Experimental Study of Noise Reduction in Swirl Diffusers Using Optimized Blade Geometry

Joanna Maria Kopania, Grzegorz Bogusławski
Lodz University of Technology, Żeromskiego Street 119, Lodz, Poland

This paper is a revised version of the authors' submitted manuscript that was peer-reviewed by at least two anonymous reviewers.

Abstract

This study investigates the aeroacoustic performance of ceiling swirl diffusers equipped with adjustable blades, with a particular focus on the influence of blade geometry and flow configuration on noise generation. Both commercial blade designs and newly developed prototype profiles manufactured using 3D printing were analyzed. Measurements were conducted in a reverberation chamber in accordance with ISO 5135 and ISO 3741 standards for three airflow rates (600, 1000, and 1400 m^3/h) and three blade positions corresponding to different flow directions. Sound power levels were determined in 1/3-octave bands to assess the acoustic performance of each configuration. The results demonstrate that blade geometry and flow direction have a significant impact on noise emission. Profiles with complex geometries, particularly profiles 1 and 3, were associated with higher sound power levels, while profiles 2 and 4 showed moderate and stable behavior. The prototype profile 5 achieved the lowest noise levels and, in certain configurations, reduced sound power below the baseline diffuser. The serrated profile 6 provided consistently low acoustic penalties, especially under specific flow conditions. The study confirms that optimized blade design and appropriate flow configuration can significantly improve the acoustic performance of swirl diffusers in ventilation systems.

Keywords: *swirl diffuser, noise, 3D printing, profile*

1. Introduction

Ceiling swirl diffusers with adjustable blades are widely used in HVAC systems because they provide effective air distribution while maintaining indoor thermal comfort. Their acoustic performance is strongly influenced by blade geometry, airflow direction, and turbulence generated within the diffuser. In swirl diffusers, the interaction between airflow and blade surfaces produces both broadband aerodynamic noise, associated with turbulence and vortex shedding, and tonal components related to flow instabilities.

Experimental acoustic measurements play an important role in validating numerical aeroacoustic models and in understanding the influence of diffuser geometry on noise generation [1–9].

Recent studies have shown that geometric features such as guide elements, mounting beams, and plenum-box design significantly affect the aeroacoustic behaviour of swirl diffusers. Numerical investigations combining large-eddy simulations with acoustic analogies demonstrated that local flow disturbances generated by these components are responsible for increased broadband noise at higher frequencies [10]. Similar conclusions regarding the influence of diffuser geometry on acoustic performance were reported in subsequent experimental studies [11].

The aim of this study was to investigate the aeroacoustic performance of ceiling swirl diffusers equipped with adjustable blades and to compare commercial blade designs with newly developed prototype geometries manufactured using 3D printing. The influence of blade geometry, blade position, and airflow rate on sound power levels was evaluated to identify blade configurations that minimize aerodynamic noise and provide practical guidelines for the acoustic optimization of swirl diffusers.

2. Experimental Study

Experimental acoustical studies were made during the airflow generated by the swirl diffuser front with adjustable blades in the reverberation room, The Institute of Power Engineering - National Research Institute - Figure 1. The reverberation room has got a volume of 237.0 m^3, an area of 231.5 m^2, and non-parallel walls. The measurements were made on a test stand built in accordance with the norm ISO 5135 [12]. The total length of the test duct system was approximately 10 m. A centrifugal fan, located outside the reverberation chamber, supplied the airflow to the measuring system. The fan rotational speed, and consequently the airflow rate, was controlled using a

Corresponding author: joanna.kopania@p.lodz.pl.

frequency inverter, allowing precise adjustment of the operating conditions. To minimize the influence of fan-generated noise on the measurements, the air supply system incorporated three absorptive silencers installed downstream of the fan. Two silencers had a length of 1.5 m, while the third silencer was 2.0 m long. The silencers, together with the straight duct sections, ensured stable airflow conditions and effectively reduced regenerated flow noise before the air reached the tested diffuser. Inside the reverberation chamber, a 1.5 m straight duct section equipped with pressure measurement ports was connected to the air supply system. A plenum box was mounted at the end of this duct, followed by the circular swirl diffuser panel fitted with interchangeable blade profiles. The plenum box was positioned 1.0 m from the chamber walls to minimize the influence of sound reflections and to satisfy the free-field installation requirements specified for sound power measurements. This configuration provided repeatable aerodynamic conditions and enabled reliable comparison of the acoustic performance of the different blade geometries.

