Asociación Argentina de Mecánica Computacional

Mecánica Computacional Vol XXXIII, págs. 3477-3485 (artículo completo)
Graciela Bertolino, Mariano Cantero, Mario Storti y Federico Teruel (Eds.)
San Carlos de Bariloche, 23-26 Setiembre 2014

COMPUTATIONAL AEROACOUSTICS ANALISYS OF AN AUTOMOTIVE ENGINE EXHAUST SYSTEM

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Keywords: Aeroacoustics, Computational, Experimental, Exhaust System, Mufflers

Abstract. Traffic is a major source of environmental noise in modern day society. Subsequently, developments of new vehicles are subject to heavy governmental legislations. The major noise source on common road vehicles during acceleration is the flow noise caused by turbulent exhaust gas. The main objective of this work is to develop an appropriate Aeroacoustic simulation method to investigate the acoustic of sound at exhaust automotive muffler. The range of validity of the method is studied by comparing the results to measurements. The Computational Aeroacoustics (CAA) results are compared with an experimental test in a vehicle during its acceleration and the mean flow model of the muffler has a satisfactory mesh with a suitable inlet boundary provided by an engine dynamometer data. The present work describes a good agreement between computational and experimental approach for the Aeroacoustics behaviour of a specific configuration of exhaust muffler.

1 INTRODUCTION

The automotive exhaust system development is complex, affecting the noise characteristics, emission and fuel efficiency of the engines. Ford Motor Company became the first to innovate the exhaust engine system and in the year of 1921 it published in the Ford Model T owner's manual an explanation for the relevance of the use of mufflers, by the follow citation: "The exhaust as it comes out from the engine through the exhaust pipe would create a constant and distracting noise were it not for the muffler. From the comparatively small pipe, the exhaust is liberated into the larger chamber of the muffler, where the force of the exhaust is lessened by expansion and discharged out of the muffler with practically no noise. The Ford muffler construction is such that there is very little back pressure of the escaping gases, consequently there is nothing to be gained by putting a cut-out in the exhaust pipe between the engine and the muffler." According that, silencer was an expansion chamber type where the backpressure produced by the silencer has a direct influence on the fuel consumption. The term "muffler" was first used in 1895 and it is commonly called "silencer" that is a "mechanism that stifles the sound of a motor or firearm" was first recorded in 1898. Most of these mechanisms are described in Lord Rayleigh's (1877 and 1878).

Among the different strategies for vehicle exhaust systems most of them haven't an external excitation source used to cancel the engine noise, called passive system. In a passive exhaust system there are essentially two common ways to attenuate the noise of the engine, the reflective and dissipative systems. The reflective systems are consisted of expansions and contractions resonators as expansions chambers and orifices, including the Helmholtz resonators where these systems have the function of scattered and cancel the incident sound. Dissipative systems convert the incident sound in heat by means of materials with high loss factor such as mineral wool or glass fibres. In addition to the mentioned about dissipative systems we can consider through-flow perforates generating flow-related losses.

The current study was carried out as part of a larger research project involving computational and experimental aspects, describing an innovative approach that can be applied to the prediction of the flow-induced noise generated by complex exhaust engine systems. The motivation of this work was driven by the new Brazilian government program, called Inovar-Auto, was created to encourage and promote the vehicle technology innovation. The computational and experimental procedure has been developed on dissipative exhaust system and it has been performed with STAR-CCM+, a CFD analysis package and the LMS Test Lab. data system acquisition.

A noise measurement of sound radiated from a vehicle is subject to a variety of sources including mechanical noise, shell vibration radiated noise and exhaust tail pipe noise, described by Pang et al. (2005). Studies from Weltens et al. (1991) and Erhard (1984) have shown that the noise radiated from the exhaust structure itself predominantly arises when large flat surfaces are present in the exhaust system. Weltens et al. (1991) and Kim et al. (2001) showed that in vehicle drive-by tests the dominant source of noise is from the engine itself, with a large contribution from the exhaust outlet or radiated from the exhaust shell structure.

