AN INVESTIGATION OF THE EFFECT OF UNEVEN BLADE SPACING ON THE TONAL NOISE GENERATED BY A MIXED FLOW FAN

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SUMMARY

The current study investigates the flow performance and sound quality for a standard and an alternative compressor design for use in a Dyson desk fan. The alternative compressor has an uneven blade spacing to spread the associated energy of a single blade passing frequency to several frequencies in order to improve sound quality. The flow performance and acoustic signature for the two designs are compared both as a standalone compressor and installed in product. A dedicated sound preference model was developed to objectively evaluate the acoustic signature of both designs. This methodology can be used to help optimise compressor design in terms of user preference.

INTRODUCTION

This research was undertaken to improve the acoustic performance of the AM01 Dyson bladeless desk fan during the product development stage. Figure 1 illustrates the patented technology [1]. A mixed flow compressor in the base of the product forces air into the circular 'amplifier' section. A thin stream of air exits the amplifier through a 1mm annular gap and the initial flow rate is amplified by entrainment of the surrounding air.

The mixed flow compressor is the predominant source of noise from the product and the most significant component of this noise is aerodynamic related. Of particular interest to the current study, are the tones induced by the blade passing frequency which can be identified in the acoustic signature of the product.

The generation of tonal noise induced by a rotor has been widely studied [2, 3] and it has been stated that the aerodynamic noise is mostly induced by the wake of each blade interacting with the stator and also the non homogeneity of the inlet flow affecting blade loading.

Human perception of this tonal noise is considered to be undesirable and to this end, its study has been of interest to the automotive and the aircraft industries [4-6]. One proposed solution to this problem and the one investigated in this paper is to unevenly space the blades of the compressor about the centre of rotation, in order to spread the single, large tone induced by the blade passing frequency into several reduced tones to improve overall sound quality. The novelty of the current study was to demonstrate this as a potential solution in the context of a domestic appliance.

The issue of sound quality has been raised for radial fans [4] and axial propellers [5]. In both studies, the overall sound pressure level in dB or dB(A) and the magnitude of the tones were the important metrics used to evaluate sound quality following changes to the blade design. A different method of evaluation is presented in the current study using psychoacoustic metrics as a more thorough and relevant means to describe changes in sound quality.

In the current study, a simple model highlighting the motivation for uneven blade spacing is described. An alternative compressor design is realised and the flow performance and acoustic signature for both compressor designs are measured experimentally as standalone units and as integrated components of AM01. Finally, the sound quality evaluation methodology using psychoacoustic metrics is described and the results of a validated human preference model are presented to allow for an objective comparison of sound quality between the two designs.

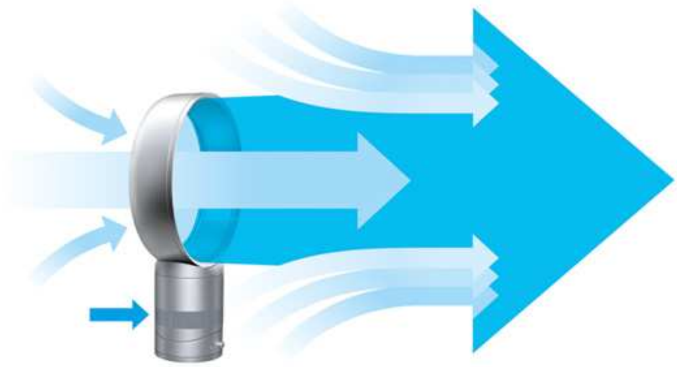

Figure 1: Illustration of the Air Multiplier^{TM} (AM01) technology

EXPERIMENTAL SET-UP

Measuring the performance of the compressor

The compressors were measured using the dedicated set-up shown in Figure 2. In accordance with the in-duct sound power measurement standard [7], two ½ inch microphones (B&K 4189) were fitted in the inlet and outlet measurement ducts. These microphones were fitted with nose cones as turbulent noise suppression devices. Note that sampling tubes were not required as the mean flow velocity was sufficiently low (2 m/s). In order to minimise the turbulent noise at the microphone location, an anti-swirl device and an airfoil shaped microphone holder were used. The anechoic termination performance was checked by determining the sound pressure reflection coefficient. Aerodynamic performance curves and global efficiency curves were measured at constant rotational speed according to the performance standard for fans [8].

