ANALYTICAL NOISE MODELLING OF A CENTRIFUGAL FAN VALIDATED BY EXPERIMENTAL DATA

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Abstract

This study focuses on the understanding and modelling of noise radiated from a mine ventilation fan, with the ultimate aim of designing quiet fan impellers. Experimental measurements are presented that suggest that the main noise sources on mine ventilation fans are related to the impeller and consist of tonal components created by the passage of the blades past the volute tongue and broadband noise associated with turbulent flow interacting with the blades. Two models, one analytical and one semi-empirical, have been developed to predict the broadband components of the radiated noise. Given the assumptions made, the computed far field noise compares well with experimental data. The models are therefore good candidates for use in designing new, quieter mine ventilation fan systems in the future.

1. Introduction

Fans are crucial for ventilation in many industries including mining, however the noise that is radiated from these fans can be intense and damage workers' hearing. Centrifugal fan noise is created mainly by the flow through the impeller and fan casing [1]. Tonal noise is generated by the interaction of the impeller blades and volute tongue at the casing outlet, with frequencies generated at the blade pass frequency (BPF) and its harmonics. Broadband noise is generated by the interaction of flow turbulence with the moving impeller blades and casing. Neise [1] provides a summary of common fan noise reduction methods, especially for the tonal component. Broadband noise reduction is more difficult as it is caused predominately by the unsteady lift forces generated by flow turbulence interacting with the leading edges of the impeller blades. Understanding and controlling this broadband noise component is still a challenge for centrifugal fans as well as axial fans and aeroengine fan blades.

There is very little available information regarding analytical modelling of broadband centrifugal fan noise, with the majority related to axial fans. In this paper, two noise models are modified for broadband fan noise prediction; the analytical model of Amiet [2-3] for airfoil turbulent interaction noise and the semi-empirical axial fan noise model of Carolus et al. [4]. The models are compared with experimental noise data obtained from a mine ventilation fan.

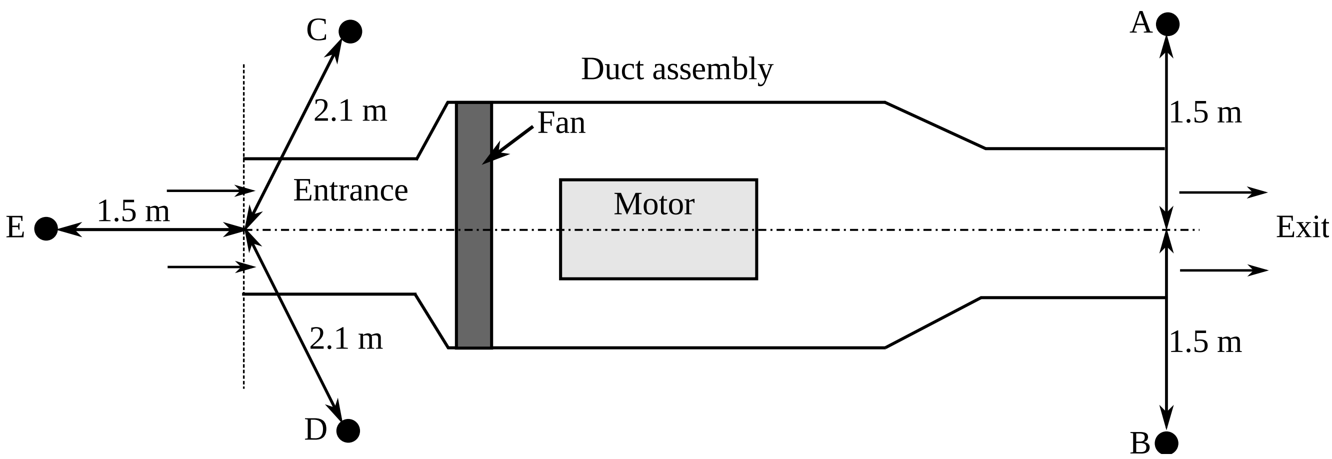
2. Experimental results

2.1 Experimental set up

Noise measurements were performed on Fan #138 at BHP Billiton's Appin coal mine, a nominal 18-cubic-metre-per-second centrifugal fan used for mine ventilation. Testing was performed in the open, on a hard asphalt surface (Fig. 1(a)) with a flow setting of 17.8 cubic metres per second. A large shed and some equipment were located within 10-20 m of the test site. The weather was pleasant and not particularly windy. Wind shields were used to reduce contamination of the acoustic signals by microphone self-noise created by the presence of wind. Testing was performed using two Gras 1/4 inch 40PH precision array microphones that were logged simultaneously using a National Instruments 24 bit CompactDAQ data acquisition system. Data were acquired using a sampling rate of 51.2 kHz. Each data record was 60 seconds in length.

