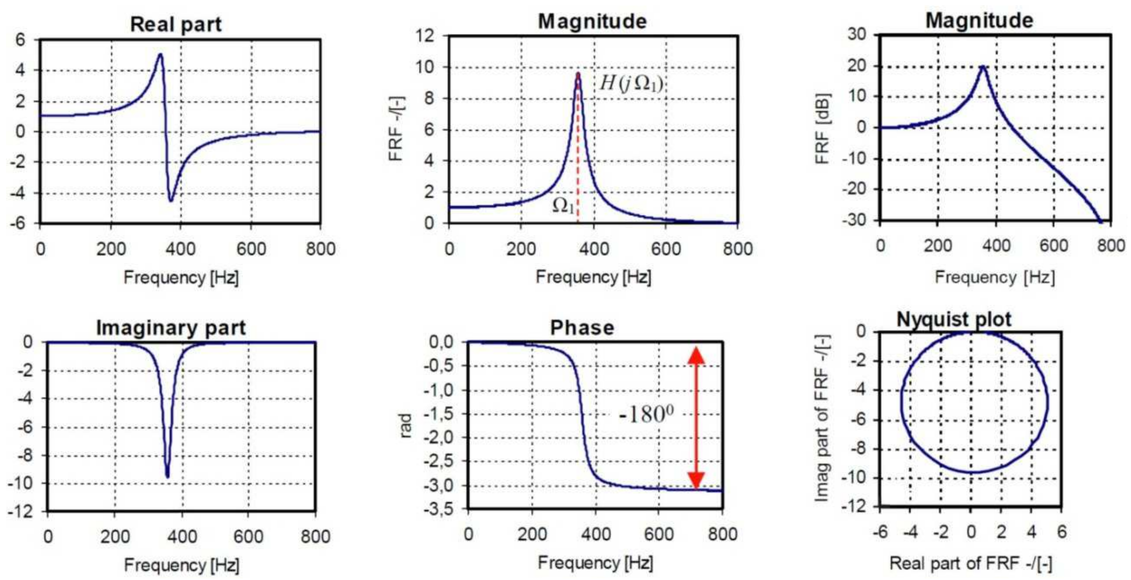


*Figure 6 Display options of FRF [24]*

In the FRF chart, where it is visualized the dependence of amplitude on frequency, and natural frequency it is determined by the maximum value of the amplitude.

For the description of the damping, there are most commonly used two types of models. It is a viscous damping and hysteresis damping. Hysteresis damping is more precise for the forced vibration of systems with more degrees of freedom but for the free vibration is problematic. More often it is for description of the damping used viscous damping.

One of most used method is called half-power point method (half-power point). This method uses a waveform of dependence of amplitude on the frequency. Determination of the proportional damping is also relatively simple. In the point of natural frequency, it is found the maximum amplitude. Next it is calculated its size of half-power (i.e. formula it is divided). In this value, it is leaded the parallel line to the horizontal axis, and where it crosses the graph of dependence, there the frequencies are subtracted (see Fig. 7)

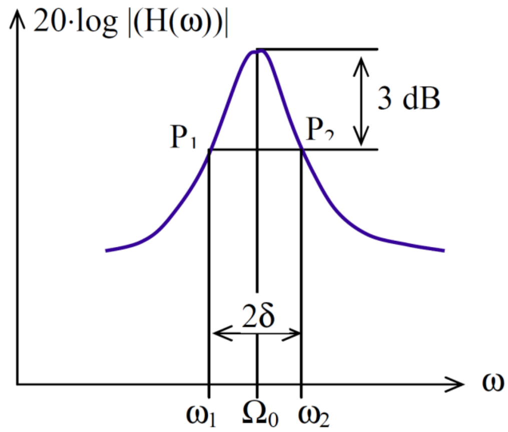

*Figure 7 Half power point [8]*

Damping ratio determined using:

$$\frac{\omega_2 - \omega_1}{\omega_0} = 2\zeta. \tag{3}$$

## 5.2 EVALUATION OF MEASUREMENT

Analysis of natural frequencies and damping is relatively easy for mass-spring-damper system. However, real cases are more complicated and complex. Therefore, it is not possible these values easily to subtract from the chart.

The first problem is that it is measured in more points. This is connected with certain complications. Despite the best efforts, it is quite likely that one or more sensors are placed close to the nodal points. In this case, the visualized waveform of FRF would be significantly distorted. Also, orientation in the maze of graphs (see Fig. 8) and determination, which peak is the right one, it is a problem. Therefore, there are the methods by which it is also possible from a large number of points accurately and quickly determine natural frequencies, damping and waveform of FRF.

The most common include [11]:

- SUM – Summation function.
- MIF – Mode Indicator Function.
- MMIF – Miltivariate MIF.
- CMIF – Complex Mode Indicator Function.

