Experimental study of acoustic performances of reactive engine mufflers

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Abstract. A well designed muffler means a compromise between its acoustic performance, backpressure, size and cost criteria. The acoustic performances of engine mufflers, in terms of insertion loss (IL) constitute the major aim of this paper. An experimental investigation of two manufactured mufflers with different inner configurations is realized. Insertion loss investigation is performed in an anechoic room and the experimental setup agrees the ISO 7235:2009 requirements. The analysis in third-octave band of measured noise gives us the sound pressure level on each frequency of interest. Sound attenuation and pressure results are highlighting the influence of mufflers inner configurations on their acoustical behaviour. A FEM analysis completes our study and simulations based on pressure results are performed in order to evaluate the critical displacements correlated to vibration eigenmodes. Our manufactured mufflers realize a good sound attenuation depending of their internal configuration.

1. Introduction

Over the years, the design of mufflers constitutes a subject of great interest in order to offer the balance between acoustic characteristics, pollutant emissions and fuel efficiency of an automotive engine. The most important component of the noise from the vehicle internal combustion engine appears on the exhaust system. The noisy gas pressure pulses are caused by the repeated opening of the exhaust valve, which allows the high pressure gas to be discharged [1].

Mufflers are widely used to attenuate the intake and exhaust noise. Though actual concerns in developing active noise control techniques exist [2, 3, 4], most mufflers use the traditional passive methods. There are two constructive options: dissipative and reactive mufflers (figure 1).

Dissipative mufflers are lined with sound absorptive materials and the travelling sound waves are largely absorbed. Reactive mufflers, often used in automotive, are meant to realize the sound waves reflection back to the source, thus minimizing their transmission towards the exit. A reactive exhaust muffler is made using flow reversal, zone changes (multiple expansion chambers), and resonant ramifications. Thus, this type of muffler is able to provide an adequate sound attenuation in frequency range from low to moderate bands [5, 6]. Their design considers the acoustic transmission theory either using the Helmholtz resonator principle or the expansion chambers. Besides the acoustic performance, it is important to consider the muffler backpressure caused by attaching it to the exhaust system. The influence of this high backpressure is felt through the engine efficiency decreasing and the fuel consumption increasing.
Therefore the main purpose of the exhaust muffler design is to reduce pressure loss and also the sound transmission loss over the frequency band of interest [7]. Summarizing the literature data, it can be concluded that a well-designed muffler must have an adequate insertion loss over the frequency range of interest, minimum or optimum restriction, moderate volume weight and cost, high durability, good styling and tonal quality [8].

![Automotive muffler: (a) reactive, (b) dissipative](image)

Figure 1. Automotive muffler: (a) reactive, (b) dissipative [9].

The paper presents an experimental study of two reactive manufactured mufflers with different internal configuration. They are analyzed from acoustic experimental point of view emphasizing the noise attenuation due to their configuration. CAD simulation is also realized in order to establish, as the literature say, a well-designed muffler.

2. Methodology and materials

The acoustic performances of a muffler with inside chambers and concentric or eccentric ducts are depicted by parameters as: the noise reduction (NR), the transmission loss (TL) and the insertion loss (IL). Noise reduction represents the sound pressure difference along the muffler. Transmission loss quantifies the difference between the incident and the output wave, in terms of sound power, supposing an anechoic end of the muffler. TL parameter is generally computed from models but the experimental measurements are difficult to be performed [10]. Insertion loss indicates the diminution of the sound power values behind the test object, owed to its insertion in the measurement chain. The standard ISO 7235: 2009 refers to the insertion loss determining of ducted mufflers, on frequency bands. Also, the sound power of the airflow through the ducts, the muffler’s pressure and transmission loss are the parameters discussed and included in this standardized methodology [11]. IL is very useful in industry, as it means the sound pressure difference without and with the muffler presence [10].

