AERODYNAMIC AND ACOUSTIC STUDIES ON A RADIAL FAN FAMILY DEVELOPED FOR INCREASED SPECIFIC FLOW RATE OF DUST-LADEN GASES

B. Kocsis\textsuperscript{1} - T. Benedek\textsuperscript{1} - P. Ferenczy\textsuperscript{2} - E. Balla\textsuperscript{1} - G. Daku\textsuperscript{1} - J. Vad\textsuperscript{1}

\textsuperscript{1} Department of Fluid Mechanics, Faculty of Mechanical Engineering, Budapest University of Technology and Economics, Bertalan Lajos u. 4 – 6., H-1111 Budapest, Hungary;
Email: vad.janos@gpk.bme.hu

\textsuperscript{2} Szellőző Művek Kft., Építész utca 8-12., H-1116 Budapest, Hungary.

ABSTRACT

The paper presents the further development of a new radial flow fan family designed for an increased resistance against the particles of solid contaminants, and for an increased specific flow rate. The rotors of these fans are equipped with straight backward-leaned blades. A CFD-aided iterative design process has served for ensuring fan efficiency meeting the legislative requirements. Measurement-based loading, efficiency and power curves are presented on surveying how rotor widening can further increase the flow delivering capability of the fans. Industrial on-site Phased Array Microphone experiments are reported on the acoustic features of the new fans, as contribution to product development. Based on these tests, preliminary studies on the potential of acoustics-based fan condition monitoring are discussed.

KEYWORDS

CONDITION MONITORING, DUST-LADEN GAS, FAN NOISE, HIGH SPECIFIC FLOW RATE, RADIAL FLOW FAN

NOMENCLATURE

Latin letters
\begin{itemize}
  \item $A$ lateral size of pressure port [m] (Fig. 2)
  \item $a$ size parameter [m] for drawing the casing spiral (Fig. 2)
  \item $B$ fan blade width (axial extension) [m]
  \item $D$ fan rotor diameter [m]
  \item $d$ distance [m] between rotor backplate and casing wall (Fig. 2)
  \item $f_{BP}$ blade passing frequency [Hz] = $f_R \cdot N$
  \item $f_R$ rotational frequency of fan impeller [Hz]
  \item $L_p$ overall sound pressure level [dB]
  \item $L_{p,n}$ sound pressure level in narrowband [dB]
  \item $L_{p,\text{oct}}$ sound pressure level in octave band [dB]
  \item $L_{p,\text{oct/3}}$ sound pressure level in third-octave band [dB]
  \item $L_{W,\text{oct}}$ sound power level in octave band [dB]
  \item $N$ rotor blade count [-]; $N = 12$ considered herein
  \item $n$ rotor speed [1/s]
  \item $P$ shaft input power [W]
\end{itemize}
\[ \Delta p_t \text{ total pressure rise [Pa]} \]
\[ q_v \text{ volume flow rate [m}^3{/s]} \]
\[ R_t \text{ radii [m] for drawing the casing spiral (Fig. 2)} \]
\[ R_T \text{ tongue radius [m] (Fig. 2)} \]
\[ u \text{ tangential flow velocity [m/s]} \]
\[ u_t \text{ rotor tip circumferential velocity [m/s]} = D \cdot \pi \cdot n \]
\[ W \text{ casing width [m] (Fig. 2)} \]

**Greek letters**

\[ \eta \text{ total efficiency [-]} = q_v \cdot \Delta p_t / \rho \]
\[ \lambda \text{ power coefficient [-]} = P / \{(D^2 \cdot \pi / 4) \cdot u_t \cdot (\rho \cdot u_t^2 / 2)\} = \Phi \cdot \Psi / \eta \]
\[ \rho \text{ fluid density [kg/m}^3\text{]} \]
\[ \sigma \text{ speed factor [-]} = \Phi^{0.5} \cdot \Psi^{-0.75} \]
\[ \Phi \text{ flow coefficient [-]} = q_v / [(D^2 \cdot \pi / 4) \cdot u_t] \]
\[ \Psi \text{ total pressure coefficient [-]} = \Delta p_t / (\rho \cdot u_t^2 / 2) \]

**Subscripts and superscripts**

1 \text{ rotor inlet} \\
2 \text{ rotor outlet (i.e. } D_2 = D \text{ for the radial fan)} \\
\* \text{ literature reference data} \\
t \text{ rotor blade tip}

