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Acoustic characteristics of a heavy duty vehicle cooling module

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ABSTRACT

Studies dedicated to the determination of acoustic characteristics of an automotive cooling package are presented. A shrouded subsonic axial fan is mounted in a wall separating an anechoic- and a reverberation room. This enables a unique separation of the up- and downstream sound fields. Microphone measurements were acquired of the radiated sound as a function of rotational speed, fan type and components included in the cooling module. The aim of the present work is to investigate the effect of a closely mounted radiator upstream of the impeller on the SPL spectral distribution. Upon examination of the SPL spectral shape, features linked specifically to the source and system are revealed. The properties of a reverberant sound field combined with the method of spectral decomposition permit an estimation of the source spectral distribution and the acoustic transfer response, respectively. Additionally, purely intrinsic acoustic properties of the radiator are scrutinized by standardized ISO methods. A new methodology comprising a dipole sound source is adopted to circumvent limitation of transmission loss measurement in the low frequency range. The sound attenuation caused by the radiator alone was found to be negligible.

1. Introduction

Despite dedicated efforts to clarify the noise generating mechanisms associated with cooling fans, there is still inadequate knowledge of its susceptibility to complex installation effects. The measured far-field sound pressure level (SPL) from a spinning fan operating in a duct-like environment is ineluctably colored by the acoustic properties of the system. Inherent acoustic phenomena like resonances and reflections of the system prevent identification of the source spectral distribution alone. Hence, performing acoustical studies of turbomachinery noise sources, comprise integration of the acoustic modulation caused by the environment, i.e. account for *acoustic installation effects*. Furthermore, the spectral distribution and magnitude of the aerodynamic noise that originates from the fluctuating forces exerted by the blades is closely linked to the inflow characteristics. Consequently, the perceived SPL visualized as peaks at discrete frequencies together with broadband components [1,2] is highly correlated with the turbulence content and the coherence in the inflow [3,4]. Additionally, obstacles placed at the inlet [5,6], an asymmetric shaped shroud [7], boundary layers on the casing [8,9] together with vortex formation in the fan tip region [10] significantly affect in particular harmonic spectral components. These are commonly referred to as *aerodynamic installation effects*.

Thus, the study of sound generating mechanisms and acoustic propagation from ducted fans involves examination of both installation effects. The aim of the present work is to investigate the effect that a closely mounted radiator upstream of the impeller has on the SPL spectral distribution radiated from an automotive cooling module. Due to the radiator's inherent pressure loss associated with its flow inhibiting properties [11] and acoustic transmission loss [12] both the aerodynamic and the acoustic conditions are amended [13,14]. Previous published material frequently omits the acoustic masking effect caused by the system and the radiator [5,14,15]. These studies solely accounted for the SPL difference caused by the combined effects as no separate analysis of the acoustic properties of the radiator or the system is included in the investigations. Also, in some instances, comparisons are made between a clean fan and a fan placed in a duct with a radiator [5].

In the present study, emphasis is on separating the impact of each installation effect. An engineering approach based on the spectral decomposition method [16–20] is enabled due to the

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unique separation of the up- and downstream sound fields. Hence, instead of formulating a computer algorithm to extract the sound source or the acoustic response of the system, the properties of the experimental facility are exploited. Additionally, the inherent acoustic properties of the radiator are investigated from flow induced noise (FIN) and transmission loss (TL) measurements. Unfortunately, radiator characteristics obtained from standardized methods fail to provide reliable information in the low frequency range. This motivated the authors to develop a new method suitable for frequencies having wavelengths much larger than the dimensions of the radiator.

This paper is arranged in the following way: Section 2 presents the theory on which the analysis is founded. In Section 3, the components constituting the cooling system are briefly described followed by the experimental techniques adopted for determining the acoustic properties of the individual components together with the complete cooling package in Section 4. Section 5 summarizes the most important results. Finally, conclusions concerning the findings are stated in Section 6.

2. Spectral decomposition

The SPL measured outside a ducted fan does not solely represent the spectral distribution of the fan but instead the combined effect of the fan and the acoustic properties (scattering and reflection) of the surrounding system. To facilitate the study of such installations, it is beneficial to isolate the effect of each contribution. For ducted fans, this can be handled through the method of spectral decomposition [16–20]. Following the formulation of Canepa et al. [20], the far-field one-sided power spectral density is given by

$$S_{pp}(f,\Omega,D_{tip},r,\varphi,\Theta)=\frac{\rho_0^2\Omega^3D_{tip}^6}{r^2}Ma^{\alpha}Re^{\beta}F^2(St,\varphi)G^2(He,\varphi,r/D_{tip},\Theta) \tag{1}$$

