Reduction of Tonal Noise in a Centrifugal Fan using Guide Vanes

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Abstract

This study investigates the effects of diffuser vane on the aeroacoustics characteristics of a centrifugal fan. It is well known that tonal noise is the dominant noise source generated in the centrifugal fan, due to the aerodynamic interaction between the rotating impeller and the stationary diffuser vane. The tonal noise occurs due to pressure fluctuation that is related to the rotating speed. The tonal noise is periodic in time where it consists of the blade passing frequency (BPF) and its harmonics. Analysis of the experimental and simulation results have shown that the stationary diffuser vane causes the tonal noise and non-rotational turbulent noise generation. However, omitting diffuser vanes leads to the increase of non-rotational turbulent noise resulting from the high velocity of the flow leaving the impeller. Hence in order to reduce the tonal noise and the non-rotational noise, guide vanes were designed to replace the diffuser vanes. The behavior of the fluid flow and acoustics characteristics was studied using computational fluid dynamics (CFD) tools.

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1. Introduction

Noise generation from centrifugal turbomachines has been a major concern to many users. Centrifugal turbomachines such as fans, compressors and turbine commonly generates tonal and broadband noise [1]. In this paper, centrifugal fan found in domestic vacuum cleaners was used for investigation. The centrifugal fan has an impeller, a diffuser, a circular casing and an electric motor as shown in Figure 1. The prominent noise in centrifugal fan is the tonal noise at the blade passing frequency (BPF) and its harmonics. The tonal is period in time which caused by the strong interaction between the impeller outflow and the stationary vane at the volute.

Many works on the centrifugal fan aerodynamic noise have been conducted over the years. One of the critical reviews is given by *Neise* which summarized the efforts to reduce passage tonal noise in centrifugal fan by improving the impeller geometry and the volute cut-off [2]. *Cudina* and *Prezelj* presented experimental work on centrifugal fan from a vacuum cleaner. They showed that the aerodynamic, mechanical, and electromagnetic noise emitted by vacuum cleaner is lowest at the best efficiency point (BEP) of the centrifugal fan [3]. *Neise* and *Koopmann* designed an acoustic resonator that can be used to reduce at the source the aerodynamic noise generated by turbomachinery [4]. Research conducted by *Choi* was focused on identifying the aspects of fluid dynamics that are associated with the noise generations in centrifugal turbomachinery [5]. Tapered trailing edge of impeller was proposed by *Jeon* which is to decrease the pressure fluctuation during the interaction between the impeller and the diffuser, thus reducing the tonal noise [6]. *Khelladi et al* presented numerical work on prediction of fluctuating pressure using Ffowcs Williams and Hawkings (FW–H) equation. They concluded that the interaction between the impeller-diffuser is the origin of the tonal noise [7].

All the previous studies have well contributed to the tonal noise generation mechanism. The objective of this work is to study the effects of replacing the diffuser vane with guide vane on the tonal noise generation. Aerodynamic and acoustics experiments were conducted on the baseline centrifugal fan with diffuser vane. Then computational fluid dynamics (CFD) simulation CFD which was validated through the experiments was used to predict the tonal noise generation of the guide vane.

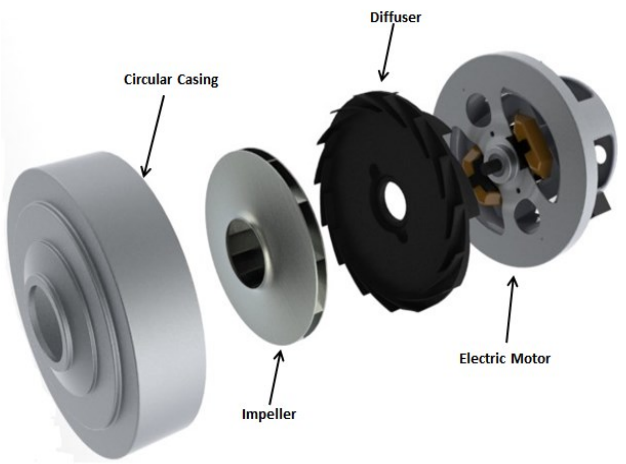


Figure 1: Exploded figure of the parts from centrifugal fan

Table 1: Aerodynamics characteristics at operating conditions

Descriptions	Value
Rotational Speed	34560 (rpm)
Flow rate	0.0479 (m^3/s)