**INTRODUCTION AND OBJECTIVES**

For transporting gases contaminated with solid particles, radial flow fans are frequently applied (e.g. Ghenaiet, 2021). In several cases, these fans are equipped with backward-leaned rotor blades (e.g. Carolus, 2003; Cory, 2005; Aldi et al., 2019). Such blade layout is applied to reduce the flow deflection inside the rotor for moderating deposit formation or erosion by the contaminants; and yet to ensure a relatively high pressure rise winning over the losses in the system caused by the two-phase flow itself as well as by the dust separation equipment (e.g. U.S. CSHIB, 2015). A further solution in moderating the adverse effects of solid particles is to apply straight, i.e. uncambered, backward-leaned rotor blades (e.g. RSES, 2009). As reported in the preliminary study in Ferenczy et al. (2022), Szellőző Művek Kft., termed herein as Company, has developed a new radial fan family of such kind, with increased resistance against dust load, whereas realizing an increased specific flow rate; i.e. the operational range of \( \Phi \) flow coefficient is relatively high. The development was carried out in collaboration with the Department of Fluid Mechanics (DFM), Faculty of Mechanical Engineering, Budapest University of Technology and Economics. DFM performed the preliminary design, iterative development aided by Computational Fluid Dynamics (CFD) data, and design as well as execution of measurements in the production hall of the Company. The development of the new LDL fan family (Szellőző, 2022), being yet available on the market, was necessitated by market demands non-coverable by the formerly existing fan product range of the Company.

This paper presents the continuation of collaboration between the Company and DFM, as supplements to the formerly reported work in Ferenczy et al. (2022). The paper adds to the open literature from the following perspectives. a) Extending the experimental experiences on how the specific flow rate, characterized by an \textit{a priori high} \( \Phi \) range, can be \textit{further increased} in an energy-efficient manner by widening the rotor (Gruber et al., 1978; Cory, 2005), for a more flexible response to case-specific demands on behalf of the industrial market. This initiative has been aided by additional CFD-aided iterative design campaigns, and measurements of extended range of validity. b) The Phased Array Microphone (PAM) technique is extensively used in basic research and in
laboratory tests related to fans (e.g. Krömer et al., 2019). This paper aims at extending the use of PAM to fan industrial “pilot plant” applications, installing and using PAM in the production hall of the Company, as part of the tests on sample LDL fans, aiding product development. c) Taking the benefits of PAM, obtaining new experimental experiences on the applicability of acoustics-based fan condition monitoring (CM), as a future perspective of CM (Goyal et al., 2018) in smart industrial fan technology, in accordance with the Industry 4.0 concept. From this point of view, this paper joins the trend reported in Tóth and Vad (2022) and Tóth et al. (2023).

The disclosed data documented in the paper are limited for confidentiality reasons.

**FAN DESIGN AND DEVELOPMENT**

A brief summary is given herein on the design considerations of the new fan family. Further details are reported in Ferenczy et al. (2022). As explained in the aforementioned reference, and illustrated in Figure 1, comparable axial and radial fan rotors are considered. Comparability means identical $D_t$, $n$, and $P$. It has been pointed out that the radial rotor tends to perform a lower flow rate and higher total pressure rise than the comparable axial rotor. Still, the specific flow rate, manifested by $\Phi$, was to be increased for the new fan family, necessitating an increase in both the inlet diameter and the outlet blade width, i.e. $D'_1 > D_1$ and $B'_2 > B_2$ – as shown in the sketch on the right-hand side of Fig. 1 –, causing a decrease in the blade aspect ratio $[(D_2-D_1)/2] / B_2$. The prescribed outward flow deflection within such low-aspect-ratio blading tends to increase the risk of flow separation over the front wall of the rotor; and the inclination for flow separation in the rotor is further increased by applying straight blades. The CFD-aided iterative design aimed at moderating such separation effects for obtaining acceptably high $\eta$ – conf. the Fan Regulation EU 327(2011) –, taking a benefit of boundary layer control by means of the leakage flow developing in the gap between the suction cone and the rotor front wall. Being in accordance with empirical formulae of DFM deduced using the Cordier diagram (e.g. Carolus, 2003), the fan family of $N = 12$ rotor blades was successfully designed and tested for $\Phi_0 = 0.18$ and $\Psi_0 = 1.0$. An additional comment to Ferenczy et al. (2022) is on $\Psi_0 = 1.0$. The Euler equation of turbomachines (e.g. Carolus, 2003) implies in an isentropic approach that $\Delta p_t = \rho u^2 u_t$. Therefore, the empirical $\Psi_0 = 1.0$ choice corresponds to the approximation of $u_{2D} \approx u_t/2$. This appears to represent a favorable design compromise for backward-leaning blades between the extreme circumstances of $u_{2D} = 0$ (no swirl) and $u_{2D} = u_t$ (solid body rotation), taking their average.

