4th Biennial International Conference on Powertrain Modelling and Control

10-11th September 2018, Loughborough UK

Design Development and Performance Evaluation of ICE Exhaust Silencer

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Abstract: The noise levels generated by an unmuffled engine exhaust system can be identified as the loudest vehicle noise source. The muffler or silencer is an essential component of the internal combustion engine exhaust system, its main function is to reduce the exhaust-generated noise to an acceptably low level. Its design development is a complex process affecting the engine efficiency and thus fuel consumption, emissions and overall noise generation. This paper focuses on the design development of a muffler for a single cylinder engine application. A 1D GT-Power model of a single valve engine was developed. Additionally, an analytical muffler preliminary design methodology was introduced. The methodology provides guidelines for muffler grade selection, sizing of different components, calculation of back pressure as a function of the exhaust gas flow rate. Two custom mufflers design concepts were developed for the single cylinder engine based on the introduced analytical methodology. Two commercial single cylinder engine muffler designs available from Yanmar and Loncin were considered for the engine performance evaluation simulation. The presented combination of analytical and numerical modelling procedures can reduce the overall length of the muffler development stage by eliminating faulty design concepts and refining the muffler's performance parameters.

Keywords - acoustics simulation, analytical modelling, back pressure, exhaust system, GT-Power, muffler design

1-Introduction

The noise pollution generated by internal combustion engines is of vital concern when used in residential areas or areas where noise levels are strictly regulated. Noise levels with a magnitude in the range of 80-90 dB are harmful to human beings. The general sources of noise in an internal combustion engine can be divided into two major groups. The first group includes the exhaust noise generation and the second group includes the mechanical noise generation. The exhaust noise is the predominant one. The most effective way to reduce the exhaust noise is by the application of a silencer (muffler) in the exhaust system. A silencer or muffler is a noise attenuation device designed to reduce the engine exhaust noise.

The continuous research in the field of engine noise reduction has led to an improvement in the performance of the modern silencers used for automotive exhaust noise attenuation. The muffler target design and performance are directly connected to the engine type and its specification [1-5] i.e. naturally aspirated, turbocharged, nominal power and rotational speed, type of fuel, number of cycles (2 or 4) etc. The engine characteristics and its performance at a nominal operational regime are determining the noise production range.

1.1- Silencer (Muffler) Types

Absorptive Mufflers [1-3]: The main advantages of those types of silencers are their low cost and low backpressure performance characteristics. Their sound attenuation mechanism is absorption. Sound waves are attenuated through the sound energy conversion into heat in the absorptive material. The most common and basic principle design of an absorptive silencer is presented in fig.1 a. The basic design includes perforated straight pipes, surrounded by absorptive materials like fibreglass, steel wool in which the noise energy is dissipated. The Absorptive silencers provide good noise attenuation in the high-frequency range, whereas in the low-frequency range the noise reduction is poor. At low frequencies, the noise dissipation can be improved through the increase of the absorptive material thickness.

Reactive Mufflers [1-3]: In comparison with the Absorptive type mufflers the Reactive type provides a higher level of noise attenuation. The noise is reduced by passing the exhaust flow through various chamber and obstacles. The general basic design of a Reactive muffler consists of a series of connected resonating and expansion chambers. The overall geometrical design of the resonators and expansion chambers reduces the sound energy at a given range of frequencies. They are widely applied in car exhaust systems where the exhaust gas flow and the noise generation vary as a function of time. The Reactive mufflers provide good noise attenuation in the low-frequency range under 500 Hz. The most common and basic principle design of a reactive silencer is presented in fig. 1 b.

1.2. Engine Noise Generation

The main fraction of the engine generated noise is due to pressure pulses released in the exhaust system. The pulses are released into the exhaust system at the end of every expansion stroke of the piston. The amplitude of the pulses varies between 0.1 bar and 0.4 bar with a duration from 2 µs to 5 µs. The spectrum of the frequency is a function of the pulse duration. Thus, the cut-off frequency varies in the range from 200 Hz to 500 Hz. The unsilenced noise generated by an internal combustion engine falls in the range 100 – 130 dB, depending on the engine type, size etc.

a. Absorptive type muffler *b. Reactive type muffler*

Figure 1 Reactive and Absorptive type mufflers

1.3-Muffler Design, Basic Requirements

During the initial phases of design development of an automotive muffler, many preliminary requirements should be taken into consideration [2-5]. Some of the requirements towards the muffler performance are - sufficient insertion loss level, back pressure (pressure drop) value fluctuation in an acceptable range, a target range of sound emission level, weight, cost, design shape and style. The design procedure of a muffler can be summarized by matching the attenuation performance of the muffler to the noise characteristics of the source, with acceptable pressure drop within a specified range.

The muffler noise attenuation performance is mainly defined by its insertion loss and transmission loss. The insertion loss is expressed as the difference between the acoustic power levels generated by the engine exhaust without and with silencing applied. The transmission loss is expressed as the difference between the sound power level in decibels near the muffler inlet and the sound power transmitted by the muffler. The design and performance development of the automotive muffler must meet the required insertion loss level.

The backpressure is defined as the additional flow resistance imposed over the exhaust gas path, resulting in an increase of the pressure drop through the muffler. Thus, this leads to a rise of the exhaust system, which is affecting the engine overall performance. A strong relationship between the muffler sound attenuation and back pressure levels exists. With the increase of the noise attenuation capabilities of a muffler, the backpressure level rises correspondingly. This is especially true for the case of the reactive type mufflers. To achieve good noise attenuation levels, the flow in a reactive muffler is forced to pass through to numerous obstacles (chambers, baffles, perforated pipes etc.). This results in high backpressure, which is reducing engine efficiency. The backpressure needs to be kept within specified limits, to maintain high engine efficiency.

1.4-Theoretical 1D CFD Muffler Acoustic Analysis

In his report, Mochkaai [6] presents an overview of the GT-Power capabilities for acoustic and dynamic exhaust system valuation and optimisation. He points out some of the software main advantages such as the relatively low computational effort even for the performance modelling of a complete intake-engine-exhaust system. The coupling of GT-Power with 3D CFD software yields an increase of the modelling procedure accuracy. This is especially noticed when the flow is highly turbulent with pronounced three-dimensional flow effect (vortex structures generation and interaction). The integrated DoE module in the simulation procedure facilitates the product development procedure by optimization of the investigated design.

Mohiuddi, Rahamn and Dzaidin [7] carried out a numerical investigation of the performance of the exhaust system of a light vehicle. The numerical simulation was performed using GT-Power. The study was focused mainly on optimizing the geometrical design of the exhaust manifold. The new optimized design of the exhaust manifold exhibits lower back pressure values thus increasing the engine performance efficiency. It should be noted that no acoustic analysis of the exhaust system was done nor any validation of the theoretical results with experimental data.

