A Resonator Noise Reduction Solution for a Centrifugal Gas Compressor

Marius Deaconu
National Research and Development Institute for Gas Turbines COMOTI, Romania
marius.deaconu@comoti.ro

Grigore Cican
Faculty of Aerospace Engineering, Polytechnic University of Bucharest, Romania | National Research and Development Institute for Gas Turbines COMOTI, Romania
grigore.cican@upb.ro (corresponding author)

Luminita Dragasanu
National Research and Development Institute for Gas Turbines COMOTI, Romania
luminita.dragasanu@comoti.ro

Laurentiu Cristea
National Research and Development Institute for Gas Turbines COMOTI, Romania
laurnetiu.cristea@comoti.ro

Received: 2 October 2023 | Revised: 24 October 2023 | Accepted: 8 November 2023

 DOI: https://doi.org/10.48084/etasr.6464

ABSTRACT

The purpose of this work is to reduce the noise generated by a compressor that conveys methane gas. After certain measurements were conducted a high level of noise was observed in the 2000-3000 Hz range, therefore a solution for noise reduction at the source is addressed and presented in this paper. The research method is based on designing resonators to be applied on the stator of a centrifugal compressor used in a natural gas distribution station. First, the calculations are made on resonators with air as the working fluid and then are validated through real measurements in a Kundt tube. After validation, the working fluid is changed with gas, calculations are made once again, and acoustic simulations are performed. To facilitate acoustic simulations and reduce computational time, a simplified stator geometry was employed. This simplified model encompassed the region starting from the rotor's gas exit, where the resonators were deployed. The purpose of the acoustic simulation was to validate the frequency range influenced by the resonators and to estimate the overall noise reduction. Depending on the operating regime of the compressor, the rotor fundamental can vary within the frequency domain of 2000 – 3000 Hz. This broadband domain requires the usage of several resonators with different resonant frequencies. The proposed solution obtained an average value of attenuation, excluding the peaks of the attenuation, in the frequency domain of 2000 - 3000 Hz, of 9 dBA. If the fundamental frequency coincides with a resonance of the resonator, higher attenuation can occur. Also, fundamental attenuation can lead to attenuation of the harmonics.

Keywords-centrifugal gas compressor; noise; acoustic; reduction; resonator keyword

I. INTRODUCTION

Natural gas will become the second major energy source by 2030, its consumption rate will be the same as that of oil by 2040while it will surpass oil and become the largest energy consumer by 2050 [1, 2]. Methane gas is a raw material for the production of other environmentally friendly fuels such as hydrogen that will constitute the fuels of the future. Therefore in [3] a methodical optimization process was devised, employing multivariate regression models created by combining input parameters within an idealized reactor.

Natural gas compressor, as a piece of important pressurizing equipment, usually has a high rotation speed, generally between 3000 and 6000 rpm, and is an important branch of rotating machinery. In [4], high-frequency measurements were utilized to investigate the occurrence of rotating instability and the onset of spike-type stall in a high-speed compressor. Depending on the compressor speed and the impellers blade number, the Blade Pass Frequency (BPF) usually falls in the 1000 - 4500 Hz frequency domain. Such reduction can be observed in [5] where the aim was to enhance the eco-friendliness of compressors and thus duct resonator acoustic arrays were designed to reduce the acoustic energy released into the environment. The human ear is most sensitive to frequencies between 2,000 and 5,000 Hz, so with the BPF falling in this domain it is easy to understand why it is considered annoying. BPF is thought to be the most critical contributor at the general noise level recorded at centrifugal compressors and its noise components originate from the upstream and downstream of the impeller circumferential flow distortions [6]. Compressor BPF for marine diesel engine turbochargers is numerically studied and experimentally investigated in [7].

