



AEROACOUSTICS CHARACTERIZATION METHODOLOGY APPLICABLE TO TURBOCHARGER COMPRESSOR

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ABSTRACT

Turbochargers have become an important part of modern high efficient engines, and soon will be a standard component. Almost all automotive and industrial diesel engines and most of the high performance SI engines are equipped with turbocharger. Even though past few decades have seen continuous performance improvement, there is still lack of wide range research on acoustical behavior of turbochargers. A turbocharger consists of compressor which is driven by an exhaust turbine. Turbocharger produces high frequency aerodynamic sound due to the high speed rotating blade. The main aerodynamic noise generating mechanisms in turbo-compressors is tonal noise at blade passing frequencies, buzz-saw noise and blade tip clearance noise. The focus will be on tonal noise which 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. Higher rotating speed will result in a more prominent blade passing noise and its harmonics. The aim of this paper is to offer a methodology on characterizing the tonal noise of turbocharger based on investigation of high speed turbo machinery, which also has similar acoustical behavior. This study will include results from commercial computational fluid dynamics (CFD) code and experimental with the sound pressure level distribution.

Keywords: aeroacoustics, centrifugal fan, tonal noise, turbocharger compressor noise.

INTRODUCTION

In the past few years, the term engine downsizing has become important in the development of automotive engines. Engine downsizing achieved typically with turbocharger which utilizes exhaust energy and in return improving the engine performance. In addition to improved fuel economy, turbocharger also contributes to emission reduction [1-8]. Turbocharger consist two main parts; compressor and turbine, fit onto a common shaft, as shown in Figure-1. Hot exhaust gases from the engine drives the turbine and this consequently work to operate the compressor [3, 4, 9-12]. The compressor draws in ambient air and compresses it before pumping it into the engine's chambers.

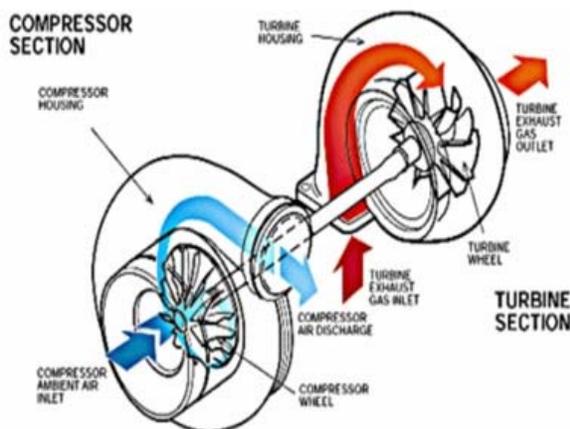


Figure-1. Schematic of main turbocharger components.

Turbochargers are known for their high frequency noise a consequent of rotating turbo wheel. Increasingly strict environmental noise level restrictions, coupled with passenger comfort requirements, means the noise control in turbochargers has become one of the important agenda in many of the current engine research programs [7, 8, 13-21]. Most of the researches aim to understand the characteristics of noise generation in turbomachinery on axial machine due to its high demand for aeronautics application. Experimental or theoretical studies on the radial machine are limited to the low-speed applications as used in air-conditioning and ventilating industries. One of the comprehensive reviews on turbocharger acoustics was published by Rämmal and Åbom [22]. According to Rämmal and Åbom [22], turbocharger noise can be divided into two categories which are passive acoustic and active acoustics. Passive acoustics is from the low frequency engine pulsations propagating through the intake/exhaust system, while active acoustics refers to high frequency aerodynamic sound [22]. The main aerodynamic noise generating mechanisms in turbo-compressors is the tonal noise at blade passing frequencies, buzz-saw noise and blade tip clearance noise which comes under active acoustics category [22, 23]. The focus of this paper will be on tonal noise which occurs due to pressure fluctuation related to the rotating speed at the compressor part of a turbocharger. Turbocharger compressor is generally consists of centrifugal compressor wheel, diffuser and housing, as shown in Figure-2. The diffuser is usually an integral part of the housing and its purpose is to increase the flow pressure, by reducing its



velocity, after the compressor wheel. This will further increase the delivery pressure into the engine [24].

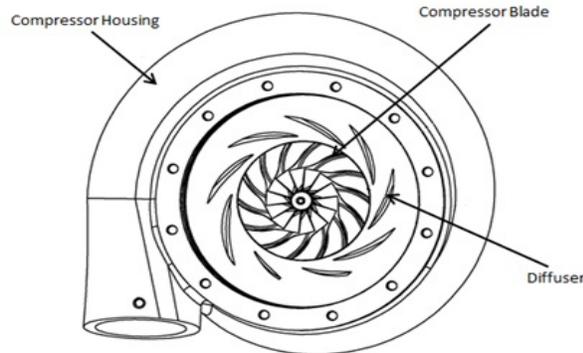


Figure-2. Components of compressor in a turbocharger.

