

Experimental Investigation of the Performance and Noise of Dyson High- Speed Fans

MSc Research Project

Dissertation

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Declaration

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I confirm that:

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Abstract

Fans and outlet guide vanes (OGV) are aerodynamic devices that can work together to produce axial flow. While their use is widespread, both can be a source of significant noise radiation in a turbulent flow. By modifying a fan and OGV provided by Dyson Ltd., several noise reduction mechanisms have been applied, and tested in the Dyson rig at the University of Southampton for acoustic and aerodynamic performance. The modified OGV designs include those with sinusoidal leading edges, slitted leading edges, and porous fan blades. The research ultimately found that all noise reduction mechanisms applied to the OGV were unsuccessful, and that the fluid-OGV interaction noise was not likely the dominant source of broadband noise. The serrated noise reduction mechanism was then applied to two fan designs, one for the trailing edge and one for the leading edge, which the former immediately showed a drop in sound pressure level (SPL). Unfortunately, due to the timing of the project and delays in manufacturing, the leading-edge serrated fan design was not manufactured in time for testing, however, both designs can be used by future researchers on the Dyson rig. This research builds off of previous findings, and can be used for further research where relevant.

Acknowledgements

Professor Phillip Joseph BSc MSc PhD, Professor of Engineering Acoustics

Dr. Joseph has been the supervisor, and the guiding figure for this project. His experience with acoustics and the Dyson rig were essential to the success of this project, as was his constant encouragement. Several papers written by Dr. Joseph and Dr. Paruchuri have been referenced in this paper.

Dr Chaitanya Paruchuri PhD, MSc (Engg), BE

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Ross Bramley – Aeroacoustics PhD MEng Acoustical Engineering

Ross has provided assistance with setting up and using the software NI Max to collect acoustic data. Ross has also provided assistance with analysing the data through the use of matlab scripts. His assistance and experience in the Dyson rig, particularly with debugging, as it often occurred, has been critical for collecting accurate and consistent data.

Mr. Kevin Smith MSc CMgr FCMI FRSA MIET MIED – Production Facilities Manager

As the manager of the production facilities Engineering, Design, and Manufacturing Centre (EDMC) at the University of Southampton, Kevin was approached for the two machining tasks: sanding the 3D printed OGVs, and the task of CNC (Computer Numerical Control) machining the two serrated fans.

Mr. Richard Dooler, CNC Programmer - EDMC, Faculty of Engineering and Physical Sciences

Richard manufactured the two serrated metal fans using the 5-axis CNC. This included converting the CAD models provided by this researcher into a set of instructions for the CNC machine to read. The serrated fan designs were amongst the more intricate parts to be manufactured in the EDMC, and his dedication and patience with the process has been much appreciated. Manufacturing a single fan took over a week, and any issues resulting in asymmetry required a restart with a new Aluminium block, which occurred four times, due to machine errors and broken CNC cutters. In the end, only one serrated fan was manufactured in time for the writing of this paper, however both fans can be used by future researchers, along with all the OGVs.

Sergi Pallejà Cabré

Sergi has provided assistance with setting up the Dyson rig, as well as providing detailed explanations on the mechanisms involved in the noise sources. Sergi has also helped with the analysis of the initial data. His encouragement and positivity have helped this researcher when it was most needed.

George Hickey

George was another MSc student who used the Dyson rig for research into liner and tip gap design. George and this researcher were often the only two researchers in the lab, and provided assistance for each other when possible.

Table of Contents

1. Introduction.....	6
1.1 Background.....	6
1.2 Research Aims	7
1.3 Literature review	7
1.3.1 Noise classification	7
1.3.3 Aeroacoustics of fans.....	9
1.3.4 Turbulence.....	11
1.3.5 Compressibility	11
2. Experimental setup - Dyson Rig.....	12
2.1 Experimental Apparatus – Dyson Test Rig.....	14
3. Methodology	15
4. Designs	17
4.1 Porous designs – noise reduction mechanisms and application	17
4.1.1 CAD Generation	18
4.2 Slitted designs – noise reduction mechanisms and application	18
4.2.1 CAD Generation	19
4.3 Serrated designs – noise reduction mechanisms and application.....	20
4.3.1 Serrated OGVs	20
4.3.2 Serrated Fans.....	22
4.4 Structural considerations	23
4.5 FEA Testing	24
4.6 Manufactured OGVs and fan.....	26
5. Results	28
5.1 OGV	28
5.1.1 Acoustic Data.....	28
5.1.2 Aerodynamic Data	29
5.2 Fan	30
5.2.1 Acoustic Data.....	30
5.2.2 Aerodynamic Data	34
6. Conclusion	34
7. Limitations	35
8. Further Study.....	35
9. References.....	36

1. Introduction

1.1 Background

Axial Fans are mechanical devices composed of a set of rotating aerofoils that compress a gas and produce axial flow. Fans can be used for thrust generation, cooling, and suction. They can be found in a variety of situation, including jet engines, powerplants, and household appliances. In addition to the axial velocity imparted on the fluid, the revolving fan also imparts a rotational (swirl) velocity onto the fluid, which is tangent to the axial velocity of the fluid. Fans can be followed by a set of Outlet Guide Vanes (OGV) (or stators for turbomachinery), which are a set of fixed aerofoils that straighten the flow by converting as much of the rotational velocity into axial velocity. A well-designed OGV will increase the axial velocity of the fluid and the upstream and downstream pressure differential further, thus making the overall process more effective and energy efficient.

Sound is produced when a gas interacts with solid objects. In the case of the fluid-fan and fluid-OGV interaction in this study, the interaction noise overlaps with the audible range of the human ear, and the sound pressure level (SPL) near the source can be significant. The sound emitted has no mechanical use, and can be disruptive for users nearby. Predictably, there has been a significant body of research dedicated to the causes of the interaction noise between aerofoil-based devices and the passing fluid, as well as design features that have the potential to reduce the (distance independent) Sound Power Level (SWL) of certain frequencies. These noise reducing mechanisms will be reviewed in this paper, as well as how they have been applied to the geometry of a fan and an OGV.

The aforementioned fan and OGV geometry used in this research project was provided by Dyson Ltd., who also provided the test rig that was used to measure acoustic and aerodynamic performance. Dyson Ltd. manufactures bladeless fans, vacuum cleaners, air purifiers, hand driers, and hairdryers (to date). These devices all require a fan (and often OGV), and therefore research in noise reduction can be used in their line of products to improve user comfort.

Research in noise reduction mechanisms is transferable to numerous aerodynamic devices. Regarding turbofan engines, fan broadband noise has become the dominant noise source, and continually increases with the ever-increasing bypass ratio. The use of large wind turbines and VTOL (Vertical Take-off and Landing) flying taxis in an urban environment is also partly restricted by the rotor-fluid interaction noise that exceeds acceptable SPL for densely populated areas.

1.2 Research Aims

The overall aim of this project is to reduce the broadband fluid-fan and OGV-fluid interaction noise by modifying the original fan and OGV design.

Objectives:

- Research existing noise reduction mechanisms, and create CAD models that apply a mechanism to the original fan and OGV design
- Test the designs structurally when needed using FEA (finite element analysis).
- Manufacture each OGV design through 3D printing, and the fan designs through CNC machining
- Test each of the (initial) fan and OGV designs in the Dyson rig at various speeds and pressure differentials, and compare the noise spectra and aerodynamic performance with that of the original design
- Determine if any design successfully reduces broad-band interaction noise without causing a significant decrease in aerodynamic performance, and optimise the designs if time permits.

1.3 Literature review

1.3.1 Noise classification

Noise produced by spinning aerofoils can be split into four categories. Monopole sources, dipole sources, and quadrupole sources. The efficiency of mechanical energy input to acoustic energy output decreases from monopole to quadrupole sources. [3]

A monopole source generates a sound wave by alternately pushing and pulling the surrounding fluid with equal force in all 3 radial directions. [3] The pressure field generated by a monopole source is shown below in figure 1. Loading and Broadband noise as seen in Figure 2 are both monopole sources. As no destructive interference occurs in a monopole source, it is the most efficient noise source of the three.

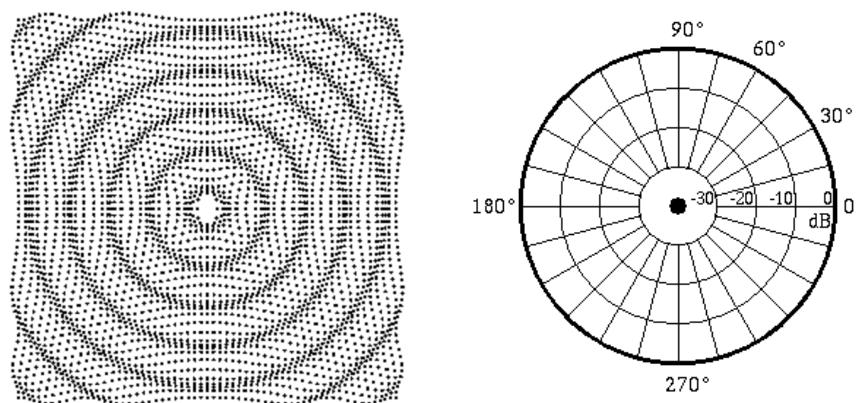


Figure 1 – Left: The pressure field produced by a monopole source. Right: The pressure field of a monopole source, represented by the bold circular line [3]

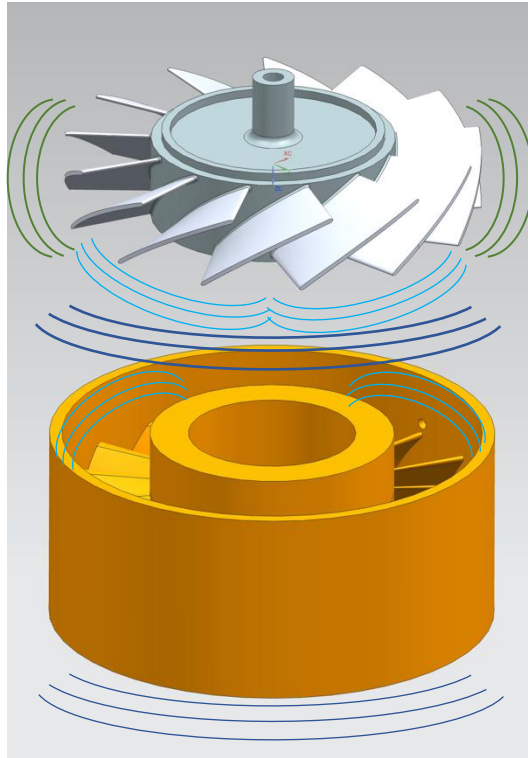


Figure 2 – Noise Sources from the fan and OGV in this study with thickness corresponding to strength. Loading and broadband noise (monopole) shown in blue, thickness and impulsive noise (dipole) shown in green, and Blade-Vortex interaction noise (Quadrupole) shown in light blue.

A dipole source is the combination of two equally strong monopole sources that are a small distance from each other (relative to the wavelength), and in opposite phase, which causes the surrounding fluid to oscillate back and forth. [3] Unlike the monopole source, the directivity pattern of a dipole source on the right in Figure 3 is not equal in all directions. Sound is radiated at 0° and 180° as well as an equally strong monopole source, but partial destructive interference occurs increasingly as the angle approaches 90° and 270° , where the two monopole sources cancel perfectly. [7]

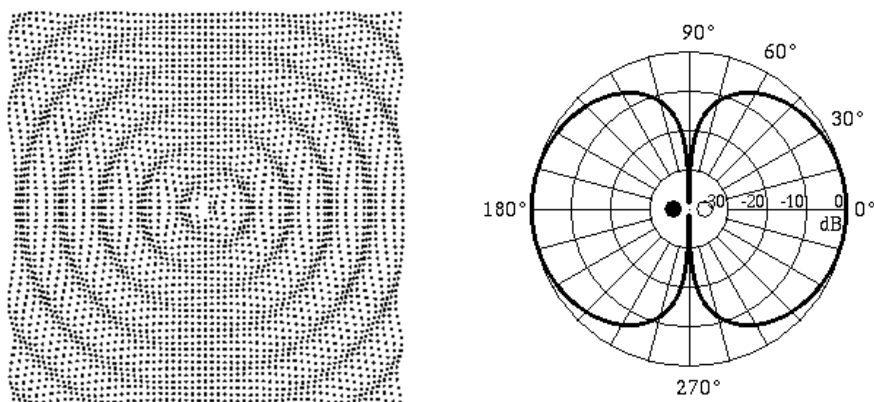


Figure 3: The pressure field produced by a dipole source and the associated pressure field on the right, represented by the bold figure of eight line [3]

Thickness noise, generated by a blade moving through the air (which increases with the blade thickness), and high-speed impulsive noise are dipole sources. Rotation noise due to blades passing a fixed point e.g., cut-off, and vortex shedding due to flow separation from blades is a dipole source. [9]

A Lateral Quadrupole (as opposed to linear quadrupole) is formed by four monopoles with alternating phase each at the corners of a square. The directivity pattern is strongest at 0°, 90°, 180°, and 270°; a line drawn from the centre of the square to each corner represents the direction to which sound is radiated strongest. However, the line from the centre to the midpoints of the sides of the square, or points in between adjacent monopoles, destructively interferes perfectly. The associated lateral quadrupole pressure field is shown on the right of Figure 4. The pressure field can be visualized as a dipole source oscillating vertically, and alternating with another dipole source oscillating horizontally.

