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1pEAAa2. Industrial Fan Noise Control using Flow Obstructions
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Fans are used in a lot of industrial processes and are sometimes a source of important noise for workers. We have focused on fans having a high tonal content, for which the Simple Silence technology can be applied, i.e. tonal fan noise control using obstructions in the flow.
Evaporator fan noise control in a cold storage room, based on previous work on fan noise control in free field is first presented. Then, in-duct gold mine underground air-extractor fan noise control is presented, an original analytical model is derived for noise generation and propagation inside the duct, and preliminary experimental results are provided.

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INTRODUCTION

A recent study [1] has shown the importance of fan noise in Quebec industries. It has been found that in underground mining industry, fan noise can be very high with global levels exceeding 100dB(A) and high tonal content. Fan noise level around 90 dBA with high tonal content were also measured in a cold storage room and in paint booth. Even if it does not exceed the Quebec noise exposure limit, workers may stop the ventilation because of the annoyance induced by the emitted noise. This discomfort could therefore lead to safety problems since the ventilation is not guaranteed anymore. It is therefore a priori important to control low frequency tonal noise since it also contributes to aural fatigue, discomfort and could lead to safety problems [1, 2, 3].

Tonal noise originates mainly from stationary non-uniform flow that causes circumferentially varying blade forces and gives rise to a considerably large radiated dipolar sound at the blade passage frequency (BPF) and its harmonics. In many instances, fans operate in a non-uniform flow: this is the case of engine cooling fans that operate behind a radiator/condenser system or in the wake of inlet guide vanes, or due to the wakes of upstream struts... For low frequency, passive techniques are bulky, inefficient and cannot be applied to the food industry due to hygienic constraints for example. A solution to reduce the tonal noise from axial fans is to create a secondary non-uniform flow interacting with the rotor. This secondary non-uniform flow creates secondary forces radiating a secondary tonal noise opposite in phase with the primary tonal noise so that the resulting sound is reduced. This can be done using adequately positioned flow control obstruction(s) [4, 5, 6, 7, 8, 9, 10, 11, 12]. Thus, the control is passive but the position of the obstruction must be adapted to adjust the magnitude and the phase of the secondary interaction mode in order to minimize the tonal acoustic radiation. The adjustment of the distance between the control obstruction and the rotor allows the secondary interaction mode magnitude to be adjusted while the adjustment of the angle of the control obstruction allows the secondary interaction mode phase to be adjusted. Once the location of the obstruction is optimal, the obstruction can be fixed. The control approach, also referred to as "SimpleSilence" technology in the following, is described in details in [8, 9, 13] in free field.

In this paper, the concept of SimpleSilence Technology is used to control noise in two situations encountered in the industry. Evaporator fan noise control in a cold storage room based on previous work on fan noise control in free field, is first presented. Then, in-duct underground air-extractor fan noise control is presented, an original analytical model is derived for noise generation and propagation, and preliminary experimental results are provided.

EVAPORATOR FAN NOISE CONTROL

The low frequency tone, emitted by 8 evaporator fans, at blade passage frequency (BPF=90Hz) was clearly audible in the cold storage room and also outside. It resulted in environmental tonal noise penalty (5 dB added to the global noise level, according to the Quebec Ministry of the Environment) in the residential area near the cold storage room. In the cold storage room, this tonal noise penalty was added to the global noise level according to the Quebec Health and Safety at Work rules. The primary non-uniform flow, that causes tonal noise, mainly comes from the interactions between the fan and the motor struts, the exchangers and the ice accretion in the upstream flow field of the fan (in the casing or on the struts supporting the motor). Based on free field model [8], for low speed axial fans, the circumferential flow pattern having the same number of lobes as the number of blades (B) will emit intensive sound at the BPF. The concept of "SimpleSilence" technology is to produce destructive interference between the primary source and a B-periodic obstruction [8, 9]. The flow control obstruction (Fig. 1-a) is located such that the secondary radiated noise is equal in magnitude but opposite in phase compared to the primary noise.
The A-weighted sound pressure spectra measured in the residential area near the cold storage room are shown in Fig. 1-b (without the obstructions and with the obstructions installed for all the 8 fans). The BPF tone was decreased by 7.6 dB. The equivalent continuous A-weighted sound pressure level was 56.8 dBA without the flow control obstructions and 52.4 dBA with the flow control obstructions, leading to an attenuation of 3.6 dB. The reduction of the broadband noise around 500Hz was unexpected. This is probably caused by rotating blade flow instabilities at the blade tip that are probably controlled by the obstructions so that the narrowband peaks around 500 Hz were decreased. CFD simulations could give more insight in the flow topology without and with the obstruction. The A-weighted sound pressure spectra measured in the cold storage room are shown in Fig. 1-c (without the obstructions and with the obstruction installed for all the 8 fans). The BPF tone was decreased by 11.3 dB in the 80 Hz third frequency band and 9.5 dB in the 80 Hz third frequency band. The equivalent continuous A-weighted sound pressure level was 89.2 dBA without the flow control obstructions and 88.4 dBA with the flow control obstructions, leading to an attenuation 0.8 dB. The global attenuation in dBA is low since the controlled tonality was filtered by the A-weighting. Global attenuation can be much more important for fans emitting tones at higher frequency, that are less filtered by the A-weighting.