In general, the aerodynamic output of turbomachines is proportional to the cube of shaft speed. Meanwhile the aeroacoustics power can be expected to be proportional to the 5th and 6th power of shaft speed [22, 23]. This indicates that measures on turbocharger noise control need to be taken into consideration as its capacity and speed are increased. In the current study, Computational Fluid Dynamic (CFD) is used to extract information on the sources of noise and consequently provide explanation on its existence [17, 25, 26]. In addition, an acoustic experiment was conducted in an anechoic chamber to obtain results for the computational validation. However, an acoustics experiment with real operating turbocharger is complicated and expensive. Thus, a high speed centrifugal fan which has similar acoustics behavior as a turbocharger compressor was used for the experiments in the current study. The main aim of this study would be to offer a methodology on characterizing the tonal noise of a high speed turbomachinery which can be equivalently applied to a turbocharger compressor.

Turbomachinery sound generation

The most common types of sound emitted are tonal noise and broadband noise. Tonal noise which also known as discrete frequency tone often relates to rotating parts of machines such as fans, turbines, compressors and pump. [27] The tonal noise component is associated with the pressure fluctuation caused by the blade passing. [28] Blade passing frequency (BPF) is governed by equation (1):

$$BPF = \frac{N_{rpm} \times N_b}{60} \quad (1)$$

where: N_{rpm} is the rotational speed and N_b is the number of the impeller blade

Broadband noise on the other hand is steady noise without the discrete frequency tones. The sounds are longer in duration and vary little over time. Broadband noise it is due to the inflow turbulence, the tip vortex, the trailing edge and the rotating stall. Figure-3 shows an

example noise spectrum of turbomachines which has tonal noise and broadband noise. In most cases of rotating machine, tonal noise the most dominant noise source. Khelladi et al. [29] Tonal noise is also periodic in time and consists of harmonics of the BPF. Higher rotor speed will produce more prominent BPF and its harmonics [22].

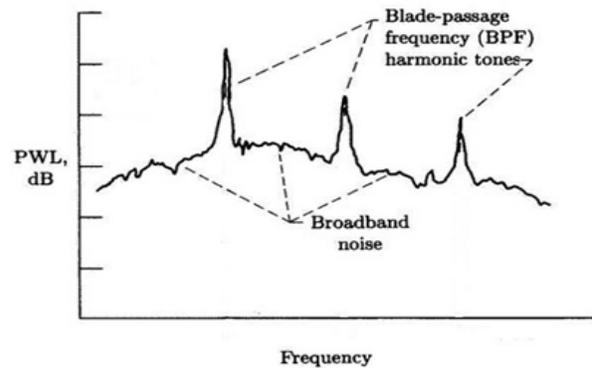


Figure-3. Noise spectrum of turbomachines.

EXPERIMENTAL METHOD

An acoustics experiment was conducted in the semi anechoic chamber located at Universiti Malaysia Pahang. Table-1 shows the geometric description of the centrifugal fan that was used in this study as an equivalent acoustic component as the turbocharger compressor. Figure-4 shows the components of the centrifugal fan. The semi anechoic chamber comply to the ISO 3745 Standard. The dimension of the chamber is 7.62m × 7.62m and its cut off frequency is 125Hz. ISO 3745 Standards recommend that the radius of the test hemi sphere should be either 1m or twice the largest source dimension. In the current, radius of hemi sphere is taken as 1 m.

Table-1. Geometric description of the centrifugal fan.

Description	Impeller	Diffuser	Return Channel
Radius of blade inlet [22]	18	53	60
Inlet blade angle (deg)	74	50	77
Radius of blade exit [22]	50	62	32
Angle of blade exit (deg)	50	62	39
Blade number	11	15	8
Blade thickness [22]	0.6	aerofoil	1

High accuracy microphones were used to measure the noise level emitted by the test specimen. In the current study, the choice of microphone was Pre-polarized Condenser type. In order to convert acoustical energy into electrical energy, microphones are used which operates on a capacitive design. For our experiment, we have chosen the Pre-polarized Condenser Microphone which operates on a capacitive design. Bruel & Kjaer



(B&K) Acoustics System was utilized which includes the Preamplifier microphone, LAN-XI data acquisition unit, and B&K PULSE Lab shop signal analysis software. The microphone was located 1m apart from the center of the fan unit in the x direction. Figure-5 shows the test setup in the semi-anechoic chamber.

