Vibroacoustic Diagnostics Based on the Experimental and Numerical Approach

Roman Zajac, Václav Otipka, Aleš Prokop, and Kamil Řehák

Abstract During the last decade, many developers in the engineering industry are focusing on diagnostics and effective eliminating of the vibration and noise. The vibration and acoustic emissions are directly related to each other. Based on this fact, it is important to take into consideration the original vibration sources (engine, gearbox), but also the radiated noise to the environment. Current time brings unprecedented possibilities in the field of laboratory measurements and computational simulations in terms of hardware and software. In the issue of vibroacoustic diagnostics, the combination of these two approaches is very often required. In the first phase, the modal properties of the structure are usually examined, and the material properties are defined according to the mutual validation from both approaches. Subsequently, the amplitudes in structural (normal acceleration of the surface) and acoustic domain (sound power level—SWL, sound pressure level—SPL) are monitored during the operating conditions. In laboratory conditions, the structure is often excited by a mode exciter. In this case, the maximum amplitudes at the specific locations are primarily recorded on the structure. This article systematically describes the vibroacoustic diagnostics of the rectangular plate. Two approaches are explained in the paper: experimental and computational—based on the finite element method (FEM). In the text below, the individual procedures of the two methodologies are described. The obtained results from modal and harmonic response analyses are validated and compared with each other.

R. Zajac (✉) · V. Otipka · A. Prokop · K. Řehák
Faculty of Mechanical Engineering, Brno University of Technology, Technická 2896/2, 61669 Brno, Czech Republic
e-mail: Roman.Zajac@vutbr.cz

© Springer Nature Switzerland AG 2021

N. Herisanu and V. Marinca (eds.), *Acoustics and Vibration of Mechanical Structures—AVMS 2019*, Springer Proceedings in Physics 251,
https://doi.org/10.1007/978-3-030-54136-1_42

1 Introduction

The vibration sensing is a current trend in every industry. Primarily, the modal properties and the acceleration amplitudes at structurally critical places are examined on the vibrating structure. However, the issues of vibration and noise are mutually connected, the sound power levels and sound pressure levels of the oscillating structure are often evaluated at the final stage [1].

In this paper, a comprehensive vibroacoustic diagnosis is briefly described. This is from the perspective of using computer simulations as well as compiling technical measurements.

2 Technical Measurement

As outlined above, a technical experiment was applied on the simple rectangular plate made of gray cast iron. The vibration testing was divided into two main parts: modal and harmonic analysis.

Individual measurements were preceded by validation of main dimensions, weighting, and pre-preparation of the structure, calibration of the used sensors (acceleration sensors, microphones, etc.).

2.1 Geometrical and Mechanical Characteristics of the Rectangular Plate

The basic dimensions of the plate were further considered in the calculation models: the geometry dimensions were measured very precisely to increase the accuracy of numerical simulation, length approx. $a = 200$ mm; width $b = 160$ mm and thickness $t = 9$ mm. The density of the plate was determined from the real weight of the shell.

As mentioned above, the material of the rectangular plate was gray cast iron. These material properties were taken into consideration in the computational model: density $\rho = 6880$ kg m^{-3}, Young's modulus $E = 169$ GPa and Poisson's ratio $\nu =$ 0285.

2.2 Modal Analysis Measurement Process

Free support was applied as the constraint boundary condition—structure was placed on the low—rigidity foam.

Pulse excitation by a modal hammer was performed—applying a shock pulse at marked points on the structure (see Fig. 1). The response scanning was mediated

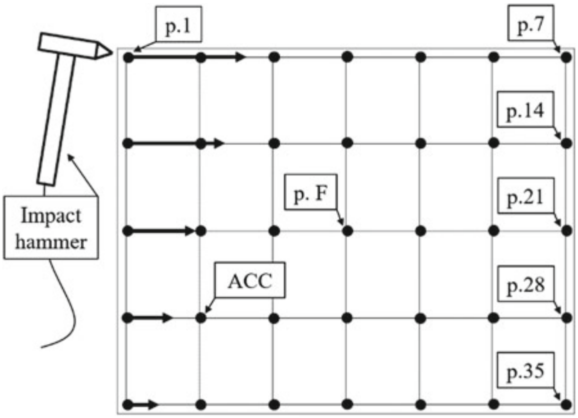


Fig. 1 Points grid

through a uniaxial acceleration sensor, which was stuck to the structure (see the position ACC in Fig. 1).

