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Numerical and experimental investigations of the unsteady aerodynamics and aero-acoustics characteristics of a backward curved blade centrifugal fan

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ABSTRACT

A numerical study of the aerodynamic and aeroacoustic behaviors of a backward curved blade centrifugal fan was conducted under two important flow conditions: BEP and 1.3 × BEP. Three-dimensional numerical simulations of the complete unsteady flow field for the whole impeller-volute configuration were used to determine the aeroacoustic sources. To locate the unsteady flow and perturbations, the near field wall pressure fluctuations at different strategic points on the volute were computed using the URANS approach. Thus the intensities and positions of the aeroacoustic sources were identified by analyzing frequency spectra. The aeroacoustic sources caused by fluctuations in the interactions of the flows leaving the impeller and volute were close to the volute tongue, and the most effective noise sources related to the flow rate were near the impeller shrouds. In addition, the unsteady flow variables provided by CFD calculations were used as inputs in the Ffowcs Williams-Hawkings equation to estimate the noise tones of the fan. The aeroacoustic calculation results showed that the volute noise was much larger than the blade noise, and the noise mainly propagated from the outlet duct of the fan. Moreover, to account for the noise propagation, three calculation methods were used by applying different solid boundaries. Compared with the other methods, the FEM method, which accounted for the complex solid boundaries, produced good agreement and showed that the complex solid boundaries cannot be neglected in aeroacoustic predictions. The calculation results showed good agreement with the experimental results.

1. Introduction

Centrifugal fans are a common type of turbomachinery that are widely used in the ventilation systems of ships' cabins and other sites, and they help provide comfortable working and living environments for people. However, the noise generated when fans run disturbs us, and the study of noise generation and propagation therefore becomes increasingly important. Previous studies have shown that pressure fluctuations on a volute surface induced by internal unsteady flows are important sources of fan casing vibrations and tone noise generation. Regarding centrifugal fans, Fehse and Neise [1] studied the noise generated by a low speed centrifugal fan. They showed that the broadband components at low frequencies were generated by classic flow separation on the impeller shroud and blades suction sides. In addition, parts of the volute tongue were identified as the noise source region. Ohta et al. [2,3] investigated the tonal noise generation of a low specific speed centrifugal fan. According to their investigation, the most effective noise source is close to the volute tongue, and the precise source region was determined using a correlation analysis between the acoustic pressure and the pressure fluctuations on the volute surface. Velarde-Suárezet et al. [4–6] employed numerical and experimental methods to investigate the pressure fluctuations on the volute surface of a frontward curved blade centrifugal fan. Their study indicated that the highest amplitude of pressure fluctuations appeared on the volute casing close to the volute tongue, and the blade passing frequency was dominant except at a position beyond the impeller outlet width. Younsi et al. [7] also studied fluctuations on a volute casing and the characteristics of tone noise in a frontward curved blade centrifugal fan. The results showed that the periodic interactions between the circumferential uniformity of the flow pattern at the impeller outlet and the volute tongue generated the tone noise. The main noise sources were

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Nomenclature

B	volute width (mm)
Z	the axial distance from the monitoring point to the volute rear casing (mm)
BEP	the best design point
BPF	the blade passing frequency (Hz)
p'	acoustic pressure (Pa)
p_{ref}	reference pressure (Pa)
$\delta(f)$	Dirac-delta distribution
$H(f)$	heaviside distribution
T_{ij}	Lighthill stress tensor (Pa)
P_{ij}	compressor stress tensor (Pa)
n	surface normal
v_n, u_n	normal surface velocity (m s^{-1})
W	weighting function
p_i	nodal sound pressure
Ω	control surface of enclosed boundary
$[N]$	global shape functions
$[\mathbf{Ka}]$	acoustic stiffness matrices
$[\mathbf{Ca}]$	acoustic damping matrices
$[\mathbf{Ma}]$	acoustic mass matrices
$\vec{Z}$	normal impedance
D_2	impeller outlet diameter (m)
Q	flow rate (m^3 s^{-1})
T_R	a blade-passing period
R_I	radius of impeller outlet (m)
R	radius of circumferential section (m)
P_T	total pressure rise (Pa)
φ	flow coefficient, $\varphi = Q/(D_2^2 u_2 \pi/4)$
ψ	total pressure coefficient
c	speed of sound (m s^{-1})
u_2	circumferential velocity of impeller outlet (m s^{-1}), $u_2 = \omega \cdot D_2/2$
ρ	density, ρ = 1.20 kg/m^3
ω	angular velocity (rad s^{-1})
SPLs	sound pressure levels (dB)

concentrated in a small region in the volute tongue region. The influence of centrifugal fan noise caused by impeller speed, volute tongue gap and the number of impeller blades has been investigated by Jeon [8]. The results revealed that the noise was mainly induced by an unsteady interaction between the flow leaving the impeller and the fixed volute tongue and that the tongue gap considerably affected the amplitudes of the power spectra.

