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VIBRO-ACOUSTIC ANALYSIS OF A SPHERICAL JOINT USING FEM

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ABSTRACT

Vibro-Acoustic analysis gives the possibility of obtaining the level of sound pressure depending on the frequency, as well as the evaluation of the acoustic pressure in a point of interest.

The monitoring can be done using a microphone placed in an area of acoustic sensitivity.

This paper highlights the possibility of monitoring the level of acoustic pressure in the vicinity of some structural elements that can vibrate under the action of external excitations.

The studied assembly is a spherical joint. All the changes in the acoustic pressure level on the control direction of this assembly can indicate structural changes at the level of the monitored joint, thus highlighting possible defects.

KEYWORDS: vibro-acoustics using FEM, spherical joint, dynamic response

1. Introduction

Acoustics is the study of generation, propagation, absorption, reflection of the sound pressure waves in a fluid medium, and some applications for this analysis include [8-10]:

- design of spikers, acoustic filters;
- underwater acoustics;
- machinery.

Vibro-acoustic analysis is used in:

- noise reduction in machinery;
- noise reduction in buildings;
- noise reduction in vehicles;
- acoustic filters.

Typical quantities of interest in vibro-acoustic analysis are:

- the pressure distribution in a fluid (air) at different frequencies - sound pressure level;
- transmission, attenuation and dispersion of acoustic waves.

A coupled acoustics analysis takes into account the interaction fluid-structure [2, 5-7].

This is the case of vibro-acoustic analysis.

The 3D vibro-acoustic calculus using Ansys Workbench requires [1]:

- define acoustics properties;
- apply acoustic boundary conditions and specific loads;
- assigning acoustic-mechanics structure interfaces;
- post-process acoustic results.

Fluid-structure interaction (FSI) is a term for acoustic and structural body interacts via boundary conditions that couples them.

The formation of coupled equation of motion for FSI is shown below [3].

The finite element equation of motion for acoustic pressure is:

$$[M_f]\{\ddot{p}\} + [K_f]\{p\} = \{F_f\} \tag{1}$$

$[M_f]$: Equivalent fluid mass matrix;
$\{\ddot{p}\}$: Vector of the second derivative of acoustic pressure with respect to time;
$[K_f]$: Equivalent fluid stiffness matrix;
$\{p\}$: Vector of unknown nodal acoustic pressures;
$\{F_f\}$: Vector of applied fluid loads.

The finite element equation of motion for structural elements is:

$$[M_s]\{\ddot{U}\} + [K_s]\{U\} = \{F_s\} \tag{2}$$

$[M_s]$: Equivalent structural mass matrix;
$\{\ddot{U}\}$: Vector of the second derivative of displacements with respect to time;
$[K_s]$: Equivalent structural stiffness matrix;
$\{U\}$: Vector of unknown nodal displacements;
$\{F_s\}$: Vector of applied structural loads.

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Additional terms and coupling matrix, $[R]$ are added to equation (1) and (2) to account for the coupling of structure and fluid.

$$[M_f]\{\ddot{p}\} + [K_f]\{p\} = \{F_f\} + \rho_0[R]^T\{U\} \quad (3)$$
$$[M_s]\{\ddot{U}\} + [K_s]\{U\} = \{F_s\} + [R]\{p\} \quad (4)$$

Matrix equation is then formed from equation (3) and (4) with added structural damping, $[C_s]$ and acoustic damping, $[C_f]$ effects.

$$\begin{bmatrix} M_s & 0 \\ \rho_0 R^T & M_f \end{bmatrix}\begin{Bmatrix} \ddot{U} \\ \ddot{p} \end{Bmatrix} + \begin{bmatrix} C_s & 0 \\ 0 & C_f \end{bmatrix}\begin{Bmatrix} \dot{U} \\ \dot{p} \end{Bmatrix} + \begin{bmatrix} K_s & -R \\ 0 & K_f \end{bmatrix}\begin{Bmatrix} U \\ p \end{Bmatrix} = \begin{Bmatrix} F_s \\ F_f \end{Bmatrix} \quad (5)$$

Sound Pressure Level or SPL is defined as follows:

$$L_p = 20\, log\left(\frac{p_{rms}}{p_{ref}}\right)$$

$$p_{ref} = 20 \times 10^{-6}(Pa)$$

2. Vibro-acoustic modeling of spherical joint using fem

The paper highlights the possibility of monitoring the acoustic pressure level in the vicinity of some structural elements that can vibrate under the action of external excitations.