In preliminary fan design, the geometries of the suction cone and the rotor front wall were taken on the basis of the literature (Osborne, 1966; Gruber et al, 1978; Carolus, 2003). The preliminary design also included the empirical prescription and optimization of rounding radii of the suction cone as well as the rotor inlet section. The relatively low blade AR envisaged a separation zone over the rotor front wall of conical geometry – to be treated using CFD –, being due to the pronounced outward deflection of flow within the front region of the rotor. Another envisaged consequence of increased rotor width was the non-uniformity of velocity inlet to the blading – also to be treated in CFD campaigns. In order to win over the aforementioned adverse effects, the concept of passive boundary layer control was considered via CFD, utilizing the a priori and inevitably occurring leakage flow between the suction cone and the rotor front wall. Although such leakage flow manifests an inevitable volumetric loss, it also offers potential benefits as follows. Since the leakage flow develops in the form of a high-momentum incoming jet, it may be utilized for energizing the boundary layer at the front part of the rotor, thus moderating the extension of flow separation, and hastening flow reattachment. In order to dominate the benefits of the leakage jet, the design of geometry and size of the gap between the suction cone and the front wall of the rotor, as well as the shaping of the convergent-divergent rotor
suction cone, was aided by CFD. Applying straight rotor blades and having a reduced AR for the rotor blading represent simultaneous challenges in the design goal of achieving a reasonably high efficiency. A primary goal of CFD was to identify and remediate separated flow regions. As detailed in Ferenczy et al. (2022), via the first part of CFD-aided iterative design of the rotor blade count, blade angle, front plate, suction cone, and the gap between the latter two, the aforementioned separated regions were significantly reduced, enabling reasonably high efficiency. Figure 2 (not to scale exactly) provides illustrative approximate sketches on the preliminary design phase, showing the longitudinal and cross-sectional views of the volute casing, indicating the position of the rotor incorporated.

Figure 1: Outline of comparative axial flow – left – and radial flow – middle – fan rotors. On the right: geometrical trends for increased $\Phi_0$.

Figure 2: Sketches on the volute casing and the rotor, in longitudinal (top) and cross-sectional (bottom) views.

Figure 3: CFD domain

In order to assess the effects of increasing the blade width, the rotor outlet has been widened by 10 and 20 percent in product development. The corresponding scenarios of “1,0 $B$”, “1,1 $B$”, and “1,2 $B$” were investigated – and labelled accordingly in the paper –, where “$B$” is the originally designed
$B_2$ outlet blade width. As Fig. 1 suggests, increasing the rotor width further reduces the aspect ratio of blading, causing an additional risk of flow separation.

**CFD STUDIES**

The response to the challenges outlined in the previous section, associated with the risk of flow separation for various reasons, deserved a careful treatment and CFD-aided iterative redesign of blading. The CFD technology is reported in Ferenczy et al. (2022); only brief comments and supplements are given herein. In ANSYS Fluent software environment, the SST $k$-$\omega$ model, and the steady-state frozen rotor approach was applied. Its suitability was confirmed via making a comparison in representative scenarios with a sliding mesh model where a time step represented $1^\circ$ rotation of the impeller. The typical cell size in the rotor domain was set to $6 \text{ mm}$, which resulted in 100 elements in the spanwise and 60 elements in the chordwise direction on the blade surface. It was intended to utilize hexahedral elements. Approximately 15 million cells were applied. The $y^+$ values were kept predominantly within the range of 30...100 for the zone of the rotor. The length of the inlet duct was set to 2 times the inlet duct diameter, and the length of the outlet duct was set to 12 times the outlet duct hydraulic diameter. A fully developed duct flow velocity profile was prescribed at the inlet with an average velocity magnitude according to the flow rate. At the outlet, constant static pressure was assumed. During the CFD studies, incompressibility was assumed, and the fluid density was set to $\rho = 1.20 \text{ kg/m}^3$. The CFD domain is illustrated in Figure 3 (previous page).

