A COMPARATIVE STUDY BETWEEN DIFFERENT HELMHOLTZ RESONATOR SYSTEMS

Md. Amin Mahmud [*1], Md. Shahriar Islam[2], Md. Zahid Hossain[3], Mir Md. Maruf Morshed[4]
[1] Dept. of Mechanical and Chemical Engineering, Islamic University of Technology (IUT), Bangladesh.
[2] Department of Mechanical and Chemical Engineering, IUT, Gazipur, Bangladesh.
[3] Department of Mechanical and Chemical Engineering, IUT, Gazipur, Bangladesh.
[4]Department of Mechanical Engineering, Jubail University College, K.S.A.

Résumé

Cet article propose un résonateur d'Helmholtz cylindrique connecté à un conduit pour en atténuer le bruit de basse fréquence et optimisé en utilisant la méthode des éléments finis. Au départ, un système de résonateur à deux degrés de liberté identiques (2-DDL) a été ajusté en termes de position relative et de l'orientation en adaptant une publication existante sur un système de résonateur unique à 2 DDL. La meilleure position relative des deux résonateurs est étudié en les plaçant en deça des ventres et noeuds de la longueur d'onde de la source sonore et de l'orientation relative optimale entre les résonateurs se trouve également à un espacement relatif optimal. Enfin, le système optimisé à deux degrées de libertés de résonateurs identiques a été comparée à trois études publiées d'un unique résonateur à 2 DDL, d'un unique résonateur à 1-DDL et à deux systèmes résonateurs à 1-DDL, et les résultats indiquent que le systèmes à deux résonateurs identiques 2-DDL peut fournir une gamme plus large d'atténuation pour une bande passante de 300 Hz. Les résultats analytiques et numériques de perte de transmission obtenus pour l'adaptation d'un unique système à2-DDL ont été validés avec les résultats expérimentaux publiés et une bonne concordance a été trouvée.

Mots clefs : bruit basses fréquences, résonateur d'Helmholtz, 1-DDL, 2-DDL, méthode des éléments finis

Abstract

This paper proposes a best fit cylindrical Helmholtz resonator system attached to a duct for attenuating the low frequency noise using finite element method. Initially, an optimized system of two identical 2-DOF resonators in terms of relative position and orientation by adapting a published single 2-DOF resonator system. The optimum relative position of the two resonators is studied by placing them at less than antinode, antinode and node of the wavelength of sound source and the optimum relative orientation between the resonators is also found out at that optimum relative spacing. Finally, the optimized system of two identical 2-DOF resonators is compared with three published studies of single 2-DOF, single 1-DOF and two identical 1-DOF resonator systems and the results indicates that two identical 2-DOF resonator system provide a comparatively broader range of attenuation for a 300 Hz bandwidth. The obtained analytical and numerical results of transmission loss for the adapted single 2-DOF have been validated with that of the published experimental result and a good agreement is found.

Keywords: Low frequency noise, Helmholtz resonator, 1-DOF, 2-DOF, Finite element method

1 Introduction

Noise sources with a dominating content of low frequencies are prevalent in many occupational environments [1]. Low frequency noise (LFN) is defined a noise with a dominant frequency content of 20–200Hz'' [1] and 100 Hz 1/3 octave band as the upper end of the LF range [2]. One significant characteristic of LFN is that individuals suffering from LFN annoyance describe it as omnipresent and impossible to ignore due to the effects of vibration which are unable to locate and difficult to tune out [3]. Closing doors and windows in attempt to diminish the effects of LFN make the noise worse, due to the propagation characteristics of LFN and the low-pass filtering effect of structures. Individuals often become irrational and anxious as attempts to control LFN fail, serving only to increase the individual's awareness of the noise [3]. The number of industrial noise sources capable of creating LFN is increasing day by day as the plant and equipment size becomes larger and automotive, aerospace industries flourishing [2].

