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STUDY OF AEROACOUSTIC BEHAVIOR OF AN AXIAL FAN: EXPERIMENTAL VALIDATION

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ABSTRACT
The work presented here forms part of a very broad specific action carried out in the LEMFI (Laboratoire d'Energétique et de Mécanique des Fluides Internes) concerning the unsteady flows in the turbomachinery of compression in general and aeroacoustic of fans in particular. The main objective is the determination of the influence of the constructive geometrical parameters on the aeraulic and acoustic global performances.

1 - NOMENCLATURE

- c_0: velocity of the sound in homogeneous and stationary environment
- $H(f)$: function of Heaviside
- Lp_{ref}: reference acoustic pressure level
- P_i: local component of the effort applied on the profile, with $P_i = \sigma_{ij} n_j$
- M_r: relative Mach number of the sources in the direction of the observer
- N: rotational speed
- n_j: normal unit vector on the surface of the profile
- p: static pressure
- u_i: component of the local velocity of the fluid
- V_n: velocity of the points of solid surface
- $\delta(f)$: Dirac function
- ρ: density of the fluid
- ρ_0: density of the homogeneous and stationary fluid
- σ_{ij}: tensor of the compression stresses
- ∇: operator gradient

2 - INTRODUCTION
The first research on the reduction of the noise produced by the turbomachinery began soon after the beginning of the aircraft industry. Several studies showed that an important share of the noise produced by this type of machines is connected directly with the stationary efforts exerted by the fluid on the rotor

blades of the machine. In 1936, Gutin [1] established the formalism allowing the prediction of the noise generated by these stationary forces in rotation.
In 1952 Lighthill [2] introduced the general theory on the flows of the jets thus giving birth to the acoustic analogy. Its interest went on the understanding and the prediction of the aerodynamic noise and particularly the noise of the jets. This work was extended by Curle [3] in 1955, to take into account the presence of the solid walls. Their interaction with the turbulent flow has as a consequence a modification of the acoustic field. So surface distributions of forces occur in this interaction and are at the origin of the aerodynamic noise of dipolar nature.
While relying on the acoustic analogy such as it was developed, J. E. Ffowcs Williams and D. L. Hawkings [4] (FW-H) exploited the idea by defining similar environment to a reality environment and in which solid surfaces, in fact the rotor blades, are replaced by mathematical surfaces.

3 - UNSTEADY PHENOMENA AND AEROACOUSTIC OF FANS

The laws of conservation of the mass and the momentum are written:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = \rho_0 V_n \delta(f)$$
$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i u_j) + \frac{\partial \sigma_{ij}}{\partial x_i} = P_i \delta(f)$$

with f the function which describes the surface of the walls and having as equation:

$$\begin{cases} f(x,t) = 0 & \text{with } f > 0 \\ \nabla f = \vec{n} \end{cases}$$

where $\vec{n}$ is the normal unit vector on surface f=0.
This new approach leads to the commonly called aeroacoustic analogy equation of FW-H. It is written:

$$\frac{1}{c_0}\frac{\partial^2 p}{\partial t^2} - \nabla^2 p = \frac{\partial}{\partial t}(\rho_0 V_n \delta(f)) - \frac{\partial}{\partial x_i}(P_i \delta(f)) + \frac{\partial^2}{\partial x_i \partial x_j}[T_{ij} H(f)] \tag{1}$$

T_{ij} tensor of Lighthill having as expression: $T_{ij} = \rho u_i u_j + \sigma_{ij} - \rho c^2 \delta_{ij}$.
The law thus expressed in (1) described the behavior of homogeneous environment and simultaneously subject to three distributions of sources represented by the second member:

- $\frac{\partial}{\partial t}(\rho_0 V_n \delta(f))$ surface distribution, called thickness noise, it is due only to the effective movements of surfaces. It is monopolar type.
- $\frac{\partial}{\partial x_i}(P_i \delta(f))$ surface distribution, called load noise, coming from the interaction of the fluid with the moving surfaces. It is dipolar type.
- $\frac{\partial^2}{\partial x_i \partial x_j}(T_{ij} H(f))$ volumic distribution, called specific noise of the flow, distributed in open space left by the moving surfaces. It is quadrupolar type.

