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Tonal noise prediction in a small high speed centrifugal fan and experimental validation

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ABSTRACT

This paper presents the work done to establish a methodology that is capable of predicting tonal noise generation in high speed centrifugal fans. The deliverables from this work emphasize on identifying and characterizing the source of tonal noise in the centrifugal fan. Computational fluid dynamics (CFD) modeling was performed using 3-D Detached Eddy Simulation (DES) to compute the unsteady flow field in the fan. The calculated time history of surface data from the CFD is then used in Ffowcs Williams-Hawkings (FW-H) solver to predict the far field noise levels. The predicted aerodynamics and aeroacoustics results are in good agreement with the experimental data acquired from the flow testing facility and the anechoic chamber. The study conducted on the centrifugal fan shows that the aerodynamic interaction between the non-uniform impeller outflow and the leading edge of the diffuser vane is the source of tonal noise generation. The impingement of the jet-wake flow structure from the impeller outflow causes periodical pressure fluctuation on the leading edge of the diffuser vanes which leads to the tonal noise generation at the blade passing frequency (BPF).

1. Introduction

The wide range application of turbomachines has led to intensive research and development towards improving its efficiency in designing highly efficient turbomachines. Most of the turbomachines operate at high rotational speed in order to fulfil the power output requirement for its given size. It is well established through many previous studies that the flow field in turbomachines are highly unsteady [1]. Hence, increase in rotational speed causes more turbulence, large scale of velocity and pressure fluctuation which in turn generates higher noise level.

The flow induced noise from turbomachines is generally characterized by broadband noise with prevailing discrete frequency tones. Among these two, the discrete frequency tones or also known as tonal noise is the most noticeable in sound spectrum compared to the broadband noise [2–8]. According to Raitor and Neise [9], the aerodynamic power output of a turbomachine is proportional to the cube of its rotational speed where else the aeroacoustics noise level rises in the fifth to sixth order. This relationship raises concern among researchers that it is necessary to consider aeroacoustics noise control measures during the design stage of turbomachines. Furthermore, it is also important that noise sources in the turbomachines are identified and characterized in order to implement suitable noise control techniques. As stated by Lee et al. [10], the best solution in reducing noise generation is to identify the noise source and completely understand its mechanism.

Earlier works in predicting flow induced noise in turbomachines were more focused on theoretical and experimental studies. One of the earliest theoretical formulations established in predicting noise from rotating machinery was the FW-H equation by Ffowcs Williams and Hawkings in 1969 [11]. The FW-H equation is an extension of the acoustic analogy developed by Sir James Lighthill which focuses on prediction of noise generated by the jet of an aircraft turbojet engine [12,13]. This extension to the acoustic analogy includes the dipole and monopole source distributions. In term of experiments, studies were conducted by making geometrical changes to the turbomachines and determining its effect on the noise generation. A very comprehensive review summarising all the research work done from 1960 to 1975 in the effort to reduce tonal noise from centrifugal fan was presented by Neise [3]. Neise and Koopmann [14,15] then experimentally studied a noise reduction method for centrifugal fan using acoustics resonator. Dong et al. [16] used particle image velocimetry (PIV), surface pressure and noise measurement to study the effects of modifying the

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Nomenclature

b_2	blade width at discharge (m)
c_0	speed of sound (m s^{-1})
d_2	blade diameter (m)
F_i	force acting on fluid (N)
H	Heaviside function
n	impeller rotational speed (rpm)
p'	sound pressure (Pa)
P	pressure (Pa)
Q	volume flow rate (m^3 s^{-1})
r	radius (m)
t	time (s)
T_{ij}	Lighthill's tensor
U	flow velocity
u_2	impeller peripheral speed (m s^{-1})
x_i	cartesian coordinates of the observer (m)
x_j	cartesian coordinates of the source (m)
Z	number of blade
$\Pi_{t,t}$	total to total pressure ratio
ϵ_R	relative error
$\delta(f)$	Dirac-delta distribution
ρ	air density
φ	flow coefficient
ψ	total pressure coefficient
φ	generic variables
μ	viscosity

Subscripts

t	total

Abbreviations

BEP	best efficiency point
BPF	blade passing frequency
CFD	computational fluid dynamics
CAA	computational aeroacoustics
dB	decibel
dB(A)	A-weighting decibel
SPL	sound pressure level

cut-off tongue and impeller geometries on the flow structure, local pressure fluctuation and noise.

