Numerical analysis of acoustic noise from an electronic cooling fan at flow disturbed by an external obstacle

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ABSTRACT

Axial cooling fans are widely used in data center dense Hard Disk Drive (HDD) storage systems. However, these fans emit high noise levels and degrade HDD performance at certain frequencies. Flow at the fan's inlet can be highly turbulent due to the wake generated by fan components such as struts, finger guards, stators/guide vanes, shrouds, and other external system components such as connectors and mounts, power cables and components, circuits etc. The wake-fan interaction also causes high tonal noise. Therefore a proper understanding of this noise mechanism will help optimize the cooling system in next-generation high-performance HDD enclosure systems. This paper focuses on studying this phenomenon using numerical simulations of a typical data center cooling fan combined with various simplified strut geometries as the obstacle. The high-fidelity Computational Fluid Dynamic (CFD) method, Large Eddy Simulation (LES), was used to obtain a transient flow field. The Ffowcs-Williams and Hawkings acoustic analogy was used to predict far-field noise.

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1. INTRODUCTION

Data is the new gold. In 2020, about 62.2 zettabytes (62.2 trillion gigabytes) of new data had been created or replicated worldwide, and this is projected to surpass 175 ZB by 2025 [1]. These newly created data is typically stored in devices such as small scale personal computers to large scale data centers. Here, data centers often use hundreds of storage devices, such as Hard Disk Drives (HHD), stacked into a single system to save space (see Figure 1). They also have their own thermal management systems to avoid overheating. Forced air-cooled thermal management systems dominate the data center market, with more than 95% of the share, because of their low unit cost and low maintenance cost [2].

Figure 1: Seagate Exos E 4U106 data center storage system

HDDs consist of one or several rapidly rotating disks that are coated with magnetic material, a servo system with an actuator arm and a magnetic head. The magnetic heads read and write data on nano-meter scale tracks on these disks [3]. However, external factors such as noise and vibration can increase R/W error at specific frequencies and reduce HDD performance [4].

Forced air-cooling systems often use axial fans because of their capability of generating a high flow rate using low power, and consequently, they generate high acoustic noise and vibration at high RPM [5]. This means that the data center HDDs are more susceptible to performance degradation when the cooling systems are at their full throttle. Primary noise sources of fans have aeroacoustic origins, i.e. complex non-linear fluctuations of the flow and their interaction with solid parts such as finger guards, blade leading edge (LE), blade trailing edge (TE), guide vanes [6]. Therefore, it is essential to understand the noise generation mechanisms during the design phase of data center HDD enclosure systems.

Tonal noise in axial cooling fans radiates as blade passage frequency (BPF) harmonics. It is principally caused by nonuniform loading on rotor blades which is usually attributed to flow disturbance due to obstructions such as finger guards, circuits, wires and fan support [7]. These obstructions generate wake, which then periodically interacts with downstream fan blades. The output noise level depends on the nature of the obstruction geometry and the shape of the blades. Obstruction geometries such as grilles/finger guards are mandatory in data center cooling systems as a safety requirement. Grilles can also be used for flow manipulation, i.e. by changing turbulence properties of incoming flow such as kinetic energy (TKE) and turbulence length scale [8]. Here the level of downstream TKE is shown to be directly proportional to the pressure drop of the grid [9].

To the best knowledge of the authors, there is no work found about grille generated fan noise effects on data center HDD enclosure systems in the available literature. However, there are few similar studies in different applications. Tian et al. (2009) performed an experimental and numerical study on aeroacoustic noise from an air conditioner unit with two different grille types; circular and rectangular [10]. They found that both grilles significantly increase the broadband noise of the system. Their analysis suggests that the noise was mainly due to vortex shedding from impeller, grille trailing edge, and inlet turbulence of the grille. Additionally, they observed that the circular grille pattern produced less noise than the rectangular pattern. Bach (2011) [11] conducted an experimental investigation on six metal grille patterns commonly used in fans and chassis (concentric circles, angled slats, honeycomb, swirl, loose swirl, turbine, mesh, stamped circles and wire). A-weighted overall sound pressure level (OASPL) values were measured at the near field, i.e. two inches from the fan. Results indicate that the wire grille produced the least noise at low fan speeds, while the mesh produced low noise at high fan speeds. In addition, they observed a significant elevation of the tonal component with the mesh pattern at high fan speeds. However, these effects have not been quantified and documented. Therefore, it requires a more thorough investigation at a wide range of frequencies, particularly those significant for HDD systems.