Herman Von Helmholtz formulated a theoretical formula for calculating the resonance frequency formula developed ; a wide volume or cavity connected with a narrow orifice or neck was created in the 1850s by Hermann von Helmholtz to detect the various frequencies or musical pitches existing in music and other complex sounds [4]. Since then, it has been extensively studied in various modifications and used in compressor lines, automotive silencers and mufflers, ventilation ducts, aerospace and architectural structures to achieve a more effective solution in low frequency noise attenuation.

* amahmud@alumni.ubc.ca
Current-Graduate Research Assistant, Acoustics and Noise Research Group, UBC, Vancouver, Canada.

The classical theory of a Helmholtz Resonator as an equivalent spring (cavity) and mass (neck) was developed by Rayleigh in 1945 [5]. Selamet et. el. [6] studied the effect of multidimensional propagation and showed deviations in resonance frequency from the lumped parameter analysis, particularly at low l/d ratios. To explain the non-planar sound propagation in both the neck and the cavity selamet et. el adopted multi-dimensional analytical techniques to study the noise attenuation in Helmholtz resonators with circular concentric cavity [7]. A mechanically-coupled resonators model mounted on a one-dimensional duct was developed for providing a wider bandwidth of attenuation and attenuating disturbances of varying frequency by Griffin et al. [8]. The relationship of closely spaced 1-DOF Helmholtz resonators in a 1-D plane was experimentally studied by Soh [9] which showed that when the relative spacing of two identical resonators was greater than a quarter wavelength apart, the transmission loss was greater than that of a single resonator while it declines as a result of the interaction between them when the relative spacing between the resonators is close (less than a quarter wavelength. D. Li [10] experimentally studied the effect of the resonator position on the noise attenuation and the relationship between two or more closely spaced different 1-DOF resonators near the interior wall of a chamber core cylindrical fairing on the basis of Soh [9]. Xu [11] developed the lumped-parameter theory for the piston-driven model system and examined the effect of geometry on the acoustic characteristics of side branch 2-DOF Helmholtz resonators. Farooqui, M. [12] investigated Helmholtz resonators of varying shapes and found that the noise attenuation in cylindrical resonator is better than others and the performance of noise attenuation increases significantly where the crest of the wave falls for the array of 1-DOF Helmholtz resonators located in same plane. S.Mekid and M.Farooqui [13] proposed a new design methodology for both single 1-DOF and 2-DOF Helmholtz resonators attached to pipelines to achieve optimized transmission loss. In our previous literature, Amin *et al*. [14] proposed an optimized two identical (same frequency) 1-DOF resonator system by a thorough investigation of the effect of the relative spacing and geometry of the cavity and neck on the transmission loss. Moreover, an optimized two different frequency 1-DOF resonator system in terms of relative position, orientation and geometry of both cavity and neck was studied by Amin *et al*. [15].

The objective of this study is to achieve the best fit cylindrical Helmholtz resonator system for a 300 Hz bandwidth by studying the effect of relative positions and orientations on the transmission loss of the two identical 2-DOF resonator system attached to a duct in a three-dimensional plane by a comparison with three other published resonator systems.

2 Method

2.1 Analytical Approach

Adaptation of a Single 2-DOF Helmholtz Resonator

The design method for a single 2-dof Helmholtz resonator has been adapted from Mekid [13].The resonance frequency (f) and transmission loss (TL) for a single 2- DOF Helmholtz resonator attached to a duct can be expressed by equations (1) and (2) [13] as follows

$$f_{1.2} = \frac{c}{2\sqrt{\pi}} \sqrt{\left(\frac{\alpha}{V_1}+\frac{\beta}{V_1}+\frac{\beta}{V_2}\right) \pm \sqrt{\left(\frac{\alpha}{V_1}+\frac{\beta}{V_1}+\frac{\beta}{V_2}\right)^2 - 4\frac{\alpha}{V_1}\frac{\beta}{V_2}}} \quad (1)$$

$$Tl = 20log_{10}\left|\frac{\alpha}{\left[2a_d\left(ik+\frac{\alpha}{ikV_1}\left(1-\frac{V_2}{V_2+V_1-\frac{V_2V_1\,k^2}{\beta}}\right)\right)\right]}\right| \quad (2)$$