The change in the acoustic pressure in the control direction of the assembly can indicate structural changes at the level of the monitored joint, thus highlighting possible defects (gaps of inadmissible sizes).

Monitoring can be done with the help of a microphone placed in an area of acoustic sensitivity.

Vibro-Acoustic analysis gives the possibility of obtaining the distribution of the acoustic pressure depending on the frequency, as well as the evaluation of the acoustic pressure in a point of interest.

The structure whose vibrations are of interest is a spherical joint presented in reference [4]; the spherical head of a steel rod is made of Structural Steel and the bearing is made of Teflon / HDPE polyethylene.

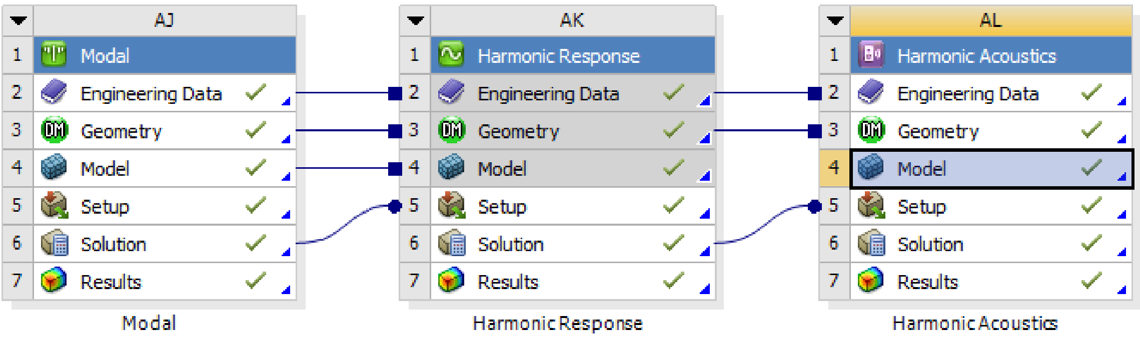

***Fig. 1.** Calculation modules for Vibro-Acoustics analysis in Ansys Workbench*

The stages of work with the related modules and the results of each module are presented in the diagram from Fig. 2; the chaining of modules in Ansys Workbench with the sharing of submodules is shown in Fig. 1.

For this purpose, the frequencies and modes of vibration are calculated in a determined range 0-10000 Hz (using MODAL module of Ansys Workbench), then with the help of the Harmonic Response module, the dynamic response of the structure to the external excitation is determined, and finally with the help of the Harmonic acoustics module the pressure distribution is obtained in the considered air enclosure.

In the Geometry submodule, the structure whose vibration is of interest is created and an area of suitable shape and size is defined around the structure, an enclosure that will be considered full of air and will become active only in the HARMONIC ACOUSTICS module (Fig. 3a).

For enclosure, a non-uniform cushion box with the dimensions given in the Fig. 3b is used.

The properties of the materials for the forced vibration analysis are given in Table 1, and the properties for the acoustic simulation (for air) are given in Table 2.

In the MODAL module, in the Model submodule, the boundary conditions for the free vibrations of the structure are imposed, obtaining the natural frequencies and natural modes in the chosen frequency range 0-10000 Hz (Fig. 4).

These results are transferred to the HARMONIC RESPONSE Module; in order to obtain the forced

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vibration, a constant force excitation on the X control direction and 5% damping are imposed.

As results, the speeds and accelerations depending on the frequency can be obtained, results that will be imported into the Acoustic Region in the HARMONIC ACOUSTICS module, the only mode in which the Air Enclosure becomes active, delimited in this module by the Radiation Boundary.

The air enclosure mesh is Adaptive with resolution 7.

Boundary conditions for the MODAL analysis are represented in Fig. 5; the horizontal arm is fixed at the end and the vertical arm has imposed frictionless support at its end.

Harmonic response corresponds to a constant force excitation applied on the steel head of the spherical joint on the command direction (Fig. 6).

