The increasing demand for high performance recreational vehicles combined with more stringent noise regulations and customer expectations lead to intake and exhaust systems design challenges, namely increased mass flow rate, reduced pressure losses and reduced space available for silencing devices along with shortened development cycles. Design choices must be made carefully to reach the performance targets while having the desired attenuation on harmonic noise related to combustion cycles without inducing excessive pressure losses and flow noise levels. Usage of predictive tools in conjunction with validation testing is becoming paramount to guide the design of such high-performance exhaust silencing devices. A discussion regarding the available simulation methodologies is presented with the associated pros and cons. All these approaches are widespread and well documented on four-stroke automotive engines, but additional challenges arise with high performance two-stroke engines, such as non-linear behavior of perforations due to grazing flow and high acoustic amplitude in addition to extended frequency range of noise predictions. Despite these new challenges, usage of simulation tools in product development allowed to identify the root cause of a problematic resonance in an exhaust silencer and to define 2 different strategies to mitigate it. One of the two design alternatives was finally selected and successfully implemented in a mass production consumer product.

Keywords: Silencer, engine, simulation, design, acoustics

1. Introduction

The increasing demand for high performance recreational vehicles combined with more stringent noise regulations and customer expectations lead to intake and exhaust system design challenges, namely increased mass flow rate, reduced pressure losses and reduced space available for silencing devices along with shortened development cycles. Design choices must be made carefully to reach the performance targets while having the desired attenuation on harmonic noise related to combustion cycles without inducing excessive pressure losses or flow noise levels. Usage of predictive tools in conjunction with validation testing is becoming paramount to guide the design of such high-performance silencing devices.
Design of internal combustion engine exhaust and intake systems involves the simulation of different phenomena in the fields of fluid mechanics, acoustics and thermodynamics. The main acoustic component is related to the pulsation of the gas generated by the periodic opening and closing of the intake and exhaust valves or ports during the filling and emptying processes. These pulsations will propagate inside the systems and be either radiated by the skins of the largest vessels (radiated noise) or at the intake and exhaust orifices (orifice noise). Silencer volume sizing, internal configuration and acoustic tuning are defined to achieve engine performance and orifice noise targets while being compatible with vehicle architecture, weight and cost objectives. The second acoustic component of importance is coming from the different noises generated by the flow. Some of them are steady like broadband jet noise but others are occasional whistling-like noises. Wave action and pressure drop inside the systems can also interact with the cylinder filling and emptying processes. Simulating these phenomena is particularly important for the design of components close to the engine such as manifolds, plenum and catalytic converter for instance.

A discussion regarding the available simulation methodologies for orifice noise is presented in section 2 with the associated advantages and disadvantages related. All these approaches are widespread and well documented on four-stroke automotive engines but additional simulation and design challenges that arise with high performance two-stroke engines will be discussed in section 3.

2. Overview of the simulation methodologies for silencer design

There is no unique software that can simulate all the physics involved in the design of the intake and exhaust systems. Instead, there is a variety of approaches which are complementary to each other and allow to predict at least two physical aspects of importance as shown in Fig. 1.

Figure 1: Illustration of the main acoustic phenomena and their simulation approaches involved in the design of internal combustion engine exhaust and intake systems.

The first approach is 1D linear acoustic in frequency domain using transfer matrices (Transfer Matrix Method, TMM) [1]. In automotive industry, in-house software usage allowed to improve the design of many products. Because it is an analytical approach it has also contributed to a better understanding of acoustic muffler tuning. On the down side, due to the underlying assumption of plane-wave propagation it is limited to low frequencies where wavelengths are large compared to the geometries under study. To predict orifice noise, it also requires experimental engine characterization data - noise source as well as flow and temperature conditions. An electric analogy based on Thevenin-Norton theorem is used [2] with
the engine modelled as a source of frequency dependent pressure $P_S$ and internal impedance $Z_S$ as illustrated in Figure 2. The intake or exhaust system is represented as an acoustic load $Z_L$ and the coupling is represented in Eq. (1). Finally, 1D fluid mechanics calculations are also needed to evaluate pressure losses.

![Figure 2: Illustration of the electric analogy based on Thevenin-Norton theorem used to represent the coupling between internal combustion engine and intake and exhaust systems.](image)

$$P_L = \frac{P_S}{1 + \frac{Z_L}{Z_S}}$$

Nowadays most of the industrial simulation are made using the time-domain non-stationary fluid mechanics approach [3] with commercial software such as GT-Power. These tools are often referred to as 1D or quasi-3D Computational Fluid Dynamics (CFD) Simulation Software. This is the only approach that combines the engine thermodynamics, exhaust and intake systems thermal transfer, flow and wave action. It allows the joint design of the engine and the different components of the systems, to optimize simultaneously orifice noise and engine performance. However, proven frequency range of validity for this approach is limited to 1kHz and it still does not consider fluid/structure interaction that can occur in the intake and exhaust systems.

