FAN TONAL NOISE REDUCTION USING CALIBRATED OBSTRUCTIONS IN THE FLOW: AN EXPERIMENTAL APPROACH

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SUMMARY

The paper presents the experimental results of a technology aiming at the cancellation of the tonal noise of fans, using calibrated obstructions placed upstream or downstream in the flow close to the fan. When placed at the optimal distance and angle, the obstruction generates pressure pulsations that interact with the fan pressure fluctuations in a destructive way, allowing the cancellation of the noise at the blade passing frequency, and, in some cases, at its first harmonic. A parametric study has been conducted on axial and centrifugal fans. Parameters that have been studied are the obstructions shape and position and the fan speed. More than 50 different configurations have been investigated.

INTRODUCTION

The possibility to use calibrated obstructions to cancel fan noise at the blade passing frequency has been proposed and demonstrate by A. Gerard and his colleagues at Sherbrooke University [1][2]. This technology consists in placing calibrated obstructions in the air flow upstream or downstream the fan. When placed at the optimal distance and angle, the obstruction generates pressure pulsations that interact with the fan pressure fluctuations in a destructive way, allowing the cancellation of the noise at the blade passing frequency, and in some cases, at its first harmonic.

Important attenuations can be achieved when the optimal position of the obstruction is driven by an automatic control means, but the cost of such control system limits the spreading of the technology. The present work is a systematic investigation of the influence of the obstructions type and position on the overall and blade passing frequency sound pressure level, mainly aiming at defining the tolerance zone around nominal parameters [3]. The size of the tolerance zone would have a great influence on the effective usability of this technology on real machines.

BPF NOISE MECHANISM

Fans generate noise following different mechanisms, witch have been summarized by Neise in his well known diagram [4]. Among this mechanisms, the one that is in concern here is the discrete part of the noise spectrum with spectral lines at the blade passing frequency and its harmonics, reference by Niese as 'Non-uniform stationary flow'. On a non uniform flow field, the local forces exerted by the blades on the fluid are unsteady, since the local angle of attack and velocity of the flow varies. This generates a discrete noise spectrum that can represent a large part of the total noise radiated by the fan.

Then, the size and shape of obstacles set upward or backward of the fan blades, (heat exchangers, motors, supports, …) as they interact with the flow, can strongly affect the amplitude of this tonal contribution, leading in general to an amplification of the noise. A. Gerard and his colleagues have shown that other obstacles, when properly chosen and placed, are able to lead to a strong reduction of the noise at the blade passing frequency.

EXPERIMENTAL SETUP

Test bench

The tests have been conducted on the CETIM fan aeroacoustics bench. This setup allows to measure fans noise and overall flow in a housing that is representative of the engine compartment of some slowly moving machines (excavators, agricultural tractors, …) [5] (Fig 1).

The fan is driven by an electrical motor.

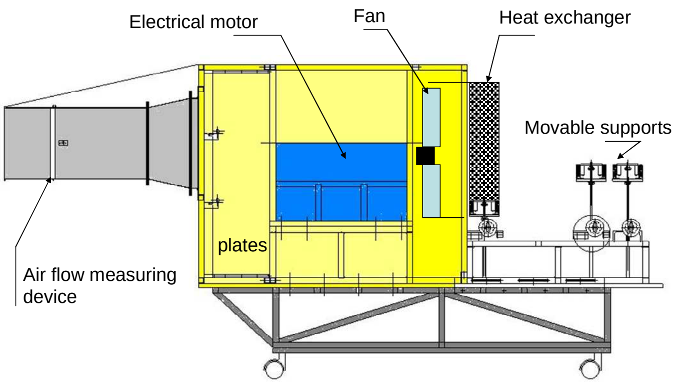

Figure 1- Fan test rig

The obstructions are positioned with the help of a two axes (axial and angular) automaton (fig 2).

Measuring devices

The data that have been recorded in each case are:

- the sound pressure level (SPL), measured on four microphones located in the center of the rectangular open faces in the front part of the test bench (Fig 2). The sound power emitted in the front part of the system is deduced from these SPL measurements and the areas of the faces.

- The overall air flow measured with a device that consists in two multi-holes Pitot wings, placed in cross position in the exit tube.

- The fan rpm measured with an optical device.