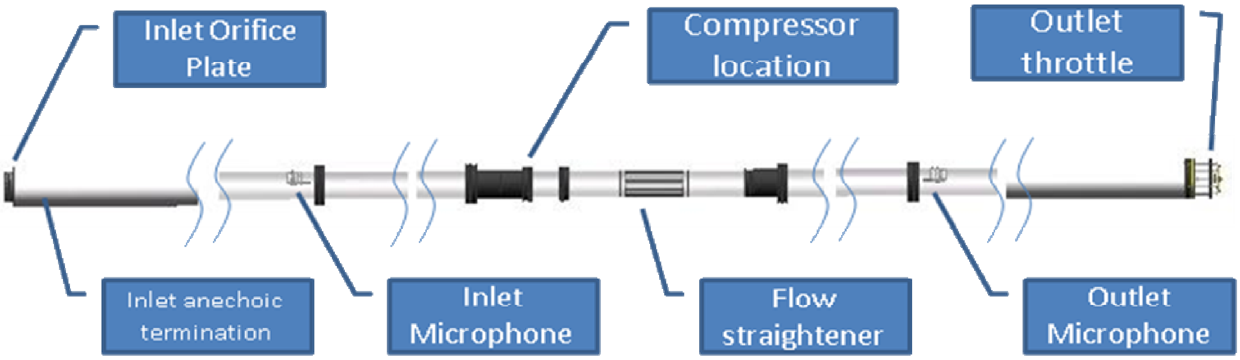

Figure 2: Aero-acoustic set-up

Measuring the performance of the Air Multiplier™ product

Two types of measurement were performed. The first was a standard sound power level measurement taken in a semi-anechoic chamber, as shown in Figure 3(a), in accordance with the standard [9]. Ten microphones were placed in a two meter hemisphere and the desk fan was located at its centre.

The second measurement was used for determining sound quality. The sound of the desk fan was recorded using a binaural recording device worn by an auditor. The desk fan was again placed in the semi anechoic chamber and the listener was positioned relative to the desk fan to represent typical usage, as shown in Figure 3(b).

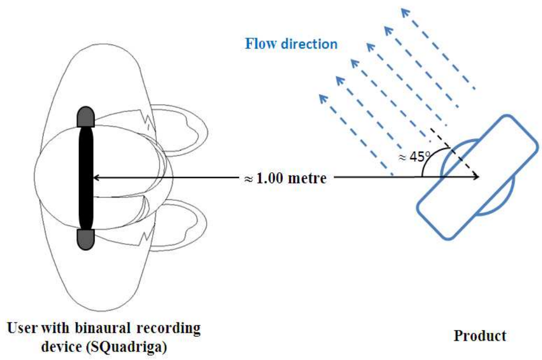

Figure 3: (a) Semi-anechoic chamber *(b) binaural recording set-up*

ALTERNATIVE DESIGN INVESTIGATION

Description of the design

Two different designs of impeller have been investigated. The first design, which is termed 'symmetric' for the purpose of this study, is a mixed flow shrouded impeller with 9 evenly spaced, identical blades. The second impeller, termed 'asymmetric' has identical hub, shroud and blade profiles as the first but the 9 blades are unevenly spaced. The circumferential offset of the blades in the asymmetric design was governed by a sinusoidal variation of up to 8 degrees from the position of the blades in the symmetric design. This design modification kept the centre of mass at the rotational centre of the impeller.

Figure 4 illustrates the difference between the two designs. Measurements using the methodologies outlined in the previous section highlight the difference in flow performance and in acoustic signature of the standalone compressors and also when installed in product.