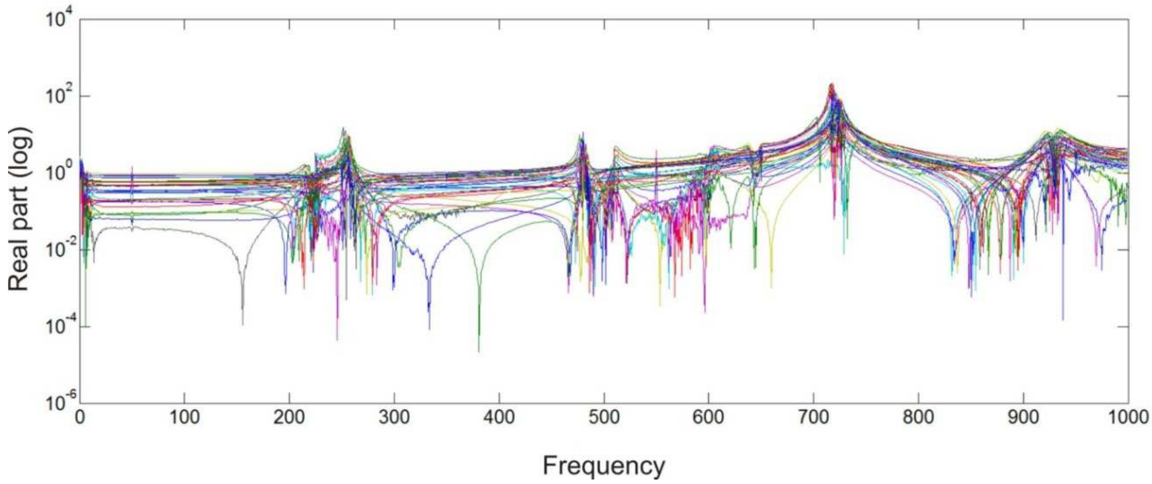


*Figure. 8 FRF of measurement points*

The most accurate method for determining the natural frequencies is the CMIF method. It bases on singular value decomposition, which consists of measured data of FRF at individual points. Singular value decomposition is defined [10]:

$$[\boldsymbol{H}] = [\boldsymbol{U}][\Sigma][\boldsymbol{V}] \tag{4}$$

Where [**H**] is the frequency response function matrix, [**U**] is left singular matrix (unitary), [∑] is singular matrix (diagonal) and [**V**] is right singular matrix.

From the options listed above it was used for processing the measurement results the CMIF method. Exported results from the measuring software have been imported into Matlab program. Source of code for processing the measured data was created from [10], where it was only a slight modification.

The resulting graph of FRF obtained using the CMIF method it can be seen in Fig. 9.

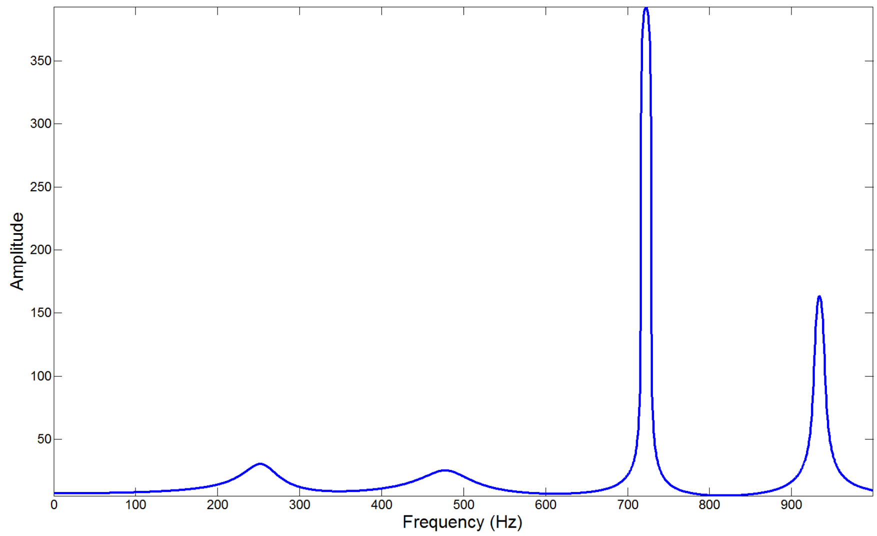

*Figure 9 Reconstruction FRF from measurement data using CMIF method*

For verification of the accuracy of the number of natural frequencies, it is used the method MAC (Modal Assurance Criterion).

The modal assurance criterion matrix is a mathematical tool to compare two vectors to each other. It can be used to investigate the validity of estimated modes.

The MAC between two mode shape vectors {ψ}_r and {ψ}_s is defined as:

$$MAC(\{\psi\}_r, \{\psi\}_s) = \frac{(\{\psi\}_r^{*T}\{\psi\}_s^{*T})^2}{(\{\psi\}_r^{*T}\{\psi\}_r)(\{\psi\}_s^{*T}\{\psi\}_s)} \tag{5}$$

The MAC will approach the value 1 if {ψ}_r and {ψ}_s are the same mode shape. It {ψ}_r and {ψ}_s are different mode shapes, the MAC value should be low, due to the orthogonality condition of the mode shapes.

In Fig. 10, it can be seen a preview of the MAC. For the calculation of the MAC in Matlab, it was used code from [10].