Centrifugal compressors use the mechanical energy to compress the working fluid and a small fraction of it is converted into acoustic energy, which consequently propagates in the whole system as noise [8]. The excessive noise of the industrial compressors used in the gas transport systems is one of the problems that both industrial areas and its populated neighborhoods are confronted with. Equipment noise reduction represents a challenge because that noise has a big impact on the staff serving these turbo-machineries and the nearby urban areas, as well. Different solutions have been proposed to eliminate centrifugal compressors noise among which one can mention the use of passive techniques (porous absorbing panels, resonators cavities) or active techniques (active noise cancelation). In [9], a methodology for experimental validation of a CFD model for predicting noise generation in centrifugal compressors is presented. In [10], a 3-dimensional CFD model of a centrifugal compressor is analyzed and in [11] fluid phenomena related to whoosh noise are studied. In [12], an experimental investigation on the metal foam for controlling centrifugal fan noise is presented. In [13], the aerodynamic noise of a turbocharger compressor mounted on a passenger car engine is examined by using turbulence numerical simulation, aero-acoustic simulation, and acoustic Boundary Element Method (BEM).

Vibration and noise characteristic of two-stage centrifugal compressors used in a refrigeration system are experimentally studied in [14] where a multi-layer micro-perforated panel absorber fixed at the outlet pipe is employed to attenuate the compressor noise. The experimental results confirm that the designed absorber can effectively reduce the noise of the centrifugal refrigeration compressor.

Most of the published studies on noise reduction of centrifugal compressors using resonator arrays were found with resonators applied at silencers, as in [15]. There testing confirmed that the duct resonator array reduces narrowband noise by at least 14 dB within the specified bandwidth (2-3 kHz) and can lower broadband noise by up to 8 dB (1-2 kHz), at pipe level [16]. A similar observation was made in [17] where the acoustic performance of the Helmholtz array ranged from 9 to 13.2 dB for the vaned diffuser.

Regarding noise reduction at centrifugal compressors, authors in [18] proposed that the resonators should be applied on the compressor stator, in the region where the air comes out from the rotor. In this respect, a series of resonators were applied on the stator and therefore an array of acoustic resonators that attenuate the acoustic energy generated by the impeller was formed. Starting from the idea of applying resonators on the stator, the present case study is based on their application on the stator of a centrifugal compressor, which is a part of a natural gas distribution station. Given the growth of the urban area that is in the vicinity of the distribution station, the noise levels recorded in the station perimeter often exceed the legislation limits. The solution is based on designing resonators to be applied on the stator of the centrifugal compressor used in a natural gas distribution station.

II. MATERIALS AND METHODS

The methodology applied in this study relies on four well defined steps. In the first step, as a starting point of the entire research, the problem is defined by establishing the frequency of interest. Solution designing as a second step is followed by design validation with air as the working fluid and the final step consists in applying the solution for the gas case. Three measurements were performed at different regimes (chosen based on the recorded statistics), with the compressor having a number of blades nb = 13, compressor shaft speed N between 9000 and 14000 rpm (150-233 Hz), obtaining a BPF between 2000 and 3000 Hz (N*), which is henceforth considered the frequency domain of interest, as shown in Figure 1.

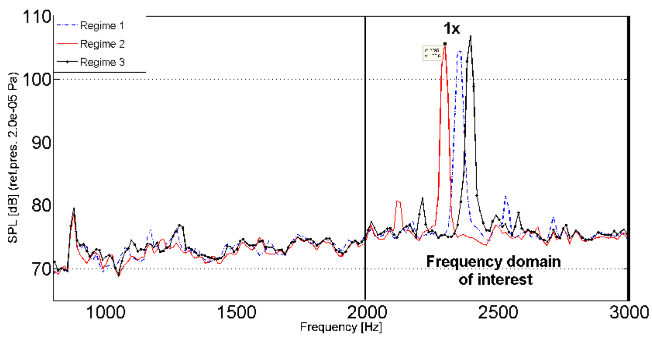

Fig. 1. BPF variation at different regimes.