where, the dimensional parameter ratios $\alpha = \frac{a_{n1}}{l_{n1}}$, $\beta = \frac{a_{n2}}{ln_2}$,and $V_1 = a_{c1}.l_{c1}$,$V_2 = a_{c2}.l_{c2}$ are the volumes of the first and the second resonator, a_{n1}, a_{n2} are the area of cross-section of the first and the second neck and l_{n1}, l_{n2} are their respective lengths, a_d is the cross-sectional area of the duct and k is the wave number. The transmission loss can be calculated using the equation 2 rewritten with respect to α and β for obtaining the optimum $\frac{\alpha}{\beta}$ or maximum transmission loss. The length of the neck is corrected by adding end correction factors considering higher wave propagation effects between the neck, cavity and duct. The end correction of a single 2-DOF Helmholtz resonator is calculated as follows [12].

$$l'_{c1} = l_{c1} + \delta_v + \delta_p \quad (3)$$

$$l'_{c2} = l_{c2} + 2\delta_v \quad (4)$$

$$\delta_v = 0.85\left(1 - 1.25\frac{\mathrm{R_{neck}}}{\mathrm{R_{volume}}}\right) \quad (5)$$

$$\delta_p = 0.82\frac{R_{neck}}{2} \quad (6)$$

This dimensions and parameters of the adapted single 2-DOF Helmholtz resonator [13] are illustrated in Figure 1 which is used to design the two identical 2-DOF resonators presented afterwards in this literature.

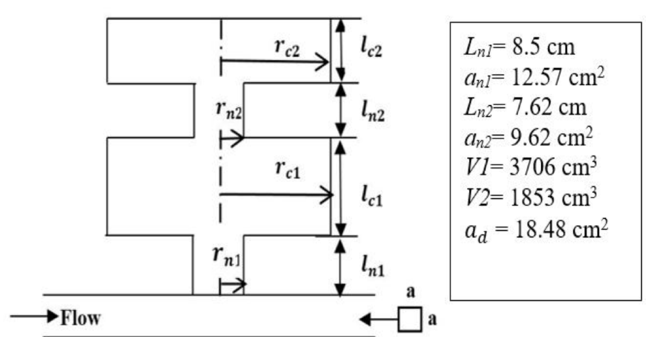

Figure 1: Single 2-DOF Helmholtz Resonator [13].

2.2 Numerical Approach

The transmission loss of two 2-DOF resonators is numerically computed using the ANSYS APDL FLUID module. A full acoustic harmonic analysis is performed for the resonator which calculates the pressure distribution at various frequencies at a specific frequency range. Uncoupled acoustic element FLUID221, a higher order 3-D 10-node solid element that exhibits quadratic pressure behavior is used for modeling the fluid medium. To avoid the complexity of numerical method, fluid structure interaction is neglected and considering structure absent, the entire finite element model of resonator system is considered as a fluid domain such as air. The material properties of air are: $\rho 0$ is density of air = 1.2041 kg/m3 and $c0$ is the speed of sound in air = 343.24 m/s. A sound source of 150 dB is considered from the noise of the jet engine at 1m distance [17] even beyond 130 dB, threshold of pain region of human being according to the Fletcher-Munson equal-loudness-level contours (1933) [18].

The main duct is considered as a square cross section of 4.3×4.3 cm like [13] and 1.5 meter long with two 2-DOF resonators designed as two cylinders which dimensions are shown in Figure 1. The mesh size is 0.11, wave/10, where the wave =1.1m corresponding to 300 Hz, maximum working frequency considered to accommodate the whole sound wave in the duct.

PML has not been used in FE model, rather surface boundary condition is applied to the nodes of the FE model which considers the sound pressure is damped at the impedance boundary and it is used to approximate infinity. Normal velocity excitation is applied on the transparent pressure wave port (an exterior surface on which incident pressure is launched into the acoustic model and the reflected pressure wave is fully absorbed by a defined matched impedance that represents the infinity) of inlet equivalent to the sound pressure of the source**.** To absorb the reflected wave on the transparent port, the impedance boundary is applied to the nodes of inlet and the radiation boundary is applied to the nodes of outlet equal to Z_0 , expressed by the equation 7 [16].

$$Z_0 = \rho 0 * c0 \quad (7)$$

Normal velocity excitation can be expressed by the equation 8 [16],[18].

$$u' = -\frac{p'}{\rho 0\ c0} \quad (8)$$

where, u' is the normal velocity excitation, p' is the equivalent sound pressure in pascal $\rho 0$ is density of air and $c0$ is the speed of sound in air.