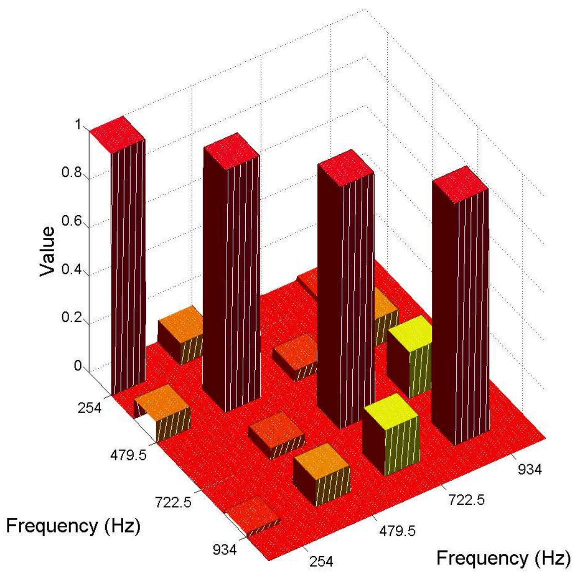

*Figure 10 Modal Assurance Criterion*

# 6 PROCESSING AND VISUALIZATION OF MEASUREMENT RESULTS AND CALCULATION

This chapter will describe the method of processing the results obtained by FEM or MBS analysis and based on this, determination of acoustic parameters of examined object. As mentioned above, for obtaining of acoustic parameters it is essential to set the normal velocities of surface vibration. Therefore, in this work as a benchmark it is used just the normal velocity.

The methodology is based on the fact that the CAD model is using specialized software discredited to finite element mesh. Using the FEM program ANSYS is then performed the harmonic analysis and its results together with a description of the mesh are processed, resulting in normal velocities of surface vibration. As a visualization tool, it is selected the program ParaView. A detailed description of the individual steps is listed below.

## 6.1 PROCESSING OF FE MESH

For the creation of the 3D parametric model, it is used program Pro/Engineer. The created model is then exported into a neutral format, which allows further processing.

For the creation of finite element mesh, it is used ICEM program. Using this program is created a mesh, defined components, contacts and material properties. Final version of the mesh is further processed in the FEM program ANSYS.

In the ANSYS program, there are on the examined object applied boundary condition and load. In the calculation, it is determined the response of the structure to the periodic excitation at the defined points. As a load, it is possible to use the results of MBS analysis, which should be converted from time domain to the frequency domain using Fourier analysis. If no results of MBS analysis are available, it is also possible to use basic load in the form of sine wave power.

In defining the parameters of calculating it shall be set the initial and final frequency and frequency step. Also, the main parameters affecting the accuracy are the damping properties, which can be obtained by the method mentioned in the previous chapter.

The results of harmonic analysis are coordinated components of deformation depending on the frequency. These results except the certain examples of models do not correspond to acoustic properties of the examined object. Therefore, it is necessary to process the results.

For that reason, it is necessary to export the results in a reasonable format in which can be further processed. For this purpose were created macro files into the ANSYS.

The first macro file exports the finite element mesh. In export, it was used information obtained in [13]. There are the information about the number of exported nodes and then are listed the numbers of individual nodes together with their coordinates. Another part of the file contains information on the walls (faces) of the individual elements, where in the first column is the number of the element and in the other columns are numbers of elements with a common wall (face). The last part is the list of elements, that contains the number of element and numbers of nodes by which is formed.

The resulting file is then processed using Matlab. As first, there are determined from the obtained file the walls (face), which form the surface of the object (there are included also the inside walls and holes in walls). Subsequently, it is determined a surface area element. A crucial step is the determination of the normals. This is divided into two steps, where in the first one it is determined the normal to the surface element by vectors formed by connecting of nodes forming the element. The second step is to determine the normal at the node. Whereas one node usually consists of more elements, in determining of the final normal it is found the final vector from vectors generated by multiplying the normal to the element containing that element.

Principle of the determination of the normals to the element it is shown in Fig. 11 where the method is presented on a single 2D example.

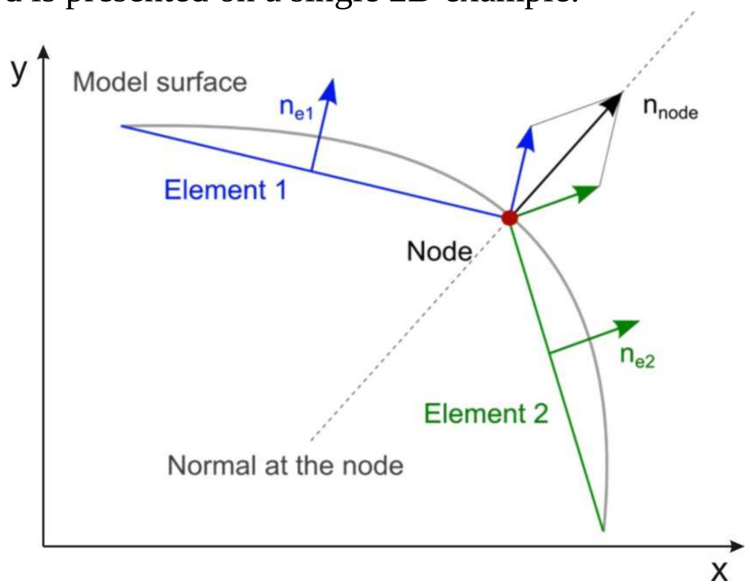


*Figure 11 Calculation of normal vector*

The second macro file exports the results of harmonic analysis. There are the information about the number of exported frequencies, number of nodes and information about what a frequency is currently exported. In the next line, there are the results of harmonic analysis for each node and coordinate components of displacement at a particular node.

Also, this file is then processed by Matlab. The values of displacement for individual frequencies are converted into velocity. From the coordinate components of velocity, it is calculated the resultant velocity vector at that node.

The last step is the conversion of the resultant velocity vector into the direction of the normal vector at that node. The principle can be seen on the 2D case in Fig. 12.