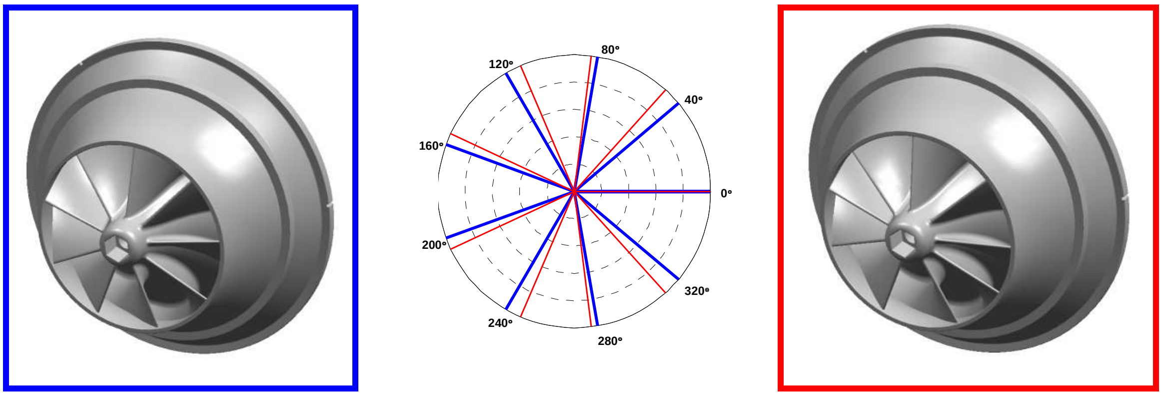

Figure 4: Two different impeller designs, symmetric (blue) and asymmetric (red)

The two designs were prototyped from nylon using a selective laser sintering process (SLS).

Motivation for the asymmetric design

The signature of the blade pass frequency induced by an evenly spaced blade impeller is well understood and can be described as a pure tone. This tone will sit exactly at the frequency f_{BP} as shown in the following equation (Eq.1):

$$f_{Bp} = Nx\frac{\omega_{RPM}}{60} \quad f_{BP} = N \times \frac{\omega_{RPM}}{60} \qquad (1)$$

Where, **N** is the number of blades and ω_{RPM} is the rotational speed expressed in number of revolutions per minute. The time domain signature of a pure tone is a sinusoid and it is possible to generate an altered time signal which takes into account the blade asymmetry, as shown in Figure *5*(a).

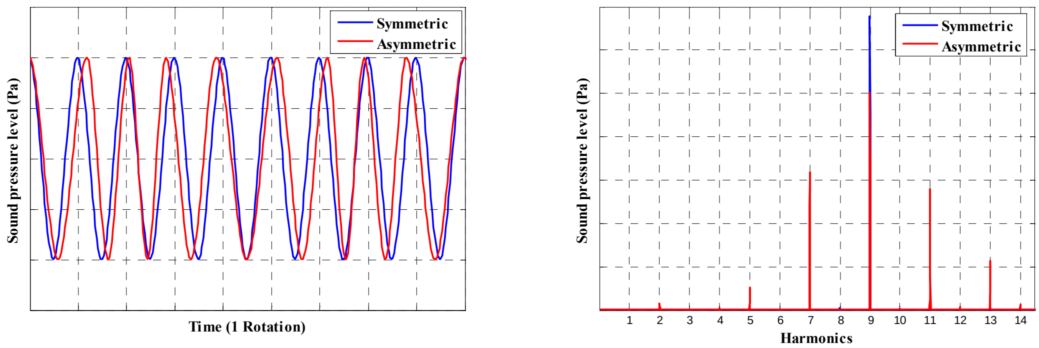

Figure 5: Blade passing induced signal in (a) time domain and (b) frequency domain for a 9 bladed design.

Performing a Fast Fourier Transform (FFT) results in the spectrum presented in the Figure 5(b). From this, the asymmetric design should provide a sound pressure reduction of 25% for the 9th harmonic and may also generate other harmonics including the 2nd, 5th, 7th, 11th, 13th and 14th.

The acoustic signature of both designs can be put simply as follows:

- The symmetric design gives a single strong tone.
- The asymmetric design gives several reduced tones.