Phillips and Orchard (2001) defined four sources of noise as being inherent to motor vehicles, being the powertrain system, induction, exhaust and tyre noise. Exhaust noise is primarily generated by the discharge of combustion products at high pressure and temperature from the engine cylinders through the exhaust valves, producing a pulsating noise. As exhaust gases flow through the exhaust system and as they exit the exhaust outlet, flow noise is produced as described by Tanaka et al. (1981). In the works of Erhard (1995) and Davies (2001) have shown that flow noise generated in exhaust systems can be amplified by resonances within the system.

According Kunz (1999) the sound pressure level at any point in space beyond the termination of an exhaust system to the atmosphere, is a direct function of the instantaneous mass flow rate from the end of the exhaust pipe and it is relative to the distance between source and microphone.

The last decades, a number of researchers have been concentrated on determining and understanding the source characteristics in the exhaust systems of engines through experimental analysis. The works of Czarnecki et al. (1948) and Davis et al. (1949) they conducted dynamometer experiments with a four stroke, opposed, six cylinders engine to investigate a number of different mufflers, including perforated tubes, straight-through and expansion chamber. Based on the two publications of Prasad and Crocker (1983) they determined the source impedance experimentally for a multi-cylinder engine exhaust system and used this information in the acoustic modeling of insertion loss and radiated sound pressure. Desmons et al. (1985) used a four-stroke four-cylinder engine to characterize the source impedance on the exhaust side.

In this work an overall method will be introduced for the computational aeroacoustics (CAA) detailing the acoustic noise generation and propagation in the near field environment of one reflective type of automotive exhaust system. The results will be shown comparing numerical calculations to the experimental data.

Computational aeroacoustics (CAA) as shown by Mohiuddin et al. (2007) refers to when sound sources and sound wave propagation are solved in a single comprehensive model. In this case, computational fluid dynamics is used to solve the sound generation and the sound wave propagation because they both follow the Navier-Stokes equations. CAA is a transient simulation of the entire fluid region encompassing the sources, receivers and entire sound transmission path in between. By rigorously calculating time-varying flow structures, pressure disturbances in the source regions can be followed and analyzed. Sound transmission is simulated by resolving the pressure waves traveling through the fluid.

The approach of splitting the flow and sound fields from each other and solving for them separately can be simplified further if the receiver has a straight, unobstructed view of each individual point that is a source of noise. Sound transmission from a point source to a receiver can be computed by a simple analytical formulation. The Lighthill acoustic analogy provides the mathematical foundation for such an integral approach. The Ffowcs-Williams and Hawkings (FWH) method extends the analogy to cases where solid, permeable, or rotating surfaces are sound sources, and is the most complete formulation of the acoustic analogy to date. A common engineering application of FWH method to the prediction of the sound radiation is in an automotive A-pillar rain gutter. In this case usually use the large eddy simulation (LES) turbulence model, predictions of the sound pressure level for this case were found to be in very good agreement with experimental data taken from the literature.

This paper will shows a comparison achieved between CAA simulation and test results for an unusual application in dissipative silencer where the simulation process is based on DES turbulence model with a compressibility model using the FWH to calculate the overall sound pressure level at a near field environment.

2 EXPERIMENTAL PROCEDURE

In order to validate the computational simulations experimental tests have been carried out on the exhaust system in the vehicle in the proving ground during for three different vehicle accelerations. The test allows measuring the sound pressure level and the noise spectrum generated by the silencer at one specific position from the tail pipe. This analysis takes into consideration all the reflective effects due the ground placed at 30 cm behind the tail pipe origin point. The aerodynamics flow around the entire vehicle shall not influence on the propagation of the irradiated sound noise. The perimeter around the vehicle should have an enough area from any kind of obstacles to don't influence the acoustic field. A global overview of the experimental test set-up is shown on the Fig. 1.