Table 1. *Properties for Vibro simulation*

Materials	Mass density (kg/m^3)	Young modulus in compression (Pa)	Poisson ratio
Structural Steel (for structure)	7850	2.1e^11	0.3
Teflon (for structure)	2160	482e^6	0.42

Table 2. *Properties for Acoustic simulation*

Material	Mass density (kg/m^3)	Sound speed (m/s)
Air (for enclosure)	1.2	343

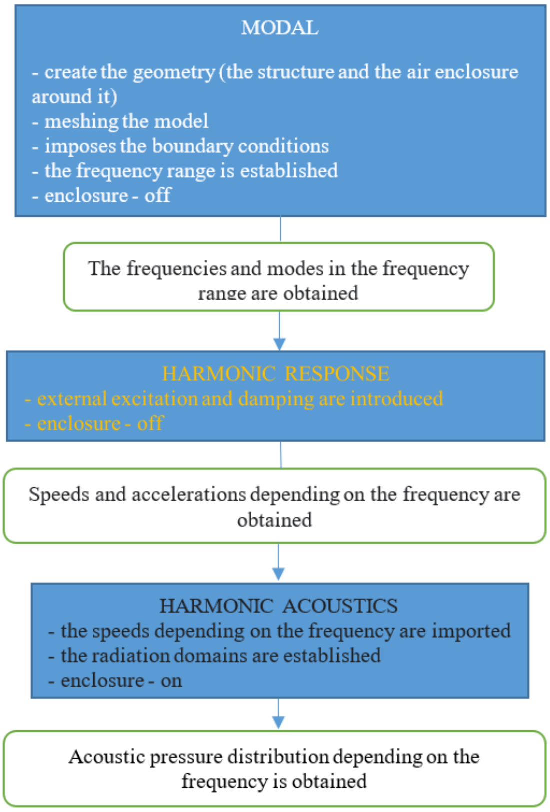

Fig. 2. *The working stages in vibro-acoustic analysis and the results obtained in each stage*

The type of element used in discretization of the structure is SOLID187 [12] (number of nodes 7485, number of elements being 1981).

SOLID187 element is a higher order 3-D, 10-node element. SOLID187 has a quadratic displacement behavior and is well suited to modeling irregular meshes (such as those produced from various CAD/CAM systems). The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element has plasticity, hyper-elasticity, creep, stress stiffening, large deflection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastoplastic materials, and fully incompressible hyper-elastic materials [12].

The enclosure was discretized in 100808 elements of FLUID221 type - see reference [7].

FLUID221 is a higher order 3-D 10-node solid element that exhibits quadratic pressure behavior. This type of element is used for modeling the fluid medium and the interface in fluid-structure interaction problems.

Meshing of simulation model for acoustic analysis should be sufficiently fine to capture the mode shapes of the model.

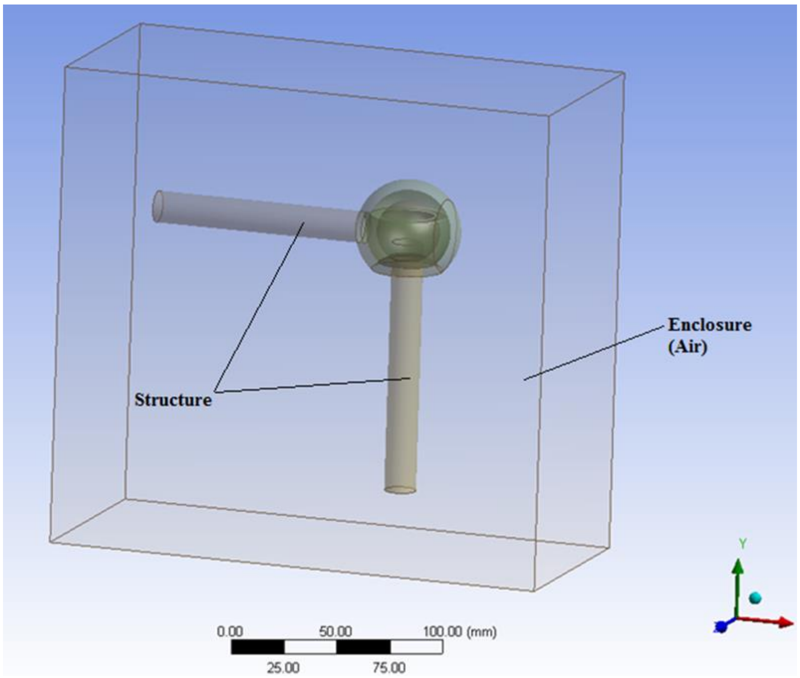

Fig. 3a. *Geometry of the system*

Details of Enclosure1	
Enclosure	Enclosure1
Shape	Box
Number of Planes	0
Cushion	Non-Uniform
FD1, Cushion +X value (>0)	30 mm
FD2, Cushion +Y value (>0)	50 mm
FD3, Cushion +Z value (>0)	30 mm
FD4, Cushion -X value (>0)	30 mm
FD5, Cushion -Y value (>0)	30 mm
FD6, Cushion -Z value (>0)	30 mm
Target Bodies	All Bodies
Export Enclosure	Yes