The generated acoustic resonator model is shown in Figure 2. Then the solution was performed using sparse solver with tolerance 1e-008, selecting the analysis type as harmonic and the frequency range from 0 to 300 Hz. From the obtained results, the sound pressure of the outlet was taken at the proximal node of the outlet and converted to output sound pressure using equation 9 [18].

Output Sound Pressure Level = $20log_{10}$(sound pressure at output/reference pressure level) (9)

which is used to measure the transmission loss (*TL)* from the sound pressure drop at the outlet from the inlet.

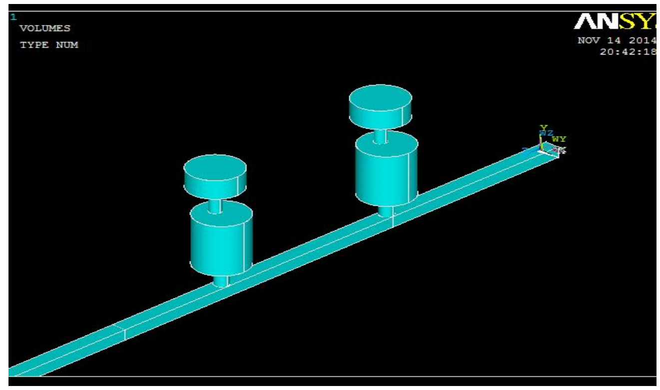

Figure 2: Numerical model of two identical 2-DOF resonators along with the duct.

2.2.1 Relative Positions of Two Identical 2-DOF Resonators

The effects of the relative positions of the two identical 2-DOF Helmholtz resonators on the noise attenuation are investigated to obtain the optimum relative position. For convenience, the relative positions between the two resonators are denoted as Z, where Z is the centre distance between the resonators, closely spaced; ($< \frac{\lambda}{4}$), antinode ; ($\frac{\lambda}{4}$) and node; ($\frac{\lambda}{2}$) where $\lambda = 1.1$ metre, wavelength corresponding to 300 Hz. maximum working frequency of, as shown in Figure 3 which are described as close, antinode and node respectively later in this paper. Both the 2-DOF resonators was modelled identically, using same dimensions of geometry of the cavity and neck as mentioned in Figure. 1 for a single 2-DOF resonator [13]. The relative positions of two 2-DOF resonators were optimized in this investigation and is considered for all further investigations of this paper.

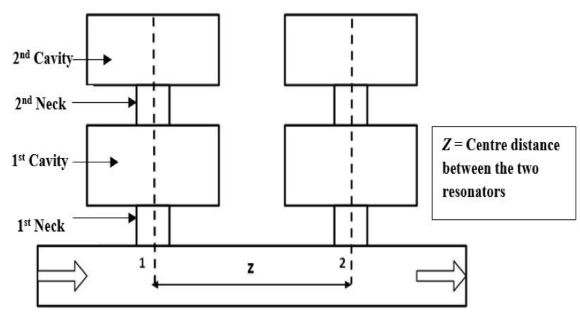

Figure 3: Relative spacing of two identical 2-DOF resonators.

2.2.2 Relative Orientation of Two Identical 2-DOF Resonators

The effects of the relative orientation of two identical 2-DOF resonators on the noise attenuation are investigated to obtain the optimum relative orientation at the obtained optimum relative position from the section 2.2.1. Initially there is no relative orientation between the two resonators. Afterwards, the 2nd resonator was rotated 90 and 180 degrees respectively about the 1st resonator as illustrated in Figure 4.

