# AN EXPERIMENTAL STUDY ON EFFECTS OF GLASS-WOOL ON TRAVELLING WAVES IN A DOUBLE-CAVITY MUFFLER

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## ABSTRACT

A muffler is a device designed to reduce the emission of the noise produced from cars engines. Reactive muffler of coupled tubes with or without absorbing materials is experimentally studied in the present paper. Finite amplitude oscillations superimposed on cold gas flow is created by a reciprocating engine of a compressor. For coupled tubes, the experimental results had been taken at different speed (different initial frequencies). Capacitive pressure transducers are used to record pressure history variations. These transducers were connected directly to computer through a Lab View soft ware. Also the power spectrum is calculated from experimental data .

In the case of pipe system arrangement consists of large cavity followed by small one without absorbing materials, it is found that there are no big change in the recorded pressure traces before and after the big cavity. In contrast, in case of existence absorbing materials embedded along the big cavity, smooth pressure waves are noticed at the exit of the large cavity. Moreover, adding glass-wool in the adopted double cavity muffler reduces the outlet noise at the muffler exit and increase in pressure downstream of the large cavity based on the generated complex pattern in the system. Results of this investigation are useful for the muffler designers in order to reduce the output noise levels.

كاتم الصوت هو جهاز مصمم للحد من انبعاث الضوضاء الناتجة عن محركات الاحتراق. هذه دراسة معملية لكاتم الصوت مكون من علبتين اسطوانتين موضوع على محيطيهما من الداخل فى صورة حلقية مادة ماصة (صوف زجاجي). العلبتين مختلفتين فى القطر (قطر ٤ ",٦") وموضوعتين على التوالى ومتصلتين بأنبوبة (٢"). هذه الدراسة تبحث فى تغيرات السريان المتذبذب خلال كاتم الصوت فى حالة وجود الصوف الزجاجي وعدم وجوده. يتم توليد السريان البارد ذو التذبذبات محدودة السعة بواسطة ضاغط ترددى يتم التركيب على مخرجه منظومة كاتم الصوت ذو العلبتين.

النتائج المعملية أخذت عند ترددات مختلفة باستخدم محولات الضغط سعوية (Capacitance Pressure Transducer) لتسجيل معدل تغيرالضغط. تم توصيل هذه المحولات مباشرة إلى الكمبيوتر من خلال مجمع بيانات باستخدام Lab View software. كما تم حساب (power spectrum) للنتائج المعملية.

وخلاصة هذا البحث: أن إضافة الصوف الزجاجي له أربعة آثار مهمة وهى تقليل سعة الموجة ، وزيادة قيمة الضغط قبل العلبة ذات القطر الصغير لكاتم الصوت ، سَلاسَة موجة الضغط بين العلبتين لكاتم الصوت وغير مُركب عليها آنتذبذات ، وانخفاض مستوى التذبذبات (الضوضاء) عند الخروج من كاتم الصوت. وبالتالي أداء كاتم الصوت فى حالة وجود الصوف الزجاجي أفضل من عدم وجوده.

نتائج هذا البحث مفيدة لمصممي كاتم الصوت من أجل خفض مستويات التذبذبات (الضوضاء). هذه هي الخطوة الأولى لإستخدام مواد بديله ومختلفه من الصوف الزجاجي أو الرغوى لامتصاص التذبذبات (الضوضاء) غير المرغوب فيها. وعلاوة على ذلك، هناك حاجة إلى استخدام الطرق العددية والتجريبية مستقبلاً لوصف وتفسير النمط المعقد لحركة هذه الموجات خلال كاتم الصوت. وعمل مقارنة بين النتائج العملية والنظرية تساعد مصممى كاتم الصوت.

*Keywords*: Noise waves; Muffler; Acoustic streaming; Sound radiation; Wave finite amplitude; Linear and nonlinear acoustics.

## 1. INTRODUCTION

A muffler placed at the end of a multi cylindrical reciprocating engine is often used to reduce the noise level. This noise is known to be the main pollutant of the present-day urban environment. There are many applications such that, pulsating flow tends to be generated in compressed gas pipelines, in which various pipe elements, such as nozzles, junctions, and diffusers, are included. Also mufflers are commonly installed in heating ventilation and air conditioning (HAVC) ducts. Compression and expansion waves generated in the pipelines pass through or reflect from these pipe elements, and as a result the complicated flow field is formed. Such wave behavior often affects adversely the pipe systems, and their elements can occasionally be broken due to fatigue failure. In exhaust and intake

pipes connected to the reciprocating engines or reciprocating compressors, similar phenomena can present significant problem. For example, small shock waves moving to and from the exhaust pipes have recently been regarded as one of the sources of exhaust noise, Endo and Iwamoto [1]. A pulsating flow in duct with a nozzle is analyzed with random choice method utilizing boundary condition for three way junction [1]. Also they generated the pulsating flow experimentally by a rotary valve from a high pressure chamber. Good agreement was found between their calculation and experiment, especially concerning the location of shock waves. They found that the reflected waves from nozzle move upstream to the rotary valve and are reflected from the rotary valve. Thus the process repeats itself until compression waves become weaker. The larger value the area ratio of the nozzle to the duct leads to a larger the difference in the amplitudes of the pressure histories.