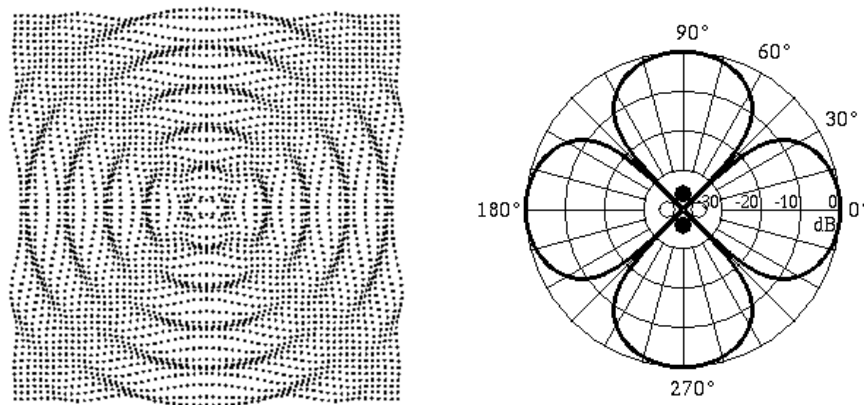


Figure 4: The pressure field produced by a lateral quadrupole source

Blade vortex interaction (BVI) noise is a quadrupole noise which occurs when approaching blades pass through the vortices of retreating blades, producing a sound referred to as "blade slap". Lighthill (1952) first showed that sound radiation by free turbulence can be represented by equally distant quadrupole sources. As turbulent flow occurs along the entire duct in this study at all fan speeds (see Equation 3), quadrupole noise will be radiated along the entire duct.

1.3.3 Aeroacoustics of fans

Fan noise can be split into tones and broadband noise. Tones appear at distinct frequencies while broadband noise is the result of a random process. On an SPL (Sound pressure level) graph, the broadband noise spectrum acts over a wide range of frequencies, and contributes most to overall SPL. It can be seen in the general curve shape, while the tones occur at distinct peaks superimposed over the broadband noise. The Blade Passing Frequency (BPF) can be calculated using equation 1 below, and will provide the frequencies at which the tonal noise can be observed. Here, n represents the rotation velocity (RPM) and t is the number of fan blades or guide vanes.

$$BPF = \frac{nt}{60} \quad \dots \textcircled{1}$$

Deterministic/Tonal Noise

Deterministic/tonal noise can be observed at multiples of the blade passing frequency (BPF), and can be split into rotor alone tones and rotor-stator interaction tones. Current literature refers to rotors and stators, which in this research would correspond to the fan, and OGV respectively.

Rotor alone tones are generated only by the rotation of the rotor, and are the result of a steady pressure field attached to the blades that rotate with the rotor. Barry *et al.* (1971) suggested that it could be caused by small variations in the incoming boundary layer thickness on surfaces leading to the rotor, or intake imperfections.

Blade-Wake interaction tones are generated by blade wakes from the rotor impinging on the downstream stators. Blade-Vortex interaction tones similarly are caused by the vortex or tangential flow velocity induced by an upstream source interacting with a downstream blade.

Broadband Noise

Blade-turbulence interaction noise may be generated when a viscous and turbulent fluid flows over the fan blades. Wright (1976) suggested that there was a minimum broadband noise generated by ducted rotors due to turbulent boundary layer shedding. Williams and Hall (1970) suggested that broadband blade self-noise was generated at the trailing edge of blades when a turbulent boundary layer was present, and it was caused by the unsteady loading on the blades and scattering. Broadband rotor alone noise can also be generated by inflow distortion, and random blade loading fluctuations induced by inflow turbulence leads to broadband noise (Barry *et al.* 1971).

Rotor-stator broadband interaction noise is also caused by blade wakes from the rotor impinging on the downstream stationary stators, similar to interaction tones. The difference is the turbulence from rotors produces random loading fluctuations on the stators and thus broadband noise.

Narrowband Noise

Narrowband noise are peaks with non-zero width (Hanson, 1974). This is a type of rotor alone noise that can occur when turbulence becomes distorted.

Installation Effects

Installation effects arise from the differences between the ideal theoretical geometrically and the physically manufactured geometry of a system. Material and geometric imperfections in the duct, fan, OGV, flow straighteners and conditioners are all possible sources of noise radiation. Dominique *et al.* (2018) splits installation effects into three components. The first component arises from the non-uniformity of the flow interacting with the fan, which causes unsteady rotating forces to act on the fan blades. These forces are the dominant noise source for fans at low tip Mach number and produce turbulence. The second component occurs when the fan rotation speed is outside the optimal value. In this study, the operating point is 4400 RPM (provided by Dyson). At speeds lower or

higher than 4400 RPM, the aerodynamics of the fan and the OGV are not optimised and their performance may fluctuate, particularly at the minimum speed (2000 RPM) and maximum speed (7000 RPM). The last component is a result of the acoustic radiation efficiency of the duct.

1.3.3 Flow induced noise in a straight duct

Tao (2016) suggests that a main source of noise from a duct arises from turbulence induced on the duct wall. This background turbulence will be present in all the tests in this study. Tao also found that the flow induced noise was dependent on the cross-sectional area and the flow speed, not the duct length. It was shown that the sound power level (SWL) in an empty duct follows the U^5 power law, where U is the flow speed.

1.3.4 Turbulence

The type of flow in the duct can be determined using the Reynolds Number equation below. For Re values greater than 4000, the flow is turbulent.

$$Re = \frac{uL}{\nu} = \frac{\rho uL}{\mu} \quad \dots \textcircled{2}$$

Here, Re is the dimensionless Reynolds number, ρ is the fluid density, u is the characteristic velocity, L is the characteristic length, which would be the duct diameter in this case, and μ is the fluid dynamic viscosity.

For the minimum fluid velocity 3.9 ms^{-1} , which is produced by the minimum fan speed of 2000 RPM with the baseline fan and OGV:

$$Re = \frac{\rho uL}{\mu} = \frac{1.2041 \times 3.9 \times 0.099}{18.13 \times 10^{-6}} = 2.56 \times 10^4 > 4000 \quad \dots \textcircled{3}$$

Therefore, the flow is turbulent in the duct. Because $Re > 10^4$, inertial forces dominate viscous forces for all fan speeds. Turbulent flow has large fluctuations in velocity, and the dominating inertial forces cause unsteady eddies to form in the flow, creating turbulence.

1.3.5 Compressibility

It can be determined if the effects of compressibility should be accounted for with regards to the flow around the fan and OGV with the calculation below.

The fluid at the tip of the fan blades could be compressible if the Mach number is greater than 0.3. Given the diameter of the fan is 99.3 mm , then at the maximum rotation speed of 7000 RPM, the tip velocity $V_{t,max}$ is:

$$V_{t,max} = \frac{d\pi RPM}{60} = 36.395 \text{ ms}^{-1} \quad \dots \textcircled{4}$$

The axial flow produced by the 7000 RPM fan speed with the baseline fan and OGV was measured to be 11.84 ms^{-1} in the preliminary tests. The fan tip velocity relative to the

fluid can be determined using a velocity triangle which combines the axial flow with the rotation of the fan:

$$V_{total,max} = \sqrt{11.84^2 + 36.395^2} = 38.17ms^{-1} \quad \dots \textcircled{5}$$

Using the Mach number equation below, the maximum tip Mach number can be determined, as well as the compressibility of the fluid in the region. Here the Mach number M is dimensionless, u is the local flow velocity (previously represented as V) and c is the speed of sound in the medium.

$$M = \frac{u}{c} \quad \dots \textcircled{6}$$

$$M_{max} = \frac{38.17}{343.21} = 0.111 < 0.3$$

Therefore, the fluid at the blade tip at the maximum fan speed can be assumed to be incompressible as $M < 0.3$. Using a similar calculation at 2000 RPM, $V_{t,min} = 10.37ms^{-1}$, $V_{total,min} = 10.80ms^{-1}$, and $M_{min} = 0.031$.

2. Experimental setup - Dyson Rig

The Dyson rig was used to collect all acoustic and aerodynamic data for this study, and can be seen below in Figure 5.



Figure 5 – the Dyson rig at the University of Southampton

The components of the rig in the region circled in green in Figure 5 are shown below in Figure 6 in an exploded view. Unseen in the assembly is the motor that passes through the centre of the OGV. The OGV simultaneously acts as an aerodynamic device to increase axial flow velocity V_z , and the structural support for the motor and fan. The sanding process for the OGVs, which will be discussed further in the manufacturing section required even sanding around the OGV to ensure that the motor and fan were exactly concentric with the line of axial flow z .

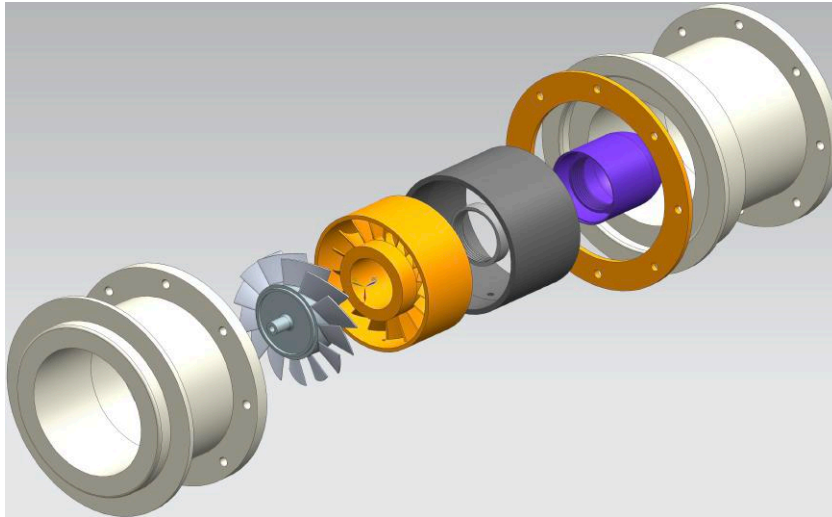


Figure 6: Exploded view of the Fan-Motor-OGV assembly

The two components that were modified in this study were the fan and the OGV, and the coordinate system and variables referenced in this paper are shown below in Figure 7.

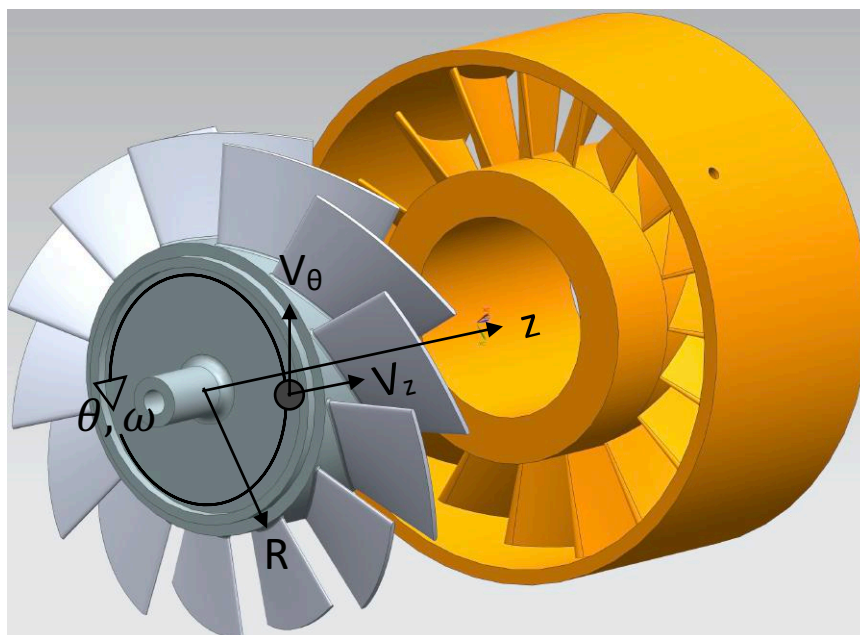


Figure 7 – Expanded view of the original fan (left) and OGV (right), with the symbols used for the coordinate system and velocities

2.1 Experimental Apparatus – Dyson Test Rig

The Dyson test rig is roughly 7 metres long and its key components are shown below in Figure X. The ascending numbers 1-10 correspond to each component in the rig viewed from left to right. 1) inlet orifice plate, 2) Fin-type flow straightener, followed by contraction section, 3) inlet anechoic termination, 4) hard-walled ducts, 5) grid-type flow conditioner, 6) upstream microphones 7) fan, motor and OGV, 8) Fin-type straightener, followed by downstream microphones, 9) outlet anechoic termination, 10) expansion section, followed by outlet and throttle. 11) NI interface, 12) Amplifier, 13) Calibrator / Manometer, 14) Motor power supply and its remote control (left).