Table 2: Geometrical characteristics of the centrifugal fan

Description	Impeller	Diffuser
Radius of blade inlet (mm)	18	53
Inlet blade angle (deg)	9.2	2.8
Radius of blade exit (mm)	51.5	62.5
Angle of blade exit (deg)	32.18	19.8
Blade number	11	15

2. Experiment Methodology

2.1 Aerodynamic Flow

In this investigation, a test bench was built to measure the aerodynamic performance of the centrifugal fan. The schematic of this test bench is shown in Figure. 2. The difference of static pressure across the orifice plate was used to obtain the flow rate through the centrifugal fan. The orifice plate was designed based on ISO 5762-2 [8]. Pressure sensors are flushed mounted at the impeller exit and diffuser exit. These pressure values will be used to validate the numerical model. The impeller rotational speed which corresponds to the universal motor rotational speed was measured by using inductive proximity sensor. Only one operating speed is considered since the centrifugal fan is from fixed speed vacuum cleaner. The vacuum cleaner uses cyclone system where the dirt particles and air particles are separated due to the centrifugal and gravitational force. Hence, the centrifugal fan constantly operates at its best efficiency point (BEP). The following uncertainties were established for the measured and calculated parameters:

I. U-Tube Manometer: ± 19.6 Pa
II. Pressure Transducer: ±1723.7 Pa
III. Inductive Proximity Sensor: ±5%
IV. K-type Thermocouple: ±2.2°C

2.2 Acoustics Measurement

The acoustic measurement was conducted in an anechoic chamber which complies with the ISO 3745 standard [9]. The floor dimension of the chamber is 4.1m × 5.2m × 3.1m and the cut-off frequency is 230Hz. For the case of interest, hemi sphere measurement surface with radius of 1m was used. The sound pressure level was measured using Gras Free Field ½" Preamplifier microphone. The microphone is calibrated using the B&K 4231 to ensure the sensitivity and accuracy is within the acceptable range. These are the uncertainties for the acoustics devices:

I. Gras ½" Free Field Pre-amplifier microphone: ± 1dB (5Hz to 10kHz) and ± 2dB (3.15Hz to 20kHz)
II. B & K sound level calibrator: ±0.2dB

The background noise of the semi anechoic chamber is 21.2 dB (A).

NO	PARTS
1	U-Tube Manometer
2-4	Pressure transducer
5	K-Type Thermocouple
6	Orifice Plate
7	Centrifugal Fan
8	Inductive Proximity Sensor

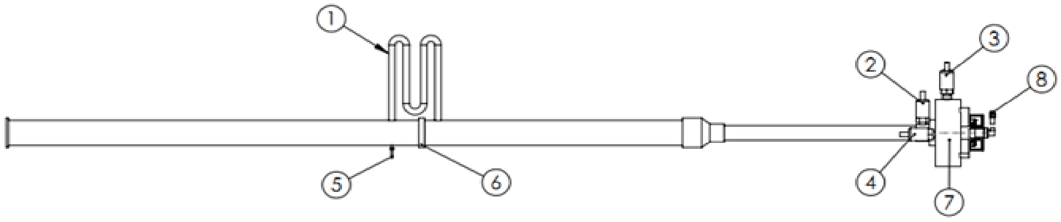


Figure 2: Schematic diagram of flow test bench

The sound pressure level (SPL) of the centrifugal fan is measured at each of the 20 positions as stated in the ISO 3745 and the overall surface-averaged SPL is computed using equation (1):

$$\overline{L_p} = 10\log\left[\frac{1}{N_m}\sum_{1}^{N_m} 10^{0.1\,L_{pi}}\right] \qquad (1)$$

Where,

N_m is the number of microphone positions.

L_{pi} is the background noise corrected sound pressure level at ith microphone positions in decibels.