(a) Evaporators with flow obstruction

(b) In the cold storage room

(c) Outside, near the residence

**Figure 1:** Sound pressure level in the cold storage room, without control (blue color) and with 8 fans controlled (red color)

No aerodynamic performance measurements were performed in the cold storage room. However, experiments on several other fans with and without flow control obstruction located in the upstream flow field showed that the control obstruction has low or almost no effect on the fan efficiency [9].

The proposed approach is well adapted to acoustically compact fans (large wavelength, compared to the source dimension), for which one unsteady lift mode has a major contribution to the radiated noise at BPF. Increasing the rotation Mach number or the radius of the fan leads to an increase of the number of the radiating circumferential unsteady lift modes contributing to
the noise at a single frequency (the modes \( B - 1 \), \( B \) and \( B + 1 \) especially contributes to the BPF radiation). In [13], a method has been proposed to combine a \( B - 1 \) and a \( B \) lobed obstruction to spatially enhance the control of the BPF for a small radiator cooling fan in anechoic conditions.

**Underground air-extractor fan noise control**

Refs. [4-12x] provide various strategies to control tonal noise that propagates in duc using flow obstructions. In this paper, it is proposed to extend previous researches dedicated to control tonal fan noise in free field to the in-duct case. An original approach was proposed in [13, 14] for the automatic positioning of a flow obstruction and multimodal obstruction combination to control tonal noise in free field, respectively. In [13, 14], the modulation created by the rotation of the obstruction allows for the primary and secondary noises to be distinguished in the frequency response of the sound field. When the primary and secondary noise (generated by the interaction between the rotor and the obstruction) are equal in magnitude, the radial extent of the obstruction or axial distance between the rotor and the obstruction is optimal. 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 following section presents a multi-stage rotors analytical model, to take into account the rotation of the obstruction.

**In-duct model**

![Figure 2: Coordinate system, from [15]](image)

According to Goldstein [15], the fluctuating density inside an infinite duct can be expressed as:

\[
p' = \frac{1}{4\pi c_0^2} \sum_{m=-\infty}^{+\infty} \sum_{n=1}^{+\infty} J_m(k_{m,n}r) e^{i m \phi} \int_{-\infty}^{+\infty} e^{-i(y_{m,n,x_1} + \omega t)} \int \frac{1}{k_{n,m}} \int_{-T}^{T} \int_{S(y\tau)} J_m(k_{m,n}r') e^{-i(m\phi - \gamma_{m,n} \gamma_1)} \left( \frac{m}{r'} f_D - \gamma_{n,m} f_T \right) e^{i \omega \tau} d\tau dS(y) d\omega
\]

where \( c_0 \) is the speed of sound, \( m \) is the azimuthal mode order, \( n \) is the radial mode order, \( J_m \) is the ordinary Bessel function of order \( m \), \( r \) is the receiver radial coordinate, \( \phi \) is the receiver
azimuthal coordinate, $\hat{\phi}$ is the emitter azimuthal coordinate in the stationary reference frame, $f_D$ is the drag force, $f_T$ is the thrust force, $S$ is the surface where the source must be integrated, $\Gamma$ is the normalized pressure autocorrelation function, $\omega$ is the acoustic angular frequency, $t$ is the time, $r'$ is the acoustic source radial coordinate and $y$ is the position source vector.

$$k_{m,n} = \sqrt{\beta^2 - \beta^2 n^2}, \quad \gamma_{m,n} = \frac{M k_{m,n}^2}{\beta^2} \pm \frac{k_{m,n}^2}{\beta^2}, \quad \text{where } M \text{ is the Mach number, } U \text{ is the flow velocity and } k_0 \text{ is the acoustic wave number.}$$