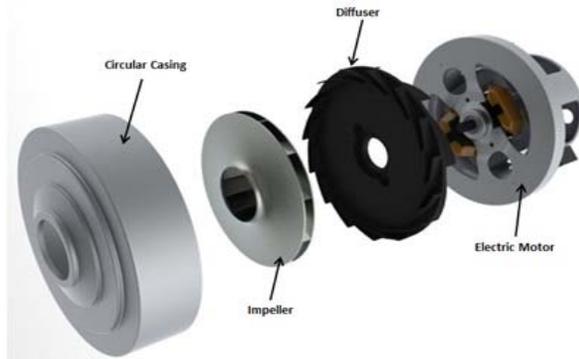


Figure-4. Exploded image of the parts from centrifugal fan.



Figure-5. Centrifugal fan measured in semi anechoic chamber.

NUMERICAL METHOD

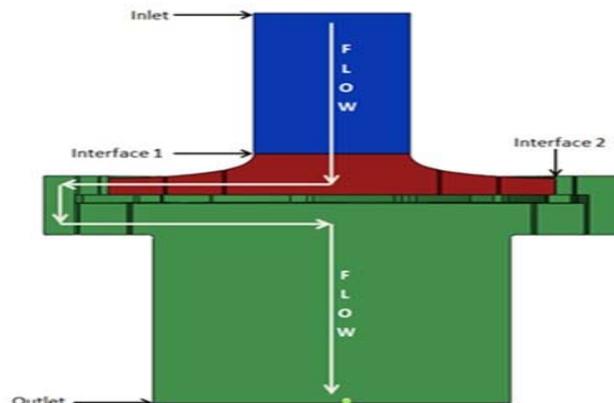
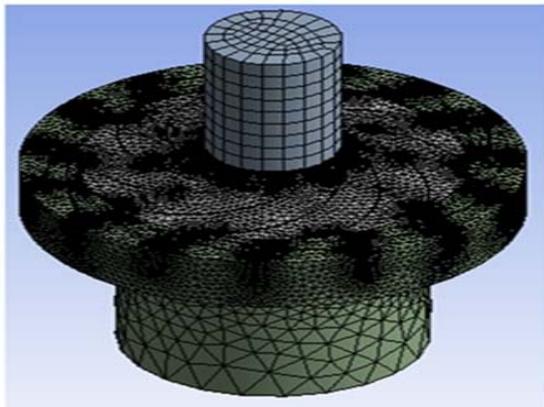


Figure-6. Computational mesh and boundary condition for the CFD.

This section presents the overview of the computational fluid dynamic (CFD) modeling using ANSYS FLUENT. Hybrid meshing with tetrahedral and hexahedral type was used, to cater for the component complexity. The computational domain is consist of 3 fluid domains: the inlet, impeller and diffuser. In order to capture the spectral content of the aeroacoustics source, time accurate calculations are necessary in setting up this turbulence modeling. Hence, the unsteady term of conservation equations second order implicit is applied. Turbulence modeling that were used are *k-w SST* variant of Detached Eddy Simulation (*DES*), as this model is capable in resolving eddies for noise prediction. In addition, *DES* also offers more practical solution. SIMPLE algorithm is used for pressure-velocity coupling and second order upwind for the turbulence [30, 31].

One of the major difficulties in CFD, which highly affects the solution accuracy, is the definition of boundary conditions. Surfaces that rotate relatively are defined as “moving wall”. The inlet and outlet are defined as “pressure inlet” and “pressure outlet”. Fluid zone in the inner volume is defined as “moving mesh” and defined 34560 rpm of rotational speed. In order to give sliding mesh property, the same surfaces in the inner and outer volume families are defined as “interface”. The interfaces type boundary conditions are applied in between the inlet and impeller surface and also between the impeller outlet and the volute inlet. This configuration is known as non-conformal interfaces [29, 32].

Using the pressure data from CFD, the noise sources resolves in the acoustics wave propagation simulation to compute the acoustics field for sound pressure level. In predicting rotating machinery noise, the Ffowc-Williams and Hawking (FW-H) equation was used. The FW-H formulation adopts the most general form of Lighthill’s acoustic analogy, and is capable of predicting sound generated by equivalent acoustic sources. [33] The receiver location as defined in the experiment which is 1m in X-direction from center before the noise Sound Pressure Level (SPL) spectrum was plotted.