Kawahashi and Arakawa [2] presented a numerical analysis of the acoustic streaming and its effect on the performance of sonic compressor. The streaming of the sound wave attenuation caused by wall friction is called "Rayleigh's streaming". In comparison with the result of one-dimensional analysis, all waveforms are similar, and there are slight differences in the pressure amplitude as the velocity amplitude of the piston increases. This result suggests that the fundamental wave motion in this calculation is not influenced by the effect of the boundary layer. The magnitude and the structure of acoustic streaming were forward to depend on the change of the velocity profile in Stokes layer with the increase of oscillation amplitude. They concluded that, as the amplitude of oscillation increases, the effective thickness of the Stokes layer decreases and the peak position of velocity shifts toward the wall. The structure of calculated acoustic streaming is changed with the amplitude of oscillation. As the amplitude increases, circulatory streaming develops and is then distorted to a complicated structure.

Erickson et al. [3] presented results of finite amplitude oscillations in closed constant diameter ducts and ducts whose cross sectional area varies with axial distance. Solutions were obtained using the Galerkin method. Shock-like waveforms exited in constant diameter ducts were caused by the generation of higher harmonics through efficient non-linear coupling with the fundamental mode. In contrast, the non-linear coupling between the fundamental mode and its' harmonic became weak in ducts whose cross sectional area varies axially, which minimizes the excitation of harmonics in such ducts. Their study had been inspired, for example, by recent interest in exiting large amplitude acoustic oscillations in compressors that use these oscillations to compress the medium and thermo acoustic engines that convert heat into acoustic oscillations, which may be subsequently used to generate electrical

energy or refrigerate a medium. To increase the efficiency of such applications, the amplitude of the oscillations exited with a given power input must be maximized. Their results illustrate shock free waveforms for the exponential horn class of ducts and classical saw-tooth waveform for cylindrical ducts. Furthermore, their observations were reinforced that the use of non-cylindrical ducts can greatly enhance maximum attainable pressure amplitudes and have a marked effect on pressure waveforms.

Yasyji et al. [4] have built a numerical analysis for sound radiation from cylindrical ducts. The structure of the radiated field is numerically obtained for unflanged, flanged, and horned-type ducts, including the effect of mean flow field. As they concluded, the computation gives reasonable results for sound radiation from variable-area ducts with flow. Increase of the wave number $\lambda$ gives a reduction of the reflection coefficient and a shifted directivity to the axis. The reflection coefficient for the unflanged duct is greater than that for the corresponding flanged duct. The change in the geometry of the duct at the open end affects the reflection coefficient and reduces the interference of radiated waves. The effect of inhomogeneous inlet flow on the reflection coefficient and the directivity of radiated field differs from that of a uniform flow. In particular, the sound pressure level at the radiated field is reduced for the case of inhomogeneous inlet flow.

A comparison between pressure pulse measured and CFD simulation using FLUENT code was studied by Hemph [5], and the correlation between measurements and simulations are found to be good. Following this validation, a new pulse converted engine exhaust manifold was created, with the goal to decrease the cylinder residual gas level for a five cylinder turbocharged gasoline engine, currently in development at Volvo Cars. The new manifold is simulated in a 1D-3D coupled CFD simulation, which had to be interrupted prematurely due to excessive convergence times. The results were inconclusive, but indicate poorer performance for the new manifold.

An active silencer to attenuate internal combustion engine exhaust noise is developed by Boonen and Sas. [6]. The silencer consists of an electrically controlled valve connected to a buffer volume. The pulsating flow from the engine is buffered in the volume and the valve resistance is continuously controlled such that only the mean flow passes to the atmosphere. This flow is free of fluctuations and consequently free of sound. The active silencer has been tested on a cold engine simulator. This device generates realistic exhaust noise with the associated gas flow using compressed air. The silencer can attenuate pulsations from engines at very low revolution speed, without passive elements reconnected between the engine

and the active silencer. They concluded that, the acoustic impedance and the source spectrum of the modeled engine are similar to those of a real engine.

An exhaust system was experimentally tested with a forced flow imposed by a flow tester by Perie et al. [7]. Particle Velocity distribution upstream is measured through LDV technique and the aero-acoustic noises created by one specific component recorded. Additionally, Computational Aero-Acoustic (CAA) simulations are also performed for the component. Their numerical results are analyzed and compared to the experimental data. The noise spectra compare reasonably well showing that the method is able to predict those noises. Flow structures creating noise are visualized through classical animation techniques as well as through modal decomposition, allowing identifying for each relevant frequency the noise source patterns as well as the resonance. This methodology is targeted to be applied at early design stages to estimate quality of exhaust components in terms of aero-acoustic noise and to imagine solutions for improvements. They concluded that some necessary improvements are identified to continue that work.

Yonezawa et al. [8] developed (in diesel engine of ships) an actual active exhaust noise cancellation system (AENC). As their results showed, the exhaust noise level could be reduced by an amount more than 20 dB. Also, the occupied space of the system was 50% less than that of the passive silencer with the same noise reduction level. By optimizing the length of pipes, the positions of microphones, speakers and by assembling a passive silencer for high frequency noise, the AENC system can be applied to various speed engines.