Step two consists in defining the analytic model for calculating the resonators. It is generally known that the resonator is an acoustic system encompassing a rigid-walled cavity filled with air and a neck through which the cavity communicates. When pressure is applied, the air filling at the neck starts to move back and forth, damping out in time. There are many studies where Helmholtz resonators are analyzed. In [19], a promising design of sound absorption panels containing acoustic resonators is proposed. In [20], focus is given on an optimized method for Helmholtz Resonators (HRs) designed to be used in room acoustics. This method utilizes a transfer matrix and the electro-acoustic analogy whereas the calculations optimization was conducted by integrating the measured data to enhance the consistency between the calculated sound absorption coefficient and the measurements taken in the reverberation room. In [21], an HR with a spiral neck is studied theoretically and numerically. Also, ways to achieve significant sound reduction in a compact solution at lower frequencies while exhibiting multiple resonance at higher frequencies are studied. For an individual resonator, it is known that the acoustic absorption is selective, meaning that it works only in a specific frequency. It can be calculated by:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{A}{V\cdot(+0.8d)}} \quad (1)$$

where c represents the sound speed, d is the resonator's neck diameter, while V is the resonator's volume.

On the other hand, to obtain a better performance in terms of larger frequency coverage, resonators with perforated plate can be used. These types of resonant structures are made of coupled resonators acquired by placing a perforated plate at a certain distance from a rigid wall. The impedance of the perforated panel Z_{pp} is computed using the Maa model [22]:

$$Z_{pp} = \frac{32\,\mu\,t}{\sigma\,d^2}\left(\sqrt{1+\frac{s^2}{32}}+\frac{\sqrt{2}sd}{32t}\right)$$

$$+\frac{j\omega t\rho_0}{\sigma}\left(1+\frac{1}{\sqrt{3^2+\frac{s}{2}}}+0.85\frac{d}{t}\right) \quad (2)$$

where μ is air viscosity [Pa/s], σ represents the porosity of the perforated panel, s is the perforation constant, equal to $10d\sqrt{f}$, ω is the angular frequency [rad/s], ρ_0 is the air density inside the perforations [kg/m^3], d is the diameter of the perforations [m], t is the distance between the centers of two perforations [m], and j represents the imaginary part of the impedance.

The total impedance of the resonator is the sum of the perforated panel and the back cavity impedances:

$$Z_{back_{air}} = -i\,\rho_0 c_0 \cot(kD)\,s \quad (3)$$

$$Z_{total} = Z_{M(MPP)} + Z_{back\_air} \quad (4)$$

where k is the wave number and D is the cavity depth. From the total system impedance, the acoustic reflection coefficient at normal incidence is determined, from which the acoustic absorption coefficient is:

$$R = \frac{(Z_{total}-\rho_0 c_0)}{(Z_{total}+\rho_0 c_0)} \quad (5)$$

$$\alpha = 1-|R|^2 \quad (6)$$

Figure 2 presents a comparison in frequency-amplitude response between an individual resonator and the perforated plate. There the main difference consists in a broadband response of the perforated plate but with smaller absorption capacity due to the perforations interaction. One of the problems when designing a resonator is given by the impurities and dirt which can be found in our specific case in gas. Designing a resonator with small neck diameter could lead to clogging of the perforation reducing the acoustic treatment effectiveness. So, a compromise between the available space and the above resonator design solutions had to be made.

In order to cover the entire domain, four frequencies were selected as references at the resonators design. These resonators were applied on the stator of the compressor, as presented in Figure 3.

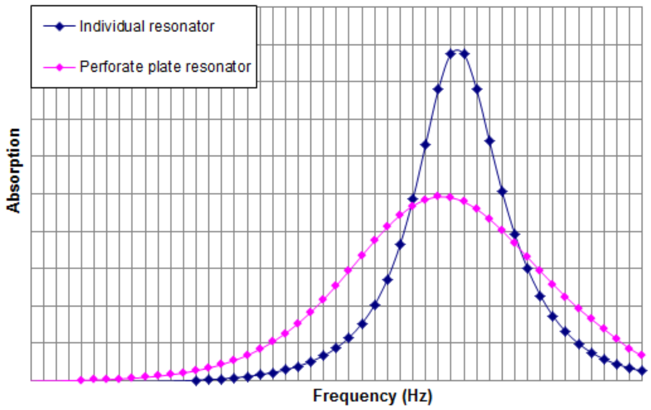

Fig. 2. Working frequency domain for τηε two types of resonators.

