TONAL NOISE CONTROL FROM CENTRIFUGAL FANS USING FLOW CONTROL OBSTRUCTIONS

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SUMMARY

Tonal noise originates from non-uniform flow that causes circumferentially varying blade forces and gives rise to a considerably larger radiated dipolar sound at the blade passage frequency (BPF) and its harmonics. The approach presented in this paper adapts a method previously developed for axial fans to control tonal noise using obstructions in the flow to destructively interfere with the primary non-uniform flow arising from stator/rotor interaction. The flow control obstruction is located such that the secondary radiated noise is of equal magnitude but opposite in phase compared to the primary noise. Experiments were carried out for a centrifugal fan test bench to validate the method for controlling BPF tonal noise by carefully positioning obstructions in a duct in the upstream flow.

INTRODUCTION

Despite the progress in the field of CFD and Numerical Aeroacoustic as preventive tools to design low noise rotating machinery, tonal noise remains a problem in fan applications and there is a need for curative techniques. Tonal noise mainly comes from the periodic unsteady blade forces and/or vane forces due to the interaction between the rotor and its environment [1]. Therefore, the environment of the fan is of major concern in the generation of fan noise. Previous papers [2,3,4] have demonstrated the efficiency of tonal noise control for axial flow fans by controlling the most radiating unsteady lift mode using flow control obstructions. The flow control obstruction is located such that the secondary radiated noise is of equal magnitude but opposite in phase compared to the primary noise. The magnitude and phase of the secondary noise are respectively controlled by the axial distance between the rotor and the obstruction and the angular position of the control obstruction.

In [2], we provided analytical tools to design the flow control obstructions in order for the blade unsteady lift generated by the rotor/control obstruction interaction to mainly contain the most radiating mode. It has been found that salient obstructions, such as low diameter rods are not adapted to control a tone without affecting its harmonics, since the circumferential harmonic content of the associated unsteady lift is high. But it is possible to design trapezoidal or sinusoidal control obstructions such that the circumferential harmonic content of the unsteady lift is low. Thus, it is possible to control one tone without affecting its harmonics.

In [3], free field experiments have shown the ability of a 6-lobed sinusoidal obstruction to control the BPF from a 6-bladed automotive fan in the whole space. In-duct experiments have shown the ability for an appropriate angular portion of the trapezoidal obstructions (35 deg) to efficiently control the BPF for different loads. Moreover, the aerodynamic performances of the automotive fan used in this study were almost unaffected by the presence of the obstruction.

In [4], a bi-harmonic control has clearly demonstrated the ability of two control obstructions to attenuate the BPF and its first harmonic (attenuation of sound pressure level up to 21 dB and 15 dB respectively).

A method has been proposed in [5] to combine several obstructions to increase the spatial effectiveness of tonal noise control, especially when the unsteady force on the low speed axial fan has significant radial and tangential components. The size of the lobes of a B and a B-1 sinusoidal obstructions has been optimized separately to adjust the secondary noise magnitude. The optimal angular position is then found for each obstruction. The B-lobed obstruction is efficient to control the BPF tone in the axial direction and the B-1 lobed obstruction was efficient in the radial direction. These two obstructions were then combined to give better spatial overall attenuation of the BPF in free field (8 dB) for a small axial fan. Finally, the linearity of the obstruction combination has been experimentally demonstrated for the fan under investigation.

Furthermore, in [5] and [6], the automatic positioning of a flow obstruction to control tonal noise from two kinds of subsonic axial fans has been theoretically and experimentally investigated. In those papers, the positioning approach sequentially finds the optimal axial and angular positions of the control obstruction to cancel a given multiple of the fan BPF. The axial positioning step uses a beat effect by slowly rotating the obstruction so that the frequency of the primary and secondary tonal noises is slightly shifted. A steepest gradient algorithm then searches the axial position [6] or radial extent [5] of the obstruction that creates an interaction noise equal in magnitude to the primary fan noise. Typically, four or five iterations are required to converge to the optimal axial location. Then the obstruction is rotated until the angular position of the obstruction creates a secondary interaction noise exactly out of phase with the primary noise. The optimal positioning approach can be used to adapt the control obstruction to possible variations of the flow.

