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Acoustical Performance of Helmholtz Resonators Used as Vehicular Silencers

This study proposes to evaluate the acoustical performance of a reactive vehicular silencer made entirely of Helmholtz resonator. Four or seven resonators are idealized to be in a series configuration, all the dimensions except cavity length and resonators are fixed. An algorithm processes inputted characteristic engine noise signal obtained from literature, identify peaks, and calculates ideal cavity length for attenuation. The transmission loss of the system is analytically calculated. Attenuation levels obtained are satisfactory. The required volume of the resonator to achieve the same resonance frequency as low frequency noise peak demands impractical cavity length, proving it to be flawed. It is suggested to use resonators of two or three different diameter for different frequency range in order to overcome this problem.

Keywords: *Helmholtz resonator, vehicular exhaust system, noise attenu–ation, muffler, reactive silencer.*

1. INTRODUCTION

The vehicular exhaust system reduces both exhaust noise and pollutants, influencing the power charac–teristics of the engine. The usual construction is shown in Figure 1, and is composed of an exhaust manifold, a catalytic converter, an exhaust-gas center silencer and an exhaust-gas end silencer, which are all connected by pipes [1].

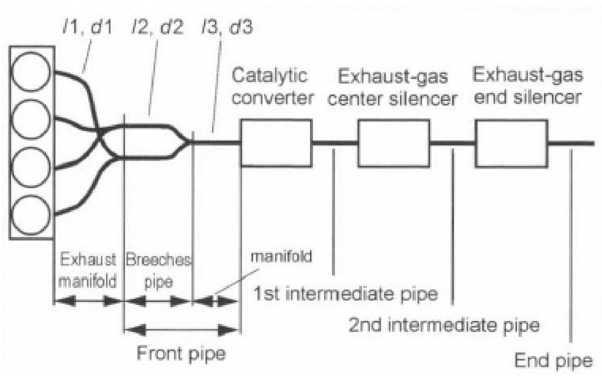

Figure 1. Schematic construction of an exhaust system. [1]

Noise from internal combustion engine can be classified into three groups: tonal noise, impulsive noise and flow noise. Exhaust flow noise is in flow noise category, and the exhaust system is responsible for between 25% and 35% of total noise from different noise sources, including engine, intake system, fan and cooling system, transmission and tires. [2-5]

Although there are some patents regarding active noise control (ANC) systems for engine generated noise [6, 7], the majority of vehicles use a combination of dis-sipative (also known as absorptive) and reactive silencers. The ANC system principle is noise cancelling by repro-ducing a sound wave at the same frequency and ampli-tude as the original noise, but 180 degrees phase shifted. Figure 2 shows a possible configuration for active noise control. It is possible to see microphones, loudspeakers and a control unit being used to identify the conditions of the system and reproduce the cancelling sound waves.

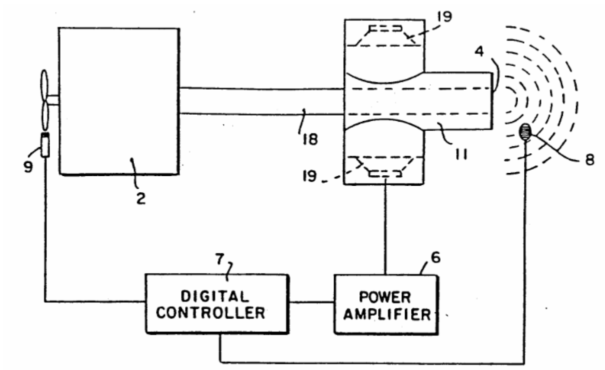

Figure 2. A patented active noise control configuration. [4]

Absorptive silencers use fibrous packing materials to fill the volume between the jacket and the perforated pipe where the gas is guided through. When gas expands into the absorption area, most of the vibration energy is attenuated by friction and converted into heat. In reactive silencers, changes in cross section or partitions cause the sound to divert, and the suppression happens when sound waves extinguish each other along two paths of different length, by being 180 degrees out of phase [1].