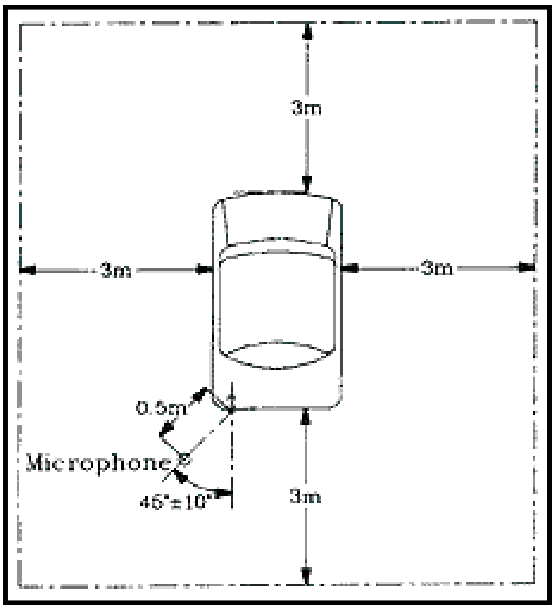

Figure 1: Vehicle experiment test set up.

The exhaust noise measurement position is in a 45° direction with the exhaust gas flow axis and the measurement distance from the tail pipe is 0.5m were made with the microphone at the exhaust outlet level from the tail pipe. For engineering method, the distance from measurement point and the tail pipe outlet to the floor shall be greater than twice of measurement distance, and there is no reflection objects in exhaust direction otherwise the distance from exhaust outlet to reflection objects shall be greater than 6 times of measurement distance.

The frequency noise of engine speed can be attributed to the increased velocity of the exhaust gasses. At 3000 rpm the 100 Hz band is the dominant frequency, which equates to the second harmonic of the engine, followed the harmonic frequencies of fourth (200 Hz) and sixth (300 Hz) harmonics as expected. At 4200 rpm the 140 Hz band is the dominant frequency, which equates to the second harmonic, followed the harmonic frequencies of fourth (280 Hz) and sixth (420 Hz) harmonics as expected. At 5000 rpm the 166.67 Hz band is the dominant frequency, which equates to the second harmonic, followed the harmonic frequencies of fourth (333.33 Hz) and sixth (500 Hz) harmonics as expected.

The Fig. 2 represents the Campbell Diagram, in which is possible to observe the measurements show that this silencer has a good broadband attenuator and the control noise well at the low frequencies and the high frequencies. It is interesting to note that the noise level between 670 Hz and 1300 Hz depends of the internal configuration and it is increased during the vehicle run up. For human hearing, this frequency band doesn't represent any contribution to the overall noise level.