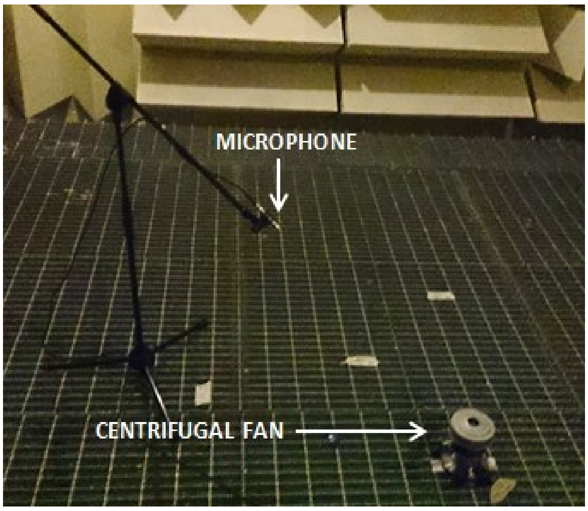


Figure 3: Centrifugal Fan Measured in semi anechoic chamber

3. Numerical Modelling

3.1 Aerodynamic Modelling

In this section the overview of the computational fluid dynamic (CFD) modelling using ANSYS FLUENT is presented. The model consists of 5 fluid domains: the upstream inlet, the gap between casing, the impeller, the volute, and the downstream outlet as presented in Figure 4.

Hybrid meshing with tetrahedral and hexahedral type was used, to cater for the component complexity. The mesh consists of 4.62 million elements. The unsteady term of the conservation equations second order implicit is applied. The turbulence modelling *k-w SST* variant of *Detached Eddy Simulation (DES)* is adopted in this modelling since it is capable in resolving eddies for noise prediction. In addition, DES models have been specifically designed to solve wall bounded flow with high Reynolds [10]. SIMPLE algorithm is used for pressure-velocity coupling and second order upwind for the turbulence [11, 12]

The inlet and outlet boundary conditions are defined as ‘‘pressure inlet’’ and ‘‘pressure outlet’’. Sliding mesh technique in used to simulate the rotation of the impeller, where the inner volume is defined as ‘‘moving mesh’’ and defined as 34560 rpm. The common surfaces in the inner and outer volume families are defined as ‘‘interface’’.This configuration is known as non-conformal interfaces [13, 7].The time step of simulation is 4.8225×10^{-6}s which represents the impeller mesh turns 1° per time step. The CFD model is then validated based on the parameters as listed in equation (2) - (4) obtained from experiments.

$$Flow\ Coefficient, \varphi = \frac{Q}{\pi b_2 d_2 u_2} \qquad (2)$$

$$Total\ Pressure\ Coefficient, \psi = \frac{\Delta P_t}{0.5\ \rho\ u_2{}^2} \qquad (3)$$

$$Total\ to\ total\ pressure\ ratio, \Pi_{t,t} = \frac{P_{t,out}}{P_{t,in}} \qquad (4)$$

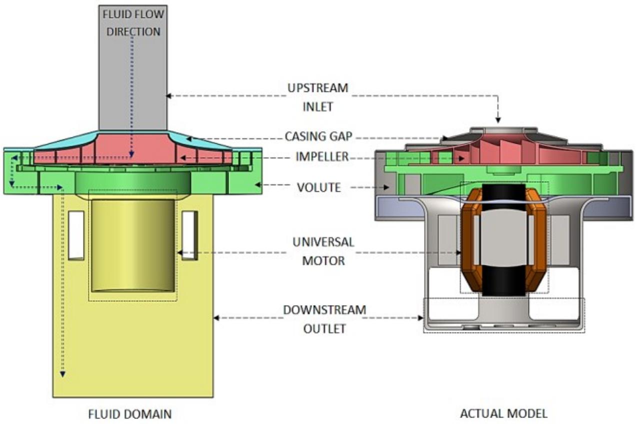


Figure 4: Cross section of fluid domain used for CFD modelling

Table 3 presents the comparison between the measured and predicted results. It's clear that the CFD under-predicts the unsteady performance of the centrifugal fan. This is due to the simplified geometry that is used to represent the universal motor in the CFD modelling. During experiment, the universal motor's commutator rotates to provide momentum to the air flow, but in the CFD modelling it is defined as stationary to reduce the computational time.