RESULTS AND DISCUSSIONS

Figure-7 shows the SPL spectrum results from CFD and experiment. The peaks at BPF (6336 Hz) and its harmonics (12672, 19008 Hz) are clearly noticeable. The steady noise without the discrete frequency noise represents the broadband noise. The sound pressure level of the tonal noise at BPF is more prominent and higher than the broadband noise.

The tonal noise generation is related to the interaction between the impeller and the stationary diffuser which cause pressure fluctuation to occur at the Interface 2 (refer to Figure-6). Referring to the Figure-8, the flow after the impeller exit can be characterized by the fact that

the pressure falls quickly at the leading edge of the diffuser. These periodic fluctuation produces tonal sound. [34] During one revolution of the impeller the pressure at the diffuser vane fluctuates periodically 11 times as shown in Figure-9. From the SPL spectrum the CFD result is higher compared to the experiment result because in CFD simulation the noise is assumed to propagate in free field conditions. This means the reflection, resonance and absorption phenomenon is not taken account in this acoustic model. The noise level presented is based only on aerodynamics of fluid flow.

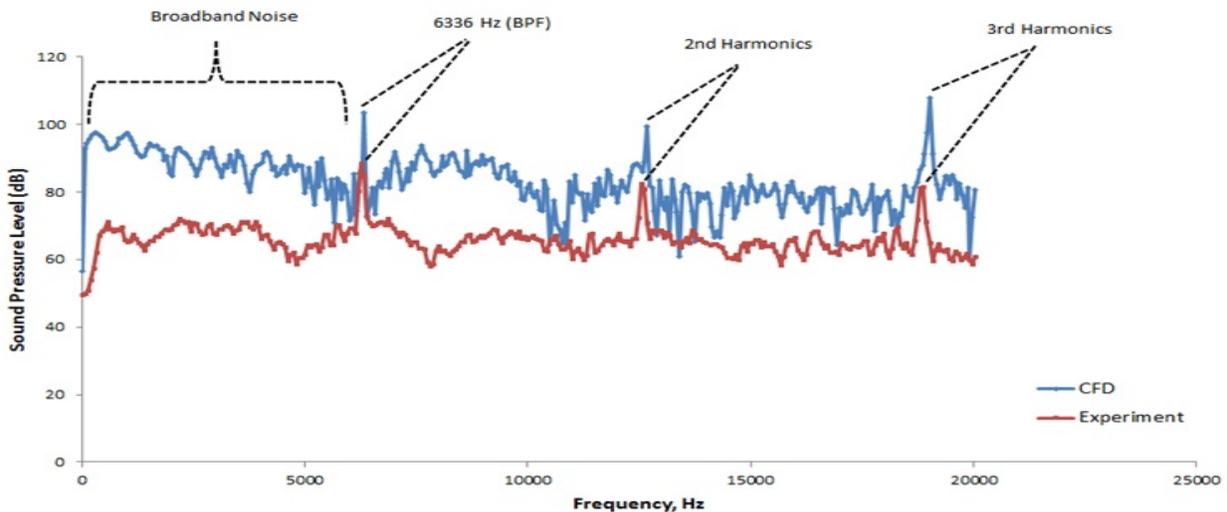


Figure-7. Comparison of noise spectrum from CFD and experiment.