The Helmholtz resonator (HR) is a type of reactive muffler: a resonance-capable spring-mass system which consists of a rigid-walled cavity of volume V acting as an acoustical spring and a neck of cross section S and length L, working as the mass of the system.

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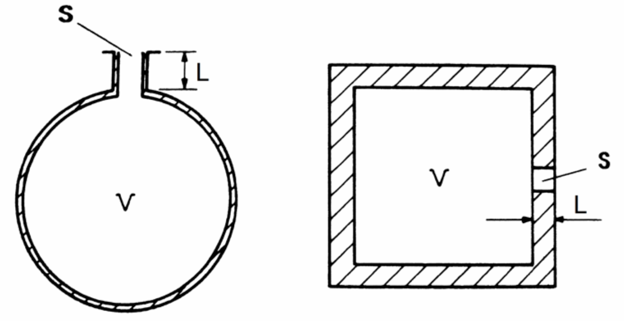

Figure 3. Two different geometry for HR. [8]

Since the fluid in the neck moves as a unit, and at low frequencies a circular opening of radius R is loaded with a radiation mass, it is necessary to find the effective length L', which is (L + 2ΔR). Therefore, it is possible to calculate the resonance frequency according to the given equation in [8-10]:

$$f_R = \frac{c}{2\pi}\sqrt{\frac{S}{V\left(L+2\Delta R\right)}} \tag{1}$$

where f_R is the resonance frequency in hertz, c is the local speed of sound in m/s, S is the cross-sectional area of the resonator in m^2, V is the resonator volume in m^3, L is the length of the neck in m, and 2ΔR the opening correction, which is approximately 1.7R for a circular opening end flanged.

Effects of the insertion of a HR in a pipe can be stu–died by the lateral opening theory. The opening causes the specific acoustic impedance (z) to change, a charac–teristic related to the medium in which the sound wave travels and its frequency, described as the ratio of acoustic pressure to particle speed [11]. When a sound wave happens to find a different z in its path, part of the incident energy (Pi) is reflected (Pr), while the other part is transmitted (Pb and Pt), as shown in Figure 4 [12].

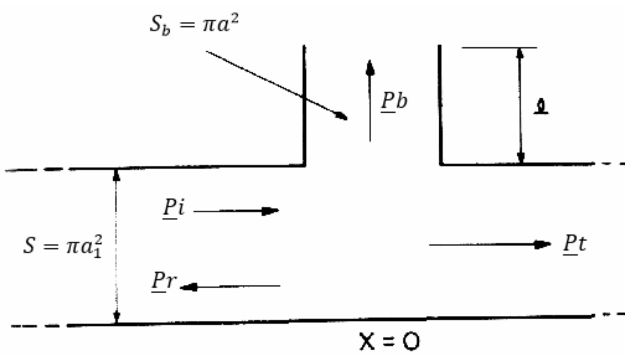

Figure 4. Representation of incident, reflected and trans–mitted acoustic energy in a pipe with lateral opening. [9]

Literature presents three independent parameters to evaluate the performance of a silencer. Insertion Loss (IL), defined as the difference between the acoustic power radiated with and without any silencer;Level Difference (LD) is the difference in sound pressure levels at two points in the exhaust and tail pipe; Transmission Loss (TL), which is independent of the source, presumes an anechoic termination and represents the difference between the power incident on the muffler and that transmitted to the anechoic termination [13]. It is possible to analytically obtain TL for a single HR, inserted on a pipe as a silencer, from the given model [12]:

$$TL = 10\log\left[1+\left[\frac{\sqrt{s_b V / L'}}{2S\left(\frac{\omega}{\omega_0}-\frac{\omega_0}{\omega}\right)}\right]^2\right] \tag{2}$$

where TL is given in decibels, S_b is the neck cross section in m^2, V the resonator volume in m^3, L' the effective length, S is the main pipe cross section in m^2, ω_0 the resonance frequency in rad/s (obtained from f_R) and ω the incident sound wave frequency in rad/s.

