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Numerical and Experimental Study on the Tonal Noise Generation of a Radial Fan

The main focus of this work is on the geometrical modifications can be made to the fan wheel and to the volute tongue of a radial fan to reduce the tonal noise. The experimental measurements are performed by using the in-duct method in accordance with ISO 5136. In addition to the experimental measurements, CFD (computational fluid dynamics) and CAA (computational aeroacoustics) simulations are carried out to investigate the effects of different modifications on noise and performance of the fan. It is shown that by modifying the blade outlet angle, the tonal noise of the fan can be reduced without impairing its aerodynamic performance. Moreover, it is indicated that increasing the number of blades leads to a significant reduction in the tonal noise and also an improvement in the aerodynamic performance. However, this trend is only valid up to a certain number of blades, and a further increment might reduce the aerodynamic performance of the fan. Besides modifying the impeller geometry, new volute tongues are designed and tested on the rig. It is demonstrated that the shape of the volute tongue plays an important role in the tonal noise generation of the fan. Moreover, in order to find out whether or not it is possible to reduce the tonal noise level through a destructive phase-shift generation, stepped tongues are comprehensively investigated by means of numerical simulations and experimental measurements. [DOI: 10.1115/1.4030498]

Introduction

Centrifugal fans with forward-curved (FC) blades are the main part of heating, ventilation, and air conditioning (HVAC) and air handling units in automotive and home appliance industries. While this type of fan is capable of delivering more air volume at lower rotational speeds than some other centrifugal fan types, noise is of great concern due to the unsteady flow inside the fan. Considering the fact that in the radial fans with FC blades a good portion of the static pressure is produced in the scroll housing [1], the efficiency of the fan is significantly affected by the shape of the housing and, without a proper volute design, the FC fan would be dysfunctional. In FC fans, the scroll housing has the same size and shape as for the other centrifugal fan types, but the volute tongue (cutoff) protrudes higher into the outlet [1]. Therefore, the tonal noise of the fan, which to a great extent, stems from the interaction of the mean air flow leaving the impeller and the volute tongue (cutoff) [2], is of great importance in this type of fans. The nonuniform velocity profile that develops above the blades (beyond the trailing edge) produces strong pressure fluctuations that lead to the generation of noise at the blade passage frequency (BPF) and its harmonics.

The following study is divided into two parts: the focus in the first part is on modifying the parameters related to the geometry of the impeller without changing the shape of the housing. The second part of the study is devoted to the shape of the volute tongue and its effects on the tonal noise generation. The reference fan wheel, illustrated in Fig. 1, is used in the second part of the study.

Shape of the blades is known to affect the noise of the fan. Changing the shape of the trailing edge of the blades led to a reduction of tonal and broadband noise in a radial fan with backward-curved blades [3]; however, the effect of this modification on the performance of the fan is not mentioned in the reference. The blade spacing is another parameter that is studied in some of the previous works; it is shown that, by using uneven blade spacing, it is possible to spread the sound spectrum over a wider range of frequencies without affecting the performance of the fan [4].

There is also a body of research on the role of the housing in the noise generated by the fan. Moreland [5] comprehensively investigated the sound power spectrum of FC fans. In his study, the enhancement of the sound power at certain frequencies was related to the acoustical resonances in the blower housing. He quantified the housing effects by performing experimental measurements for two configurations, i.e., with housing (housed) and without housing (unhoused). In order to make sure that the spectral peaks were only related to the housing geometry, he performed the sound measurements for three different fan speeds and noticed that the enhancements were independent of the fan speed. His experimental results showed that the peaks diminish from the spectral curves when operating the fan without housing at a constant rotating speed. Neise extensively commented on this study and rejected the concept of comparing housed and unhoused fans [6]. Neise substantiated his claim by performing some measurements and showed that the unhoused FC fan has completely different characteristics (comparing to the housed fan); hence it is not meaningful to compare these configurations in view of acoustical effects.

The volute tongue (cutoff) has received a lot of attention and has been the focal point in some of the previous studies. Diminishing the cutoff effects by increasing the clearance between the impeller and the cutoff was one of the earliest methods [2,7]. Although this method helps to reduce the tonal noise of the fan, it negatively affects the fan efficiency. However, there is not necessarily a direct relation between the rotor–stator clearance and the efficiency of the FC fans (e.g., the greater the distance, the less the efficiency) [8]. Changing the cutoff height is another parameter which is reported to affect the tonal noise of the fan, in particular when the cutoff has a straight shape [9]. Inclining either the impeller blades or the cutoff geometry is also addressed as an effective method in reducing the tonal noise of the FC fans [2,10,11].

In addition to the studies mentioned above, active source cancellation by means of resonators [12,13] or loudspeakers [14] has been researched, and it was possible to effectively reduce the tonal

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noise component of radial fans. However, the drawback of this method is that attaching these add-on devices on the fan makes it massive and bulky, and provisions for mounting them should be made in advance.

Experimental Measurements

In order to obtain the characteristic curves and evaluate the performance of the fan, it was installed on a chamber test rig in accordance with DIN/ISO 5801 [15]. The performance data, such as pressure rise across the fan, and shaft power were collected for a range of flow rates at a rotating speed of 1000 rpm [16,17]. Acoustic measurements were carried out using the in-duct method in accordance with ISO 5136 [18,19]. For the noise measurements, the fan was placed in a semi-anechoic room, while an anechoic termination was attached to its outlet (Fig. 2). The anechoic termination prevents the formation of the axial standing waves, and as a result, the sound measurement becomes independent of the axial location [19]. The sound radiated from the outlet of the fan was measured up to 5000 Hz by using three high precision microphones manufactured by Microtech Gefell GmbH (applicable in the range of 10 Hz–40 kHz with a maximum sound pressure level (SPL) up to 158 dB [20]). These microphones were calibrated in advance of the measurements by means of a Brüel & Kjær sound calibrator (Type 4231) with a reference SPL of 94 dB and calibration accuracy of ±0.2 dB [21].

