ISSN 2395-1621

Design of Reactive Silencer

#1Mandar S. Narsale, #2Milind S. Ramgir

1mandar.narsale@gmail.com
2milindramgir@yahoo.co.in

#12Department of Mechanical Engineering, J.S.P.M.'s Rajarshi College of Engineering, Pune-33, SavitribaiPhule Pune University

ABSTRACT

Reactive silencers, which are commonly used in automotive applications, reflect the sound waves back towards the source and prevent sound from being transmitted along the pipe. Reactive silencer design is based either on the principle of a Helmholtz resonator or an expansion chamber, and requires the use of acoustic transmission line theory. Structure of current silencers involves a drawback which is essentially focused on the fact that silencers are large if they are to be acoustically effective, which in turn means that they are heavy in relation to the power of engines. More specifically, this problem is because current silencers require a large volume of air to be able to dissipate the frequencies generated by the exhaust gases from the engine. Therefore, in order to obtain a certain acoustic performance a very large and therefore very heavy silencer is required. The dimensions required for silencers are also determined by the need to maintain certain counter pressure levels. The objective of the invention is to obtain a silencer with optimal acoustic performance in relation to its size or volume, as well as a reduction in the counter pressure to the flow of gases, i.e. while at the same time reducing the noise, it also minimizes the power it takes away from the engine of the vehicle. By Mathematical modelling we will calculate dimensions of the silencer.

***Keywords*:- Reactive Silencer, Helmholtz Resonator, Expansion Chamber, Acoustic Performance.**

ARTICLE INFO

Article History

Received:30th September 2015

Received in revised form :

2nd October 2015

Accepted:6th October, 2015

Published online :

9th October 2015

I. INTRODUCTION

Exhaust system is effectively a series of pipes and boxes specially designed to channel emissions away from the front of the vehicle, reduce engine noise and maintain optimum fuel efficiency. The importance of maintaining a healthy exhaust system and the impact a faulty exhaust system can have on the environment is not something people automatically think about. All exhausts produce 6 gases as emissions; of the six three are less harmful (nitrogen, carbon dioxide and water vapour) and three are toxic (carbon monoxide, nitrogen oxide and nitrogen monoxide). The job of the exhaust, catalytic converter and its monitoring system is to maintain the correct balance of emissions, check the engine is running efficiently and move the emissions away from the vehicle occupants. Under existing regulations a Police officer can warrant the removal of any vehicle from the roadways on the suspicion that it is producing excessive amounts of pollutant gases from the exhaust. Also, if your vehicle exhaust system is broken and noisy your vehicle will come under police scrutiny. Your vehicle will also fail its MoT test if the exhaust system has a fault resulting in incorrect emissions levels being recorded. A vital part of today's exhaust systems are the 'Catalytic Converters'. These change the properties of noxious gasses produced by the engine combustion chambers in to a more environmentally friendly emission. All petrol cars manufactured from 1993 have catalytic converters fitted and all diesel cars from 1997. The gas emission level for your vehicle is set by the vehicle manufacturers and enforced at the time of a MoT test; however there has been much debate within the European Commission about whether these settings should become mandatory. If a mandatory level is to be underpinned by law the result could be further regulatory controls to ensure all motorist keep their vehicle emissions within a specified tolerance level.

How does an exhaust silencer work? The pulsating flow from each cylinder's exhaust process of an automobile petrol or diesel engine sets up pressure waves in the exhaust system-the exhaust port and the manifold having average pressure levels higher than the atmospheric. This varies with

the engine speed and load. At higher speeds and loads the exhaust manifold is at pressures substantially above atmospheric pressure. These pressure waves propagate at speed of the sound relative to the moving exhaust gas, which escapes with a high velocity producing an objectionable exhaust boom or noise. A suitably designed exhaust silencer or muffler accomplishes the muffling of this exhaust noise. The basics of silencing can be understood by recalling a few principles of physics. The velocity of sound in the gas at a given temperature is directly proportional to the square root of the product of the pressure and the ratio of the specific heats (at constant pressure to that at constant volume), and inversely to the square root of the density of the gas. As the temperature varies, the velocity also varies directly as the temperature by another square root law involving the product of the coefficient of thermal expansion of the gas and the temperature. The exhaust noise can be reduced appreciably by providing resonance chambers to offset the noise wave effects. This is accomplished by the principle of the Helmholtz resonator. In principle, it comprises the exhaust pipe, which goes through the large volume of a chamber. The axial holes in the exhaust pipe enclosed by the