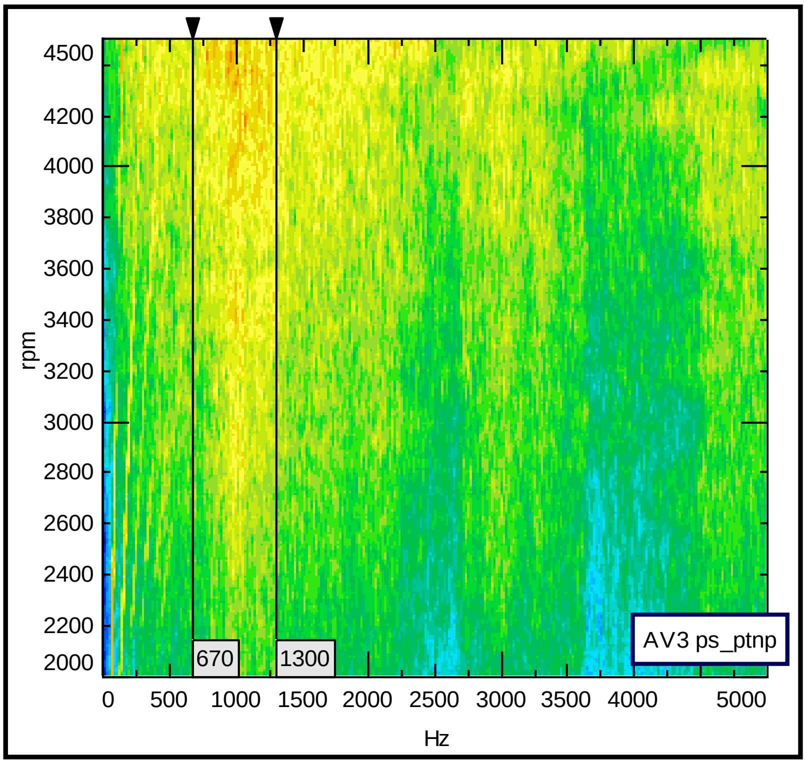

Figure 2: Campbell Diagram frequencies band

3 COMPUTATIONAL PROCEDURE

The computational methodology developed and validated in this work had used an experimental data obtained from an engine dynamometer to measure the thermodynamics variables used as input boundary conditions and the noise irradiated sound was measured as described in the section below used for the Aeroacoustics correlation. The location of the probe points for the numerical analysis was positioned in the same spatial coordinates of the experiment. The small influences about the aerodynamic noises under the generated exhaust flow noise weren't predicted in the current simulations.

Internal schemes of silencers are the results of a combination of several components, such as absorbing materials and reflective elements that are used to ensure the acoustic attenuation. For this work the silencer are consisted by reflective elements, such as perforated pipes and baffles, that attenuate the sound pressure level by reflection and interference of the acoustic waves.

In this approach, compressibility formulation is applied to calculate the propagation of pressure waves together with the fluid dynamics, inside and outside the silencer, at the same time obtained the sound quantities for any node in the computational domain. For this modeling is required a second-order accurate with 20 cells per acoustical wavelength to reach a good agreement with the real acoustic signal.

The thermal effects are considered due the temperature variations of the silencer walls with the environment and with between the inside chamber. The study is treated as being non-adiabatic, meaning there is a thermal exchange by forced convection and thermal diffusion.

A solution is sought in the time domain with segregated implicit solver with the second order implicit time stepping method. A time step of 2.5x10-4 seconds was chosen, which is accurate enough for the frequency range limited below 5.0 kHz. The total simulation time for spectral data comparisons is 0.5 seconds.

The LES turbulent model has less computationally demanding than DNS. Agoing to achieve a high numerical accuracy keeping together less demanded computer power a hybrid RANS-LES model was developed for researches where an unsteady RANS-model (URANS) is used for shear-layers and LES is used for unsteady regions such as when the flow separates from a solid boundary as showed by the work of Spalart et al. (1992). This is referred to as Detached Eddy Simulation (DES) have been shown to be suitable for capturing both narrow-band and broad-band flow excitations in a wide range of aerospace and automotive applications. The SST k-ω model is one RANS-models that was modified for DES. Menter et al. (2003) showed the reason why these two models were chosen to be implemented in DES model.

One important issue with the DES-formulation occurs at the point of switch from RANS to LES inside the boundary-layer, thus the turbulent viscosity may be underestimated and consequently the skin-friction coefficient also becomes underestimated could happen a premature separation point. The proposed Delayed DES (DDES) utilizes a shielding function to preserve RANS in the entire boundary-layer. Two years later Shur et al. (2008) presented an Improved DDES (IDDES) which contains an LES wall-model where it has the feature of treating the boundary layer flow, that means, if the boundary-layer flow contains turbulent structures only the part of the boundary-layer closest to the wall is assigned for RANS-modeling while the rest is modeled using LES wall-model. Therefore the IDDES and SST k-ω turbulent models have been used for the case study presented in this paper.