All of the microphones were equipped with slit-tube windscreens (namely, turbulence screen or sampling tube microphones [19]) to suppress the effect of turbulent pressure on the microphones and to increase the signal-to-noise ratio (see Fig. 3). The microphones were installed in the measurement duct that connects the fan outlet to the anechoic termination (see Fig. 2). This duct had a diameter of 170 mm; therefore according to Eq. (1) [18], the results were influenced by the effects of modes beginning from approximately 1172 Hz at the best efficiency point (BEP) of the fan (i.e., 450 m^3/hr)

$$F_{\text{cut-on}} = 0.586\frac{c}{D}\sqrt{1-\left(\frac{U}{c}\right)^2} \tag{1}$$

At each operating point, the final result was obtained by calculating the average of ten sequentially performed single measurements (each taking 35 s). The average SPL recorded by the corresponding microphones was calculated according to the following equation:

$$\overline{L_p} = 10\log\left[\frac{1}{N}\sum_{i=1}^{N}10^{0.1L_{\text{pi}}}\right]dB + C \tag{2}$$

where N is the number of microphones, L_{pi} is the time-averaged SPL of different microphones, and C is the total correction factor, including the combined frequency response correction for the microphones with sampling tube [18]. The reliability of the experimental results was verified by repeating the reference measurement in ten different days, which also provided a good basis for estimating the standard deviations.

Numerical Setup

In order to study the aerodynamic performance and the noise characteristics of the fan, CFD and CAA simulations were performed. The simulations were carried out by means of the commercially available CFD package STAR-CCM+. The geometry of the fan was discretized into approximately 12.5×10^6 polyhedral elements (cells). According to a previously conducted study [17], the characteristic curves as well as the flow field of the fan can reliably be predicted by using a mesh that consists of approximately 4×10^6 cells (60% in the rotor and 40% in the stator). However, noise prediction demands employing a more sophisticated grid that, for example, satisfies the standard criterion of 20 cells per the smallest wavelength [22,23]. Figure 4 illustrates the mesh used for the numerical simulations in this study. The mesh was generally more refined in the rotating region (maximum cell size of 0.8 mm in the rotor, while 3 mm in the stator), and special attention was paid to the grid parameters on the interface between the rotor and the stator (i.e., internal interface with in-place topology). The CFD grid had globally low Reynolds wall resolution ($y+\leq1$ in the rotor and $y+\leq2$ in the stator), and the grid was tailored so that an unsteady flow leaving the impeller channels could be captured properly.

CFD simulations were performed on a 64 core server (2.3 GHz), and modeling each rotation of the impeller took approximately 3840 central processing unit-hr. Unsteady CFD simulations were started from the converged steady-state solutions and were continued for eight rotations of the impeller. The parameters used in running the CFD solver were: Ideal gas, compressible flow, second order convection scheme, all $y+$ wall treatment, and the segregated solver with second order temporal discretization accuracy. The shear–stress transport k-ω model was used in both steady-state and unsteady (transient) simulations. In the latter, the improved delayed detached-eddy simulation was employed. Detached-eddy simulation (DES) is inherently a combination of Reynolds-averaged Navier–Stokes (RANS) and large eddy simulation (LES), and it is one the most widely used approaches in the aeroacoustic simulations [24]. Nonreflective boundary conditions (also called freestream), which avoid

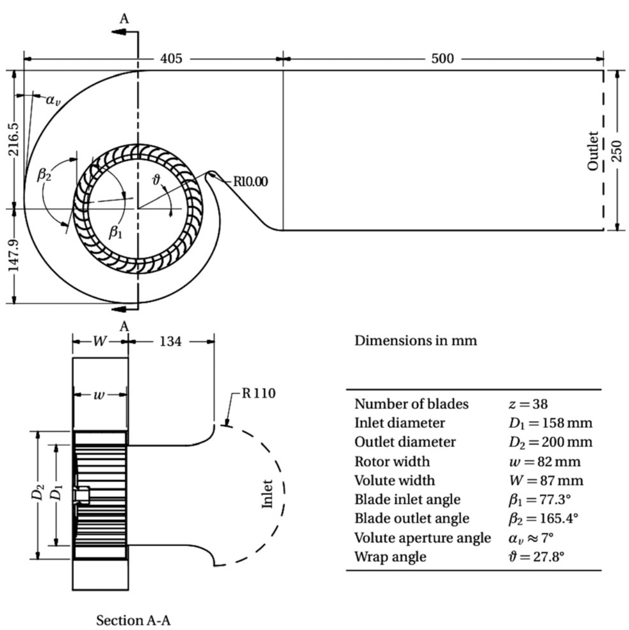

Fig. 1 Geometrical dimensions and parameters of the fan (reference model) [8]

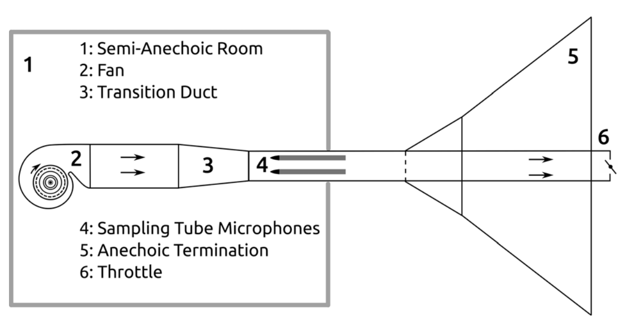

Fig. 2 Schematic illustration of the experimental setup

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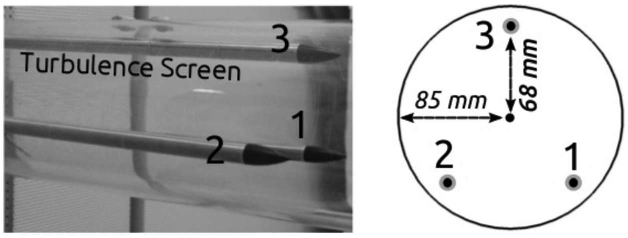

Fig. 3 Distribution of the slit-tube microphones in the measurement duct calculated according to DIN 5136

pressure reflections, were defined at the inlet and outlet of the fan. Unsteady CFD simulations were initialized by using coarser time-steps for the early rotations and finalized using a small time step that could properly resolve ten points in the wave-form at 2400 Hz.

