

Studies On The Effects Of Interfering Structural Vibrations On Underwater Acoustic Vector Sensor

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Abstract- In this paper, we have reported the FEM studies that analyse the effect of unwanted interfering structural vibrations on the performance of an underwater Acoustic Vector Sensor (AVS) using COMSOL Multiphysics simulation tool. The acoustic particle velocity of the medium is generally measured indirectly using a tri-axial accelerometer based AVS which is also sensitive to the unwanted mechanical vibrations transmitted to it through the suspending structure. Thus, it is essential to understand the effects of these transmitted vibrations on the AVS, especially when it is mounted on a moving platform like an autonomous underwater vehicle (AUV). The simulation results are presented and insights are drawn.

Keywords: Acoustic Vector Sensor, Finite element method.

1. Introduction

An AVS is a compact sensor that measures a four-dimensional vector quantity consisting of the acoustic particle velocity of the water medium in three dimensions and the scalar pressure. It is primarily used to sense the acoustic intensity vector and estimate the direction of one or more radiating acoustic sources in the medium. The acoustic particle velocity of the medium is generally measured indirectly using a tri-axial accelerometer based AVS which is also sensitive to the unwanted mechanical vibrations transmitted to it through the suspending structure. It is essential to understand the effects of these transmitted vibrations on the AVS, especially when it is mounted on a moving platform like an autonomous underwater vehicle (AUV). This could also reduce the effective SNR and thus can introduce deviation in the measurement of the direction of acoustic sources. Especially it is not desired when these vibrations overlap with the frequencies of the desired acoustic source signal and may impose extra challenges. Thus it is vital to study how potential vibration patterns are transmitted from the suspending platform to the accelerometer embedded in the AVS. The inferences will allow to design appropriate isolation mechanisms to reduce mechanical coupling between the AUV platform and the AVS to prevent the unwanted vibrations resulting in more reliable AVS measurements.

Section II describes the methodology. Simulation setup and details are provided in section III. The results and inferences are given in section IV. The section V concludes the paper.

2. Methodology

2.1 Accelerometer based AVS design

In figure 1, the tri-axial accelerometer based underwater AVS is shown. It consist of a tri-axial accelerometer encased by a syntactic foam sphere which ensures neutral buoyancy. The outer casing fulfils the purpose of withstanding the hydrostatic pressure. In between these two layer is the gap which is filled with visco-elastic rubber which acts as a acoustically transparent medium and allows the signal to pass with very insignificant attenuation. Further, there is neck with connecting edge with holes. This provide the passage or wires to data aquisition unit. The suspending structure is connected to AVS through the Flanges screwed with the O-ring inbetween them. The suspending structure mainly consisted of rods with flange termination for mounting the AVS to platforms like AUV.

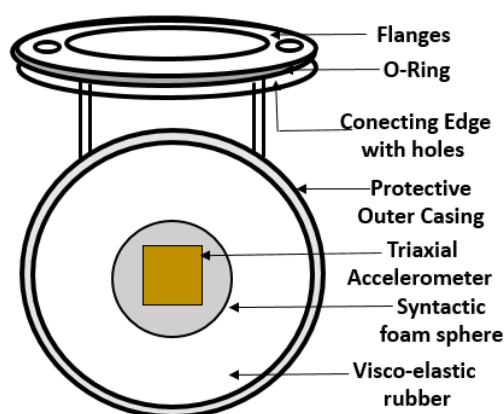


Figure 1. Tri-axial accelerometer based underwater AVS.

2.2 Background Theory

The frequency range of the interfering structural vibrations can vary depending on the platform considered for mounting. For example, for AUV as the mounting platform it will depend on the factors like design, material properties, the propulsion systems, operational conditions and the equipment it

carries within it. For our analysis purpose we have considered the range 10-1000Hz which is usually covers the vibrations sources from the propulsion system , machinery and some resonant frequencies of suspending structure. Eigen frequency analysis of AVS with and without suspending rod structure is carried out for better understanding of the resonant frequencies and mode shape present.

3. Simulation Setup

To formulate this problem in COMSOL Multiphysics, the Acoustics and Structural Mechanics modules are used to model the interaction of different physical phenomena. Two distinct studies are merged: one that performs a structural vibration analysis of the mounting structure along with the AVS, and another wave propagation analysis that focuses on acoustic analysis to compute the pressure and acceleration measurements by the AVS deployed in water medium.