Chamber allows the gases to vibrate with the large mass of the gases in the chamber (forming a spring-mass vibrating system) and generate the sound of the same frequency but in opposite phase to that which has to be nullified (called anti-sound). To achieve this muffler volume should be proportioned to the engine piston displacement, and inversely proportioned to the engine speed and the square root of the number of engine cylinders. The usual length to diameter (l/d) ratio of the resonator is about 4:1 to 8:1. A small l/d ratio muffler attenuates the sound well for a narrow frequency band, whereas the large l/d muffler attenuates the sound to a lesser degree but over a wider frequency band. The effectiveness of the exhaust system in silencing the exhaust depends also on the relative lengths of the exhaust pipe (from the exhaust manifold to the muffler) and the tail pipe. A ratio of 1:2 is better than 4:1, and 1:1 is the poorest ratio. Since the narrow frequency range limits the resonant chamber application, other features are incorporated in the resonant chamber to produce friction effects and filter off noise effects of other offending frequencies. Provision of baffles, resonator mufflers with end baffles, resonator with center baffle chamber and four-chamber muffler are illustrative examples. In early stationary engines, muffling of the sound was accomplished by allowing the gases to expand by changing the direction of flow or by cooling them with injected water

II. OBJECTIVES

As is well known, current silencers intended for being intercalated in a vehicle exhaust system are structured, among other technologies, by means of a casing based on a tubular frame with two end covers. Said casing houses more or less complicated designs of resonators, an expansion chamber, Helmholtz resonators, etc., in addition to a material that can absorb high frequency noise, generally fiberglass. The most common silencers have an intake pipe leading to the chamber defined by the mentioned casing, through which pipe the gases generated by the engine of the vehicle enter the silencer, possibly one or more intermediate pipes, and finally a substantially cylindrical outlet pipe traversing the cover opposite to the cover traversed by the intake pipe, projecting there from. Internal separators divide said chamber and form the support for the intermediate pipes. The flow of gases in this type of silencers is guided, i.e. the gases pass through multi-resonators. This structure involves a drawback which is essentially focused on the fact that silencers are large if they are to be acoustically effective, which in turn means that they are heavy in relation to the power of engines.More specifically, this problem is because traditional silencers require a large volume of air to be able to dissipate the frequencies generated by the exhaust gases from the engine. Therefore, in order to obtain a certain acoustic performance a very large and therefore very heavy silencer is required. Furthermore, the dimensions required for silencers are also determined by the need to maintain certain counter pressure levels.

The object of the invention is to obtain a silencer with optimal acoustic performance in relation to its size or volume, as well as a reduction in the counter pressure to the flow of gases, i.e. while at the same time reducing the noise, it also minimizes the power it takes away from the engine of the vehicle.

Introduction of charcoal in exhaust circuit to absorb free carbon elements thereby reducing the residual carbon content of exhaust gases.

III.LITERATURE REVIEW

Rogerio Cora, CristianeAparecida Martins, Pedro Teixeira Lacava [1], Evaluate the performance of the Helmholtz resonators to control acoustic instabilities inside combustion chambers in their work. Findings are:The presence of the resonator may change the chamber acoustic behavior. The resonators, in spite of the attenuations close to the design frequency, can amplify other frequencies that originally did not have considerable amplitude. The attempt to increase the conductance was not observed experimentally, the interval of good performance was narrow close to the design frequency 577 Hz, between 560 and 590 Hz.