In order to experimentally validate the CFD methodology, the elaborated CFD technique was applied to the commercially available LDH fan family of the Company, for which the Company is in possession of detailed data of characteristic and efficiency curve measurements. The LDH fans are relevant from the perspective of the CFD validation, thanks to their following features being similar to the LDL fan family presented herein: same basic construction (volute casing, suction cone, conical front plate of rotor); radial fan rotor with backward-leaned blades; straight blades suited to dust load. The CFD technique has been considered valid on the basis of the following experiences. i) The CFD tool was capable for predicting $\eta$ at the best efficiency point within the range of measurement uncertainty. ii) In the vicinity of the best efficiency point, the CFD tool was able to predict the $\mathcal{P}$ values as well as the slope of the $\mathcal{P}(\Phi)$ characteristic curve mostly within the range of experimental uncertainty.

**Figure 4** presents illustrative exemplary details about the rotor mesh. At the top, enlarged views of the mesh near the connections of blade leading edge / rotor front plate and blade trailing edge / rotor backplate are shown. At the bottom, a segment of the mesh over the backplate, incorporating the refined mesh near the blades, is presented. Flow separation is most pronounced over the blade suction side and the front plate. The mesh was properly refined in the vicinity of these surfaces, as illustrated in the figure.

**Figure 5** shows representative CFD plots of radial velocity for the various rotor widths for intermediate fan development steps. In the upper row of the CFD plots, related to the meridional plane, separation zones are indicated by arrows, demonstrating that the flow does not follow the shape of the conical front plate of the rotor but is detached from that, forming stagnant fluid regions. Contrary to the healthy radially outward flow, featuring high positive (i.e. outward) radial velocity, such stagnant fluid regions are characterized by near-zero or even negative radial velocity (i.e. reverse flow toward the axis of rotation). Such attributes of separation zones are also visible in the lower row of the CFD plots, especially near the suction side of the blade tips. As the figure and its caption suggests, increasing the rotor width tends to increase indeed the flow transporting capability of the
blading. Flow separation zones inevitably occur over the rotor front wall as well as in certain regions over the suction side of the straight blades; but such separation can be moderated in a purposeful iterative design of the blading, also exploiting the boundary layer refreshing capabilities of the leakage flow entering in the gap between the suction cone and the rotor inlet.

Figure 4: CFD mesh details. Top left: near the connection of the blade leading edge and the rotor front plate. Top right: near the connection of the blade trailing edge and the rotor backplate. Bottom: over the rotor backplate.

Figure 5: CFD radial velocity plots. From left to right: [1,0 B and $\Phi = 0.200$]; [1,1 B and $\Phi = 0.207$]; [1,2 B and $\Phi = 0.214$]. Upper row: meridional plane. Lower row: rotor cross-section in the plane at half of outlet blade width.
AERODYNAMIC MEASUREMENTS

A test fan of the new LDL fan family was manufactured with the following characteristics: $D_2 = 630$ mm, nominal rotor speed $n = 1470$ 1/min, direct-driven by an 11 kW 4-pole asynchronous electric motor. In order to experimentally justify the effects of rotor widening, representative sample rotors of $1.0 B$, $1.1 B$, and $1.2 B$ were manufactured. The experimental technique reported in Ferenczy et al. (2022) has been supplemented with the possibility of extending toward higher flow rates in the measurements for obtaining the loading, efficiency, and power curves. For this purpose, Pitot-static probe traversing was incorporated for pressure rise and flow rate measurements, with consideration of the guidelines documented in standard ISO:5801(2017). The test rig selected is a type “C” (ISO:5801) ducted-inlet, free-outlet configuration. With knowledge of the gas density, volume flow rate, and cross-section of the pressure-side port of the fan, the outlet dynamic pressure can be approximated, and thus, estimation can be given to the total pressure rise $\Delta p_t$ and total efficiency $\eta$, as presented in this paper. Barometric pressure, air temperature, rotor speed, and electric input power to the driving motor were measured, as reported in Ferenczy et al. (2022).