The present work is an extension of the work presented at the INTER-NOISE 2021 conference [12]. Previously, authors validated the LES and Ffowcs-Williams and Hawkings (FW-H) based noise prediction model using a fan that is typically used in a large scale data center HDD enclosure system. Theoretical background and the methodology of validation can be found in [12]. In this paper, the authors focus on evaluating fan noise effects due to simplified grille structures close to the fan's inlet.

2. COMPUTATIONAL FLUID DYNAMICS AND COMPUTATIONAL AEROACOUSTICS SIMULATION SETUP

The cooling fan used in one of Seagate's HDD enclosure system has a complex design (see Figure 2). It has a five-bladed front rotor which rotates anti-clockwise and a four-bladed rear rotor which rotates clockwise. A seven-bladed stator is located between the rotors to guide the air flow towards the rear fan and hold the electric motor. These rotors and stator are covered with a cylindrical shroud. Previously validated open fan configuration [12] has been used as the reference case. Here, the front rotor rotates at 12960 RPM, and the rear rotor rotates at 13740 RPM. Several inlet grille geometries have been tested with the same reference case. The CFD domain consists of a pressure inlet and a pressure outlet, where gauge pressure is defined as P = 0 Pa. The fluid domain has been split into four separate subdomains with sliding surface boundaries. The motion of the rotor was simulated by rotating the mesh with respect to a fixed coordinate system at a given constant RPM. The grille domain and stator domain are defined to be stationary. All wall boundaries, grille, stator, hub, rotors and shroud are defined as no-slip walls. Computational meshes for each domain were generated separately and combined using ANSYS ICEM CFD. Here, the near-wall mesh was constructed using structured hexahedral inflation layers, and the far-field was constructed using unstructured tetrahedral finite volume cells.

The flow field was simulated using LES following the approach described in [12]. The characteristic chord length is about 3 cm. The maximum Mach number and Reynolds number near the tip is about 0.18 and 1.2×10^5. The rotor blade was considered as an acoustically compact object within the interested frequency range up to 10 kHz. Second order methods were used for temporal discretization. A bounded second order central differencing scheme was used for spatial descretization. An initial transient simulation was run, using the bounded second order implicit method, at $\Delta t = 1 \times 10^{-5}$ s , for 10000 time steps. The typical cell Courant-Friedrich-Lewy (CFL) number is below 3. Here each time step consists of 10 inner iterations. Flow convergence of the simulation was observed using the volume flow rate at the outlet and showed that the simulation had reached a statistically steady state at a flow time of 0.1 s. After that, the simulation ran for an additional 32768 time steps, and acoustic source data was extracted, from surfaces aligned with walls, at each time step. These simulations were conducted using the Irish Center for High End Computing (ICHEC) Kay cluster [5]. Each simulation ran on 160 processors in parallel and took approximately three days to complete. For the FW-H calculations [13], integral surfaces were assumed to be aligned