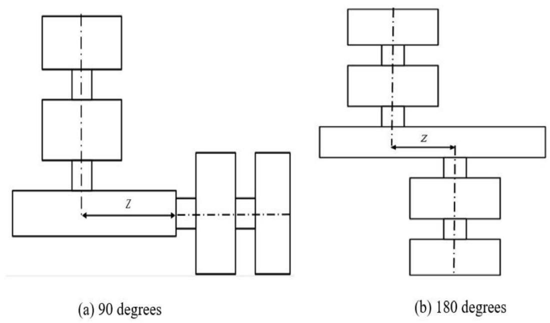

Figure 4: Relative orientation of two 2-DOF resonators.

3 Results

3.1 Relative Positions of Two Identical 2-DOF Resonators

As can be seen in Table 1 and Figure 5 that the obtained 1st and 2nd resonance frequencies, *f1* and *f2* of two identical 2-DOF resonators are identical, 73 Hz and 166 Hz respectively for all 3 relative positions. The numerical results of maximum *TL* are 83.61 dB, 86.22 dB and 92.24 dB at *f1,* 73 Hz relative to that of 63.15 dB, 69.12 dB and 73.80 dB at *f2,* 166 Hz for close, antinode and node position respectively for a 150 dB sound source which indicates that the *TL* is relatively high at and around the both resonance frequencies. Moreover, it is very interesting to observe that at both resonance frequencies, there is nearly a 6 dB and 10 dB increase in *TL* at node position from the antinode and close position respectively .However, *TL* drops to zero i.e. sound pressure level increases at 59 Hz and 151 Hz for close; 55 Hz and 151 Hz for antinode; 45 Hz, 151 Hz and 227 Hz for node position due to anti resonance effects before rising sharply at around the 73 and 166 Hz. These results exhibit that the resonators at node position provide a wider bandwidth of noise attenuation compared to that of antinode and close position respectively.

Therefore, it can be summarized from the results that the transmission loss of two identical 2-DOF resonators can significantly vary with change of their relative positions. The optimum relative position for two identical 2-DOF resonators achieved from this analysis is node distance apart corresponding to the wavelength of 300 Hz, maximum working frequency.

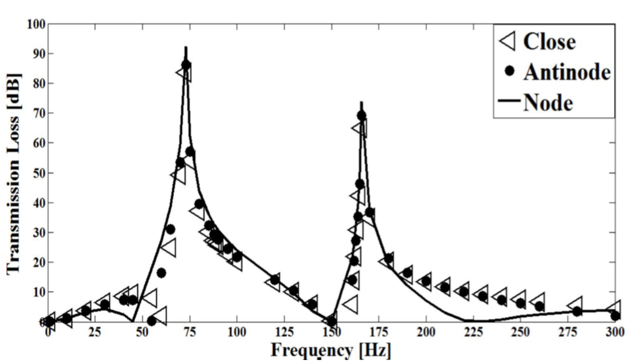

Figure 5: A comparison of numerical transmission loss for various relative positions of two identical 2-DOF resonators.

Table 1: Numerical transmission loss at two resonance frequencies for various relative positions of two identical 2-DOF resonators.

Relative Positions	Transmission Loss (*TL*), dB	
	1st Resonance Frequency (*f1*), 73 Hz	2nd Resonance Frequency (*f1*), 166 Hz
Close	83.61	63.15
Antinode	86.22	69.12
Node	92.24	73.80

3.2 Relative Orientation of Two 2-DOF Resonators

As depicted in Figure. 6 and Table 1, the numerical results of maximum *TL* are 92.24 dB, 91.63 dB and 87.33 dB at *f1,* 73 Hz compared to that of 73.15 dB, 72.16 dB and 65.65 dB at *f2,* 166 Hz for inline, 90 degree and 180 degrees respectively for a 150 dB sound source. At the first resonance frequency, there is nearly a 1 dB and 4 dB decrease in *TL* for inline orientation from the 90 and 180 degree relative orientation respectively while it drops nearly by 2 dB and 6 dB at the second resonance frequency. These results indicates that although the variation of *TL* is insignificant at 90 degree from inline orientation, it varies quite significantly in 180 degree relative orientation due to the opposing effects of the resonators.