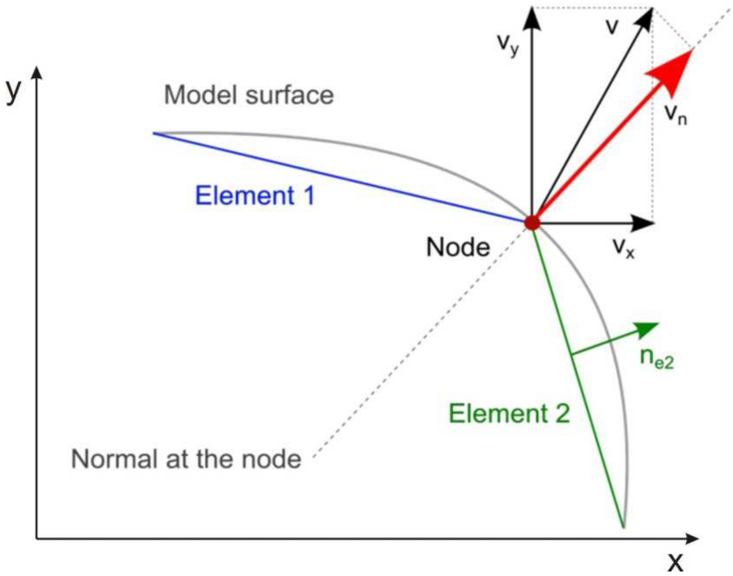


*Figure 12 Calculation of normal velocity*

## 6.2 ANALYSIS OF THE VELOCITY OF SURFACE VIBRATION

From the results obtained by the previous method, it is detected surface vibration at specified frequencies in certain nodes. However, for more detailed evaluation it is sometimes necessary to find out the integral characteristics in a certain frequency range or to have available one number representing the properties of the examined object. In the mutual comparison, it is also used the waveform of velocity of vibration throughout the structure depending on frequency.

Determination of individual acoustic parameters depends on whether it is a node, component or eventually complete assembly of the analysed object. However, in each of these cases are analysed the normal velocities and the surface of the object. Distribution and mutual connection can be seen in Fig.13. [14]

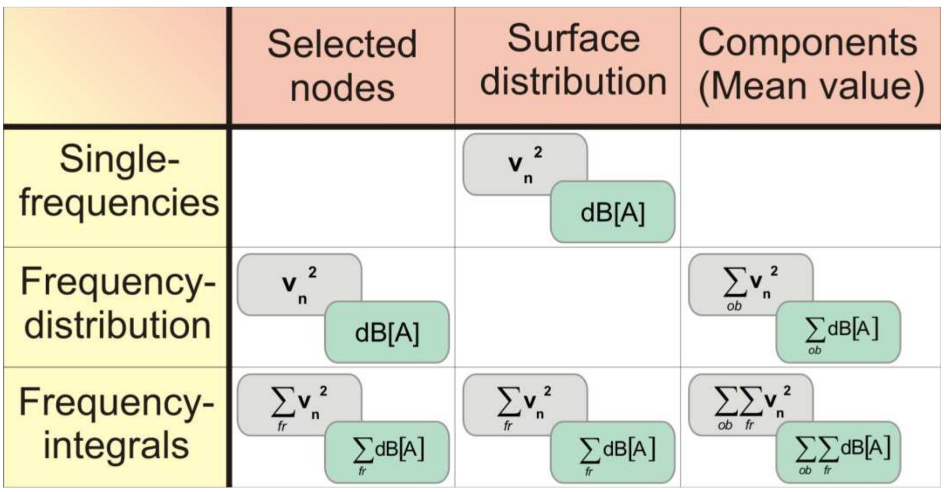


*Figure 13 Overview of possible results [14]*

To calculate the acoustic parameters are according to [14] used the following relations:

Mobility:

$$E[x^2] = \frac{1}{2}\sum_{f_1}^{f_2} v_n, \tag{6}$$

where $v_n$ is normal velocity, $f_1$ is lower frequency and $f_2$ is upper frequency.
Surface noise level in dB [14]:

$$L_v[dB] = 10\log\frac{(E[x^2])_{f_1}^{f_2}}{{v_{ref}}^2}. \tag{7}$$

## 6.3 VISUALIZATION OF RESULTS

After processing of the measurement results and determination of the acoustic parameters, it is, in addition to an integral number, describing the level of acoustic noise areas or graph of dependence of vibration on the frequencies, also the visualization of the entire surface in 3D space.

In the initial phase of the testing of chosen procedure, the visualization was carried out using Matlab. However, this solution showed significant impracticability for large objects, because that from size of the mesh around 3000 nodes, there was an inability to rotate and zoom in Figure in MATLAB program.

For this reason, it has been chosen as a postprocessor, program ParaView. This program is free, and it is used to visualize a wide range of scientific applications.

For purposes of this work, it was chosen as the input file, the file with suffix *.vtk. Description of this file is given in [15]. The advantage of this file is that it allows visualizing different types of elements and also the results of analyses related to the nodes or elements, by using the colour range. Another advantage is the fact, that this file uses for the description of the geometry a similar principle as ANSYS and thus the conversion of the exported file from ANSYS into the ParaView, was relatively easy to apply. Writing of *.vtk file was also carried out using Matlab.

# 7 EXAMPLES OF PRACTICAL APPLICATION OF METHOD

This chapter will show examples of using the above listed procedures to real objects. In the first part, it is a single flat metal plate with ribs. Next section describes the cover of valvetrain for 4 cylinder diesel engine Zetor. In the last part, there is the sample of processing the results of MBS analysis for 6 cylinder diesel engine Zetor.