Igolkin A.A., Kruchkov A.N., Shakhmatov E.V. [2], The efficiency of using mufflers for reducing the exhaust noise of power engineering equipment is justified.A mathematical model is developed which makes it possible to calculate the efficiency of a noise muffler and the time of pressure drop in a pneumoreceiver with regard to an exhaust noise muffler installed. Design pressure dependences agree well with the experimental data, which testifies to the adequacy of the proposed calculation procedure.

F.D.Denia, E.M.Sánchez-Orgaz, J.Martínez-Casas, R.Kirby [3],It is necessary to include the temperature effects when modeling the acoustic behavior. For high material flow resistivity, increasing mean temperature have been shown to deliver a general reduction in the sound attenuation. For axial and radial thermal gradients, although axial temperature variations have exhibited a reduced impact. Therefore, a suitable representation of the thermal effects is required to avoid an over estimation of the silencer performance.For less resistive materials, an increase in temperature and/or thermal gradient has led to a slight drop in the silencer performance in the low to mid frequency range but the opposite trend has been found at higher

frequencies, the transition point shifting to higher frequencies as the temperature gradient and/or mean flow rise.

Xiang Yu, Yuhui Tong, JiePan, Li Cheng[5], proposes sub-chamber optimization for the design of a silencer. A theoretical basis is presented for a description of the overall transmission loss (TL) of the silencer, using the TLs of each of the cascade-connected multiple sub-chambers and the interactions between them. Three typical sub-chamber configurations are considered, representing the effects of varying geometrical parameters, adding internal partitions, and introducing perforated liners. The characteristics of the sub-chambers, influences of the parameters, and the limits of design are investigated to provide guidelines for optimization. It is demonstrated from this study that:By connecting sub-chambers with optimized TLs to tackle different frequency regions, a desired broadband attenuation performance can be achieved.The proposed design scheme with sub-chamber optimization greatly reduces the design variables and calculation costs compared with global optimization, thus offering wider scope in silencer design.

G. Montenegro, A. Onorati, A. Della Torre[6], describes the development and application of different non-linear models: a coupled 1D–multiD model and a coupled 1D-quasi-3D model, to predict the silencer behavior in the time and frequency domains. Second order time and space discretization were adopted in the 3D and quasi-3D approaches, whereas specific coupling strategies were developed to realize the interface between them and the 1D model. In particular, since the 3D relies on a collocated grid discretization, a Riemann solver based method was developed to realize the coupling with the 1D code, while a cell overlapping procedure was exploited to interface the 1D code with the quasi-3D method, in order to fit with the pseudo-staggered grid arrangement. Both a white noise and a single impulse boundary condition have been imposed upstream of the pipe system to excite the wave motion. The integrated 1D–multi D and the quasi-3D approaches were applied to predict the transmission loss of reactive and dissipative mufflers in which the pressure waves can be significantly non-planar, to point out the influence of higher order modes on the acoustical performance.