Before investigating the users' acoustic preference between the two designs when installed in the product, it was necessary to ensure that the compressor flow characteristics were similar and if the anticipated acoustic signatures could be measured experimentally. Therefore, compressor measurements were carried out prior to product integration.

Compressor measurements

The difference between the two compressors is illustrated by the performance curves presented in Figure 6(a). For the same rotational speed, the total pressure developed ΔP_{Tot} was compared against the flow rate **Q** in litres per second (l/s). For a given flow rate, the asymmetric compressor developed approximately 10 Pa more pressure than the symmetric design.

It is also interesting to define the point of maximum global efficiency for these compressors. Note that the efficiency η can be calculated from the following expression (Eq.2):

$$\eta = \left(Q \times \Delta P_{Tot} \times 10^{-3}\right) / W_{Elect} \qquad \eta = \left(Q \times \Delta P_{Tot} \times 10^{-3}\right) / W_{Elect} \quad (2)$$

Where W_{Elect} is the electrical input power in Watts. From Figure 6(b), the point of maximum global efficiency occurs at approximately 24 l/s for both compressors.

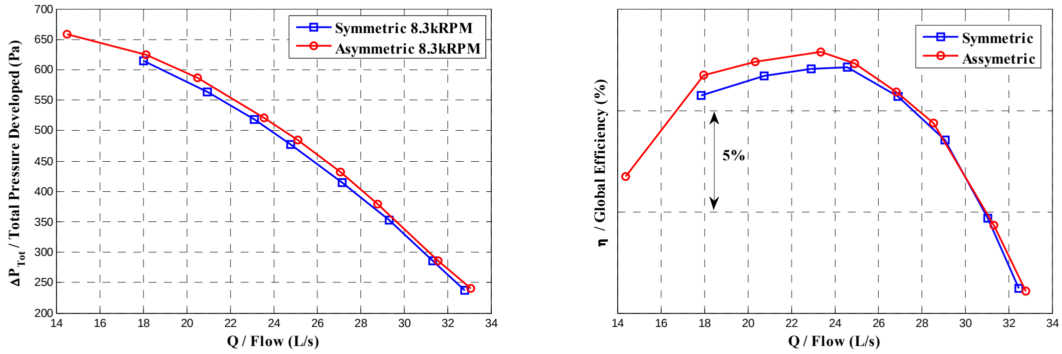

Figure 6: (a) Performance curves and (b) efficiency curves of compressors

Generally, despite these small differences which are likely to be due to hand assembly tolerances, the compressor performance curves are very similar.

The primary goal of the asymmetric design was to improve the acoustic signature of the compressor unit. Initially, the sound power level for both designs was measured. The asymmetric compressor showed a reduction of less than 1 dB(A) in the overall sound power, as shown in Figure 7. Note that the flow rate corresponding to the minimum sound power level is offset from that for the peak global efficiency.

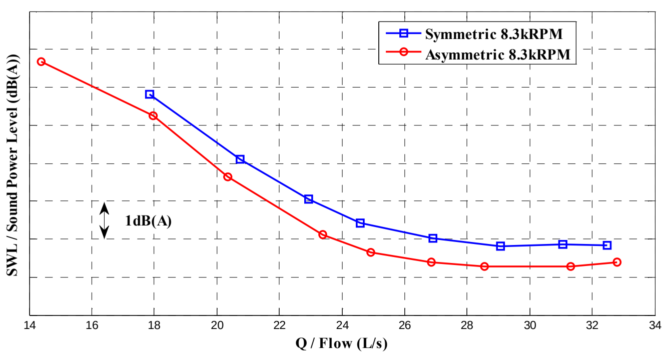

Figure 7: Sound power level of compressors

The aero-acoustic test differentiated between the inlet and the outlet acoustic signatures. However, to evaluate the overall acoustic signature, the inlet and outlet signatures were combined. As the current study was primarily driven by acoustic performance, the measurements were taken at 29l/s corresponding to the minimum sound power level for both compressors. The acoustic spectrums are presented in Figure 8.