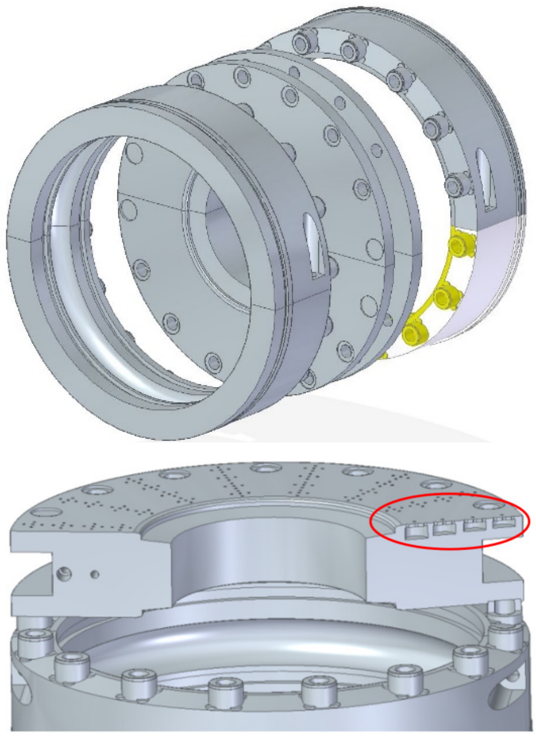

Fig. 3. Stator segment (top – no resonator applied, bottom – highlithed resonators on the stator).

Step three consisted in validating the selected solutions by means of measurements. For this step, the resonators frequencies were modeled for air (c = 340 m/s) and were manufactured from a thermoplastic aliphatic polyester (PLA, Poly) at the 3D printer, as in Figure 4. The resonators were measured in impedance tube, Figure 5, using ISO 10534-2 method that is based on transfer function between the two microphones. Moreover, the reflection coefficient, from which the acoustic absorption coefficient is resulted, is calculated [22].

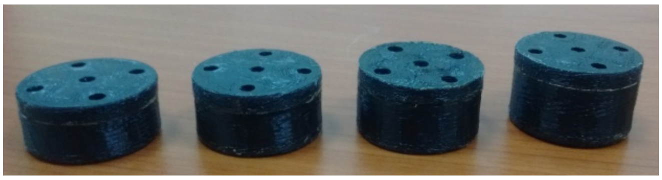

Fig. 4. Resonators designed for 2000 Hz – 2600 Hz.

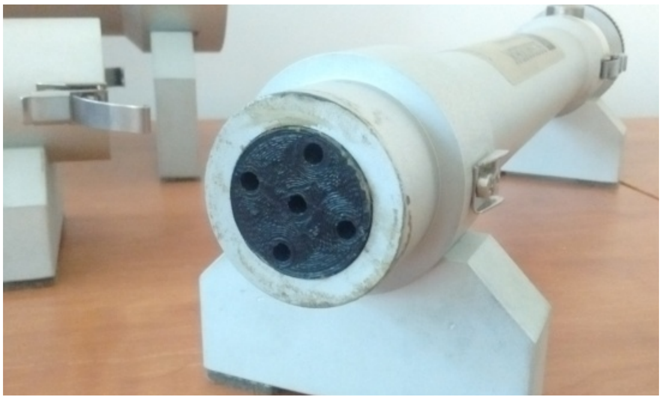

Fig. 5. Resonator tests in impedance tube.

In Figure 6, one can easily see that the designed resonators reduce the noise at the intended frequency, therefore the design principles were appropriately chosen, even though small deviations (max. deviation obtained: 18Hz) due to the manufacturing process were spotted.