The following measurements are done for every configuration:

- SPL and flow measurements for a stabilized fan speed with the obstruction far enough to have a negligible influence on the system. This is the reference measurement.
- Systematic SPL and flow measurements for a stabilized fans speed on a mesh covering the range of axial and angular positions of the obstruction.
- SPL and flow measurements for the fan speed varying in a ramp from 0 to 2200 rpm in 369 seconds, without the obstruction, and with the obstruction located in its nominal axial and angular position.

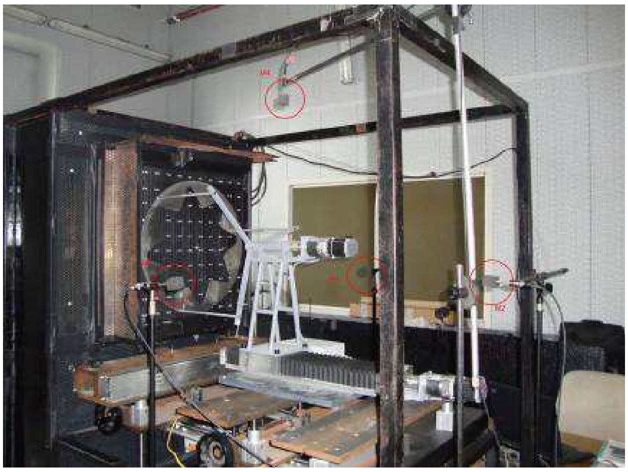

Figure 2- Microphones M1 to M4 location, and automaton.

Fans

Two fans have been used: an axial fan with 8 blades, diameter 600 mm, and a backward centrifugal fan with 6 blades, diameter 570 mm (fig 3).

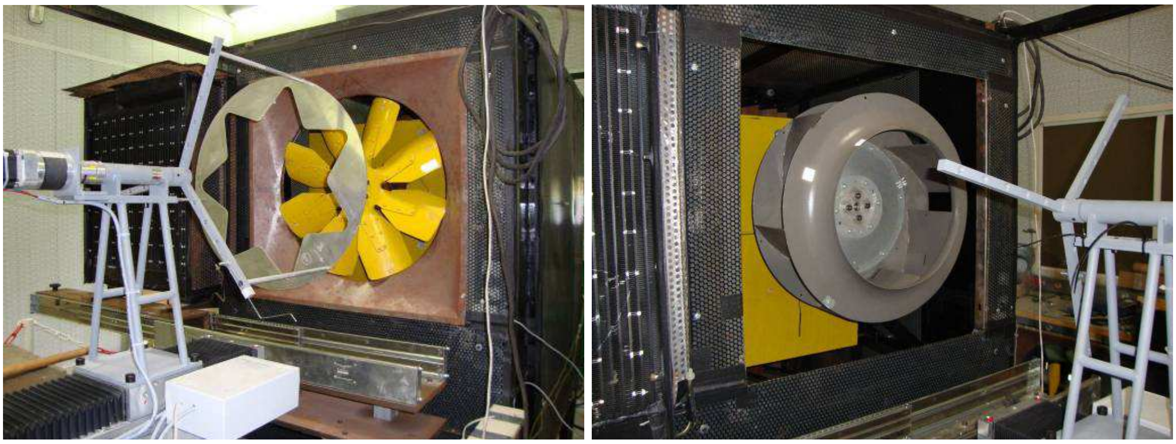

Figure 3- Axial (left) and centrifugal (right) fans.

Obstructions

Two different set of obstructions have been used.

The first set is composed of periodic indentations disposed at the periphery of a disk, with a smooth sinusoidal shape (M type) or a sharp dissymmetrical shape (D type). The number of indentations range from N-1 to N+1, where N is the number of blades of the fan.

The second set is composed of a unique obstacle with the shape of a portion of disk (C type) (Fig 4).

Figure 4- Example of M type (left), D type (center), and C type (right) obstructions.

TEST RESULTS

Fifty five configurations have been tested, 30 with the axial fan, and 25 with the centrifugal fan. Only a subset of the results is presented here.