It is convenient to express the source integral in terms of a coordinate system $\zeta$ that rotates with the blades. The cylindrical coordinates corresponding to this frame are $r', \psi'$ and $\phi'$ as $\psi' = \psi - \Omega^{(k)} t$. Using the coordinate system $\zeta$ and according to Eq (1), the fluctuating density caused by the interaction between a stage $k$ and a stage $l$ can be expressed as:

$$[\rho']^{(k)-(l)} = \frac{1}{4 \pi c_0^2} \sum_{m=-\infty}^{+\infty} \sum_{n=1}^{\infty} \sum_{p=-\infty}^{+\infty} \sum_{p=1}^{+\infty} J_m(k_{m,n} r') e^{i m \phi} e^{i n \psi} k_{m,n} \int_{-\infty}^{+\infty} e^{-i \gamma_{m,n} x_1 + \omega t}$$

$$\times \int_A J_m(k_{m,n} r') e^{i (m \psi - \gamma_{m,n} \epsilon_0)} \int_T A(r') (\frac{\partial}{\partial r'} \bar{F}^{(k)-(l)}_{\Omega^{(k)}} - \gamma_{m,n} \bar{F}^{(k)-(l)}_{\Omega^{(k)}}) e^{i (\omega - m \Omega^{(k)}) t} d\tau d r' d\phi' d\omega$$

where $A$ is the projected area of the blades on the rotational plane of the fan. The net thrust and drag forces per unit projected area acting on the blades of a stage $k$ (rotating at angular velocity $\Omega^{(k)}$) at the point $r', \phi'$ due to its interaction with a stage $l$ rotating at $\Omega^{(l)}$ are decomposed into Fourier series as follows:

$$F_{\alpha (k)-(l)} (r, \tau) = \sum_{p=-\infty}^{+\infty} F_{\alpha (k)-(l)} (r, \tau) e^{-i p (\Omega^{(k)} - \Omega^{(l)}) \tau}, \quad \text{for } \alpha = T, D$$

$$P_{\alpha (k)-(l)} (r, \omega) = \int_T F_{\alpha (k)-(l)} (r, \tau) e^{-i p (\Omega^{(k)} - \Omega^{(l)}) \tau} d\tau, \quad \text{for } \alpha = T, D$$

These forces can typically result from the viscous wakes from an upstream stage $l$ that impinge on a downstream stage $k$ or from potential flow interaction between two close stages [15]. Using the formula $\lim_{T \to +\infty} \int_{-T}^{+T} e^{i (\omega - p (\Omega^{(k)} - \Omega^{(l)}) + m \Omega^{(k)}) t} d\tau = 2 \pi \delta (\omega - p (\Omega^{(k)} - \Omega^{(l)}) + m \Omega^{(k)})$, Eq. (2) becomes:

$$[\rho']^{(k)-(l)} = \frac{1}{2 c_0^2} \sum_{m=-\infty}^{+\infty} \sum_{n=1}^{+\infty} \sum_{p=1}^{+\infty} J_m(k_{m,n} r') e^{i m \phi} e^{i n \psi} k_{m,n} \times \left( m \bar{D}_{n,m,p} - \gamma_{m,n,p} \bar{T}_{n,m,p} \right)$$

with $\phi_0 = \frac{2 \pi b^{(k)}}{B^{(k)}} + \psi_0$, $k_{m,n,p} = \sqrt{\frac{p (\Omega^{(k)} - \Omega^{(l)}) + m \Omega^{(k)}}{c_0}} \gamma_{m,n,p} = \frac{M k_{m,n}^2}{\beta^2}$, $\bar{D}_{n,m,p} = \int_A J_m(k_{m,n} r') e^{i \gamma_{m,n,p} \psi} \bar{P}_D d\psi$, $\bar{T}_{n,m,p} = \int_A J_m(k_{m,n} r') e^{i \gamma_{m,n,p} \psi} \bar{T}_P d\psi$.