Sileem and Nasr [9] studied experimentally the behavior of finite amplitude oscillations in open constant area duct and constant area duct provided with variable area portions. The acoustic wave was exited by an oscillating piston whose frequency equals the natural frequency of the open end constant area duct. Waveform of almost constant amplitude was obtained. In convergent-divergent area case, the wave amplitude declines to a certain value before reaching the quasi-steady waveform. The maximum wave amplitude and the waveform amplitude decreased along the duct. It was anticipated that the waveform was established when a balance occurs between the input energy, viscous and heat transfer energy losses, energy dissipation loss (in case of shock wave formation) and the acoustic energy radiation to the atmosphere. The power spectrum calculations showed that the fundamental frequency of gas oscillation was equals the piston oscillating frequency. Higher harmonics appear in some cases near the end of constant area duct. The energy content of these harmonics was subtracted from that of the fundamental one. The appearance of higher harmonics is an indication of wave steepness.

El-Sharkawy and Nayfeh [10] studied analytically and experimentally the sound wave propagation through a circular duct in the presence of an expansion chamber. They concluded that, the performance of an expansion chamber was greatly affected by the expansion ratio, the sound frequency and the chamber length. The plan wave theory was valid at low frequencies and small expansion ratio. Expansion-chamber mufflers were efficient in attenuating low-frequency sound, which make them ideal for automotive silencing application.

Dissilhorst and Wijngarden [11] studied experimentally and theoretically the effects of an open tube edge on wave amplitude and acoustic energy. They referred to their previous study on acoustic waves in open end tubes with sharp edge. In that work, they stated that jet was formed at the open end during out flow and boundary-layer separation takes place during inflow associated with vortex formation in the pipe near the open end. This causes energy dissipation during inflow. They demonstrated their concept by the sketch in Fig. 1. For open rounded edge tube, they showed the disappearance of vortex and boundary layer-separation. They stated also that, for open end tube rounded edge during out flow, jet formation occurs and the pressure (p) approximately equals the ambient pressure ($P_o$).

$$p = p_o \qquad (1)$$

During the inflow no separation takes place and Bernoulli's theorem gives the following exit condition;
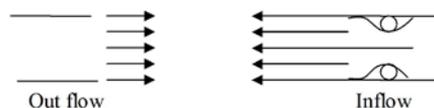


**Fig. 1** Out and inflow boundary condition [10]

$$p - p_o = -\frac{1}{2}\rho_o u^2 \qquad (2)$$

Where $p_o$ and $\rho_o$ are the ambient pressure and density while, $p$ and $u$ are the pressure and velocity at tube exit.

Middelberg et al. [12] studied different configurations of simple expansion chamber mufflers, including extended inlet/outlet pipes and baffles. They modeled such case numerically using Computational Fluid Dynamics (CFD) in order to determine their acoustic response. They showed that the lower frequencies below approximately 1 KHz are in close agreement to another experimental data. For frequencies above 1 KHz their predicted attenuation is above the published experimental results, however the trend is followed closely.

Roeck and Desmet [13] studied two aeroacoustic properties which experimentally analyzed for simple expansion chamber carrying a mean flow which is uniform in time. They showed that the noise generation inside the expansion chamber had a

broadband nature and mainly occurs in the downstream direction, scaling with third power of velocity. This indicates a dipole aerodynamic noise generation mechanism caused by the interaction of turbulence with the expansion chamber walls.

Vanelderen et al [14] studied experimentally an acoustic filter in general, and automotive muffler applications in particular, based on an active two-port formulation, and validated on various muffler configurations. They used a simple rectangular expansion chamber with no flow and an artificial loudspeaker excitation. They concluded that, aerodynamic noise generation is of growing importance with increasing flow velocities and can be damped out by the different interior components of the muffler, such as sound absorbing materials and a partitioning of the expansion chamber.

Finally, the objective of the current study is to examine the effect of glass-wool adapted in pipe system arrangements on the generated complex wave pattern and in turn on the generated noise level from this system. This goal resulted from our review of the article available where previous studies were not exposed to this point.

## 2. THE EXPERIMENTAL APPARATUS AND MEASUREMENTS

The experimental test rig, which has been used in this investigation, is described in the present section. The set-up has been fabricated, constructed, and established in the heat engine laboratory of the mechanical power engineering department, faculty of engineering, Menofya University, Shebien El Kom, Egypt. The pipe system (muffler) is placed at the end of the reciprocating system (compressor). The system consists of two cavities located in the pipe systems as shown in Fig.(2-a). The first cavity has different diameter than the second one. The aim of the experimental investigation was to obtain experimental data to know the pressure traces before and after each silencer duct (cavity chamber) of muffler (see Fig. (2)) (reactive silencer is composed of coupled tubes) in different cases. A glass-wool is used to fill the perimeter of the adopt cavities. The experimental data is obtained with and without glass-wool existence (see Fig. (2-b)). The results are taken at different initial frequencies (17.75, 21.75, and 29.13 Hz) of a reciprocating compressor. The initial frequencies were calculated via the measured rotational speed of the compressor crank shaft, which measured by a tachometer. The pulsating flow direction generated from the reciprocating compressor interact first with the large cavity and then travelling through the smaller one, as seen in Fig (2-a). Two stages reciprocating compressor is used. The commercial glass-wool is used. Its properties as follows; apparent density=29.89 Kg/m$^3$, actual density=524.76 Kg/m$^3$,, and porosity = 0.943. Three capacitive pressure transducers {Model SA, Data Instruments, Action MA01720, USA 0-100

PSIS, accuracy ±1%} are used to record pressure history variations at three stations (1,2, & 3) Fig.(2). These capacitive pressure transducers are connected directly to computer through a Lab View software. The data is recorded in files and plotted as will be shown later. The uncertainties of the measured pressure shown herein are estimated according to the procedures given by Coleman and Steele [15] and Steele and Schneider [16]. It is found to be within ± 2.5% for confidence interval of 95.45%.