Figure 6 presents the measurement data. For clarity, $5^{th}$ order polynomial trend lines were fitted to the experimental data points using the least-squares method, and these trend lines are shown in the figure. The absolute experimental uncertainty of the data in Fig. 6 is conservatively estimated at 95 % confidence level as follows, in the ranges surrounding the best efficiency points: for $\Phi$ ± 0.003; for $\Psi$ ± 0.01; for $\eta$ ± 0.03; for $\lambda$ ± 0.01. For a clear and well-visible presentation of the trends in Fig. 6, without any mismatching and masking effects, the symbols of measurement data were omitted from the graphs. It is noted that the measurement data points scatter in the vicinity of the trend lines mostly within the experimental uncertainty ranges reported above. The changes due to varying the rotor width are in fair quantitative agreement with the empirical guidelines in Gruber et al. (1978). The figure demonstrates the flow-increasing capability of widening the rotor: at a fixed $\Psi$ abscissa value, $\Phi$ tends to increase with $B$. An inevitable consequence of increasing $B$ is the deterioration of efficiency. However, with careful CFD-aided redesign as outlined before, the realized $\eta$ values are still in accordance with the efficiency requirements set in the Fan Regulation EU 327 (2011), and the fans can be utilized at reasonably high efficiencies over fairly broad operational ranges. The $\lambda(\Phi)$ power curves suggest the possibility of choosing a drive operating safely, without an overload.

PHASED ARRAY MICROPHONE STUDIES

The advanced acoustic experimental methodology of beamforming, used otherwise in laboratory studies (e.g. Benedek et al., 2022), was applied in the production hall of the Company. The benefits of beamforming were utilized for enabling noise source localization despite the acoustically contaminated environment. Such industrial environment provides an adequate venue for preliminary studies to acoustics-based CM that is to be applicable under non-laboratory but real industrial circumstances.

In what follows, two operational states of the aerodynamically tested sample fans are discussed, termed as “Nominal” and “Throttled” states, and characterized by $\Phi=0.18$ and $\Phi=0.08$, respectively. The Nominal state corresponds to the design point – being also the best efficiency point – of the original fan design for $1.0 B$, as confirmed in Fig. 6, and reported in Ferenczy et al. (2022). The Throttled state represents flow rate reduction below the peak pressure point. From fan CM point of view, being discussed in the next section, the Nominal state represents a normal, desirable scenario. Contrarily, the Throttled state imitates an abnormal scenario of fan overthrottling, occurring e.g. due to erroneous system design underestimating the losses, or due to control anomalies. Such abnormal
scenarios are to be either avoided or to be recognized and eliminated e.g. with the aid of acoustics-based CM.

The acoustic tests presented in the following sections are related to $n = 1470$ 1/min. The characteristic frequencies playing a major role in the discussion are $f_R = 24.5$ Hz and $f_{BP} = 294$ Hz.

For extending the experiences on PAM applications in industrial environments, a planar microphone array consisting of 24 miniature microphones placed in logarithmic spirals, manufactured by Optinav (2021), has been installed and applied to the sample fans. This PAM is extensively used at DFM for fan research and development (e.g. Benedek et al., 2022). The PAM has been placed perpendicularly to the rotor axis, at a distance of $3.2D$, i.e. 2.0 m, from the rotor mid-plane. The data acquisition has taken 30 s for each measurement, with a sampling rate of 44100 Hz. Beamforming has been carried out in the Beamform Interactive plugin of the ImageJ software (Optinav, 2021), in one-third octave bands between 111 and 17888 Hz, using the delay-and-sum [Images a), b), and g) of Figure 7] as well as the TIDY [Images c) to f) of Figure 7] algorithm. Beamforming was performed in a stationary reference frame.

In supplementary studies, beamforming spectra were produced, containing the peak beamforming values for the frequency bins under consideration. In three-octave-band representation, the absolute value of difference between the beamforming spectra and the one-microphone spectra in Figs. 9 and 11 to 13 (see later) was less than 1 dB in most cases. Even considering the maximum deviations, they fell well within the variance range presented in Fig. 9. On this basis, the authors judged sufficient and reasonable to include the above listed spectral distributions, and to omit the presentation of beamforming spectra for the purpose of this paper.