In a psychoacoustic study [7], the ability of tonal noise control from an automotive fan to enhance acoustic comfort inside and outside the vehicle has been proven. Jury testing (composed of 20 occidental persons) proved that even if the loudness difference is low between two sounds with and without tonal noise control, the perception difference can be significant if the tonality difference is large. The control of the first two tonal noise frequencies enhances the acoustic comfort in the vehicle. Moreover, it has been concluded that the analysis of the specific loudness (ISO 532 B) as a function of critical band rate can also provide pertinent information to predict the effect of the tonal noise control. Indeed, it is important to first reduce the part of the noise that produces the largest area in the loudness pattern. It can be used as a "Loudness Pattern Equalization" technique to control tonal fan noise.

Other methods of tonal noise control from centrifugal are described in [8]: for example, an increase of the distance between the rotor and the tongue, if possible, in order to reduce the pressure fluctuations on the tongue and on the rotating blades. The use of circular and rectangular volute or an increase of the radius of the tongue curvature is also suggested. To reduce the noise, one can also try to minimize the acoustic propagation, by avoiding cut-on duct modes, by adding acoustic foams, resonators, or by actively controlling the tonal noise using loudspeakers as presented in [9].

The approach presented in this paper adapts the method previously developed for axial fans to centrifugal fans using obstructions in the flow to destructively interfere with the primary non-uniform flow. The analytical model of tonal noise radiation from centrifugal fans in free field based on Ffowcs-Williams and Hawkings's analogy is first analysed. Then guidelines for the circumferential modes of the unsteady forces to be controlled are given. Finally, experimental results are presented.

ANALYTICAL MODEL

Primary noise model

The fundamental mechanisms of noise generation are the same for axial and centrifugal fans. In the Ffowcs Williams and Hawkings's analogy [1], an aeroacoustic source of noise can be decomposed in three terms. The first term is associated with a quadrupole source that represents the generation of noise due to turbulent volume sources and corresponds to the solution of Lighthill's analogy [1]. For fan noise, this quadrupole source is significant only if the blade tip Mach number exceeds 0.8 [1] and is therefore irrelevant for the low subsonic fan under investigation in this study, for which blade tip Mach numbers do not exceed 0.15. The second term is related to dipole source caused by unsteady forces exerted by the solid surface on the fluid. This is the well-known "loading noise" or "dipole noise", the principal cause of subsonic fan noise [1]. The last term is equivalent to a monopole radiation due to the volume displacement effects of the moving sources, also called thickness noise. The efficiency of the thickness noise is poor at low fan rotational speed since the circumferential phase velocity of the fluid pressure fluctuations generated by the moving blades is well below sonic velocity [1]. Therefore, the main source of noise from subsonic fans comes from the forces exerted by the fluid on the static components (vanes, casing,…) or rotating parts of the fan, that act as dipolar sources in the Ffowcs Williams and Hawkings's analogy [1]. Periodic forces (steady rotating forces or unsteady rotating forces due to non-uniform but stationary flow) lead to discrete tone generation at BPF and its harmonics while random forces (such as turbulent boundary forces) lead to broadband noise.

The centrifugal fan is considered in free field, and reflections, diffractions, scattering as well as the casing effect are not taken into account. The coordinate system and the forces exerted by the blade are described in Fig. 1.

The Ffowcs Williams and Hawkings analogy's applied to compact rotating dipoles yields to the acoustic pressure in the frequency domain at the m^{th} harmonic of the BPF produced by a centrifugal rotor composed of B equivalent and equidistant blades rotating at angular velocity Ω [10]:

$$p'^{(m)}_{load1}(\mathrm{x}) = \frac{imB^2\Omega}{4\pi c_0 r_0} e^{imB(r_0/c_0)\Omega} \sum_{k=0}^{k=+\infty} e^{i(mB-k)(\varphi-\pi/2)} \times \left(i\sin(\theta) J'_{mB-k}(A) F_r^{(k)} + \left[\cos(\theta) F_a^{(k)} - \frac{mB-k}{mBM_{rs}} F_t^{(k)}\right] J_{mB-k}(A) \right) S \quad (1)$$

where A=$mB\Omega(r_s/c_0)\sin(□)$ and $M_{rs}= r_s\Omega/ c_0$; c_0 is the sound velocity, r_s is the source radius;

$J_{mB-k}(A)$ is the Bessel function of the first kind of order mB-k, and its derivative $J'_{mB-k}(A)=(1/2)(J_{mB-k-1}(A)- J_{mB-k+1}(A))$, where k is the circumferential harmonic order of the forces. $(r_0,□,\varphi)$ is the acoustics spherical coordinate system and $F_r^{(k)}, F_a^{(k)}, F_t^{(k)}$ are respectively the k^{th} circumferential order of the radial, axial and tangential forces.