Table 3: Comparisons between experiment and simulation for aerodynamic validation

	φ	ψ	$\Pi_{t,t}$
EXP	0.136	0.604	1.129 (Pa)
CFD	0.129	0.594	1.125 (Pa)
$\epsilon_R(\%)$	4.86	1.67	0.35

3.2 Aeroacoustics Modelling

The Ffowc-Williams and Hawking (FW-H) equation was used to predict the rotating machinery noise. This equation is an exact rearrangement of the continuity and the momentum into the form of an inhomogeneous wave equation [14].The FW-H formulation adopts the most general form of Lighthill's acoustic analogy, and it is capable of predicting sound generated by equivalent acoustic sources [15, 10]. The FW-H equation is written as in equation 5. The first term represents the monopole, the second represent the dipole and the third term represents the quadrupole source. [16]

$$\left(\frac{1}{c_0{}^2}\frac{\partial^2}{\partial t^2} - \nabla^2\right)p = \frac{\partial}{\partial t}[\rho v_n \delta(f)\nabla f] - \frac{\partial}{\partial x_i}[n_i P \delta(f)\nabla f] + \frac{\partial^2}{\partial x_i \partial x_j}[T_{ij} H(f)] \qquad (5)$$

The comparison between the predicted SPL and the equivalent value from experiments is shown in Figure 5. The SPL from modelling has similar pattern with the experimental values but there are deviations in their magnitude. These deviations are because in aeroacoustics formulation the noise is assumed to radiate into free field. The effects of scattering, diffractions, and reflections of casing are not considered. [7, 17]

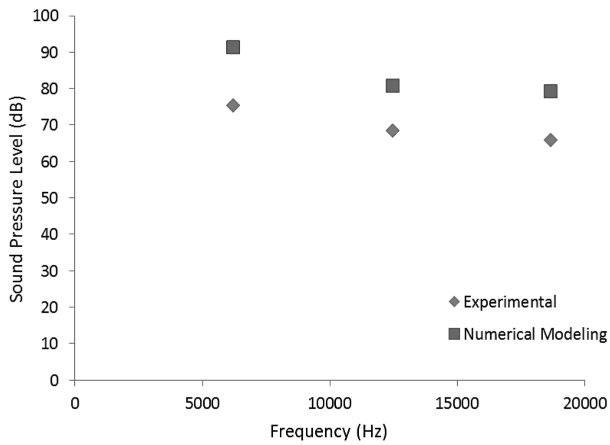
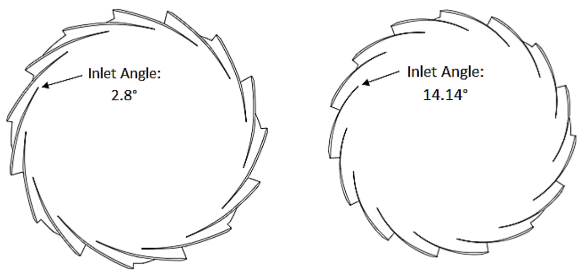


Figure 5: Comparison between measured and calculated tonal noise sound pressure level

4. Guide Vane Design

In order to reduce impingement of the impeller outflow on the leading edge the vane is redesigned. The leading edge angle for the guide vane was determined using impeller discharge velocity triangle. Slip factors were considered in the guide vane design. According to *Dixon*, even under ideal conditions the relative flow leaving the impeller will not be perfectly guide by the vane and the flow is said to slip. [18]. The inlet angle of the guide vane was determined as 14.14°. By using the guide vane, the impeller outflow is expected to have better flow guidance and impingement on the leading edge will be reduced.



(a) (b)

Figure 6: Geometry of the tested prototypes (a) Diffuser (b) Guide Vane

5. Results and Discussion

The SPL of the tonal noise from numerical modeling comparing the diffuser vane and guide vane is presented in the Figure 7. The guide vane shows reduction in the tonal noise at the fundamental BPF and 3rd harmonics. Tonal noise at the fundamental BPF was reduced by 4.2 dB, where else for 3rd harmonics it was 16.5 dB No reduction is achieved for tonal noise at 2nd harmonics.

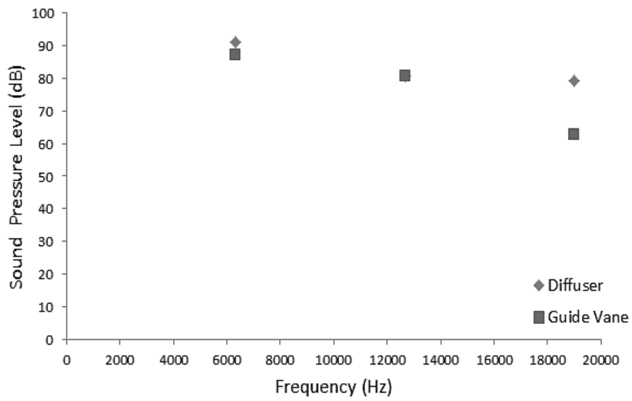


Figure 7: SPL frequency spectrum of diffuser and guide vane

Table 4: Comparison between diffuser vane and guide vane

	1st BPF (dB)	2nd BPF (dB)	3rd BPF (dB)
Diffuser Vane	91.2	80.8	79.3
Guide vane	87.0	80.7	62.8
Difference	4.2	0.1	16.5