In subsonic flow, the noise due to the aerodynamic efforts applied on the surface of the blades is very dominant compared with the other types of noise [5, 6]. So any source other than the dipolar source is negligible. The acoustic pressure solution of the equation of FW-H (1), becomes after development:

$$p(x,t) = \frac{1}{4\pi}\int_S \left[\frac{R_i}{R^2(1-M_r)c_0}\left(\frac{\partial}{\partial \tau}\left(\frac{P_i}{(1-M_r)}\right)\right)_{\vec{x}}\right] dS$$

The load of the profile P is equivalent to the point of considering aerodynamics to $\rho u^2 d^2$, it becomes:

$$p \propto \frac{1}{4\pi r c_0}\frac{\rho u^2 d^2 f}{(1-M_r)^2}$$

By introducing the number of Strouhal, S_t, defined by $S_t = f\frac{d}{u}$ where f is a frequency characterizing the period of the phenomenon, one obtains:

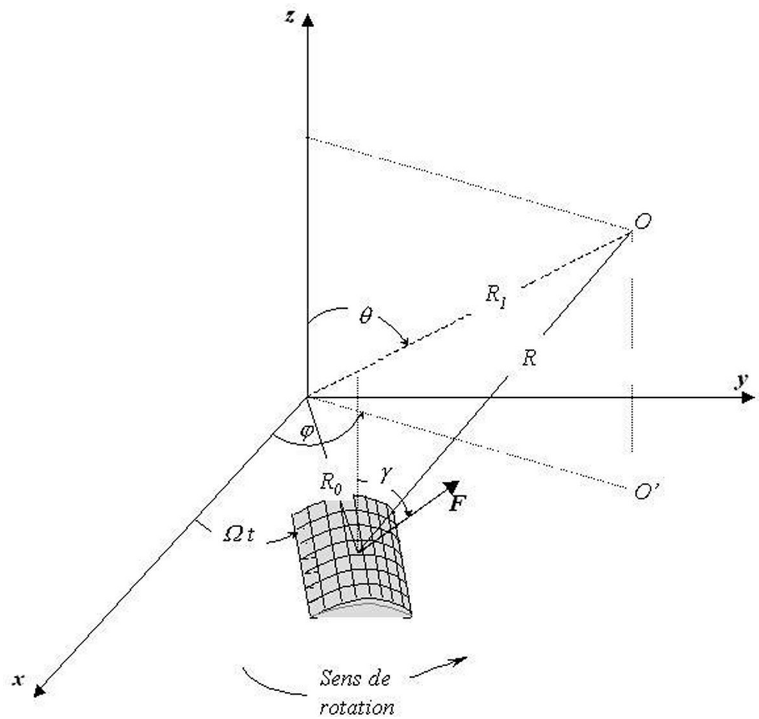

Figure 1: System of co-ordinates.

$$p \propto \frac{1}{4\pi r c_0} \frac{\rho_0 S_t u^3 d}{(1 - M_r)^2}$$

This expression reveals, concerning the acoustic power, a law in N exponent 6 since this same quantity is homogeneous with the square of the pressure.
In the very usual case of an insulated axial rotor, working at adapted flow conditions, the noise connected with the passage of the blades is accessible today by computation (Figs. 2, 3 and 4).
The first stage of this work consists in computing the field of pressure between the blades of the machine [7, 8, 9] making it possible to determine the distribution of the unsteady aerodynamic efforts on the surface of the profiles. The main part of the noise produced by the rotating machines comes from the fluctuating component of these efforts which it is advisable to determine in a precise way.
The level of the broadband noise is still not easily foreseeable. It depends on the type of fan (axial, centrifugal, mixed flow), on its immediate environment (the inlet conditions, distance between rotor and stator) load of the blade, velocity of rotation and operating point: partial flow rate or design point.
For a complete stage made up of a rotor cascade followed by a stator cascade, the interaction rotor-stator (Fig. 4) can become dominant under certain unsuited operating conditions of one or the other of the cascade.
The organization of the flow and the aerodynamics of the profiles guaranteed best required acoustic discretion. The viscous effects, the secondary loss of adaptability of the flows and another harmful aerodynamic effect to optimization appear apart from the nominal rate flow. The phenomena of wake at exit of the rotor contributes in a considerable way to increase the sound level of the machine. The amplitude of this phenomenon decreases in an important way according to the axial distance rotor stator.

4 - EXPERIMENTAL STUDY OF AN AXIAL FAN

The developed computational tools, based on the flows between the blades, allowed to define with an increasingly satisfactory precision, the fine kinematics of the internal flow. However, the complexity of unsteady phenomena governing the aeroacoustic behavior of the fans requires a theoretical approach and at the same time, in the current state of the progress of the work, an experimental validation.
The main aeroacoustic results presented relate to an axial fan (with 5 blades and 394 of specific speed) whose velocity of rotation varies from 500 to 2300 rpm (Fig. 5).