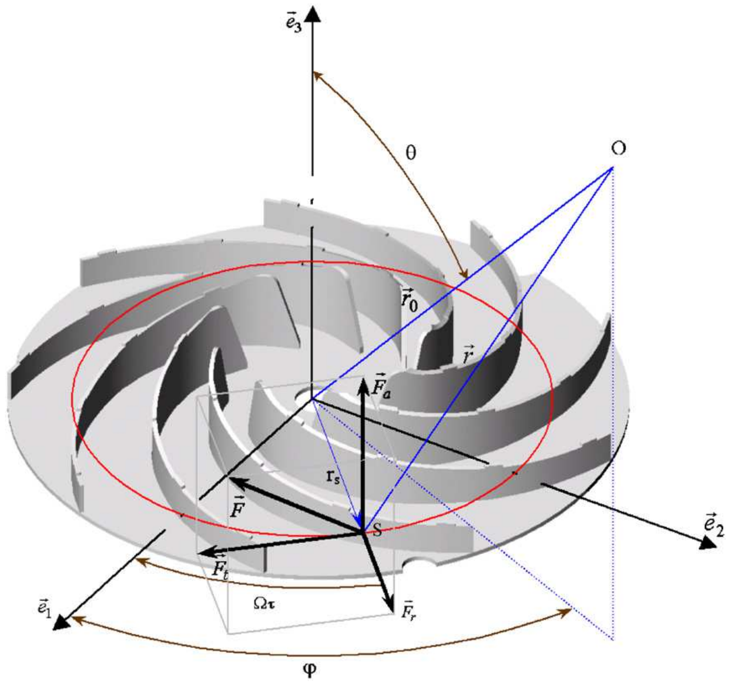

Figure 1 : Coordinate system [10]

In Eq. (1), there are Bessel functions of the first kind of order *mB*-*k* and combination of Bessel functions of the first kind of order *mB*-*k*-1 and *mB*-*k*+1. Since high order Bessel functions have negligible magnitudes compared to low order Bessel functions, the negative circumferential orders *k* for positive *m* in Eq. (1) can be neglected.

For centrifugal fans, even if the axial forces are negligible compared to radial and tangential forces, the radiation from those axial forces can be significant, especially for *k*=*B* (Fig. 2, middle). Indeed, if *k*=*B*, the radiation of tangential forces is null because $\frac{mB-k}{mBM_{rs}}=0$ and the radiation factor of the axial forces $\cos(\theta)J_{mB-k}(A)$ is especially high in the axial direction ($\theta = 0 \pm n\pi$). For *k*=*B*, the radiation of radial forces is dominant in the fan rotation plane $\theta = \pi/2 \pm n\pi$. Even if the radiation factor of radial forces $i\sin(\theta)J'_{mB-k}(A)$ is lower than the radiation factor of axial forces, the magnitude of the acoustic pressure integrated in the whole space, radiated by radial forces can be greater than the one radiated by axial forces if it is assumed that $F_r >> F_a$ for centrifugal fans.

For $k \neq B$, especially for *k*=*B*-1 and *k*=*B*+1 (Figs. 2 left and right), the radiation factor magnitude is much higher for radial and tangential forces than axial forces. Whereas the radiation factor phase of tangential forces are identical for *k*=*B*-1 and *k*=*B*+1, the phase of radial forces are opposite. It means that, in the case of an impulse excitation $F_r(t)=\delta(t)$ and $F_t(t)=\delta(t)$ (or $F_r^{(k)} = F_t^{(k)} = 1 \;\forall\, k$), the radiation of tangential forces interferes constructively and the radial forces interfere destructively.

Thus, to control the tonal noise emitted at BPF, the control of one or several of these forces can be efficient: radial forces of order *B*-1<*k*<*B*+1, the axial forces of order *B* and the tangential forces of order *B*-1 and *B*+1. The control of the axial forces is expected to be effective in the axial direction and the control of radial and tangential forces is expected to be efficient in the rotational plane.