Figure 6: Measurement-based dimensionless loading, efficiency, and power curves for various rotor widths.
Figure 7 shows exemplary beamform maps as results of the PAM campaign. The beamform maps are plotted over photographs of the measured scene, which were taken with a camera installed in the center of the microphone array. In the photographs in Figs. 7 and 8, a grey rubber sheet can be seen above the drive motor. This sheet provided temporary insulation to the electric box located at the top of the motor, the cover of which was removed, in order to provide access for the electric input power meter in the aerodynamic experiments. The rotor width, operational state, and frequency range associated with each map are specified above the maps. The discussion refers to the labels a) to g) of the maps. a) represents the beamform image of broadband outlet noise, manifested by a noise spot at the outlet section. The presence of the outflow jet raises the suspicion that the microphones detect not outlet noise but aerodynamic pseudo-sound. In the investigated cases, however, the PAM was located at a distance from the jet being sufficiently large (see above) for avoiding the exposure of the microphones to any pressure fluctuations originating from the jet flow. Beamforming performed for the various scenarios – not only for the presented one – consequently and repeatably localized this noise spot to the outlet section, i.e. without any randomness that could be dedicated to pressure fluctuations generated by the jet. The above, supplemented by the authors’ physical perception by audition on the site, confirms that Image a) presents the spot of outlet noise indeed. b) shows an example in which pronounced noise is generated at the door of the access hatch configured on the volute casing, and this noise exceeds the outlet noise in the studied frequency range (small noise spot at about “half past one o’clock” at the periphery of casing). This draws the attention to the need of properly sealing the door, in order to avoid whistling noise due to outward air leakage, and to properly tighten the door for avoiding additional vibration and the noise associated. Upon a possible future demand of evaluation, the distinction between vibrational and aeroacoustic noise sources can be made through comprehensive mechanical analysis and the evaluation of typical vibration patterns, being beyond the scope of the present paper. c) → d) → e) present the noise localized to the volute tongue in the sequence of increasing rotor width. The noise (spots at about “half past ten o’clock” near the volute tongue region) appears in the narrow band incorporating \( f_{\text{BP}} = 294 \) Hz, i.e. it is the acoustic signature of rotor-stator interaction between the rotor blades and the tongue. It increases with rotor width (see the scales), i.e. it is a sort of fingerprint of the given construction. Its detection is promising from acoustics-based CM point of view since an internal flow phenomenon, i.e. rotor-stator interaction, manifests itself in a localized manner in the noise pattern recorded from outside, via the casing wall. c) → f) demonstrate that the rotor-stator interaction noise, localized to the near-tongue region, tends to increase with throttling. This outlines a potential to identify the throttling state of a fan in acoustics-based CM by inspecting the frequency band in the vicinity of \( f_{\text{BP}} \). The increased noise near \( f_{\text{BP}} \) due to increasing the pressure rise is in accordance with VDI 3731-2 (1990) [there: “middle-pressure” → “high-pressure” fans]. g) demonstrates that the noise related to the cooling fan of the driving electric motor dominates in the studied frequency range (noise spot at the perforated cover of the motor). The noise spot does not extend to the entirety of the perforated cover but is shifted above the center of rotation. This can be dedicated to the following: i) Asymmetry in inflow conditions to the cooling fan: the inflow from below is restricted by the motor base as well as by the vicinity of the ground, whereas free inflow occurs from above from a practically unlimited space. ii) Apparent geometrical asymmetries in the motor / cooling fan / perforated cover assembly. iii) Spatial uncertainty in the source localization capabilities of the beamforming technique.
Figure 7: Representative scenarios from the PAM measurements
PRELIMINARY STUDIES IN ACOUSTICS-BASED CONDITION MONITORING