Figure 1(b) provides a schematic diagram of the test setup. The microphones were placed in the following locations: A, B aligned to the exit and on each side of the duct assembly, located 1.5 m from its centre; C, D aligned to the fan, again on each side of the duct assembly, located 1.5 m from its centre; E located in front of the entrance and aligned to the centre of the duct assembly.

(a) Photograph of experiment



(b) Schematic diagram showing microphone positions

Figure 1. Experimental set up for Fan #138

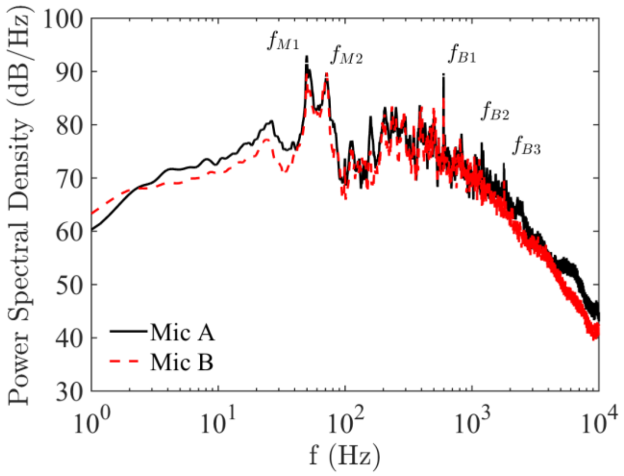
2.2 Noise spectra

The power spectral density measured at each microphone is presented in figure 2. The frequency resolution of all power spectral densities is 0.8 Hz. Major tones are labelled, which are associated with tones created by machinery noise (f_{M1}, f_{M2}) and the motion of the blades past the volute tongue (the BPF and its harmonics, f_{B1}, f_{B2}, f_{B3}). These frequencies as well as their nature are summarised in Table 1, and the reasoning behind these classifications is below.

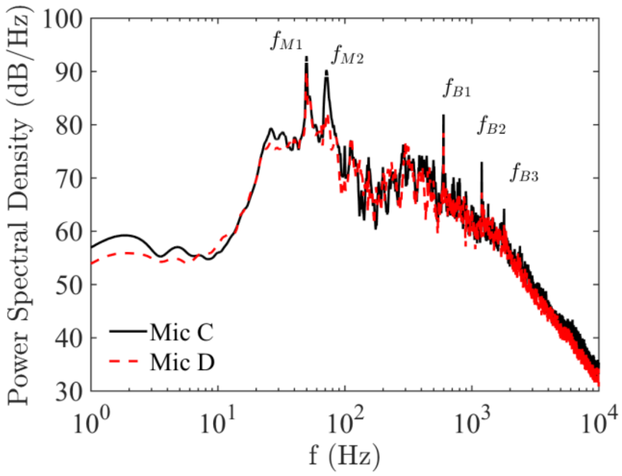
The electric motor turns at 2975 rpm, corresponding to a rotational frequency of 49.6 Hz, which is clearly visible in the spectra. Hence, the peaks at 49.8 Hz and 72 Hz are most likely due to machinery noise (bearings, etc). The impeller itself induces peaks that occur at frequencies 595, 1190 and 1785 Hz. This is confirmed by a calculation of the Blade Passing Frequency (BPF) and its harmonics f_{Bk} , k = 1, 2, 3 defined by