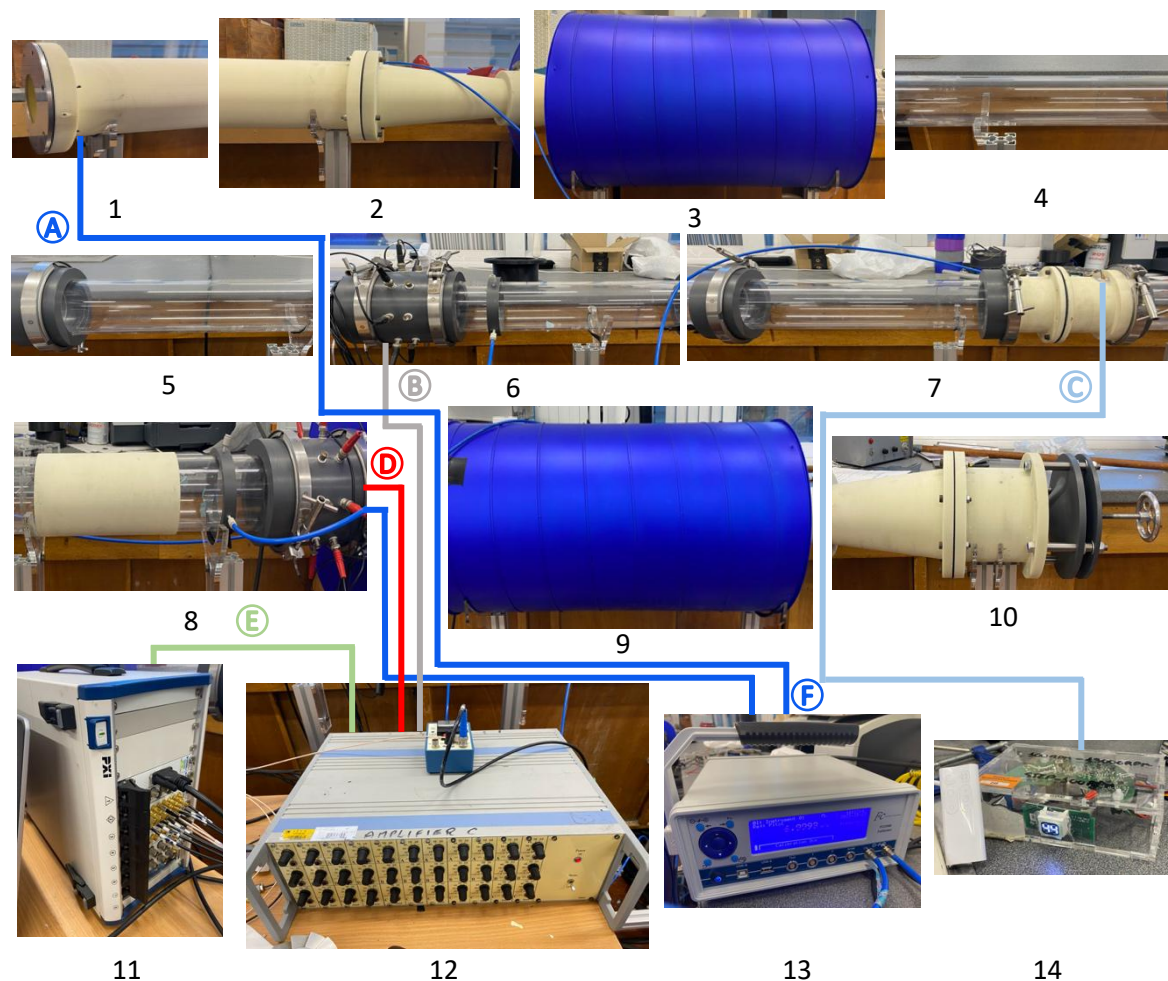


Figure 8 – Detailed setup of the Dyson right and data collection apparatus

The inlet orifice pressure and downstream pressure are connected to the calibrator / manometer (13) via tube connections at ① and ② respectively. The upstream and downstream microphones are each connected to the amplifier 12) via the connections ③ and ④ respectively, each connection representing 8 microphone cables. The motor power supply is connected to the motor via the power cable ⑤. The upstream and downstream microphone data is transmitted to the NI interface by connection ⑥, which represents all 16 cables from the upstream (1-8) and downstream (9-16) microphones. The NI interface (11) transmitted all acoustic data to the computer (not shown).

The flow straighteners help eliminate surface and sub-surface vortices and reduces vibration. The flow straighteners reduce disturbances in the flow while the conditioner allows a redistribution of the speed profile, at each point of the section of the pipe. [4] Flow straighteners, such as tube bundles, are effective at removing swirl from a flowing fluid, but less effective at normalizing longitudinal velocity variations from the fluid. Isolating conditioners, such as plate type, are effective at both removing swirl and redistributing the flow profile symmetrically. Plate type conditioners typically produce less pressure loss than tube bundle types. [5] These flow conditioners restore the flow to a uniform profile when the velocity profile is disturbed or altered by disturbances such as bends, diameter change or protrusions into the flow, all of which are present in the Dyson rig. [5]

3. Methodology

1. Initial acoustic and aerodynamic data from the baseline fan and OGV were collected.
2. Noise reduction mechanisms were researched, and mechanisms that could practically be applied to the geometry of the fan and OGV were chosen - the CAD models provided by Dyson were solid bodies, and lacked the step-by-step process that these designs were created from, to protect intellectual property. Therefore, the design modifications had to be applicable to a 3D body, rather than a more easily modified bottom-up CAD part.
3. A set of designs were planned for each noise reduction mechanism. For example, introducing slits onto the leading edge of the OGV was a chosen noise reduction mechanism, and its set of designs tested a range of slit dimensions. This would allow the effect of the slit as well as the slit length/width to be studied. While several noise reduction mechanisms such as slits, serrations, and porosity have had success in a variety of situations, the optimal dimensions of the noise reduction mechanism are mostly applicable to those parts in the study. The equations that give rise to the optimal dimensions of leading-edge serrations on a particular propeller or flat plate for example, may be significantly different to that of the fan and OGV in this study. Currently, there is no established equation for the optimal dimensions of leading-edge serrations that can be applied to any propeller or fan that includes variables such as tip diameter, advanced pitch, the number of blades, rake angle, skew angle, chord and pitch angle, and rotational velocity. The method employed in this study involves an experimental approach, with further design optimisation to be conducted based on the results of the initial designs.
4. The initial OGV and fan designs were created on Siemens NX. The method for generating these designs can be seen in section 4 -Designs.
5. The Initial OGV designs were manufactured first. 3D printing was the chosen method for manufacturing (the baseline OGV provided by Dyson was also 3D printed).
6. FFF (fused filament fabrication) 3D printing was initially used for two designs Serrations-B and Slits-A, but difficulties in sanding the parts, necessitated the switch to Stereolithography (SLA) 3D printing, which was used for Porous-A, Serrations-A, and Serrations-C.
7. All 5 initial OGV designs were tested in the Dyson rig using the metric in table 1.

8. Following the limited (acoustic) effect of the design modifications on the initial OGV designs, the remaining OGV designs were not manufactured and tested, and instead resources were put towards testing serrated fan designs.
9. Following a similar process for the OGV designs, the fans modified on Siemens NX.
10. FEA testing on ABAQUS was conducted to ensure that the fans could withstand the centrifugal stress at the maximum RPM.
11. Both fans were manufactured using a Haas VF-7/40 5-axis CNC in the EDMC (Engineering, Design, and Manufacturing Centre) at the University of Southampton.
12. The trailing edge fan design was tested following the same metric for the OGV. Unfortunately, manufacturing for both the serrated leading edge and trailing edge design had to be restarted a total of 5 times (to date) and the leading-edge serrated design was not manufactured in time for testing.

The metric below in Table 1 was used to assess the performance of each fan and each OGV. The first set of data involved collecting the SPL at various fan rotation speeds. The *operating point* of the fan (4400 RPM) is the rotation speed Dyson have used to optimise the fan and OGV. To better understand how each OGV and fan generally performed, a wide range of fan rotation speeds were used. The lowest speed was 2000 RPM, and the highest speed was 7000 RPM. While the motor can spin at a maximum of 13,000 RPM, speeds higher than 7000 RPM were not investigated, as the motor itself became a dominant source of noise, and therefore the acoustic data may not have been the direct result of Fluid-Fan or Fluid-OGV interaction, but the motor itself. In addition to the acoustic data collected for each Fan and OGV for each rotation speed, the average flow speed was also collected using the Portable Calibrator and Precision Manometer (FC0560).

In addition to the RPM data collected, another set of pressure differential data was collected. In these tests, the throttle (at the end of the test rig, see Figure 5) was adjusted to constrict the size (surface area) of the pressure outlet, allowing the acoustic and aerodynamic performance of each fan and OGV to be collected at various pressure differentials (dP) at the operating point (4400 RPM). This data was particularly useful for determining the acoustic and aerodynamic performance of each fan and OGV when the flow encountered resistance, e.g., when a Dyson vacuum cleaner was almost full.

Table 1 – Metric used to collect data for all OGV and fan designs

		Fan Operating Point (4400RPM)				
		Design				
Pressure Difference (dP)	Baseline	Serra-A	Serra-B	Serra-C	Slits-A	Porous-A
50.00						
68.75						
87.50						
106.25						
125.00						
	(dP)	Design				
Fan Speed (RPM)	Baseline	Serra-A	Serra-B	Serra-C	Slits-A	Porous-A
2000						

3000						
4000						
4400						
5000						
6000						
7000						

4. Designs

4.1 Porous designs – noise reduction mechanisms and application

Porous aerofoils have been experimented used in a variety of experiments due to the potential for noise reduction. Previous experiments typically place a set of equally spaced rows of equally spaced holes perpendicular to the chord and span of the aerofoil, which allow air to pass from the high-pressure area (lower surface) to the low-pressure area (upper surface). The introduction of porosity can reduce broadband noise, however, the increase in drag from the holes will decrease aerodynamic performance, and is the main disadvantage of their use.

Geyer et al. [13] suggested that the holes may allow turbulence kinetic energy to be dissipated in the holes. Furthermore, the turbulence structures are displaced from the surface of the aerofoil, thereby reducing the pressure fluctuations on the surface, which is a source of far-field noise. The holes also allow flow to be injected into the boundary layer and increase its thickness, thereby leading to an effective increase in the leading edge, which is known to reduce Blade-Turbulence interaction noise. [13]

The diameter of the holes in the fully porous design in Figure 9 below was set to 1mm to limit the disruption to the flow and drag around the vanes. These holes were spaced 1 cm apart along the span and chord of the vanes. Partially porous aerofoils with only one or two rows of holes were shown by Joseph et al. [1] to potentially reduce noise, with the added benefit of a lower increase in drag. The partially porous design with two rows of holes can be seen on the right in Figure 9. The hole size was increased from 1 mm to 2.5 mm to allow for more communication between the upper and lower surface of each vane.