[5]https://www.ichec.ie/about/infrastructure/kay

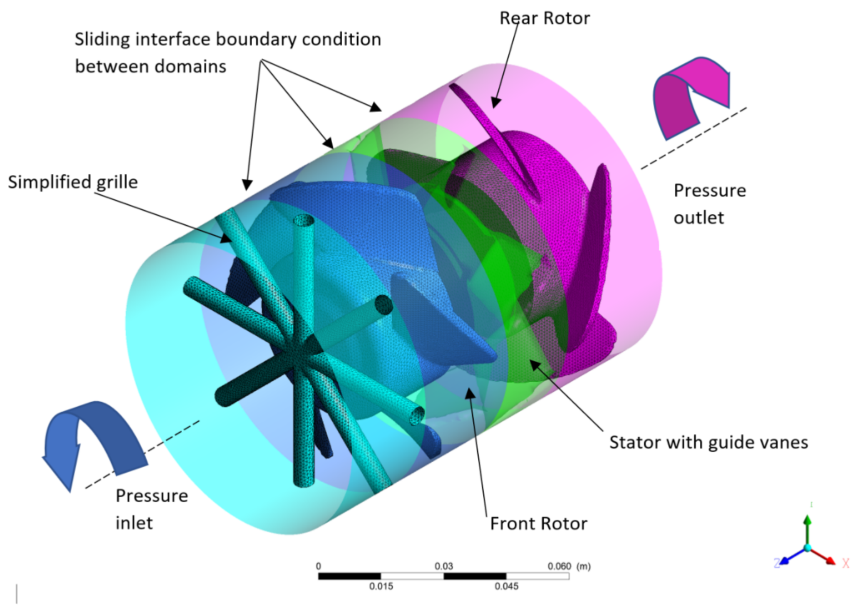


Figure 2: Geometry and boundary conditions of the contra-rotating fan used for the initial noise prediction model. Cyan: Domain with the grille, Blue: Domain with the five bladed front rotor, Green: Domain with the seven bladed stator, Magenta: Domain with a four bladed rear rotor. The shroud is defined with no-slip wall boundary condition and is not shown for clarity.

with the solid surfaces of the fan, which include: front fan, stator, rear fan and shrouds. Quadrupole acoustic sources located in the fluid domain are ignored, assuming their effects are negligible for subsonic flow. Numerical acoustic receiver points were placed at the exact locations as in the experiments.

Initially, grille (c) was placed 9 mm from the leading edge (LE) of the front blade (see Figure 3). This is the grille location on a typical data center cooling fan. Then, the distance from LE further increased to 18 mm and 27 mm to simulate the effects of the grille location. Three main grille rod diameters have been used to simulate diameter effects: D = 1.25 mm, 2.5 mm, and 5 mm. Here, D =2.5 mm is approximately the maximum thickness of the front rotor aerofoil. Grilles: single rod - a, two rods - b and four rods - c with a diameter D = 5 mm have been used to evaluate noise characteristics at different grille porosities. Additionally, a circular grille (ring - d) has been compared with rod - c to evaluate the anisotrophic effects of turbulence. Here, the ring is placed at an 85% rotor radius [14].