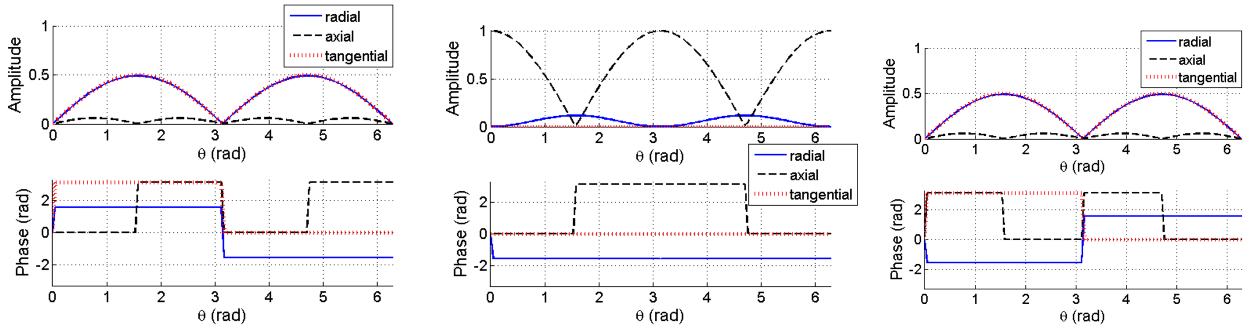

Figure 2: Magnitudes (up) and phases (down) of the radiation factors $\left[i\sin(\theta)J'_{mB-k}(A)\right]$; $\left[\cos(\theta)J_{mB-k}(A)\right]$ and $\left[(mB-k)/(mBM_{rs})J_{mB-k}(A)\right]$ associated to radial, axial and tangential forces respectively, as a function of θ. Left: k=B-1, middle: k=B and right: k=B+1. Ω=50 rad.s^{-1}, r_s=0.2m, c_0=340m.s^{-1}, m=1, B=8.

Control model

In this paper, the upstream obstruction does not radiate sound, but rather, the periodic wakes in the downstream flow field of the obstruction impact the rotor that radiates tonal noise.

Contrary to the control of axial flow fan using flow control obstruction, several modes can contribute significantly to the primary noise for the centrifugal fans. Moreover, the free field hypothesis can never be fully satisfied for such fans. Yet, to formulate the control problem, we make the following hypothesis:

- Free field radiation
- The principal mechanisms of control come from the interaction between an upstream obstruction and a downstream rotor
- The rotor radiates more than the fixed parts
- Thickness noise is negligible
- A *V*-lobed obstruction excites the circumferential mode *k=V* and its harmonics of radial $F_{sr}^{(k)}$, tangential $F_{st}^{(k)}$ and axial $F_{sa}^{(k)}$ secondary forces with different weightings.
- The secondary forces and the primary forces ($F_{pr}^{(k)}$, $F_{pt}^{(k)}$, $F_{pa}^{(k)}$) can be linearly added

Thus, the total noise can be written as follows:

$$\begin{aligned} p_t^{\prime(m)}(\mathbf{x}) = \quad & p_p^{\prime(m)}(\mathbf{x}) + p_s^{\prime(m)}(\mathbf{x}) \\ = \quad & \frac{imB^2\Omega}{4\pi c_0 r_0} e^{imB(r_0/c_0)\Omega} \sum_{k=0}^{k=+\infty} e^{i(mB-k)(\varphi-\pi/2)} \\ & \times (i\sin(\theta)J'_{mB-k}(A)\{F_{pr}^{(k)} + F_{sr}^{(k)}\} \\ & + \left[\cos(\theta)\{F_{pa}^{(k)} + F_{sa}^{(k)}\} - \frac{mB-k}{mBM_{rs}}\{F_{pt}^{(k)} + F_{st}^{(k)}\}\right] J_{mB-k}(A))S \end{aligned} \tag{3}$$

where the subscripts *p* and *s* respectively denotes primary and secondary forces. Since the obstructions considered in this paper are *V*-periodic: $F_{sr}^{(k)}=0$, $F_{st}^{(k)}=0$ and $F_{sa}^{(k)}=0$ if $k \neq nV$ where *n* is a positive integer.