The fact that HR attenuation response is narrow-banded limitsits usage as a passive noise control device for broadband noise. Since it is possible to change the resonance frequency of a HR by modifying its geometry, many studies have been made toward understanding the effects of an adaptive system. Enable the resonator to tune itself according to external conditions, such as air temperature (affecting the speed of sound, and as a consequence the resonance frequency), or the input noise frequency, makes it possible to enlarge working range. Different resonator geometry and control system might be adopted, as well as the number of resonators, according to the project limitations and requirements, elevating its complexity [14-18].

The aim of this study is to analyze some possible settings for a vehicular silencer made entirely from HR analytically, considering noise from different engine speed. Since Helmholtz resonator actuates in a very specific frequency, and its effect decreases rapidly when diverging from this frequency, it is necessary to use more than one resonator in order to cover a higher number of noise peaks. The resonators will be considered in series configuration that means total noise reduction is a summation of each individual resonator attenuation. As there are many dimensions which can affect resonator tuning, it is important to set some of them up, and leave just one working variable. Both neck and resonator cavity are cylindrical, and cavity height is altered in order to achieve the same natural frequency as the noise peak frequency. A 4-resonators and a 7-resonators assembly are taken into consideration.

2. METHODS

From data available in Figure 5, it was possible to identify noise peaks for different engine speed, and their respective frequency and amplitude. For this study, it was considered values for six engine speeds: from 2000 rpm to 7000 rpm, in 1000 rpm steps. Frequency range considered goes up to 800 Hz.

Sound pressure level of peaks, given in dB, were converted to Pascal, making it possible to reconstruct the signal through wave equation in posterior stage. An algorithm was developed to simulate the acoustic attenuation performed by four or seven HR disposed in series configuration. Figure 6 shows what was idealized.

The algorithm objective is to, for a given scenario (number of resonators, gas temperature in the resonator, noise characteristics, neck diameter and length, main pipe diameter and cavity diameter), indicate the ideal cavity length, modifying its volume andsetting the reso–nance frequency to the same as the loudest noise peak. Each resonator acts on one peak.The input for a reso–nator is the output of the previous one, which will act on the new highest peak. Iteration number is equal to the quantity of resonators. Low amplitude broadband noise was added to the signal in order to make it more similar to reality, then signal wave is converted to spectrum using native FFT function.

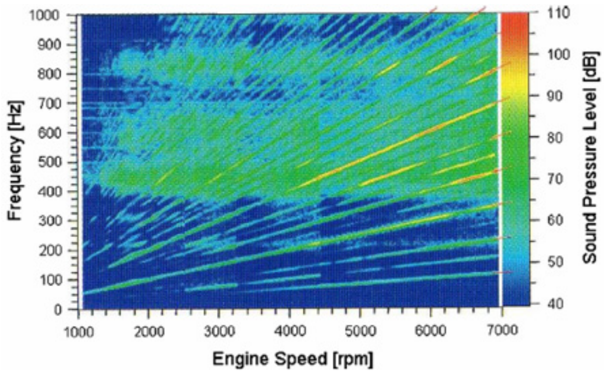

Figure 5. Waterfall plot relating engine speed, noise frequency and sound pressure level. [18]

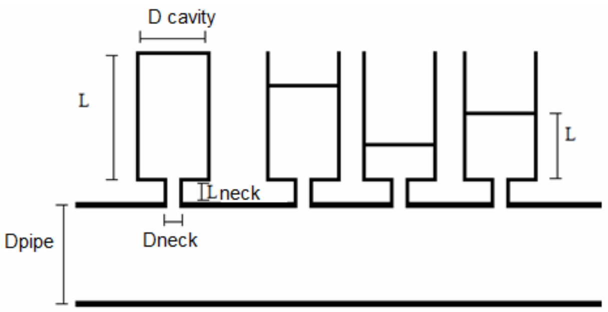

Figure 6. Representation of the 4-HR configuration.