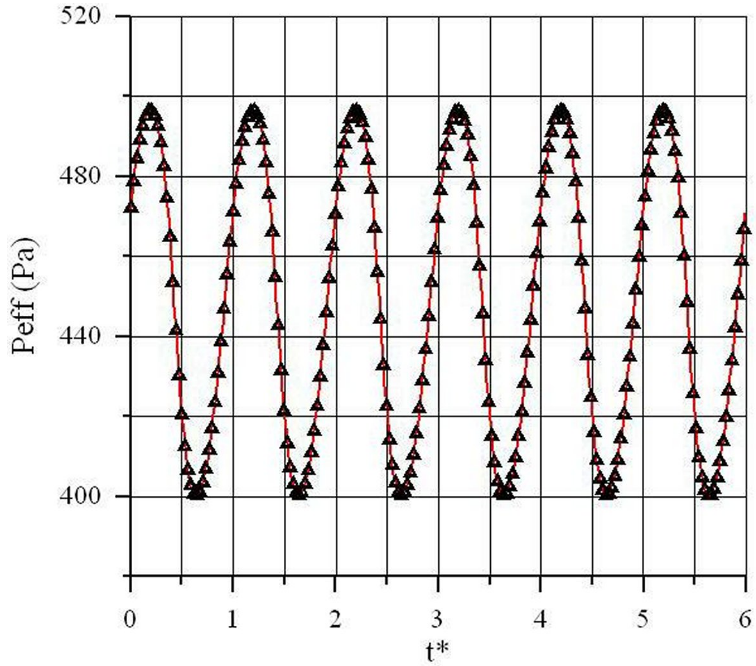

Figure 2: Computation of the field of pressure downstream at 20 mm of the trailing edge of an axial cascade.

In accordance with the European standard ISO 5801 describing the aeraulic and acoustic tests of pusher fans, we made up circuit standardized with a 352 mm diameter. It was equipped with aerodynamic devices of high precision, three numerical micromanometers coupled to a system of automatic acquisition gives access quickly to the various pressures. A double tube of Pitot of Furness type makes it possible to measure the flow rate.

On the acoustic level, the test bench was equipped with 3 microphonic probes 0.5 inch G.R.A.S with antivent cone and preamplifiers ICP. The various data are collected and analyzed using a card with format PC ISA supporting a DSP 32 bits in floating point with a resolution of 3201 pts and 4 entries of signal real time.

This whole of analysis and acoustic data acquisition makes it possible to carry out the measuring in conformity with the standard ISO 5136 relating to the determination of the acoustic power radiated by fans in duct (Fig. 6).

4.1 - Aeraulic performances

The variations of flow rate of the fan are obtained by diaphragms of various diameters. Distribution of axial velocity presents a strong radial variations should be considered in integration allowing to calculate the flow rate.

Figs. 7 and 8 give the global aeraulic performances: flow rate vs static pressure and flow rate vs total efficiency for various velocities of rotation.

4.2 - Acoustic performances

Acoustic measures are carried out starting from the three microphones located at approximately 4 m downstream from the fan. One has access to the following elements respectively:

Acoustic pressure:

In Fig. 10, we can observe a traditional spectrum of the acoustic pressure at 2000 rpm and 1800 m^3/h (design point): blade passing frequency and various harmonic are well underlined.

Weighted global level of acoustic pressure:

The global level weighted A of the acoustic pressure varies strongly versus the flow rate by presenting a minimum around the design point (Fig. 9). These global performances result from a series of records according to the velocity of rotation, diagrams of the type diaphragm iso of Fig. 12.

A comparative analysis was carried out by taking into account the contribution of the discrete frequency noise and the broad band noise to the total sound level. The results presented on Fig. 13 show the variation of the level of global acoustic pressure according to rotation speed of the fan. It is notorious that the contribution of fundamental is about 2 dBA.

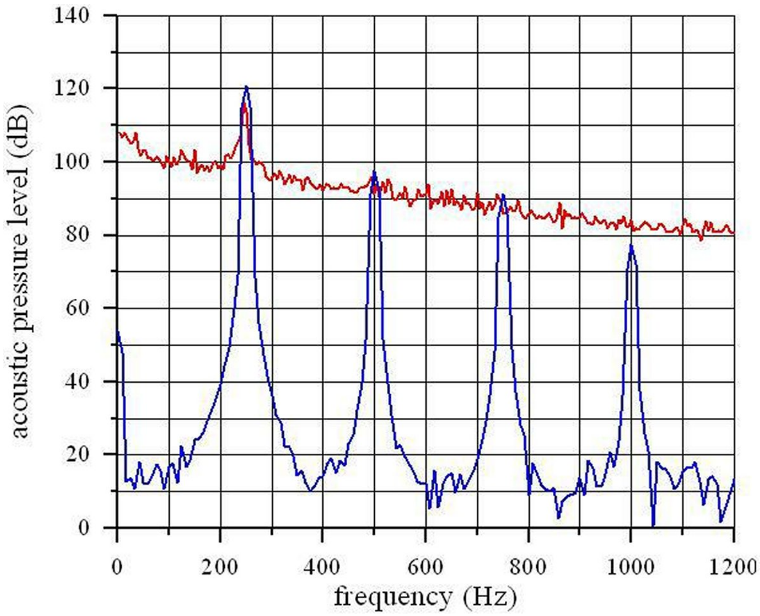

Figure 3: Spectral analysis of the signal and comparison with the experimental result.