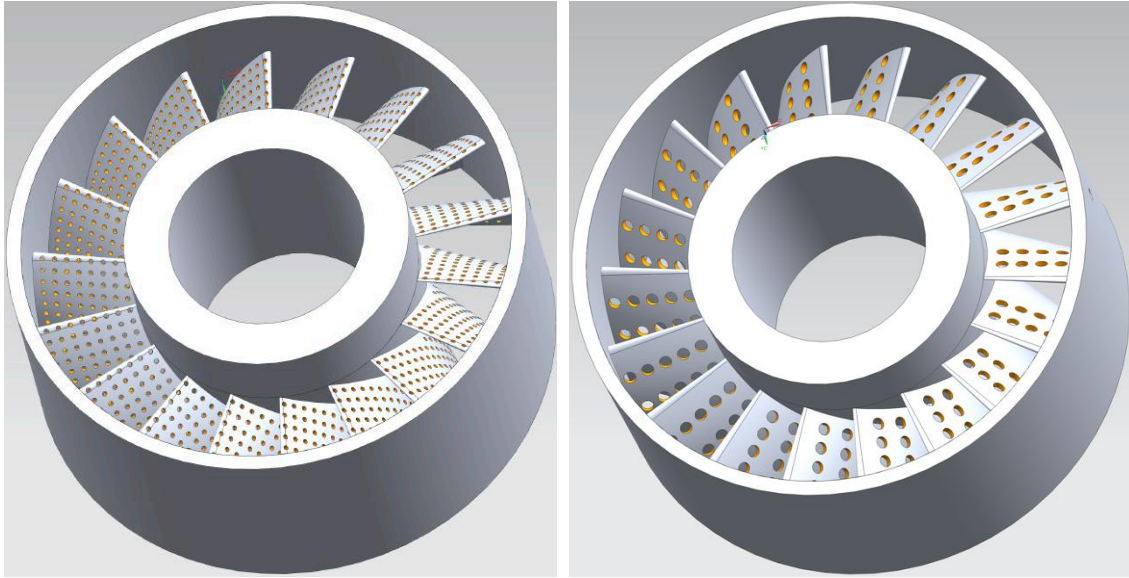


Figure 9 – Porous Designs: Fully porous design (left), and partially porous design (right).

4.1.1 CAD Generation

To ensure that all the holes penetrated the vanes perpendicularly, a plane was created for each row of holes, and a set of equally spaced cylinders were drawn perpendicular to the plane. The cylinder set was then copied radially 17 times and subtracted from the original geometry.

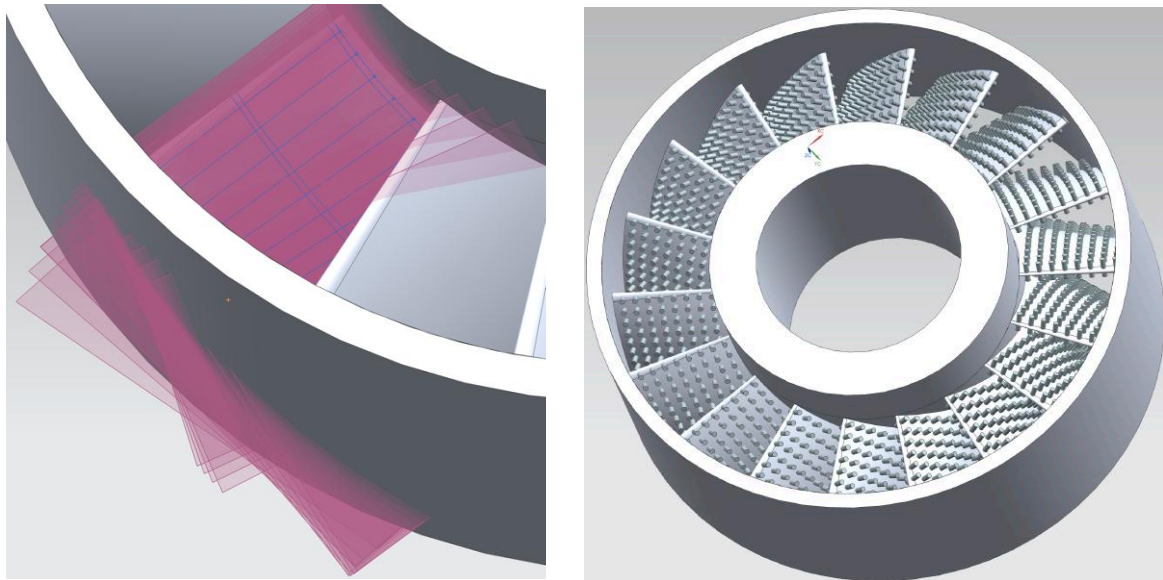


Figure 10 – The construction process of the Porous-A design (fully porous).

4.2 Slitted designs – noise reduction mechanisms and application

Introducing a set of equally spaced narrow slits onto the leading edge of aerofoils or flat plates is another noise reduction mechanism that has been tested. The findings in the experiments conducted by Paruchuri et al. [2] were consistent with the hypothesis that the slits produce two equally strong source regions at each side of the slit, which then destructively interfere and reduce noise radiation.

Finding the exact slit length and slit height for optimal noise reduction is not straightforward, and depends heavily on the aerofoil shape and the flow characteristics. This is further complicated due to the negative aerodynamic effect of wide and deep slits, which essentially remove material from the vane and reduce its lifting potential. Narrow slits in [2] with a width of 0.7 mm to 2.2 mm were shown to be most successful in reducing SPL, compared to wider slits up to 10.3 mm wide, and on average showed a reduction of 8 dB. Slit heights in the same experiment ranged from 9.9mm to 49.5mm with the highest broadband noise reduction with the longest height.

Previous research in slitted aerofoils suggested that narrow and long slits would be best for noise reduction, however aerodynamic performance was best with thinner and shorter slits. For this study, three slitted designs for the OGV were created, and can be seen in Figure 11: Slit-A had a slit width $w = 1 \text{ mm}$, and a slit length $2h = 5 \text{ mm}$. Slit-B had a slit width $w = 1 \text{ mm}$, and a slit length twice as long $2h = 10 \text{ mm}$. Slit-C had a wider slit width $w = 1.5 \text{ mm}$, and a slit length $2h = 5 \text{ mm}$.

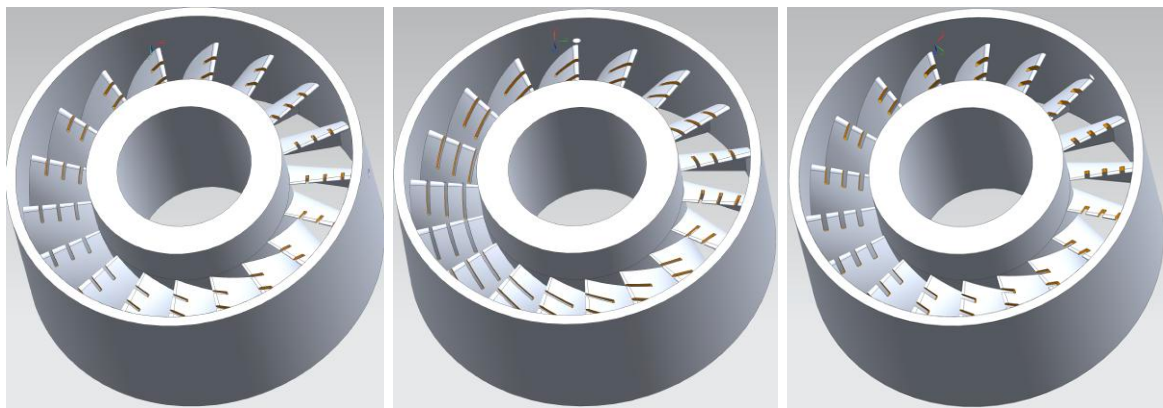


Figure 11 – OGV slitted designs: Slit-A (left), Slit-B (middle), and Slit-C (right).

4.2.1 CAD Generation

A set of circles were drawn in the radial plane R , and were projected onto the upper surface of a single vane. The projected curves were then projected onto a plane roughly tangent to the OGV surface where the slits were located. The curves were lengthened beyond the leading edge, and the opposing edge was shortened such that the overlapping area of the curve and the vane was equal to the slit height and width. A face was filled in and thickened into a solid body which was copied radially 18 times for each vane, and then subtracted from the baseline OGV. The construction curves and planes can be seen in Figure 12.

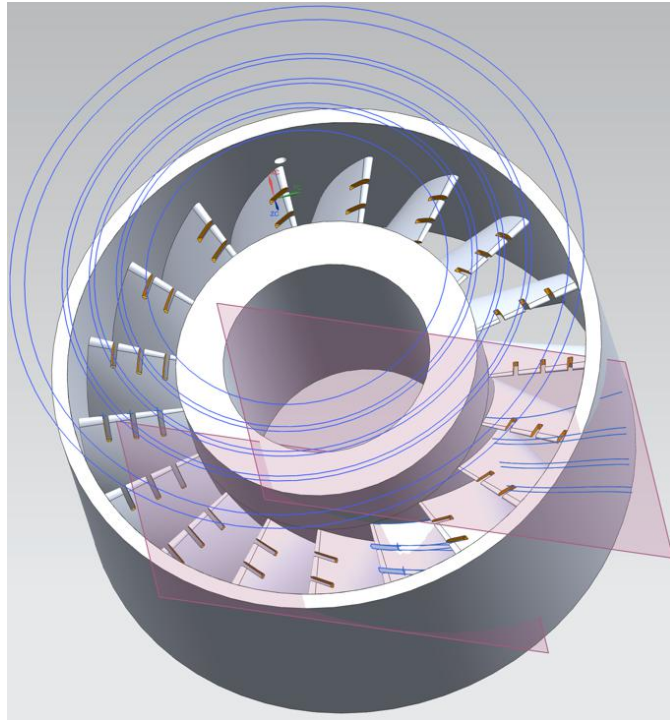


Figure 12 – The construction curves and planes of the Slits-C design.

4.3 Serrated designs – noise reduction mechanisms and application

4.3.1 Serrated OGVs

Summarised in [14], the noise reduction mechanism in leading edge sinusoidal serrations works by reducing the source strength all points along the leading edge (through destructive interference) apart from the root location, where the source strength is unchanged. This effect results in a single compact source distribution that is concentration at the serration roots, and significantly reduce broadband turbulent leading-edge interaction noise.

As with previous noise reduction mechanisms, the optimal wavelength and height of the serrated leading-edge noise reduction mechanism depends on the overall curvature in three axes of the aerofoils, and is unique to each geometry. Therefore, to provide a starting point for serrated leading edge optimisation process for the OGV in this study, four designs were drawn, and three of which were printed in the initial set of OGVs.

In both designs below in Figure, the amplitudes of the sine curve are equal; the peak-to-peak value $2h = 10 \text{ mm}$, and the wavelength $\lambda = 3 \text{ mm}$ for Serra-A and $\lambda = 5 \text{ mm}$ for Serra-B.

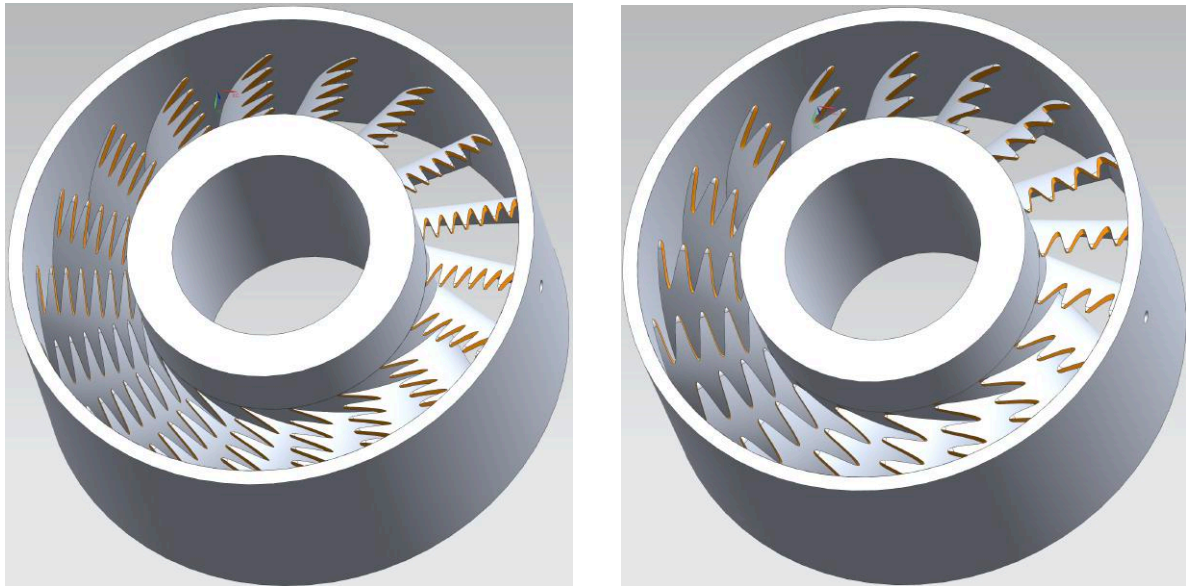


Figure 13 – OGV Leading edge serrated Designs: Serra-A (left) and Serra B (right).