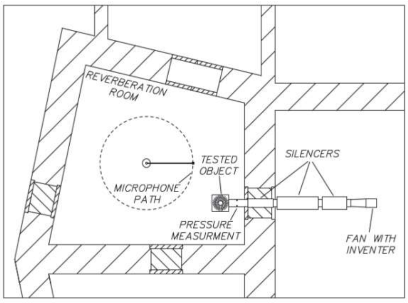


Figure 1. Scheme of reverberation room with studied object.

This study investigated five different commercial blade (vane) profiles installed in a ceiling swirl diffuser, each modifying the airflow and its acoustic behavior. The profiles differed in geometry, including variations in curvature, thickness, and edge shape, ranging from smooth aerodynamic forms to more complex serrated or contoured designs (Figure 2).

Profile	Cross-sectional geometry	Blade length (without mounting) [mm]	Maximum blade width [mm]	Maximum thickness [mm]	Number of guide ribs	Guide rib radius [mm]	Reinforcing rib	Trailing/Leading edge design
Profile 1	Standard commercial profile	100	30.0	3.0	3	20	Yes, height = 3.5 mm	Upper surface chamfer, 3 mm
Profile 2	Modified commercial profile	100	30.0	3.0	2	13	None	Lower surface chamfer, 3 mm
Profile 3	Simplified profile	100	30.0	3.0	None	–	Yes, height = 3.0 mm	Upper surface chamfer, 3 mm
Profile 4	Reinforced profile	100	30.0	3.0	2	13	Yes, maximum height = 10 mm, radius = 187.3 mm	Upper surface chamfer, 3 mm
Profile 5 (Prandtl-D)	Biomimetic Prandtl-D profile	100	30.5	4.0	None	–	None	Smooth trailing edge
Profile 6 (Prandtl-D with serrated trailing edge)	Biomimetic serrated profile	100	30.5	4.0	None	–	None	Serrated trailing edge; serration height = 5 mm, pitch = 5 mm

Figure 2. Geometrical characteristics of the investigated swirl diffuser blade profiles. All blade profiles have a total length of 110 mm, including the mounting section. The dimensions presented below refer only to the blade itself (without the mounting element).

Some profiles were shaped to guide the airflow more gently, while others introduced stronger disturbances to enhance mixing. Special attention was given to the leading edge (LE) and trailing edge (TE) of the blades, as these features significantly influence turbulence and noise generation (Figure 3).

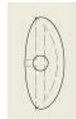
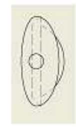
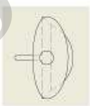
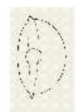
Profile	Cross-sectional geometry	3D model of profile
Profile 1 (commercial blade with three guide ribs)		
Profile 2 (commercial blade with two guide rib)		
Profile 3 (single reinforcing rib profile)		
Profile 4 (streamlined reinforced profile)		
Profile 5 (Prandtl-D blade)		
Profile 6 (serrated Prandtl-D blade)		

Figure 3. Diffuser profiles used in the research.

The prototype blades (profile 5 and 6) were manufactured using 3D printing technology, allowing for precise control over their geometry and enabling rapid testing of different configurations.

All blade profiles were mounted in a standard circular ceiling swirl diffuser panel with 48 holes for profile blades (Figure 4), which directs supply air into the room and allows for the adjustment of blade positions (straight, left-flow, and right-flow).