## 3. RESULTS AND DISCUSSION

Before considering the results, one should recognize the according to the sudden contraction or sudden expansion of the area (due to reactive muffler), many compression and rarefaction wave transmitted, reflected, and interactions will be created. Thus, the very complicated flow field occurs. This complicated mechanism is still one of the complex problem that need intense computational and experimental work and require an experimental work to interpret this important wave structure. A detailed discussion of the experimental results will be presented in the present section.

The first set of the results show the pressure histories at different initial frequencies (17.75, 21.57, and 29.13 Hz), and different stations (1, 2, and3) for large cavity followed by small one without absorbing materials, as shown in Figs. (3-5). Figures illustrate that: - there is no big difference in pressure before and after the big chamber for all forced initial frequency in this case. Higher harmonics appears in all pressure waves for initial frequency 17.75 Hz and this almost due to the irritation in engine at lower speeds. Higher harmonics decrease when forced initial frequency increased. The amplitude increases for forced initial frequencies 21.57, and 29.13 Hz as shown in Fig.(6). At high initial frequencies, the amplitude at two stations (before and after the large cavity) is nearly the same due to the multi interaction between the wave inside the cavity and the area of discontinuity (junction). Slow compression and fast expansion are noticed for frequency 17.75 Hz. The energy in higher harmonics transfers to the 1$^{st}$ mode, initial frequency 21.57 Hz compression and expansion times are almost equal but at higher initial frequency 29.13 Hz the time of compression decreases and the time of expansion increases. This could be explained as follow, when speed increases the velocity of compressed air increases and this leads to increase of compression wave speed (u + a). However during the closing of cylinder valve the velocity of flow (u) becomes zero. This leads to longer time for relaxation (expansion). When the smaller chamber is downstream the back pressure increases, at location 2 for this case. When the change during compression occurs fast, the process deviates from isentropic condition (very strong compression wave near to shock wave). However, during expansion the process is still isentropic. The

deviation from isentropic leads to wave deformation. Small chamber increases back pressure and decreases exit pressure because of transmitted pressure is larger.

Figure (6) shows the amplitude versus the initial frequency at different stations for large cavity followed by small one without absorbing materials. From experimental data, the pressure amplitude is calculated as follows: $[(P/P_{atm})_{max} - (P/P_{atm})_{min}]$. It is noticed that, at high initial frequency the amplitude before and after the large chamber is nearly the same. At the inlet of small chamber, when incident waves interact with sudden expansion they are partly reflected and partly transmitted at the junction. The strength of reflected and transmitted waves depends on the area ratio [10]. At the outlet of large chamber, the same behavior is observed. Thus according to this very complicated interaction between the transmitted wave from exit cross-sectional area of large chamber and reflected waves from the inlet cross-sectional area of small chamber, the equality of amplitude is noticed at high initial frequencies. This is believed to be due to interact between traveling waves from the inlet (or outlet) large

chamber cavity, small chamber cavity, and muffler duct exit. These interactions and interference of waves lead to wave energy lost or gain, then the wave amplitude changes [9].

The second set of the results illustrate the oscillating pressure histories at different initial frequencies with absorbing material (glass wool) in large chamber as shown in Figs. (7-9). Figures (10-12) are drawn at different initial frequencies with the presence of absorbing material in both large and small cavity chambers. It's noticed that also, wave deformation decreases with speed increasing. This is almost due to the irritation in engine at lower speeds. In these cases the wave energy is partially absorbed and the rest reflected or transmitted or interacts with another waves. These actions depend on the condition of boundary (the existences of glass-wool in large cavity or in both cavities). These waves interaction with elastic boundary is very complicated.
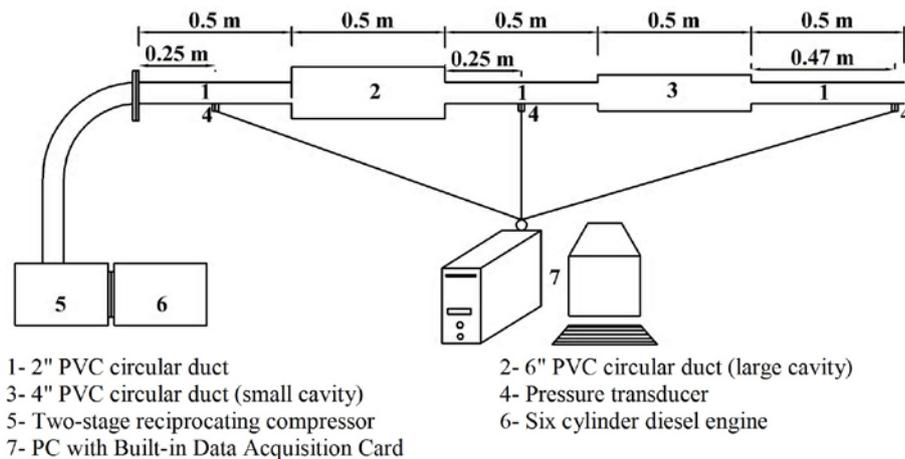