where f is the frequency, Ω is the rotational speed, D_{tip} is the impeller tip diameter, r is the distance to the receiver and Θ is the inclination angle, measured from the rotational axis. The non-dimensional quantities, namely the Mach number Ma, Strouhal number St, Helmholtz number He, Reynolds number Re and flow coefficient φ are

$$\begin{aligned}
Ma &= \Omega D_{tip}/2a_0\\
St &= f/f_{BPF}\\
He &= fD_{tip}/a_0\\
Re &= \Omega D_{tip}^2/2\nu\\
\varphi &= 8Q/\pi\Omega D_{tip}^3
\end{aligned} \tag{2}$$

where f_{BPF} denotes the blade passing frequency, a_0 is the speed of sound, ν is the kinematic viscosity and Q is the volume flow rate. The source term $F(St,\varphi)$ is purely aerodynamic, describing sound source mechanisms related to pressure and velocity fluctuations within the flow region. It is independent of propagation effects which are described by the product $Ma^{\alpha}G^2(He,\varphi,r/D_{tip},\Theta)$. The term Ma^{α} is associated with aeroacoustic radiation efficiency and depends exclusively on the source properties (acoustic compactness and coherence of the source). The acoustic response function $G(He,\varphi,r/D_{tip},\Theta)$ describes the acoustic properties of the cooling unit and all solid objects scattering the sound field. Besides a strong dependency on the receiver's position, a possible change in source position is given through the dependency on flow coefficient φ. Both functions F and G are made non-dimensional in the formulation. The SPL spectrum can be approximated from Eq. (1) for a small frequency band Δf

$$L_p(f,\Omega)\cong 10\log_{10}\frac{S_{pp}(f,\Omega,D_{tip},r,\Theta,\varphi)\Delta f}{p_{ref}^2} \tag{3}$$

where p_{ref} = 20 μPa. The effect of Re on the generated noise is negligible for low-Mach number fans ($\beta = 0$), see e.g. [17,18]. Additionally, for a fixed receiver position and constant flow coefficient, F and G will be functions of St and He, respectively. Hence, an expression for the two functions in Eq. (1) is given as

$$\begin{aligned}
20\log_{10}F(St)+20\log_{10}G(He) &= L_p(f,\Omega)-10\log_{10}\left(\Omega^{3+\alpha}\right)-C(\alpha)\\
C(\alpha) &= 10\log_{10}\left[\frac{D_{tip}^{6+\alpha}\rho_0^2\Delta f}{2^{\alpha}a_0^2r^2p_{ref}^2}\right]
\end{aligned} \tag{4}$$

The left-hand side represents unknown terms while all terms on the right-hand side are known from experiments. The separation is feasible as the terms on the left-hand side are inherently related to two different frequency scales, St and He. In principle, in the absence of propagation effects, e.g., when the source radiates into a reverberation room, the spectra scaled with $10\log_{10}\Omega^{3+\alpha}$ and plotted versus St, should collapse on a single curve. This aspect is also discussed later in more detail in Section 5.4.

3. Description of the cooling module components

In order to simplify the study of features of noise emanating from a low-speed fan, the number of components is reduced in relation to a realistic underhood compartment. The modular test section consisted of an automotive radiator, a shroud, a fan, a frame and a hydraulic motor; see Fig. 1 left to right.

To the left in the figure, the modular assembly is shown followed by each component separated. All parts come from a Scania bus cooling installation. The modular set-up enables separate measurements of the different components, as well as measurements of the complete setup. The radiator is of parallel-plate type and has a frontal area of approximately 0.60 m^2 (0.85 m × 0.70 m). The cooling operation of the radiator was not in use, that is, the radiator acted as a passive device during the measurements. The radiator has a thickness of 50 mm and the distance to the fan is 80 mm. As in realistic applications, the shroud is mounted closely to the radiator in order to guide the flow towards the fan. Two different fans are tested where one is common for truck applications and one is common for buses. Both are 750 mm in diameter, have a blade chord ranging from 0.14 m to 0.16 m and are equipped with 11 blades but differ in shape; see Fig. 2. The tip clearance is 5 mm. The fans were driven by a hydraulic motor with a maximum pressure of approximately 230 bar.

4. Experimental setup

The measurements comprise acoustic properties like SPL, sound power level (SWL), flow induced noise (FIN), transmission loss (TL) and insertion loss (IL). The modular test section is mounted in the wall between an anechoic and a reverberation room; see Fig. 3. The air was sucked by the fan, from the anechoic room through the radiator into the reverberation room. The setup enabled separation of the radiated sound up- and downstream in accordance with the installation in a heavy vehicle application where the outside is separated from the engine bay. The wall where the fan is mounted has an area of 9 m^2 (3 m × 3 m). To minimize the inflow disturbances to the radiator, the inlet side opening is designed with rounded edges (radius 50 mm) i.e. around the perimeter of the radiator; see Fig. 3(a).