## 7.1 THE VERIFICATION OF PROCESS FOR A SIMPLE CASE

Flat metal plate was selected because of its easy production and the possibility of a simple verification of models.

Verification of the chosen methodology is implemented on two experimental models. The first model is a flat metal plate and the second one is a metal plate of the same dimensions but with added rib in the form of welded belts. Experimental models can be seen in Fig. 14

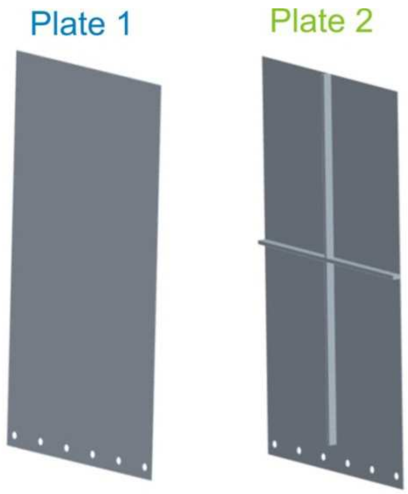


*Figure 14 Experimental models*

On the experimental models, it is created a mesh of points at which the measurement is then performed. Excitation is positioned eccentrically to the center of metal plate. Mounting is solid and fixed. Measured samples are securely fixed to a steel plate. Electrodynamic vibration exciter is placed freely by using steel chains.

For scanning of displacements, it is used the laser scanner POLYTEC 4000 Series Laser Vibrometer, as the recording device it is used Compact PULSE data acquisition unit. For saving, processing, visualization and control, it is used software PUPLSETM 13. The real experimental arrangement can be seen in Fig. 15.

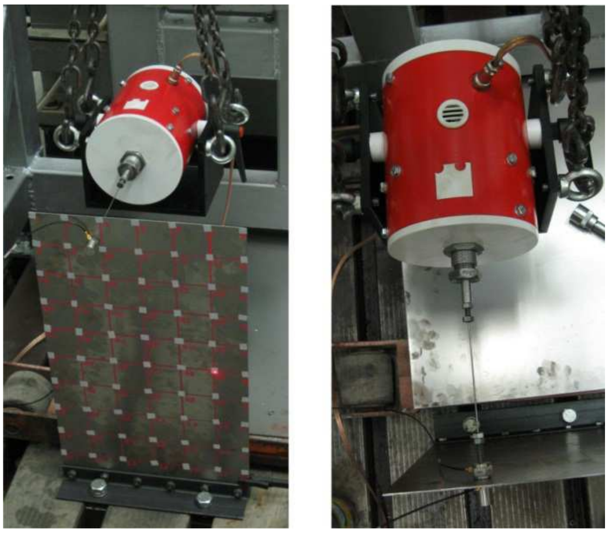

*Figure 15 Real experimental arrangement*

The next step is the comparison of velocity of surface vibration investigated by measuring and calculated using the harmonic analysis when using the sine wave power 10 N. In Figure 16, it can be seen the results for the frequency of 30 Hz, 110 Hz and 260 Hz.