Optimization analysis of the sound pressure level of an internal combustion engine is carried out by Vaidya and Hujare [8]. The theoretical investigation was made with the GT-Power software. The acoustic optimisation was done through the introduction of a resonator to the intake system and a modification of the air filter box design. The transmission losses (TL) were significantly increased in the frequency range of 50 – 300 Hz, thus reducing the overall sound pressure level. The analysis was performed under a variety of different operational conditions and engine types. There is no comparison of the theoretical results with experimental data.

2-Main Objective of the Investigation

The general objective of the investigation is to establish a robust computational procedure for preliminary design development and evaluation of ICE exhaust muffler. Thus, the following set of tasks is formed:

- Develop an analytical design methodology based on fundamental governing theories (aerodynamics, acoustics) and empirical data for sizing the exhaust muffler main components;

- Conduct analytical and 1D CFD noise performance and backpressure level analysis at the early design stages of the exhaust muffler under different operational conditions.

3-Analytical Methodology for Muffler Sizing

3.1-Selection of Muffler Grade

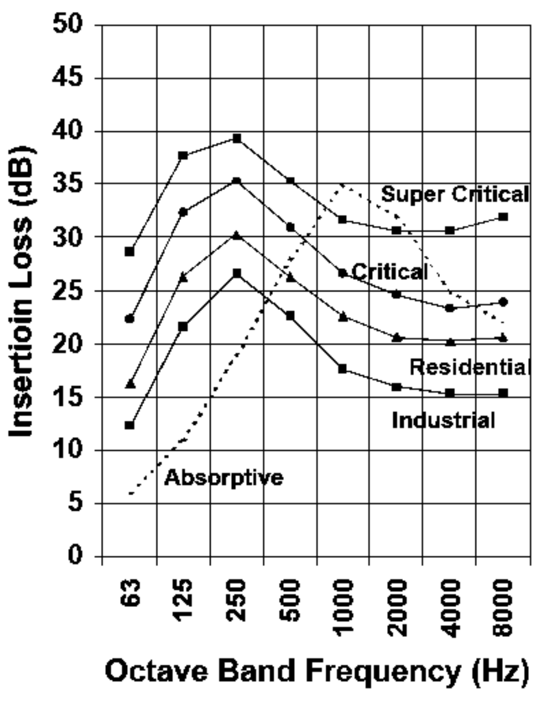


Figure 2 Insertion Loss as a function of frequency [5]

The selection of a suitable muffler for a given application begins with the identification of its type. The different mufflers grade types are presented in Table 1 according to ASHRAE Technical Committee 2.6 [5].

The insertion loss as a function of the frequency for the different grade types of mufflers is presented in fig. 2. The insertion loss values are approximated due to their dependency on the operational conditions and the engine type. As can be seen in the operational range of low frequency the reactive type muffler has the best performance, whereas the absorptive type performance is lacking. The muffler grade selection includes the following four steps presented below.

Step-1. Unsilenced Noise Level (UNL): The unsilenced exhaust noise level for most of the internal combustion engines falls into the range from 100 dB to 120 dB, measured at 1 m distance from the exhaust pipe outlet.

Step-2. Calculation of the exhaust noise criterion: The following dependency is used for the exhaust noise criterion calculation

$$ENC = RNC - 5, dB, \quad (1)$$

where ENC is Exhaust Noise Criterion, RNC is Required Noise Criterion.

***Table 1**. Muffler Grade Types [5]*

Grade Type	**IL (Insertion Loss)**	**Body/Pipe D_{MC}/d_{INLET}**	**Length/Pipe L_{MC}/d_{INLET}**
	dB	-	-
Industrial/Commercial	15 to 25	2 to 2.5	5 to 6.5
Residential Grade	20 to 30	2 to 2.5	6 to 10
Critical Grade	25 to 35	3	8 to 10
Super Critical Grade	35 to 45	3	10 to 16

Step-3. Calculation of the unsilenced exhaust noise at receiver location: The distance correction with an assumption of free-field spreading is achieved by:

$$L_P(X_r)_{UNL} = L_P(X_0) - 20\ log\left({}^{X_r}/_{X_0}\right), \quad (2)$$

where X_r is the reflection distance, X_0 is the distance at which the UNL is measured.

Step-4. Calculation of the required insertion loss of the muffler: This calculation is done according to the equation

$$IL = UNL - ENC + 5, dB. \quad (3)$$

The required insertion noise level is obtained by subtracting the receiver noise criterion from the unsilenced receiver noise level. A safety factor of 5 dB is added to prevent eventual performance fail of the muffler. After obtaining the insertion loss value the grade type of the muffler can be selected from Table 1 [5].

3.2-The Target Engine Performance Parameters

A simple one cylinder, four stroke, gasoline engine GT-Power model was built for evaluation of the muffler acoustic and exhaust flow performance. The baseline engine used for the numerical model development is a single cylinder gas engine. Its innovative cylinder head design utilises a single overhead valve per cylinder and uses a reed valve in the intake system to throttle airflow.

The engine can run on a variety of fuels, offering good power density against other existing small engines within the marketplace and significant flexibility for a wide range of potential applications including stationary natural gas-powered generators, fixed speed pumps, HEV range extenders, small natural gas-powered vehicles, industrial engines, dedicated biogas applications.

Table 2 *Required Initial Data*

Initial Data Specification	Value	Units
Engine Speed	3,600	min^{-1}
№ of Cylinders	1	-
№ of Cycles	4	-
Engine Power	7.4	*hp*
Engine Displacement	0.435	*l*
d_{INLET}, Inlet Pipe Diameter	38	*mm*
T_{INLET}, Temperature at Inlet of Muffler	535	*°C*
T_{INTAKE}, Intake Temperature	25	*°C*

The analytical muffler sizing methodology requires initial inputs, derived from the targeted IC engine specifications and performance data. The required initial data for the analytical design procedure is presented in Table 2.