Axial fan, periodic obstructions

The figure 5 presents an example of the sound level map obtained by sweeping an obstruction on its axial and angular positions. The colour level represents the sound pressure difference in dB, average on the 4 microphones, between the reference configuration (no obstruction effect) and the configuration with the obstruction, at the blade passing frequency (BPF). The angular step is 5 degrees, and the axial step is 5 mm. An axial value of 0 represent a position very close (5 to 6 mm) to the heat exchanger.

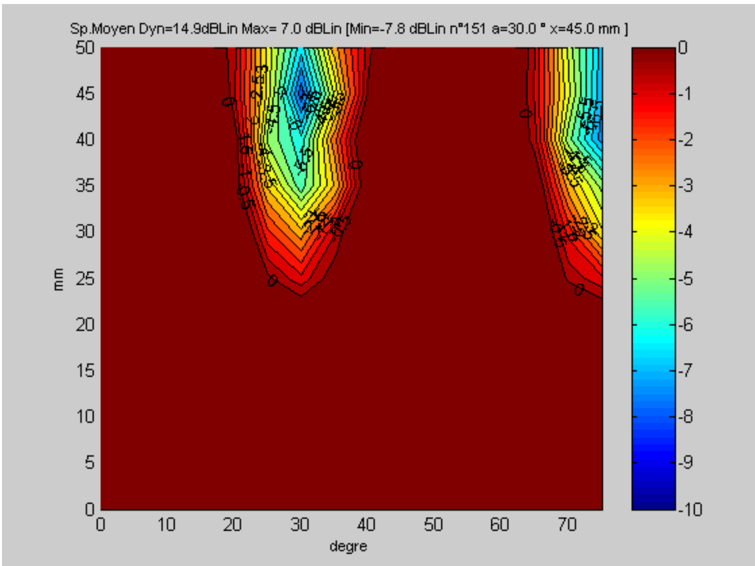

Figure 5- Example of Sound Map (D type with 8 indentations).
Horizontal axis: angle, degree, vertical axis: displacement, mm

One sees than, in that case, a maximum attenuation of 10 dB is obtained at the BPF, for an obstruction rotated 30 degrees and moved back 45 mm from its reference location.

The following pictures (Fig 6) presents the results obtained with the best M type and D type obstructions, for a fan speed of 2200 rpm.

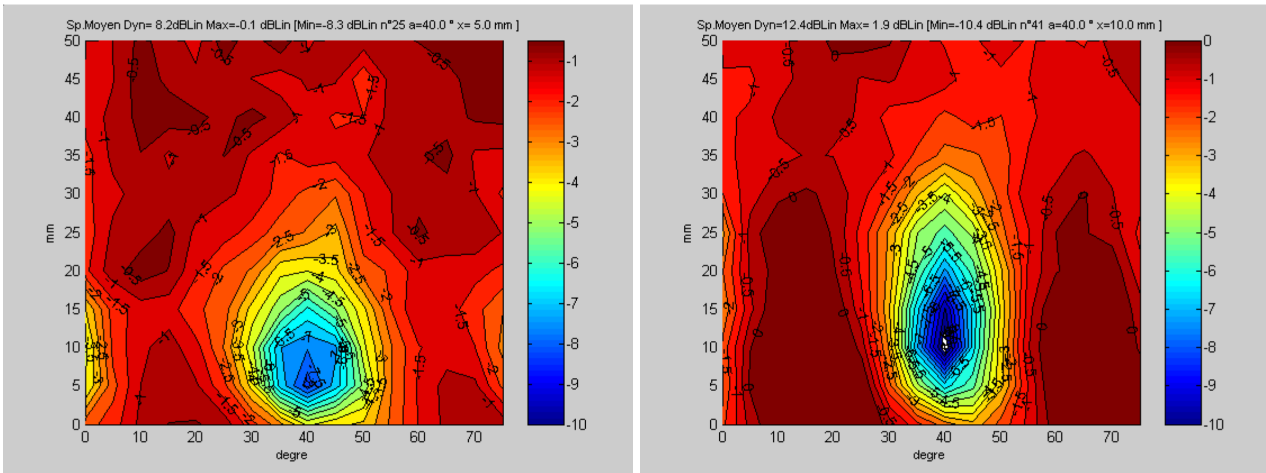

Figure 6- Sound Map obtained with the best M type (left) and D type (right) obstructions.

