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Labyrinth quarter-wavelength tubes array for the reduction of machinery noise

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Abstract. Anthropogenic noise from navigation is a major contributor to the disturbance of the acoustic soundscape in underwater environments. The noise generated by ship's machinery exhibits energetic tonal harmonic peaks at multiples of the rotating and firing frequency, that occur in the 20-200 Hz frequency range and difficult to control with classical soundproofing materials. Quarter wavelength tubes (QWT) can be a concrete solution since their absorption peaks are harmonic odd integers of the first resonance frequency. The main issue of QWT is their tuning length, which equals 1.43 m for a 60 Hz resonator. The problem is solved by coiling the tube into a labyrinth. Three labyrinth quarter wavelength tubes are tuned respectively at 60, 90 and 120 Hz. Samples are printed with filament 3D additive manufacturing techniques using PLA and tested with a square impedance tube designed for low-frequency measurements. Measurement results are in good agreement with analytical and numerical predictions. An array including four 60 Hz, four 90 Hz and four 120 Hz labyrinths QWTs is finally tested.

Introduction

The most important source of noise of an internal combustion engine powered vehicle or equipment is usually the engine itself. The sound disturbance produced by a ship is dominated by machinery noise at low speed, and by cavitation and propeller noise at higher speeds.

The low frequency spectrum of the noise produced by an internal combustion engine is dominated by the engine rotating and firing frequencies and the associated harmonics [1]. Most of the internal combustion engines are four-stroke engines and those will be under consideration in this work. The explosion frequency f_{ex} of a four-stroke engine with N cylinders at a given speed rpm can be calculated following:

$$f_{ex} = \frac{N}{2}\frac{rpm}{60}, \tag{1}$$

Let's take as an example a diesel engine that is a 6 cylinder - 4 stroke engine. With symmetric firings, 6 explosions occur on a complete cycle (two crankshaft rotations, and thus three ignition events occur per crankshaft revolution). Vibration measurements were conducted on a CAT C9 diesel engine at *Innovation Maritime*. With an average rotation speed of 1800 rpm, the engine exhibits an explosion frequency $f_{ex} = 90$ Hz. This frequency and its harmonics (multiples of f_{ex}) are generally the most energetic peaks in the noise and spectrum spectra. The two other important noise generation mechanisms are those associated with the rotation frequency of the crankshaft (in this case, $\frac{rpm}{60} = 30$ Hz) and those generated by the ignition imperfections or unbalance on a complete cycle (typically half of the rotation frequency, $\frac{1}{2} \cdot \frac{rpm}{60} = 15$ Hz). In Figure 1, significant peaks are identified each 15 Hz with a dominance at f_{ex} and its harmonics. This frequency spectrum is common to diesel engines used in the marine industry. Quarter wavelength tubes could be an efficient solution to limit this disturbance. Indeed, a quarter wavelength tube (QWT) is an open-closed tube (Figure 2a) that has resonant frequencies f_{qwt} when the tube length L is an odd-integer multiple of the quarter of the acoustic wavelength $\lambda = c_0/f$:

$$f_{qwt} = \frac{(2m-1)c_0}{4L}. \qquad \boldsymbol{m = 1, 2, 3 \ldots} \qquad (2)$$

At their successive harmonic resonances, large sound absorption can be achieved, above all for the first harmonics. This behavior is well adapted to the defined problem of reducing the harmonic noise generated by reciprocating engines. One issue is related to the required length for a QWT. To cope with this, the QWT is stretched into a labyrinth (Figure 2b) or following a spiral path, depending on the application, which guarantees similar sound absorption properties compared with a straight tube.

Theory and numerical implementation for labyrinth resonators

The acoustic impedance of a labyrinth resonator (LR) is studied according to the analytical approach proposed by Magnani et al. [2], where the labyrinth resonator is evaluated as a perforated plate followed by a QWT. The QWT of length $\boldsymbol{L}$ has an impedance equal to $\boldsymbol{Z_{QWT} = -jZ_{eff}\cot(k_{eff}L_{eff})}$, based on Low Reduced Frequency theory (LRF) [3], which studies the sound wave propagation with a lossy Helmholtz equation which takes into account viscous and thermal dissipation by modelling the effective density $\boldsymbol{\rho_{eff}}$ and speed of sound $\boldsymbol{c_{eff}}$, and consequently the effective impedance $\boldsymbol{Z_{eff} = \rho_{eff}c_{eff}}$ and the effective wavenumber $\boldsymbol{k_{eff} = \omega/c_{eff}}$. $\boldsymbol{L_{eff} = L - (4-\pi)\frac{d}{2}(n-1)}$ is the effective length to tune a labyrinth resonator, with $\boldsymbol{n}$ number of branches and $\boldsymbol{d}$ width of the channel [4].