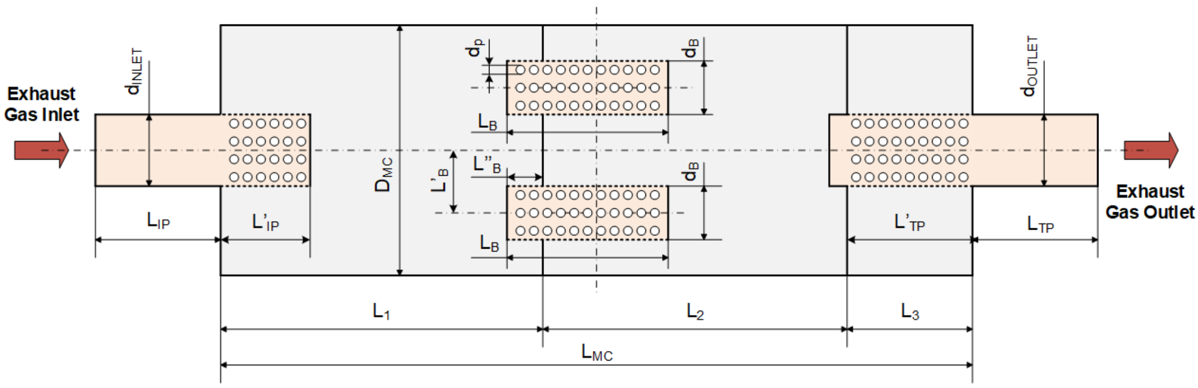


Figure 3 Muffler design schematics and key dimensions

However, there is some data that is valid for all engine types. Numerous experimental investigations are pointing that the exhaust noise frequency range for either diesel or gasoline engine is falling into the spectrum 200 – 500 Hz. An example schematic of a muffler with its main geometrical dimensions denoted is presented in Figure 3.

3.3-Engine Exhaust Frequency Band of Noise Generation

To preliminary evaluate the target spectrum of exhaust noise generation frequencies, the investigated engine maximum power and speed are required (Table 2). The identification of the engine exhaust noise level frequency band helps the estimation of the transmission loss (TL) value range [2, 9]. The calculation follows the dependencies:

- Cylinder Firing Rate (CFR), for 4 stroke engines:

$$CFR = \frac{Engine\ RPM}{120}\ [Hz] \tag{4}$$

- Cylinder Firing Rate (CFR), for 2 stroke engines:

$$CFR = \frac{Engine\ RPM}{60}\ [Hz] \tag{5}$$

- Engine Firing Rate (EFR):

$$EFR = N.CFR\ [Hz] \tag{6}$$

where *N* is the engine number of cylinders.

3.4- Muffler Main Components Dimension According to ASHARE Technical Committee 2.6 Methodology [5]

After identifying the appropriate muffler grade type in line with the procedure presented in Section 3.1, the muffler key components can be sized. The reference data presented in Table 1 is considered for the calculations of the muffler's key components geometrical sizes. If the muffler falls into the grade type "Critical Grade" (Table 1), the following calculations for its length and diameter can be made:

$$L_{MC}/d_{INLET} = 8 \div 10, \tag{7}$$

$$L_{MC} = (8 \div 10).d_{INLET}, \tag{8}$$

$$D_{MC}/d_{INLET} = 3, \tag{9}$$

$$D_{MC} = 3.d_{INLET}, \tag{10}$$

where d_{INLET} is the exhaust system inlet pipe diameter.

3.5-Muffler Length Determined by the Resonance Method

The resonance method can be adopted for muffler chamber length [10-13]. Maximum attenuation of the propagating wave occurs when:

$$L_{MC} = \frac{n.\lambda}{4}, \tag{11}$$

where, λ [m] is the sound wavelength and n = 1, 3, 5… are odd integers. To maintain the overall size within reasonable limits, the odd integer value is often taken as n = 1.

The wavelength λ is directly related to the frequency spectrum by the speed of sound. Thus, the maximum attenuation of the sound wave occurs at frequencies which correspond to the muffler's chamber length. The operational frequency range for the exhaust system is obtained from the engine performance data.

The length of the muffler chambers is calculated in correspondence with user-specified operational frequencies. The frequencies can be acquired analytically, experimentally, from reference data etc. The default frequency span for most of the different types of engine exhaust systems falls into the range $f = 200 \div 500, Hz$ (Fig. 2). The wavelength λ is calculated for the different user-specified frequencies at constant operational temperature by:

$$\lambda = \frac{\upsilon_c}{f}, \tag{12}$$

$$\upsilon_c = \sqrt{\frac{\gamma T R}{M}}, \tag{13}$$

where υ_c is the speed of sound (in air), γ is the air adiabatic constant, T is the absolute temperature, R is the universal molar gas constant and M is the molecular mass of air.

3.6-Muffler Chamber Length with Consideration of the Exhaust Gas Temperature

An important factor affecting the exhaust system design and performance is the exhaust gas temperature [12, 13]. The influence of the exhaust gas temperature on the muffler length can be evaluated by the following dependency:

$$0.5\left(\frac{49.03\sqrt{°R}}{2\pi f}\right) \le L \le 2.6\left(\frac{49.03\sqrt{°R}}{2\pi f}\right), \tag{14}$$

where °*R* is the absolute temperature of the exhaust gas (Rankin temperature), *f* is the muffler nominal frequency.

If the maximum exhaust gas temperature is not available, the following values are recommended: for gasoline engines, T = 650 °C; for diesel engines, T = 490 °C.

3.7-Muffler Volume Estimation

The theoretical muffler volume estimation is one of the first steps in the initial concept design procedure [1, 2, 10, 12, 14]. Two analytical approaches are adopted in the initial evaluation of the muffler volume.

According to L. H Billey [10, 14], muffler volume for four-stroke engines can be estimated by the analytical dependency:

$$V_m = \frac{K V_P}{N}\sqrt{\frac{1}{n}}, \tag{15}$$

where V_m is the muffler volume in l, V_P is the engine swept volume in l, N is the engine speed (min^{-1}), n is the number of cylinders, K is a constant, reflecting the type of vehicle and its application: K = 5,000 for agricultural tractors; K = 10,000 for urban and rural vehicles and bulldozers; K = 35,000 for trucks; K = 50,000 for buses and cars.

Dean G. Thomas [10, 14], recommends the following dependency for preliminary estimation of the muffler volume:

$$V_m = \frac{QV_PN}{1{,}000\sqrt{Tn}}, \qquad (16)$$

where V_m is the muffler volume in l, V_P is the engine swept volume in *l*, N is the engine speed (min^{-1}), n is the number of cylinders, T is the number of strokes, Q is a constant, reflecting different conditions and requirements. The value for Q can be chosen in the range of Q = 5 - 6.

These formulas are analytically derived and meant to be incorporated into a broad range of industrial and commercial problems. The results obtained by these empirical dependencies serve the purpose of preliminary design estimation of the muffler volume.

3.8-Baffle Pipes Design

The sum of the cross-section areas of the baffle pipes must remain equal or larger compared to the inlet exhaust pipe cross-section [11-13]. The diameter of the baffle pipes is calculated as follows:

$$d_B = \left(\frac{d_{INLET}}{N_B}\right).B_{GF}, \qquad (17)$$

where d_{INLET} is the diameter of the exhaust system inlet pipe, N_B is the number of baffle pipes and B_{GF} is the baffle pipe diameter growth factor. The growth factor values can be chosen from the range $B_{GF} = 1 - 1.4$.