Pressure monitors, which were placed in the discharge of the fan (see Fig. 5), provided time-accurate data for the noise predictions. In addition, the Ffowcs Williams–Hawkings (FW–H) method was used to predict the noise radiated from the fan. The FW–H receiver was placed near the outlet, and all of the solid surfaces in the domain were assigned to it as the impermeable surfaces (sources of noise). The FW–H equation is an exact rearrangement of the continuity equation and the Navier–Stokes equations into the form of an inhomogeneous wave equation with two surface source terms, i.e., a thickness source (monopole) and a loading source (dipole) as well as a volume source term (quadrupole) [25]. The volume source was neglected in this study; mainly due to the fact that in the subsonic fans, the dipole source originating from the unsteady forces exerted by the impeller on the air inside the fan is the dominant source of noise [26,27]. Moreover, as will be seen subsequently, the CFD grid limits the range of the sound predictions to approximately 1.2 kHz. Therefore, it was not necessary to include the volume sources in the FW–H method, since their contributions are more important in the higher frequency range. Nonetheless, neglecting the volume sources might lead to an underestimation in the FW–H results.

Results and Discussion

The study is divided into two parts: the first part highlights some of the simple geometrical modifications that can be made to the fan wheel to reduce the tonal noise, whereas the second part focuses on the shape of the cutoff in the housing of the fan. In both parts, the influence of the corresponding modifications on the static efficiency of the fan is also addressed.

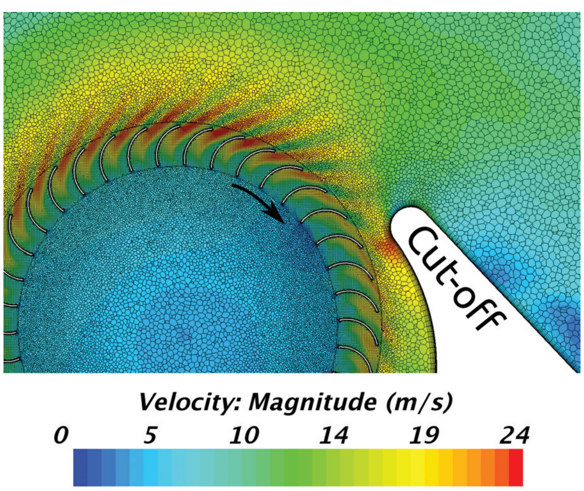

Fig. 4 Unstructured mesh with 12.5×10^6 polyhedral cells

Since the numerical and the experimental signals were analyzed by means of two different programs, it was necessary to determine the ensuing differences in the SPLs. In order to do so, a reference experimental signal was processed by STAR-CCM+ (i.e., the numerical signal processor) and the result was compared with that obtained from the experimental signal processor. This comparison revealed a difference level of 2–4 dB (depending on the frequency) in the SPLs. This should be taken into account when comparing the experimental and the numerical results. Moreover, the comparison showed that at the BPF of the fan (i.e., 633 Hz), the experimental SPL is 0.8 dB higher than the numerical level. The corresponding value has been added to the tonal noise level of the fan predicted by the numerical simulations in this section. Another point worth mentioning is that due to the excessive computational costs, the numerical signals are often shorter than the experimental signals. As a result, the numerical and the experimental signals have two different energy levels and should not be compared with each other without a proper normalization. Therefore, in order to ensure that the power spectral density (PSD) is conserved, the numerical Fourier transformed results were normalized to the same value (i.e., 1.28 s) similar to what was used by the experimental signal processor. The normalization is performed according to the following equation:

$$|\mathrm{FFT}_1|\sqrt{T_1} = |\mathrm{FFT}_2|\sqrt{T_2} \tag{3}$$

In terms of the SPL, Eq. (3) can be recast into the following form:

$$\mathrm{SPL}_2 = \mathrm{SPL}_1 + 10\log\left(\frac{T_1}{T_2}\right) \tag{4}$$

where T_1 is the length of the real signal and $T_2 = 1.28$ s.

Part I: Blade Modifications. In FC fans, generation of the tonal noise is associated with strong pressure fluctuations generated by the nonuniformity of the velocity profile around the blades (above the trailing edge) [2]. Therefore, any geometrical modification that imposes uniformity on the shape of the velocity profile has the potential to decrease the tonal noise of the fan. The first method investigated in this study was to increase the number of blades. Figure 6 illustrates the velocity profiles above the blades (near the trailing edge) of the impellers with different number of blades. As can be seen, by increasing the number of blades the velocity profile becomes more uniform, whereas decreasing it has the contrary effect and increases the difference between the highest and the lowest velocity magnitude on the profile.

Figure 7 illustrates the relation between the number of blades and the modeled turbulent kinetic energy (TKE). The results are captured on a cylindrical surface above the blades (showed

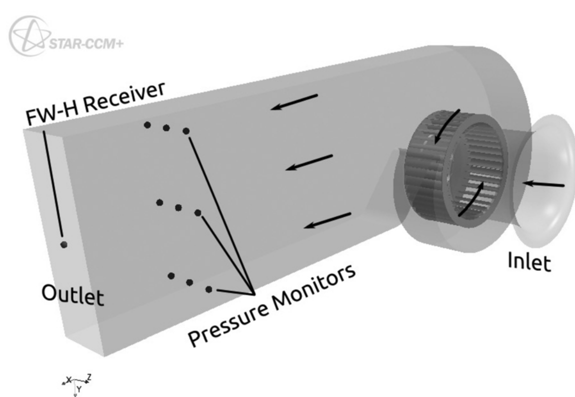

Fig. 5 The noise radiated from the fan was predicted using the pressure monitors placed in the discharge of the fan as well as a FW–H monitor installed near the outlet

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unrolled here). It is apparent that increasing the number of blades not only creates a more homogeneous TKE distribution but also reduces its level, especially in the vicinity of the cutoff that significantly contributes to the tonal noise generation. Based on the steady RANS results (Figs. 6 and 7), it is expected to have a lower tonal noise component by increasing the number of blades. Despite lowering the tonal noise level, increasing the number of blades can deteriorate the aerodynamic performance of the fan. Therefore, in parallel to the acoustic aspects, the efficiency of the fan should always remain in focus. Figure 8 illustrates the static efficiency of the fans with different number of blades in comparison with the reference model. The static efficiency was calculated using the pressure rise across the fan, airflow, torque, and the rotational speed of the fan (i.e., 1000 rpm) according to the following equation:

$$\eta_{\text{static}} = \frac{\Delta p \dot{V}}{2\pi M n} \tag{5}$$

According to Fig. 8, it is evident that reducing the number of blades reduces the obtainable efficiency of the fan, especially in the overload range (flow rates higher than 450 m^3/hr). On the other hand, increasing the number of blades boosts the aerodynamic performance of the fan across its operating range.