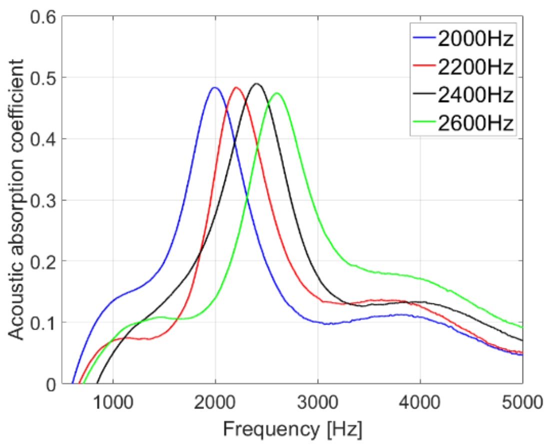

Fig. 6. Acoustic absorption coefficient for the four resonators (case: air).

Step four was to model and optimize a series of resonators for gas compressor for a speed of sound in methane of c = 462 m/s ($\lambda_{methane} > \lambda_{air}$). In order to evaluate the noise reduction of resonators for gas compressor, a finite element method was used. For acoustic simulations, a simplified geometry of the stator was utilized so as to reduce the time calculation. A simplified model was made for a region chosen from the exit of the gas from the rotor (see Figure 3), where the resonators were applied. The objective of the acoustic simulation was to verify the frequency domain on which the resonators act and to estimate an overall noise reduction.

It is known that similar resonators must not be placed very close to each other because they interfere, and their efficiency is lowered. When having two similar resonators close to one another the frequency for witch they were designed is shifted and as consequence the system does not work on the intended frequency domain. Even a relation between resonators holes and high TL (Transmission Loss) for a given frequency domain was defined in [23]. In this case, it cannot be applied due to the small distances available in the stator region. Given the specific geometry of the compressor stator, three solutions were proposed for resonator placing (in line of slightly shifted between them), see Figure 7. The mesh of the selected domains is presented in Figure 8.

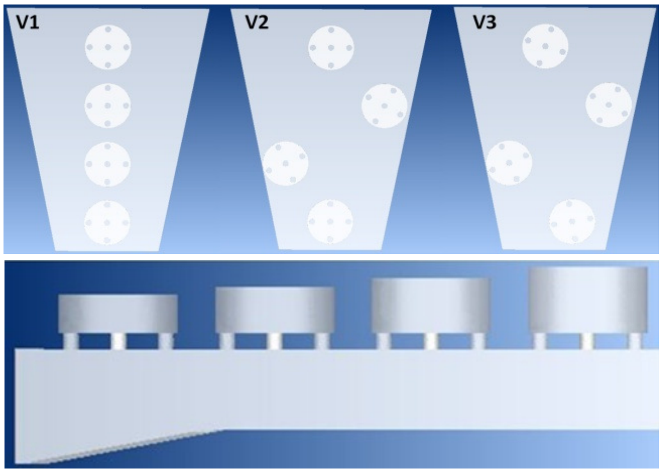

Fig. 7. CAD simplified model (top: front view, bottom: side view).

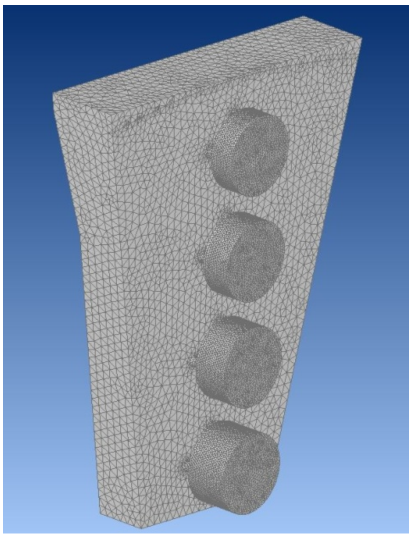

Fig. 8. Mesh of the simplified stator CAD model.

Numerical simulations were performed by ACTRAN software, which uses the Finite Element Method. For this problem the acoustic module was used and the wave propagation was employed. The mesh was modeled by a rule of 8 tetrahedral elements per the wavelength of the highest studied frequency. Considering a speed of sound of 462 m/s and a maximum frequency of 2800 Hz, the wavelength is 0.165 m, which leads to a maximum element dimension of 0.02 m. Special attention was given to the calculation grid in the area of the resonator perforations (the neck of the resonator), in order to be able to capture the visco-thermal phenomena that occur when the resonance is reached. Thus, elements with dimensions up to 0.3 mm were utilized in the resonators neck.