$$f_{Bk} = k\, N\, \Omega \tag{1}$$

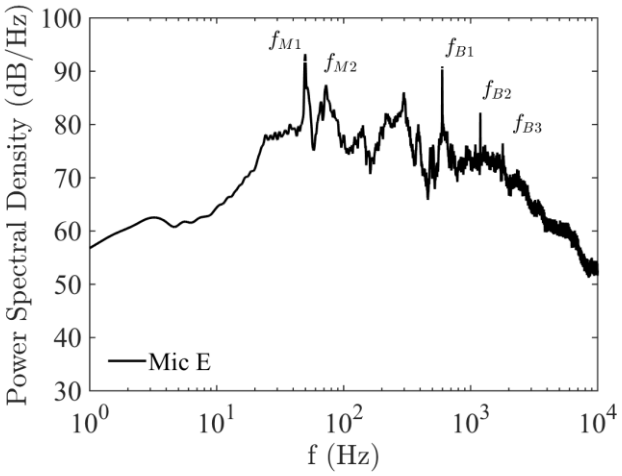
where Ω is the rotational frequency (49.6 Hz) and N is the number of rotor blades (12). Note that the fundamental frequency is for k = 1.



(a) Microphones A and B



(b) Microphones C and D

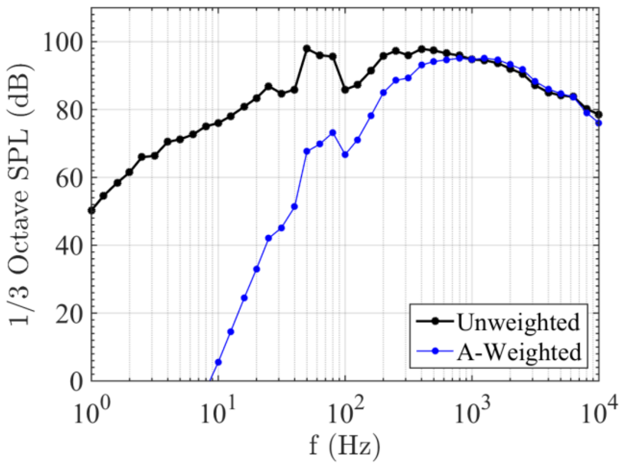


(c) Microphone E

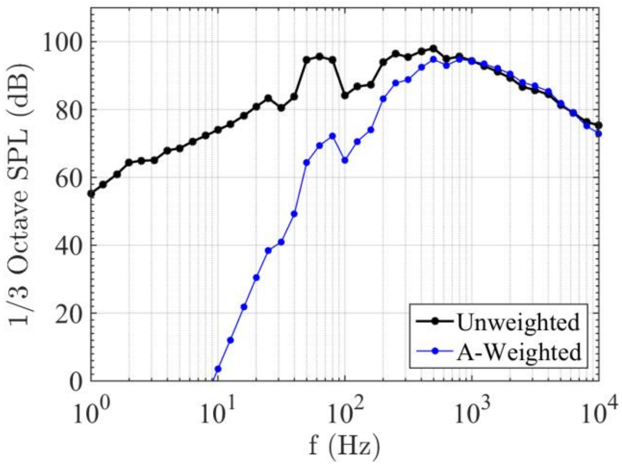
Figure 2. Power spectral densities at each microphone location

Table 1. Peak frequencies for all measurements

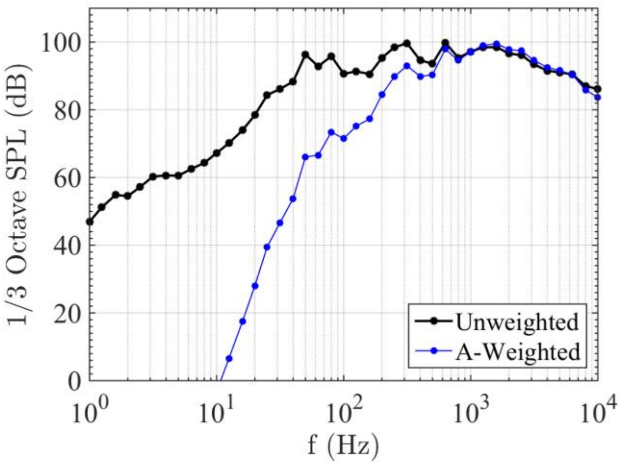
	f_{M1} (Hz)	f_{M2} (Hz)	f_{B1} (Hz)	f_{B2} (Hz)	f_{B3} (Hz)
Microphones A, B (Fig. 2(a))	49.8	71.3	596.7	1194	1790
Microphones C, D (Fig. 2(b))	49.8	71.9	597.3	1195	1791
Microphone E (Fig. 2(c))	49.6	72.9	596.9	1194	1791
Calculated Frequency (Eq. 1) (Hz)	49.6	72	595	1190	1785
Type of noise source	Machinery noise	Machinery noise	Fan BPF k = 1	Fan BPF k = 2	Fan BPF k = 3