The following values are invariable in this experi–ment: temperature is set to 500°C [12], main pipe diameter is 63.5 mm, neck diameter is 7 mm, neck length is 30 mm and cavity diameter is 40 mm. After the peak frequency isidentified, and considering the inva–riable dimensions, the algorithm finds the ideal cavity length. From the obtained dimensions and noise spect–rum, Equation 2 is used to calculate the transmission loss and graphic response is given.

3. DISCUSSION AND RESULTS

Sound pressure level (SPL) described here are all global values for each engine speed. Noise level tends to get higher as speed increases, to the point that an attenuated noise from 5000, 6000 and 7000 RPM are still higher than noise before attenuation for lower speeds. Table 1 shows an overall view of attenuation results for each engine speed when using four resonators.Noise reduction was more effective when engine speed is 4000 rpm, and less effective for 7000 rpm.

Although the increased resonator quantity in the configuration shown in Table 2, the minimum attenuation value has not changed drastically. For this configuration, lowest attenuation happens at 2000 RPM and the highest at 3000 RPM.

Table 1. Total attenuation for each engine speed, 4-resonator setting.

4 HR			
Engine speed (RPM)	Initial SPL (dB)	Final SPL (dB)	Attenuation (dB)
2000	74.4	66.0	8.4
3000	90.1	73.5	16.6
4000	90.1	70.8	19.2
5000	100.8	84.5	16.3
6000	110.8	95.6	15.2
7000	115.8	107.8	8.1

Table 2. Total attenuation for each engine speed, 7-resonator setting.

7 HR			
Engine speed (RPM)	Initial SPL (dB)	Final SPL (dB)	Attenuation (dB)
2000	74.4	65.8	8.6
3000	90.1	69.4	20.7
4000	90.1	69.7	20.4
5000	100.8	81.8	19.0
6000	110.8	92.6	18.2
7000	115.8	99.9	16.0

Figure 7 shows graphically the original spectrum in blue, and in red the attenuated one for 2000 RPM using four HR. It is noticeable that the resonators could suppress the major peaks and return sound pressure level of those frequencies to the same as the other freq–uencies, resulting in a global SPL of 66 dB.Comparing to figure 8, where seven resonators are used, there is no big difference in final SPL.

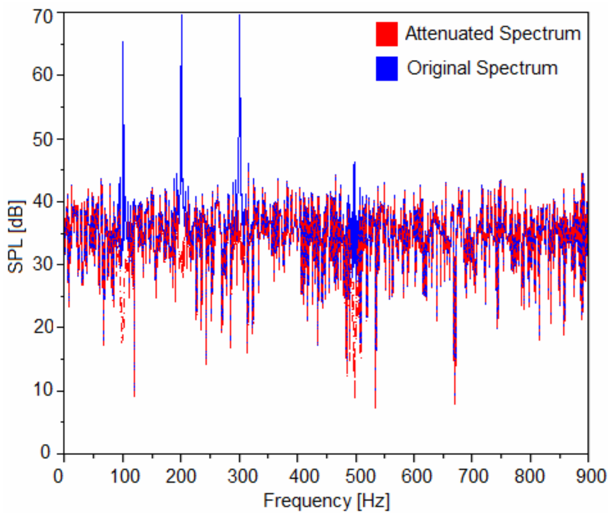

Figure 7. Attenuation at 2000 RPM, 4-HR.

Since there is no big peak left, HR actuates in frequencies that have amplitude very close to the remaining frequencies. In this case, the attenuation will have no expressive impact in global noise level, as there are many other frequencies at the same level.

The main issue encountered regards required dimen–sions, sometimes not physically compatible with a vehicular silencer. One example ofthis problem happens when engine speed is 3000 RPM, shown in Table 3. Each peak is identified by its frequency, its respective amplitude and calculated cavity length for optimal noise reduction for each frequency.