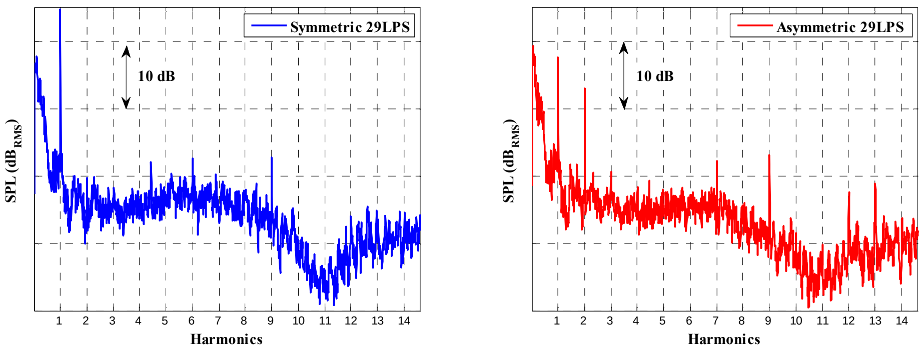

Figure 8: Compressor acoustic signatures of the symmetric design (left) and the asymmetric one (right)

For the asymmetric design, the predicted attenuation of the 9th harmonic is not apparent. As expected, the only harmonic observed from the symmetric design is the 9th, whereas the asymmetric design gives rise to the 2nd, 7th, 9th and 13th harmonics. However, not all of the expected harmonics are apparent, possibly masked by the broad band noise. The 6th harmonic of the symmetric design and the 12th harmonic of the asymmetric design are known to be induced by the dc brushless motor.

For both configurations the fundamental is induced by the imbalance of the impeller. This tone is transmitted through the structure and will be mostly damped in product as the compressor is soft mounted. The next step in the study was to understand the implications of these two designs when installed in product.

In product measurements

Due to motor manufacturing variability, the rotational speeds for the symmetric and asymmetric compressor units were not identical and were measured as 8220rpm and 8340rpm, respectively. The fan law (Eq.3) can be used to calculate the flow rate **Q** and the pressure rise Δ**P** developed by both designs for measured rotational speed ω_2 if their values are known at ω_1. Since the impellers were characterized prior to compressor assembly, corrected values of **Q** and Δ**P** can be determined for the compressors.

$$\frac{\omega_1}{\omega_2} = \frac{Q_{\omega_1}}{Q_{\omega_2}} = \sqrt{\frac{\Delta P_{\omega_1}}{\Delta P_{\omega_2}}} \qquad (3)$$

The Figure 9 shows the corrected performance curves together with the restriction curve of the desk fan amplifier section.

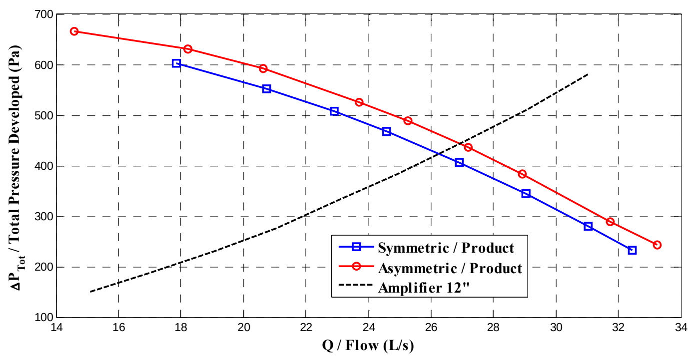

Figure 9: In product performance curve and restriction curve of the amplifier

Assuming negligible losses arising from the product inlet, the flow rates were 26.2L/s and 26.9L/s for the symmetric and asymmetric designs, respectively. As way of confirmation, the same flow rates were also inferred from measurement of the pressure in the amplifier.

The sound power level for the symmetric and asymmetric designs was measured in a semi-anechoic chamber as previously described and the asymmetric design was quieter by 0.6 dB(A).

The acoustic signature for each compressor unit when installed in product was investigated by performing a binaural recording as previously described. Figure 10 shows the acoustic spectrum for the left ear recording for both compressor designs (the same conclusions can be drawn from the right ear).