The computational domain has been created with a hybrid mesh consisting of hexahedral elements on the wall regions for boundary layer resolution and the polyhedral cells are used to fill the core volume of the entire domain. The computational volume mesh contains 2.0 millions of cells with the maximum size of 10 mm and minimum size of 0.05 mm. In addition to the mesh a volumetric refinement zone is added to the plenum from downstream of the tail pipe until the microphone point, to avoid dissipation of the pressure wave.

The computational work excludes the internal combustion engine up to the inlet to the silencer tube in the transient simulation. To provide a good description of the mean flow, the inlet conditions are prescribed by experimental tests data from dynamometer, representing the real physical conditions in the selected engine speeds. A non-reflective boundary condition was applied for the inlet and outlet boundaries to avoid wave reflections interferences provided by the pressure wave propagation.

During the simulation the pressure data is sampled at the specific point and the resulting noise level spectra are then generated from the sampled pressure time history. The post-processing for the experimental measurements data was performed with equivalent signal processing used for the computational simulation.

4 RESULTS

The comparison between the experimentally measured data and the CAA prediction is showed for a probe point by Figures 3, 4 and 5. The point spectra are in good agreement with measurements for the whole range of frequencies from 20Hz up to 1.0 kHz been able to capture the main harmonic frequencies.

The noise level is composed by the acoustical and fluid induced components in this way the objective of the currently study is to develop the fluid dynamics inside and outside the muffler through boundary conditions originated from experimental tests and simultaneously to capture the acoustic propagation in the near field at downstream of the tail pipe. The agreement in the acoustical spectra at the external probe give assurance that the chosen mesh, turbulence model and time step size settings result in an efficient propagation of the acoustic field.