(a) Microphone A



(b) Microphone B



(c) Microphone E

Figure 3. 1/3 octave SPL measurements, unweighted and A-weighted

The noise measurements are presented in 1/3 octave bands, both unweighted and A-weighted in figure 3. A-weighting significantly reduces the significance of the machinery noise components, as they are approximately 10-20 dB below the broadband components above 250 Hz. Further, the BPF tones are narrow and consequently do not contain much acoustic energy relative to the broadband components. Hence the tones that appear in the power spectral densities are not present in the 1/3 octave results, which indicates that broadband noise above approximately 250 Hz is the most important noise source from the fan. It is likely that this noise source is due to flow turbulence interacting with the impeller blades.

3. Noise Model

Two models are developed to model the broadband noise generated by the fan: one from the theory of Amiet and one semi-empirical model (SEM). Amiet's theory [2-3] aims to predict the radiated noise of an airfoil due to turbulence interaction with the leading-edge. It is well suited to slender objects such a rotor blade. The far-field noise is obtained from the derivation of the cross-power-spectral density of surface pressure on the airfoil surface to far-field sound by using the techniques of Curle [5].

The axial fan SEM of Carolus *et al.* in [4] has been modified for use with centrifugal fans. Table 2 shows the parameters used for each noise model. For both the Amiet and Carolus models, the turbulence model used was the Karman model [3-4].

Both models are adapted here for the industrial centrifugal fan. For brevity, the detailed mathematical descriptions of these models have been omitted from this paper and can be found in the referenced papers. The Carolus model predicts sound power from an axial fan, while the Amiet model was developed to predict the power spectral density at a point in the free-field about a stationary airfoil. Hence, the Amiet model was modified so that it calculated sound power from 8 radial segments of each blade, via a surface integral in the far field. The Carolus model was unmodified, with the impeller fan blade dimensions used directly.

A free field transfer function, based on a spherical wave exiting the duct, was used between the fan impeller to the observer location, and of r distance, has been considered as analytical and equal to $1/(4\pi r^2)$. Parameters for both models are given in Table 2.