3.9-Tailpipe Design

The exhaust tail-pipe has resonances that can amplify the engine generated noise [10, 12, 13]. To avoid amplification of the engine noise a short tail-pipe should be used. According to equation (11), resonance occurs when the length of the pipe is $L = n\lambda/2$. The odd integer n is taken as $n = 1$. The total length of the tail-pipe must be $L_{TP} < n\lambda/2$. The length of the tail-pipe can be calculated by the following dependencies:

$$L_{TP} = \left(\frac{n\lambda}{2}\right)0.7. \qquad (18)$$

- Optimal tail-pipe length:

$$L_{TP,OPT} = \frac{\upsilon_c}{4f_{TP}} - \frac{d_{TP}}{2}, \qquad (19)$$

where $f_{TP} = \frac{\omega.N}{120}$ is the firing frequency of a four-stroke engine or $f_{TP} = \frac{\omega.N}{60}$, in the case of a two-stroke engine, ω is the engine speed in RPM, N is the engine number of cylinders, d_{TP} is the tailpipe diameter and υ_c is the speed of sound.

3.10-Exhaust System Back Pressure Evaluation

The theoretical calculation of the back-pressure exact value is a challenging task [1, 2, 12]. This is due to the complex geometrical structure of the exhaust system combined with the dynamic nature of the pulsating exhaust flow. The pressure drop along the muffler can be predicted with an acceptable level of accuracy by commercial CFD software packages. The detailed numerical modelling of the exhaust flow is a computational and time-consuming process. For robust, preliminary evaluation of the muffler's back-pressure the following analytical dependencies can be used:

- Exhaust Flow Rate:

$$Q_{EFR} = \frac{E_D V_E T_e RPM}{T_i C_E}, \tag{20}$$

where, E_D is the engine displacement in m^3, V_E is the engine volumetric efficiency constant (V_E = 0.4 naturally aspired, V_E = 1.2 blower scavenged, V_E = 1.15 turbocharged), T_e is the exhaust gas temperature in °R, RPM is the engine speed in min^{-1}, T_i is the intake gas temperature in °R, C_E is a cylinder numbers dependant constants (C_E = 1 two cylinders, C_E = 2 four cylinders).

- Exhaust Gas Velocity:

$$V_{EG} = \frac{Q_{EFR}}{S_{EP}}, \tag{21}$$

where, Q_{EFR} is the exhaust mass flow rate in m^3/s, S_{EP} is the cross-section area of the exhaust pipe in m^2.

- Pressure-drop:

$$\Delta p = C_{\Delta p} \rho \frac{v^2}{2} \frac{T_i}{T_e}, \tag{22}$$

where, $C_{\Delta p}$ is pressure drop coefficient, which depends on the silencer model and application.

Besides the pressure-drop along the silencer, the losses introduced by other components should be considered. The frictional losses of the exhaust system piping should be taken into account in cases with long piping routes and/or very small diameters. An estimation of the frictional losses in the piping system can be derived from:

- Smooth pipes:

$$C_{SP} = 0.0148 \left(\frac{L}{d}\right), \tag{23}$$

where *L* is the length of the pipe and *d* is the pipe diameter.

- Rough pipes:

$$C_{RP} = 0.032 \left(\frac{L}{d}\right). \tag{24}$$

The influence of the elbows and turns over the overall pressure losses is evaluated by their representation as an equivalent length of straight pipe. It is generally recommended to avoid these elements due to their relatively high-pressure loss. The pressure losses generated by other system components, such as expansions, entrances and exits can be obtained from reference graphical and tabular data.

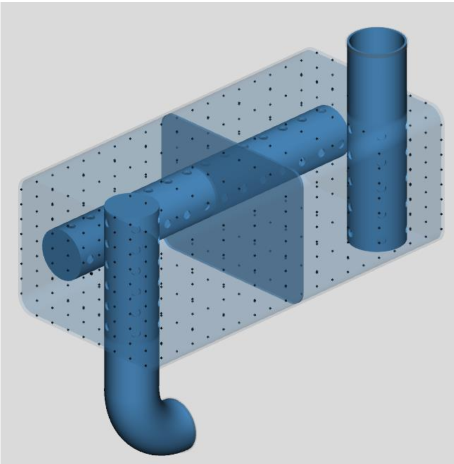
There are three major approaches to the overall pressure loss evaluation. The first one is an evaluation of the pressure loss only along the muffler. The second is pressure loss determination, including entrance and/or exit losses. And the third approach is an evaluation of the pressure loss across the whole system, including elbows, fittings, turns and all piping.

3.11-Analytical Sizing and Performance Results & Generation of 3D Muffler Models

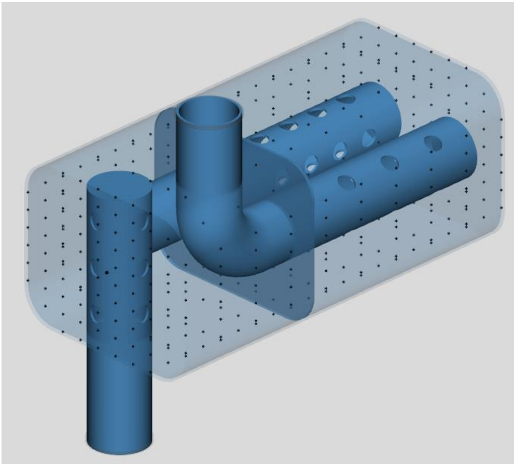
The obtained analytical results are presented in Table 3. For some muffler components, the characteristic dimension value is given in the range from min to max.

Table 3 *Analytical results summary*

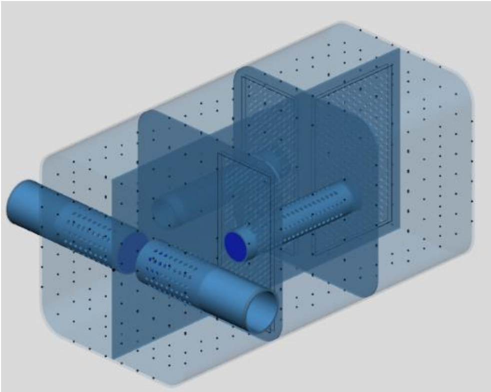
Muffler Chamber				Baffle Pipes			Tail Pipe			Pressure Drop	
Length, L		Muffler Volume		Diameter	Diam. perforations	Porosity	Diameter	Length		ΔP	
L_{min}	L_{max}	VM_{min}	VM_{max}	d_B	d_P	σ	d_{TP}	L_{TPmin}	L_{TPmax}	kPa	mH_2O
m	m	l	l	mm	mm	-	mm	m	m	kPa	mH_2O
0.331	1.720	4.31	6.04	22.8	3	16.12	38	0.363	1.169	0.295	0.030