The numerical noise predictions of the corresponding models are shown in Fig. 9. The results belong to the pressure monitors in the discharge of the fan (see Fig. 5). For all cases, 2880 samples were recorded for a total signal length of 0.12 s at a sample rate of 24,000 Hz. The signals were broken into two blocks, representing a signal resolution of approximately 17 Hz; the Hanning window was also used to minimize spectral leakage. Comparing the simulation result of the reference model with the available experimental result, there is good agreement up to St = 1.5 and fairly a close correlation up to St = 2; afterward, the deviation becomes higher due to the limitations imposed by the CFD grid.

Table 1 presents the comparison between the tonal noises of different impellers predicted by the numerical simulations. The transient results ratify the conclusion previously drawn based on the RANS results, i.e., increasing the number of blades leads to a tonal noise reduction.

The second modification made on the impeller was changing the blades outlet angle (i.e., 165 deg in the reference model). Numerical simulations were performed for three other fan wheels with the outlet angles of 155 deg, 160 deg, and 170 deg. Figure 10 illustrates the performance curves of the fans with various blade outlet angles. As can be seen, reducing the outlet angle reduces the efficiency of the fan, particularly at higher flow rates. On the other hand, increasing the outlet angle improves the efficiency in the higher range of the flow rates, while the changes at the other operating points are negligible.

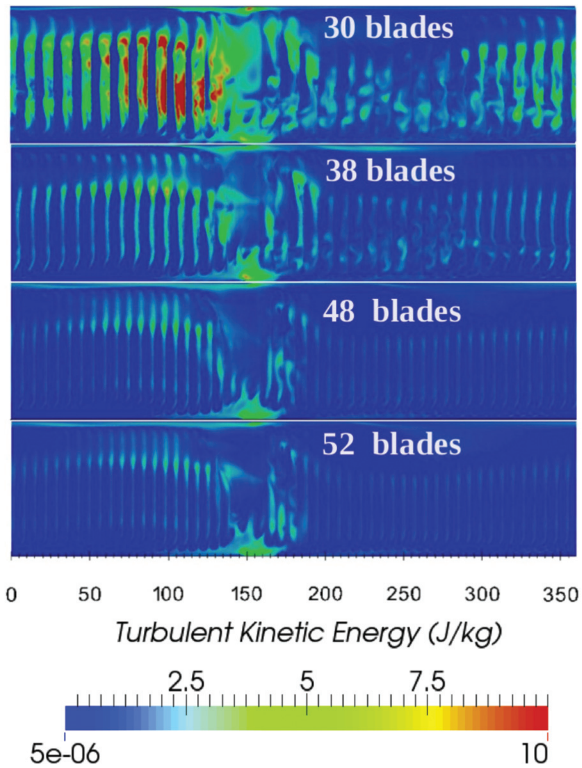

Fig. 7 The effect of changing the number of blades on the distribution of the TKE above the impeller

According to the numerical results, size of the flow separation in the impeller channels becomes smaller as the outlet angle increases (see Fig. 11). This is a favorable trend from the aerodynamic point of view and leads to a better performance of the fan. Modifying the blade outlet angle has an influence on the flow leaving the impeller channels and changes the velocity profile above the blades; therefore, it affects the tonal noise generation of the fan. Numerical predictions (see Fig. 12 and Table 2) indicate that the maximum level of tonal noise is generated by the model with an outlet angle of 160 deg, and either reducing or increasing the corresponding outlet angle leads to a tonal noise reduction. This observation is in agreement with the results of the experimental measurements conducted for a similar fan by Leist et al. [28].

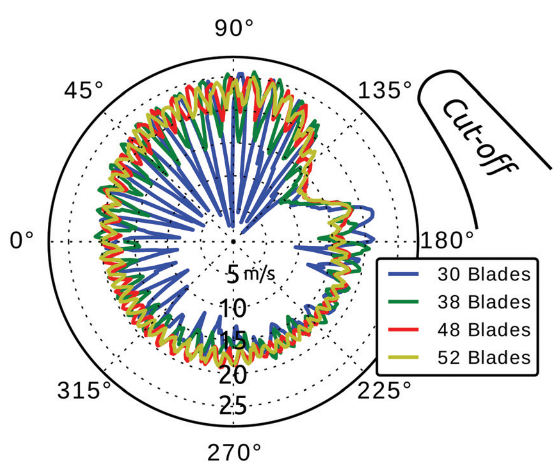

Fig. 6 Influence of changing the number of blades on the shape of the velocity profile above the impeller (results from numerical simulations)

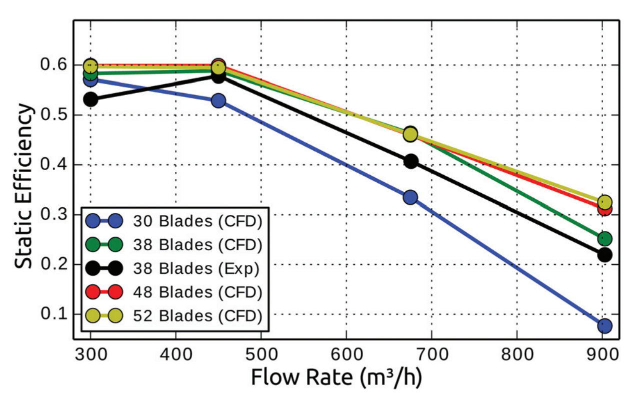

Fig. 8 Static efficiency versus flow rate of the fans with different number of impeller blades