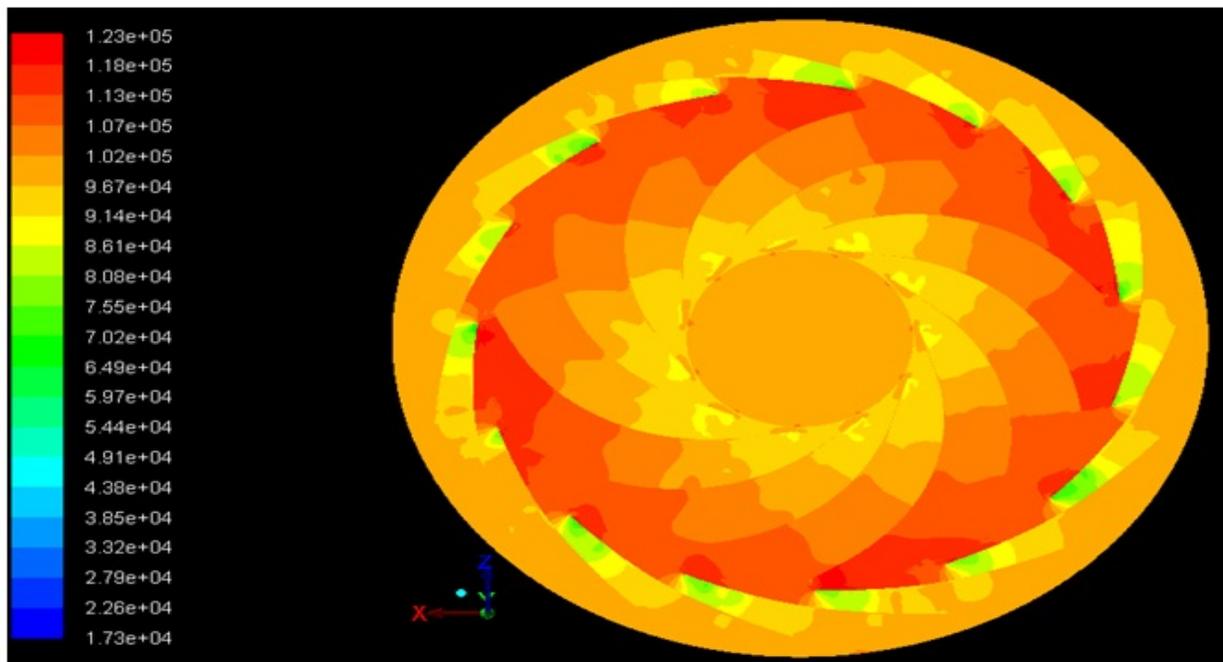


Figure-8. Pressure contour of the centrifugal fan.

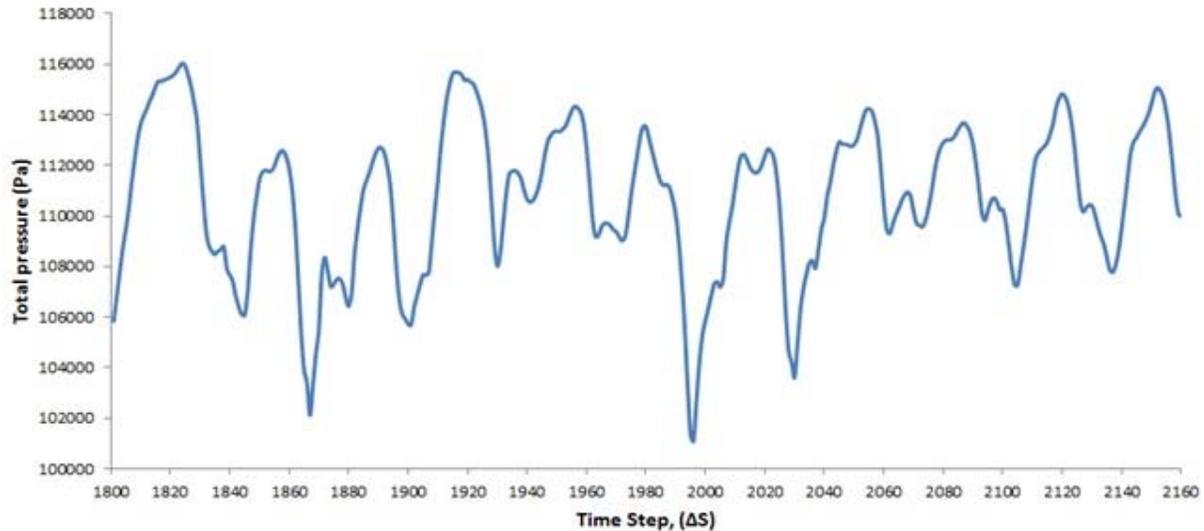


Figure-9. Pressure fluctuation at the diffuser vane.

CONCLUSIONS

In this paper, the acoustics behavior in a centrifugal fan is investigated which provides methodology on characterizing the tonal noise in a turbocharger. An experimental and numerical (CFD) study is described to identify the tonal noise generation in centrifugal fan. The acoustics measurement in a semi anechoic chamber was used as a validation for the CFD. In overall it shows that the dominant noise in a centrifugal fan is the tonal noise. The interaction between the impeller and the stationary diffuser are the source of noise generation. Hence, with this finding it is expected that similar phenomenon to occur in the turbocharger compressor where the interaction between the compressor wheel and diffuser will cause tonal noise. The knowledge from this study is a first step towards designing a low noise turbocharger compressor.

ACKNOWLEDGEMENTS

The authors would to thank The Ministry of Higher Education Malaysia and Universiti Teknologi Malaysia for Flagship Grant Vot 01G49.

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