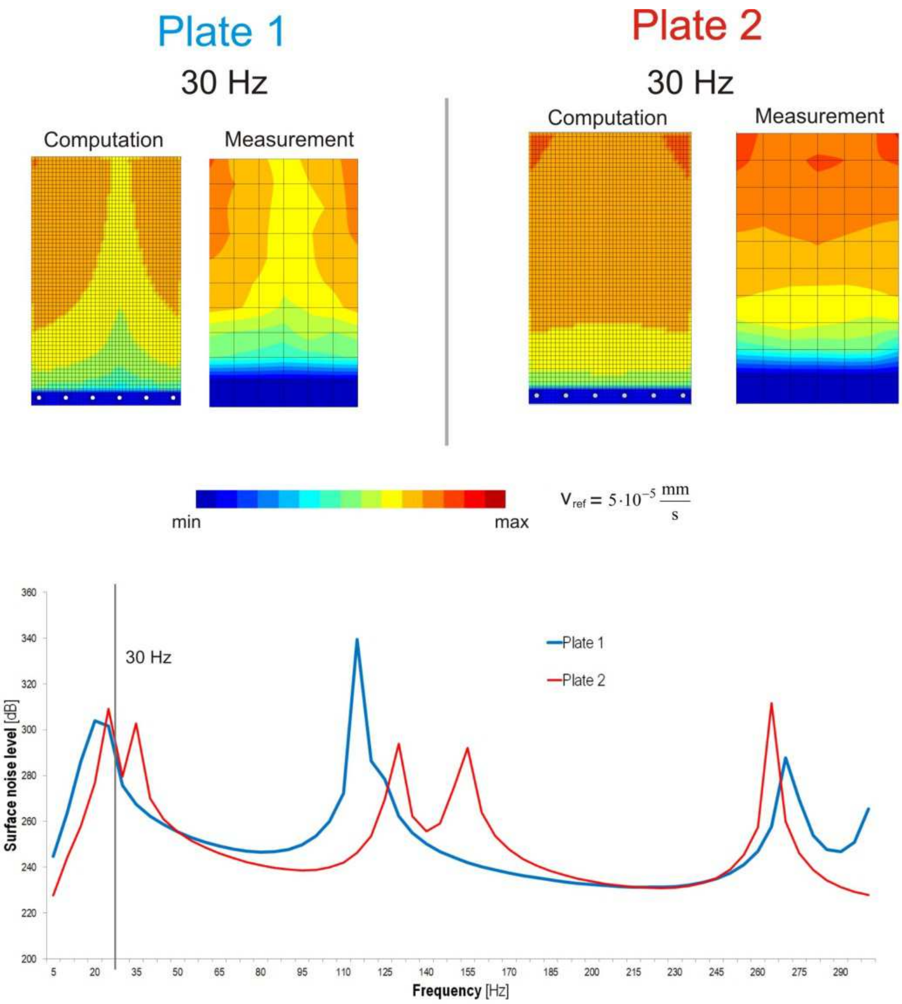


*Figure 16 Comparison results of calculations with experiment*

Visualization of calculated velocity of surface vibration for examined objects throughout the frequency range is shown in Fig. 17.

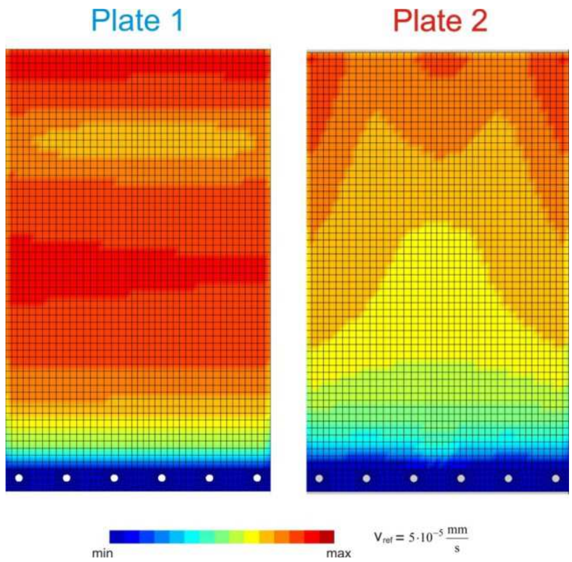


*Figure 17 Calculated integral mobility*

The detected results qualitatively meet the expectations. The metal plate with added rib is in terms of vibration and noise more favourable. The calculated and measured values in decibels are relatively high. This is because of that, the loading force was in comparison with the stiffness of metal plate vast and metal plate had a significant displacement (in several mm). This way of the load it has been chosen because of the possibility of visual control of the real vibration, where real form of metal plate deformation at the selected frequencies is identical to the calculated and measured normal velocities.

## 7.2 THE COVER OF VALVETRAIN OF THE ZETOR ENGINE

Another example is the detection of dynamic properties of the cover of valvetrain four-cylinder diesel engine Zetor. Preview of the cover can be seen in Fig. 18.

*Figure 18 Cover of valvetrain of the Zetor*

The measurement is performed using the same apparatus as in the previous case. The difference is in the excitation signal and also in how to fix the cover and the electrodynamic vibration exciter. In the previous case, it was used the continuous frequency sweep. In the measurements of the cover, it was used the white noise. Arrangement of the measuring chain it can be seen in Fig. 19.

*Figure 19 Arrangement of the measuring*

The measurement results are again processed using Matlab and comparison of natural frequencies can be seen in Table 1.

*Tab. 1 Results of measurement and calculation*

| Number of Mode | Measurement | Calculation | Damping Ratio |
|---|---|---|---|
| 1 | 254 [Hz] | 285 [Hz] | 0,076 |
| 2 | 480 [Hz] | 457 [Hz] | 0,017 |
| 3 | 722 [Hz] | 730 [Hz] | 0,013 |
| 4 | 934 [Hz] | 893 [Hz] | 0,012 |

This measurement is carried out in order to determine the dynamic properties, which are subsequently used in the following example.

*Tab. 2 Damping variable*

| Variable | Value |
|---|---|
| $\alpha$ | 250,7 |
| $\beta$ | -3,19·10-6 |
| $\zeta$ | 0,031 |

## 7.3 VIRTUAL ENGINE

The last sample is the application of the evaluation and visualization on the six-cylinder engine Zetor. The input data represent a finite element mesh, alpha, beta, proportional damping, and as the load there are used the results of the virtual engine.

Due to the fact, that the entire engine block is made of the same material as a cover of valvetrain in the previous case, the dynamic properties alpha, beta and proportional damping are applied from the above used results.

For the calculation of virtual engine in MBS but also in measuring, it has been the engine fixed on silent blocks. These silent blocks were simulated using support in the extreme points of the engine. For the supports, it is defined the stiffness of 50 000 N/mm and the damping of 100 Ns/mm.

From the results of simulations of virtual engine are obtained coordinate components of forces in the bearings of crankshaft and the forces in the liners of cylinders. For the purposes of harmonic analysis, it is necessary to transfer these forces from time domain to the harmonic components. The transfer is carried out by using the Fourier analysis.

Harmonic analysis is made from 0 Hz to 1000 Hz with the step of 10 Hz. Due to the fact that this is a comprehensive model containing elements of 228 085 and 190 927 nodes, the harmonic analysis is performed only for two revolutions 1600 min-1 and 2200 min-1.

Given that during the experiment and writing of this work, it was not available any measuring device for visualizing of the surface vibration (acoustic camera), the calculated results have been compared with results obtained in [1]. In that publication is the Campbell diagram of normal velocity at the point of the engine block surface.

In the figures below, there are pictures representing the comparison of normal velocity vibration of engine's surface for the selected revolutions at selected frequency.