The results clearly shows that the change of relative orientation has a significant effect on the transmission loss of two identical 2-DOF resonators particularly at the opposite direction, 180 degree relative orientation. The optimum relative orientation for two identical 2-DOF resonators achieved from this analysis is inline, no relative orientation.

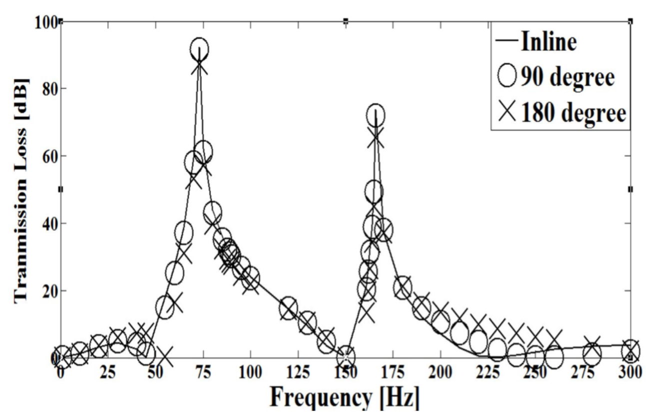

Figure 6: A comparison of numerical transmission loss for various relative orientations of two identical 2-DOF resonators.

Table 2: Numerical transmission loss at two resonance frequencies for various relative orientations of two identical 2-DOF resonators.

Relative Orientations	Transmission Loss (*TL*), dB	
	1st Resonance Frequency (*f1*), 73 Hz	2nd Resonance Frequency (*f1*), 166 Hz
Inline	92.24	73.80
90 degree	91.63	72.16
180 degree	87.33	65.65

3.3 Comparison of Various Resonator Systems

The numerical transmission loss and resonance frequencies of the optimum two identical 2-DOF resonator system (Resonator A) i.e. node position and inline orientation found from sections 3.1 and 3.2 are compared with that of a single 2-DOF [13] (Resonator B), single 1-DOF [07] (Resonator C), and optimum two identical 1-DOF resonator system [14] (Resonator D) as shown in Figure 7. Moreover, the octave band numerical transmission loss of all 4 resonator systems at their respective resonance frequencies within 300 Hz are presented in Table 4 respectively. It should be mentioned that the geometry of the duct, and all the above resonator systems, cavity and neck are quite similar considering a 150 dB sound source for all cases. This findings from the above comparison can be summarized as follows-

Table 3: A comparison of octave band numerical *TL* of two identical 2-DOF resonators with published resonator systems.

Resonator No	Transmission Loss (*TL*), dB at Octave Band (Hz)				
	16	31.5	63	125	250
A	2.46	4.35	33.65	12.11	1.73
B	0.89	3.43	16.95	4.26	2.28
C	1.75	3.57	15.06	24.53	1.71
D	2.66	4.51	11.91	11.50	3.95

The resonance frequency of both 1-DOF resonator systems make a significant drift to 89 Hz from the 1st resonance frequency of both the 2-DOF resonator systems, 73 Hz. However, the first and second resonance frequencies of both the single and two 2-DOF system remains same like the single and two 1-DOF resonators as shown in Figure 7.

The numerical result of *TL* both two identical 2-DOF and 1-DOF resonator system at their resonance frequencies are approximately double in contrast with a single 2-DOF and two identical 1-DOF resonator system respectively as shown in Table 3.

The numerical *TL* of both the 1-DOF resonator systems is nearly 5 dB higher than that of both the identical 2-DOF resonators at their respective resonance frequencies shown in Table 3.

Octave band numerical *TL* presented in Table 4 shows that the difference of *TL* for all four resonator systems are quite negligible in frequency bands, 16, 32.5 Hz and 250 Hz. However at 63 Hz, *TL* of two identical 2-DOF resonator is around 17 dB higher than a single 2-DOF resonator; 18 dB higher than two identical 1-DOF resonators and 22 dB higher than a single 1-DOF resonator system. Hence, it can be concluded that among the 4 resonator systems, the noise attenuation performance of two identical 2-DOF resonator system is best at 63 Hz.