A vibroacoustic Finite Elements Method (FEM) model can be used to compute radiated noise [4] with acoustic noise source input data obtained either experimentally or from the non-stationary fluid mechanics method mentioned above in a similar fashion as per 1D TMM. A FEM approach is also useful when it comes to simulate the higher order acoustic modes taking place inside mufflers and chambers [5] or complicated geometries which can become difficult to discretize accurately using 1D software.

Most of the acoustic issues on intake and exhaust systems can be solved by using 1D-CFD and FEM. What is not addressed with these approaches is mainly related to aerodynamic noise. It is now possible to simulate jet noise or whistle with several 3D non-stationary models [6] but at the cost of increased computational resources.

As already stated at the beginning of this section, there is a variety of approaches which are complementary to each other in the sense they allow to predict at least two physical aspects of importance, as shown in Figure 1. Each approach also has some pros and cons as summarized in Table 1.
Table 1: Overview of the available simulation methods with pros and cons.

<table>
<thead>
<tr>
<th>Approach</th>
<th>Pros</th>
<th>Cons</th>
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<tbody>
<tr>
<td>1D linear acoustics</td>
<td>- Low calculation expense</td>
<td>- Limited to low frequencies</td>
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<td></td>
<td></td>
<td>- Does not predict engine performance</td>
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<td></td>
<td></td>
<td>- Requires engine characterization</td>
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<tr>
<td></td>
<td></td>
<td>- Does not consider fluid/structure interaction</td>
</tr>
<tr>
<td>Time-domain 1D CFD models</td>
<td>- Allows engine performance predictions</td>
<td>- Valid below 1kHz</td>
</tr>
<tr>
<td></td>
<td>- Does not require engine characterization</td>
<td>- Does not consider fluid/structure interaction</td>
</tr>
<tr>
<td>Acoustic FEM</td>
<td>- Simulates the higher order acoustic modes</td>
<td>- Does not predict engine performance</td>
</tr>
<tr>
<td></td>
<td>- Can predict fluid/structure interaction</td>
<td>- Requires engine characterization</td>
</tr>
<tr>
<td>3D non-stationary CFD models</td>
<td>- Allows performance predictions</td>
<td>- Computationally expensive</td>
</tr>
<tr>
<td></td>
<td>- Allows flow noise prediction</td>
<td>- Does not consider fluid/structure interaction</td>
</tr>
</tbody>
</table>

There is a lot of literature regarding performance and acoustic simulation for four-stroke engines which are widespread in the automotive industry, but additional simulation and design challenges arise with high performance two-stroke engines. Most of the predictive developments are limited to engine cycle simulation and the literature regarding acoustic simulation is very limited. To investigate this situation, the existing approaches were compared in the context of a two-stroke engine in section 3.

3. Challenges in a two-stroke engine exhaust system.

3.1 Non-linear effects in exhaust muffler

Exhaust ports on performance two-stroke engines are generating higher and more aggressive pulsations than valves of four-stroke engines resulting in higher noise levels (10 to 15 dB) on a larger spectrum due to higher harmonic contents (beyond 2kHz instead of 1kHz).

Using a linear acoustics model, a muffler (MUFF1) was designed to perform better than the reference muffler (MUFF2) based on calculated acoustic transmission loss (TL). As presented on Fig. 3, in MUFF1 pipes and chambers are coupled through high porosity (40%) perforated areas. Acoustic attenuation tests on a conventional acoustic bench confirm simulations with a higher TL shown on Fig. 3a, but these results conflicted with orifice noise measurement results on engine dynamometer (Fig. 3b), where MUFF1 showed poor acoustic performance with higher orifice noise levels than MUFF2.

Figure 3: Performance comparison between two mufflers. (……..) MUFF1, (▬▬) MUFF2.
(a) Transmission Loss on acoustic test bench (b) Tailpipe noise on engine dynamometer
The poor performance of MUFF1 is attributed to the non-linear behavior of the perforated pipe under operating condition with the engine which reduces the coupling between pipes and chambers. This phenomenon is well known on micro-perforation or with low perforation porosity [7,8], but it was never met on a configuration such as MUFF 1 which is commonly seen on automotive exhaust silencers.

As a matter of fact, in linear acoustics, perforated sections are generally represented using equivalent “homogeneous” transfer impedance $Z_p$ of the perforated sheet separating adjacent fluid domains per Eq. (2).

$$Z_p = \frac{\Delta p}{V} = R_p + j \cdot X_p$$  (2)

Several semi-empirical approaches have been proposed using this representation [7,8]. By testing perforation on a concentric resonator, Dickey [9] noticed that behavior is changing depending on acoustic velocity. With high acoustic particle velocity through perforation, the behavior is highly non-linear, and the perforated area behaves like a resistive plate. A comparison between transfer impedance calculated using Mechel’s formula [7] and Dickey non-linear formulation can be seen on Fig.4 for actual operating condition with silencer connected to the engine. This can explain why the higher acoustic attenuation observed with MUFF1 in the case of the acoustic test bench without flow on Fig. 3a does show up in orifice noise measurements on Fig. 3b with the actual engine.