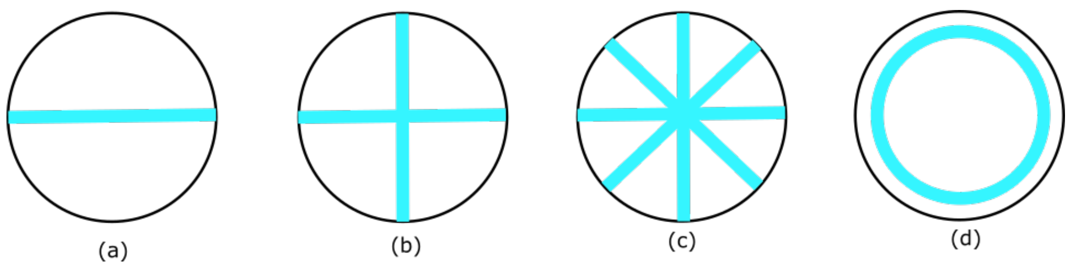
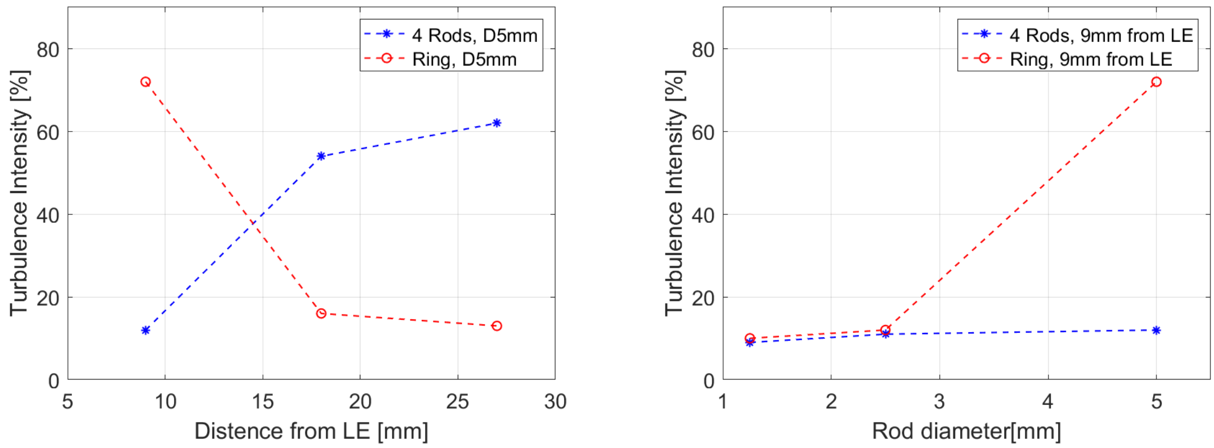


Figure 3: Grille geometry is simplified to a (a) single rod, (b) 2 rods (c) 4 rods and (d) ring. These grills were located at 9 mm, 18 mm and 27 mm from the leading edge of the front rotor.

3. RESULTS

Fans show a significant noise response when they are exposed to an obstacle geometry at the inlet. Selected results from this study are presented in this section.

Figure 4 show the average turbulence intensity (TI) at the inlet of the fan. The circular grille (ring) generated the highest TI (72%) when it was placed close to the fan inlet. The TI values are notably reduced when it is moved away from the LE. On the contrary, the radial grille (4 rods) generated the lowest TI (12%) when it was closest to the fan inlet. However, it increased up to 62% when the grille was moved away from the fan. These significant turbulence effects of the circular grille were predominant only when the diameter was larger than the rotor blade thickness. At small diameters, circular grille and radial grilles show very similar results.



(a) (b)

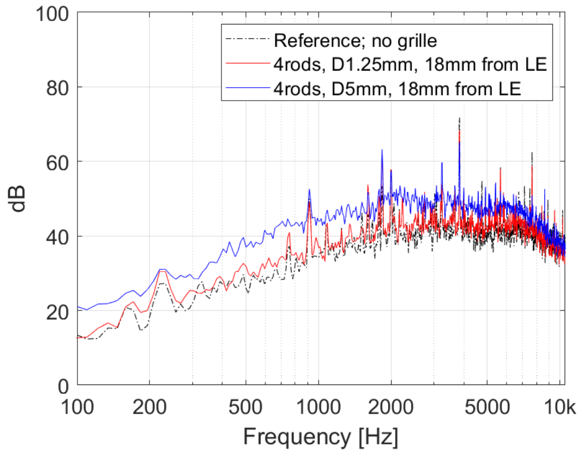
Figure 4: Averaged turbulence intensity ($\bar{u'}/\bar{U}$) at the inlet of the fan.

Figure 5 summarize the noise response at a receiver that is located 1 m from the inlet of the fan. All studies with a grille show an increased noise with respect to the reference case. This increase in the noise has a broadband nature. However, in most cases, the tonal noise shows a slight decrease compared to the reference case, and thus Overall Sound Pressure Level (OSPL) stay approximately the same. Therefore, OSPL does not give a good representation of the noise at the receiver.