We have postulated a priori that the angle and the distance between the rotor and the obstruction located in the inlet (parallel to the rotation plane of the rotor) can respectively control the phase and the magnitude of the secondary forces, and subsequently the secondary tonal noise. The magnitude of the forces should have monotonic decrease as a function of the axial distance.

EXPERIMENTAL RESULTS

The experiments have been conducted in duct in order to locate the obstruction in the upstream flow field (Fig 3). The cavity in which the fan is mounted and the duct have strong influences on the acoustic radiation and differs from the free field model given in the previous section. However, the periodic radial, tangential and axial forces can still be considered as the sources of tonal noise.

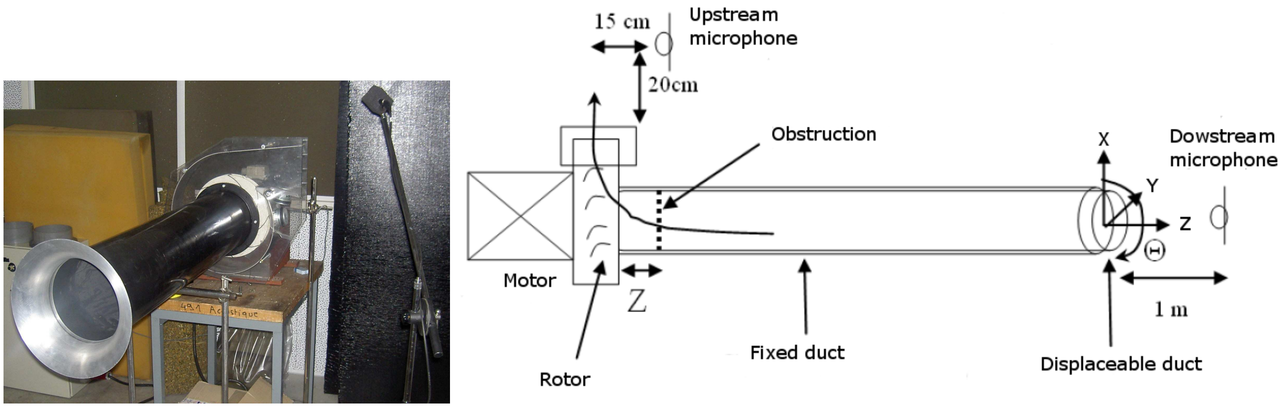

Figure 3: Experimental set-up, picture (left) and schematic representation (right)

The rotor has B=8 regularly spaced blades and its rotational velocity is 3000 R.P.M. The inlet flow comes from the duct, and the outlet flow is lateral. A strong interaction between the tongue and the rotor is the main contributor to the primary tonal noise.

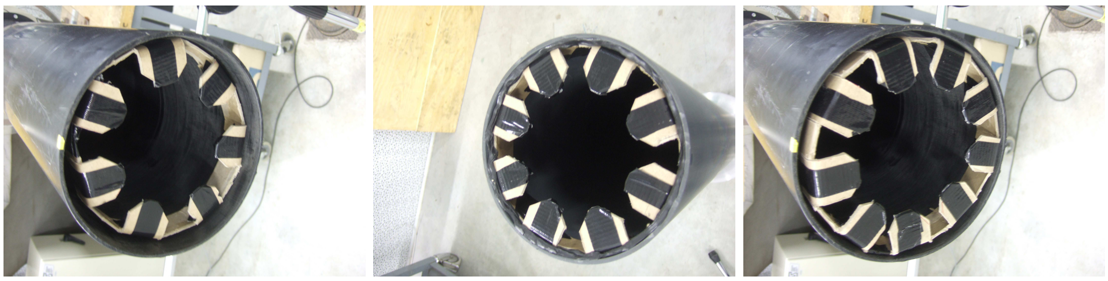

Figure 4: The three obstructions tested in this study, B-1 lobed obstruction, a B-lobed obstruction and a B+1 lobed obstruction

To locate the obstruction axially and circumferentially, a first duct is inserted in a second duct (See right hand side of Fig. 3). The second duct is fixed in the laboratory referential frame. The obstruction is fixed in the first duct (See Fig. 4), which can be slided inside the second duct. Two microphones were used to measure the acoustic pressure at 1m upstream the fixed duct and downstream, at 0.2m the rotation plane and slightly shifted from the flow.

