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Effect of the rotor–stator gap variation on the tonal noise generated by axial-flow fans

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ABSTRACT

The effect of the rotor–stator axial-gap on tonal noise generated by an axial-flow fan employed for automotive cooling systems has been studied. A fan equipped with a 9 blade rotor and a 18 vane stator positioned at several axial-gaps has been tested in a hemi-anechoic chamber. The acoustic measurements have been performed during rotational speed ramps. To analyze the experimental data, the propagation function, obtained by means of the spectral decomposition, has been compared with the velocity-scaled SPL and the phase angle, evaluated at the 1st and 2nd blade passing frequency harmonics. Opposite to what is commonly observed, the SPL due to the rotor–stator aerodynamic interaction does not monotonically decrease with the axial-gap and at the shortest gaps it may not be scaled with a single power of the rotational speed. The listed quantities have been plotted versus frequency, rotational speed, and axial-gap. Their analysis provides a detailed picture of the investigated phenomena and supports the assumption that the observed behavior is due to acoustic effects and not to the aerodynamic noise generating mechanism.

1. Introduction

Tonal noise is an important component of the noise generated by axial-flow fans employed in automotive cooling systems. It may be due to the ingestion of large scale inlet turbulent structures, Homicz and George [1], Majumdar and Peake [2], Hanson [3], or to the aerodynamic rotor–stator interaction Lowson [4], Tyler and Sofrin [5], Kaji and Okazaki [6]. Compared to broadband noise, it is often a major cause of annoyance, due to its periodic nature.

Basing on established knowledge [4,6], the designer expects that increasing the axial-gap between rotor and stator allows to decrease the tonal noise and hence he seeks for a trade-off between module compactness and noise reduction. Other important aspects are the scaling properties, i.e. the possibility of relying on simple relations expressing the dependence of the SPL on the rotational speed, Neise and Barsikow [7]. Conversely, an unexpected dependence on both rotational speed, Quinlan and Krane [8], and axial-gap, Canepa et al. [9], has been observed, resulting in difficulties in deciding how to proceed with the design.

This strongly happens to the noise due to rotor–stator interaction since it has a high temporal coherence. Indeed, the tonal components excite the acoustic response of the system (fan assembly and test environment) at precise frequencies, possibly resulting in relevant attenuation or amplification if small variations of the rotational speed take place. This phenomenon has been reported by Margetts [10] and Canepa et al. [11]. In real cooling units, the fan often operates at variable speed and the tonal noise peaks sweep the acoustic response function of the system. As a consequence, irregularities in the growth and in the decrease of tonal components during the fan operation may be heard, resulting in an important cause of annoyance. However, such irregularities could also depend on variations in the generating mechanism strength. Due to such a complicated behavior, tests taken at a limited number of axial-gap values, [9], resulted in interesting indications on the features of the phenomenon but could not provide a detailed picture of it. In the present work, only one stator geometry has been considered, the axial-gap has been systematically varied with a reduced step, and attention has been focused on the first and the second BPF harmonics only. This has allowed to perform a deeper and more complete study. Furthermore, installation effects have been discussed more deeply.

Experimental techniques based on the employ of simple instrumentation and facilities, i.e. tests with one or few microphones in hemi-anechoic chambers, are usual in the automotive field. This limits the available information and, at the same time, may introduce further undesirable acoustic phenomena, Canepa et al. [11] and Roger [12]. In the present paper, amplitude and phase of the

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Nomenclature

a_0	speed of sound (m/s)
c	blade chord (m)
C_n	constant used in the scaling of SPL_n (dB)
d	rotor–stator axial-gap (m)
D_{hub}	rotor hub diameter (m)
D_{tip}	rotating shroud inner diameter (m)
D_n	scaled SPL (departure of SPL_n from the linear dependence on $10\log_{10}\Omega$) (dB)
f	frequency (Hz)
G	propagation function (dB)
He	Helmholtz number based on D_{tip} and on a_0, $= f D_{\mathrm{tip}}/a_0$ (–)
K, K_n'	constants used in the SPL scaling (dB)
l_{refl}	length of the path followed by reflected waves
Ma_{tip}	Mach number based on u_{tip} (–)
n	multiple of the harmonic order of tonal noise components, $n = St / \Delta St$ (–)
OASPL	overall sound pressure level (linear, ref. 20 μPa) (dB)
Q	volume flow rate (m^3/s)
r	distance of the receiver to the source (m)
R	radius of a blade airfoil (m)
$S_{pp}(f,\Omega)$	power spectral density of the received acoustic pressure (Pa^2 s)
SPL	sound pressure level spectrum (ref. 20 μPa, Δf = 12.5 Hz) (dB)
SPL_n	sound pressure level at the nth BPF harmonic (ref. 20 μPa) (dB)
St	Strouhal number based on the rotational frequency, $=60f/\Omega$ (–)
t	time of reception of the acoustic wave (s)
u_{tip}	peripheral speed at the blade tip (m/s)
z_R	rotor blade number (–)
z_S	stator vane number (–)
α	exponent in the scaling of the SPL at variable Ω (–)
β	exponent in the scaling of the constant-Ω SPL spectrum (–)
Δf	bandwidth employed in the SPL computation
Δp	static pressure rise (outlet static pressure minus the ambient pressure) (Pa)
ΔSt	non-dimensional bandwidth employed in the SPL_n computation
θ	angular distance between the noise source and the reflecting tape (deg)
τ	time delay (s)
ϕ	phase angle (ref. *tacho pulse* emission) (deg)
φ	flow coefficient (–)
ψ	pressure coefficient (–)
Ω	rotational speed (r/min)