At the analyzer, two active channels were used. In the first channel, the impact hammer was applied and on the second channel, the ACC sensor was involved. The recording of the transfer functions at marked points and subsequent evaluation of modal properties were followed.

2.3 Harmonic Analysis Measurement Process

Measurement was performed in a fully anechoic chamber—examination of the acoustic features. The whole measuring stand is shown schematically in Fig. 2 with positions of important parts of the measuring chain according to [2, 3].

The structure was hung by silk lines (position e.5)—free boundary condition assumption.

The structure excitation via a modal vibration exciter (e.1)—the exciter was firmly attached to the lodge. The modal exciter was wrapped with a special sound-absorbing material (e.2). The transmission of the excitation signal to the structure was performed via the excitation rod (e.3). It was structurally connected through a force transducer (e.4) that was glued to the structure. A harmonic signal of the specified amplitude and frequency was transmitted to the structure.

There were considered two types of response sensors. In terms of the structural domain, the normal surface accelerations were recorded by a laser vibrometer (e.6). From the acoustic point of view, two condenser microphones (e.7) were used, placed 500 mm away from the vibrating structure in a defined position.

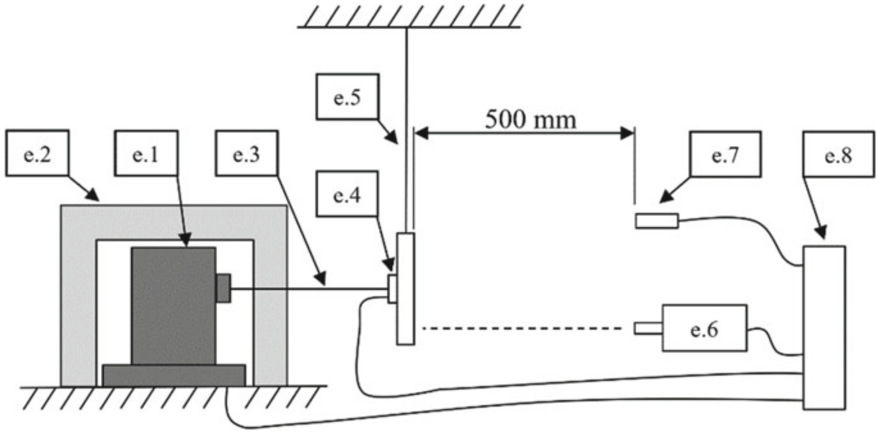


Fig. 2 Measurement set up

In this technical measurement, five channels were on: force sensor, excitation signal, two microphones, and laser vibrometer. In Fig. 2 is shown the position (e.8)—the analyzer, it is one of the most important parts of the measuring chain. Generally, the analyzer communicates between the hardware and the software part of the measuring apparatus. It mediates the measured signals and delivers them in digital form to the evaluation software.

BK CONNECT software from Brüel & Kjaer mediated the whole measurement, recording, and post-processing of the data.

3 Computational Approach

In parallel with the experiment, a computational model was created in FEM-based software. In the software environment, a CAD model with validated dimensions was initially created. In the calculations, the material properties of the given structure were considered, which were partially modified in the initial phase by comparing the modal shapes from the measurement, measuring of dimensions, and mass. Further, the computational modal analysis was followed by a harmonic analysis in the structural and acoustic domains. The numerical models were created based on boundary conditions considered from a technical experiment [4].

3.1 Harmonic Response in Structural Domain

At the very beginning, the modal and harmonic analysis was interconnected in the computational model. A modal superposition method could be used during the solution process.

The finite element model was created by using the solid186 and solid187 elements. Based on the sensitivity analysis, two elements were considered through the thickness of the plate. The final discretized geometry model contained approx. 3200 elements in the structural analysis.