The diffuser panel served as the base structure, ensuring consistent boundary conditions for comparing the acoustic performance of each profile. This setup made it possible to evaluate how both blade geometry and airflow direction affect the overall noise emission of the diffuser.

Measurements were conducted for three volumetric flow rates: 600, 1000, and 1400 m³/h. The flow rate was

adjusted by varying the rotation frequency of the fan motor using a connected three-phase inverter. Flow velocity was verified using a Testo 420 balometer. The three-position of blades were studied, as shown in Figure 5.
Air discharge directions can be modified by adjusting the blade angle; for this study, the swirl blades were set to 45° (left-flow and right-flow) to create an external swirl. For profile 6 the leading edge of the air flow from inside the box and the trailing edge are marked.
The generated noise was determined by the sound power level, measured and calculated in accordance with the PN-EN ISO 3741 standard [13]. A Norsonic Nor140 measurement set with Nor850 software and a Nor265 rotary table was employed. Sound pressure was measured at twelve uniformly spaced points along a circle with a radius of 1.7 m. Data were recorded in 1/3-octave bands within the frequency range of 100 Hz to 10,000 Hz. The measurement duration was set to 15 seconds for the test samples and 30 seconds for the background noise.

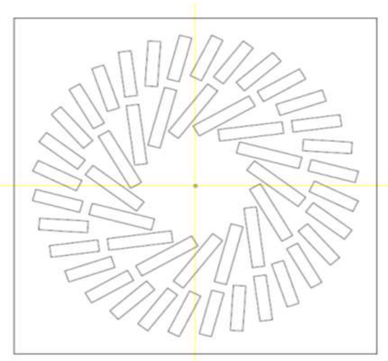


Figure 4. Diffuser panel with 48 holes for profile blades

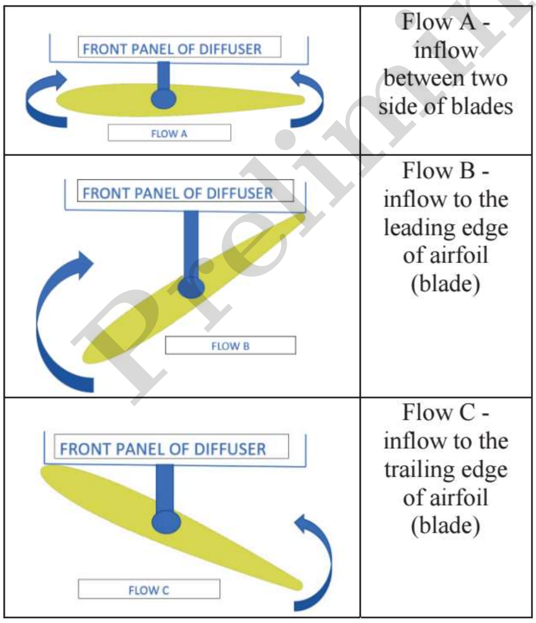


Figure 5. Scheme of inflow between studied blades and front panel of the diffuser.

Reverberation time was determined using four omnidirectional loudspeaker positions combined with three microphone settings spaced every 120°. All sound power level calculations were performed using a pre-configured calculation sheet. Before and after each measurement series, the background noise level was recorded and the system was calibrated using a Brüel & Kjær 4231 calibrator. Following each measurement, the temperature, relative humidity, and atmospheric pressure were recorded to provide the necessary data for sound power level corrections.