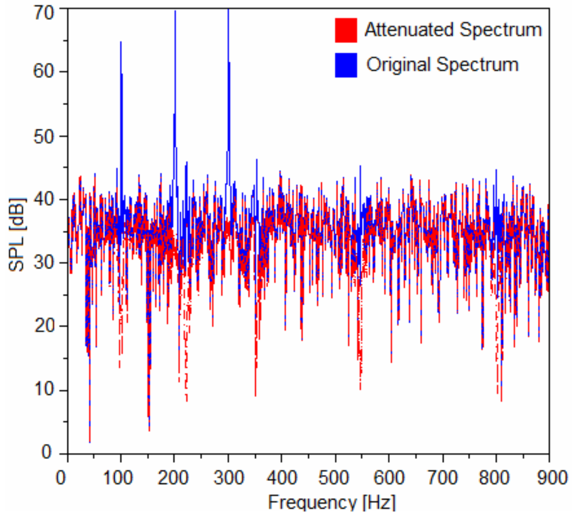

Figure 8. Attenuation at 2000 RPM, 7-HR.

Table 3. Overview for attenuation at 3000 RPM.

<table>
<tr><td colspan="6">3000 RPM</td></tr>
<tr><td>Max amplitude (dB)</td><td>(Hz)</td><td>L (mm)</td><td>(Hz)</td><td colspan="2">SPL (dB)</td></tr>
<tr><td>89.1</td><td>451</td><td>42.2</td><td>450.99</td><td>Initial</td><td>Final</td></tr>
<tr><td>69.5</td><td>301</td><td>94.8</td><td>300.99</td><td>90.1</td><td>73.5</td></tr>
<tr><td>65.3</td><td>151</td><td>376.8</td><td>150.99</td><td colspan="2">Total attenuation (dB)</td></tr>
<tr><td>59.9</td><td>76</td><td>1489.5</td><td>75.97</td><td colspan="2">16.6</td></tr>
</table>

Figure 9 presents noise spectrum for this rotation, and the resonator attenuation.

For the standard resonator dimensions used, it would be necessary 1.49 m to achieve ideal volume to attenuate 76 Hz. As a matter of fact, the lower the frequency, the higher the volume necessary, especially below 100 Hz. There is a difference of 150 Hz from 451 Hz and 301 Hz, and it is necessary 52.6 mm to change the resonance fre–quency between these two frequencies. From 151 Hz to 76 Hz there is a difference of 75Hz, and the difference in length to achieve the resonance frequency is over 1100 mm.

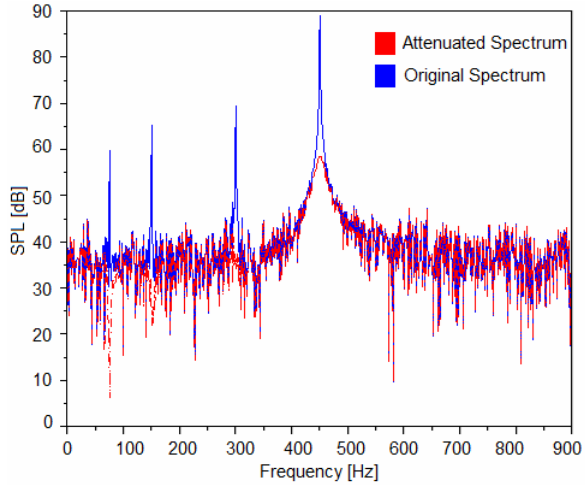

Figure 9. The four highest peaks are attenuated by HR. (@3000 rpm)

A solution proposed was to change the diameter of the cavity. Table 4 presents the effect of changing the cavity diameter from 40 mm to 100 mm for frequency considered low and high for this system.

Table 4. Required length to achieve necessary volume for different diameter.

Frequency (Hz)	D = 40 (mm)	D = 100 (mm)
76	1489.5	237.5
101	842.2	135.0
701	17.4	2.8
801	13.3	2.1

At the same time that a bigger diameter was bene–ficial for lower frequencies, it takes the higher frequen–cies to a very sensitive area, where a minor mispla–cement changes drastically the resonance frequency.