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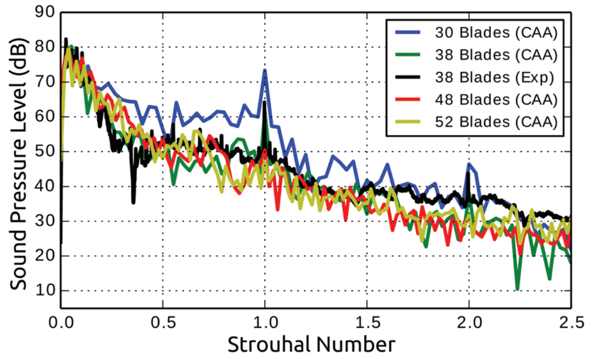

Fig. 9 Spectral noise analysis of the fans with different number of impeller blades; St = 1 corresponds to the blade passing frequency

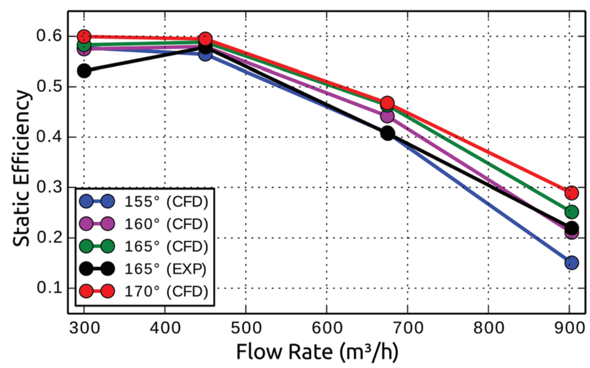

Fig. 10 Static efficiency versus flow rate of 38 blade impellers with different outlet angles

Part II: Cutoff Modifications. The second part of the study focuses on the geometry of the cutoff (volute tongue); the reference fan wheel with $z = 38$ and $\beta_2 \approx 165$ deg was used in this part. Transient surface data (i.e., FFT of surface pressure) obtained from the numerical simulation of the reference model (see Fig. 13(*a*)) reveal that the cutoff surface does not uniformly contribute to the noise generation at the BPF of the fan. Dividing the cutoff geometry into two halves: the part which is near the impeller hub contributes more to the tonal noise generation than the other part (i.e., on the shroud side). Moreover, from monitoring the pressure on the hub-side of the cutoff (see Fig. 14), it can be inferred that the pressure trends are in-phase condition. Keeping this in mind, this part of the study aims to answer the following questions whether or not:

(1) It is possible to generate a destructive phase shift at the cutoff?
(2) It is feasible to reduce the tonal noise of the fan through a phase-shift generation?

In order to answer these questions, the cutoff geometry was split into four equal pieces, and different standing combinations of them were experimentally tested. Each cutoff arrangement was designated by a unique serial number that provides useful information about its geometry. Figure 13 illustrates the pressure fluctuations as well as the SPLs of the cutoff arrangements investigated in this study. The abbreviation LHLH, for instance, denotes the relative position of each segment, from the hub-side toward the shroud side (L stands for low and H for high). The first number represents the circumferential difference in the heights of the segments; 8.25 mm represents half a blade-to-blade spacing and 16.5 mm is exactly equal to one impeller passage, respectively. The other two numbers, which were remained unchanged throughout the study, denote the radius of the tongue (10 mm) and the impeller to tongue clearance (16 mm). Figure 15 depicts the effect of the stepped tongues on the static efficiency of the fan obtained from the numerical simulations. As can be seen, there is an insignificant reduction in the static efficiency in the lower range of the flow rates. However, after the BEP of the fan, the deviations become more apparent, and this trend is more pronounced in the 16.5 mm tongues. According to the pressure patterns obtained from the cutoff monitors (see Figs. 16 and 17), the height difference between the high (H) and the low (L) cutoff segments significantly affect the quality of the phase-shift. It is evident that the 16.5 mm tongues are not able to generate the desired phase shift effects, and the corresponding inability is independent of the arrangement. Moreover, by comparing the pressure patterns of the 8.25 mm tongues in Figs. 16 and 17, it becomes evident that a more destructive phase-shift is generated by the LHLH-8.25-10-16 arrangement. A phase-shift is also apparent in the pressure patterns of the HLHL-8.25-10-16, yet not as effective as it is in the LHLH arrangement.

Based on the numerical results, a definite answer can be given to the first question, and it is evident that a destructive phase-shift can be generated at the cutoff by using the stepped tongues. Nevertheless, the method proposed herein (stepping the cutoff) is not the only way of generating a phase-shift. According to Ref.

Table 1 Comparison between the tonal noise components of the fans with different number of impeller blades

Number of blades	BPF (Hz)	FW–H (dB)	Pressure-monitor (dB)
30	500	71	73
38	633	60	61
48	800	50	50
52	867	46	47

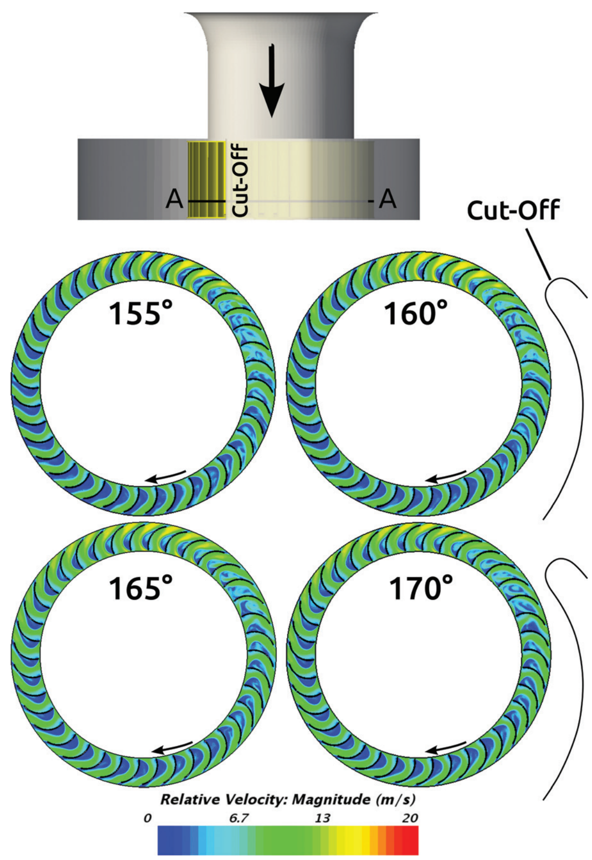

Fig. 11 The effect of changing the blade outlet angle on the size of the flow separation between the impeller channels