The structure in the analyses was not bound by any constraints—free boundary condition. The force load was applied—the force by which the structure was excited was used from the laboratory testing. See position *p. F* in Fig. 1, where the force transducer was placed in technical measurement and in parallel the location where the force was applied in the numerical model. The damping parameter was also applied from the modal experimental analysis. The damping ratio was included in the computational model.

Evaluation of the normal acceleration in points based on a technical experiment was given.

3.2 Harmonic Response in Acoustic Domain

The structure-acoustic analysis linking approach was chosen when creating the acoustic model. Creating a 3D model—the rectangular plate model has been cut into the sphere volume that formed the acoustic space. Furthermore, only the acoustic space was used in whole analysis. Creation of FE mesh—in the model discretization, the minimum element size $L_{\min}$ [mm] was chosen in acoustic model according to (1) from [5]:

$$L_{\min} = \lambda/5 = c/(5 * f_{\max}), \tag{1}$$

where λ [m] is wavelength,

c [m/s] is the speed of sound,

$f_{\max}$ [Hz] is considered frequency maximum in analysis.

In the acoustic analysis, the fluid220 and the fluid221 elements were used, which are a high order 3-D solid elements that exhibit quadratic pressure behavior. The final discretized model contained approx. 93,500 elements in the acoustic analysis.

Boundary conditions—in Fig. 3 are shown the boundary conditions used in the acoustic numerical model. On the left side of the picture, the import of surface normal velocities from structural harmonic analysis is depicted. In the middle part of the model, a specific envelope was applied. From this surface area, the acoustic features outside of the FEM model were approximated. An important boundary condition of the "Infinite elements" was applied to the outside surface of the acoustic space. In

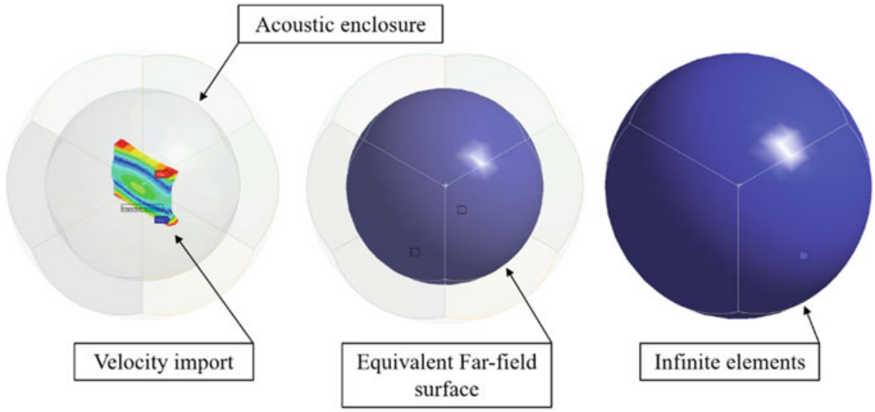


Fig. 3 The acoustic computational model

this way, free propagation of acoustic waves into space was ensured—simulation of fully anechoic chamber.

The microphones were placed according to the experimental measurement.

4 Evaluation of the Results

Through both approaches: by technical experiment and numerical model, it is possible to compare and validate results among themselves.

By the modal analysis, the eigenmodes of the structure were defined. This means its eigenfrequencies, eigen shapes, and modal damping (from lab measurement). In Table 1, two eigen shapes are compared. According to the pictures, it is possible to define and assign a custom shape from experiment and numerical calculation based on color scaling. The percentage difference in own frequencies did not exceed half a percent, which can be considered as very accurate compatibility.

After the evaluation of modal properties, the harmonic analysis was performed. At specified points on the vibrating structure, normal acceleration values were recorded and evaluated. In Table 2, the acceleration amplitudes are compared. From the acoustic point of view, the sound pressure levels (SPL's) were recorded by two microphones in a technical measurement. These values were subsequently validated by a computational model. A comparison of SPL's values is shown in Fig. 4. On the radar chart, the different curves of SPL's are shown around the structure. From the FEM model is possible to compare sound pressure in planes to see the maximum values depending on the eigen shape. In Table 3, the acoustic pressure in two different planes is compared.