The applied boundary conditions in FEM software were: the bottom surface of the mesh was considered as the plane wave acoustic excitation, the properties of methane were applied to the fluid, and the surface above was considered a free field surface with no reflection back into the computational grid. Therefore, the acoustic TL was computed knowing the acoustic power generated and the acoustic power transmitted out of the model. All the other surfaces were considered as hard walls with perfect reflection.

III. RESULTS AND DISCUSSION

The acoustic simulations results highlighted once more that the position of the resonators influences both the acting frequency of the resonators and their effectiveness (Figure 9). The objective of this work was to achieve the highest attenuation in the frequency domain of 2000-3000Hz. Consequently the first version (V1) with the resonators in straight line, obtained the highest attenuation in this domain.

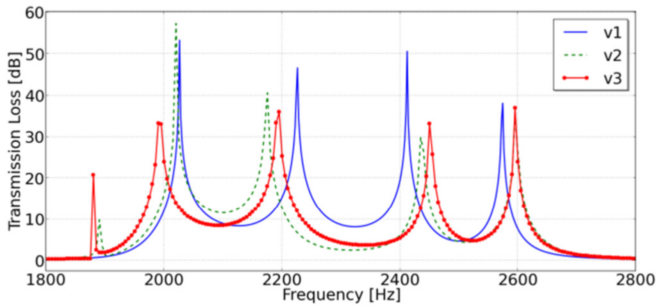

Fig. 9. Transmission loss obtained for the simplified model.

For a better understanding of the shift in the resonant frequencies of the resonators and the difference of attenuation level for each resonator, an acoustic pressure plot inside the fluid domain of the V1 is presented in Figure 10.

At the frequency of 2000 Hz, the acoustic pressure inside the first resonator is in the opposite phase with the generated wave and the resonator competence is high. Also, in the resonators' neck, the acoustic velocity of the air is high and this can lead to interferences between the resonators. Only the second resonator is situated in pre-resonance domain leading to a small influence on the effectiveness of the first resonator. At the other frequencies, the resonators with lower frequency produce a phase modification that influences both attenuation and resonant frequency.

IV. CONCLUSIONS

Depending on the operating regime of the compressor, the rotor fundamental frequency can vary within the frequency domain of 2000 – 3000 Hz. This broad band domain requires the usage of several resonators with different resonant frequencies. The proposed solution obtained an average value of attenuation of 9 dB, excluding the peaks of the attenuation, in the frequency domain of 2000 - 3000 Hz. If the fundamental coincides with a resonance frequency of a resonator, higher attenuation can occur, but also fundamental attenuation can lead to attenuation of the harmonics. Different geometrical arrangements of resonators were performed, and the results obtained by numerical simulation and experimental evaluation provide a way for topological optimization.