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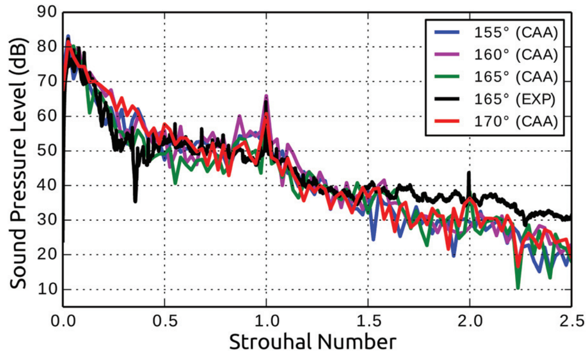

Fig. 12 Spectral noise analysis of the impellers with different blade outlet angles; St = 1 corresponds to the blade passing frequency

Table 2 Comparison between the tonal noise of the fans with different blade outlet angles β_2

β_2	FW–H (dB)	Pressure-monitor (dB)
155	61	62
160	64	65
165	60	61
170	59	61

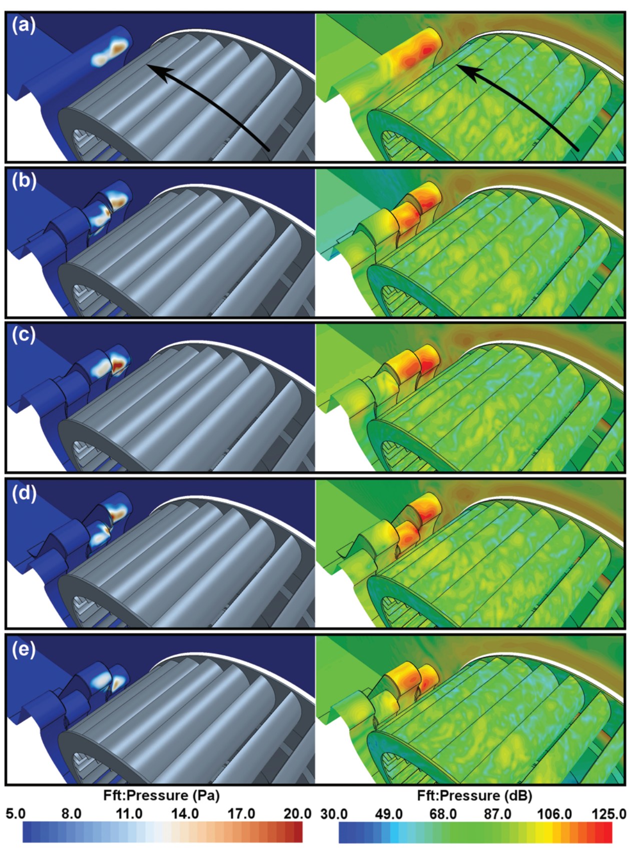

Fig. 13 Transient surface data representing the pressure fluctuation and the SPL of different cutoff configurations at the blade passing frequency while the fan operates at its BEP [29]. (*a*) 0-10-16 (Reference), (*b*) HLHL-8.25-10-16, (*c*) LHLH-8.25-10-16, (*d*) HLHL-16.5-10-16, and (*e*) LHLH-16.5-10-16.

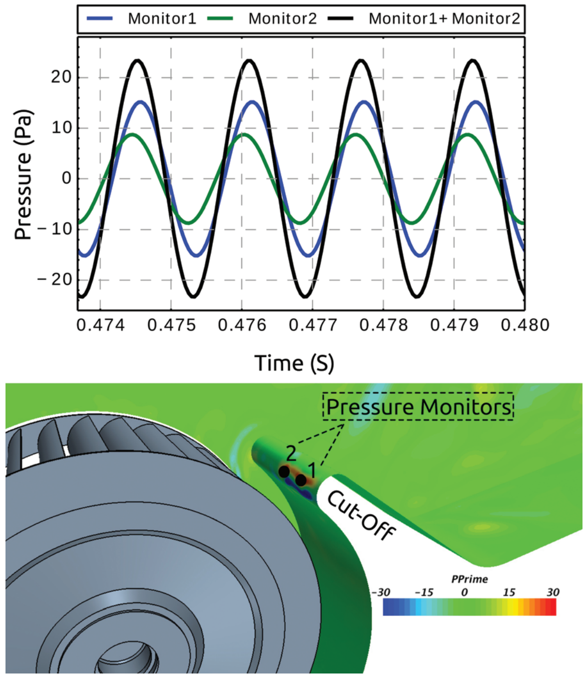

Fig. 14 The pressure patterns (top) obtained from the monitors installed on the reference cutoff at the displayed positions (bottom). The curves are "in-phase" condition; the scalar shows pressure fluctuations [29].

[2], inclining either the cutoff or the impeller blades is also an effective way of phase shift generation, as briefly mentioned in the introduction. Considering the inherent difficulties, inclining the impeller blades should be a troublesome method, but inclining the cutoff geometry seems to be a feasible method that can be investigated in the future. According to the results obtained herein, it can be expected that the inclination angle should play a leading role in this context. It should also be noted that the effectiveness of the inclined tongues in reducing the tonal noise component is already confirmed in the case of a FC fan with relatively large inlet and outlet diameters (refer to Ref. [10]).

In addition to the numerical simulations, experimental measurements were performed to find the answer of the second question. The noise measurements were conducted at four different flow rates, i.e., 250, 350, 450 (BEP), and 550 m^3/hr. Table 3 summarizes the measurement results of the reference model at different operating points at the BPF of the fan and its first harmonic.