Two further serration designs were created below in Figure 14. Serra-C has the same peak to peak value but the longest wavelength $\lambda = 10 \text{ mm}$, on the right, another design had half the amplitude of the other designs, but the same wavelength as Serra-A. This design was to be printed based on the results of the other serrated designs. Due to the absence of change in SPL for all initial designs the half amplitude design was not printed.

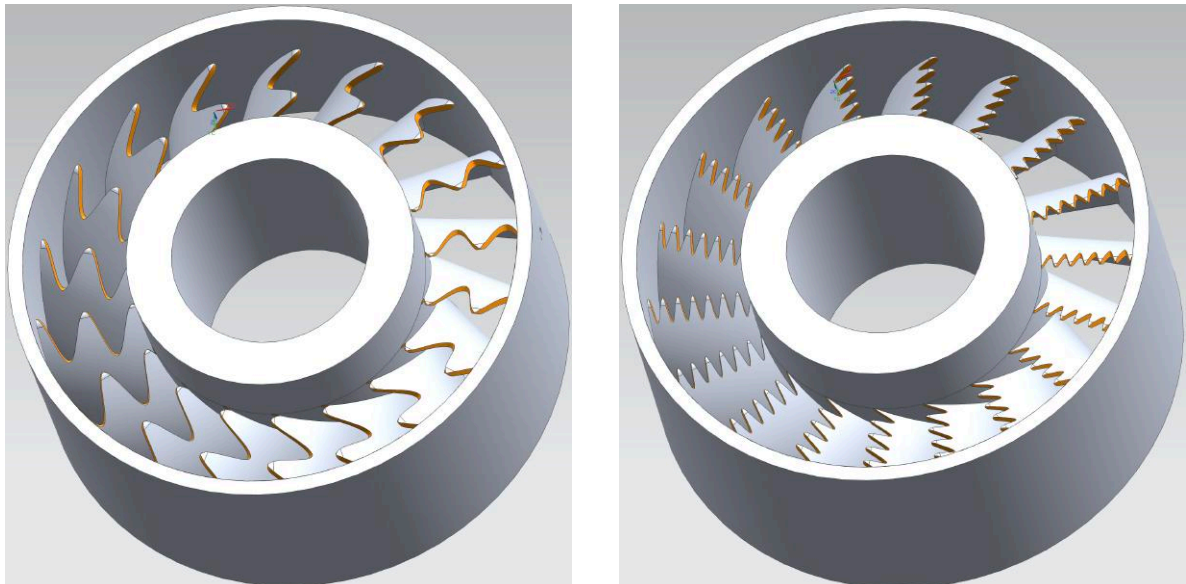


Figure 14 – Leading edge serrated Designs: Serra-C (left) and Serra-A half amplitude (right).

The serrated designs were drawn by first plotting a sinusoidal wave on a plane tangent to the vane near the leading edge. The wave was thickened into a body which was copied

radially 17 times for each vane, and then subtracted from the original OGV. A similar process was used for the serrated fan designs, with the addition of adding a spline to the tip of the blade that smoothened the transition from the sign wave to the edge of the blade.

4.3.2 Serrated Fans

As mentioned in Biedermann et al. [20], transferring serrated edges from fixed aerofoils and flat plates to rotating aerofoils in propellers and fans can also produce similar noise reduction benefits, however, finding the optimal serration dimensions may be more problematic due to the interaction between adjacent blades, as well as the more complex geometry of spinning aerofoils. The experimenters in [20] found that significant noise reduction occurred at the blade tips of leading-edge serrated rotors, and attributed the lower SWL (Sound Power Level) to the reduced blade tip vortex strength.

Experiments conducted by Liang et al. [15] indicated that trailing edge serrations on an impellor achieved significant trailing edge noise reduction at low frequencies, and broadband noise reduction of 3dB. Regarding serrations on a flat plate in a low to moderate Reynolds number flow $1.6 * 10^5 < Re < 4.2 * 10^5$, trailing edge serrations were able to reduce noise by 13dB in the narrowband frequency due to a reduction of vortex shedding at the trailing edge. [16] Another study conducted by Chong and Vathylakis involved adding serrations to the trailing edge of an aerofoil. It was claimed that “the interaction between the vortical structures and turbulent boundary layer redistributed the momentum transport and turbulent shear stress near the serration, and thus affected the self-noise radiation.

Previous work on serrated fans have shown that the optimal serration amplitude and wavelength for rotating aerofoils can vary significantly, case to case. In order to investigate the effect of the serration amplitude or peak-to-peak value $2h$, three leading edge fan designs were drawn with varying peak-to-peak values, the first of which can be seen below in Figure 15 on the left with $2h = 3mm$, and the second design on the right had $2h = 6mm$.

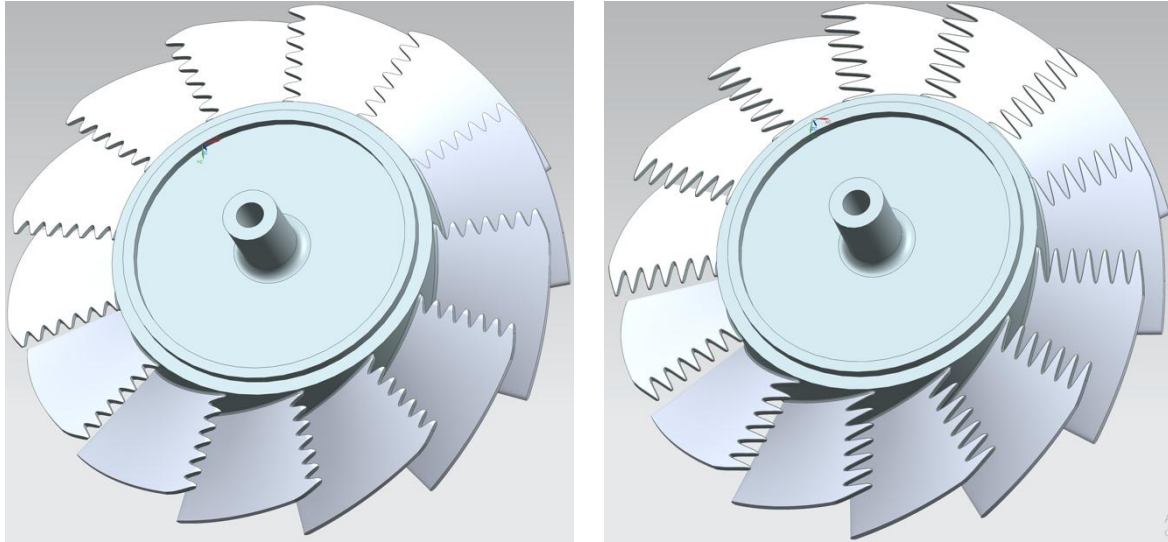


Figure 15 – Fan leading edge serrated Designs: Fan-LE-Serra-A (left) and Fan-LE-Serra-B (right).

The third design in figure 16 below on the left, with the highest peak-to-peak value of peak-to-peak value $2h = 10\text{ mm}$. To ensure a fair test was given to all three designs, the wavelength $\lambda = 3\text{ mm}$ was unchanged.

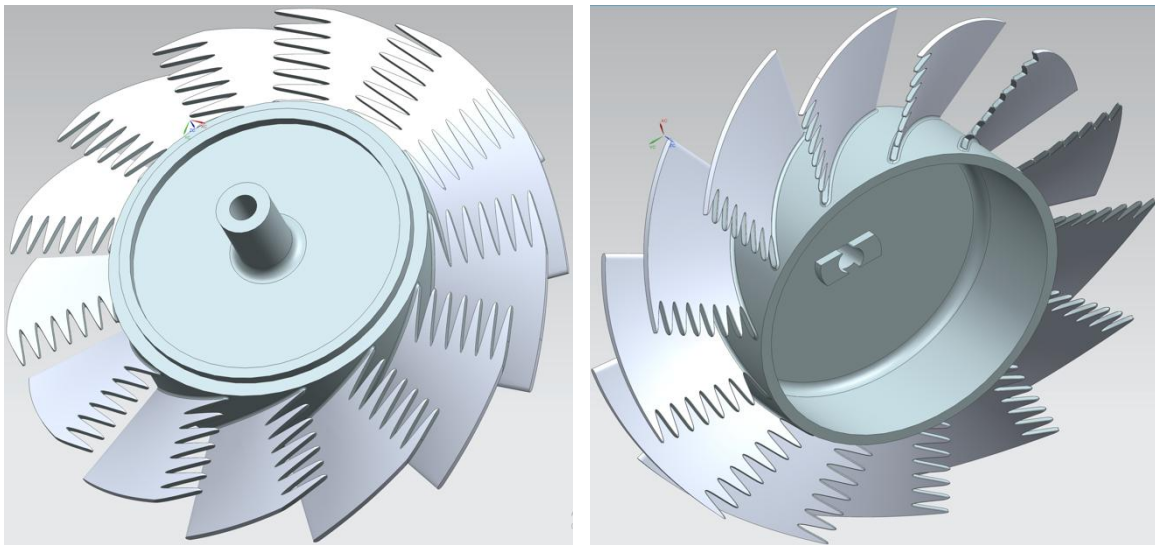


Figure 16 – Fan leading edge serrated Designs: Fan-LE-Serra-C (left), and fan trailing edge serrated design Fan-TE-Serra-A Serra-A (right).

4.4 Structural considerations

The design modifications to the fan and the OGV remove material from the original (baseline) geometries, and therefore have the potential to invalidate any structural testing conducted by Dyson. The high rotational speed of the fan will exert significant centrifugal force on the structure. The resulting stress, can cause the material to yield, permanently deform, or even fracture completely. To prevent damage to the components in the test rig, Finite Element Analysis (FEA) on the modified fan geometries was conducted. The OGV is

stationary in the experimental setup and therefore is not subject to the same centrifugal force. As only the passing fluid and vibrating motor are able to cause stress, no FEA testing was conducted, however, a general limit of 25% was placed on the maximum amount of material removed from each vane.

4.5 FEA Testing

Each fan was to be machined from a single block of 6082 T6 Aluminium using a 5-axis CNC. This process took one week per fan, and before resources were committed to the manufacturing of each block, the strength of each fan was tested on FEA software ABAQUS to ensure that they could withstand the centrifugal force from spinning without fracturing or deforming significantly.

The maximum rotation speed in this study is 7000 RPM, or $733.04 \text{ rad s}^{-1}$. To ensure that a minimum safety factor of 1.5 was reached, the centrifugal force placed on the fans were 150% of the maximum force they could be expected to experience, which corresponded to an angular velocity equal of $\sqrt{1.5}\omega_{max} = 897.7804 \text{ rad} \cdot \text{s}^{-1}$, according to the centrifugal equation 7 below.

$$F = m\omega^2 r \quad \dots \textcircled{7}$$

The material parameters set for 6082 T6 Aluminium included a Young's Modulus $E = 70 \text{ GPa}$, a Poisson's ratio $\gamma = 0.33$, and a density $\rho = 2.7 \text{ gcm}^{-3}$ [12].

The fan that experienced the highest deformation and Von Mises stress was the leading-edge serrated fan with $2h = 10 \text{ mm}$. This was to be expected as this design had the highest amplitude, and the root location between the serrations had the tightest radius, and would be expected to experience the greatest stress of all designs. The simulation process of this fan is discussed below.

The mesh for this simulation was composed of tetrahedral element types, which was found to reduce cell skewness at the peaks of the serrations; highly skewed cells in the mesh could decrease accuracy and destabilize the solution. A coarse, medium, and fine mesh were created by setting the global seed value to 1.5 mm, 1 mm, and 0.6 mm respectively. The seed value for the fine mesh was set as low as possible without exceeding the 250,000-node limit of the academic license. The displacement and Von Mises stress converged as the mesh was made finer. The difference in maximum displacement between the coarse, medium, and fine mesh was 8 microns and 6 microns respectively. This result indicated that further refining the mesh would be unlikely to significantly affect the accuracy of the simulation.

In Figure 17 below on the right, the two boundary conditions placed on the fans are shown. The green arrows pointing in the R (radial) direction represent the centrifugal force from spinning. The region near the axis of rotation z is the encast  boundary condition (fully fixed with no movement and rotation in all axes) on the centre of the fan. While the fan in reality is allowed to rotate in the rotational direction θ , the encast  boundary condition would be adequate, as the fan in the model was not rotating (but has the centrifugal force of spinning).