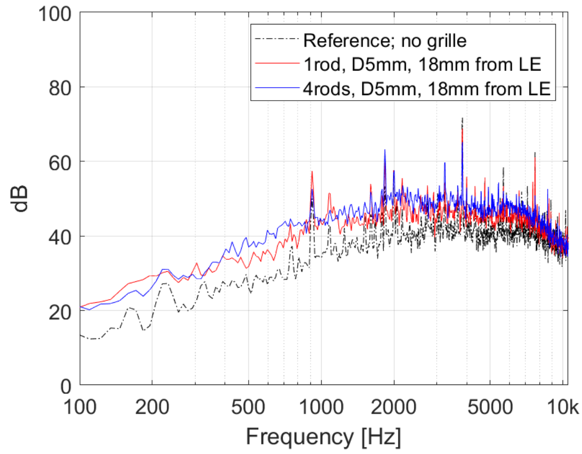
Figure 5a depicts that rods with diameters higher than the maximum thickness of the blade can significantly increase the broadband noise. This broadband noise increase is approximately 6 dB. High TI at the inlet may cause this noise to increase. This also agrees with the observations by references [10,11]. In addition to the broadband noise, there are noticeable tones shown in the spectra, particularly in cases where the grille is closest to the LE. Most of those tones do not appear in the reference case. When the grille is moved further away, they tend to disappear under the elevated broadband noise spectra. Those tones that do not disappear are somewhat reduced when the rod diameter is higher.

Noise levels increase when the number of rods increases from 1 to 4 (see Figure 5b). Increasing the number of rods or increasing rod diameter influences the area of the inlet and increases the pressure drop. Here, the pressure drop created by the grill is directly proportional to the turbulence kinetic energy (TKE) of the wake [9]. When these high energy turbulence eddies interact with the rotor blades, they emit high broadband noise levels. Therefore it is aerodynamically and aeroacoustically beneficial to have grilles with a few rods at diameters smaller than the maximum thickness of the blade.

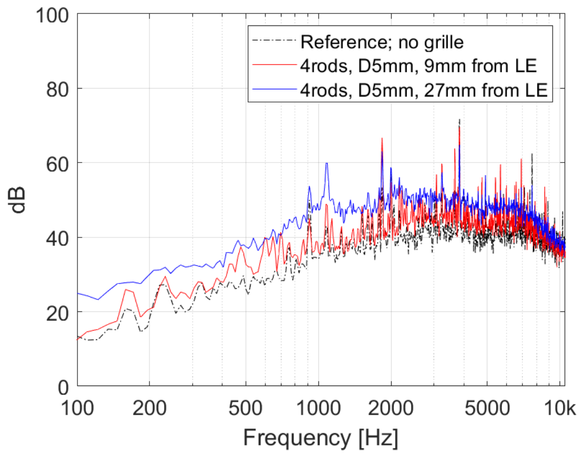
Figure 5c shows the noise response of the radial grille (4 rods) at a different distance from the LE. This also agrees with the TI results shown in the Figure 4a. This describes that the turbulent fluctuations with a predominantly tangential nature, i.e. from radial obstacles (rod), tend to grow with the distance. On the contrary, the circular grille show reduced TI when the grille is further from the rotor LE. Whereas the turbulent fluctuations with a predominantly radial nature, i.e. from circular obstacles (ring), tend to reduce with distance.