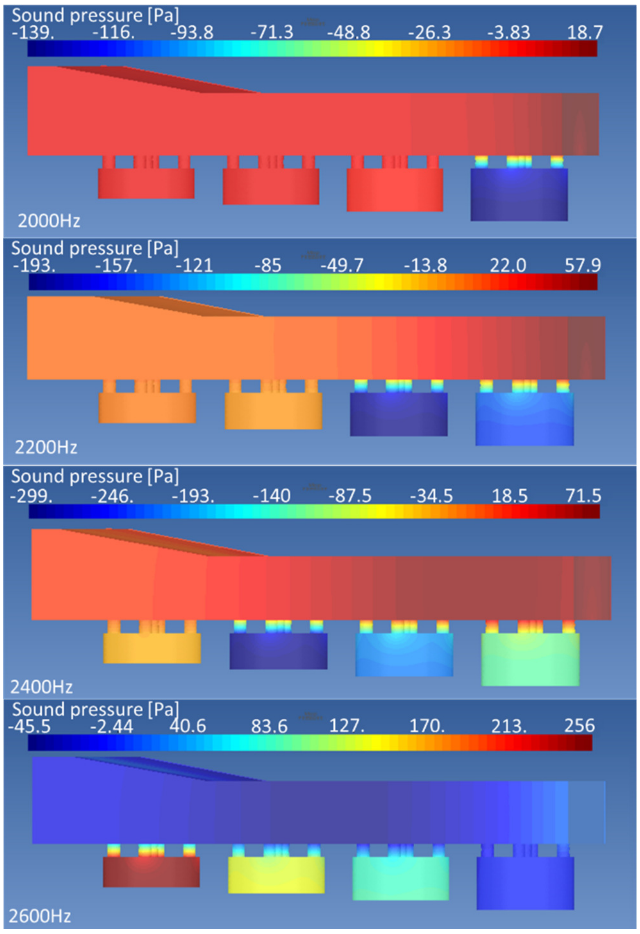

Fig. 10. Sound pressure field inside the V1 model.

The acquired results are encouraging for the continuation of the research, even if from a technological point of view, the proposed solution is not so easy to be applied at the moment.

ACKNOWLEDGMENT

The data presented and analyzed in this report were obtained with the help of COMOTI's Research and Experiments Center in the field of Acoustic and Vibrations staff and facility and the work was carried out through the Core Program within the National Research, Development, and Innovation Plan 2022-2027, implemented with the support of MCID, project no. PN23.12.01.02.

REFERENCES

[1] Z. R. Chong, S. H. B. Yang, P. Babu, P. Linga, and X.-S. Li, "Review of natural gas hydrates as an energy resource: Prospects and challenges," *Applied Energy*, vol. 162, pp. 1633–1652, Jan. 2016, https://doi.org/10.1016/j.apenergy.2014.12.061.

[2] "2015-2020 China Natural Gas Industry Forecast and Investment Strategy Research Report," China Commercial Industrial Research Institute, Report CI 389603, Nov. 2014.

[3] E. C. Romao, A. F. Siqueira, and J. A. Martins, "Numerical Simulation and Optimization of Methane Steam Reforming to Maximize H2 Production: A Case Study," *Engineering, Technology & Applied Science Research*, vol. 13, no. 2, pp. 10255–10260, Apr. 2023, https://doi.org/10.48084/etasr.5632.

[4] J. Schreiber, B. Paoletti, and X. Ottavy, "Observations on Rotating Instabilities and Spike Type Stall Inception in a High-Speed Multistage Compressor," *International Journal of Rotating Machinery*, vol. 2017, Aug. 2017, Art. no. e7035870, https://doi.org/10.1155/2017/7035870.

[5] Z. Liu and M. J. Kuzdzal, "Noise Control of an 11,000 Horsepower Single Stage Pipeline Centrifugal Compressor," in *ASME Turbo Expo 2007: Power for Land, Sea, and Air*, Mar. 2009, pp. 1489–1496, https://doi.org/10.1115/GT2007-27422.

[6] J. T. Kim and I. Iman, "Compressor noise attenuation using branch type resonator," *The Journal of the Acoustical Society of America*, vol. 90, no. 3, Sep. 1991, Art. no. 1708, https://doi.org/10.1121/1.401709.

[7] C. Liu, Y. Cao, W. Zhang, P. Ming, and Y. Liu, "Numerical and experimental investigations of centrifugal compressor BPF noise," *Applied Acoustics*, vol. 150, pp. 290–301, Jul. 2019, https://doi.org/10.1016/j.apacoust.2019.02.017.

[8] S. Ionescu, A. Toma, R. Codoban, N. Enescu, and I. Magheti, "Acoustic Measurments and Scenarios to Reduce Noise at a Natural Gas Compression Station," in *20th International Danube-Adria-Association-for-Automation-and-Manufacturing Symposium*, Vienna, Austria, Nov. 2009.

[9] A. Broatch, J. Galindo, R. Navarro, and J. García-Tíscar, "Methodology for experimental validation of a CFD model for predicting noise generation in centrifugal compressors," *International Journal of Heat and Fluid Flow*, vol. 50, pp. 134–144, Dec. 2014, https://doi.org/10.1016/j.ijheatfluidflow.2014.06.006.