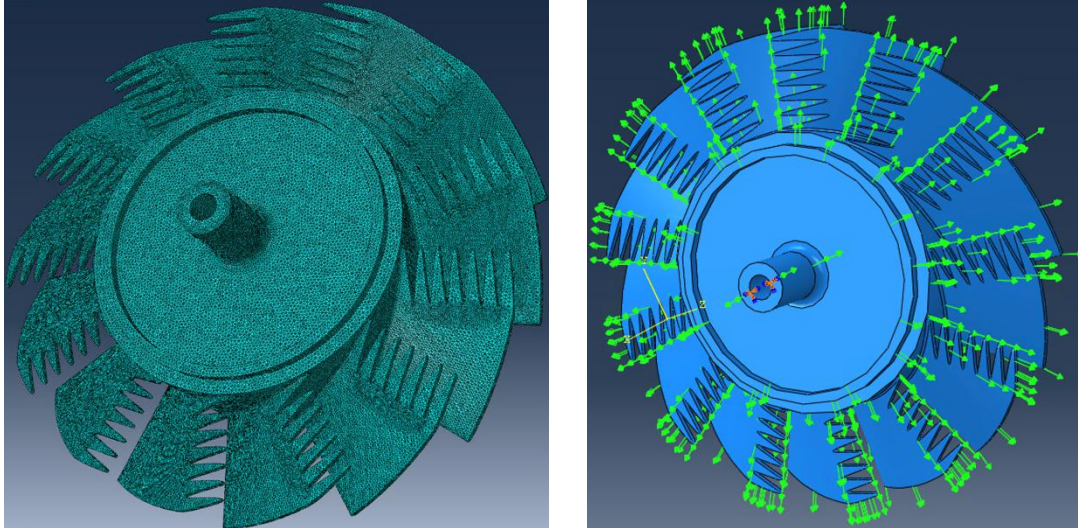


Figure 17 – Fine mesh (left) and boundary conditions (right) for the 10mm amplitude serrated leading edge fan design. Both images are taken from ABAQUS.

The total deformation was plotted, with the maximum deformation occurring at the node represented by the white point in Figure 18. This location is furthest from the axis of rotation, and would therefore experience the greatest centrifugal force. The maximum deformation of 38 microns was very low, and would not be visible to the naked eye. This result was expected from a material with high specific strength.

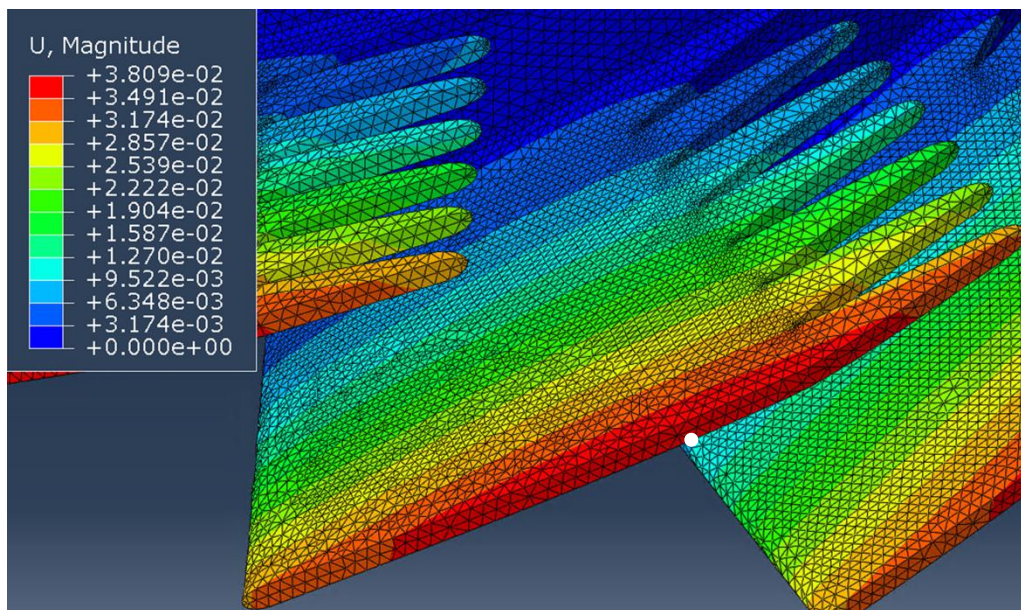


Figure 18 – Total deformation on the 5 mm amplitude ($2h = 10\text{ mm}$) leading edge serrations fan design.

The 250 MPa yield strength (stress) of 6082 T6 Aluminium represents the maximum stress before the material begins to yield, permanently deform, or fracture. The Von Mises stress plot in figure 19 below revealed that at 150% of the maximum centrifugal force exerted on the fan, the yield strength would not be exceeded. The maximum value of 197 MPa

occurred at the root of the serration closest to the hub. The stress at each subsequent root location decreased towards the tip. This result was expected given that material towards the tip is supported by material towards the hub. The root of each serration was also expected to have the highest stress as the lines of force passing from the leading edge towards the rest of the blade would concentrate around these tight corners.

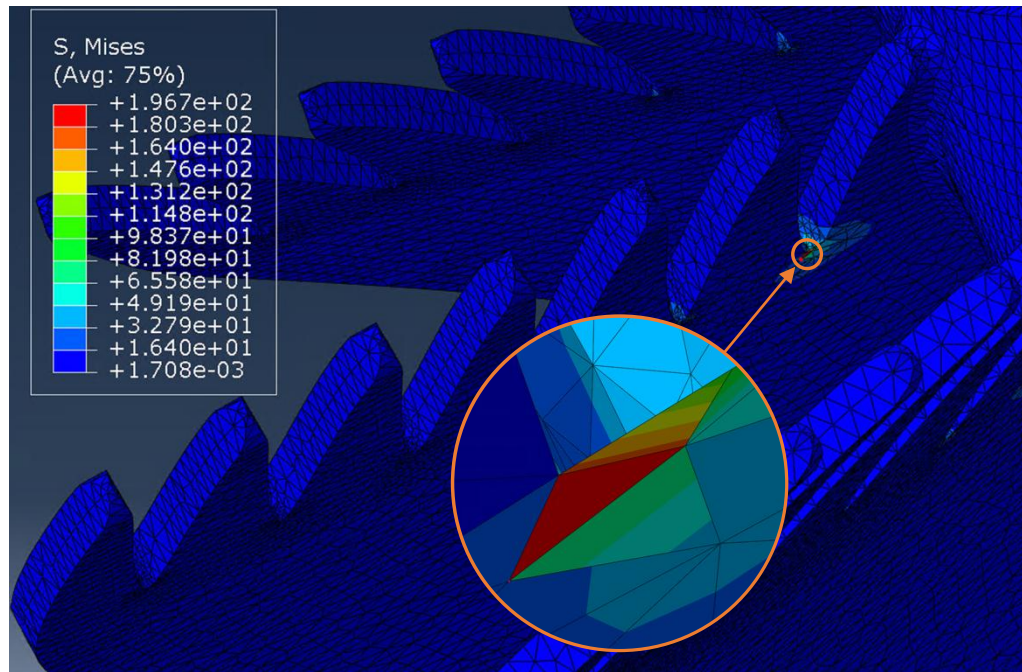


Figure 19 - Von Mises stress on the 10 mm Amplitude leading edge serrations fan design.

4.6 Manufactured OGVs and fan

The initial 5 OGVs were printed, and can be seen below in Figure 20. The Serra-B and Slits-A design (black) were printed via FFF 3D printing, while the porous design, the Serra-A and Serra-C design were printed via SLA 3D printing. All designs needed sanding to ensure that the fit between the OGV and the metal ring (See figure 6), and the fit between the OGV and the motor was tight and coaxial.



Figure 20 - Top left: Serra-A, top middle: Serra-B, top right: Serra-C, bottom left: Porous-A, bottom middle Slits-A, bottom right – original OGV from Dyson

Below in Figure 21 is the completed trailing edge fan design. The acoustic data regarding this fan was generally positive, however in making the two initial fans, four restarts had to be initiated due to machine errors or broken cutters leading to asymmetry in the fan. Fans with asymmetries would cause the fan to vibrate unpredictably, and had to be discarded.



Figure 21 – The manufactured fan with a trailing edge serration, clamped into the 5-axis CNC.

5. Results

5.1 OGV

5.1.1 Acoustic Data

The data recorded by the 8 upstream and 8 downstream microphones were read and analysed using matlab. The Acoustic data from the OGV tests generally showed limited to no change in SPL levels across all frequencies for both the upstream and downstream microphones. Figure 22 below shows the SPL at the operating point with open throttle. All SPL plots for the OGV had very few tones as they lack distinct peaks.

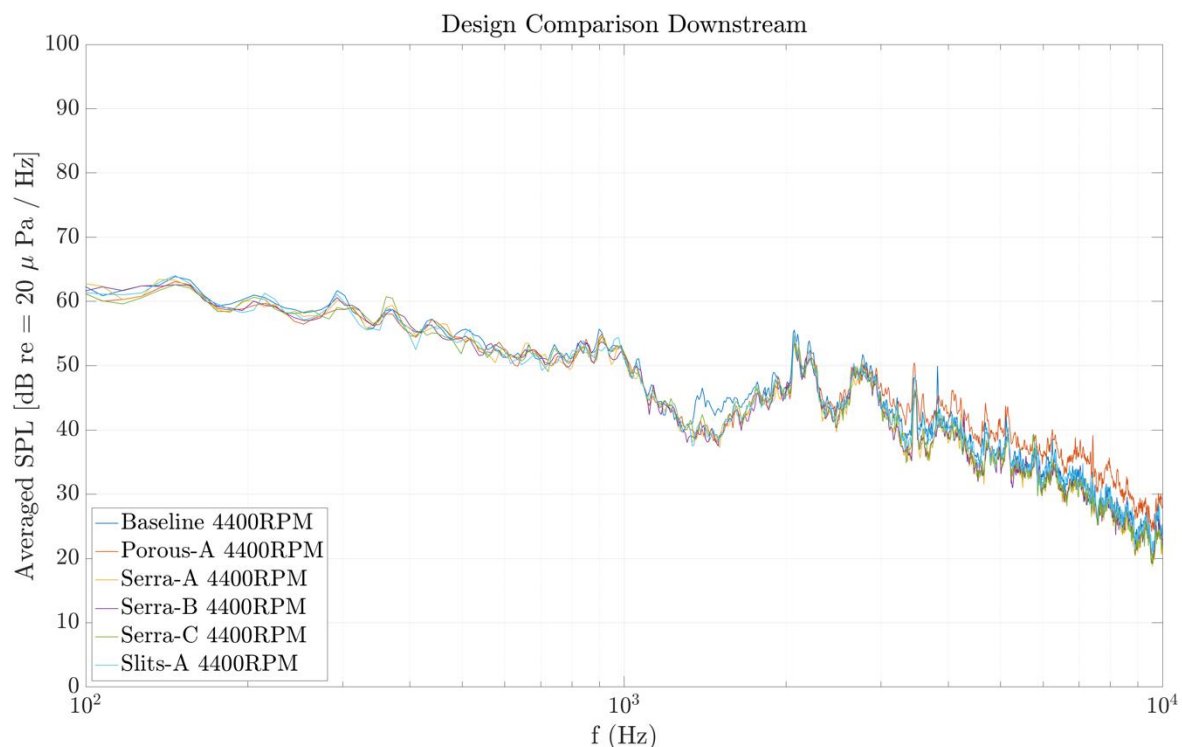


Figure 22 – Averaged SPL across all downstream microphones at 4400 RPM open throttle for each OGV

As it can be seen above, there is a very narrow region between $1.4 * 10^3 \text{ Hz}$ and $1.7 * 10^3 \text{ Hz}$ where the average difference in SPL for each of the modified OGV designs and the baseline OGV is 3.1dB. Beyond this range of frequencies however, the broadband noise appear to be very similar for all OGVs tested, including the baseline. While not shown here, the plots for all other RPM values also show limited noise reduction apart from that in the same narrow region above. At Frequencies $2 * 10^3 \text{ Hz}$ and above, an increase in SPL can be seen for the Porous-A design. Regarding upstream microphones, there is no difference in SPL at any frequency for all RPM speeds and pressure differentials.

Figure 23 below shows the plot comparing the SPL for all OGVs at the operating point and at constant pressure differential 106.25 dP, which was achieved by adjusting the throttle at the back of the Dyson rig.