IV.SYSTEM DESIGN

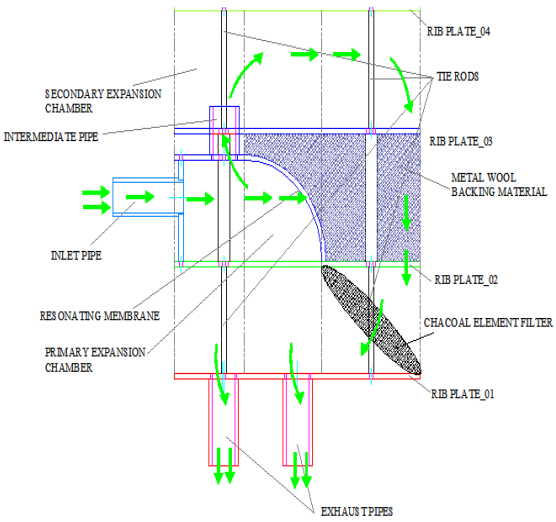

Fig. 1 Schematic of reactive silencer

A. Working of the Reactive Silencer:

An exhaust silencer for automobile vehicles, of the type incorporating a casing determining a gas tight chamber, based on a tubular frame and two end covers, and of those having a gas intake pipe and at least one outlet pipe for such gases, characterized in that the gas intake is located in the middle area of the tubular frame, said intake opening out into an expansion chamber, on the back wall of which the gases collide, which back wall being carried out in a multi-_resonating membrane outside of which the chamber formed by the casing is filled in this area with an absorbent material, two outlet pipes being arranged on one side of the mentioned intake, perpendicular thereto and connected therewith through the mentioned expansion chamber, which outlet pipes internally house respective high frequency multi-_resonators, whereas a Helmholtz resonator neck is arranged opposite to said outlets which opens out into an anti-_resonating chamber located opposite to the gas outlets.

An exhaust silencer for automotive vehicles according to claim 1, characterized in that arranged between the multi-_resonating membrane and the absorbent material there is a layer of steel wool stabilizing said absorbent material, also having provided that this absorbent material also surrounds both the outlet pipes and the resonator neck.

An exhaust silencer for automotive vehicles according to the previous claims, characterized in that the empty anti-_resonating chamber affects a little less than half the casing opposite to the outlet pipes, said anti-_resonating chamber together with the resonator neck defining a dampener for the frequencies causing noise that bothers the ears, while at the same time the multi-_resonating membrane filters high and middle frequencies as the gases collide on it.

The charcoal filter acts as active carbon absorber that frees the exhaust gases discharged to atmosphere of carbon particles suspended as soot in the exhaust gases.

The Reactive silencer design can be divided into two main parts, namely:

- Design of muffler (mechanical aspects of design related to sound wave –energy dissipation & noise reduction)
- Designof resonator
- Tailpipe Design

B. Design of muffler

Muffler Selection:

In order to select a suitable muffler type, some basic information are necessary regarding how industrial mufflers work. Industrial mufflers, (and mufflers in general), attenuate noise by two fundamentally different methods. The first method, called reactive attenuation - reflects the sound energy back towards the noise source. The second method, absorptive attenuation – absorbs sound by converting sound energy into small amounts of heat. There are three basic industrial muffler types that use these methods to attenuate facility noise – reactive silencers, absorptive silencers and anyone or both of them combined with resonator. The proper selection of a muffler is

performed by matching the attenuation characteristics of the muffler to the noise characteristics of the source, while still achieving the allowable muffler power consumption caused by muffler pressure drop. Fortunately, industrial noise sources separate primarily into three different categories with specific characteristics. The first category covers sources that produce mainly low-frequency noise, yet can typically tolerate relatively high-pressure drops. Engines, rotary positive blowers, reciprocating compressors, and rotary screw compressors are types of these sources. It is simply the nature of these machines to produce low-frequency noise and have pressure-volume relationships that are quite tolerant of system pressure drop. These machines are perfectly suited for reactive (chambered) silencers. The second category of noise sources are those that produce mainly high-frequency noise and have performance that is very sensitive to system pressure losses. These sources are almost always moving or compressing a fluid with spinning blades. Examples include centrifugal fans, compressors, and turbines. By definition, this type of equipment is best treated with absorptive silencers for both low and higher temperature applications. Resonators can be used to remove tones from the exhaust spectrum. There are two major industrial facility applications that fall outside these categories, and are best silenced with specific combination reactive-absorptive mufflers. These sources are high-speed rotary positive blowers and high-pressure vents. Both sources have substantial high and low frequency noise content, and can tolerate moderate pressure drop. As a general rule, reciprocating or positive displacement machines should be attenuated with reactive silencers, and centrifugal equipment should use absorptive silencers. For all remaining major noise sources, combined reactive-absorptive silencers are appropriate with many designs available to choose from.