Subscripts

BB	related to broadband noise
e	wave emission
filt	filtered by means of the propagation function
n	related to the *n*th harmonic order
orig	computed by the spectrum analyzer
prop	related to propagation
ref	emission of the *tacho pulse* (s)
source	related to the acoustic source
T	related to tonal noise

acoustic pressure measured during speed ramps are studied according to the theory described in [11], and the spectral decomposition method, Bongiovì and Cattanei [13], is employed to highlight the acoustic response of the whole system.

Finally, the obtained results may provide useful information for more theoretical studies such as the ones of Abid et al. [14] or Trabelsi et al. [15].

2. Experimental facility, data processing method, and tests organization

2.1. Tested fan

The measurements have been taken on a fan whose rotor is made of polyamide with glass fibers, Fig. 1. The rotor is provided with $z_R = 9$ evenly spaced blades and with a cylindrical rotating shroud of external diameter D_{tip} = 460 mm, the hub has a diameter D_{hub} = 181 mm, and the blade chord c varies between 65 mm at the hub and 72 mm at the tip.

The stator employed in the present work is evenly-spaced and is composed by z_S = 18 constant-thickness, cambered vanes of 30 mm chord, Fig. 1. The axial-gap between the rotor blade trailing edge and the stator vane leading edge at the rotor tip, d, has been varied in the range 12–27 mm (d/c_{tip} = 0.167–0.375) with 1 mm steps, resulting in 16 different position. The choice $z_S = 2z_R$ is not realistic since in actual fans z_R and z_S are never integer multiples but it enhances the SPL at the low BPF harmonics. Indeed, the stators employed in production units are usually composed of non-evenly spaced vanes and struts, thus resulting in tonal noise components at all BPF harmonics. In the present case, the standard gap is d = 23 mm, which equals to about one axial chord of the blade. Other details of the experimental facility are reported in Canepa et al. [16].

The design performance are a flow rate Q = 1.081 m^3/s and a static pressure rise $\Delta p = p_{\mathrm{out}} - p_0$ = 320 Pa at Ω = 2725 r/min with p_0 the total pressure at the rotor inlet and air at ambient conditions (T_0 = 20 °C and p_0 = 101,300 Pa). This results in a flow coefficient $\varphi_{des} = Q/(u_{tip}\pi D_{tip}^2/4) = 0.095$ and a pressure coefficient $\psi_{des} = \Delta p_{des}/(0.5\rho_0 u_{tip}^2) = 0.134$, where u_{tip} is the blade tip speed. In the present investigation, the fan has been operated at free-discharge conditions, for which $\varphi = 0.164$ and $\psi = 0$.

The rotor is driven by a PC-controlled brushless servomotor (Danaher AKM42E-ANCNR-00, rated power 1.14 kW at 640 V) supported by a steel structure which allows a precise positioning of the rotor by means of a 3-axis system. The structure is fixed to a 690 mm × 710 mm rectangular wooden panel. The motor is quieter and more stable than the brushless motors employed in the production units. Its noise does not interfere with the aerodynamic one and the only noticeable effect is a sharp peak at $f \cong 16$ kHz, which does not influence the noise object of the present study. The wooden panel has a central circular hole and is supported by a metal frame which realizes a free-discharge condition. The tip-clearance geometry (with 5 mm axial and radial gaps) is obtained inserting different aluminum rings in the hole. In the present test configuration, at free-discharge conditions, the tip-leakage noise is negligible compared to the tonal one, Canepa et al. [16]. The stator has been inserted in the most external ring and no heat exchanger has been mounted.

2.2. Test configuration

The measurement campaign has been carried out in the DIME hemi-anechoic chamber. Below 100 Hz, the SPL spectra may be affected by the loss of anechoicness of the test environment, and, at a so low frequency, they must be carefully treated. Usually,