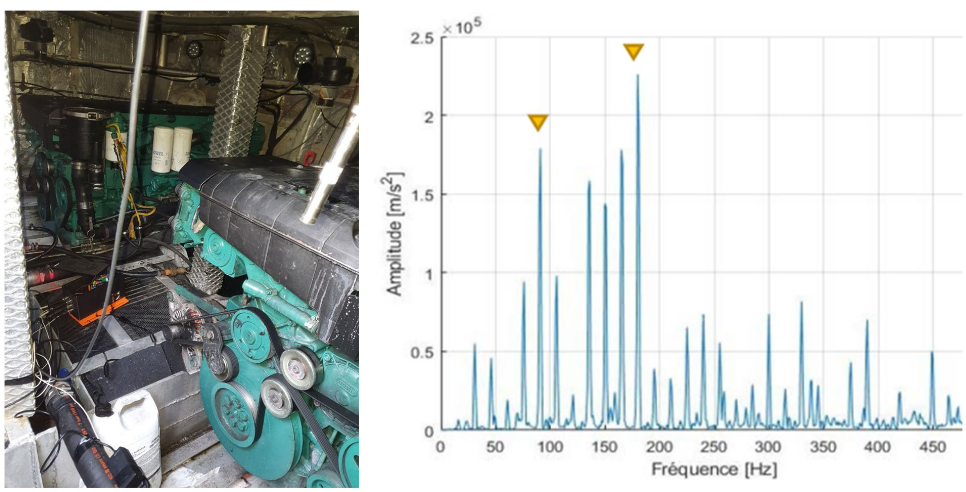

Figure 1: Result of a vibration measurement conducted on the crankshaft axis of a C9 diesel engine. The yellow triangles indicate the firing frequency of the pistons (90 Hz), as well as its first harmonic (180 Hz).

The plate of thickness $\boldsymbol{t_d}$ with the square inlet hole of side-length $\boldsymbol{d}$ is defined through Johnson-Champoux-Allard (JCA) approach [5]. The impedance of the labyrinth resonator $\boldsymbol{Z_{LR}}$ is:

$$Z_{LR} = \frac{1}{\phi_{inlet}}\left[Z_{d,JCA}\frac{-jZ_{QWT}\cot\left(k_{d,JCA}t_d\right) + Z_{d,JCA}}{Z_{QWT} - jZ_{d,JCA}\cot\left(k_{d,JCA}t_d\right)}\right], \quad (3)$$

where $Z_{d_{JCA}} = \rho_{JCA}c_{JCA}$ and $k_{d_{JCA}} = \omega/c_{JCA}$ are respectively the impedance and the complex wavenumber of the perforated plate, with $c_{JCA} = \sqrt{K_{JCA}/\rho_{JCA}}$, ρ_{JCA} and K_{JCA} effective speed of sound, density and bulk modulus. $\phi_{inlet} = A_{hole}/A_{plate}$ is the perforatio ratio between the hole area and the plate area. Labyrinth resonators are positioned inside a box to have several peaks at the desired frequencies. According to the electro-acoustic analogy, this implies that the impedance of the global system Z_{tot} and the sound absorption coefficient α_{theory} are respectively:

$$Z_{tot} = \left[\sum_{i=1}^{N}\frac{1}{Z_{LR,i}}\right]^{-1} \qquad \alpha_{theory} = \frac{4Re(Z_{tot}/Z_0)}{|Z_{tot}/Z_0|^2 + 2Re(Z_{tot}/Z_0) + 1} \quad (4)$$

The sound absorption of the box is experimentally evaluated with a 1 ft x 1 ft impedance tube: a speaker placed at one end of the tube excites it with a normal plane wave radiation, and the sample is placed at the opposite end, backed by a rigid wall. Two microphones separated by a distance s evaluate sound pressure in the tube, with P_2 at a distance x_2 respect to the sample, and P_1 at a distance $x_1 = x_2 + s$. The sound absorption coefficient under normal incidence is estimated according to the ISO 10534-2 1998 standard. Numerical simulations mimic this experimental measurement. Analyses are made with COMSOL Multiphysics, *Pressure Acoustics Module*, with the impedance tube and labyrinth walls considered as rigid.