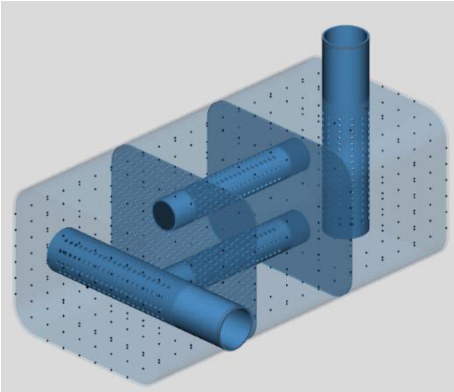
a. Loncin single cylinder ICE muffler



b. Yanmar single cylinder ICE muffler



c. Custom Design 1 single cylinder ICE muffler



d. Custom Design 2 single cylinder ICE muffler

Figure 4 3D models of custom designs and commercial mufflers for single cylinder ICE application

The max values are preferable because they yield better acoustic performance. But, due to size constraints, the smaller components dimensions values are preferable. The calculated value for the pressure losses Δp across the general concept of the muffler is approximately $0.3\ kPa$. This is a relatively low value, which is taken only as an indication of the overall backpressure change due to design and operational changes. Low backpressure indicates a high level of noise generation and the inability of the muffler to sufficiently attenuate for the sound wave propagation.

The analytical results give a good initial estimation of the muffler size and performance regarding its application and operational conditions. But for more in-depth theoretical analysis, a 1D CFD modelling approach is adopted. The modelling is done with the GT-Suite 2018 software pack [15].

Based on the analytical results and size constraints, two 3D muffler model designs have been developed. The geometrical models were created with the software GEM3D, which is a module of the GT-Suite 1D modelling software pack. Two 3D models of commercial, single cylinder engine muffler designs available from Yanmar and Loncin were also created. The GEM3D models of the investigated mufflers are presented in figure 4. The muffler key component dimensions are shown in Table 4.

Table 4 *Muffler key components dimensions*

Parameter	Units	Yanmar	Loncin	Custom Design 1	Custom Design 2
Length	mm	250	250	300	300
Width	mm	125	125	150	150
Height	mm	105	95	150	120
Number of baffle pipes	-	1	1	2	2
Number of baffle plates	-	1	1	3	2
Baffle plates perforations	mm	-	-	4	-
Baffle plates porosity	%	-	-	20	-
Baffle pipe diam.	mm	34	32	25	25
Baffle pipes perforations	mm	11	6	3	3
Baffle pipes porosity	%	4	6.7	20	20
Inlet diam.	mm	34	32	32	32
Inlet perforations	mm	11	6	3	3
Outlet diam.	mm	34	35	32	32
Outlet perforations	mm	11	6	3	3
Wall thickness	mm	2	2	2	2
Rock wool thickness	mm	10	5	10	10

4-GT-Power Numerical Model

The intake and exhaust ports and intake and exhaust system pipework were modelled using detailed CAD geometry to ensure the gas path was represented as accurately as possible.

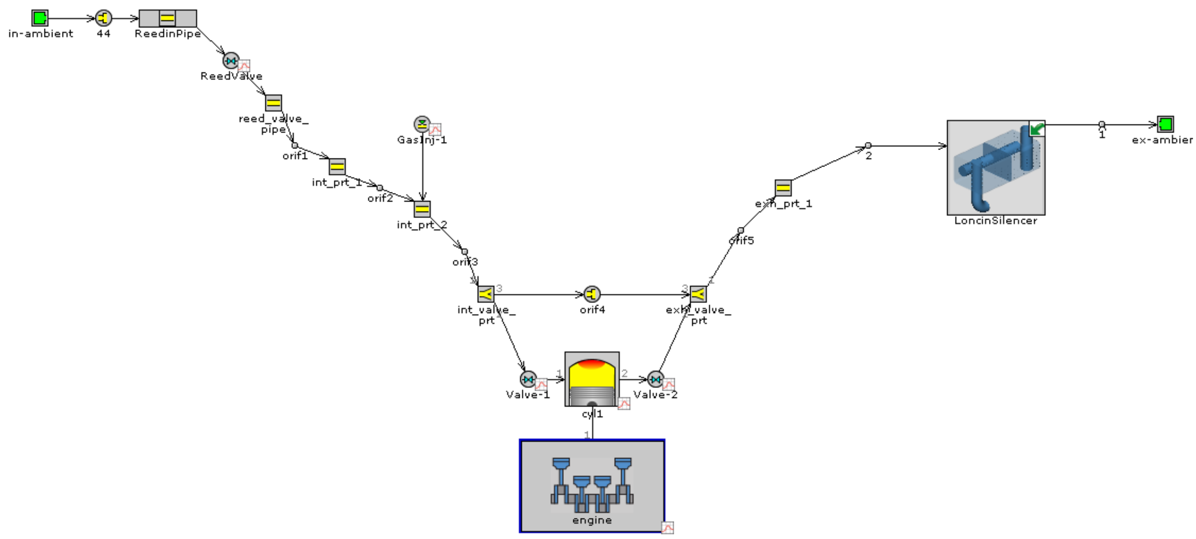


Figure 5 GT-Power model of the single cylinder gas engine

STL files of the cylinder head were created from the CAD model and imported into the GT-Suite software GEM3D for discretisation. GEM3D is a piece of software within the GT-Suite that can be used to discretise a set of complex flow paths into several flow components, which can subsequently be exported into GT-Power.

Individual components were created by discretising the complete flow path into separate pipes or flow splits, with attention being paid to the intake and exhaust ports and the small area where flow can leak from the intake port to exhaust port (or vice versa depending on the pressure difference between within the two flow systems) above the valve head. The discretised flow paths were then converted into effective pipe geometries for use in GT-Power. The final structure of the single cylinder ICE GT-Power model is presented in Figure 5.

4.1-Physical Sub-Models

The GT-Power model is supplemented by several sub-models that are used to represent the complex physical processes that occur within the engine.

Combustion model - the combustion event is to be modelled via a non-predictive methodology, utilising the in-cylinder pressure profile to calculate the burn rate for a given condition.

Poppet valve model - key input data for modelling a cylinder head poppet valve is the variation in the lift with time and the variation in valve discharge coefficient with lift used to describe the effective valve flow area. For the current model, default GT-Power discharge coefficient vs valve lift curves are adopted.

Reed valve model - a reed valve is used in the intake system to throttle airflow. The valve is modelled using a lookup table to describe the variation in valve lift as a function of the pressure differential across the valve. The valve dynamics are modelled via a simplified spring-damper system.

In-Cylinder heat transfer - the in-cylinder heat transfer coefficients are approximated using the 'WoschniGT' heat transfer model which closely follows the classical Woschni [16] heat transfer model.