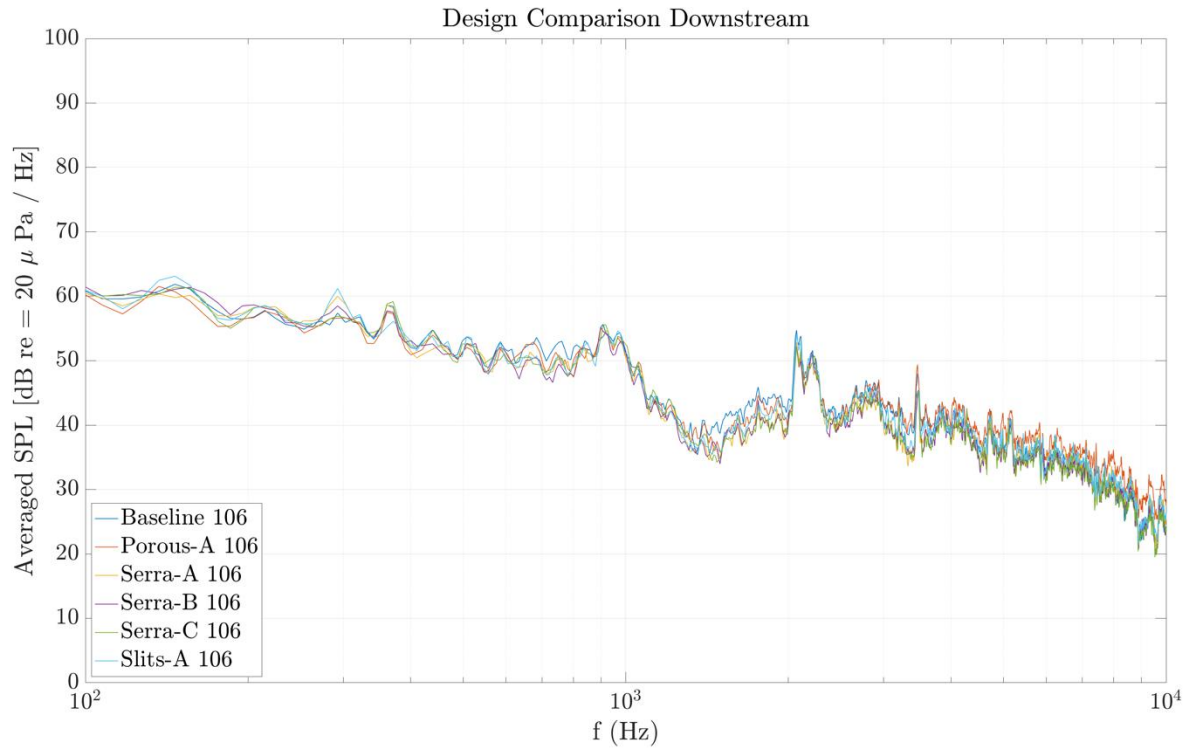


Figure 23 – Averaged SPL across all downstream microphones at 4400 RPM open throttle for each OGV

The SPL graph shows a limited reduction in the SPL across the same range of frequencies, and no reduction for lower and higher frequencies for all OGVs. Similarly, at frequencies greater than $2 \times 10^3 \text{ Hz}$ there is a visible increase in the SPL for the (fully) porous aerofoil.

5.1.2 Aerodynamic Data

While both the fan and OGV are located between the inlet orifice and downstream pressure sensors, the change in pressure differential can be assumed to be the result of changing the OGV only, if the baseline fan is used in all measurements. The same assumption is made when comparing fan designs by using the same baseline OGV in all measurements.

The flow speed shown on the manometer was recorded for the various RPM at open throttle and are shown below in Table 2.

Table 2: Fluid velocity V_z for each OGV

Fan Speed (RPM)	Fluid velocity $V_z \text{ (ms}^{-1}\text{)}$					
	Baseline	Serra-A	Serra-B	Serra-C	Slits-A	Porous-A
2000	3.90	3.94	3.92	3.94	3.90	3.74
3000	6.05	6.06	6.11	6.10	6.02	5.76
4000	8.19	8.27	8.23	8.25	8.17	7.87
4400	9.07	9.16	9.13	9.13	9.01	8.71
5000	10.34	10.43	10.44	10.41	10.29	9.93
6000	12.52	12.62	12.62	12.60	12.46	12.00
7000	14.69	14.81	14.80	14.77	14.62	14.07

The serrated leading edge OGVs had an unexpected improvement in aerodynamic performance. Each serrated design generally increased the pressure difference between the inlet orifice and the downstream pressure gauge (downstream of the OGV) with increasing RPM, see Figure 24 below. The Serra-A OGV with the shortest value for λ had the highest increase in dP at 7000 RPM (2.1 dP).

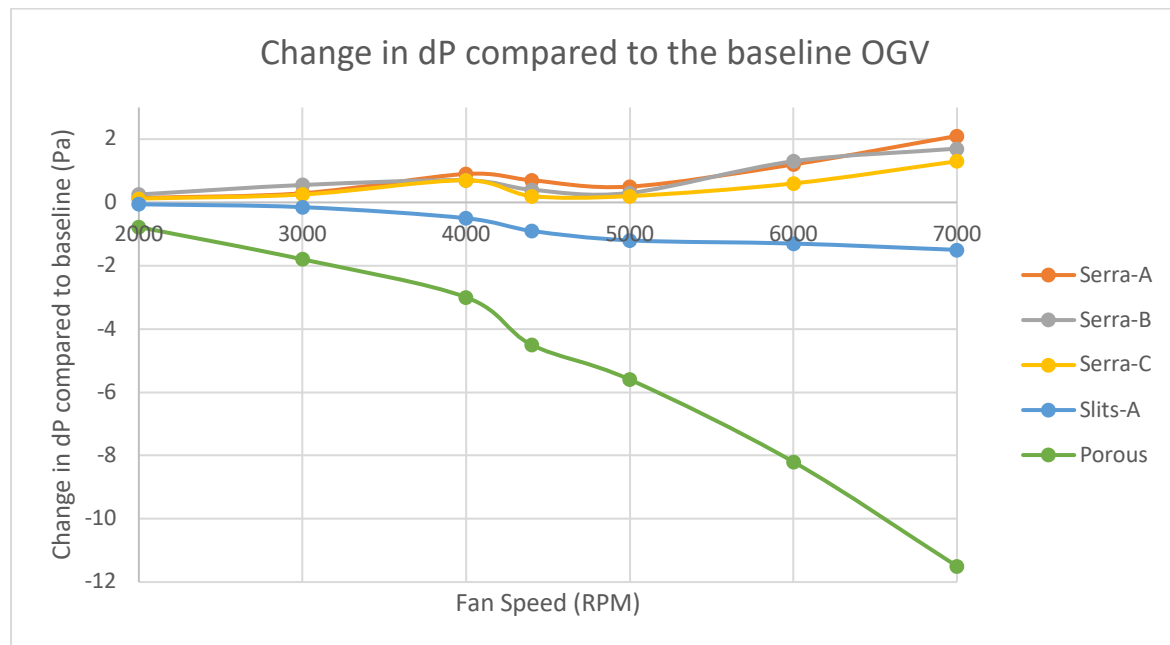


Figure 24 - Change in dP compared to the baseline dP for each OGV

The fully porous design had a negative impact on aerodynamic performance, and decreased in aerodynamic performance as the rotation speed increased. The slitted design marginally decreased the pressure differential. A noticeable peak was observed around the 4000 RPM mark, suggesting that the OGV may perform best at this point, rather than 4400 RPM, at which the change in dP drop for all OGVs.

5.2 Fan

5.2.1 Acoustic Data

All microphones were calibrated, and the connections between the microphones, the amplifier, the NI Interface, and the computer were checked several times to ensure that acoustic data was collected accurately. However, microphone 8 was providing a flat signal, and although each of the cables that led from the microphone 8 slot to the computer were changed several times, the issue was not solved. The issue was likely an issue on the microphone 8 slot on the amplifier, as the microphone 8 signal was present when the slot on the amplifier was swapped with another. Using this slot however meant the loss of data from another microphone. The data from microphone 8 is therefore absent in all tests. Microphone 1 provided an exceptionally high SPL value around the 10^1 Hz region, which dropped to a very low value around the $1.1 \times 10^1 \text{ Hz}$ region. The graph also included several extended and unusual linear sections from 10^1 Hz to the $4 \times 10^1 \text{ Hz}$ region. However, the SPL value for frequencies greater than $1.2 \times 10^2 \text{ Hz}$ were similar to that of

other microphones, and it was concluded that only the SPL values between 10^1Hz and the $1.2 * 10^2\text{Hz}$ region were not reliable. The SPL for microphone 4 were similar to that of other microphones apart from a region between $1.2 * 10^2\text{Hz}$ and $1.4 * 10^2\text{Hz}$, however as the difference was small, the data was not discarded. The rig is symmetrical about the centre line of the pipe, however the requirement for power to reach the motor necessitates a power cable that enters the pipe upstream from the downstream microphone 1 location. This wire is the only source of asymmetry, and its effects were considered when assessing the reliability of the data. The significant deviations in the SPL for microphone 1 were too large to be a result of the power cable.

Using the BPF formula, the BPF for both the fan and the OGV were calculated for each rotation speed, and plotted below in figure 25. The BPF is circled on the subsequent SPL graphs.

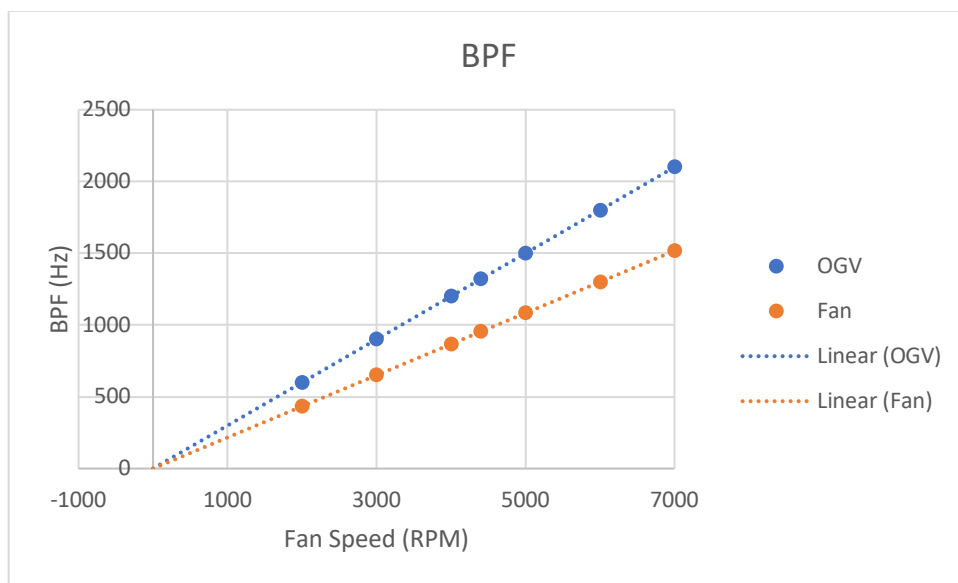


Figure 25 – The BPF for the Fan and the OGV

Regarding both the upstream and downstream microphones in the dP tests, the difference in SPL between the baseline fan and the serrated trailing edge fan lowered as the pressure differential increased. The most noticeable difference between the baseline and the serrated design can be seen in the 50dP downstream plot comparison below in Figure 26.

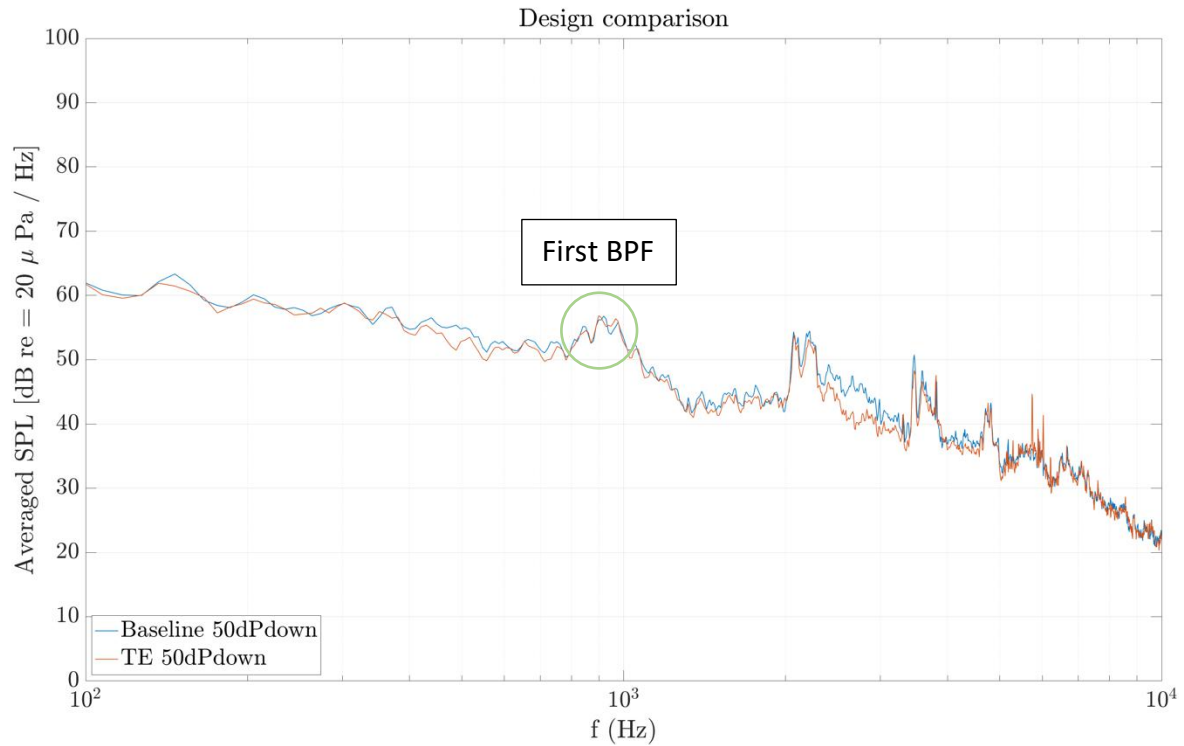


Figure 26: Averaged SPL plot of the two fans at 50dP and 4400 RPM

Noise reduction occurred most significantly in the $2.4 * 10^3 \text{ Hz}$ to $3.4 * 10^3 \text{ Hz}$ region, however, there is a minor region of noise reduction between $3 * 10^2 \text{ Hz}$ and $5 * 10^2 \text{ Hz}$. The noise reduction occurred at similar frequencies for other pressure differentials, however as previously mentioned, the noise reduction reduces with increased dP.