Table 1 Eigen frequencies

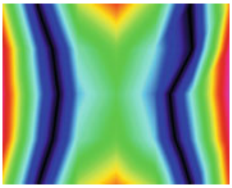
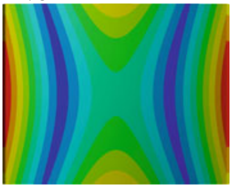
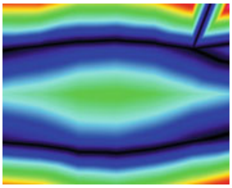
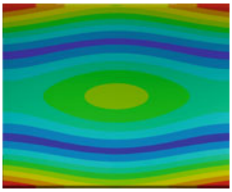
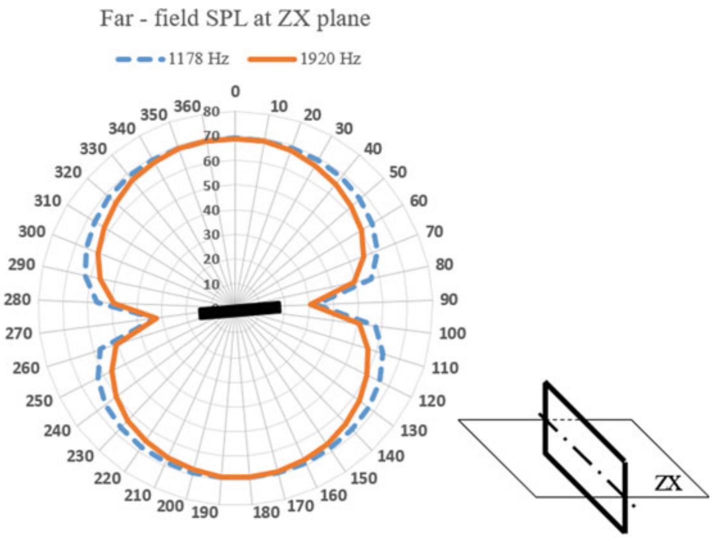
Mode	Experiment	Ansys	Difference
2	1180 Hz 	1178 Hz 	−0,17%
3	1926 Hz 	1920 Hz 	−0,31%

Table 2 Frequency response in the structural domain at selected points for 2nd and 3rd mode shape

Frequency response [m s^{-2}]													
1178	Point	p1	p2	p3	p4	p8	p9	p10	p11	p15	p16	p17	p18
	Exp	26.3	0.6	21.8	29	33.1	8.48	12.9	20.4	36.6	11.8	9.16	17.1
	FEM	29.6	0.2	23.6	32.6	37.5	8.5	13.9	22.5	40.8	11.8	10.5	18.9
	Diff (%)	12.6	−63.5	8.2	12.5	13.4	0.0	8.1	10.2	11.4	−0.1	14.2	10.7
1920	Point	p1	p2	p3	p4	p8	p9	p10	p11	p15	p16	p17	p18
	Exp	38.5	31.2	25	22.3	0.285	2.21	6.79	9.09	16.4	18.8	23.2	25.7
	FEM	37.9	32.7	27.1	27.1	1.4	3.0	7.6	9.8	16.8	19.1	23.4	25.4
	Diff (%)	−1.7	4.7	8.5	21.6	390.5	35.7	12.3	7.3	2.7	1.6	0.9	−1.0

5 Conclusion

The vibroacoustic properties of the simple structure were analyzed by technical measurement and numerical simulation. Modal properties were evaluated in the first phase. Based on the comparison of results from two approaches, the material model was partially modified. In the subsequent phases, the frequency responses of the structure in the structural and acoustic domains were evaluated in precisely specified locations. The results from numerical simulations were mainly influenced by load condition and damping factor, which was obtained from modal measurement. The differences between technical measurement and FEM model were compared in percent in this paper. Based on the results the numerical method can be used to



