The design of noise attenuation for gas turbine GTS 22 engine

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Abstract

Nowadays, awareness of global society to the green concept where life should be in harmony with nature has been increasing. One of the ideas in the green concept is to reduce excessive sound exposures or noise. Within this context, gas turbine GTS 22 engine is one of mechanical system that generates a very high sound pressure level. Hence, it is instructive to develop noise controls for the gas turbine in order to have acceptable sound pressure level. This research is focused on the design of muffler as noise control treatment after carefully analyzing the dominant noise spectrum of the gas turbine. In this paper, numerical approach is employed to design the muffler. The results are then validated by experimental ones. Moreover, noise reduction (NR) is used to assess the muffler performances. The design of muffler in this research contains all of combination treatment: area cross section expansion chamber, perforated system (micro perforated panel), porous material, partition chamber and concentric tube. The chosen final design of muffler is combination treatment design where NR 38.77 dB is present. This NR leads to sound pressure level of 80.82 dB or 68.95 dBA exist after the muffler applied on the exhaust of gas turbine. The deviation of experimental results compared to numerical ones is 7.14% in average. Moreover, the pressure drop (ΔP) of the muffler is only 0.4042% which is acceptable to keep the mechanical performance.

1. Introduction

Nowadays, awareness of global society to the green concept where life should be in harmony with nature has been increasing. One of the ideas is to reduce excessive sound exposures or noise where related to the health and quality of human life. In this study, the gas turbine GTS 22 engine was used as a noise source. GTS 22 is two spool gas turbine engine which has two shafts are known as low pressure compressor and high pressure compressor. The compressor and turbines can rotate at their optimum speed which will match the local airflow inside the engine. The differences on speed of airflow can be generated the noise. Based on noise measurement, the engine SPL is about 118.5 dB at 160 Hz. Muffler are applied at the exhaust section. There are two approach used for design the muffler, they are in numerically and experimentally. As a reference for this research is uses [1-3]. A hybrid silencer is proposed by introducing MPP into the plate to elicit the sound absorption in order to compensate for the deficiency in the passband caused by the insufficient sound reflection in a certain frequency range due to weaker plate stiffness [4]. The method was used in this study is numerical.
2. Materials and Methods
In this research a combination treatment have been applied, below are brief explanation of each treatment,

2.1. Expansion chamber
The mechanism of expansion chamber to reduce noise is uses continuity. It will appear a discontinuity cause by the changing of cross section area. The attenuation process come from the difference in acoustic pressure which makes a wavefront. A wavefront will miss-match its impedance through different cross section area or called reflected wave [5].

\[ S_1 u_1 = S_2 u_2 \] (1)

2.2. Perforated system (micro-perforated panel)
In this research is uses MPP (micro-perforated panel) Dah-You Maa model with perforation type homogeneous pattern [6]. MPP works by using a helmholtz resonator mechanism which has two main parameters that affect its performance, namely cavity depth and perforation. MPP utilizes the viscothermal effect. MPP is useful to reduce noise on low frequency. These formulations were used to design MPP Dah-You Maa model with homogeneous pattern,

Total impedance,
\[ z = r + jx_m \] (2)

Acoustic resistance,
\[ r = \frac{32 \eta t}{\sigma \rho_0 cd^2} \left( 1 + \frac{k^2}{32} \right)^{\frac{1}{2}} + \frac{\sqrt{2} k d}{c} \] (3)

Acoustic reactance,
\[ x_m = \frac{\omega t}{\sigma c} \left( 1 + \left[ 1 + \left( \frac{k^2}{2} \right)^{\frac{1}{2}} + 0.85 \frac{d}{c} \right] \right) \] (4)

Sound absorption coefficient,
\[ \alpha = \frac{4r}{(1+r)^2 + (x_m - \cot \omega D/c)^2} \] (5)

Perforation ratio formula,
\[ \sigma = \left( \frac{\pi}{4} \right) \left( \frac{d}{b} \right)^2 \] (6)

2.3. Porous material
The attenuation of porous material arises from several mechanisms. One mechanism is viscous boundary layer and the other comes from heat conduction due to the transition of isothermal, isentropic areas, fiber vibrations and friction. Delany-Bazley uses for understand the works of porous material [6,7].