We have used an anechoic room for our IL measurements. The experimental setup, according to the above mentioned standard, includes a centrifugal fan with variable power providing the airflow, sound-source and sound-receiving equipments. The setup also comprises flow rate and pressure drop devices. A white noise is generated by a Brüel & Kjær sound source, and then it is amplified and transmitted using a broadband loudspeaker to the tested object, in our case the ducted reactive muffler. The sound is received by a Brüel & Kjær soundmeter via microphones and connected to data acquisition board (National Instruments). Further, the sound wave is forwarded to the LabVIEW soft to processes the acquired data. The sound pressure analysis was realized in third-octave frequency bands. The standard provides nominal frequencies (63÷8000 Hz, taking into account of the cut-off frequencies) on which the sound pressures values dB (A) can be evaluated. Thus, the sound attenuation, in terms of insertion loss, is determined for frequencies band of interest, as the difference between the measured data acquired without and with the muffler insertion in the system. This brief description of the standardized methodology and experimental setup are detailed discussed in [12].

We have realized two mufflers with three and two chambers, denoted with “muffler 1” and “muffler 2”, respectively. They are made of steel elements: muffler bodies, plates, ducts. The body of the
muffler is a cylinder with 200 mm inner diameter, 206 mm outer diameter and 400 mm length. The muffler caps and the inner plate have 200 mm diameter and 3 mm thickness.

The “muffler 1” includes three ducts (inlet, outlet and an intermediate one), with 42 mm outer diameter and 39 mm inner diameter. Inlet and outlet pipes have a length of 325 mm and the intermediate pipe has a length of 250 mm. The perforated holes of the intermediate duct have 4 mm diameter. The “muffler 2” includes two similar ducts (inlet and outlet) with 42 mm outer diameter, 39 mm inner diameter and a length of 325 mm.

Figures 2 and 3 present the specified mufflers’ manufacturing and figure 4 presents the experimental setup.

3. Results and discussion
The above procedure was used to investigate the mufflers’ insertion loss, thus highlighting their noise attenuation level. The measurements were performed for different power variations of the centrifugal fan (2V÷10V, 2V ratio) which overlap the white noise from the sound-source [11].
Figure 5. Insertion loss provided by the manufactured mufflers.

Figure 6. Comparison between the muffler’s attenuation.

Figure 7. Pressure – airflow variation.

Figure 5 presents the noise attenuation (IL) provided by both mufflers, following the procedure. It can be seen that the maximum attenuation (about 50 dB) is reach at the 500 Hz frequency for all the imposed variable flows. It is also obvious that “muffler 1” offers a better attenuation than “muffler 2”.

Moreover, this situation is highlighted in figure 6 which compares the mufflers’ sound attenuation for two situations of power variation (4 V and 10 V). It can be noted a very good sound attenuation especially in the middle frequencies (250-2000 Hz) which is a strong feature of these manufactured
mufflers, taking into account that the human ear has the most sensitivity in this range than in low or high frequencies [13, 14]. The results agree the literature, but we specify that only the flow noise attenuation performance was the subject of our study. The results correspond to 0% engine load on chassis dynamometer recordings. Some research shows that the flow sound covers the pulsation noise generated by the high speeds of engine and the reverse situation occurs for low engine speeds [15]. This will be a future work.

It is known that, in a muffler conception, attenuating the noise is linked to the backpressure increasing which diminishes the engine power [8]. Therefore, a balance noise-pressure must be kept. Figure 7 presents pressure-air flow evolution for both mufflers. "Muffler 2" allows airflow and pressure a little higher than "muffler 1", and it should be noted the generally design requirements of mufflers that is the pressure values decreasing, but keeping the flow values within the same range.