Exhaust Muffler Grades:

Industrial/Commercial:

IL = 15 to 25 dBA
Body/Pipe = 2 to 2.5
Length/Pipe = 5 to 6.5
Residential Grade:
IL = 20 to 30 dBA
Body/Pipe = 2 to 2.5
Length/Pipe = 6 to 10

Critical Grade:
IL = 25 to 35 dBA
Body/Pipe = 3
Length/Pipe = 8 to 10

Super Critical Grade:
IL = 35 to 45 dBA
Body/Pipe = 3
Length/Pipe = 10 to 16
IL= Insertion Loss, i.e., the level of sound reduction after attaching the muffler.
The super-critical grade muffler generally represents the "top of the line" for reactive mufflers. Fig. 4.1 below shows a 3-chamber critical grade muffler. It achieves its "super-critical" status primarily from its length, as much as 16 x pipe diameters.

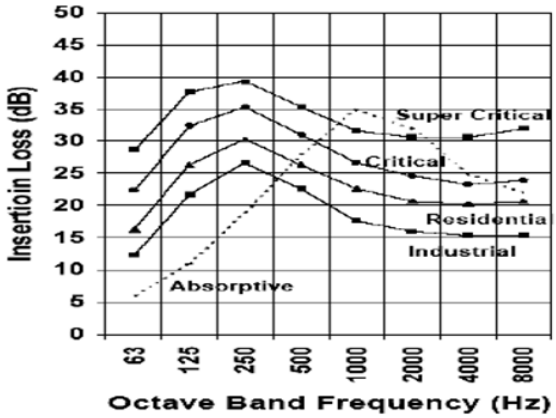

Fig.2 Insertion loss (dB) versus frequency (Hz) curve

Fig. 4.1 shows the approximate insertion loss as a function of frequency for the various grades of mufflers. All values are approximate since no muffler has repeatable IL performance from engine to engine. It can be noted that the IL performance of the absorptive silencer is the best in the frequency region where reactive mufflers start to deteriorate.
Designing and Calculation of Muffler:
A muffler have been designed which is of supercritical grade type and includes all the three attenuation principles i.e., reactive, followed by absorptive type muffler, and a side branch resonator. The interesting events of the design are continuous volume reduction of chambers in the reactive part, the flow pipe cross-sectional area is maintained constant throughout, a layer of insulation outside the reactive part, the placing of side branch resonator compactly, option for tuning the resonator using a screw and cylinder.
Design Data:
For the experiment, an existing petrol engine has been used. Calculations are done on the basis of data collected from the engine; Specifications of the engine available for testing are as follows:
Specifications of Engine:
Make: Crompton Greaves
Model: IK-35
Engine is two strokes Spark ignition engine with following specifications:
Bore 'diameter: 35 mm
Stroke: 35 mm
Capacity: 34 cc
Power output: 1.2 BHP at 5500 rpm
Torque: 2.72 N-m @ 5000 rpm
Dry weight: 4.3 kg
Ignition: Flywheel magneto
Direction of rotation: Clockwise (looking from driving end)
Carburetor:'B' type
Cooling: Air Cooled engine
Lubrication: Mist–via petrol
However, some data are applicable to all engines. For designing, the following data are required.
1. Sound characteristics (Without silencer)
Rpm of the engine= 5500
Load Sound level

Without any load 9.2 kg 104.5 dbA
50% load 15 kg 106.5 dbA
100% load 24 kg 107 dbA

2. Sound analysis with frequency analyzer (to obtain the dominating frequency)

Two dominating frequencies, the low level and the
High levels have been obtained. These are:
Frequency Level Frequency (Hz)
Low 270
High 40000

3. Diameter of exhaust pipe of engine/inlet pipe of muffler

The Exhaust Pipe diameter: 1.0 inch (25.4 mm) ----this is in accordance to the standard mounting flange on the engine exhaust.

4. The theoretical exhaust noise frequency range

From various experiments is has been found that the theoretical exhaust noise frequency is 200-500Hz.

C. Reflective Part Design

S1 = Exhaust pipe diameter = 1.0 inch
The dimensions to determine are that of the chamber
Length L and the body diameter S2.To determine L, Two methods have been used. They are as follows:

1. First method used to determine L

Maximum attenuation occurs when,
$L = n\lambda/4$…….(1)
Where, λ = wavelength of sound (m or ft.)
n = 1, 3, 5, (Odd integers)
Since λ is related to frequency by the speed of sound, one can say that the peak attenuation occurs at frequencies which correspond to a chamber length.