3.1 Structural vibration analysis

We have created a 3D geometry of the AVS as shown in figure 2. In the model we have not included the full rod that is used to mount the AVS, rather for simulation purpose its end portion having the flange connector with two holes only is considered.

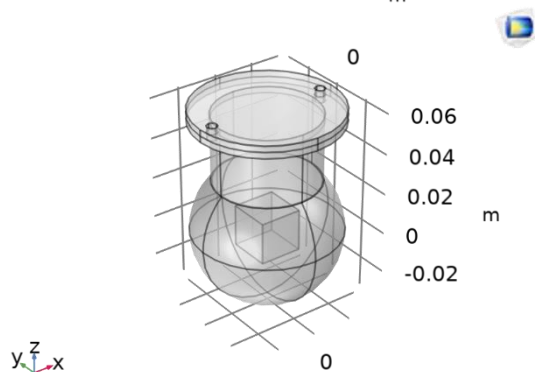


Figure 2. 3D geometry of the AVS model

Next the material properties such as density, speed of sound, Young's modulus, Poisson's ratio, etc. are defined for each domain for accurate mechanical behavior representation. Then we set up the physics. For structural vibration analysis, we employed Solid Mechanics interface. The upper plate of flanges is considered to be fully constrained to represent the fixed connect to the mounting platform. And the two holes in it are modelled with rigid connectors. For meshing, we have used free tetrahedral mesh discretize the geometry to ensure accurate results and meshed modal is shown in the figure 3.

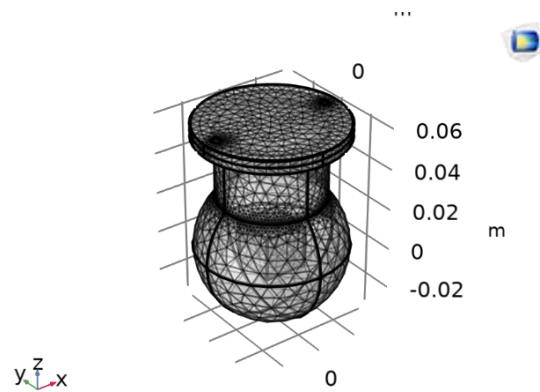


Figure 3. 3D geometry of the AVS meshed

For structural vibration analysis, a predefined study available in COMSOL Multiphysics is being used, which consists of the eigenfrequency analysis that generates a reduced order model (ROM) based on which further computations are done. Also, we have created the model including suspending rod of length 50cm that will be used to mount the AVS to the platform via flanges and performed its eigenfrequencies analysis. Figure 4 shows the 3D geometry modal and meshed geometry.

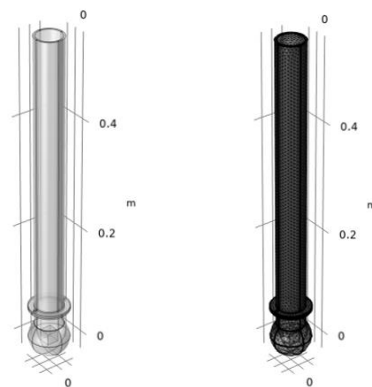


Figure 4. AVS with suspending rod structure: 3D geometry (left) and meshed geometry (right)

3.2 Acoustic analysis

For acoustics, we have used Pressure acoustic physics interface in the time domain. In geometry, the structure shown in figure 2 is further enclosed by a sphere of radius 1m. The sphere is filled with water. The figure 5 and 6 shows the complete 3D model and meshed model considered for acoustic analysis. To make it a far-field environment, Perfectly Matched Layer (PML) is used, which acts like an absorbent material for acoustic waves. Acoustic solid interaction is utilized here and the acoustic structure boundary multiphysics coupling node is used to defined the coupling between the pressure acoustic and the solid domain.

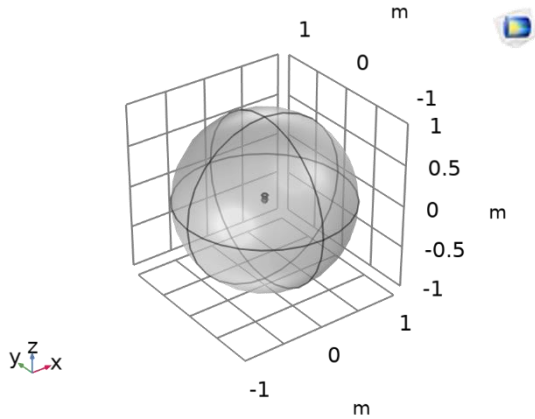


Figure 5. 3D geometry of the AVS and its suspending structure.