Assuming that each rotor blade of stage $k$ is subjected to the same perturbation coming from stage $l$, only shifted in time and space, then the unsteady loading is the same on each blade in magnitude but shifted in phase (this hypothesis is valid if the perturbation from stage $i$ is not time varying in the reference frame of stage $l$); and since $\sum_{m=1}^{+\infty} e^{-i (m+\phi) \beta^{(k)}} = \sum_{m=1}^{+\infty} B_{\phi}^{(k)} \delta (\phi + m \Omega^{(k)})$, the acoustic density field caused by forces acting on the $k^{th}$ rotor blades coming from a $l^{th}$ rotor periodic flow is:

$$[\rho']^{(k)-(l)} = \frac{1}{2 c_0^2} \sum_{s=-\infty}^{+\infty} \sum_{p=-\infty}^{+\infty} \sum_{p=1}^{+\infty} J_m(k_{m,n} r') e^{i (s \beta^{(k)} - p \phi) \beta - \gamma_{n,m,p} \psi} \times \left( m \bar{D}_{n,m,p} \right)^{(k)-(l)}$$

$$\times \left( \left[ m \bar{T}_{n,m,p} \right]^{(k)-(l)} \right)$$

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with \( k_{n,m,p} = \sqrt{\left( \frac{sB^{(k)} \Omega^{(k)} - p\Omega^{(l)}}{c_0} \right)^2 - \beta^2 \gamma_{m,n}} \), \( \gamma_{n,m,p} = \frac{M}{\beta} \left( \frac{sB^{(k)} \Omega^{(k)} - p\Omega^{(l)}}{c_0} \right) \pm \frac{k_{n,m,p}}{\beta^2} \),

\( \hat{D}_{n,m,p}^{(k)-(l)} = \int_A J_m(k_n \varphi) e^{i(r_n \varphi_r - m \phi)} \hat{P}_p^{(k)-(l)}(\varphi, \psi) \, d\varphi \, d\psi \),

\( \hat{T}_{n,m,p}^{(k)-(l)} = \int_A J_m(k_n \varphi) e^{i(r_n \varphi_r - m \phi)} \hat{P}_p^{(k)-(l)}(\varphi, \psi) \, d\varphi \, d\psi \).

A remarkable and useful result of this multi-stage model is the angular frequency given by \( \omega = sB^{(k)} \Omega^{(k)} - p\Omega^{(l)} \). The modulation created by the rotation of an obstruction (considered as a stage) allows for the primary and secondary noises to be distinguished in the frequency response of the sound field. Finally, the cut-off relation is given by:

\[
\left( \frac{sB^{(k)} \Omega^{(k)} - p\Omega^{(l)}}{c_0} \right)^2 < \beta^2 \frac{\gamma_{m,n}}{\gamma_{n,m,p}} \text{ or } \left| \frac{sB^{(k)} M_l^{(k)} - pM_l^{(k)}}{sB^{(k)} - p} \right| < \sqrt{1 - M^2}
\]  

Eqs. (6-7) are consistent to the Goldstein model for one rotor interacting with steady wakes if \( \Omega^{(l)} = 0 \). The in-duct fluctuating density from rotor/stator, stator/rotor and rotor/rotor interactions are particular cases of the model described in this section.

Finally, if the fluctuating density radiated by \( K \) radiating rotors due to flow disturbance originating from \( L \) rotors can be linearly superimposed, the total pressure field can be written

\[
\rho'(x_1, \omega) = \sum_{k=1}^{K} \sum_{l=1}^{L} \left[ \rho_l^{(k)-(l)}(x_1, \omega) \right].
\]

In the following sections, the modulation generated by the interaction between a rotating obstruction and a rotor is not presented, thus \( \Omega^{(l)} = 0 \).

**Control Approach**

In a duct, the cut-off relashionship gives the modes \( m, n \) that propagate inside the duct at frequencies \( sB^{(k)} \Omega^{(k)} \). The primary modes can be controlled by superimposing a periodic secondary forces \( F_{sec.p} \) of equal intensity but opposite in phase relative to the most radiating circumferential primary forces components \( F_{prim.p} \). Assuming that the primary and secondary inflow velocity fields (thus the forces) can be linearly added, the total fluctuating density \( \rho_{tot}(x_1, \omega) \) is the sum of the primary sound field \( \rho_{prim}(x_1, \omega) \) and the secondary sound field \( \rho_{sec}(x_1, \omega) \), thus \( \rho_{tot}(x_1, \omega) = \rho_{prim}(x_1, \omega) + \rho_{sec}(x_1, \omega) \). The linear assumption has to be checked in further investigations.

The secondary tonal noise magnitude should increase as the axial distance \( Z \) between the rotor and the control obstruction decreases or as the radial extent \( \Delta R \) of an obstruction increase. The optimal axial location \( Z_{opt} \) or the optimal radial extent \( \Delta R_{opt} \) of the obstruction can be found when the magnitudes of primary and secondary tonal noise are equal. The phase of the secondary tonal noise is a priori a linear function of the angular location \( \phi \) of the obstruction. The optimal angular location \( \phi_{opt} \) of the control obstruction is found when the primary and secondary tonal noise are opposite in phase, thus \( \rho_{tot}(x_1, \omega) \) is minimized.