1- 2" PVC circular duct
3- 4" PVC circular duct (small cavity)
5- Two-stage reciprocating compressor
7- PC with Built-in Data Acquisition Card

2- 6" PVC circular duct (large cavity)
4- Pressure transducer
6- Six cylinder diesel engine

**Fig. 2-a** Schematic diagram for the experimental setup



CASE 1: Large cavity followed by small one without absorbing materials (LESE)



CASE 2: Large cavity with absorbing material followed by small one without absorbing materials (LWSE)



CASE 3: Large cavity followed by small one with absorbing materials in both (LWSW)
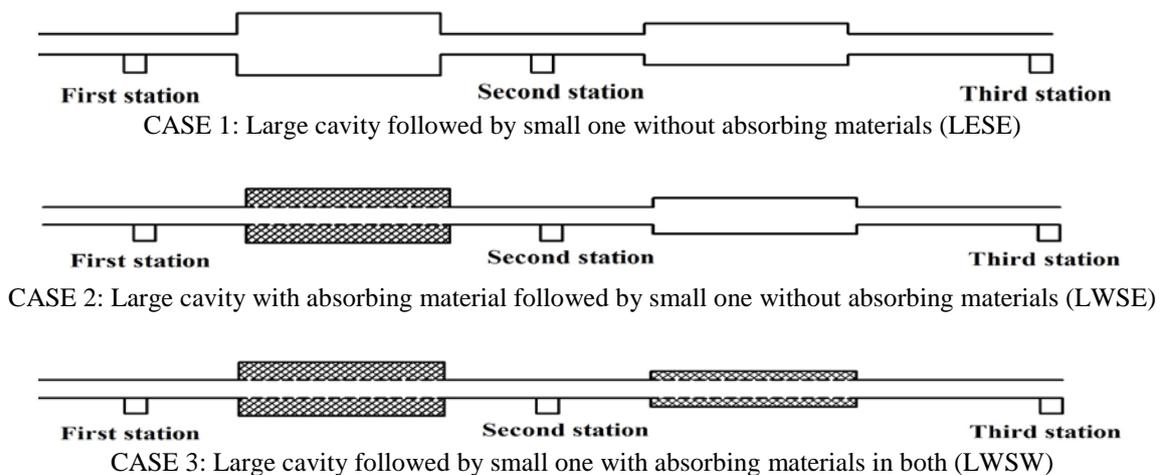
**Fig. 2-b** Schematic diagram for three test cases

The average value of pressure is increasing at the first station with the presence of glass-wool in the large cavity. These effects are due to the existence of elasticity of the foam and multi wave interaction. The region of the glass-wool existence works like a flexible tube wall. Smooth pressure wave after the large cavity chamber (station 2) is noticed. This smooth pressure wave is due to the existence of glass-wool. Glass-wool works as an elastic boundary condition. This elasticity effects are lead to absorption of higher harmonics in the wave form. It is notice that the pressure is almost constant at last station (3).

Figures (13) and (14) show the amplitude versus initial frequency at different stations for large cavity followed by small one in the case of large cavity with glass-wool and two cavities with glass-wool respectively. It's clear that the amplitude at station 1 increases until certain initial frequency then decreases. But at station 2 the corresponding pressure increases. At certain frequency the amplitudes at station 1 and 2 are the same. This means that, at station 1, and due to the glass-wool, the higher harmonic is absorbed through glass-wool and the results of waves interaction is a compression wave and rarefaction wave. At station 2 in the case of glass-wool in both cavity chambers, the glass-wool in two cavities absorbs all higher frequencies and leads to increase the amplitude and smooth pressure wave.

The power spectrum is calculated from experimental data is given in Figs. (15-17). The power spectrum calculation shows that the fundamental frequency of flow wave oscillation equals to the initial frequency. Higher harmonics appear in some cases as shown earlier. The energy content of theses harmonics is subtracted from that of fundamental one. The appearance of higher harmonics is an indication of wave steepness.

## 4. CONCLUSIONS

The effect of glass-wool insertion in one or two chambers of the double-cavity muffler on the traveling of waves through the selected pipe system has been studied.

It may be concluded that the addition of glass-wool has four important effects, a reduction in wave amplitude, increases the mean pressure value upstream the small cavity chamber muffler, smooth pressure wave between two cavity chambers muffler, and low noise at muffler exit. Thus the reactive muffler of coupled tubes with absorbing materials gives better performance than the other without absorbing materials from exit pulsating pressure point of view.

This is the first step towards the research for the use of alternative materials as different glass-wool or foam to absorb undesired noise. Moreover, intensive computational and experimental works are needed to describe and interpretation the complex wave pattern

interaction mechanism in these important pipe systems (muffler). The coupling between the experimental and computational work will help the muffler designer.