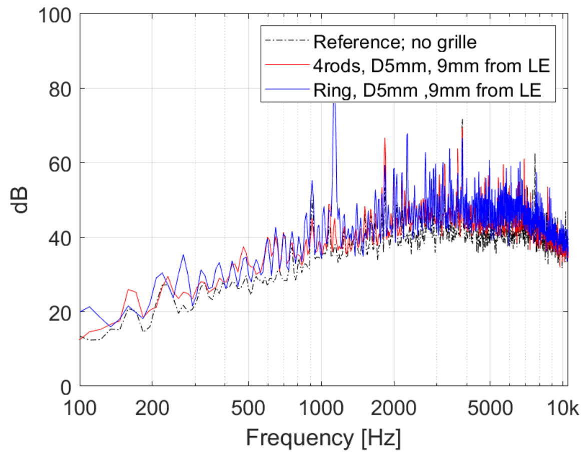
(a) Diameter



(b) Porosity



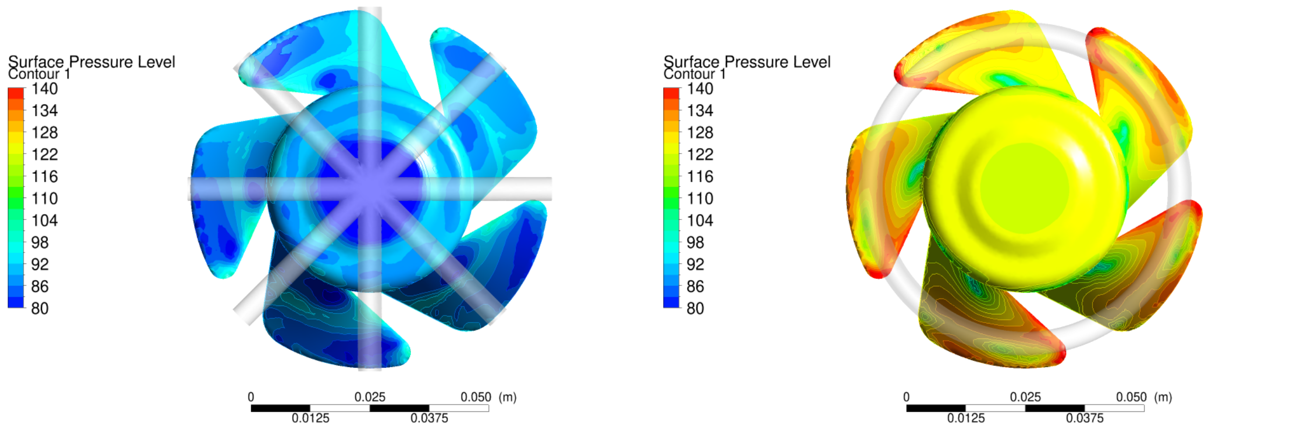
(c) Position



(d) Type

Figure 5: Aeroacoustic noise spectra at 1 m from the front of the fan calculated using modified FW-H as in the reference [12]. Noise effects due to the flow disturbance was analysed in terms of grille diameter (a), porosity (b), position (c) and grille type (d).

Another interesting observation is the high noise level due to the circular grille placed at 9mm from LE. Here, the OSPL increase about 7 dB mainly because a significant tone appears at 1135 Hz. Additionally, a number of high SPL harmonics appear throughout the spectrum. Figure 6 depicts the surface pressure levels, at the tall peak (1135 Hz) observed in Figure 5d. Here, surface pressure levels depict aerodynamic surface pressure data on a log scale relative to 2×10^{-5} Pa. Surface acoustic sources are remarkably high on the case with a circular grille. Additionally, it shows intensive acoustic sources at the leading edge, at about 80% of the radius and at the tip of the blade on the path of the wake that propagates from the circular grille. It has been shown in the literature that the intense acoustic sources of rotational machinery applications such as wind turbines are located at 75% - 95% of the blade diameter [14, 15]. Therefore, having an intense flow disturbance on fans in the outer region can adversely affect their noise characteristics. Therefore, to minimize noise, the grilles need to be carefully designed.



(a) 4-rods

(b) Ring

Figure 6: Surface pressure level [dB] of the front rotor, at 1135 Hz. Grille geometry is shown as a semitransparent overlay.

4. CONCLUSIONS

The radial obstacles (rods) nearest the leading edge that generates turbulence predominantly in tangential directions showed the minimum noise output. The circular obstacles (ring) nearest the leading edge that generates turbulence predominantly in radial directions showed a high tonal noise output. Therefore, circular grille geometries larger than the maximum thickness of the rotor blade at the outer radius must be avoided, particularly in the vicinity of the leading edge. In general, it is beneficial to have grilles with rod diameters smaller than the maximum thickness of rotor blades and place them far from the fan. It is beneficial to have an aerodynamically smooth flow path from HDD arrays to cooling fans in data center HDD enclosure systems to minimize the noise due to turbulent inflow.