Surface impedance,
\[ z_n = \frac{p_0 u_n}{u_n (0)} \] (7)

Normal specific impedance,
\[ z'_n = \frac{z_n}{\rho_0 c_0} \] (8)

Ratio between reflected wave and incident wave,
\[ R = \frac{(z_n \cos \phi - 1)}{(z_n \cos \phi + 1)} \] (9)

Sound absorption coefficient,
\[ \alpha(\phi) = 1 - |R|^2 \] (10)
2.4. **Partition chamber**

On partition chamber, the expansion chamber will be separated into two or more chambers. In this research, the type of partition chamber is separate expansion chamber in two. It is chosen by considering the weight factor. Then will be varied the hollow space thickness or partition wall length of partition chamber [8].

2.5. **Concentric tube**

Concentric tube works is the same as the helmholtz resonator and micro perforated panel. Concentric tube utilizes the viscothermal effect. In MPP the ratio of perforation should not be more than 20% whereas in concentric tubes the minimum perforation ratio is 60% [6].

2.6. **Sound Pressure Level and Noise Reduction**

In this research there are two formulations used in analyzing the design capabilities of noise attenuation, namely Sound Pressure Level (SPL) and Noise Reduction (NR) [5,8,9].

\[
SPL_{overall} = 10 \log \left( 10^{\frac{SPL_1}{10}} + 10^{\frac{SPL_2}{10}} + 10^{\frac{SPL_3}{10}} + \cdots \right)
\]

\[
NR (dB) = SPL_{source\ overall} - SPL_{receiver\ overall}
\]

2.7. **Methods**

Methods that used in this research is numerical and experimental. Two approach have been used for numerical methods, there are numerical methods based on computational fluid dynamic for air flow case models and finite element method for acoustic case models [9,10]. In simulating gas flow, k-epsilon turbulence model has been used. However the design of this muffler is only viewed from the value of the acoustic performance in gas flow conditions equal to zero. Input data for numerical simulation process are the data of power turbine output gas flow from existing simulation results and SPL data or noise spectrum obtained through measurements on GTS 22 gas turbine engine directly using sound level meter (SLM) at 70000 rpm engine speed for approximately 30 seconds. The measurements gain the highest SPL of 118.5 dB at frequency 160 Hz or 119.59 dB for the overall frequency measurement were used the 1/3 octave band frequency with cut off bandwidth from 100 Hz to 800 Hz.

An experiment muffler model has been developed to validate the numerical methods. The test focused only on acoustic characteristic at zero air speed. Sample muffler in this test are made of acrylic with thickness of 2 mm as a walls and partitions material also uses rockwool with density of 60 kg/m³ as porous material. The process of testing of the sample muffler uses 2 rooms, i.e. reverberation chamber and anechoic chamber (figure 1). In the reverberation chamber a sound source is placed while the sample muffler is placed in the anechoic chamber. Those rooms are separated by a partition consists of gypsum and rockwool material with a density of 60 kg/m³. SLM is used to obtain SPL data on a decibel scale (dB) from an omni-directional speakers (dodecahedron) as a source of noise. In the SPL data collection process, real time SPL data from source and receiver are recorded. From the two existing data, the value of each overall SPL is then calculated. After that, the NR is also calculated [6,8].
3. Results and Discussion