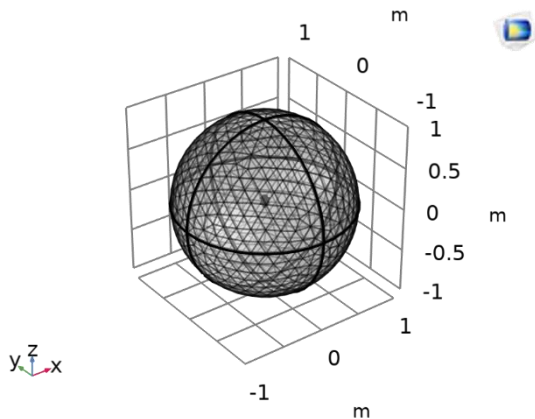


Figure 6. 3D geometry of the AVS meshed.

A background pressure field of 1Pa is applied in the frequency range 10 to 1000Hz. Table 1 lists the parameters used for simulation.

Table 1: Simulation Parameters

Parameters	Values
Dimension of accelerometer	0.0203 m
Density of Accelerometer sphere	2640
Time of study	0.004 s
Sampling Frequency	50000 1/s
Radius of AVS sphere casing	0.0185 m
Density of AVS material	1210
Frequency of operation	10 to 1000 Hz
Speed of sound in water	1500 m/s
Elevation angle (theta)	90 deg
Azimuth Angle(phi)	0 deg

4. Simulation Results and Discussion

Figure 7-10 shows the results for eigenfrequency analysis for both the geometries with and without rod under undamped condition. We have compute 12 eigenfrequencies out of which first two sets are presented here. For model 1 in figure 2, results lie in range 2332 to 38583.5 Hz Hz and for model 2 in figure 4, the eigenfrequencies covers the range 144.16 to 4334.3 Hz.

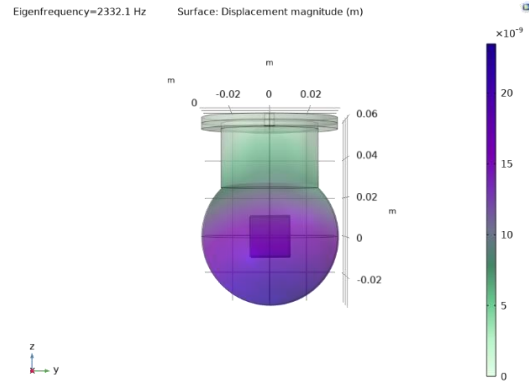


Figure 7. Modal 1: Displacement (meters) at eigenfrequency 2332 Hz.

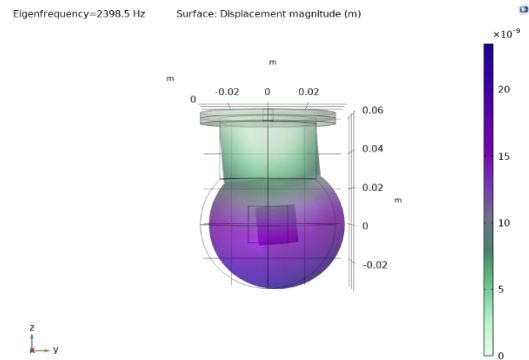


Figure 8. Modal 1: Displacement (meters) at eigenfrequency 2332 Hz.

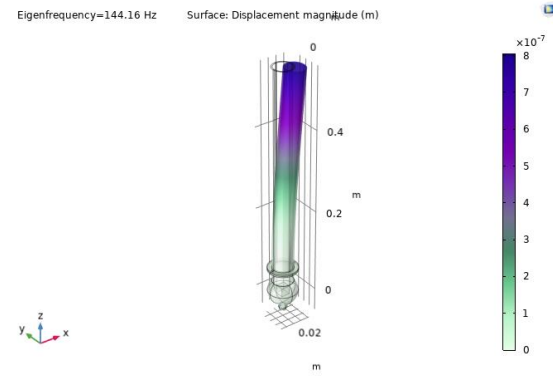


Figure 9. Model 2: Displacement (meters) at eigenfrequency 144.16 Hz.