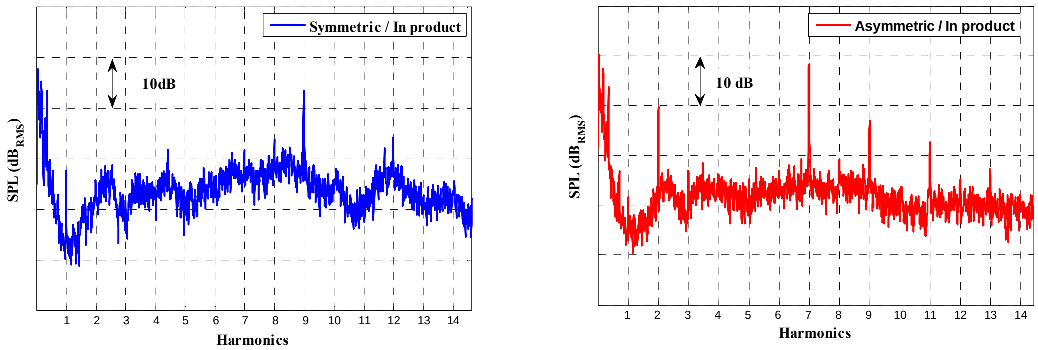

Figure 10: Left ear binaural recording

Firstly, by comparing the signatures of the compressors shown in Figure 8, to those installed in product shown in Figure 10, it is apparent that the fundamental has been significantly damped. Secondly, comparing both designs in Figure 10, the 9th harmonic is 6.5 dB lower for the asymmetric design. Finally, all the expected tones, except the 5th harmonic, are apparent in the asymmetric design signature. It is interesting to note that the order of the tones in terms of magnitude for the asymmetric design installed in product is different for that for the compressor, shown in Figure 8. This difference is likely to be due to the transfer function of the amplifier.

From these results, it is reasonable to conclude that the preferred compressor design would be asymmetric, providing better flow performance and lower overall sound power level. However, a sound quality investigation was necessary to understand the user preference of these different acoustic signatures.

SOUND QUALITY APPROACH

A sound quality investigation was undertaken to establish an objective measure to describe the subjective preference of listeners. The outcome of this investigation provided a dedicated preference model for quantifying the acoustic user preference for desk fans.

A schematic of the method employed is shown in Figure 11.

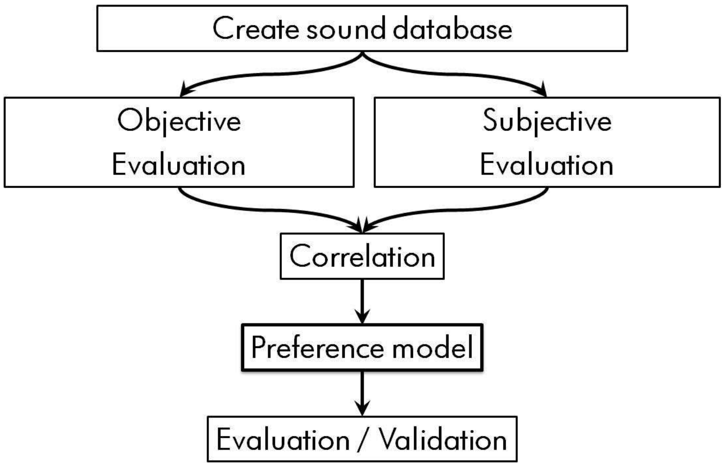

Figure 11: Methodology to define a preference model

28 desk fan sound signatures were generated and objective psychoacoustic metrics computed from this time domain data. A total of 14 metrics were produced including those commonly used such as dB, dB(A), loudness, sharpness, tonality, roughness, and fluctuation strength.

Subjective measurements were elicited by conducting a perceptive test. During a test the listener was played multiple sound recordings and asked to rank them in order of preference. This gave rise to a 'preference index' with a value between 0 and 100 with 0 tending toward the least preference and 100 tending toward the greater preference for a particular sound recording. The perceptive test used to develop the current perception model included 10 sound recordings and 16 listeners. This number of listeners was sufficient for the preference model to converge whereby further listener tests yielded no significant change to the model.