Another interesting fact that might happen depen–ding on the noise characteristics is to have the resonator to actuate on the same frequency for more than once. An example can be seen at engine speed 4000 RPM, in which there is a double attenuation around 400 Hz, in Figure 10. The attenuated peak value becomes again the highest value when the fourth resonator is actuating. The reason why it is important to attenuate the same frequency twice instead of going for a different one can be explained by the fact that sound pressure level is given in dB, which is a logarithmic scale. Differently from a linear scale, in logarithmic scale the summation behaves in a different way – the result is mostly influenced by the higher value. This being said, it is pointless to attenuate other value but the highest.

It is possible to notice that the second attenuation at 400 Hz has created two new peaks, something plausible given the attenuation shape. When using three more resonators, at some point these peaks became the highest ones, and they too were treated. Figure 11 shows when this happens.

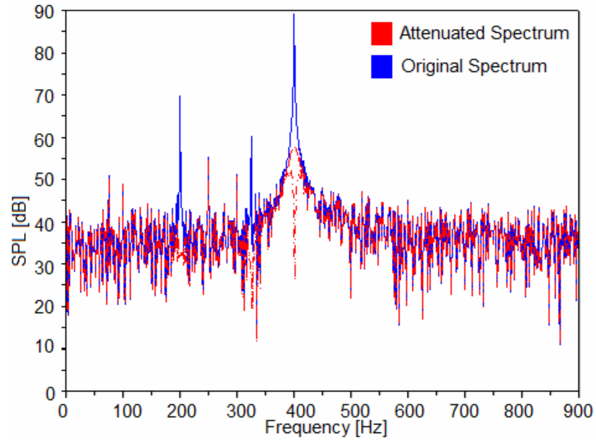

Figure 10. Double attenuation can be seen around 400 Hz. Engine running at 4000 RPM, 4 resonators.

The global noise level grows as engine speeds up. Originally 74.4 dB, the sound pressure level reaches up to 115.8 when engine is running at 7000 RPM. In this speed, the spectrum is ruled by seven very distinct peaks. Naturally, in this case four resonators cannot cover all the peaks, as shown in Figure 12, and the use of seven HR is ideal.

The difference between using four or seven resonators is almost 8 dB. The spectrum before and after attenuation is presented in Figure 13.

Unsurprisingly, as expected from a Helmholtz reso–nator behaviour, attenuation works better for lower frequencies [8], and Figure 13 makes it notable. Noise reduction at 126 Hz is visibly greater than at 801 Hz.

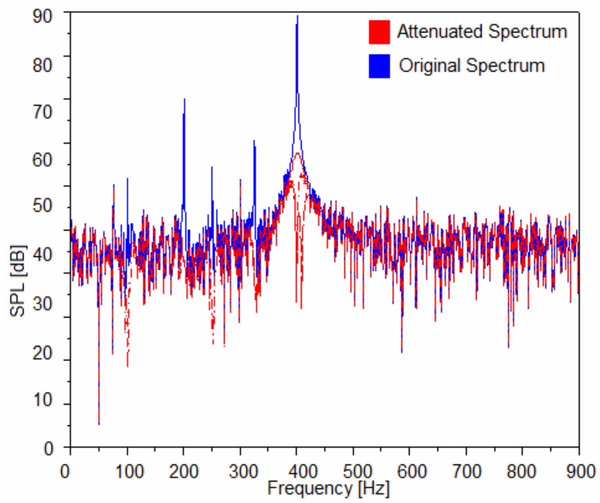

Figure 11. Double attenuation can be seen around 400 Hz. Engine running at 4000 RPM, 7 resonators.