The SPL plot for 68.75 dP is shown below in Figure 27. Here, the noise reduction at the lower frequencies is the most significant in all tests.

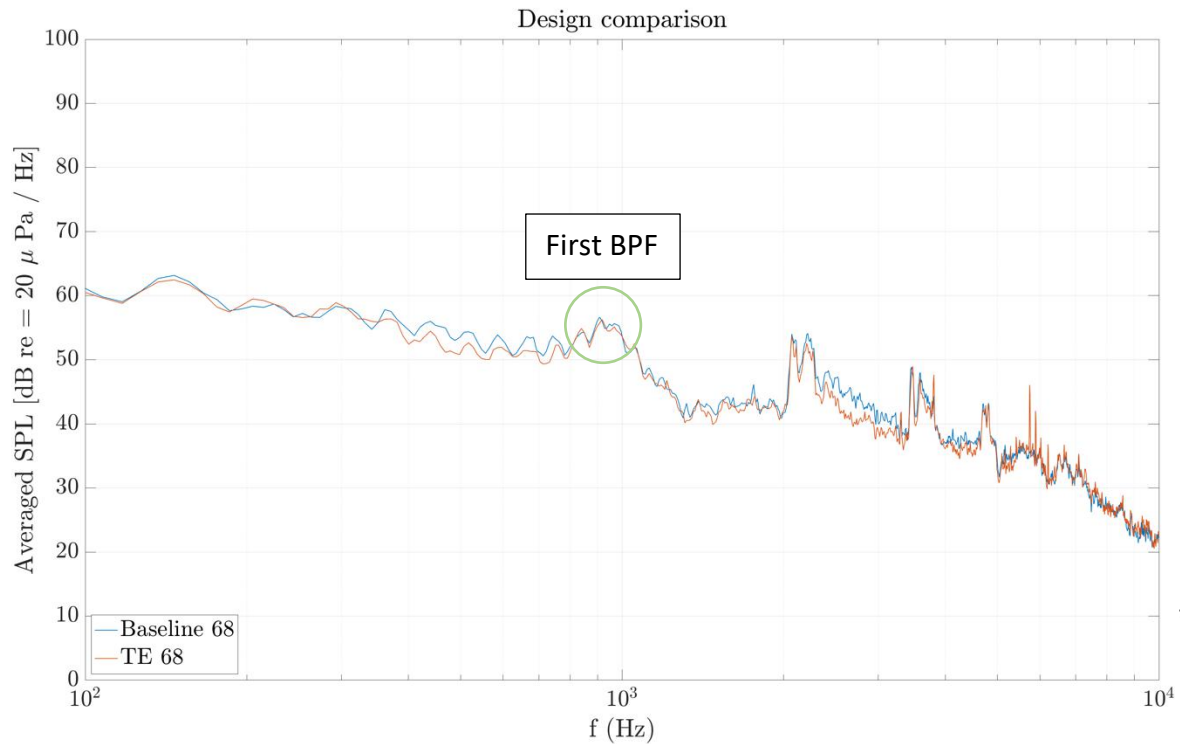


Figure 27: Averaged SPL plot of the two fans at 68.75dP and 4400 RPM

As seen in Figure 28 below, the level of noise reduction between the $13 \times 10^3 \text{ Hz}$ to the $24 \times 10^3 \text{ Hz}$ region is an average of 3.1 dB. As the dB scale is logarithmic, this reduction corresponds to a 50% noise reduction for those frequencies.

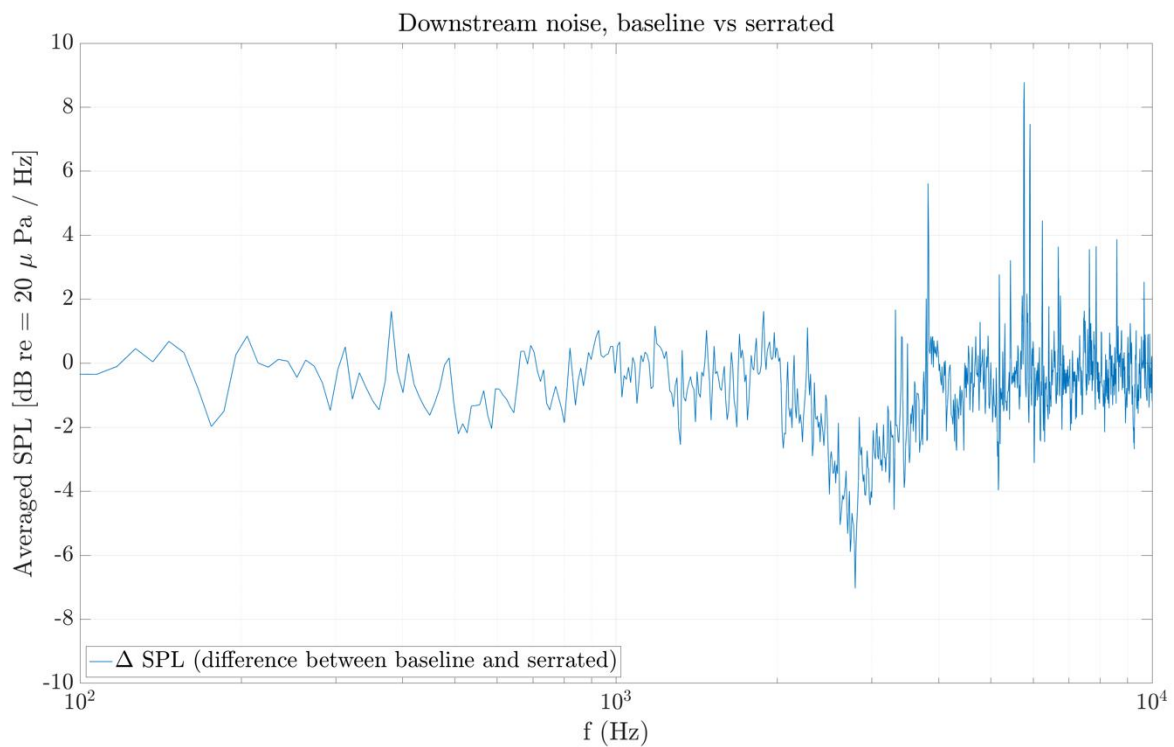


Figure 28: Change in SPL between the two fans at the operating point at open throttle

5.2.2 Aerodynamic Data

The flow speed recorded in the experiments is shown below in Table 3, with the % difference from the baseline to the serrated design in the right column. Overall, the serrated design reduced the aerodynamic performance. the reduction in velocity ranged from -2.98% to 3.62%. This value is low however its effect would be noticeable in terms of suction power for Dyson vacuum cleaners or jet velocity for Dyson hand dryers. To achieve the same axial flow with the quieter serrated fan, more power will be needed for a higher RPM.

Table 3: Axial flow velocities for each fan at various fan speeds, with open throttle.

Fan Speed (RPM)	Axial Flow Speed $V_z \text{ ms}^{-1}$		% Difference from baseline
	Baseline fan	Serrated Trailing Edge fan	
2000	3.02	2.93	-2.98
3000	4.76	4.59	-3.57
4000	6.49	6.3	-2.93
4400	7.21	6.96	-3.47
5000	8.28	7.98	-3.62
6000	10.06	9.71	-3.48
7000	11.84	11.48	-3.04

6. Conclusion

Regarding the modified OGV designs, a limited reduction in SPL was observed in the frequencies between $1.4 * 10^3 \text{ Hz}$ and $1.7 * 10^3 \text{ Hz}$ where a 3.1dB drop was present. However, for the remaining frequencies for all OGV designs across all RPM speeds with open throttle, and all pressure differentials with a fixed 4400 RPM, there was no noise reduction. The primary objective to reduce the broadband noise through modifications to the OGV design was not reached in this study. Aerodynamic OGV data indicated that all serrated OGV increased the downstream and inlet orifice pressure difference compared to that of the baseline OGV, with the shortest wavelength $\lambda = 3 \text{ mm}$ showing the greatest increase of 2.1dP. While the primary aim of the design modifications was to reduce broadband noise, the aerodynamic improvement in the serrated designs is of value, as there is no apparent trade-off in terms of the noise produced, or mechanical strength. The porous design produces more noise and was much less efficient aerodynamically. The slitted design produces negligible effect on noise and aerodynamic performance (both slightly negative).

The lack of effect from the changes to the OGV design may suggest that the OGV is not the dominant source of broadband interaction noise, perhaps the fan instead is the main source. – the main source is most likely the fan, (and partially the motor)

Because dB is a logarithmic scale, a reduction in SPL observed in the trailing edge fan design of 3dB is corresponds to a 50% reduction in sound power. Therefore, the reductions seen in 4400 RPM open throttle correspond to a 50% drop in SPL.

7. Limitations

The use of different materials in the OGV was likely to affect surface smoothness. The designs printed using fused filament fabrication (FFF) 3D printing had a rough surface that resulted from the structure being built upon layers of narrow tubes. OGV designs printed using Stereolithography (SLA) were much smoother, with the surface similar to that of a rubber. While using one method of manufacturing for all OGV would create a fair test, the change to SLA was necessary as sanding was required to fit the OGV in the metal ring tightly, and the OGV could not be printed exactly to the dimensions of the ring due to the ± 0.5 mm tolerance of the printer which could cause the OGV to rattle in the metal ring. FFF printed parts were tough but brittle, and when clamped on the sanding belt, the risk of cracking was evident. This necessitated manual sanding with a file, and due to the strength of the filament, each design required 10 hours of sanding. The SLA printed parts were soft and could be sanded by machine or manually in two hours.

Excessive vibration can create resonance with the fan, motor, and mounting structure. This noise was noticed at 5000 RPM (with both fan designs), and intensified for even higher speeds. It may be the result of imperfect installation, mechanical imbalances of the fan/motor, mounting structure from manufacturing. It may also be the result of fatigue and wear that has accumulated from years of experiments on the Dyson test rig.

Electric motor noise can arise from mechanical and electrical sources. The mechanical sources include cogging torque, acoustic resonance modes, which is a result of internal air space, and structural vibration modes. The electrical sources include the motor inductance. This could have negatively affected the results of the SPL graphs, as at higher speeds the noise produced by the motor would be stronger [11].

8. Further Study

Fan noise can be adjusted by adjusting blade count, and blade shape optimisation. The motor used in the Dyson rig could also be experimented with, by increasing magnet pole count to reduce cogging torque. [11] Another currently experimental technology is ironless core motors which would eliminate cogging torque. [11] At high RPMs the broadband noise radiated by the motor is significant.

The swirl velocity downstream of the fan decreases in strength due to friction with the duct wall. Another area to research could be in how the distance between the fan and OGV affects the vortex-OGV interaction noise, and the aerodynamic performance of the fan-OGV combination.

Materials can also muffle noise by absorbing sound energy. It may be possible to experiment with different noise absorbing materials such as acoustic fibreglass or rubber [17] for the fan and the OGV, with greater freedom given to the OGV as it is not placed under centrifugal stress.

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