Fig. 3b. *The dimensions of the enclosure*

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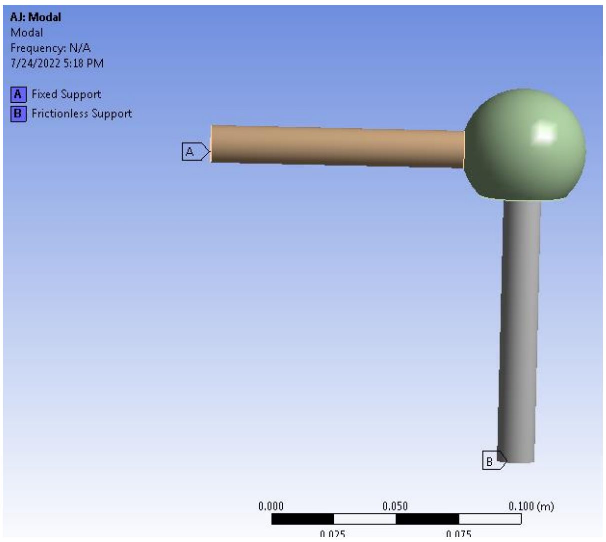

Fig. 4. *Boundary conditions*

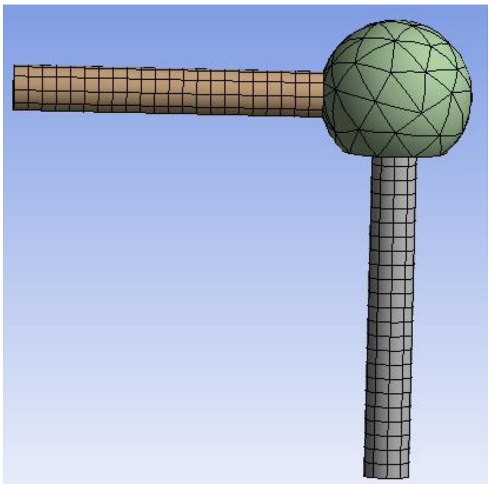

Fig. 5. *The mesh of structure*

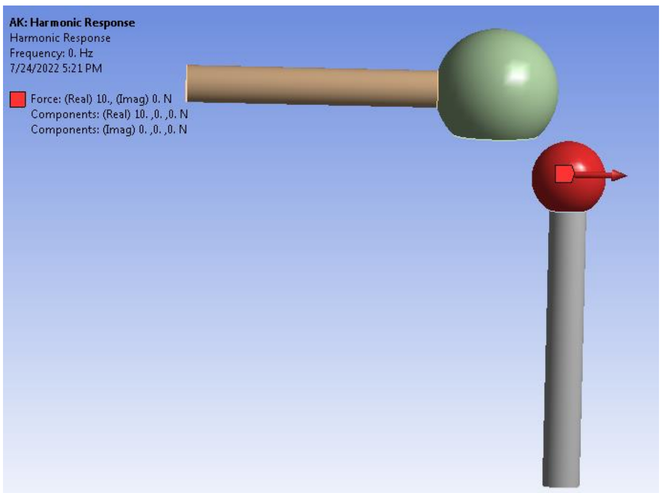

Fig. 6. *Harmonic excitation*

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3. Results and discussions

In Fig. 7 and Fig. 8 are the representations of the speed amplitudes as functions of the frequency, respectively of the accelerations as functions of the frequency (Bode diagrams).

These diagrams show the frequency of 2000 Hz and the corresponding sweeping phase angle for speeds and accelerations.

Fig. 9 shows the distribution of imported speeds at the frequency of 186 Hz.

Fig. 10 gives the sound pressure level distribution (for excitation Force = 10 N) in a section; in the Fig. 11 the distribution of sound pressure level corresponds to the excitation Force = 100 N.