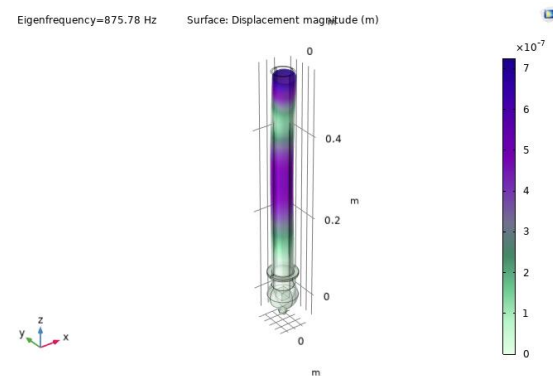


Figure 10. Model 2: Displacement (meters) at eigenfrequency 875.78 Hz.

Random excitations are introduced into the structure via two defined rigid connectors, applied as loads in the x- and z- directions. The input PSD encompasses random vibrations within the frequency range of 10 Hz to 1000 Hz. Various variations within this range are taken into account to ensure a comprehensive analysis. Figure 11-15 illustrates the power spectral density (PSD) of these input loads for various combinations considered. Specifically, considering the primary source of vibrations, the propulsion system, which typically operate at 500-2000 rpm. Thus, the corresponding frequency considered within 10 to 40 Hz. To address this scenario, the input PSD shown in Figure 16 has been utilized. Additionally, we have explored variations in the amplitude of these PSDs to observe their effects.

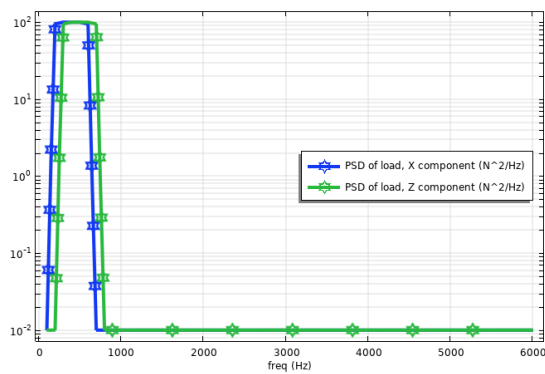


Figure 11. PSD of loads applied (N^2/Hz) (y-axis in log scale).

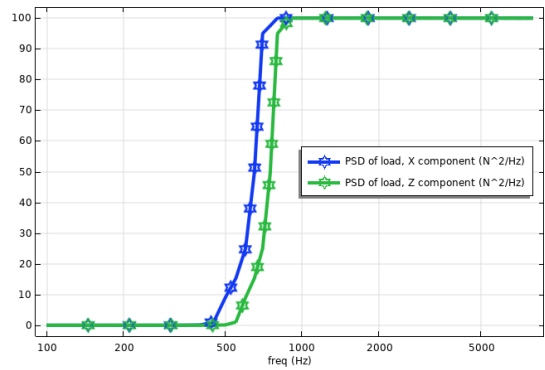


Figure 12. PSD of loads applied (N^2/Hz).

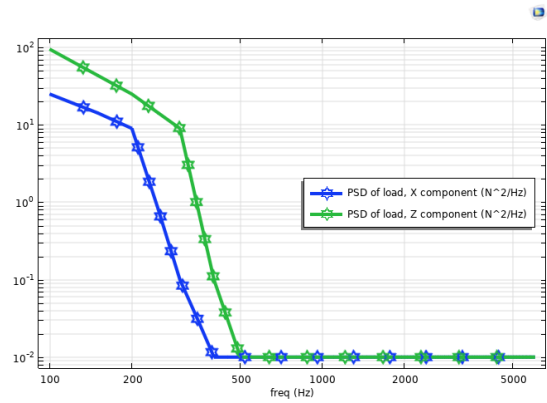


Figure 13. PSD of loads applied (N^2/Hz) (x- & y-axis in log scale).

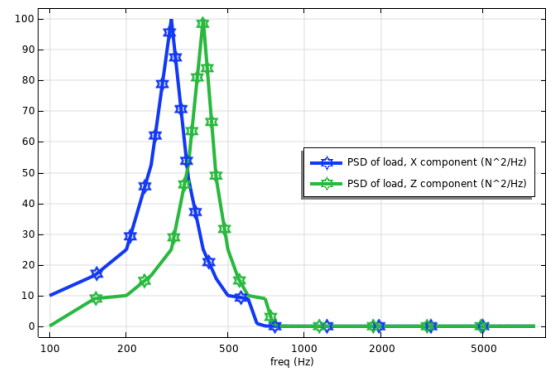


Figure 14. PSD of loads applied (N^2/Hz).