The best results are obtained with an obstruction having a number of indentations equal to N-1, where N is the number of blades of the fan. The angular sensitivity is similar for the two types, while the D type appears to be less sensitive to the axial location. In both cases, the optimal location is close to the heat exchanger.

The sound pressure spectra at the optimal location are presented in figure 7.

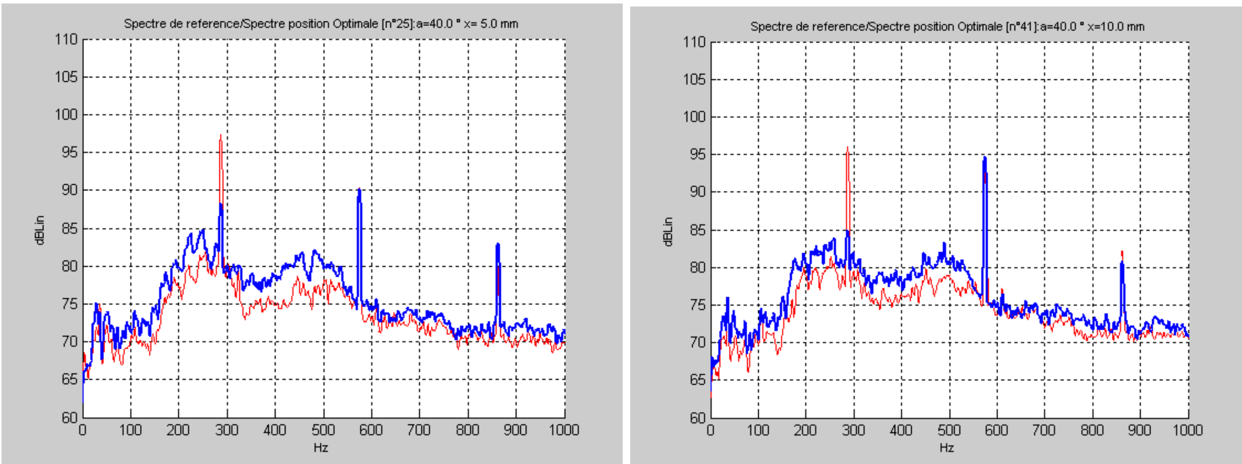

Figure 7- Sound pressure spectrum at optimal location for the best M type (left) and D type (right) obstructions.

In these configurations, only the BPF is modified, the following harmonics stay unchanged. While the sound level at BPF can be efficiently reduced (10 dB), the consequence on the overall sound power level is far less. The maximum gain on the overall sound power is 1.8 dB(A) for the M type, and 0.3 dB(A) for the D type.

When varying the fan speed for an obstruction in its optimal location for the given speed, one notices that the obstruction stays efficient in a range of some hundreds of rpm around the optimal speed. Beyond that range, the effect of the obstruction can become negative. Figure 8 shows an example of such a behaviour in the case of D type obstructions with 8 indentations.

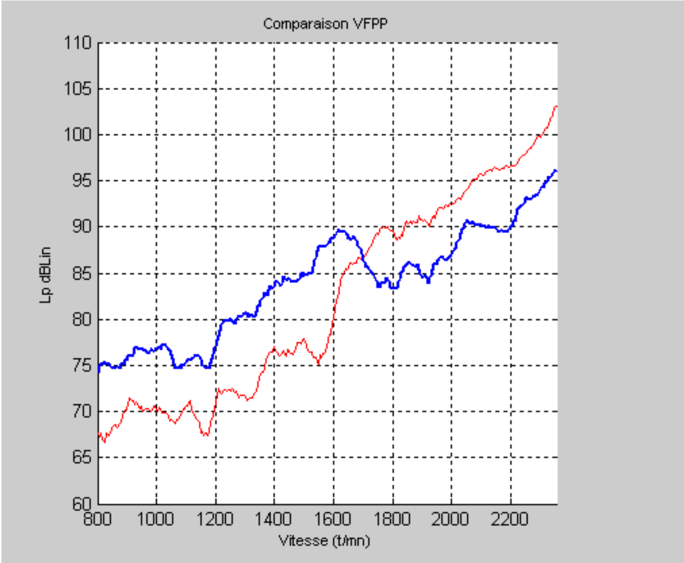

Figure 8- Sound pressure level for an obstruction in fixed position, when varying the fan speed (D 8 type)
SPL without obstruction (red curve) and with obstruction (blue curve)

In the proposed example, the obstruction keeps its efficiency from 2200 to 1700 rpm, but increase the BPF level by 8 dB for the velocity under 1600 rpm. Anyway, using this device in a fixed position would allow reducing by 8 dB the sound level in the zone where it is the most noisy.