The correlation between the objective sound quality metrics and the preference index was carried out using a least square linear regression method. Regressions with one and two parameters (metrics) were investigated.

The method generated multiple solutions to the preference model dependent upon the number of metrics considered. In order to pick the most relevant solution, three requirements for the solution were defined as follows:

- The coefficient of determination needed to be greater than 90%;
- The preference model needed to be statistically relevant (complying with the T-test and the F-test) [10];
- The model needed to be logical. For example, if the loudness of the recording decreased then it is logical that the preference index should increase.

The preference model which best fit these requirements is defined by the following equation (Eq.4):

$$\textbf{Preference Index} = -\alpha \textbf{ x Loudness} - \beta \textbf{ x Tonality} + \gamma \quad (4)$$

The coefficients α, β and γ are positive constants. The **Loudness** is expressed in "sones" and computed according to the standard [11, 12] and the **Tonality** is expressed in "tu" [13].

The coefficient of determination for this preference model is 98.5% and the correlation between this model and the subjective results is illustrated in the Figure 12(a).

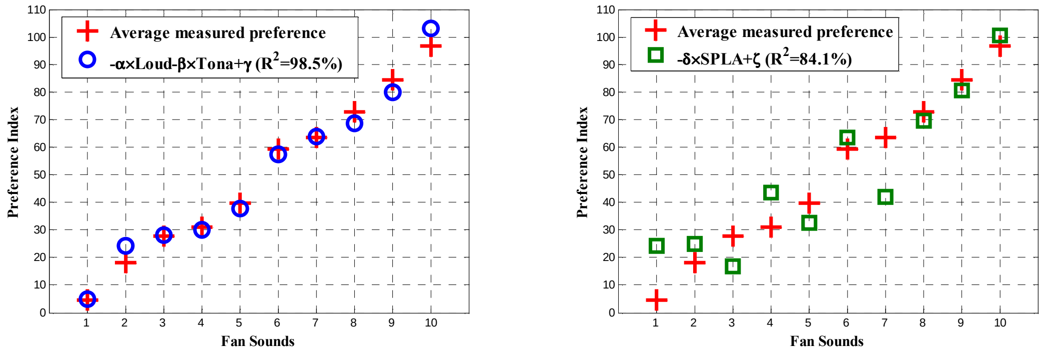

Figure 12: (a) Final preference model and (b) SPLA preference model

For comparison, a simpler preference model using only the A-weighted sound pressure level (**SPLA)** is also presented in Figure 12(b) and the coefficient of determination for this preference model is 84.1%.

To evaluate the model, further perceptive tests were performed using 18 new fan recordings and 52 new listeners. The average correlation of the preference model as described in the equation (Eq.4) with the new subjective results was 86.1% whereas the preference model using only SPLA gave a correlation of 65.2%. From this, it is apparent that the model using loudness and tonality more accurately describes subjective preference than the model relying solely upon SPLA.

Applying the preference model to recordings made of the symmetric compressor installed in product gave a preference index of 20 whilst the asymmetric design in product gave a preference index of 6. Therefore, in terms of user perception, we can speculate that the symmetric design would actually be preferable to the asymmetric design.

CONCLUSION

Blade spacing asymmetry has been previously suggested as a potential solution to reduce the blade passing frequency induced tone in an effort to improve the sound quality of radial and axial compressors [4, 5]. The current study investigated the value of this concept when applied to a Dyson desk fan mixed flow compressor. To this end, the flow performance and the acoustic signature of a symmetric and an asymmetric design were compared as standalone compressor units and also when installed in product.

A dedicated sound quality preference model was defined to objectively evaluate the acoustic signature of both designs. It was interesting to note that, despite having a higher sound power level, the symmetric design gave rise to a higher preference index suggesting that the asymmetric design had made no improvement to sound quality.

The preference model, developed specifically for desk fans in the current study can be used as an important tool to further understand and optimise the asymmetric design concept.

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