3. Results and discussion

The acoustic performance of the tested blade profiles varies depending on both the airflow rate and the type of flow configuration (A, B, or C). At 600 m³/h, Profile 5 exhibits the lowest sound power level in flow B (43.8 dB), making it the quietest option under these conditions, while Profile 3 and Profile 1 are among the loudest, especially in flow A (above 52 dB). For flow C at the same airflow, Profiles 5 and 6 show relatively low noise levels (around 46 dB), whereas Profile 1 remains significantly louder.
At 1000 m³/h, Profile 5 again performs best acoustically, particularly in flow B (58.7 dB), while Profiles 1 and 3 generate the highest noise levels across most flow types, exceeding 62–65 dB. Profiles 2 and 4 show moderate acoustic performance, with relatively stable values across all flow directions.
At the highest airflow rate of 1400 m³/h, Profile 5 remains the quietest in flow B (68.1 dB) and also performs well in flow C, while Profile 6 shows competitive results in flow C (69.8 dB). In contrast, Profiles 1 and 3 consistently produce the highest sound power levels, particularly in flow A, where values exceed 74 dB.
Overall, flow B generally results in the lowest noise levels for most profiles, while flow A tends to be the noisiest configuration. Among all tested geometries, Profile 5 demonstrates the best acoustic performance across different airflow rates, whereas Profiles 1 and 3 are the least favorable in terms of noise emission.

Table 1. Sound Power Levels (LwA) of Swirl Diffuser Blade Profiles for Different Airflow Rates and Flow Configurations.

600 m3/h			
position of blade	flow A	flow C	flow B
baseline	45.2	45.2	45.2
profile 1	52.7	49.8	47.7
profile 2	48.9	47.0	46.7
profile 3	53.0	48.3	49.6
profile 4	49.8	47.2	47.0
profile 5	51.5	46.1	43.8
profile 6	48.1	46.0	47.8
1000 m3/h			
position of blade	flow A	flow C	flow B

baseline	59.4	59.4	59.4
profile 1	65.5	62.7	62.5
profile 2	62.5	62.3	61.7
profile 3	65.9	62.7	64.7
profile 4	63.8	62.1	62.0
profile 5	64.6	61.1	58.7
profile 6	63.2	60.5	62.1
1400 m3/h			
position of blade	flow a	flow C	flow B
baseline	69.0	69.4	69.4
profile 1	74.7	72.3	71.0
profile 2	71.9	71.5	71.2
profile 3	75.1	71.8	72.7
profile 4	72.4	71.3	71.2
profile 5	73.1	69.9	68.1
profile 6	72.3	69.8	72.1

The tested blade profiles were evaluated in relation to a reference (baseline) diffuser configuration without profiles - empty holes. The comparison was based on differences in sound power level, expressed as the change between the diffuser equipped with profiles and the baseline diffuser – as in Equation (1).

$$\Delta L_{wA} = L_{wA\,profiles} - L_{wA\,baseline} \qquad (1)$$

where:
LwA baseline - the level of the A-sound power in the frequency band considered for the baseline panel, without profile;
LwA profiles - the level of the A-sound power in the frequency band considered for the studied variant of profiles.
This allowed the researchers to determine whether the introduced blade geometries increased or reduced noise emission. The reference case ensured consistent conditions for assessing the acoustic impact of each profile.

Table 2. Difference in Sound Power Level (ΔLwA) Relative to Baseline Diffuser.

600 m3/h			
position of blade	flow A	flow C	flow B
profile 1	7.5	4.6	2.5
profile 2	3.7	1.8	1.5
profile 3	7.8	3.1	4.4
profile 4	4.6	2	1.8
profile 5	6.3	0.9	-1.4
profile 6	2.9	0.8	2.6
1000 m3/h			
profile 1	6.1	3.3	3.1
profile 2	3.1	2.9	2.3
profile 3	6.5	3.3	5.3
profile 4	4.4	2.7	2.6
profile 5	5.2	1.7	-0.7
profile 6	3.8	1.1	2.7
1400 m3/h			
profile 1	5.3	2.9	1.6
profile 2	2.5	2.1	1.8
profile 3	5.7	2.4	3.3
profile 4	3	1.9	1.8
profile 5	3.7	0.5	-1.3
profile 6	2.9	0.4	2.7