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## 6. LIST OF ABBREVIATIONS

LSE    Large cavity followed by Small one without absorbing materials (Empty)

LWSE   Large cavity with glass-Wool followed by Small Empty one

LWSW   Large cavity followed by Small one with glass-Wool in both

1, 2, 3   Three different initial frequencies (17.75, 21.75, and 29.13 Hz)
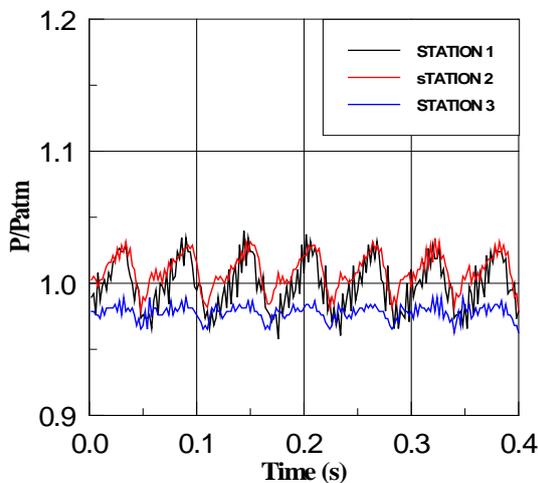


**Fig. 3** Pressure traces at initial frequency = 17.75 Hz for Large cavity followed by small one (muffler) without absorbing materials (LSE1)
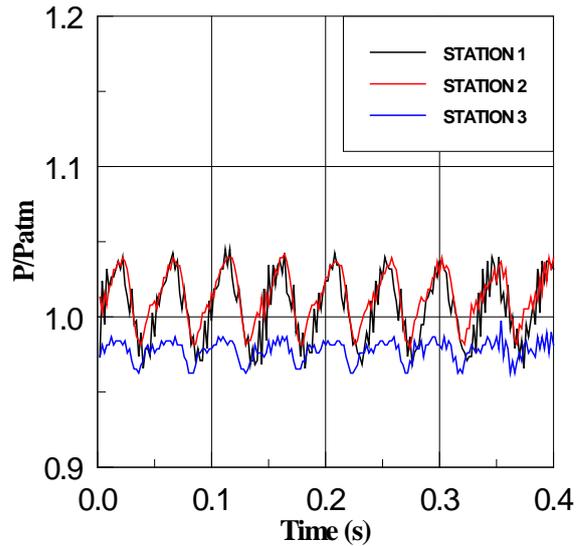


**Fig. 4** Pressure traces at initial frequency =21.57 Hz for Large cavity followed by small one (muffler) without absorbing materials (LSE2).
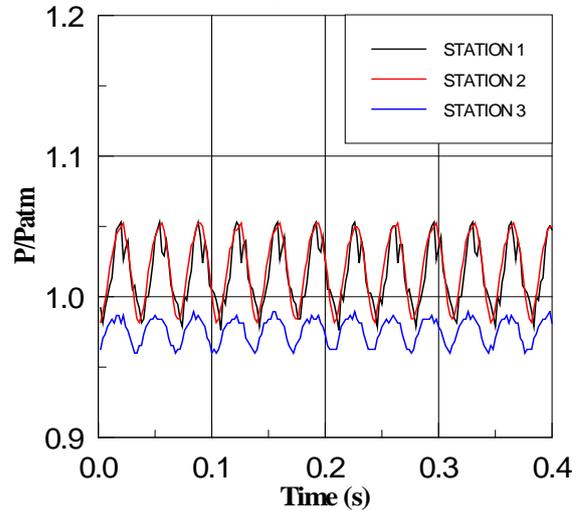


**Fig. 5** Pressure traces at initial frequency =29.13 Hz for Large cavity followed by small one (muffler) without absorbing materials (LSE3)
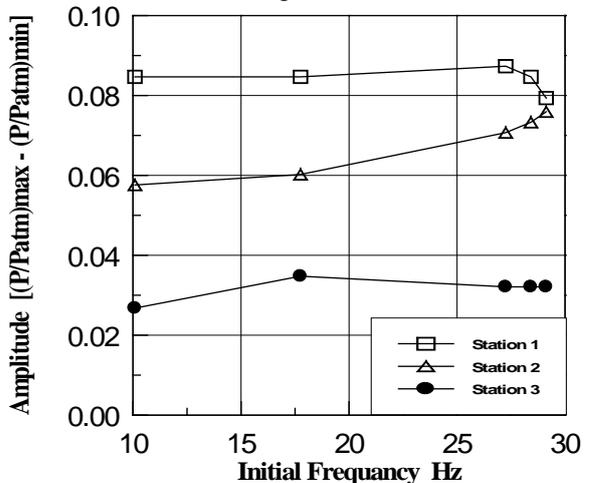