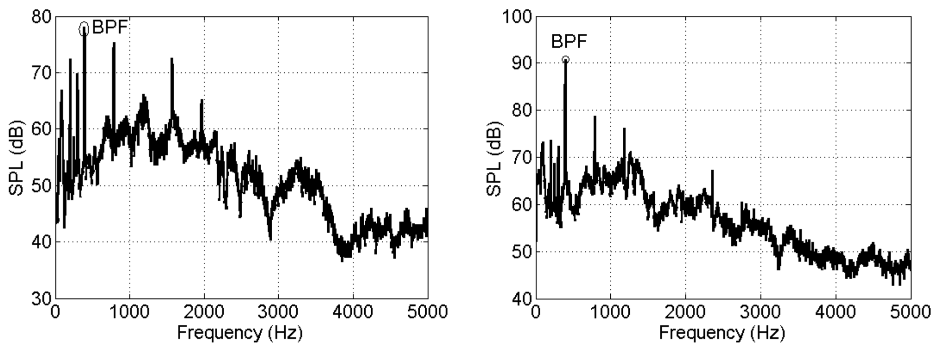

Figure 5: Primary sound power level, upstream (left) and downstream (right)

Three obstructions were tested, a *B*-1 periodic, a *B*-periodic and a *B*+1 periodic, as shown in Fig. 4.

The primary acoustic spectra measured upstream and downstream are shown in Fig. 5. The BPF tone is $B\Omega$=400Hz and its harmonics are $mB\Omega = m \times 400\,\text{Hz}$ (where $m \geq 2$). The motor was responsible for the subharmonic tones below 400Hz.

As shown in Fig.3, *Z* is the axial distance between the rotor and the control obstruction and *Θ* is the angular position of the control obstruction. The obstruction is translated by 1cm increment in the axial direction from *Z*=7cm to *Z*=13cm. The 9-lobed obstruction is rotated by 4.2° step from *Θ*=0° (vertical axis Y) to *Θ*=42°, and the 8-lobed and 7-lobed obstruction from *Θ*=0° to *Θ*=46.2°.