Further developing the experiences gathered with the PAM, it is investigated herein how the acoustic signal obtainable by a single microphone can be exploited for CM purposes. Except for Fig. 10 (see the explanation there), the sound pressures for each spectral distribution are non-dimensionalised using the same reference value purposefully customized to the actual comparison, and are presented in level form. Figure 8 illustrates the location of the individual PAM microphones relative to the fan over the PAM plane. Figure 9 shows an example for the variance of spectra delivered by the individual microphones in Fig. 8, as function of microphone location. The levels obtained using the various microphones are indicated with hairline segments within each band, represented by various colors on the grayscale assigned to the microphones. The minima and maxima are shown using thicker black lines. In accordance with Tóth and Vad (2022), the figure draws the attention that, when designing acoustics-based CM using e.g. a single microphone, the placement of this microphone is to be purposefully selected, and such choice can be effectively aided by using PAM, a priori containing an extensive set of microphones for such sensitivity studies. For example, if inspection of rotor-stator interaction noise deserves an increased attention in CM – as inspired by Fig. 7 –, Fig. 9 demonstrates that the detectability of the band containing \( f_{BP} = 294 \) Hz has an approx. 9 dB variance as function of microphone location; and therefore, it promises the possibility of selecting a microphone placement being the most optimum for this purpose.

![Figure 8: Position of PAM microphones relative to the fan.](image)

![Figure 9: 1,0 B; Throttled state, \( L_{p \text{oct}/3} \) [dB]. An example for the sensitivity of detected spectra to microphone location.](image)

For the following discussion, the signal by microphone No. 18 in Fig. 8 has been selected for further processing, since this microphone is located the closest to the dominant noise source of outward flow. The microphone data was planned to be compared to literature reference data \( L_{Woct}^{*} \), and VDI 3731-2 (1990) provides such data on fan noise. The comparable \( L_{Woct}^{*} \) data are obtained as follows, conf. Tóth et al. (2023). Spectral data originating from the Nominal state, i.e. normal scenario, can be compared with empirical spectra from VDI 3731-2 (1990) obtained for fans operating near their best-efficiency point, i.e. representing also aerodynamically normal states. Since the speed
factor is $\sigma = 0.42$ for the fan under investigation (Ferenczy et al., 2022), it can be classified as a middle-pressure fan within VDI 3731-2 (1990) [Eq. (10), Fig. 6], and the $L_{\text{W}^{*}}$ spectrum can be obtained accordingly. Figure 10 shows the comparison. It is to be emphasized that only the shapes of the $L_{\text{W}^{*}}$ and $L_{p \text{oct}}$ spectra are to be compared for CM purposes, and therefore, they are allowed to be normalized in conjunction with each other; i.e. may be purposefully shifted along the dB scale. In accordance with the method in Tóth et al. (2023), the maximum of $L_{\text{W}^{*}}$, located within the octave band incorporating $f_R = 24,5$ Hz, was set to 0 dB, and so was set the level of $L_{p \text{oct}}$ for the same band as well (conf. Fig. 10). The agreement between the two graphs is fair, in the view that the standard deviation of $\approx 3$ dB specified for the empirical $L_{\text{W}^{*}}$ formula in VDI 3731-2 (1990) can be regarded as its uncertainty. This agreement suggests that the VDI database has a potential to be used as reference characterizing normal cases in CM. The level of the octave band incorporating $f_{BP} = 294$ Hz is outstanding, being in accordance with Fig. 7. At higher frequencies, the measured spectrum falls below the reference spectrum. This is dedicated to the fact that the noise is recorded through the flat side wall of the fan casing. On the basis of VDI 2081-1 (2001) [Chapter 6.1], the sound insulation of the flat wall increases at higher frequencies.

**Figure 10: 1.0 $B$; Nominal state. A comparison between measured $L_{p \text{oct}}$ (ladder-type solid line) and $L_{\text{W}^{*}}$ reported in VDI 3731-2 (1990) for middle-pressure fans (dashed line) [dB], in normalized representation.**

**Figure 11: 1.0 $B$; Nominal state. Spectra in narrowband, third-octave-band and octave-band data representation [dB].**

It is investigated in Figure 11 whether narrower-band representations of measurement data in Fig. 10 can aid the recognition of phenomena being of significance from CM point of view. Such investigation gives an aid to economization of data acquisition in future CM. The figure shows the outstanding narrowband peak at $f_{BP} = 294$ Hz but, apart from this, gives no evidence for the necessity to make narrowband refinements in the data representation.

**Figure 12** shows the spectra for the Throttled case in various resolution. One may presume that the octave band incorporating $f_R = 24,5$ Hz is outstanding e.g. due to an apparent rotor imbalance.
performing mechanical excitation on the entire construction, and thus generating noise. Nevertheless, the narrower-band representations show that the outstanding character appears at a frequency being higher than \( f_R \), at about 35 Hz. N. b. this peak appears, although at moderate level, also for the Nominal state in Fig. 11. This envisages the occasional necessity of applying data of resolution being higher than an octave-band one in CM, in order to avoid misleading evaluation. The narrowband peak at \( f_{BP} = 294 \) Hz appears also here, with an amplitude being higher than that for the Nominal case.