In Fig. 12 is given the Microphone position (-50 mm, -50 mm, 0 mm) for Far field SPL Mic evaluation.

In Fig. 13 is represented comparatively SPL Mic vs Frequency for excitation force of 10 N and of 100 N; the SPL (sound pressure level) charts are similar, only the sound pressure level is different. The position of the Microphone is -50 mm, -50 mm, 0 mm.

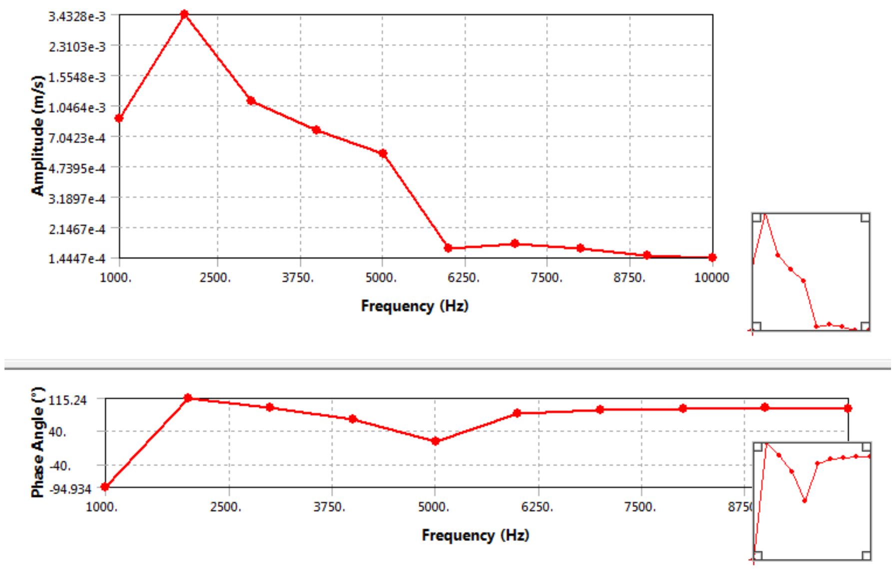

Fig. 7. *Velocities vs frequencies*

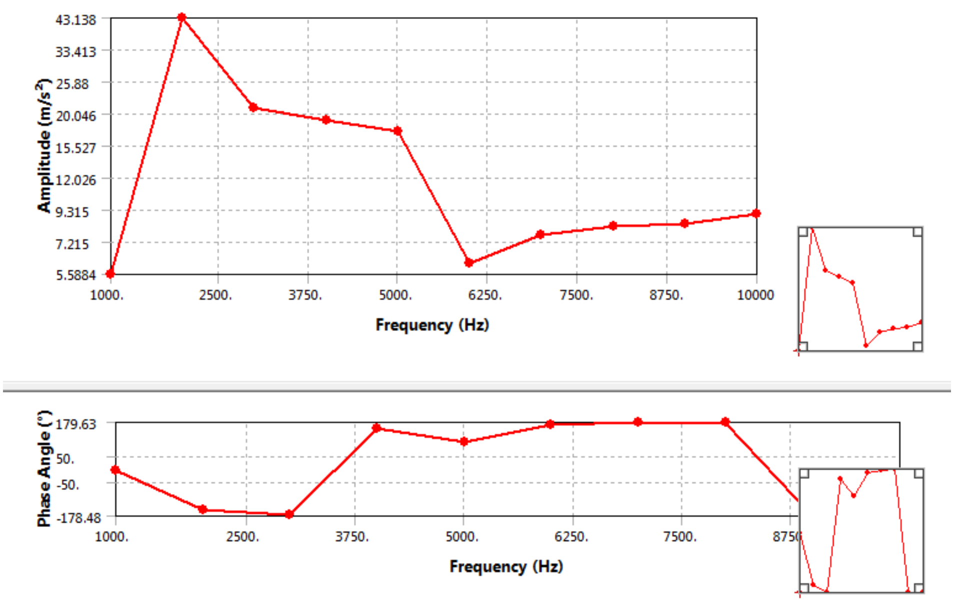

Fig. 8. *Accelerations vs frequencies*

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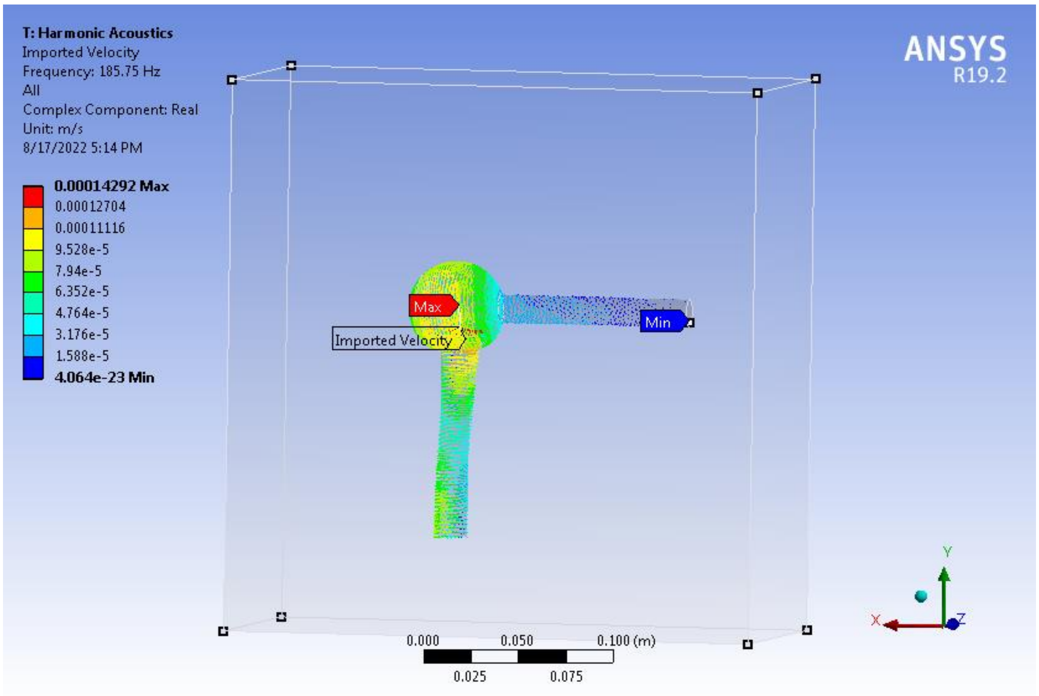