Table 2. Parameter values for both noise models

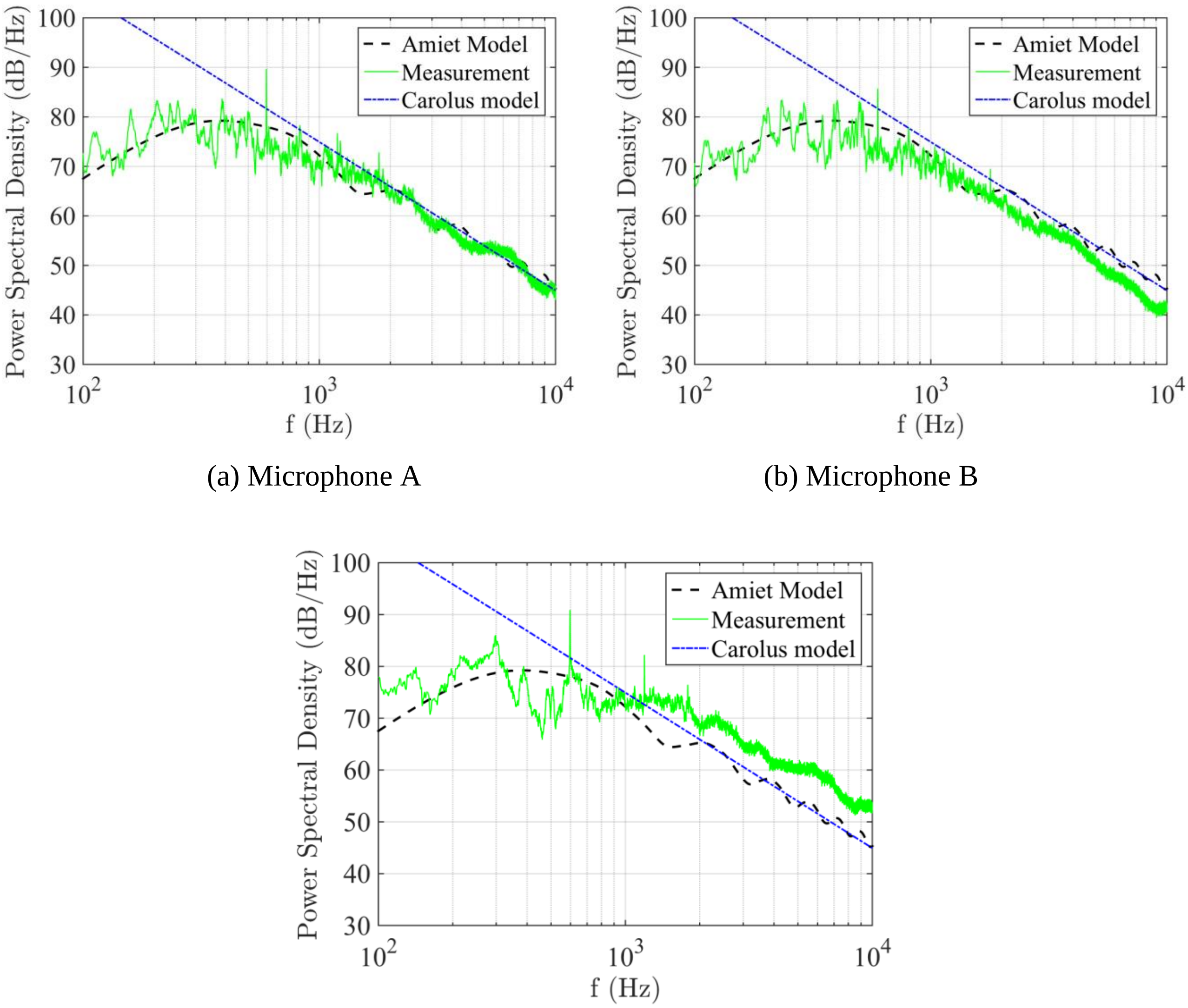
Parameter	Value
Density (kg/m^3)	1.225
Sound velocity (m/s)	340
Rpm motor speed (rpm)	2976
Turbulence Intensity	0.1
Turbulence Length scale (m)	0.025
Mean axial flow velocity (m/s)	5
Integral axial length scale (m)	0.035
Number of Impeller Blades (N)	12

Figure 5 compares the experimental and numerical results for both models. The results are, on the whole, in good agreement. Amiet's model is the most sophisticated as it uses a complex, frequency-dependent turbulence-to-lift-force transfer function that takes into account finite chord-scattering effects. The Carolus model, on the other hand, uses only a relatively simple linear relationship between the fluctuating turbulence and the lift. The result is the Amiet model is able to recover the shape of the broadband spectrum over the entire frequency range of interest. The Carolus model is only able to accurately predict the higher-frequency components of the noise spectrum, due to the assumptions used in modelling the unsteady lift.

6. Conclusions

This paper presents experimental measurements and analytical noise predictions of an industrial centrifugal fan used for mine ventilation. Experiments have been done on a mine site. Machinery noise is responsible for noise at low frequency whereas the aerodynamic noise from the impeller itself creates dominant broadband noise, greater than 250 Hz. Tones created by the passage on the impeller blades past the volute tongue (the BPF and its harmonics) are clearly visible in the power spectral densities; however, they are not apparent in the 1/3 octave results due to their limited amount of acoustic energy. Two models are presented to predict the broadband noise, one from the classical theory of Amiet, and the other the semi-empirical model of Carolus dedicated to axial fans. Good agreement is shown between both models and experimental data that emphasise the usefulness of such analytical models. The Amiet model provides better low-frequency predictions due to its more sophisticated treatment of the unsteady lift.

Both models are considered good candidates for use in designing new, quieter mine ventilation fan systems. Further work is required to better characterise the incoming flow turbulence level and distribution as well as investigating a variety of new, lower noise blade shapes.



(a) Microphone A

(b) Microphone B

(c) Microphone E

Figure 5. Numerical and experimental comparisons

Acknowledgements

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