TABLE I.
CALCULATED WAVELENGHTS FROM FREQUENCIES

Frequency	**λ=C/f (m)**	**λ (inch)**	**N=odd integer**	**L(inch) L=n λ/4**
N(Min) 200 Hz	0.5 (λ_{max})	19.6 (λ_{max})	1	4.9
			3	14.7
N(Max) 500 Hz	0.6 (λ_{min})	23.6 (λ_{min})	1	5.9
			3	17.7

From table I, we can determine the length for the engine exhaust specifications length is within the range of minimum 4.9 inches to maximum of 17.7 inches, as maximum conditions as never achieved because engine is always operated under load. Hence length of the muffler smaller section is considered to be 5 inches and length of the overall section can be considered to be 17.7 inches or 450 mm

2. Second method: Range of chamber length considering the temperature of exhaust gas

Another factor which must be considered in expansion chamber design is the effect of high temperature of exhaust gases. This factor can easily be included in the design by using the following equation:
$0.5(49.03\sqrt{°R})/2\pi f \leq L \leq 2.6\ (49.03\sqrt{°R})/2\pi f$ ……. (2)
Where, $\sqrt{°R}$=absolute temperature of the exhaust gas
f = frequency of sound (Hz)
Let the temperature of exhaust is assumed to be 300º F or 759.7º R ,
Putting this value in equation (2) one obtains,

$0.5(49.03\sqrt{759.7})/2\pi 270 \leq L \leq 2.6\ (49.03\sqrt{759.7})/\ 2\pi 270$
(Here, f =270Hz for low frequency reactive muffler)

0.4 ft. ≤ L ≤ 2.04 ft.
From the 1st method, L = 17.7 inch = 1.47 ft.
So the condition of 0.4 ft. ≤ 1.47 ≤2.04 ft. is satisfied

Other parts of reactive muffler design

It has always been considered that the flow path diameter does not reduce at any point. Otherwise, there would be a possibility of back pressure. That is why; the following equation has been used to determine the diameter of the smaller pipes, which are at the outlet of the first two chambers.
$\pi S1^2/4 = \pi d1^2/4 + \pi d2^2/4$
Where, d1 and d2 are smaller pipe diameters.
As both pipes are of the same diameter, one gets,
d1 = d2 = 1.06 inch ≈ 1 inch.
Now, the total length L has been divided into three
Small chamber lengths L1, L2, and L3.
As the pressure is dropping from the first chamber to the next, we reduced the length slightly from the first to the third.

D. Design of Resonator

Resonators are used to attenuate low frequency noise. The principle is to create an opposite phase waveform to nullify each other. The simplest way to produce a wave of opposite phase is to put a reflective obstacle at a distance of $n\lambda/4$, where n = 1, 3,…..other odd integers. A value of n = 1 has been used in the design. A damped resonator has been designed in form of thin membrane of brass as a result, the length of the resonator increases or decreases due to deflection under the action of shock wave. The range of length has been determined in the following way.
The velocity of sound increases 0.6m/sec with an increase in temperature of 1º C. The velocity of sound at 0º C is 332 m/sec. For the design purpose, the minimum temperature has been assumed to be 20º C and the maximum to be 300º C.From previous section, the dominating frequency for low range is obtained to be 270 Hz. The range from 220 to330 Hz has been chosen here. A table has been constructed with various combinations to get the minimum and the maximum wavelengths, which can cover the above range.

TABLE II.
LENGTH OF THE RESONATOR FOR VARIOUS FREQUENCIES

Temp (˚C)	Vel (C) (m/sec)	F(Freq.) (Hz)	λ=C/f (m)	λ (inch)	L= λ/4 (inch)
(min) 20	(min) 344	(max) 330	(min) 0.25	(min) 9.8	2.45
		(min) 220	(max) 330		
(max) 300	(max) 512	(max) 330	(min) 0.250	(Max) 14.1	3.525
		(min) 220	(max) 330		

From Table II, Lmax and Lmin have been found to be:
2.45 inch, i.e., and 3.525 inch, respectively.
Thus the total resonator length becomes,
Lres = Lmax + Lcyl + clearance for lubrication
= 3.525 + 10” +1” = 14.525 inch.