But at 125 Hz, numerical *TL* of two identical 1-DOF resonator is around 12 dB higher than two identical 2-DOF resonators; 20 dB higher than a single 2-DOF resonator and 13 dB higher than a single 1-DOF resonator system. Hence, among the 4 resonator systems, the noise attenuation performance of two identical 1-DOF resonator system is best at 125 Hz.

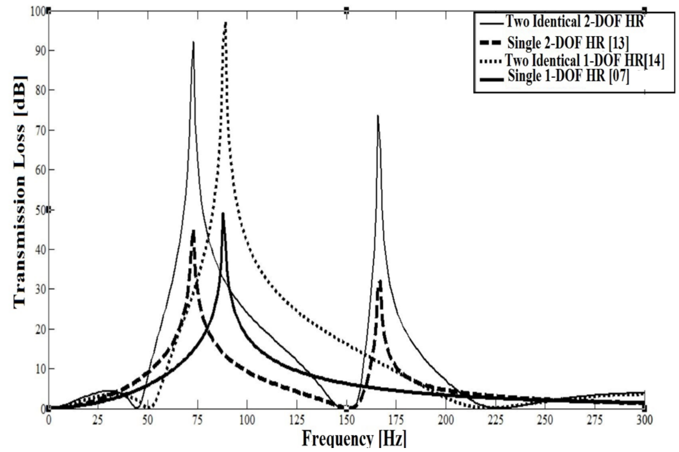

Figure 7 : A comparison of numerical transmission loss of two identical 2-DOF resonators with three published resonator systems.

4 Validation

A single 2-DOF Helmholtz resonator used by Mekid et al. [13] is used to validate the analytical design method described in section 2.1 using MATLAB. The same resonator is used in the ANSYS simulation described in section 2.2 to validate the level of transmission loss obtained analytically which is also validated with the published experimental results [13]. The analytical and numerical results of transmission loss matches well with the experimental results as shown in Figure 8.

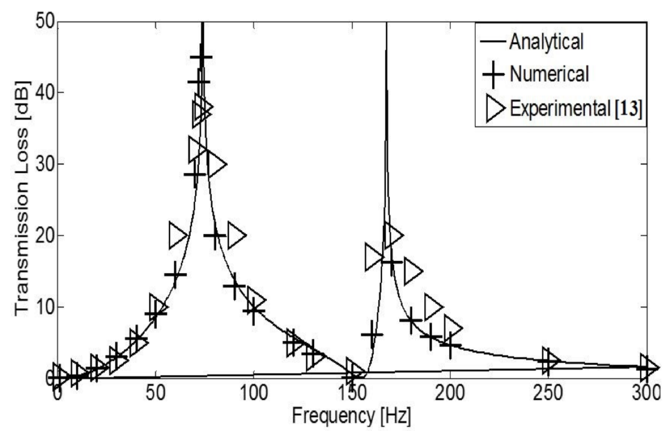

Figure 8: A comparison of transmission loss with published experimental results for a single 2-DOF resonator [13].

5 Conclusion

This work demonstrates that the relative positions and orientations have a significant effect on the low frequency noise attenuation performance of two identical 2-DOF cylindrical Helmholtz resonator system. An optimum system has been achieved for the above system in terms of relative positions and orientations: relative position: node of wavelength corresponding to the maximum working frequency, relative orientation: inline, no rotation. Moreover, the comparison between the numerical transmission losses of the 4 resonator systems shows that the noise attenuation performance of two identical 2-DOF and 1-DOF resonator systems are comparatively much better than that of both single 1-DOF and 2-DOF resonator system respectively. However, two identical 2-DOF resonator is recommended as the best fit resonator system at the low frequency range within 300 Hz since it provides a wider bandwidth of attenuation and shows better performance at most low frequencies within 300 Hz although the choice of resonator can vary depending on the target frequency of the noise source. However, it could be difficult to place the resonators at node distance apart if the length of the duct is less than the wavelength corresponding to the sound source, at particularly higher frequencies.

Future work could be further optimization of the proposed two 2-DOF resonator system by changing the geometry of the cavity and neck of the resonator similar to our published studies of two identical [14] and different frequency 1-DOF resonators [15].