The influence of these obstructions on the air flow depends on the distance between the obstruction and the heat exchanger. Figure 9 shows that the difference in air flow for a system without obstruction, and with the obstruction at its nominal position (M type with 8 indentations) is around 3.5%.

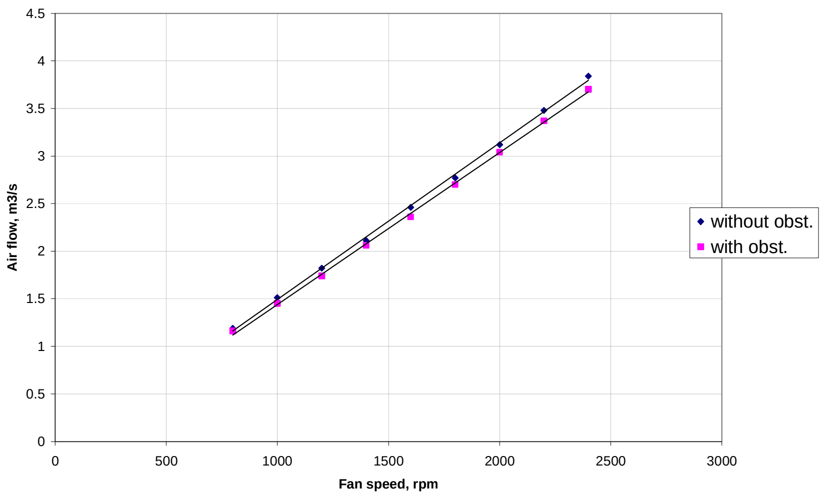

Figure 9- *Influence of the obstruction on the flow (M 8 type).*

If one would wants to compensate these decrease of flow by an increase of rotational speed, as the noise power scales roughly to 60 lg N where N is the rotational speed, these would results in a noise power increase of 0.9 dB.

Axial fan, unitary obstructions.

The typical behavior of an obstruction composed of a sole part of reduced size (C type-Figure 4) is shown in the following pictures (Fig 10 to 11)

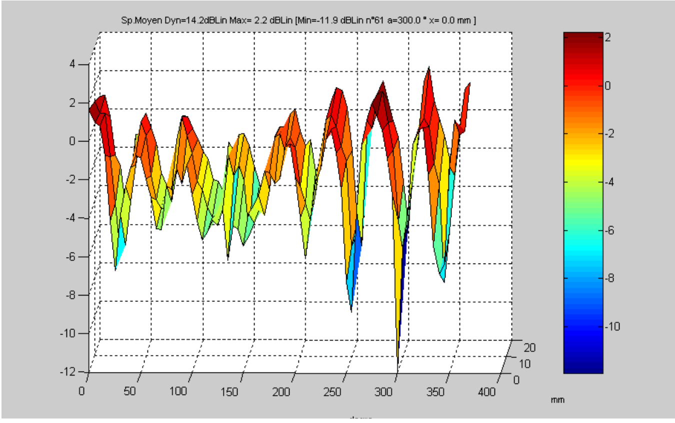

Figure 10- Sound pressure level at BPF for C3 obstruction. Influence of angular and axial position. Horizontal axis is the angle in degrees and the second axis is the distance in mm