Figure 8 shows comparison of pressure fluctuations at the leading edge of the diffuser vane and the guide vane. One will find there are 11 peaks for one period of rotation which corresponds to the number of impeller blades. Pressure fluctuations do exist at the leading edge of the guide vane but the amplitude is smaller compared to the leading edge of the diffuser vane. As mentioned before, the tonal noise generations are due to the interaction between the impeller trailing edge and the diffuser leading edge. The impingement of impeller outflow and the wake/jet flow structure on the leading edge of diffuser vane causes periodic pressure fluctuation which leads to the generation of tonal noise at the blade passing frequency. By using the guide vane, the interaction area of the impeller outflow on the leading edge becomes smaller. Reducing the interaction area causes reductions in the pressure fluctuation strength. In term of performance, the guide vane does not show apparent change compared to the original diffuser vane. The mass flow rate obtained through simulation was 0.051 kg/s which is similar to the mass flow rate using the diffuser vane.

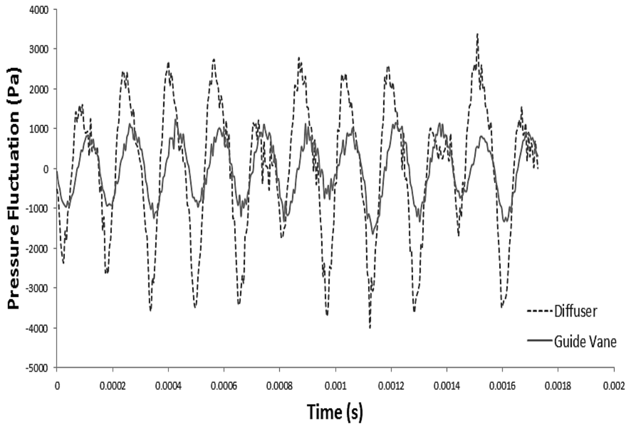


Figure 8: Comparison of pressure fluctuation at leading edge between the diffuser and guide vanes

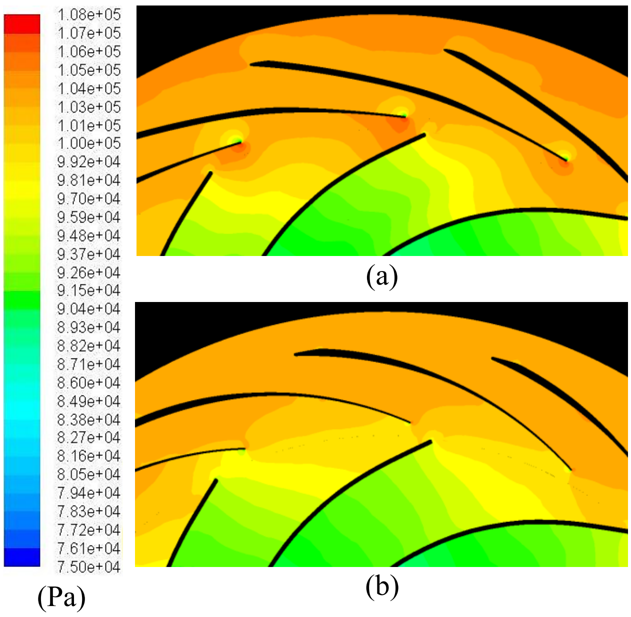


Figure 9: Contour of static pressure on leading edge of stationary vane during interaction (a) Original diffuser vane (b) Guide vane

6. Conclusion

The effects of guide vane on the tonal noise of centrifugal fan were computed in this work. The experimentally validated numerical modelling shows that the tonal noise in centrifugal fan can be reduced by changing the leading edge inlet angle. The tonal noise at the fundamental BPF was reduced up to 4.2dB and the 3rd harmonics reduced about 16.5dB. By determining the proper leading edge, the impingement of the impeller outflow can be reduced. As consequence, pressure fluctuation is suppressed, leading to the reduction of the tonal noise generation. This work is part of an on-going effort to design a quieter centrifugal fan.

Acknowledgement

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