The two mufflers were modeled in the CATIA design environment, where a number of simulations were made to find out, for each of them, the pressure influence on resonance frequencies. Using the most defavorable measured pressures (320 Pa for "muffler 1" and 335 Pa for "muffler 2"), some simulations were performed in order to evaluate the critical displacements correlated to vibration eigenmodes. Material properties used for "muffler 1" and "muffler 2" are, Young’s modulus of elasticity $E=2\times10^5$ MPa, Poisson Ration $= 0.266$, material density $= 7860$ Kg/m$^3$, yield strength $250$ MPa. Constraints are applied at the inlet tube of the muffler at clamping location. In Figure 8 and Figure 9 are presented the Von Mises stress and the translational displacement for "muffler 1" and "muffler 2", respectively. First six natural frequencies are extracted by using a Block Lanczos algorithm. The first six frequencies obtained are presented in table 1. Figure 10 shows the first mode of vibration which is bending about Z axis with natural frequency 932.3Hz for "muffler 1" and 893.4Hz for the other. Obvious, due to its configuration with three chambers, "muffler 1" has lower natural frequencies than "muffler 2". The simulations are useful for exhaust mufflers designing in order to avoid their resonance.
Figure 10. Mode 1: (a) “muffler 1”, (b) “muffler 2”.

Table 1. Natural frequencies of the mufflers.

| Mode order | 1   | 2   | 3   | 4   | 5   | 6   |
|------------|-----|-----|-----|-----|-----|-----|
| Frequency (Hz) “muffler 1” | 932.3 | 937.1 | 1049.2 | 1071.0 | 1347.5 | 1571.3 |
| Frequency (Hz) “muffler 2” | 893.4 | 913.0 | 1495.0 | 1631.5 | 1837.2 | 2109 |

4. Conclusions
Our experimental investigation on manufactured mufflers acoustic performance, in terms of insertion loss, demonstrates the importance of their interior configuration. The sound attenuation difference between the two muffler models is more noticeable in the middle frequencies (250-2000 Hz) when a 16 dB gap is obtained in favour of “Muffler 1”, with three chambers. This fact emphasizes the idea of a complex interior configuration when designing a muffler. It would be advisable to include a larger number of expansion or resonance chambers so that the sound to be gradually dispersed for achieving the best possible attenuation. On the other hand, the sound attenuation is in competition with the pressure drop. The results show that a large airflow and pressure are obtained using a simpler configuration (“Muffler 2”). Also, the resonant frequencies must be known. Finally, it can be concluded that a car muffler design is an exhaustive study balancing between acoustic performance, minimal backpressure, size and cost. A further work intends to realize a specific muffler CFD analysis corroborated to experimental and theoretical aspects on engine dynamics.

References
[1] Potente D 2005 Proc. of Acoustics (Western Australia: Busselton) 153-158
[2] Anthony M, Chang G Y and Kuo S M 2017 Proc. of APSIPA Annual Summit and Conference 1-5
[3] Li K, Zheng S, Yang D and Takeharu T 2003 Tsinghua Science & Technology 8(5) 577-81
[4] Reddy K A 2017 Materials Today: Proceedings 4(8) 7313-34
[5] De Lima K F, Lenzi F and Barbieri R 2017 Appl. Acoust. 72(4) 142-50
[6] Chiu M C 2010 Appl. Acoust. 71(6) 495–505
[7] Chao S and Liang H 2017 Appl. Acoust. 116 291-6
[8] Munjal M L 2014 Acoustics of ducts and mufflers (New York: John Wiley & Sons) 416
[9] https://auto.howstuffworks.com/muffler.htm
[10] Tao Z and Seybert A 2003 A review of current techniques for measuring muffler transmission loss 10.4271/2003-01-1653
[11] EN ISO 7235: 2009 Acoustics- Measurement procedures for ducted silencers- Insertion loss, flow noise and total pressure
[12] Bujoreanu C and Benchea M 2017 MATEC Web of Conferences 112 07001
[13] Ver I L and Beranek L L 2006 Noise and Vibration Control Engineering: Principles and Applications (New York: John Wiley & Sons) 501
[14] Cory W T W 2005 Fans & Ventilation: A Practical Guide (Berlin: Elsevier) 424
[15] Yasuda T, Wuc C, Nakagawa N and Nagamura K 2013 Appl. Acoust. 74 49-57