These results indicate that the absolute sound power level depends on both blade geometry and airflow rate, whereas Table 3 isolates only the influence of blade geometry by eliminating the contribution of the diffuser itself. Profiles 1 and 3 consistently exhibit the highest positive differences across all flow rates and configurations, indicating that these geometries significantly increase noise, particularly in Flow A. In contrast, Profiles 2 and 4 show moderate and relatively stable increases in sound power levels, with smaller deviations across different flow configurations.
Profile 5 demonstrates a clear dependence on flow direction; while it shows small positive differences in Flows A and C, it yields negative values in Flow B across all airflow rates. This indicates that Profile 5 can actually reduce noise levels below the baseline under this specific configuration. Profile 6 (the serrated blade profile) generally results in low noise increases, particularly in Flow C, where the differences are among the smallest recorded (as low as 0.4–0.8 dB at higher flow rates).
A key observation is that Flow A tends to produce the highest noise increase for all profiles, whereas Flows B and C are acoustically more favorable. For Profile 6, Flow C appears to be the optimal configuration, as it minimizes additional noise across all airflow rates. Although Profile 6 does not reduce noise below the baseline, it maintains a very low increase, which is highly beneficial for practical applications. The advantage of the serrated blades in Profile 6 lies in their ability to reduce turbulence intensity and smooth the airflow, leading to lower acoustic penalties compared to other geometries.
Based on these findings, Profile 5 appears to be the most promising option for commercial swirl diffusers. It provides the lowest sound power levels and is the only solution that achieves negative ΔLwA values (in Flow B), meaning it can perform more quietly than a diffuser without blades. Profile 6 is also a recommended alternative due to its consistently low acoustic penalties and predictable performance, especially in Flow C. Conversely, Profiles 1 and 3 should be avoided in commercial applications, as they repeatedly generate the highest noise increases.

The data further suggest that flow arrangement is critical: Flow A is generally the least favorable acoustically, while Flows B and C are superior. From a design perspective, optimal blade constructions should minimize turbulence and flow separation by avoiding unnecessary transverse elements, sharp obstructions, or additional guide features that disturb the airflow. Simple, aerodynamically smooth blades appear to be the most advantageous, particularly when combined with a configuration similar to Profile 5 or the serrated concept of Profile 6. Ultimately, the most acoustically efficient commercial diffuser would utilize low-disturbance blade geometry operating in a flow configuration corresponding to B or C.

3.1 Aeroacoustic interpretation of the measured results.

Although the present study is based exclusively on experimental measurements of sound power levels, the observed acoustic trends can be interpreted using established aeroacoustic mechanisms reported in the literature. Aerodynamic noise in low-Mach-number flows is primarily associated with turbulence-induced pressure fluctuations, vortex shedding, and the interaction of unsteady flow structures with solid surfaces, as described by Lighthill, Howe and Blake [14–16].

The experimental results demonstrate that blade geometry strongly influences the generation of turbulent structures downstream of the diffuser. Profiles 1 and 3, incorporating multiple guide ribs and reinforcing elements, consistently produced the highest sound power levels. Although no flow visualization was performed, these geometric discontinuities are expected to promote boundary-layer separation and the formation of coherent turbulent wakes, leading to increased pressure fluctuations and broadband aerodynamic noise [14–16]. Similar aeroacoustic sources associated with guide elements were identified by Ostmann et al. [10].

Profiles 2 and 4 exhibited intermediate acoustic performance. Their reduced number of guide ribs or more streamlined reinforcing ribs likely weakened flow separation and vortex formation, resulting in lower sound power levels than Profiles 1 and 3.

The best acoustic performance was achieved by the biomimetic Prandtl-D blade (Profile 5). Its smooth airfoil-like geometry promotes gradual flow turning and delays boundary-layer separation, reducing wake thickness and turbulence intensity. Consequently, lower pressure fluctuations were generated, explaining why Profile 5 produced the lowest sound power levels and even outperformed the baseline diffuser in Flow B.

