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Acoustic signature of flow instabilities in radial compressors

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A B S T R A C T

Rotating stall and surge are flow instabilities contributing to the acoustic noise generated in centrifugal compressors at low mass flow rates. Their acoustic generation mechanisms are exposed employing compressible Large Eddy Simulations (LES). The LES data are used for calculating the dominant acoustic sources emerging at low mass flow rates. They give the inhomogeneous character of the Ffowcs Williams and Hawkings (FW-H) wave equation. The blade loading term associated with the unsteady pressure loads developed on solid surfaces (dipole in character) is found to be the major contributor to the aerodynamically generated noise at low mass flow rates. The acoustic source due to the velocity variations and compressibility effects (quadrupole in character) as well as the acoustic source caused by the displacement of the fluid due to the accelerations of the solid surfaces (monopole in character) were found to be not as dominant. We show that the acoustic source associated with surge is generated by the pressure oscillation, which is governed by the tip leakage flow. The vortical structures of rotating stall are interacting with the impeller. These manipulate the flow incidence angles and cause thereby unsteady blade loading towards the discharge. A low-pressure sink between 4 and 6 o'clock causes a halving of the perturbation frequencies at low mass flow rates operating conditions. From two point space-time cross correlation analysis based on circumferential velocity in the diffuser it was found that the rotating stall cell propagation speed increases locally in the low pressure zone under the volute tongue. It was also found that rotating stall can coexist with surge operating condition, but the feature is then seen to operate over a broader frequency interval.

1. Introduction

Downsized reciprocating internal combustion engines (ICE) in combination with turbocharging play an important role increasing the energetic efficiency and reducing emission levels. In the wide range of engine operating conditions, the turbocharger compressor can become a very important noise contributor [1]. This audible discomfort is caused by the acoustic pressure fluctuations, with the origins in the unsteady pressure loads, some of which driven by the developed flow instabilities interacting with the moving or stationary solid surfaces in the compressor [2,3].

Rotating stall and surge instabilities may occur during normal driving conditions [4], e.g. gear shifting or sudden acceleration. Experimental measurements by Kabral and Åbom [5] provides scientific evidence that these off-design flow instabilities cause

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substantial sound pressure levels.

Without a complete description of the temporal and spatial evolution of the flow instabilities and the unsteady pressure loads on solid surfaces there is a lack of knowledge regarding the sound generation mechanism. This restricts the optimization of compressors with respect to acoustic performance [6,7]. Rämmal and Åbom [8] reported that such sources can be categorized according to their characteristics and are commonly referred to as monopoles, dipoles, and quadrupoles. A monopole source is associated with temporal variations caused by motion of a discontinuity forcing a net volume flow (e.g. impeller blade surface). A dipole source corresponds to pressure fluctuations on a solid surface induced by e.g. flow structure interaction with the surface or flow separation. Quadrupole sources originate from turbulent fluctuations or by varying tangential shear stresses on surfaces.

Several distinct acoustic noise features have been reported in the literature. However, a quantification of the acoustic sources has not been carried out. A distinct mode is associated with the blade passing frequency (BPF), i.e. the rotating order times the number of impeller blades. Narrowbanded tip clearance noise has been reported by Ref. [9] to dominate over the blade passing tones. It has been reported to occur at approximately 50% of the rotating order frequency (RO) [10] for some compressor designs and may be more prominent at lower impeller speeds. Additionally, it has been suggested to emanate from secondary flow motion in the gap between the compressor casing and the impeller blades similar to axial compressors. Moreover [10] hypothesized for an existence of a relationship between rotating flow instability and tip clearance noise. However, in the work by Ref. [11] the tip clearance size is reported to have negligible effect on the noise generation at off-design operating conditions. Mendonca et al. [12] found that the occurrence of stalled impeller blades is related to a rotating instability and with associated amplified noise levels. The presence of narrowbanded noise at higher harmonics to the shaft rotational speed has been reported by Ref. [13]. Another noise source, evident at near-surge line operating conditions, is coined “whoosh noise” [14,15], stretching over several orders of kHz. It was observed that the “whoosh noise” is more apparent at near-surge operating conditions than at actual deep-surge operating conditions, see Refs. [2,14].

Karim et al. [16] related high flow incident angles at the leading edges of the impeller with noise generation and suggested that rotating flow structures in the blade passages are causing the so called “whoosh noise”. In contrast [14], observed the main noise source to be localized further downstream in the compressor outlet piping. All these studies reveal the challenge associated with connecting the perceived acoustic noise with the actual generation mechanism. One of the best-known descriptions for noise assessment of rotating devices such as impellers is the Gutin analogy [17]. An alternative theory is according to [18]. Both models have in common that the net force produced by each blade is assumed constant. The resulting force distribution yields a source signal with a dipole character. It consists of the fundamental blade passing frequency including higher harmonics. With simplifying assumptions such as compact source field, the analogy suggests a high directionality of the far-field sound, i.e. zero intensity perpendicular and co-planar to the impeller disc. From observations, this never happens, because effects from unsteady flow disturbances will tend to manifest in a broadband spectrum. Therefore, a deterministic acoustic model is not completely satisfactory to describe the emitted noise for centrifugal compressors. An obvious shortcoming is the inability of the reduced order models to account for sound associated with low-frequency tonalities. In fact most of the audible frequency range is unaddressed. More predictive capabilities were demonstrated in Ref. [11] using unsteady Detached Eddy Simulation (DES). The computed pressure fluctuation spectra were reported to show good trends with experimental data.

The current work is looking to reveal the interconnectivity between the flow field instabilities, the acoustic sources, and the observed emitted sound levels. High-fidelity compressible Large Eddy Simulation (LES) calculations are employed in order to quantify the developing flow instabilities as the compressor operating conditions approach the surge line. This methodology, i.e. LES has also demonstrated predictive quality for a range of operating conditions as compared to probe point spectra obtained experimentally, see Refs. [19,20]. Acoustic sources are determined using the Ffowcs Williams-Hawkings (FW-H) equation [21]. The distribution of the acoustic source signatures are extracted using modal decomposition methods. Further, the acoustic sources are analyzed and linked with the generating flow structures. From the theoretical stand point of view Howe [22] derive that radiated power due to pulsating pipe flow is proportional to:

$$\frac{\overline{W}_m}{\rho_0 U^3 D^2} \propto M \tag{1}$$

where ρ_0 - air density, U - mean flow speed, D - pipe diameter, c_0 - sound speed, M - Mach number, and subscript m refers to a monopole source. The expression in Eq. (1) is obtained by assuming that the frequency of the monopole source is proportional to U/D. Moreover, the power is normalized with $\rho_0 U^3 D^2$ so that radiation efficiency is only proportional to the Mach number. The rotating blade forces can be represented as a dipole radiation. The radiation efficiency from a dipole source is also derived in Howe [22] shown to be proportional to the cube of the Mach number:

$$\frac{\overline{W}_d}{\rho_0 U^3 D^2} \propto M^3 \tag{2}$$

where subscript d refers to a dipole. The Lighthill stress tensor behaves as a quadrupole acoustic source. For this Howe derives that the quadrupole radiated power scales as:

$$\frac{\overline{W}_q}{\rho_0 U^3 D^2} \propto M^5 \tag{3}$$

where the quadrupole is indicated with subscript q. It needs to be emphasized that Eqs. (1)–(3) relates only to the effectiveness of the source type with respect to the Mach number. It does not convey the actual magnitude of the source in the present