Fig. 9. *Distribution of imported velocity*

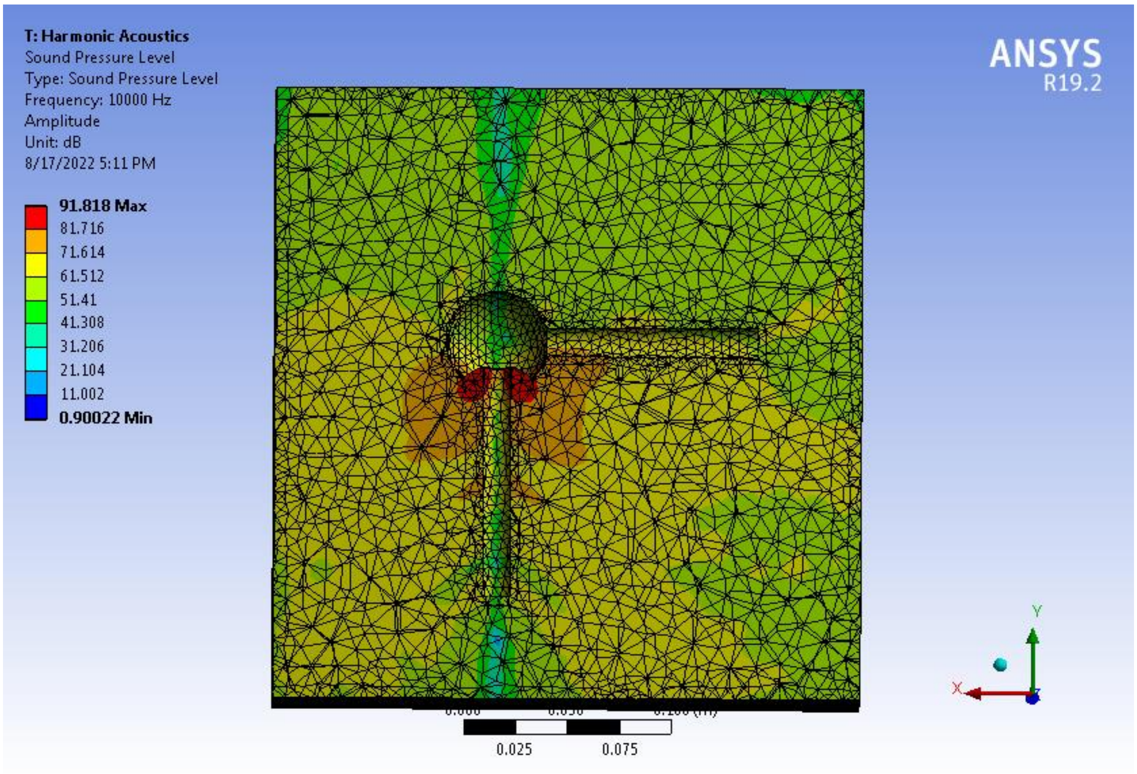

Fig. 10. *Sound pressure level (excitation Force = 10 N) - section in the middle of the enclosure*

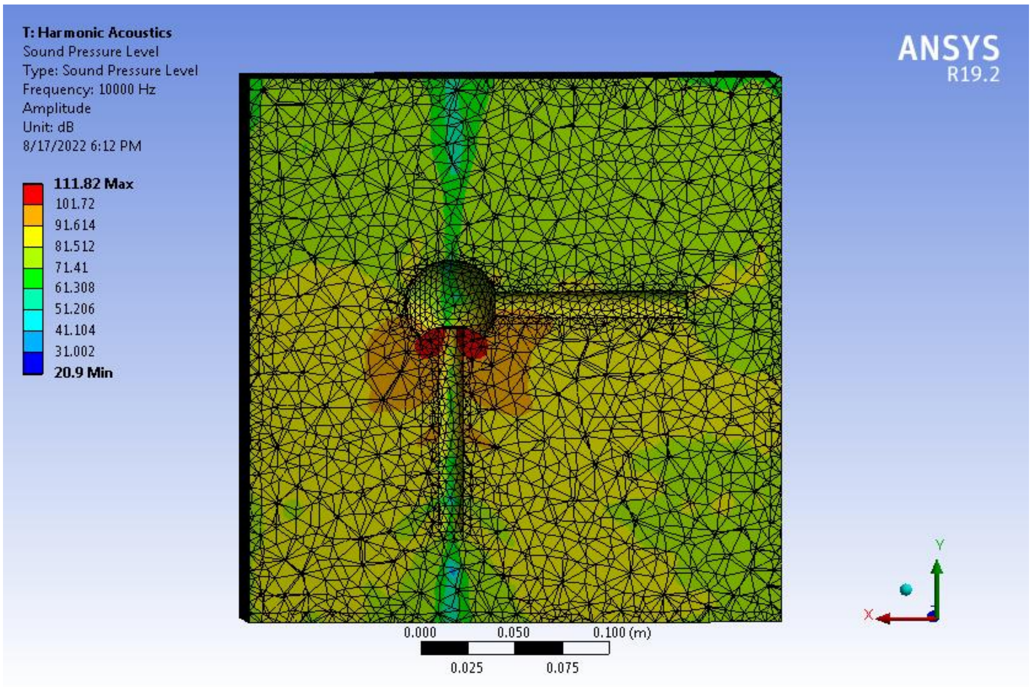

Fig. 11. *Sound pressure level (excitation Force = 100 N) - section in the middle of the enclosure*