[10] V. Dragan, I. Malael, and B. Gherman, "A Comparative Analysis Between Optimized and Baseline High Pressure Compressor Stages Using Tridimensional Computational Fluid Dynamics," *Engineering, Technology & Applied Science Research*, vol. 6, no. 4, pp. 1103–1108, Aug. 2016, https://doi.org/10.48084/etasr.696.

[11] A. Lipej, "Challenges in the Numerical Analysis of Centrifugal Pumps: Energetic, Cavitation and Dynamic Characteristics," *Engineering, Technology & Applied Science Research*, vol. 12, no. 1, pp. 8217–8222, Feb. 2022, https://doi.org/10.48084/etasr.4647.

[12] C. Xu and Y. Mao, "Experimental investigation of metal foam for controlling centrifugal fan noise," *Applied Acoustics*, vol. 104, pp. 182–192, Mar. 2016, https://doi.org/10.1016/j.apacoust.2015.11.014.

[13] C. Zhong, L. Hu, J. Gong, C. Wu, S. Wang, and X. Zhu, "Effects analysis on aerodynamic noise reduction of centrifugal compressor used for gasoline engine," *Applied Acoustics*, vol. 180, Sep. 2021, Art. no. 108104, https://doi.org/10.1016/j.apacoust.2021.108104.

[14] Y. Mao, C. Fan, Z. Zhang, S. Song, and C. Xu, "Control of noise generated from centrifugal refrigeration compressor," *Mechanical Systems and Signal Processing*, vol. 152, May 2021, Art. no. 107466, https://doi.org/10.1016/j.ymssp.2020.107466.

[15] M. J. Newman *et al.*, "Verification of a Duct Resonator Array for Larger Pipe Diameters," in *Inter-noise 2014*, Melbourne, Australia, Nov. 2014.

[16] Z. Liu and M. J. Kuzdzal, "Noise reduction of a multistage export / reinjection - Dresser-Rand," 2002, [Online]. Available: https://www.yumpu.com/en/document/view/10372341/noise-reduction-of-a-multistage-export-reinjection-dresser-rand.

[17] Zheji Liu and D. Lee Hill, "Centrifugal Compressor Noise Reduction By Using Helmholtz Resonator Arrays.," *Proceedings of the 30th Turbomachinery Symposium*, pp. 89–94, 2001, https://doi.org/10.21423/R1TQ0Z.

[18] "European Patent Application," EP 1 340 920 A1, Sep. 2003.

[19] J.-S. Chen, Y.-B. Chen, Y.-H. Cheng, and L.-C. Chou, "A sound absorption panel containing coiled Helmholtz resonators," *Physics Letters A*, vol. 384, no. 35, Dec. 2020, Art. no. 126887, https://doi.org/10.1016/j.physleta.2020.126887.

[20] D. Jun, O. Nespesny, J. Pencik, Z. Fisarova, and A. Rubina, "Optimized method for Helmholtz resonator design formed by perforated boards," *Applied Acoustics*, vol. 184, Dec. 2021, Art. no. 108341, https://doi.org/10.1016/j.apacoust.2021.108341.

[21] X. Shi and C. Ming Mak, "Helmholtz resonator with a spiral neck," *Applied Acoustics*, vol. 99, pp. 68–71, Dec. 2015, https://doi.org/10.1016/j.apacoust.2015.05.012.

[22] M. Deacon, G. Cican, L. Cristea, and L. Dragasanu, "Analysis of High Porosity Micro Perforated Panel Using Different Methods," *UPB Scientific Bulletin, Series D*, vol. 80, no. 3, pp. 141–152, 2018.

[23] S.-H. Seo and Y.-H. Kim, "Silencer design by using array resonators for low-frequency band noise reduction," *The Journal of the Acoustical Society of America*, vol. 118, no. 4, pp. 2332–2338, Oct. 2005, https://doi.org/10.1121/1.2036222.