**Fig. 6** Amplitude versus Frequency (Hz) at different stations, Large cavity followed by small one (muffler) without absorbing materials
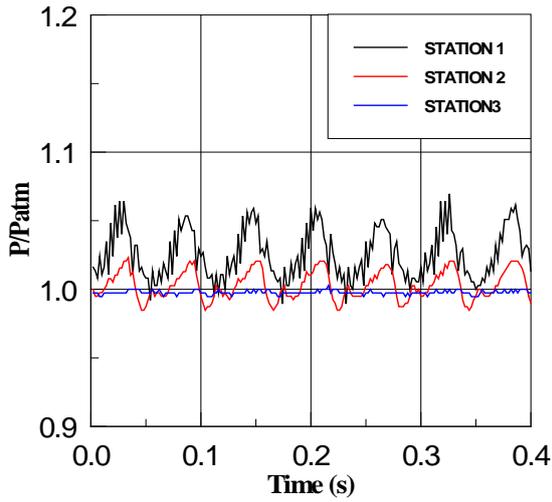
**Fig. 7** Pressure traces at initial frequency =17.75 Hz for Large cavity with absorbing materials (glass-wool) followed by small one (muffler) (LWSE1)
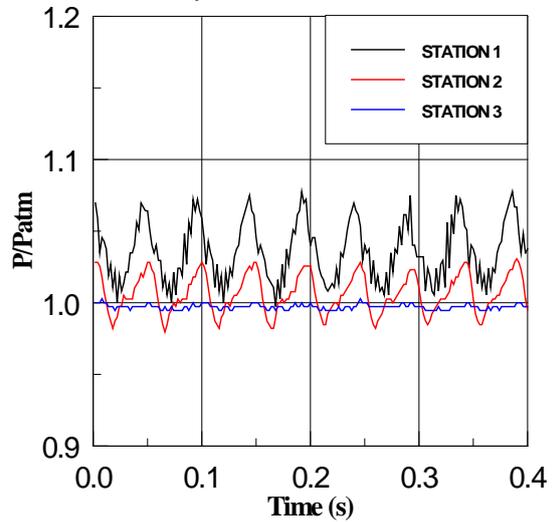


**Fig. 8** Pressure traces at initial frequency =21.57 Hz for Large cavity with absorbing materials (glass-wool) followed by small one (muffler) (LWSE2)
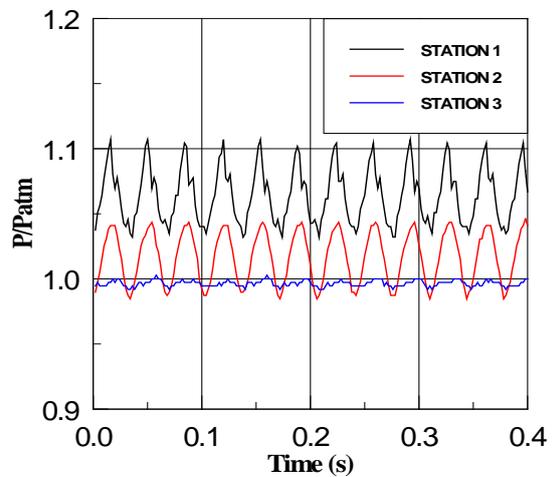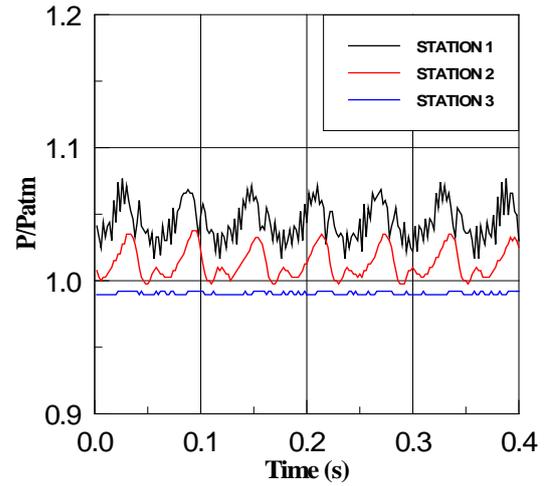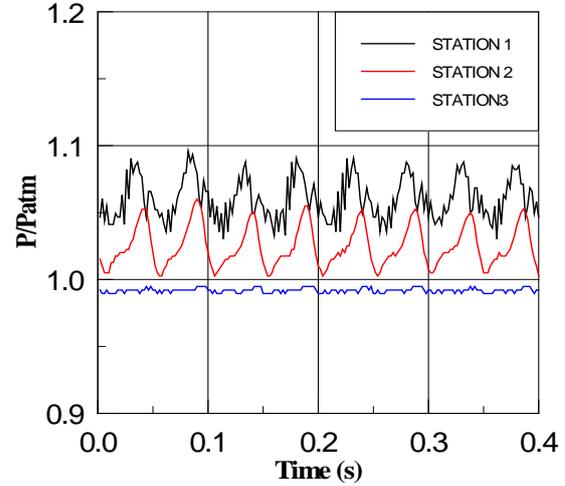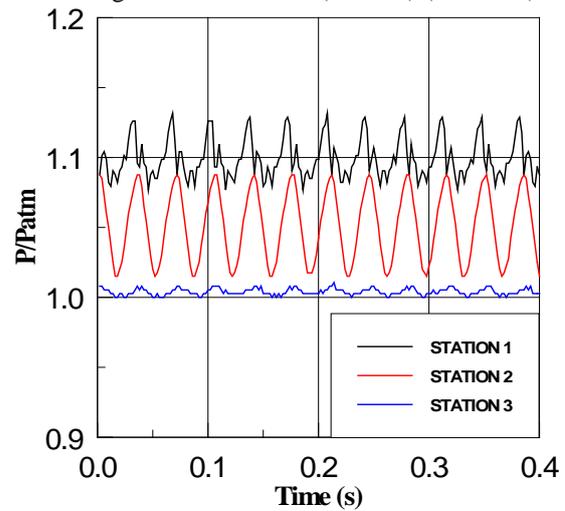


**Fig. 9** Pressure traces at initial frequency =29.13 Hz for Large cavity with absorbing materials (glass-wool) followed by small one (muffler) (LWSE3)



**Fig. 10** Pressure traces at initial frequency =17.75 Hz for Large cavity followed by small one with absorbing materials in both (muffler) (LWSW1)