The calculation result of NR for the expansion chamber is shown in Figure 2a. For numerical simulation process the expansion chamber is varied based on the cross section ratio between the inlet muffler area and the expansion chamber area. The ratio expansion chamber starts from 1 – 3 with an interval of 0.25. In Figure 2a there are ratio of cross section expansion chamber that give a negative value NR. This negative value means that there is a reflection wave effect from sound pressure instead of absorbing the acoustic pressure from the sound source. This result agrees with the theory of the expansion chamber that the wider the area of the cross section, the lower value of the acoustic pressure. Hence on designing expansion chamber area of the muffler, it is not only by choose the highest NR value but also seen that for some frequencies the SPL value are not decrease of (attenuated). If in a certain frequency the SPL value has not been attenuated, this means the sound wave coming in is experiencing excessive reflection, so that the SPL value is not reduced but constant or even increased. From the simulation results, it was decided to choose a cross section area with a ratio of 3.

Figure 2b is the calculation results of NR for porous material. For numerical simulation of porous material, the density and thickness are varied. The density variations are 40, 60, 80, 100 and 120 kg/m³ and the thicknesses variation are 25, 50 and 100 mm. Based on the numerical simulation results, the porous material with a thickness of 50 mm and density of 40 Kg/m³ is selected as a design of noise muffler GTS 22 gas turbine. This selection is also based on other considerations such as the overall weight. The highest NR value was given by porous material with a thickness of 100 mm and a density of 40 Kg/m³ but if using porous material with that thick and density it will increased the weight of the muffler. One thing that can be synthesized from the simulation results above is that the greater the density value of a porous material, the less absorption or attenuation ability. The effect that arises from increasing density is the sound pressure wave (acoustic pressure) will be more reflective or less able to be absorbed. Porous material with higher density values is good to be used as insulation rather than as an absorption material. Porous material is used to increase noise reduction capability for medium and high frequencies.

Figure 2c is the NR calculation result of the perforated system in the form of MPP. The numerical simulation process of MPP is conducted by varying on the thickness of the panel. MPP 1 design has a thickness of 1 mm whereas MPP 2 design has a thickness of 0.5 mm. According to the simulation results above, it is known that MPP design 1 gives a higher NR value compared to MPP design 2. This means that MPP design 1 has better noise reduction capability compared to MPP design 2 in the same frequency range. This is in accordance with the existing MPP theory that MPP works by using the Helmholtz resonator mechanism. At Helmholtz resonator there are several main parameters that affect its
performance, namely the cavity depth and perforation. The frequency range and absorption coefficient value are not only determined by the air cavity but also determined by the parameters of perforation including the pore or hole diameter and the distance of pore and the panel thickness.

Figure 2d is the calculation result of NR for partition chamber. Numerical simulation of partition chamber is conducted on varying the thickness or length of the partitions. Partition chamber 1 design has a thickness or 10 mm partition distance (length), while partition chamber 2 design has a thickness or partition distance (length) of 30 mm. As the results of the simulations that have been done above, it is known that design partition 1 gives a higher NR value compared to partition chamber design 2. But it needs to be noted that partition chamber design 2 provides better attenuation on target frequency (160 Hz) than design 1.

Figure 2e is a NR calculation result of concentric tube. The difference between two kind design of concentric tube is the diameter of the perforation hole. Concentric tube 1 design has a 3 mm perforation hole diameter while concentric tube 2 design has a 5 mm perforation hole diameter. According to the NR calculations that have been done above, it is known that the concentric tube design 1 gives a higher NR value compared to concentric tube design 2. This means that the concentric tube 1 design has better noise reduction capability compared to the concentric tube design 2 in the same frequency range.

Figure 2. NR results of acoustic attenuation treatment in numerical simulation. (a) NR for expansion chamber, (b) NR for porous material, (c) NR for MPP, (d) NR for partition chamber, (e) NR for concentric tube and (f) NR for combination treatment
Figure 2f is the calculation result of NR for overall combination of noise reduction treatment. It obtains a value of 38.77 dB or SPL down to 80.82 dB. When it is converted to dBA scale then it becomes 69.94 dBA. After doing the aerodynamic simulation process, the velocity of air through out the muffler is 9.1296 m/s. Hence the pressure drop value of the gas flow in the muffler can be calculated using the Bernoulli equation for incompressible flow. The value of the pressure drop (ΔP) of the muffler GTS 22 gas turbine engine is 0.4042% which is acceptable in relation to maintaining the mechanical performance of the engine.