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REFERENCES

[1] M. Shirer and J. Rydning. Data creation and replication will grow at a faster rate than installed storage capacity, according to the IDC global datasphere and storagesphere forecast. *International Data Corporation*, March 2021. Available at: https://www.idc.com/getdoc.jsp?containerId=prUS47560321.

[2] L. Bearn. The practical business case for data center sustainability. In *Open Compute Project Educational Webinar Series, 14 March*. Open Compute Project, 2022. Available at: https://www.opencompute.org/events/past-events/ocp-educational-webinar-the-practical-business-case-for-data-center-sustainability.

[3] Exceeding capacity, speed and performance expectations. *White Paper by Seagate*, 2011. Available at: https://www.seagate.com/files/staticfiles/docs/pdf/whitepaper/seagate-acutrac-TP624.1-1110US.pdf.

[4] Hard disk drive performance degradation susceptibility to acoustics. *White Paper by ASHRAE*, 2019. Available at: https://www.ashrae.org/file%20library/technical%20resources/bookstore/hard-disk-drive-performance-degradation-susceptibility-to-acoustics.pdf.

[5] S. V. Garimella, T. Persoons, J. A. Weibel, and V. Gektin. Electronics thermal management in information and communications technologies: Challenges and future directions. *IEEE Transactions on Components, Packaging and Manufacturing Technology*, 7(8):1191–1205, 2017.

[6] S. H. Wasala. *Numerical Analysis and Aeroacoustic Simulation of Noise from Wind Turbines*. PhD thesis, The University of Auckland, 2016. Available at: http://hdl.handle.net/2292/32089.

[7] K. L. Gee and S. D. Sommerfeldt. Multi-channel active control of axial cooling fan noise. In *INTER-NOISE and NOISE-CON Congress and Conference Proceedings*, volume 2002, pages 880–885. Institute of Noise Control Engineering, 2002.

[8] P. E. Roach. The generation of nearly isotropic turbulence by means of grids. *International Journal of Heat and Fluid Flow*, 8(2):82–92, 1987.

[9] R. I. Loehrke and H. M. Nagib. Experiments on management of free-stream turbulence. Technical report, Illinois Inst Of Tech Chicago, 1972.

[10] J. Tian, H. Ouyang, and Y. Wu. Experimental and numerical study on aerodynamic noise of outdoor unit of room air conditioner with different grilles. *International Journal of Refrigeration*, 32(5):1112–1122, 2009.

[11] M. Bach. Effects of grill patterns on fan performance/noise. *Puget Systems*, 2011. Available at: https://www.pugetsystems.com/labs/articles/Effects-of-Grill-Patterns-on-Fan-Performance-Noise-107/. Accessed on: 29/04/2022.

[12] S. Wasala, Y. Xue, L. Stevens, T. Wiegandt, and T. Persoons. Numerical simulations of flow induced noise from a dual rotor cooling fan used in electronic cooling systems. In *INTER-NOISE and NOISE-CON Congress and Conference Proceedings*, volume 263, pages 1308–1319. Institute of Noise Control Engineering, 2021.

[13] J. E. Ffowcs Williams and D. L. Hawkings. Sound generation by turbulence and surfaces in arbitrary motion. *Phil. Trans. R. Soc. Lond. A*, 264(1151):321–342, 1969.

[14] S. H. Wasala, R. C. Storey, S. E. Norris, and J. E. Cater. Aeroacoustic noise prediction for wind turbines using large eddy simulation. *Journal of Wind Engineering and Industrial Aerodynamics*, 145:17–29, 2015.

[15] S. Oerlemans, P. Sijtsma, and B. M. López. Location and quantification of noise sources on a wind turbine. *Journal of Sound and Vibration*, 299(4-5):869–883, 2007.