Friction modelling - the mechanical friction of the engine is modelled using the Chen-Flynn [17] model. The Chen-Flynn model is an empirically derived model that calculates friction mean effective pressure as a function of peak cylinder pressure, mean piston speed and mean piston speed squared.

4.2-Model Validation

The developed gasoline engine model was validated against a limited amount of experimental data. The engine experimental testing was still ongoing at the time of the present study being finalized. The validation results are presented in Table 5.

The Power and Brake Specific Fuel Consumption (BSFC) numerical results are both within a 9% deviation margin compared to the experimental data at the correct air-fuel ratio and fuel injection characteristics.

This computational error is explained by the utilization of a non-predictive combustion model and it was considered reasonable for the purpose of this study.

Table 5 *Validation results for the gasoline engine model*

Parameter	Units	Experimental Results	GT-P Predictions	% Difference
Engine Speed	*RPM*	2,200	2,200	0.0
Torque	*Nm*	26.0	23.9	-8.2
Power	*HP*	8.02	7.38	-8.0
BSFC	*g/kWh*	216	216	0.1
AFR (inducted)	*No Unit*	23	23	-0.1
SOC	*°BTDC*	19	19	0.0
Av Inj. Mass Flow Rate	*g/s*	0.36	0.33	-7.8
Pulse Duration	*ms*	5.3	5.0	-5.5

5-Results and Discussion

The influence of the exhaust muffler design over the engine output torque at different engine speeds is presented in Figure 6. In total, four single-cylinder muffler designs were investigated, two commercial and two custom designs (CD). The commercial mufflers are manufactured by Yanmar and Loncin. To highlight the effect of the exhaust muffler over the engine performance a set of five straight pipes of various lengths were also considered for the investigation. The straight pipes lengths are varied between 250 mm and 2000 mm.

With the increase of the engine speed the torque generation increases for all straight pipe lengths. The engine torque is reaching its peak at about 2,000 RPM, while further increase of the speed results to torque generation remaining relatively uniform. The increase of the aerodynamic resistance of the 2 m pipe at high engine speeds, above 3,000 RPM leads to deterioration of the torque generation. The shortest pipe 250 mm has a negative effect over the torque through the whole engine speed range. This is explained by the influence of the reverse flow effect on the gas dynamics in the cylinder. With the increase of the pipe length, the effect of the reverse flow is subsiding.

In the range of the low engine speeds, up to a 1,000 RPM the highest torque values are achieved by the Yanmar muffler and the 2 m pipe, respectively about 19 Nm and 25 Nm. At 4,000 RPM, the torque generated by the engine equipped with the Yanmar muffler reaches about 7 Nm, a decrease of 65%. The averaged torque produced by the engine with the Yanmar muffler is about 13.5 Nm. The Loncin and the CD1 mufflers have an identical effect over the torque generation. The engine torque curves for both mufflers have an ascending branch in the low RPM operational area, with a peak at about 2,000 RPM. Past 2000RPM, the curve follows a descending trend, reaching its minimum at about 4,000 RPM. The averaged torque generation with the Loncin and CD1 mufflers are 11.4 Nm and 10.9 Nm respectively. The CD2 muffler design has a negative impact over the torque generation. The torque curve is following an ascending trend with its minimum at 1,000 RPM and maximum at 4,000 RMP. The averaged torque generation is about 6.7 Nm.

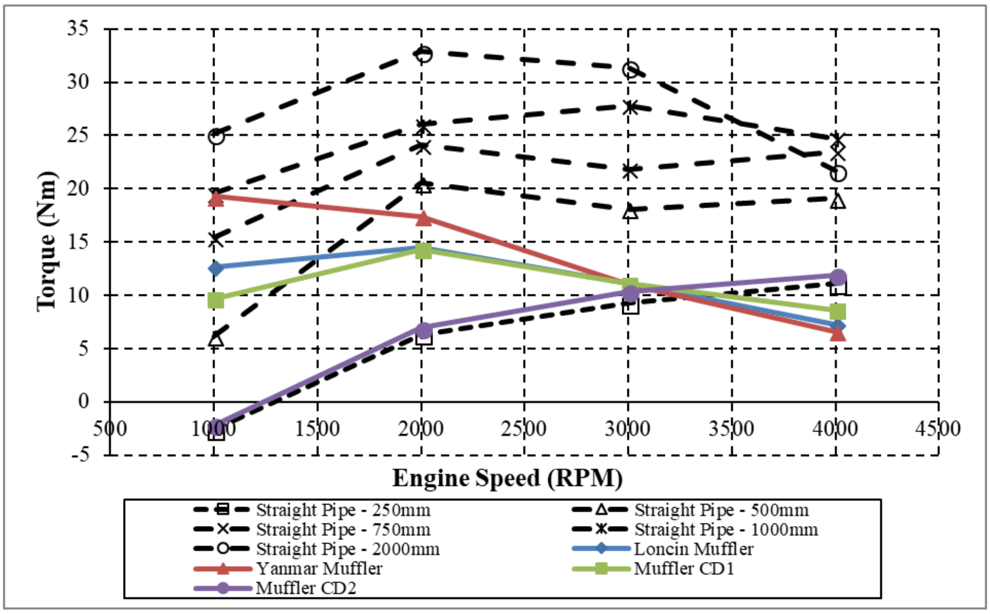


Figure 6 Influence of the exhaust mufflers over the engine torque at different engine speed

Figure 7 presents the overall exhaust system pressure losses at different engine speeds for the different types of mufflers and straight pipes of variable lengths. All pressure loss curves are following an ascending trend with their minimums at 1,000 RPM and maximums at 4,000 RPM. The averaged value of the pressure losses for the 2m straight pipe and the Yanmar muffler are 0.0576 bar and 0.057 bar. This is due to the high frictional losses in the long straight pipe and the pronounced gas flow restriction imposed by the Yanmar muffler. In the case of the Yanmar muffler, the high values of the pressure losses indicate an increase in its back pressure. The increase in the backpressure of the muffler results in improved noise attenuation performance. But, the balance between the backpressure and noise attenuation should be kept in reasonable margins. The recommended backpressure value should not exceed 0.4 bar for engines output power less than 50 hp according to VERT (Verification of Emission Reduction Technologies). The averaged pressure losses for the Loncin, CD1 and CD2 mufflers are 0.032 bar, 0.018 bar and 0.003 bar. The averaged pressure loss of the Yanmar muffler is about 43% higher in comparison with the Loncin muffler. The averaged pressure loss of the CD1 muffler is about 83%.