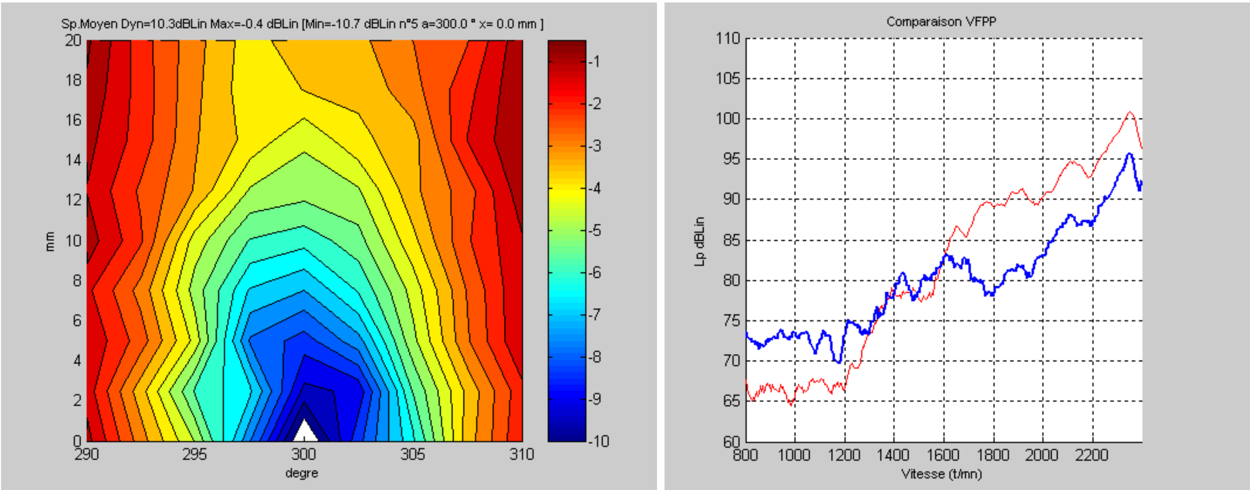

Figure 11- Best C type obstruction (C5). Sound map around nominal position at 2200 rpm (left) and sound level when varying fan speed (right), obstruction at fixed position Red: without obstruction, blue: with obstruction..

Figure 9 shows that several angular positions can lead to a significant reduction of the BPF level. At the best location, the result is very sensitive to the angular (+-1 degree) and to the axial position (+-2 mm). However, the sensitivity to the fan speed is smaller (figure 9). The influence of this type of obstruction on the air flow is very low.

Centrifugal fan

The general trends for the centrifugal fan are similar to those observed for the axial fan. However, some differences can be noticed.

The efficiency of the obstruction on the BPF tends to be larger than what has been observed for the axial fan. The performances of the M and D type are very similar (Figure 12). The best results are achieved with the obstructions having the same number of indentation as the number of fan blades.

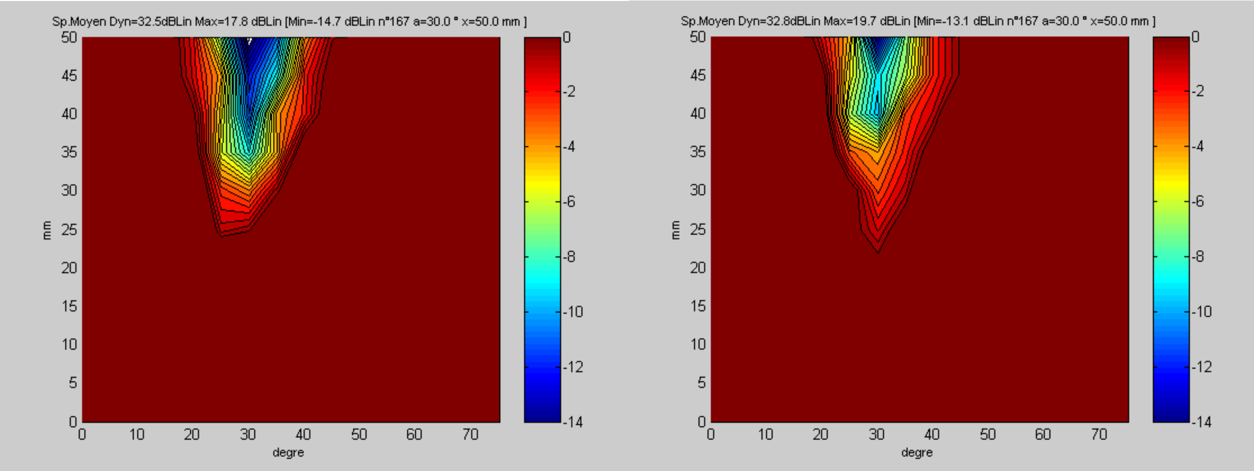

Figure 12- Sound map for the best M (left) and D(right) type obstruction with the centrifugal fan at 2200 rpm.