![Figure 4: Comparison between Mechel linear and Dickey non-linear transfer impedance formulations for silencer perforation under operating conditions with engine.](image)

The second issue related to non-linearity is the accuracy of the simulations. As shown in Fig. 5, GT-POWER tailpipe noise predictions for silencer design MUFF2 agree with measurement in low frequency for the first engine orders (EO), but not on higher orders which correspond to frequencies at which muffler attenuation is increasing. Even if this model is non-linear, these results mean that the non-linear behaviour described previously is not included in the perforation model of the commercial software. It was decided to build a linear model of the muffler that includes a non-linear perforation impedance. Dickey’s semi-empirical formula was used and acoustic velocity inside the pipe was simulated with GT-POWER. As shown by Fig. 5, including non-linearities led to a much better agreement on higher orders.
3.2 Solving mid-frequencies issues on an exhaust muffler

As it was shown in the preceding section, two stroke engine exhaust noise frequency range is large, and it is common to meet mid and high frequency issues. In this case, an issue on tailpipe noise at 1070Hz was experienced with the 8th engine order (8EO) as shown on Fig. 5. A FEM model of the muffler presented in Fig. 6 shows that this resonant behaviour is associated with half wave resonance of the upper chamber. Two solutions were investigated with FEM models: the first one by modifying the muffler’s internal architecture and the second one by integrating a Helmholtz resonator in the silencer design.

3.2.1 Muffler tuning analysis and architecture modifications

To understand the tuning associated with resonance at 1070 Hz, chambers were analysed separately. As shown on Fig. 6a, this problem is associated with half-wave resonance inside the upper chamber. It is possible to cancel this effect by exchanging inlet and intermediate pipe locations within the chamber. This solution was tested on tailpipe noise and showed a dramatic reduction at 1070Hz, but another issue raised at 1900Hz. As shown in Fig. 6a and Fig. 6b, simulations show it was corresponding to an anti-resonance in upper and lower chambers. The solution was to move the quarter-wave resonance in lower chamber from 1200 Hz to 1900 Hz by shortening it of 30mm (See Fig. 6b). With these modifications, a good balance between all engine orders on tailpipe noise was achieved.

Figure 5: Tailpipe noise of a two-stroke engine. Comparison between measurements (▬▬), GT-POWER time domain model (▬▬) and frequency model including non-linear behavior of perforated areas (…….)

Figure 6: FEM analysis of muffler architecture influence on acoustic resonance in upper and lower chambers (▬) Original architecture, (-----) Modified architecture (a) Upper chamber transmission loss. (b) Lower chamber transmission loss.
3.2.2 Helmholtz resonator design

As the architecture modifications identified in 3.2.1 to achieve required attenuation at 1070 Hz were not compatible with project schedule and manufacturing tools already built, an alternate option to control the problematic behavior was to implement a Helmholtz resonator compatible with the existing components as shown in Fig.7. The added feature was designed in such a way that it did not increase pressure drop which would have been detrimental to engine performance and was tuned using a FEM model to achieve the desired acoustic attenuation as seen on the TL curve from Fig.7. Exhaust orifice noise measurements were performed on original design and modified design with resonator and demonstrated that the implemented design modification brought significant improvement at the targeted frequencies without any drawback on other engine harmonics of interest as demonstrated on Fig.8.

Figure 7: Half-wave resonance identified at 1070 Hz with the FEM model on original model (up left) of the muffler and effect of Helmholtz resonator integration (bottom left) on TL curve

Figure 8: Measured tailpipe noise of original design (—) and design with added resonator (---). Up to 13dB reduction at 1070 Hz without any drawback on other engine harmonics of interest.
4. **Conclusions**

The increasing demand for high performance recreational vehicles combined with more stringent noise regulations and customer expectations lead design challenges. Usage of predictive tools in conjunction with validation testing is becoming paramount to guide the design of such high-performance silencing devices. Design of internal combustion engine exhaust and intake systems involve the simulation of different phenomena in the fields of fluid mechanics, acoustics and thermodynamics and there is no unique software that can simulate all the physics involved.

A discussion regarding the available simulation methodologies was presented in section 2 with the associated advantages and disadvantages related summarized in Table 1. All these approaches are widespread and well documented on four-stroke automotive engines, but additional simulation and design challenges arise with high performance two-stroke engines as demonstrated in section 3. Nevertheless, usage of simulation tools allowed to identify a problematic resonance in an exhaust silencer and to define 2 different strategies to mitigate it. One of the two design alternatives was finally selected and successfully implemented in a mass production consumer product.

5. **Acknowledgments**

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**REFERENCES**