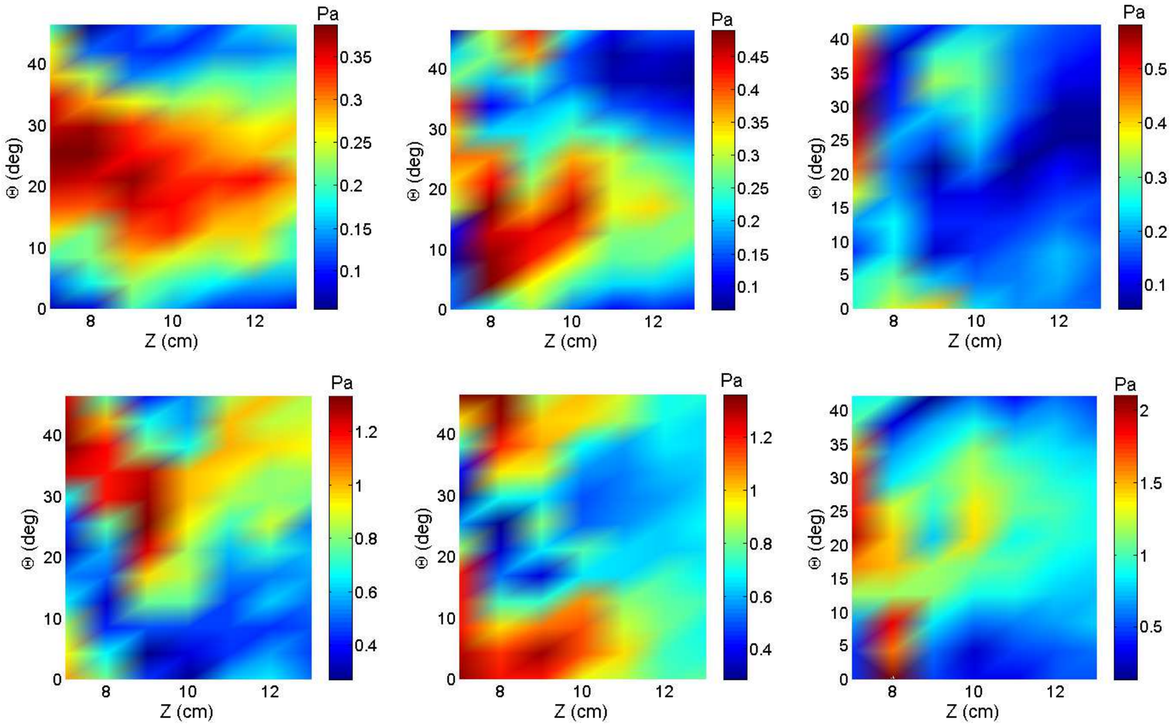

Figure 6: Downstream (top) and upstream (bottom) error surface magnitudes at BPF. Left: B-1=7-periodic obstruction, middle: B=8-periodic obstruction and right: B+1=9-periodic obstruction.

The magnitude of the error surfaces (total acoustic pressure in Pa as a function of *Z* and *Θ*) measured for the three obstructions are shown in Fig. 6. Upstream error surfaces are shown at the top and the downstream error surfaces are shown at the bottom, for the 7-lobed obstruction (left), 8-lobed obstruction (middle) and the 9-lobed obstruction (right). These error surfaces are a bit noisy, but exhibit typical interference patterns. If the primary and secondary noise are in phase, the total acoustic pressure is increased, if the primary and secondary noise are out of phase, they interfere destructively and the acoustic pressure is decreased, with an angular period of $1/V$, where V is the number of lobes of the obstruction. The acoustic pressure is minimal if the primary and secondary noises are out of phase and of the same magnitude. The magnitude of the secondary noise increases as the axial distance *Z* between the rotor and the obstruction decreases. In Fig. 6, the swirl of the flow is observable for small distance *Z*, especially for the *B*-1 and the *B*-lobed obstructions. Indeed, the angular position *Θ* for which the error surface is maximal (or minimal) varies with the axial distance *Z*, which means that the phase of the secondary acoustic pressure varies with *Z* (since the deflection of the flow between the lobes of the obstruction depends on the swirl, thus on *Z*).

Fig. 7 shows the upstream specific loudness without obstruction (primary loudness) and with the B+1 located at *Z*=9cm and *Θ*=42° (total loudness). This clearly shows that the fourth critical band rate which contains the BPF has been reduced. The loudness pattern has been equalized and the total loudness without control was 137 sone and with control 127 sone. The psychoacoustic impact of this control is high [7].

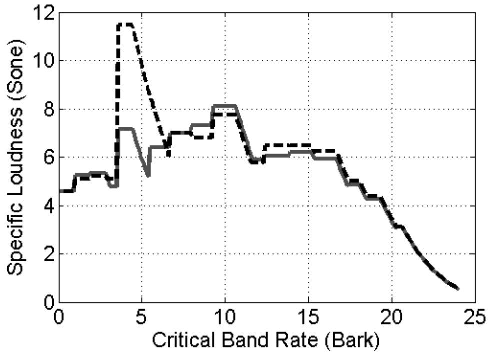

Figure 7: Specific loudness measured at the upstream microphone.
Without obstruction (dashed line) and with the B+1 lobed obstruction at Z=9cm and Θ=42° (plain line).

CONCLUSION

The feasibility of BPF tonal noise control from a centrifugal fan has been demonstrated analytically in free field and experimentally in duct. The analytical model of the noise radiated by a centrifugal fan in free field proposed in [10] showed that the B-1$^{\text{th}}$ and B+1$^{\text{th}}$ circumferential orders of the radial and tangential forces and the B^{th} circumferential order of the axial and radial forces are largely responsible for the tonal noise emitted at the BPF. The experiments showed that the control is possible with a B-1, B or B+1 lobed obstruction located in an upstream duct. The equalization of the loudness pattern proves that the control is efficient from a perceptive point of view.

Further work should consider a combination of obstructions following the method proposed in [5] to even better control tonal noise. Further experimental investigations should be in free field to validate the analytical model presented in this paper.

Finally, experimental work in progress on smaller centrifugal fans confirmed the efficiency of the control method using obstruction integrated in the inlet part of the casing of the fan without decreasing the inlet area. Thus the aerodynamic performance of the fan should remain almost equal with the original and modified inlet.

Other experiments [11] have shown the negligible impact of control obstructions to the aerodynamic performance for bigger centrifugal fans.

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