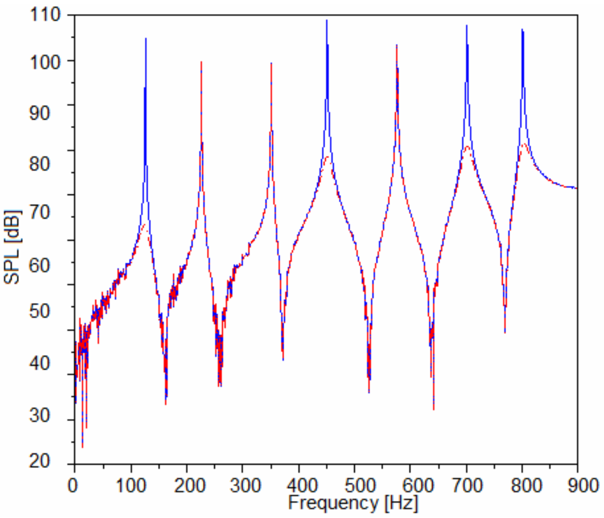

Figure 12. Attenuation at 7000 RPM, 4 resonators.

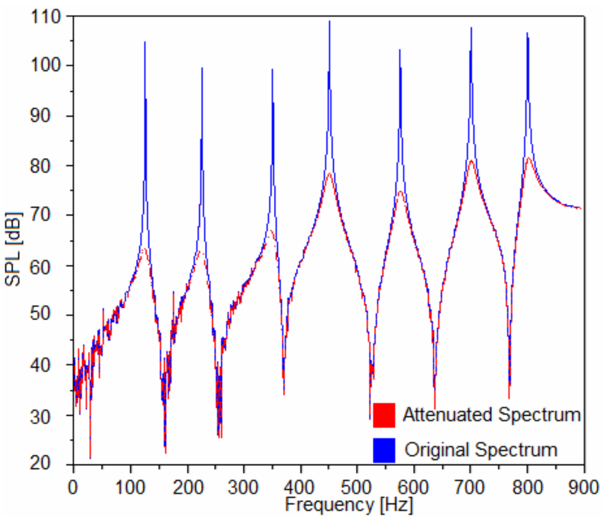

Figure 13. Attenuation at 7000 RPM, 7 resonators.

4. CONCLUSION

From the acoustics point of view, a vehicular silencer made solely from Helmholtz resonators is possible, and can achieve satisfactory attenuation level. Even though, spatial restrictions are a concern. In this matter, working with a single resonator size proved to be insufficient, so having resonators of two or three distinct diameters for different frequency ranges should fix this issue, leaving the smallest one for higher frequencies, the medium size for intermediate frequencies and the largest one for lower frequencies.

The effectiveness of the attenuation depends mostly on the noise characteristics. As a HR is a narrowband filter, it works better when it is used on peaks much higher than average noise level. The location of the peaks also affects final attenuation level: the lower the frequency of the peaks, the better, as HR does not work so well for higher frequencies. For these reasons, the increased quantity of resonators might bring expressive more attenuation power, as for 7000 RPM, where there were clearly peaks not covered by the resonators when using only four – problem solved when using 7 HR. Or it makes no big difference at all, as for 2000 RPM, when considering, it is being used 75% more compo–nents for a raise of just 0.2 dB in noise reduction.

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АКУСТИЧНЕ ПЕРФОРМАНСЕ ХЕЛМХОЛТЦ-ОВИХ РЕЗОНАТОРА КАО ПРИГУШИВАЧА НА ВОЗИЛУ

Л.Р. Мартинс, Г.П. Гуимараеш, К. Фрагаса

Ово истраживање предлаже процену акустичких перформанси реактивног пригушивача возила нап–рављеног у потпуности од Хелмхолтц-ових резона–тора. Четири или седам резонатора су идеа–лизовани да буду у серијској конфигурацији, све димензије осим дужине резонатора су фиксиране. Алгоритам обрађује карактеристичан сигнал буке мотора добијеног из литературе, идентификује врхове и израчунава идеалну дужину шупљине за атенуацију. Губитак преноса система се аналитички израчунава. Добијени нивои атенуирања су задово–љавајући. Потребна запремина резонатора за пости–зање исте резонантне фреквенције, као што је низак фреквентни шумски пик захтијева неујед–начену дужину шупљине, што доказује да је то погрешно. Предлаже се коришћење два или три резонатора различитог пречника за различите фреквентне опсеге како би се превазишао овај проблем.