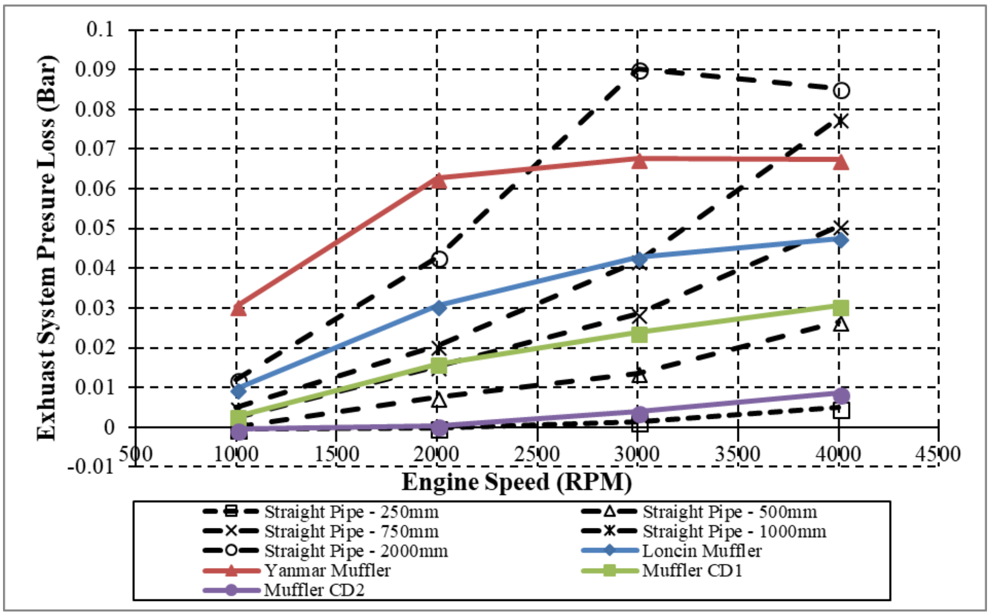
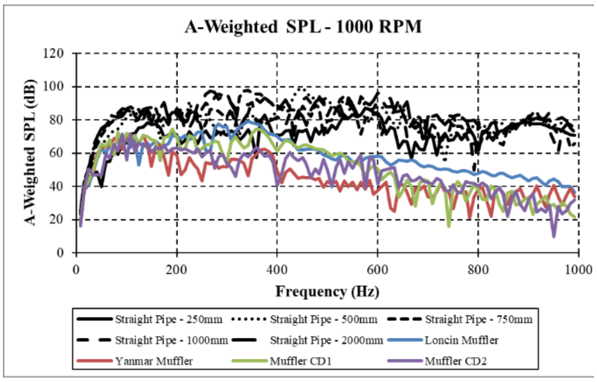


Figure 7 Exhaust system pressure loss at different engine speed

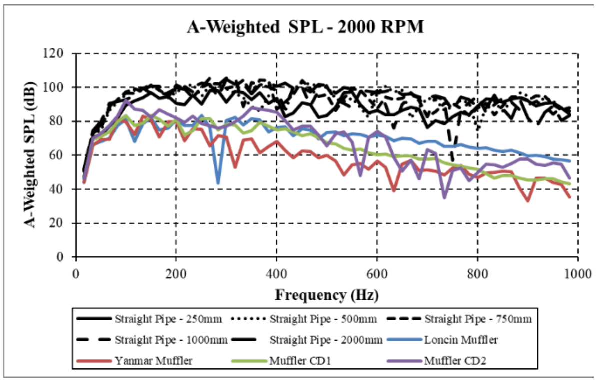
The A-Weighted sound pressure level as a function of frequency at different constant engine speeds is presented in Figure 8. The noise generation levels produced by the exhaust system equipped with straight pipes of various lengths are pronounced. The averaged value of the sound pressure level for the straight exhaust pipes is presented in Table 6. The average value increases with both an increase in the exhaust pipe length and the engine speed. According to the resonance method (Section 3.5), the length of the pipe is directly related to the exhaust system sound attenuation capabilities.

***Table 6** Averaged A-Weighted SPL (dB) at different pipe lengths*

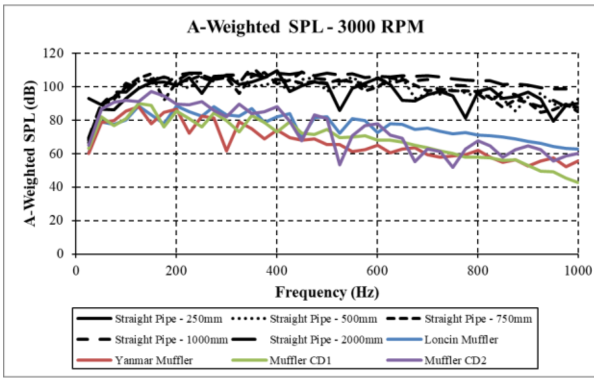
Engine speed, RPM	Averaged A-Weighted SPL (dB) at different pipe lengths				
	Pipe 250 mm	**Pipe 500 mm**	**Pipe 750 mm**	**Pipe 1,000 mm**	**Pipe 2,000 mm**
1,000 RPM	70.6	78.2	79.2	80.7	82.2
2,000 RPM	88.02	92.05	92.86	91.97	93.8
3,000 RPM	96.25	97.77	97.9	99.6	103.08
4,000 RPM	102.14	103.76	105.1	108.59	109.56



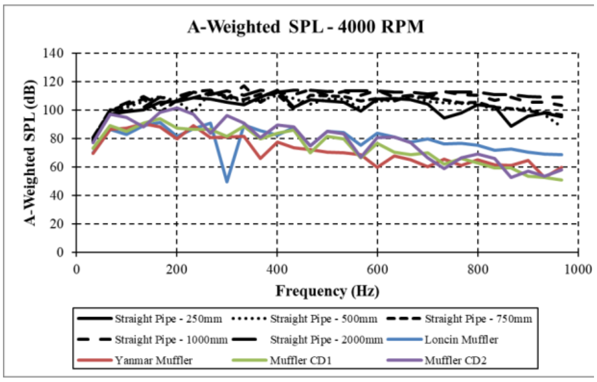
a. A-Weighted SPL at 1,000 RPM



b. A-Weighted SPL at 2,000 RPM



c. A-Weighted SPL at 3,000 RPM



d. A-Weighted SPL at 4,000 RPM

Figure 8 A-Weighted sound pressure level at different constant engine speeds

The results show that the noise attenuation deteriorates with the increase of the length. At the engine speed of 1,000 RPM, the difference between the averaged sound pressure level for the 250 mm and the 2,000 mm pipes is about 16.4%. This percentage gradually decreases with the rise of the engine speed. At speed 2,000 RPM the difference between the shortest and the longest pipes is 6.6%. For the engine speeds 3,000 RPM and 4,000 RPM, the percentage increase is 7.1% and 7.2% respectively. With the increase of the engine speed, the sound attenuation effect of the exhaust pipe length weakens. The noise generation gradually increases for all exhaust pipe lengths with the rise of the engine speed, due to the increase of the engine firing rate and gas flow rate. For the 250 mm exhaust pipe, the averaged noise level rises by 44.7% from 1,000 RPM to 4,000 RPM. For all other exhaust pipe lengths, the increase of the average noise level for the same operational range of engine speeds falls between 32.6% and 34.6%.