**Figure 13** presents a comparison between the Nominal and Throttled cases, in third-octave-band representation. It suggests that the Throttled state has an increased noise emission than the Nominal state. Indeed, the overall sound pressure level obtained from the measurements was \( \approx 4 \) dB higher. This is in quantitative accordance with VDI 2081-1 (2001) [Fig. 13] extrapolating the noise increment due to fan throttling indicated in the figure to the flow rate ratio of \( \Phi / \Phi_D = 0.08/0.18 = 0.44 \). Fig. 13 demonstrates the formerly discussed peak in between 30 \( \text{Hz} \)…40 \( \text{Hz} \), and an increase in the amplitude in the band enclosing \( f_{BP} = 294 \) Hz. Both of these can be regarded as signatures of abnormal operation in CM for this specific case study, imitated herein with the Throttled state. According to the figure, the third-octave-band representation appears to be of sufficient resolution for discovering such signatures.

![Figure 12: 1,0 B; Throttled state. Spectra in narrowband, third-octave-band and octave-band data representation [dB].](image1)

![Figure 13: 1,0 B. Comparison of the two operational states in third-octave-band representation [dB].](image2)

**CONCLUSIONS AND FUTURE REMARKS**

The further development related to the new LDL fan family of Szellőző Művek Kft., and an extension of its experimental studies, involving aerodynamic measurements as well as acoustic investigations, is reported herein. The paper adds to the open literature in the following topics.

1. The customized fan design, aiming at performing a relatively high specific flow rate, results in a moderate aspect ratio of rotor blading. As instructed by the Company, the rotor was designed with straight, i.e. uncambered, backward-leaned blades, in order to moderating either deposit formation or erosion of the blading due to solid contaminants. As a potential means for further increasing the specific flow rate, widening of the rotor was considered. All of the three
The aforementioned items tend to hasten flow separation within the rotor. A CFD-aided iterative redesign methodology was applied for moderation of such adverse effects, with the purposeful involvement of boundary layer-refreshing capabilities of the leakage inflow to the rotor, developing in the gap between the suction cone and rotor inlet section. Measurements on the datum fan as well as on ones with 10% and 20% widened rotors demonstrated the success of redesign, showing the gain in aerodynamic performance, whereas keeping the efficiency at levels being sufficiently high to correspond to the EU 327-2011 Fan Regulation. Measurement-based loading, efficiency and power curves were presented.

2. Phased Array Microphone (PAM) measurements were carried out on the sample fans, under “pilot plant” circumstances, i.e. in the production hall of the Company, demonstrating the capability of the PAM technique and extending the related experiences in an industrial acoustic environment. Noise related to the outflow, to the door of the access hatch on the volute casing, to the cooling fan of the driving electric motor, and to rotor-stator interaction related to the volute were localized, as a basis for guidelines in fan noise reduction. From acoustics-based condition monitoring (CM) point of view, it is promising that the noise of rotor-stator interaction, being an internal flow phenomenon, was possible to detect externally, and was found to increase with rotor width and with increased throttling.

3. Preliminary studies were carried out from the perspective of acoustics-based CM, in the representative Nominal and Throttled states, corresponding to normal and abnormal operation, respectively. A sensitivity analysis was made on the effect of microphone location to the detected spectra. Literature reference data from VDI 3731-2 (1990) were involved for comparison, and were found to represent the normal condition fairly well. The comparative evaluation between the two operational scenarios was carried out over octave-band, third-octave-band, and narrowband spectra. Remarkable signatures of the abnormal state were recognized in the third-octave spectra. This adds to the experimental database for future establishment of acoustics-based fan CM in case-specific applications.

4. The acoustic study presented herein is planned to be supplemented with comprehensive CFD campaigns, as e.g. in Benedek et al. (2022), as well as by a comprehensive mechanical analysis for making a distinction between vibrational and aeroacoustic noise sources. The effectiveness of the acoustics-based CM technique of future development can be enhanced e.g. by involving deep neural networks, as summarized in Tóth et al. (2023).

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