Figure 18 depicts the cumulative SPLs of the measurements performed at different operating points for each cutoff arrangement. As can be seen, stepping the cutoff geometry generally increases the broadband noise level of the fan. The HLHL-8.25-

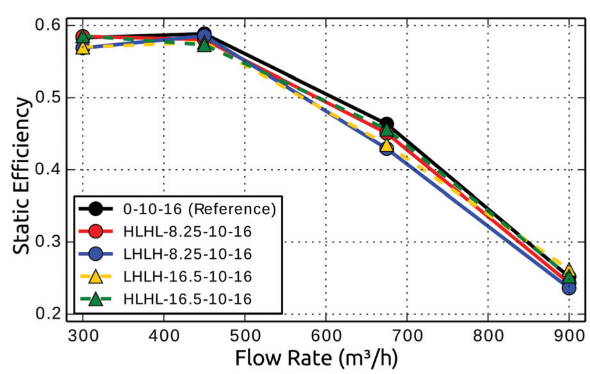

Fig. 15 Static efficiency versus flow rate of different cutoff models

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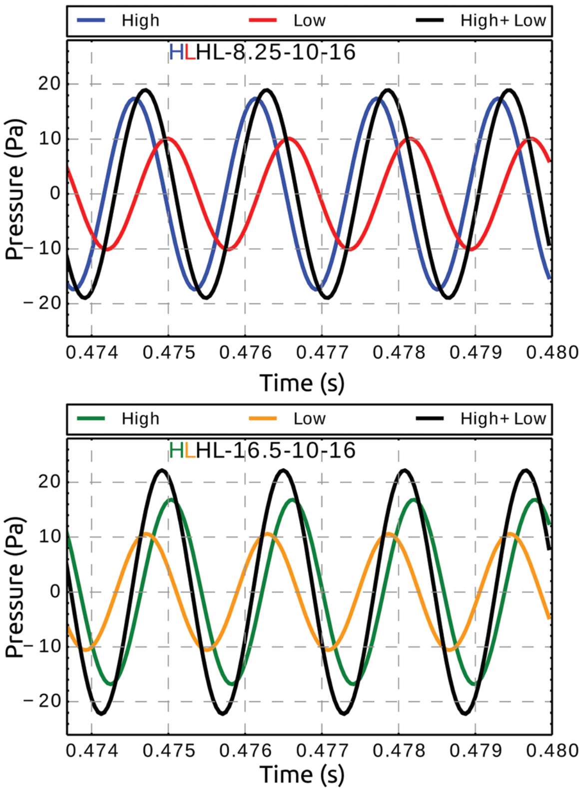

Fig. 16 The pressure patterns obtained from the monitors installed on the leading H and L segments of the HLHL tongues with 8.25 mm (top) and 16.5 mm (bottom) height difference

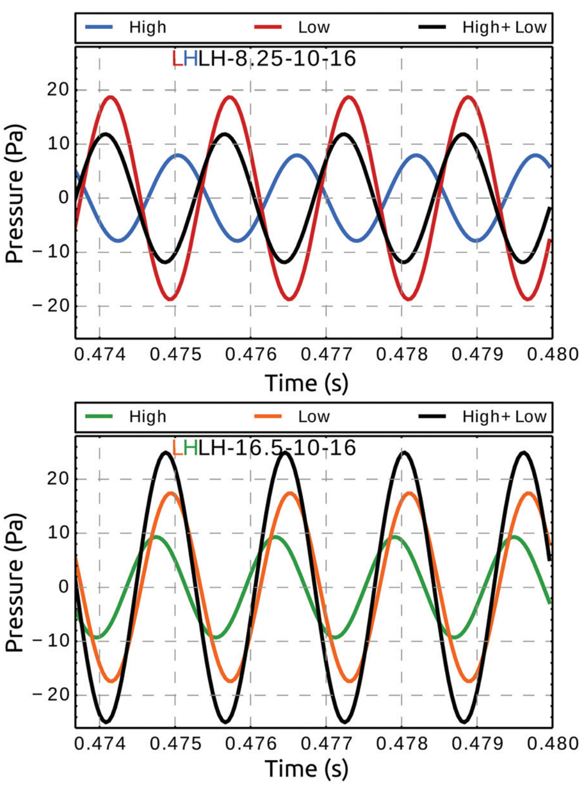

Fig. 17 The pressure patterns obtained from the monitors installed on the leading L and H segments of the LHLH tongues with 8.25 mm (top) and 16.5 mm (bottom) height difference

Table 3 SPLs of the reference fan at the BPF and its first harmonic at different operating points [29]

Flow rate (m^3/hr)	SPL at BPF (dBA)	SPL at first harmonic (dBA)
250	48.7 ± 1.5	33.0 ± 0.4
350	61.7 ± 1.8	39.0 ± 1.9
450	62.7 ± 1.5	43.0 ± 1.5
550	66.7 ± 2.7	46.5 ± 2.0

10-16 is the arrangement that shows a very similar trend to the reference tongue, whereas the LHLH-16.5-10-16 has the highest SPL in most of the frequency bands.

Table 4 presents the numerical and the experimental noise levels of the stepped tongues at the BPF. An effective reduction in the tonal noise level is only evident in the case of the HLHL-8.25-10-16, whereas the corresponding changes in the SPLs of the other cutoff arrangements are within the uncertainty range of the experimental measurements. Both of the numerical methods predict an increment of 4 dB for the LHLH-8.25-10-16 arrangement, although the experimental results show only a negligible increment.