Nonetheless, with the advancement in computational technology, numerical simulations such as computational fluid dynamics (CFD) and computational aeroacoustics (CAA) enable noise prediction in turbomachinery to be more feasible. Thus through the combination of experimental work and numerical simulation, deeper understanding on the unsteady flow field and the aerodynamic noise generation mechanism in turbomachinery can be achieved [17]. Jeon et al. [5] introduced an incompressible two dimensional discrete vortex method (DVM) to analyse the unsteady flow field of centrifugal fan. Then, using the calculated unsteady force data in the fan flow region, the noise radiations are predicted through FW-H equation. A two-part paper was then published by Langthjem and Olhoff [18,19] to study the flow induced noise phenomenon in a two-dimensional centrifugal pump where they adopted the DVM to estimate the strength of the dipole sources due to the unsteady surface force. In order to predict the noise generation within the volute of the centrifugal pump, they employed the newly developed boundary element method (BEM).

In order to further improve understanding of the noise generation in an unsteady flow field, Liu et al. [20], Ballesteros-Tajadura et al. [6], Khelladi et al. [21], and Mao et al. [22] performed three-dimensional numerical calculations on centrifugal fan. They employed Reynolds Averaged Navier-Stokes (RANS) calculation to solve the unsteady flow of the centrifugal fan and the aeroacoustics modelling was performed using the FW-H equation. The results from their investigations provide better qualitative predictions compared to the two-dimensional results presented by previous researchers.

Despite the knowledge accumulated over the past few decades on the noise generation mechanism in centrifugal turbomachines, the prediction of noise generation in such complex flow is still difficult especially in high speed turbomachinery and requires much deeper understanding. In this paper, a methodology that is capable of identifying and characterising the tonal noise source in a high speed centrifugal fan is presented. This paper is an advancement of the previously published papers by Paramasivam et al. [23,24] which focuses on the reduction of tonal noise through the application of guide vane and tapered guide vane.

The centrifugal fan adopted for this study mainly consists of an impeller with 11 blades, diffuser vanes, a circular casing and a universal motor as shown in Fig. 1. The aerodynamic and the geometrical characteristics of the centrifugal fan are presented in Tables 1 and 2.

2. Experimental methodology

2.1. Aerodynamic performance measurement

In order to obtain the aerodynamic performance of the centrifugal fan, a flow bench as shown in Fig. 2 was setup. A straight PVC pipe was fitted at the inlet of the centrifugal fan and an orifice plate which was designed in accordance to ISO 5167-1 and ISO 5167-2 [25,26] was installed at the mid-point. The flow rate of the centrifugal fan was obtained based on the static pressure difference across the orifice plate using U-tube manometer. Pressure transducers are flushed mounted at the inlet of centrifugal fan, exit of the impeller and the exit of diffuser – to measure static pressures across the centrifugal fan. Fig. 3 shows the location of pressure transducers on the centrifugal fan. In addition to the pressure measurement, the temperature of the air flow was also measured simultaneously. Furthermore, an inductive proximity sensor was used to measure the impeller rotational speed which corresponds to the universal motor speed.

The following uncertainties were established for the measured and calculated magnitudes:

I. U-Tube Manometer: ±19.6 Pa
II. Pressure Transducer: ±1.7 kPa
III. Inductive Proximity Sensor: ±5%
IV. K-type Thermocouple: ±2.2 °C

Since the centrifugal fan constantly operates at its best efficiency point (BEP), only one operating speed was considered in this work. The measured rotational speed was 34,560 RPM and since the number of blade is 11, thus the blade passing frequency (BPF) of the centrifugal fan would be 6336 Hz. The BPF can be determined using Eq. (1):