Moreover, it could be of future interest to validate this work experimentally to determine the level of the transmission loss can be achieved in practice since several important factors like background noise levels, coupling effects of fluid and structure, damping effect of the material of the duct, viscous losses due to friction of the oscillation air at the neck, attenuation due to transmission and frictions in structures, increase of sound pressure level due to reflection at the duct opening are ignored in this study to avoid difficulties in the numerical simulation which may have some effect in achieving the desired result.

References

[1] K. Persson Waye, On the Effects of Environmental Low Frequency Noise, *Doctoral Thesis*, Department of Environmental Medicine, University of Gothenburg, ISBN 91-628-1516-4, 1995.

[2] Rushforth, I., Moorhouse, A. and Styles, P. "A case study of low frequency noise assessed using DIN 45680 criteria''. *noise notes*, *3*(1): 3-18, 2000.

[3] DeGagne, D. C., & Faszer, A. C. " Controlling Low Frequency Noise Using a Passive Silencer''.

[4] Helmholtz, H. von. "Theorie der Luftschwingungen in Röhren mit offenen Enden." *Journal für die reine und angewandte Mathematik,* 57: 1-72, 1860.

[5] J. W. S. Rayleigh, "The Theory of sound'', 2 : 293, Dover, New York, 1945.

[6] A. Selamet, N. S. Dicky, J. M. Novak, " Theoretical, computational and experimental investigation of Helmholtz resonators with fixed volume: lumped versus distributed analyses''. *Journal of sound and vibration,* 187(2) : 358-367, 1995.

[7] A. Selamet, P. M. Radavich, N. S. Dickey,J. M. Novak," Circular concentric Helmholtz resonators''. *The Journal of the Acoustical Society of America,* 101 (1) : 41-51, 1997.

[8] S. Griffin, S. A. Lane, S. Huybrechts, "Coupled Helmholtz resonators for acoustic attenuation''. *Journal of vibration and acoustics*, 123(1) : 11-17, 2001.

[9] Y. P. Soh, E. W. T. Yap, B. H. L. Gan, " Industrial Resonator Muffler Design''. *Level 4 Thesis, University of Adelaide*, Australia, 2001.

[10] D. Li, " Vibroacoustic behavior and noise control studies of advanced composite structures''. *PhD Thesis, University of Pittsburgh,* USA, 2003.

[11] M.B. Xu, A. Selamet and H. Kim, "Dual Helmholtz resonator". *Applied Acoustics*, 71: 822-829, 2010.

[12] M. Farooqui, "Noise reduction in centrifugal compressors using Helmholtz resonators''. *Masters-thesis*, King Fahad University of Petroleum and Minerals, Saudi Arabia, 2012.

[13] S. Mekid, M. Farooqui, "Design of Helmholtz resonators in one and two degrees of freedom for noise attenuation in pipelines''. *Acoustics Australia*, 40(3), 2012.

[14] M. A. Mahmood, M. S. Islam, M. Z. Hossain, M. M. M. Morshed, "Noise Attenuation by Two One Degree of Freedom Helmholtz Resonators''. *Proceedings in 10th Global Engineering, Science and Technology*, 202, 2015.

[15] Md. Amin Mahmud, Md. Shahriar Islam, Md. Zahid Hossain, Mir Md. Maruf Morshed " Improvement of Noise Attenuation in a Duct Using Two Helmholtz Resonators''. *International Review of Mechanical Engineering (IREME),* 9(1): 231-236, 2015.

[16] ANSYS Mechanical APDL Acoustic Analysis Guide, Release 15.0, *ANSYS Inc,* 2013.

[17] Winer, Ethan. "1". The Audio Expert. New York and London: Focal Press. ISBN 978-0-240-82100-9, 2013.

[18] Suzuki, Yoiti, and Hisashi Takeshima. "Equal-loudness-level contours for pure tones." *The Journal of the Acoustical Society of America* 116(2): 918-933, 2004.

[19] Rienstra, Sjoerd W., and Avraham Hirschberg. "An introduction to acoustics." Eindhoven University of Technology 18:19, 2013.