Numerical research on centrifugal fan aeroacoustics are usually divided into two important steps: the first step is obtain the noise source by solving the unsteady flow field, for which Computational Fluid Dynamic methods are used, and in the next step, the Lighthill [9,10] acoustic analogy or its special formalisms such as the Ffowcs Williams–Hawkings (FW–H) equation, Lowson equation and Curle equation are applied to predict the far-field noise. Younis et al. [11] predicted a frontward multi-blade centrifugal fan noise tone generated from impeller monopole and dipole sources using the FW–H equation. Khelladi [12] established an impeller noise model by applying the Lowson equation to solve for the monopole and dipole noise excited by the impeller and vane diffuser of a high-speed centrifugal fan. To obtain the aerodynamic noise source of an industrial centrifugal fan, Liu [13] employed the LES method to solve the unsteady flow field, and in their study the fan dipole discrete noise and quadrapole broadband noise were predicted using the FW–H equations and Powell vortex theory, respectively. However, the simulation error was much greater due to the solid boundary effects of the casing, which were neglected in their calculations. To consider the solid boundary effects of the casing, a newly developed Kirchhoff–Helmholtz boundary element method (BEM) was employed by Tournour [14] and Cai [15,16] to take the noise propagation into account. Indeed this BEM method discretizes the Lighthill equations by applying a free field Green function integral. At present, the Green function integral method can only solve problems with simple geometric boundaries, and those complex boundaries must be simplified in the free field. Without doubt, this simplification does not consider reflection and scattering effects in the noise propagation. To overcome this disadvantage, Kaltenbacher [17] introduced a Finite Element Method (FEM) based on the Lighthill variation formulation, in which the propagating noise can be spatially discretized to account for the interactions between the structure and noise, which makes it suitable for all complex geometries. Kang [18] employed this method and successfully predicted the impeller dipole noise of an automobile air conditioner centrifugal fan. The results showed that there was a good agreement if the volute casing solid boundary was considered in the simulation. However, the predictions referred to above neglected the effect of the blade solid boundary in the noise propagation. In fact, the blade solid boundary cannot be overlooked in noise propagation.

In the current investigation, both numerical and experimental studies were employed to obtain the characteristics of aerodynamics and aeroacoustics of a high-speed centrifugal fan. Thus, the overall performance and local behaviors of the centrifugal fan were studied. In addition, the acoustic behavior was studied using the FW–H formalism coupled with URANS data. In addition, experimental measurement, which used sound pressure level approximate engineering method, was gathered to validate the numerical results.

2. Description of the fan and experimental procedures

2.1. Test fan description

The test machine is a ventilating centrifugal fan with four main components (conical bell mouth, shrouded impeller, volute casing with conventional tongue and conical flow rates throttle) driven by an AC inverter motor with adjustable angular speed between 0 and 3600 r/min; the design rotational speed is specified as 2920 r/min. Fig. 1 shows the profile of the components. The main dimensions and characteristics of the investigated fan for this study are presented in Table 1. The fan inlet is open to ambient air. The tests for characterizing the aerodynamic and acoustic behaviors of the fan system were made in a resilient installation to fulfill ship system noise and vibration requirements (according to standards GJB4058-2000 China [19] and GB-T1236-2000 China [20].) Fig. 2a shows the details of the test system installation and data collection procedure. The following maximum measurement errors were obtained for the different magnitudes: total pressure ±2% (±10 Pa), flow rate: ±2% (±0.05 m^3/s) and shaft power ±2% (±50 W).

2.2. Dynamic pressure measurement

The Dynamic Pressure Testing System (DPTS), which included XCQ-080-5G Kulite high frequency dynamic pressure sensors, standard power supplies, 8300 AU amplifiers and AVANT