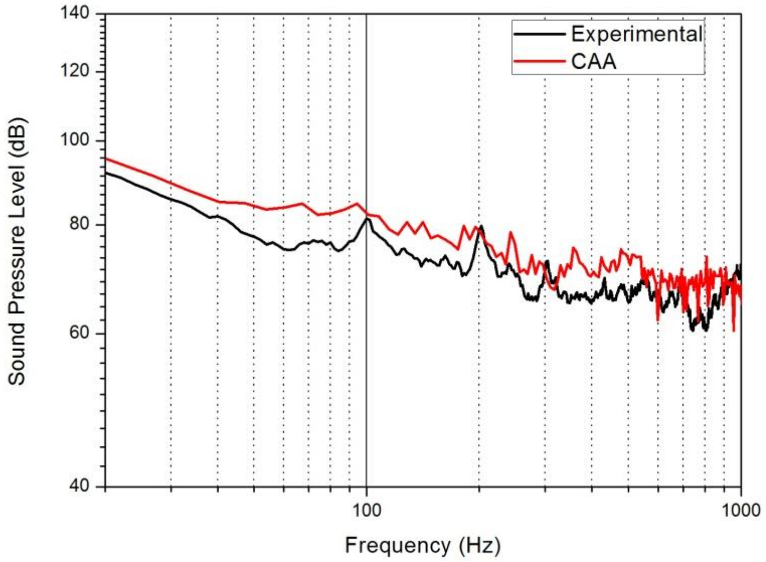

Figure 3: CAA Prediction and Test Data for SPL at 3000 RPM

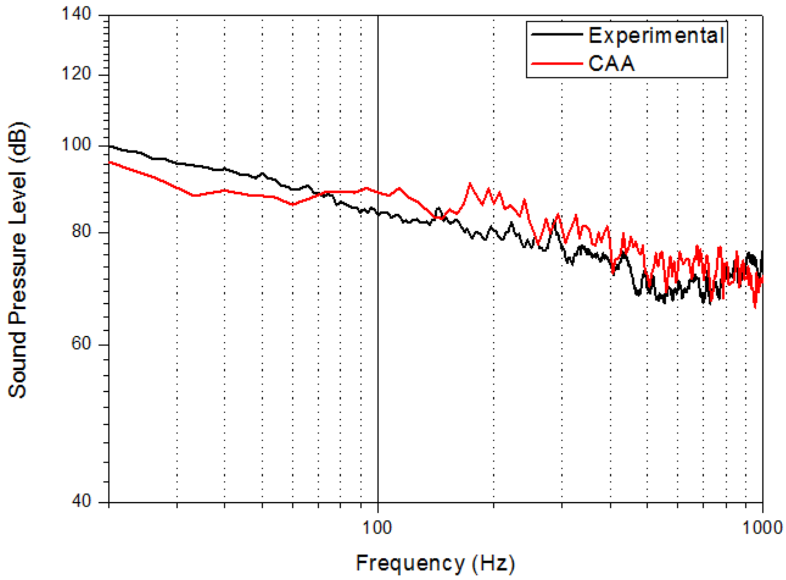

Figure 4: CAA Prediction and Test Data for SPL at 4200 RPM

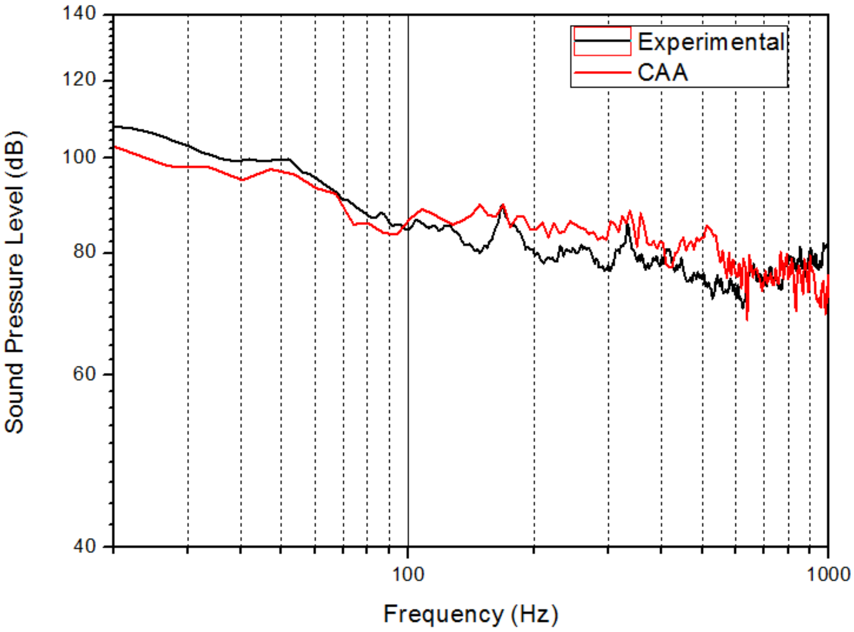

Figure 5: CAA Prediction and Test Data for SPL at 5000 RPM

5 CONCLUSIONS

This work presents an innovative simulation method approach for aeroacoustic analysis that has been developed and validated for a specific exhaust muffler configuration, focusing on the prediction of the acoustic field considering the dissipative, heat transfer and mean flow effects. The main objective of this study is to identify the acoustic source and capture the sound propagation in the near field downstream of the tail pipe.

The simulation showed a close agreement for the main harmonics frequencies in comparison of experimental sound pressure level spectra up to 1.0 kHz at captured near-field probe in the space location. These satisfactory results were carried out by the selecting DES turbulence model with the appropriate inlet and outlet boundary conditions.

The assurance from the simulation results presents an essential opportunity to optimize the internal configuration of an exhaust muffler for the noise sound reduction. Further studies will be focused on applying this method to different mufflers configurations, including a full exhaust line system and noise source generated by the engine.

ACKNOWLEDGMENTS

The authors would like to thank the FIAT CHRYSLER AUTOMOBILES.

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