As can be seen from Figure 8 the sound generation level produced by the exhaust system equipped with the different design types of mufflers has a descending trend. The mufflers are outperforming the straight pipes of various lengths, due to their prominent noise attenuation capabilities. The averaged A-Weighted SPL for all mufflers at different constant engine speeds is presented in Table 7. The sound level for all exhaust mufflers is rising with the increase of the engine speed, due to the increase of the engine firing ratio, high gas flow rates inducing prominent aerodynamic noise. The overall averaged noise levels throughout the whole engine operational regime are as follows: Loncin 70.28 dB, Yanmar 60.72 dB, CD1 64.98 dB, and CD2 67.41 dB. The lowest averaged values throughout all engine speeds belong to the Yanmar muffler. The Loncin muffler is about 14.3% louder in comparison with the Yanmar muffler. The custom designed mufflers CD1 and CD2 are 6.5% and 9.92% respectively louder than the Yanmar muffler. The second-best sound attenuation performance belongs to the custom designed muffler CD1. The worst performance belongs to the Loncin muffler. All investigated mufflers are reflective types, their internal structure influences the exhaust gas dynamics, thus determining their acoustic and resistive performances.

Table 7 *Averaged A-Weighted SPL (dB) with different muffler types*

Engine speed, RPM	**Averaged A-Weighted SPL (dB) with different muffler types**			
	Loncin	**Yanmar**	**CD1**	**CD2**
1,000 RPM	57.81	45.14	51.54	48.23
2,000 RPM	69.92	58.84	64.78	68.19
3,000 RPM	76.31	67.2	68.92	74.49
4,000 RPM	79.28	71.72	74.69	78.73

The relative sound pressure level fluctuations at different constant engine speeds are given in Table 8.

The relative sound fluctuation is evaluated by the following relation:

$$\psi = \frac{SPL_{Max} - SPL_{Min}}{\frac{(SPL_{Max} + SPL_{Min})}{2}} \tag{25}$$

where SPL_{Max} is the maximum sound pressure level and SPL_{Min} is the minimum sound pressure level.

Table 8 *Relative sound pressure level fluctuation with different muffler types*

Engine speed, RPM	Relative sound pressure level fluctuation with different muffler types			
	Loncin	Yanmar	CD1	CD2
1,000 RPM	1.26	1.17	1.29	1.51
2,000 RPM	0.62	0.81	0.63	0.89
3,000 RPM	0.37	0.51	0.71	0.61
4,000 RPM	0.59	0.53	0.59	0.63

The sound pressure level fluctuations are decreasing with the rise of the engine speed for all muffler designs. At low engine speeds, the exhaust flow rate is lower, thus the exhaust gas velocity is relatively low. This leads to a rise of the turbulence intensity of the gas flow, massive vortices are formed, detached from the muffler wall and transported downstream. Those irregularly formed and detached structures are increasing the pressure and velocity fluctuations along the mufflers. This is directly related to the fluctuation of the sound pressure level. With the increase of the engine speed the exhaust gas velocity rises. The vortex areas are growing smaller and the size of the detached vortex structures decreases, due to the rise of the gas flow energy allowing it to stay predominantly attached to the walls.

The sound pressure level fluctuations generated by the Loncin muffler are decaying with the increase of the engine speed. At 4,000 RPM its relative sound pressure level fluctuation is reduced by 53.17% in comparison with the fluctuations at 1,000 RPM. The Yanmar muffler fluctuations drop by 54.7% with the engine speed increasing from 1,000 RPM to 4,000 RPM. The fluctuation reduction for the two custom-designed mufflers CD1 and CD2 is 54.26% and 58.28% respectively. The best sound pressure fluctuation performance belongs to the custom designed muffler CD2.

Conclusions

In this paper, a procedure for the design development and performance evaluation of internal combustion engine exhaust silencers was introduced. The design development methodology is based on the fundamental gas dynamic and acoustic theories. The methodology is focused on muffler design development for single cylinder internal combustion engines. The introduced methodology provides guidelines for muffler grade selection, sizing of different components, calculation of back pressure as a function of the exhaust gas flow rate. This methodology is a tool allowing for fast initial sizing of an exhaust muffler. The outputs of the components sizing procedure are presented in a range from a minimum to a maximum value. The maximum value component sizes are resulting in better acoustic performance from the muffler. But, the overall size constraints implied by the engine specifics are driving the muffler components sizing towards the minimum side of the range. Based on the introduced methodology, two custom designed mufflers for single cylinder engines were developed.

A 1D GT-Power model of a single cylinder engine was developed for evaluating the acoustic and flow performance of the custom designed mufflers. Additionally, several pipes of various lengths were also considered for the exhaust system performance valuation. All 3D CAD models of the four mufflers were created in GEM3D, discretised and imported to the GT-Power 1D engine model.

The highest values of the generated engine output torque are reached via the application of the straight pipes of various lengths throughout the whole engine operational range. The exhaust system pressure losses rise with the increase of the pipe lengths. Regardless of their favourable effect over the torque generation, the straight pipes exhibit a poor acoustic performance for all investigated operational regimes with averaged values in the range 80 – 110 dB.

Regarding the effect of the mufflers over the engine torque generation performance the Yanmar muffler provides the best performance with 13.57 Nm averaged torque followed by the Loncin and CD1 mufflers with 11.3 Nm and 10.91 Nm. The lowest averaged value of the exhaust system pressure loss belongs to the custom design muffler CD2 0.003 bar. The highest value is reached with the Yanmar muffler, 0.032 bar. The higher the back pressure the better the acoustic performance. The balance between them should be kept in appropriate margins according to the muffler application.

It was found that the muffler with the best noise attenuation capabilities is the Yanmar muffler with an averaged A-weighted sound pressure level of 60.7 dB. Its performance is followed by the custom designed mufflers CD1 and CD2 with 64.9 dB and 67.4 dB averaged sound pressure levels. The Loncin averaged sound pressure level is 70.8 dB. The relative sound pressure level fluctuations for all muffler types are decreasing with the increase of the engine speed. The overall fluctuation reduction with the increase of the engine speed from 1,000 RPM to 4,000 RPM within the boundaries 53.1% - 58.3%. The numerical study shows that the muffler with the best performance is the Yanmar muffler followed by the custom designed muffler CD1 and the Loncin muffler.

The presented combination of analytical and numerical modelling procedures can reduce the overall length of the muffler development stage by eliminating faulty design concepts and refining the muffler's performance parameters.

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