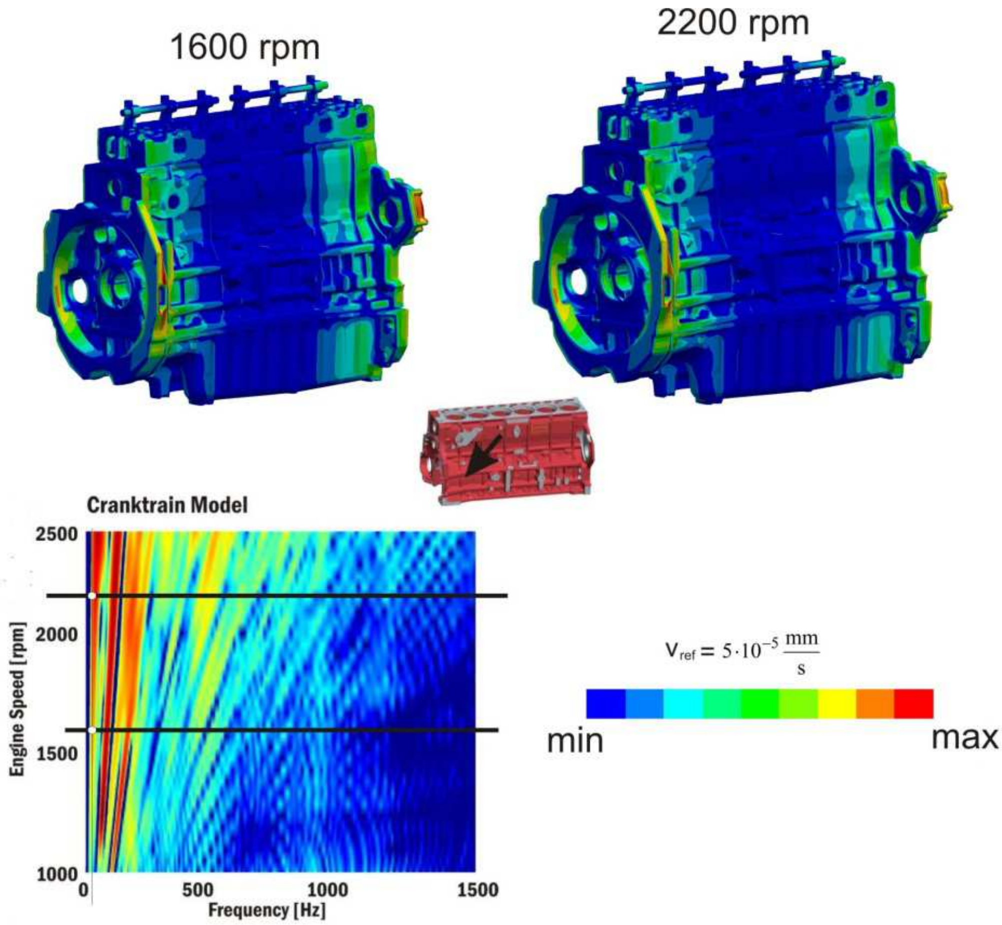


*Figure 20 Mobility for 30 Hz*

From the above picture, in comparison with the results in the Campbell diagram it can be seen, that for the visualized point are the results qualitatively consistent.

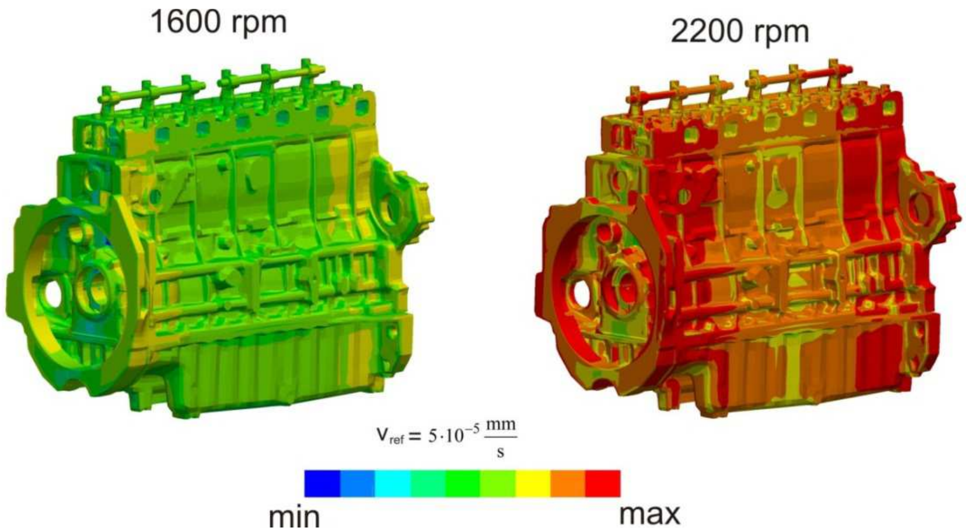


*Figure 21 Surface noise level*

Integral vibration of surface for the frequency range and revolutions it is shown in Fig. 21. The characteristic value for the whole engine, it is the mobility, which in case 1600 min-1 is 90,66 dB and in case 2200 min-1 is 133,65 dB.