**EXPERIMENTS**

**Experimental set-up**

Fig. 3 shows the axial fan used in this experiment (the experiments presented in this paper have been performed in a gold mine but this picture shows the installation at Sherbrooke university). An underground air-extractor axial fan was installed in a duct. On the upstream side of the fan, a positioning device allowed for the obstruction to be moved in the axial and angular directions. The fan had \( B = 10 \) regularly spaced blades and its rotational velocity was 3345 RPM.
corresponding to a BPF at $B \Omega/(2\pi) = 590$ Hz. $B^{(j)} = V = 8$ regularly spaced outlet guide vanes (OGV) was located downstream the rotor. The internal diameter of the fan was 33 cm while its external diameter was 53.3 cm. The diameter of the duct was 55.9 cm. The stagger angle of the blades was around $\chi = 30^\circ$ and the blade chord was 12.7 cm.

**Figure 3:** Experimental set up at Université de Sherbrooke with the underground air-extractor axial fan.

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**Control**

![Control Diagram](image_url)

**Figure 4:** Sound pressure level at BPF, 2BPF and 3 BPF as a function of the angular position and the radial extension of the flow obstruction

The first step is to estimate the propagating modes in the duct using the cut-off inequality Eq. (7) at each frequency to be controlled. In order to reduce the propagating sound field inside the duct, one or several of these modes can be controlled using sinusoidal or trapezoidal obstructions having the same circumferential periodicity as the azimuthal order of the forces generating the propagating acoustic modes.

To control each of these acoustic propagating modes $m$ separately, the obstructions must have
p = sB^{(k)} - m lobes. In the preliminary study shown in this paper, only a 10-lobed obstruction was used to partly control the plane wave propagating at BPF.

From the geometrical characteristics of the fan and the duct, the modes that can propagate are |m| < 2 at BPF (s = 1) the modes |m| < 4 at 2BPF (s = 2).

The measurements were performed in a 130 m underground gold mine gallery in Val d’Or, Québec. A trapezoidal obstruction (10 trapezes, with an inner radius of 14.6 cm) has been tested to control the plane wave in the duct. Fig. 4 shows the sound pressure level at the BPF, 2BPF and 3BPF as a function of the angular position and the radial extension of the flow obstruction (outer radius of 5.7 cm, 7 cm and 8.3 cm) at two microphones located in the underground gallery, located at 3 m upstream and downstream the fan.

Fig. 4 shows that the 10-trapezoidal obstruction has an impact on the BPF tone but also at 2BPF and 3BPF. At BPF, attenuations of 13 dB and 11.5 dB have been achieved at the upstream and downstream microphones respectively for the same angular position of the obstruction, but for a slightly different radial extension of the obstruction. This compromise between upstream and downstream control has already been observed for other fans.

CONCLUSION

Two types of axial fans have been controlled in this study using SimpleSilence technology: evaporators axial fans used in a cold storage room and an underground in-duct air-extractor fan.

For an acoustically compact fan in free field, the spectral content and the directivity of the primary tonal noise is due to a combination of spinning modes caused by the unsteady blade lift along the circumferential direction of the rotor. Usually, for largely subsonic fans with B blades, the most radiating modes are the sB-lobed mode, which radiate like an axial dipole at frequency sBΩ. Controlling these modes can lead to large noise attenuation. However, when radial or tangential forces are significant, the unsteady lift modes sB-1 and sB+1, and possibly lower and higher modes, generate acoustic radiation, especially in the rotor plane. For an in-duct fan, the spectral content and the propagating tonal noise is also due to a combination of spinning modes caused by the unsteady blade lift along the circumferential direction of a rotor stage. However, unlike the tonal noise emitted from fans in free field, the modes that can propagate inside the duct have to respect the cut-on condition of the duct.

The model presented in this paper generalizes the model proposed by Goldstein [15] by including a multi-stage rotor in an infinite duct. Future experimental investigations can be based on this model, such as automatic positioning of flow obstruction (adapted from [14]), multi-modal obstruction design (adapted from [13]) and in-duct modal characterization from the modulation signature caused by the interaction between the rotor and a rotating obstruction (adapted from [16]). Further investigations are in progress at Université de Sherbrooke to validate this model and to control tonal noise from the underground in-duct air-extractor fan.

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