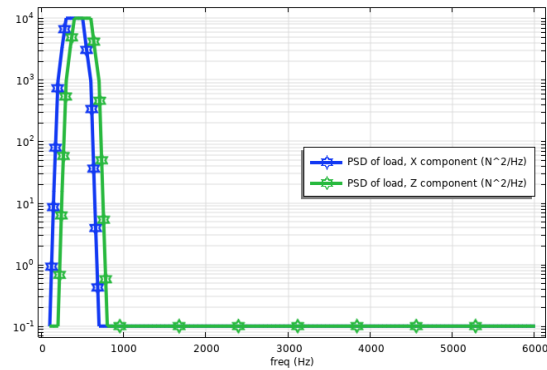


Figure 15. PSD of loads applied (N^2/Hz) for case of amplitude variation (y-axis in log scale).

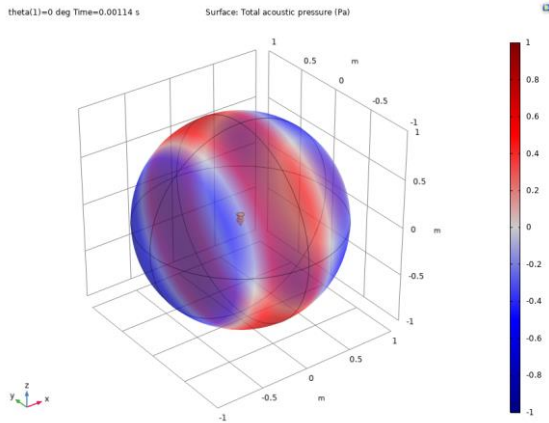


Figure 17. Total acoustic pressure field in the model (angle $\theta=0$ deg).

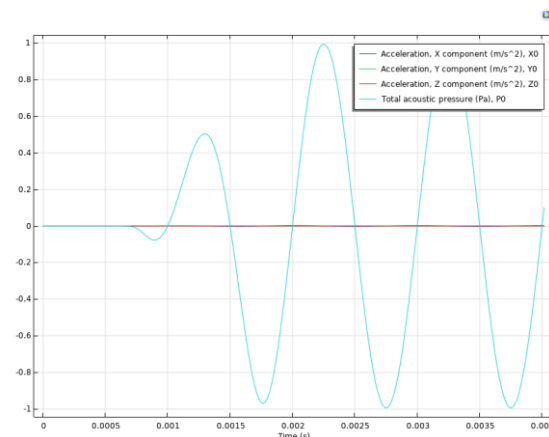


Figure 18. Plot of measurement obtained in acoustic domain.

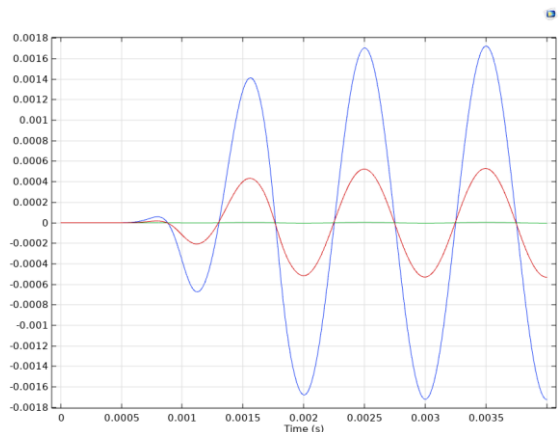


Figure 19. Plot of measurement obtained in acoustic domain. Zoomed to visualize acceleration measurement.

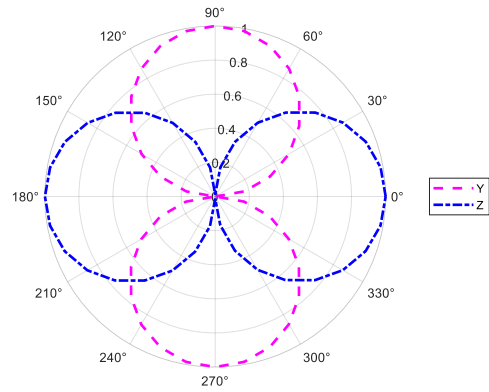


Figure 20. Beam pattern of y- and z- channel of AVS in azimuth plane at frequency $f=1$ kHz and varying angle θ from 0 to 360 deg with step size of 10 deg.

5. Conclusion

The paper presented the results of the simulation studies in COMSOL and provided insights into how to arrive at a suitable design of the AVS that reduces or not overlap with the interfering structural vibrations frequency range.

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