E. Tailpipe Design

According to equation (1), resonance occurs when L = nλ/2. So, for an economical construction, the value of n may be taken as 1. Then the tailpipe must be less than λ/2 i.e. 3”

V.CONCLUSION

This paper contains mathematical calculation of dimensions of the silencer as the need is to maintain certain counter pressure levels.The design of different parts of silencer is done with taking into account all the required dimensions and noise control requirement. The proper selection of a muffler is performed by matching the attenuation characteristics of the muffler to the noise characteristics of the source, while still achieving the allowable muffler power consumption caused by muffler pressure drop. The review of different types of performance measurement techniques of silencer is done.

ACKNOWLEDGMENT

The authorsare thankful to J.S.P.M’s RajarshiShahu College of Engineering, for providing me all the facilities which are vital in doing this project

REFERENCES

[1]Rogerio Cora, CristianeAparecida Martins, Pedro Teixeira Lacava, “Acoustic instabilities control using Helmholtz resonators”, Applied Acoustic, Vol.77, pp.1-10, 2014.

[2]Igolkin A.A., Kruchkov A.N., Shakhmatov E.V. “Calculation and Design of Exhaust noise mufflers for Power Engineering Equipment ”, 8th International Symposium TRANSPORT NOISE AND VIBRATION 4–6 June 2006, St. Petersburg, Russia, pp.1-10.

[3]F.D.Denia, E.M.Sánchez-Orgaz, J.Martínez-Casas, R.Kirby, “Finite element based acoustic analysis of dissipative silencers with high temperature and thermal-induced heterogeneity”, Finite Elements in Analysis and Design, Vol.101, pp. 46-57, 2015.

[4]Soon-Hong Park, “Acoustic properties of micro-perforated panel absorbers backed by Helmholtz resonators for the improvement of low-frequency sound absorption”, Journal of Sound and Vibration, Vol.323, pp.4895-4911, 2013.

[5]Xiang Yu, Yuhui Tong, JiePan, LiCheng, “Sub-chamber optimization for silencer design”, Journal of Sound and Vibration, Vol.351, pp.57-67, 2015.

[6]G. Montenegro, A. Onorati, A. Della Torre, “The prediction of silencer acoustical performances by 1D, 1D–3D and quasi-3D non-linear approaches”, Computers & Fluids, Vol. 71, pp.208–223, 2013.

[7]Lixi Huang, “Attenuation of low frequency duct noise by a flute-like silencer”, Journal of Sound and Vibration, Vol.326, pp.161-176, 2009.

[8]ChunqiWanga, Lixi Huang, “Analysis of absorption and reflection mechanisms in a three-dimensional plate silencer”, Journal of Sound and Vibration, Vol.313 , pp.510–524, 2008.

[9]S.K. Tang, “On Helmholtz resonators with tapered necks”, Journal of Sound and Vibration, Vol. 279, pp. 1085-1096, 2005.

[10]A. Selamet, I.J. Lee, N.T. Huff, “Acoustic attenuation of hybrid silencers”, Journal of Sound and Vibration, Vol. 262, pp.509–527, 2003.

[11]Bryan Willson, Justin Mick and Stephanie Mick “Development of an Externally-Scavenged Direct-Injected Two-Stroke Cycle Engine” SAE TECHNICAL PAPER SERIES 2000-01-2555pp.1-8.

[12]Byung R. Kim “VOC Emissions from Automotive Painting and Their Control: A Review” Environ. Eng. Res. 2011 March, Vol. 16(1),pp 1-9

[13]Timothy V. Johnson “Diesel Emission Control in Review” SAE Int. J. Fuels Lubr., Vol.2, pp.1-12.