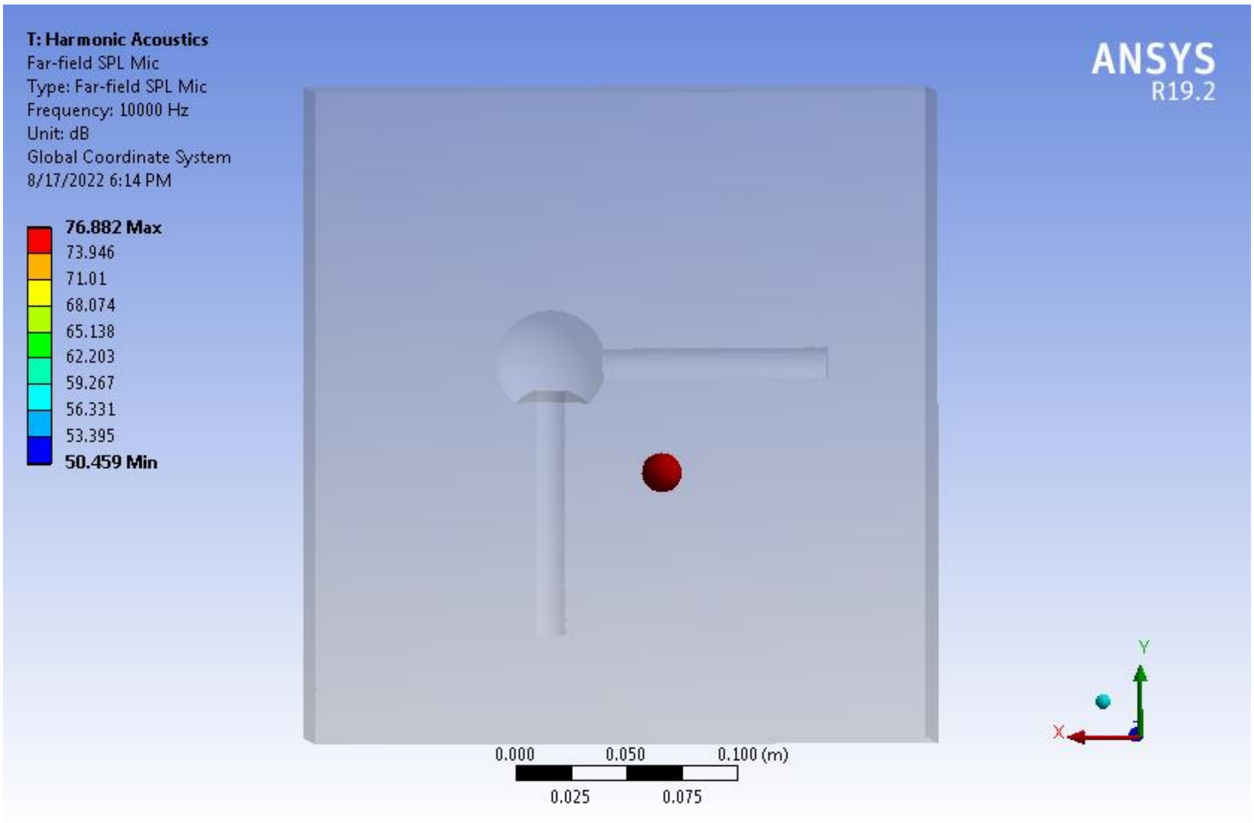

Fig. 12. *Microphone position -50 mm, -50 mm, 0 mm*

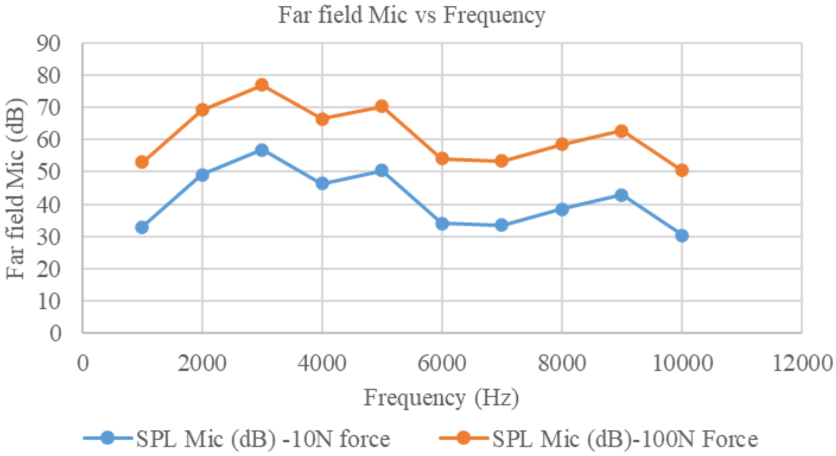

Fig. 13. *SPL Mic for Excitation in the position -50 mm, -50 mm, 0 mm for excitation force 10 N and 100 N*

4. Conclusions

Figures 10 and 11 show that the acoustic pressure level depends on the excitation intensity; in Fig. 10 the acoustic pressure level corresponds to the exciting force F = 10 N, and in Fig. 11 the acoustic pressure level corresponds to the exciting force F = 100 N.

For a certain position of the microphone and different excitations, the variation curves of the SPL are similar, but the acoustic pressure level depends on the intensity of the excitation force (Fig. 13).

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