Figure 2: a) Quarter wavelength tube excited by a plane wave radiation (PWR); b) labyrinth resonator excited by a PWR.

Results

Three labyrinth resonators are modelled with their fundamental resonance peak at 60, 90, 120 Hz, and 3D printed to form a box with multiple resonators to have high sound absorption at 60, 90, 120 and 180 Hz. The latter is the second harmonic of the 60 Hz labyrinth. Their height (thickness of the sample) is fixed at 100 mm, to place them in a limited space. The 60 Hz labyrinth has lateral dimension of 97 mm times 97 mm, the 90 Hz has lateral dimension of 145 mm times 49 mm, and the 120 Hz has lateral dimension of 73 mm times 73 mm. A wooden box is designed to include all the resonators is a single unit of 1 foot by 1 foot area, that can be directly installed inside the impedance tube (of 1 foot by 1 foot square section), with four 60 Hz, four 90 Hz and four 120 Hz LRs (Figure 3a). In Figure 3b), the experimental sound absorption coefficient of the box with multiple resonators is plotted, comparing it with theory and simulation. Resonances are clearly distinguished even for experiments, with the sound absorption at each peak always bigger than 0.7

up to 500 Hz. Each tone is at its precise tuned frequency. In addition, their combination gives non-zero absorption in the entire frequency range studied.

Conclusion

The multi-resonator box shows convincing sound absorption properties and opens interesting perspectives. Experimental results are compared with analytical and numerical methods: the experimental peaks appear at the predicted frequencies, with an amplitude difference that can be attributed to the rigid wall hypothesis. Future developments will be pursued with an experimental campaign, in a water basin or ideally in the engine cabin of a ship.

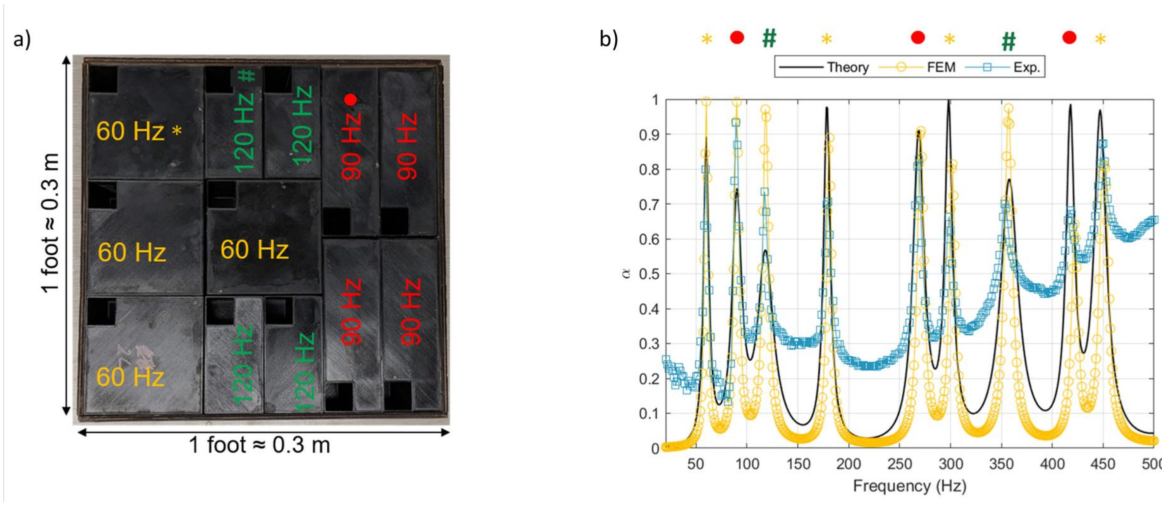

Figure 3: a) Picture of the multi-resonators box; b) Sound absorption of the labyrinth resonator box excited by a normally incident plane wave – comparison of analytical, numerical, and experimental results.

Acknowledgments

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