A sample muffler was used as a tool to validate the numerical result. The sample muffler in this experiment is composed of several noise reduction treatments there are expansion chamber with ratio 2, porous material 50 mm thick and density of 60 Kg/m³ also partition chamber with a thickness or partition distance (length) of 30 mm. In Figure 3a there are differences in the NR values of the two methods used. This is probably because of the effect of the bulkhead between the reverberation chamber and the anechoic during the test. The inlet of the sample muffler is not parallel to the edge of the reverberation chamber and also there is a rockwool at the tip of the sample muffler inlet. Furthermore in Figure 3b, it can be seen that the trendline of the NR value for each frequency can be said has similarities. From the acquisition of NR above and the percentage level of difference or margin error between numerical method (simulation) and the test method (experiment). The percentage of the level of difference (margin error) between the numerical method (simulation) and the test method (experiment) in acoustics is 7.14% which also means that the design approach used in this study can be applied or used.

Figure 3. NR sample muffler. (a) NRoverall sample muffler and (b) NR per frequency sample muffler

The muffler design for GTS 22 gas turbine engine has several noise attenuation treatments that can be used effectively:
1. Expansion chamber ratio 3 by considering the weight factor, gives NR 6.48 dB.
2. Porous material uses 100 mm thick and density of 40 Kg/m³ gives NR 16.16 dB.
3. Perforation system using the MPP with Dah - You Maa model gives NR 6.43 dB.
4. Partition chamber obtains NR 5.76 dB.
5. Concentric tube obtains NR 6.43 dB.
6. NR of combined treatment is 38.77 dB or 80.82 dB SPL (68.94 dBA).
7. The pressure drop (ΔP) is 0.4042% which can able to maintain the mechanical performance.
8. The margin error in acoustics is 7.14% which means the design approach used in this study can be applied or used.

4. Conclusion
The design of muffler in this research contains all of combination treatment: area cross section expansion chamber, perforated system (micro perforated panel), porous material, partition chamber and concentric tube. The chosen final design of muffler is combination treatment design where NR 38.77 dB is present. This NR leads to sound pressure level of 80.82 dB or 68.95 dBA exist after the muffler applied on the exhaust of gas turbine. The deviation of experimental results compared to numerical ones is 7.14% in
average. Moreover, the pressure drop ($\Delta P$) of the muffler is only 0.4042% which is acceptable to keep the mechanical performance.

### Nomenclature

- $d$: decibel
- $dBA$: decibel A weighting
- $NR$: noise reduction (dB or dBA)
- $MPP$: micro – perforated panel
- $SLM$: sound level meter (dB SPL)
- $SPL$: sound pressure level (dB or dBA)
- $\Delta P$: pressure drop (%)
- $\eta$: kinematic viscosity, (kg/ms)
- $t$: panel thickness, (m)
- $\sigma$: perforation ratio, ($\%$)
- $\omega$: angular frequency, $2\pi f$ (rad/s)
- $\rho_0$: gas density, (kg/m$^3$)
- $c$: speed of sound, (m/s)
- $d$: perforation diameter, (m)
- $k$: perforation constant
- $D$: cavity depth, (m)
- $b$: hole to hole center of perforations (m)
- $z_n$: surface impedance (Pa.s/m)
- $p$: complex acoustic pressure amplitude (Pa)
- $u_n$: normal acoustic particle velocity (m/s)
- $z'_n$: normal specific impedance (Pa.s/m)
- $R$: ratio between reflected and incident wave
- $\alpha$: sound absorption coefficient

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