Profile 6, featuring a serrated trailing edge, also exhibited favourable acoustic behaviour, particularly in Flow C. According to Amiet and Brooks et al. [17,18], serrations reduce the spatial coherence of vortices by splitting the wake into smaller turbulent structures, thereby decreasing the efficiency of aerodynamic sound radiation.

Flow direction also influenced the acoustic response. Flow A generally generated the highest noise levels due to stronger shear-layer development, whereas Flows B and C promoted smoother momentum redistribution and lower sound power levels. Although these mechanisms were not directly verified experimentally, the measured trends are fully consistent with classical aeroacoustic theory. Future work should combine acoustic measurements with flow visualization techniques, such as PIV or LES, to validate the proposed mechanisms.

4. Conclusions

The conducted study demonstrated that the acoustic performance of ceiling swirl diffusers with adjustable blades strongly depends on both blade geometry and flow configuration. The experimental results confirmed that certain bade profiles can significantly increase noise emission, particularly under unfavorable flow conditions such as flow A. Profiles 1 and 3 were identified as the least acoustically efficient, consistently generating the highest sound power levels across all tested airflow rates. In contrast, Profiles 2 and 4 exhibited moderate and stable acoustic behavior, making them more suitable for general applications.

The most favorable results were obtained for Profile 5, which showed the lowest sound power levels and, in flow B, achieved noise reductions below the baseline diffuser. This indicates that appropriately designed blade geometry can not only minimize acoustic penalties but also improve overall noise performance. Profile 6, featuring serrated edges, also demonstrated good acoustic characteristics, particularly in flow C, where it minimized additional noise across all airflow rates. The improved acoustic performance of the Prandtl-D profile (Profile 5 and 6) is attributed to delayed flow separation and reduced wake thickness, whereas the serrated trailing edge promotes vortex decorrelation, reducing the efficiency of acoustic radiation. Conversely, profiles incorporating multiple guide ribs and reinforcing elements generate stronger turbulent structures and larger coherent wakes, leading to higher broadband noise levels.

The results further highlight the importance of flow direction, with flows B and C being significantly more advantageous than flow A from an acoustic perspective. These findings suggest that diffuser design should prioritize smooth, aerodynamically optimized blade shapes while avoiding unnecessary structural elements that may increase turbulence. The use of advanced manufacturing techniques, such as 3D printing, enables the development and testing of innovative blade geometries with improved acoustic properties. The observed experimental trends are consistent with classical aeroacoustic theory, indicating that reductions in flow separation, wake thickness and vortex coherence constitute the primary mechanisms responsible for the lower sound power levels measured for the optimized blade geometries. Overall, the study provides practical guidelines for the design of low-noise swirl diffusers and confirms that both blade geometry and flow configuration are key parameters in achieving optimal acoustic performance in HVAC systems. The obtained results demonstrate that relatively simple modifications of blade geometry can reduce diffuser noise by more than 1 dB relative to the baseline configuration and by up to 7–8 dB compared with unfavorable blade geometries, without modifying the diffuser housing itself.

5. Acknowledgments

The authors would like to express their gratitude to the Institute of Power Engineering – National Research Institute (Instytut Energetyki – Państwowy Instytut Badawczy) in Łódź for providing access to the acoustic testing facilities. Special thanks are extended to the staff for their technical support during the measurement sessions.

References

[1] C. Cravero and D. Marsano, "Instability Phenomena in Centrifugal Compressors and Strategies to Extend the Operating Range: A Review", Energies, vol. 17, no. 5, p. 1069, 2024. https://doi.org/10.3390/en17051069

[2] A. Corsini, F. Rispoli, A. Sheard, & T. Tezduyar, "Computational analysis of noise reduction devices in axial fans with stabilized finite element formulations", Computational Mechanics, vol. 50, no. 6, p. 695-705, 2012. https://doi.org/10.1007/s00466-012-0789-4