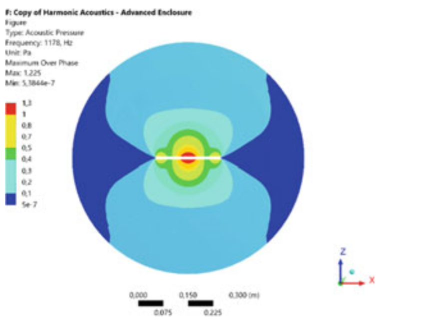
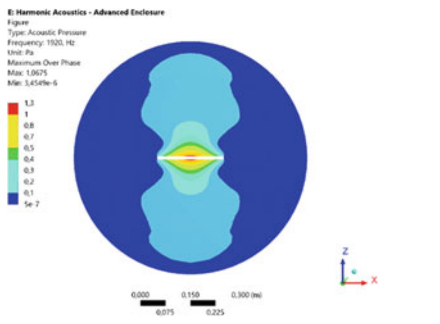
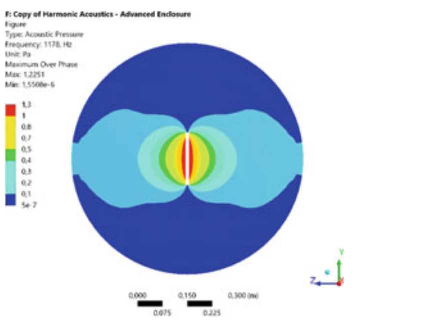
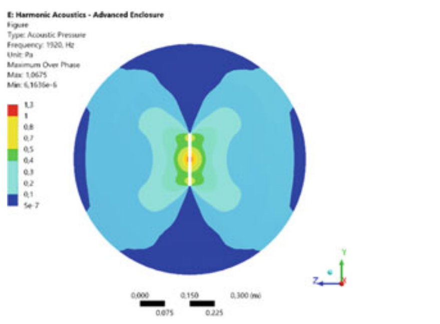
SPL Comparison [dB]						
	Experiment		Ansys		Difference [%]	
Mode	Mic1	Mic2	Mic1	Mic2	Mic1	Mic2
2. mode shape	67,8	70,48	68,288	67,783	0,72%	-3,83%
3. mode shape	69,6	67,1	68,43	63,742	-1,68%	-5,00%

Fig. 4 Sound pressure level

evaluate the acoustic behavior of structure after correlation with technical experiment. Without validation of process, the numerical simulation can show the benefit of change, but without correct values.

Compared to [4], some modifications were applied in acoustic computational model to increase the accuracy. Greater emphasis was placed on the creation of the FE "mesh" based on the size of the element relative to the investigated frequency range. Another change was the boundary condition "Equivalent Far-field Surface", which more effectively and more accurately determines the SPL's outside the "acoustic mesh". This methodology will be used for other structures, which is more complex [6].

Table 3 Acoustic pressure

1178 Hz—*ZX* plane	1920 Hz—*ZX* plane
	
1178 Hz—*ZY* plane	1920 Hz—*ZY* plane
	

Acknowledgements The research leading to these results has received funding from the project FV 18-56 granted by specific university research of the Brno University of Technology, Faculty of Mechanical engineering. The authors gratefully acknowledge this support.

References

1. V. Misun, in *Vibrace a Hluk*. (Academic Publishing House CERM, s. r. o., Brno, 2005). ISBN: 80-214-3060-5
2. D.J. Ewins, in *Modal Testing: Theory and Practice*. (Research studies Press, Letchworth, England, 1984)
3. R. Nový, Hluk a chvění, České vysoké učení technické, Praha (2009)
4. R. Zajac, K. Řehák, A. Prokop, Investigation of the harmonic response on the simple structure in structural and acoustic domain by experimental and numerical approach, in Transport Means 2018. Transport Means. (University of Technology, Kaunas, 2018), pp. 669–672. ISSN: 1822-296X

5. R.D. Blewins, in *Formulas for Natural Frequency and Mode Shape*. (Krieger Publishing Company, USA, 2001). ISBN: 9781575241845
6. J. Tůma, Transmission and gearbox noise and vibration prediction and control, in The Sixteenth International Congress on Sound and Vibration (Krakow, 2009)