Transient surface data shown in Fig. 14 helps to shed some further light onto the source of the corresponding increment in the numerical SPL. Accordingly, there is a considerable increase in the SPL as well as in the pressure fluctuations of the leading L segment of the LHLH-8.25-10-16 arrangement. This observation suggests that generating a phase-shift between the cutoff segments might be a useful method, yet it is not solely sufficient to reduce the tonal noise. In particular, when stepping the cutoff increases the pressure fluctuations, as is the case, for example, in the LHLH-8.25-10-16 arrangement, the tonal noise may become even

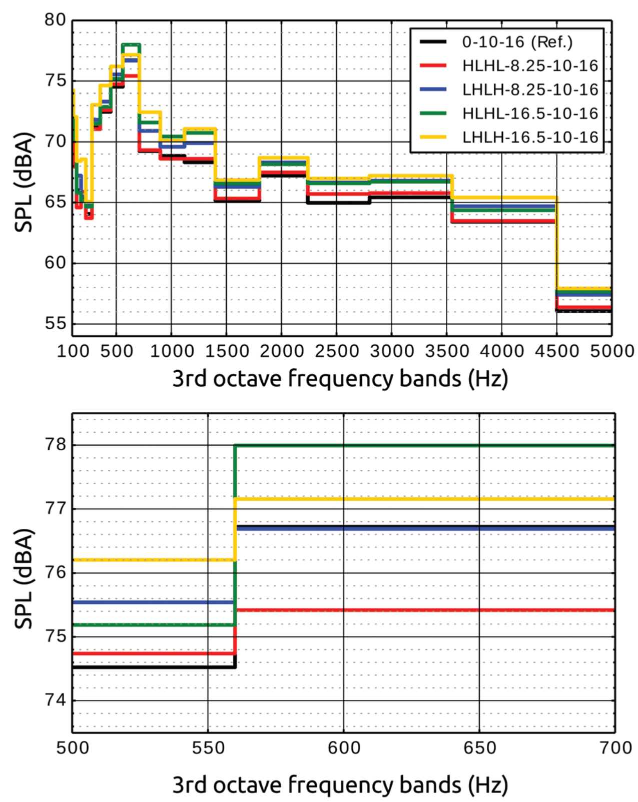

Fig. 18 Comparison between the experimental results of different cutoff arrangements. The plot shows the cumulative SPLs measured at four different flow rates. The inset at bottom focuses on the frequency band that includes the blade passing frequency.

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Table 4 Tonal noise comparison between the experimental and the numerical results of different stepped tongues at the BEP of the fan

Cutoff	Experiment (dBA)	FW–H (dBA)	Pressure-monitor (dBA)
0-10-16 (Reference)	62.7	58.3	59.4
HLHL-8.25-10-16	55.9	50.4	51.8
LHLH-8.25-10-16	63.3	63.0	64.5
HLHL-16.5-10-16	64.8	55.6	59.4
LHLH-16.5-10-16	62.2	56.3	59.8

higher, and the existing phase-shift between the segments cannot help to reduce it. Keeping this in mind, the relationship between the existence of a destructive phase-shift between the cutoff segments and the tonal noise level becomes even more perplexing. Therefore, the answer to the second question raised at the beginning of this section cannot be as straightforward as to the first one. Making a conclusive judgment about this topic demands further investigation. Particularly, it is essential to conduct new experimental measurements by means of surface mounted pressure transducers to further delve into the essence of the phase-shift between the cutoff segments and its relationship with the tonal noise level.

Conclusions

In this work, different methods are investigated to reduce the tonal noise of a radial fan with FC blades. It is shown that the tonal noise of the fan can effectively be reduced by making some geometrical modifications. On the impeller side, the parameter which plays an important role is the number of blades. It is demonstrated that increasing the number of blades not only leads to a tonal noise reduction but also improves the aerodynamic performance of the fan. According to the simulation results, increasing the number of blades considerably reduces the TKE above the impeller blades, which is a favorable trend concerning the acoustical aspects. However, using this approach demands some further investigation into the connection between the number of blades and the efficiency of the fan. In fact, it is always a challenging task to find an optimum number of blades, which satisfies both the noise and the aerodynamic requirements. The next modification investigated was changing the outlet angle of the impeller blades. Although this modification was not as effective as changing the number of blades, it certainly shows the potential benefits of such a modification, especially in cases where the fan design is not optimal and demands rigorous enhancements. Based on the simulation results, increasing the outlet angle of the blades leads to the contraction of the flow separation between the impeller blades, which can improve both the aerodynamic and the aeroacoustic characteristics of the fan.

On the volute side, it is shown that, by employing stepped cutoff geometries, the tonal noise can be reduced. The stepped tongues investigated in this study consist of various arrangements with different heights. According to the simulation results, the stepped tongues featuring half a blade-to-blade height difference between their segments negligibly affects the static efficiency of the fan, whereas the arrangements with one blade passage height difference has more evident effects, and at some flow rates (particularly in the mid to high range), they can deteriorate the aerodynamic performance of the fan. According to the numerical results, it can be concluded that it is possible to produce a destructive phase-shift between the cutoff segments. A successful phase-shift generation was only evident in the case of the stepped tongues with half a blade-to-blade circumferential difference between their segments. Experimental results approved the effectiveness of one of the stepped tongues (i.e., HLHL-8.25-10-16) in reducing the tonal noise. However, further investigation, including experimental measurements using pressure transducers, is required to find out whether the corresponding reduction is justifiably related to the phase-shift between the cutoff segments or it is a by-product of stepping the cutoff.

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Nomenclature

BEP = best efficiency point
c = speed of sound (approximately 340 m/s)
C = total correction factor
CAA = computational aeroacoustics
CFD = computational fluid dynamics
D = diameter of the measurement duct (mm)
D_1 = inner diameter of impeller (mm)
D_2 = outer diameter of impeller (mm)
DES = detached eddy simulation
F = frequency (Hz)
FC = forward-curved
FW-H = Ffowcs Williams–Hawkings
HVAC = heating, ventilation, and air conditioning
$\overline{L_P}$ = average sound pressure level in dB (Ref = 20 μPa)
LES = large eddy simulation
M = torque (Nm)
n = revolutions per second
N = number of microphones
PSD = power spectral density (Pa^2/Hz)
RANS = Reynolds-averaged Navier–Stokes
SPL = sound pressure level in dB (Ref = 20 μPa)
St = Strouhal number
T = signal length (s)
TKE = turbulent kinetic energy (J/kg)
U = mean flow velocity (m/s)
$\dot{V}$ = flowrate (m^3/s)
w = rotor width (mm)
W = volute width (mm)
z = number of blades
α_v = volute aperture angle
β_1 = blade inlet angle
β_2 = blade outlet angle
Δp = pressure rise (Pa)
η_{static} = static efficiency
ϑ = wrap angle

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