**Fig. 11** Pressure traces at initial frequency =21.57 Hz for Large cavity followed by small one with absorbing materials in both (muffler) (LWSW2)



**Fig. 12** Pressure traces at initial frequency =29.13 Hz for Large cavity followed by small one with absorbing materials in both (muffler) (LWSW3)
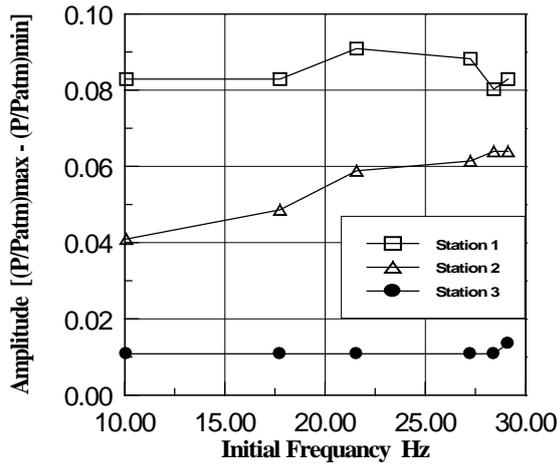
**Fig. 13** Amplitude versus Frequency (Hz) at different stations, Large cavity with absorbing materials followed by small one (muffler)
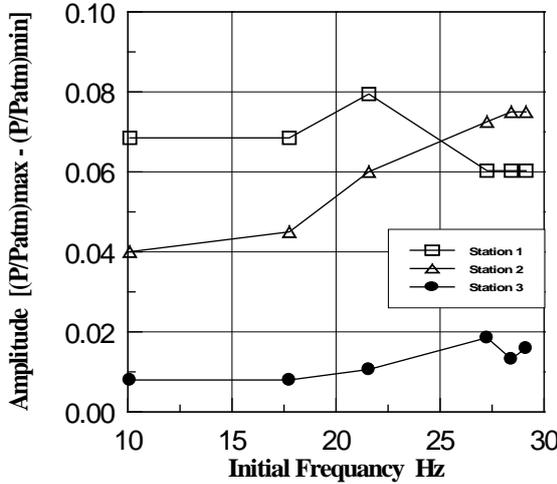


**Fig. 14** Amplitude versus Frequency (Hz) at different stations, Large cavity followed by small one with absorbing materials in both (muffler).
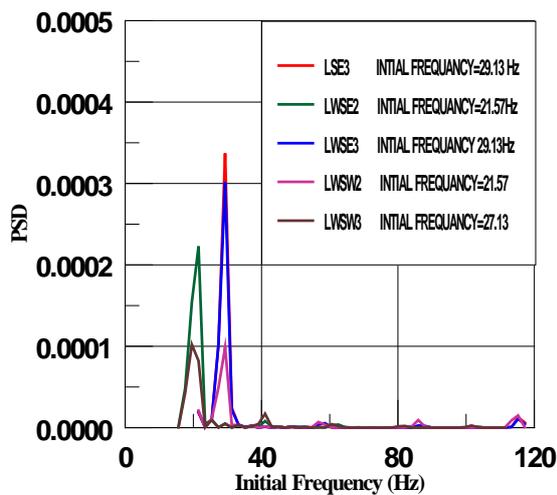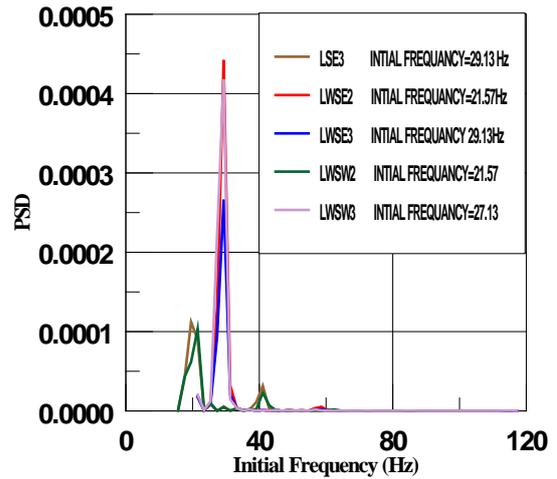


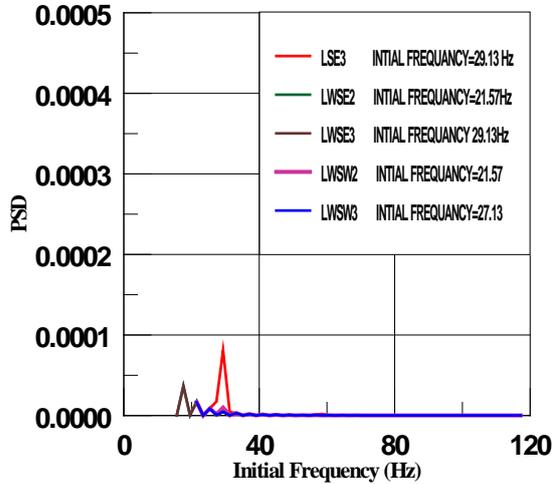**Fig. 16** Power spectrum density (PSD) versus frequencies at the second station



**Fig. 17** Power spectrum density (PSD) versus frequencies at the third station.



**Fig. 15** Power spectrum density (PSD) versus frequencies at the first station