The nominal distance between the obstruction and the heat exchanger is greater in the case of the centrifugal fan. The axial and angular sensitivity is similar for the two types of fan.

The efficiency at BPF is almost independent of the centrifugal fan speed (figure 13). One can note that the influence on the total sound level can be significant at high speed (-3 dB at 2200 rpm).

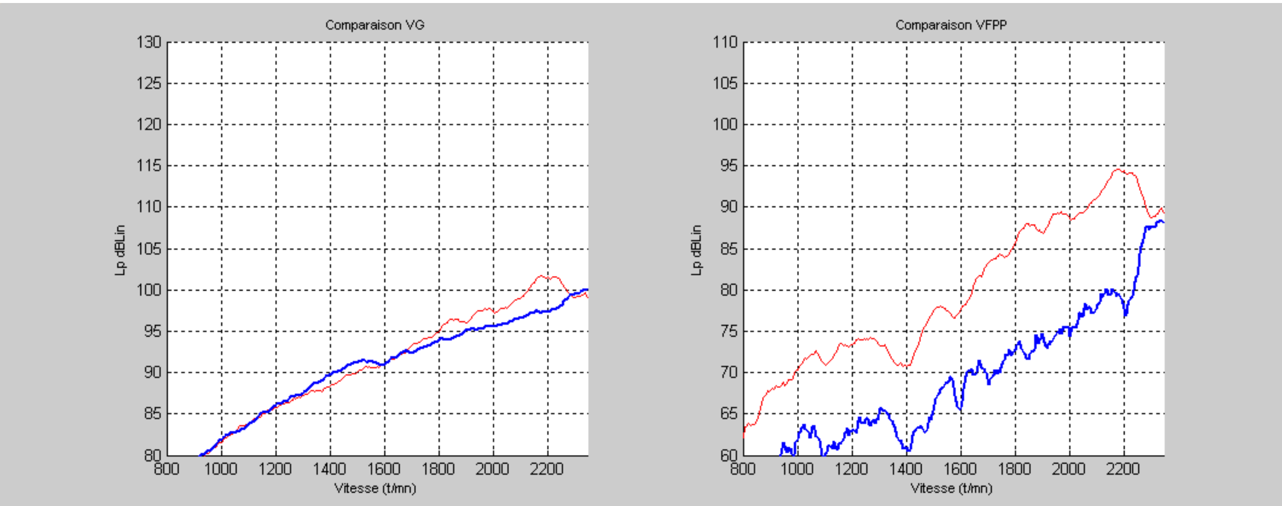

Figure 13- Influence of the fan speed on the overall sound level (left) and at BPF (right). M type obstruction with 6 indentations. Red: without obstruction, blue: with obstruction.

C type obstructions are a bit less efficient than the other types. The gain at BPF is in the range of 11 to 13 dB, compared to the 13 to 15 dB obtained with the M and D type.

As the nominal distance between the obstruction and the heat exchanger is greater in the case of the centrifugal fan, the influence of the obstruction on the flow is lower compared to the axial fan case. The flow decreases by 1% to 2% when the obstruction is in nominal place. To compensate for this flow reduction by an equivalent increase of rotational speed would results in a noise power increase of 0.25 to 0.5 dB.

CONCLUSIONS

Applied to a fan cooling system that is representative of what is use on slowly moving machines such as excavators or agricultural tractors, the use of various types of obstruction upward of the fan has shown that a 10 to 15 dB reduction on the sound power level at blade passing frequency is achievable.

As only the BPF is strongly affected by the obstructions, the maximum noise reduction to be expected on the overall sound level is small: 2 dB(A) for the axial fan, and up to 4 dB(A) for the centrifugal fan.

These performances are obtained by adjusting the obstruction position in a fairly tiny zone, the precision required being of the order of 1 or 2 degrees in angular position, and 3 to 5 mm in axial position. However, once the location properly set, the performances are kept for large variations of the fan speed, covering several hundreds of rpm

Work has still to be done to define obstructions shapes that would allow enlarging the working area. Numerical simulation should be an efficient tool to go towards shape optimisation, as the first results that have been obtained on ongoing work on Computational Aeroacoustics seems encouraging [6].

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