[3] D. Zante, G. Podboy, C. Miller, & S. Thorp, "Testing and Performance Verification of a High Bypass Ratio Turbofan Rotor in an Internal Flow Component Test Facility",, p. 51-65, 2007. https://doi.org/10.1115/gt2007-27246

[4] E. Envia, "Prediction of model scale turbofan exhaust tone noise", International Journal of Aeroacoustics, vol. 15, no. 4-5, p. 395-429, 2016. https://doi.org/10.1177/1475472x16642134

[5] Y. Biba, Z. Liu, & D. Hill, "Pipeline Compressor Redesign With the Consideration of Both Noise and Performance",, 2000. https://doi.org/10.1115/ipc2000-266

[6] S. Lohse, J. Schuler, F. Fischer, J. Seume, & S. Moreau, "On the Aeroacoustic Scaling of Axial Flow Turbines", International Journal of Gas Turbine Propulsion and Power Systems, vol. 15, no. 3, p. 9-17, 2024. https://doi.org/10.38036/jgpp.15.3_9

[7] J. Taylor, "Complete flow conditioning gauzes", Experiments in Fluids, vol. 60, no. 3, 2019. https://doi.org/10.1007/s00348-019-2682-9

[8] J. M. Kopania, G. Bogusławski, K. Wójciak, and P. Gaj, "The noise study of ceiling swirl diffuser with serrated adjustable blades," Vibrations in Physical Systems, vol. 35, no. 2, Art. no. 2024202, 2024. doi: 10.21008/j.0860-6897.2024.2.02.

[9] T. Sinnige, T. Stokkermans, D. Ragni, G. Eitelberg, & L. Veldhuis, "Aerodynamic and Aeroacoustic Performance of a Propeller Propulsion System with Swirl-Recovery Vanes", Journal of Propulsion and Power, vol. 34, no. 6, p. 1376-1390, 2018. https://doi.org/10.2514/1.b36877

[10] P. Ostmann, M. Kremer, and D. Müller, "Identification of the Aeroacoustic Emission Source Regions Within a Ceiling Swirl Diffuser," Acoustics, vol. 7, no. 1, p. 9, 2025. doi: 10.3390/acoustics7010009

[11] J. M. Kopania, G. Bogusławski, K. Wójciak, and P. Gaj, "The acoustic performance of the different types of plenum boxes for ceiling panels," Vibrations in Physical Systems, vol. 36, no. 1, Art. no. 2025103, 2025. doi: 10.21008/j.0860-6897.2025.1.03.

[12] ISO 5135:2020: Acoustics — Determination of sound power levels of noise from air-terminal devices, air-terminal units, dampers and valves by measurement in a reverberation test room.

[13] PN-EN ISO 3741:2011; Acoustics – Determination of sound power levels and acoustic energy levels of noise sources based on sound pressure measurements – Precision methods in reverberation chambers

[14] M. J. Lighthill, "On Sound Generated Aerodynamically. I. General Theory," Proceedings of the Royal Society A: Mathematical, Physical and Engineering Sciences, vol. 211, no. 1107, pp. 564–587, 1952. https://doi.org/10.1098/rspa.1952.0060

[15] M. S. Howe, Acoustics of Fluid–Structure Interactions, Cambridge University Press, Cambridge, UK, 1998. https://doi.org/10.1017/CBO9780511625872

[16] W. K. Blake, Mechanics of Flow-Induced Sound and Vibration, Vol. I and II, Academic Press, New York, 1986.

[17] D. P. Brooks, T. F. Brooks, D. S. Pope, and M. A. Marcolini, Airfoil Self-Noise and Prediction, NASA Reference Publication RP-1218, NASA Langley Research Center, Hampton, VA, USA, 1989.

[18] R. K. Amiet, "Noise Due to Turbulent Flow Past a Trailing Edge," Journal of Sound and Vibration, vol. 47, no. 3, pp. 387–393, 1976. https://doi.org/10.1016/0022-460X(76)90948-2