# 8 CONCLUSION AND DISCUSSION

In presented thesis, it is at the beginning evaluation of the current status. In the next section, it is presented a design of electrodynamic vibration exciter, which is used for experimental verifying of calculations and determination of dynamic properties of the structure. The principle of the method bases on processing the results of harmonic analysis. Detailed description gives the chapter 5. In the last chapter is on three examples shown a possible use of the method and presented potential pitfalls and problems. One of the examples is purely experimental and others present method of solving practical problems.

The above mentioned examples describe the method and the way of determining the acoustic properties of either the powertrain or its individual components. Accurate qualitative comparison was not possible to make due to the insufficient laboratory equipment. However, the method has adequate compliance of experiment and measurements for qualitative comparison.

By this procedure, it is possible to determine the quantitative levels of generated acoustic emissions and thus the mutual comparison of individual engineering designs. This would require the use of modern methods of visualization, which is the acoustic camera. On the other hand, it should be also included in the calculations other elements that influence the noise and vibrational characteristics of objects or the whole structure. To the most basic, it is possible to include the impact of the operating fluid, changing of properties depending on temperature and as well impact of internal interaction of excitation to the spread of vibration and noise.

The advantage of this method is mainly the relative simplicity and speed of comparison of individual load step, variants or other modifications. In using of implementation as indicated in this work, is this solution unsatisfactory for complicated structures (see the time of the calculation of engine). Speed up of calculation by using the FEM is possible only by reduction of model. This can be achieved by simplifying or omitting structural details or by coarser mesh. However, both solutions involve the increasing of uncertainty of the calculation. The second way to speed up, it is to use for processing of the calculated results another way. As appropriate, it appears to rewrite a debugged source code either from Matlab to C + + or to Fortran. In them, it is possible to speed up the calculation in some cases more than 10 times.

This thesis presents initial analysis and a one of possible approaches to the analysis of acoustic properties. It is not based on direct evaluation of generated noise, but on the evaluation of surface vibration. The work is the basis for future development to solving the issue. The development could move towards the use of normal velocities to obtain the changes of the acoustic pressure around the investigated object. Another option is to focus on increasing the accuracy of calculation results. One option is also extension of the method on the determination of the dynamic properties of the system, and creation a basic database of these parameters that would be used in the design of new engineering solutions in the future.

The objectives of the dissertation thesis, which was specified at the beginning, have been met. The result is a designed method of evaluation of noise (vibration), with the basic application and presentation of the results. The whole method is created in such a way that allows its use by the free software; therefore it is open to further development.

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# 10 CURRICULUM VITAE

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**EDUCATION**

| | |
|---|---|
| 09/2007 – | Brno University of Technology, Faculty of Mechanical Engineering, Institute of Automotive Engineering.<br>Subject of Doctoral Thesis: „ Analysis of Powertrain Acoustic Properties “.<br>Supervisor: prof. Ing. Václav Píštěk, DrSc. |
| 09/2009 – 06/2011 | Brno University of Technology, Faculty of Business and Management.<br>Master Thesis: „Selection of Information System for Small Company“. |
| 09/2002 – 06/2007 | Brno University of Technology, Faculty of Mechanical Engineering, Institute of Machine and Industrial Design.<br>Master Thesis: „Optimization of Cabin Mounting“. |

**PROJECTS**

| | |
|---|---|
| 2011 – 2013 | Development of methods suitable for vibration reduction of drive units. |
| 2009 – 2012 | Research and development of two-stroke diesel engine with opposed pistons, a member of the research group. |
| 2010 | Development of methods for determining the mechanical losses, Standard grant of Specific research FME BUT FSI-S-10-2, member of research group. |
| 2009 | Laboratory for comprehensive analysis of frictional losses in internal combustion engines. |

**TEACHING**

Leadership training of subjects:
Institute of Machine and Industrial Design
  CAD II
  Design and
Institute of Automotive Engineering
  Computational models
  Computational methods
  Team Project

**PRODUCTS**

2011 AMBRÓZ, R.; PROKOP, A.; NOVOTNÝ, P.: UADI shaker V1; Electrodynamic vibration exciter.

2010 AMBRÓZ, R.; NOVOTNÝ, P.: SBLoader; SBLoader.

**EXPERIENCE**

| | |
|---|---|
| 02/2011 – | NETME Centre |
| 04/2012 – 06/2012 | Engineering Center Steyr |
| 01/2009 – 03/2012 | AKTechnology |
| 08/2006 – 08/2009 | Zetor |
| 06/2005 – 08/2006 | ABB |

**SKILLS**

**Driving Licence:** A1,AM, A, B1, B, C, T

**Languages Skills:**
Slovak Language – native
Czech Languages – fluent
English Languages – pre-intermediate

**SW skills:**
Very good – MS Office, Inventor, SolidWorks, Pro/Engineer
Good – ANSYS, MathCAD, AutoCAD, COREL, MATLAB
Basic –ADAMS, ANSYS ICEM, HTML, ParaView

# 11 ABSTRACT

This thesis deals with the analysis of the acoustic properties of powertrain. Introduction section contains the theoretical analysis of the physical principles of vibrations and noise and an overview of the current state of solving problems in the area of powertrain design.

Main part of the work describes the engineering design of electrodynamic vibration exciter, design of evaluation of measurement results and also the method of processing the results. Acoustic properties are evaluated by normal velocity of surface vibration.

In the final part of this work, there are presented the samples of the use of the method in practical examples connected with the design of powertrain. The whole solution is a comprehensive method of analysis suitable for quick evaluation of the impact of engineering changes on the acoustic properties.