Classification of the fan:
In accordance with the constructive data, it is interesting to note that the whole of the results obtained allows to give the evolution of the level of global noise according to the velocity of rotation. The exponent associated with the rotation speed varies with the flow rate, Φ2, Φ4 and Φ7 (Fig. 13). The average value of this exponent is around 4 which indicates to us that the sources of noise are mainly dipole and quadrupole. Fig. 13 makes it possible to position the fan compared to other industrial achievements.

5 - CONCLUSIONS

The dimensional analysis carried out on recent modeling relating to the noise at rotational frequency allowed to establish laws of variation of the sound power level versus the velocity of rotation. The non compact of the profiles is not taken into account. A complementary work is carried out in parallel to analyze the aerodynamic non-uniformity of the profile.
The experimental results obtained on the axial fan show that the acoustic signature of this one results in the emergence of the discrete frequency compared to the broad band noise due mainly to the turbulent flow.
The developed methodology to characterize a fan calls for unsteady loads on the blades. An approach in potential flow allows a very interesting quantitative analysis concerning the acoustic spectrum concerning the rotational noise.

ACKNOWLEDGEMENTS

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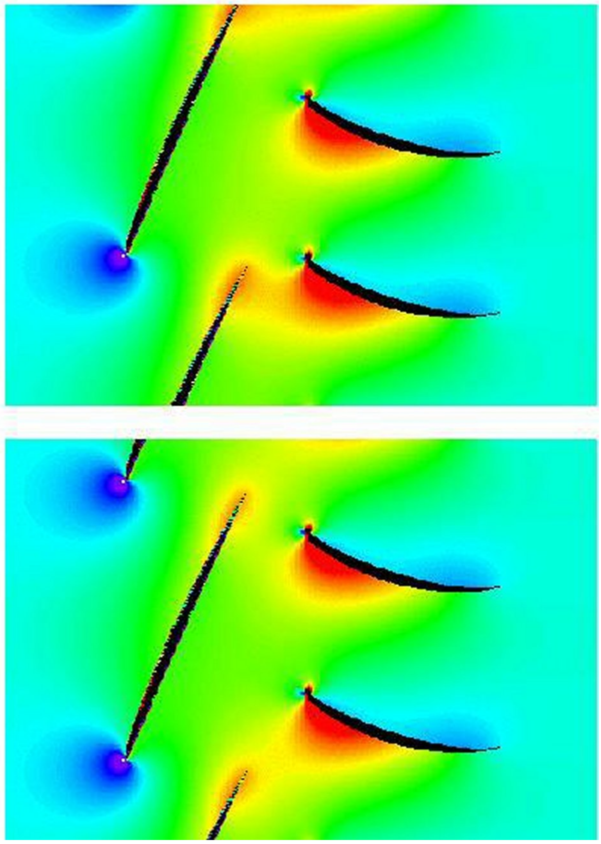

Figure 4: Blade to blade computation - field of absolute velocity with interaction rotor-stator.

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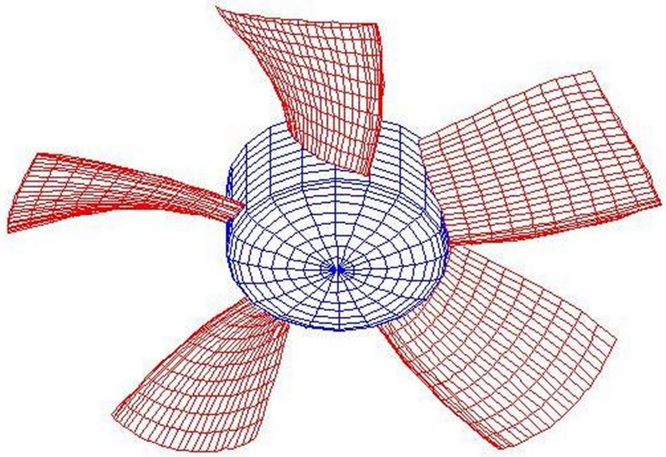

Figure 5: Tested axial fan.

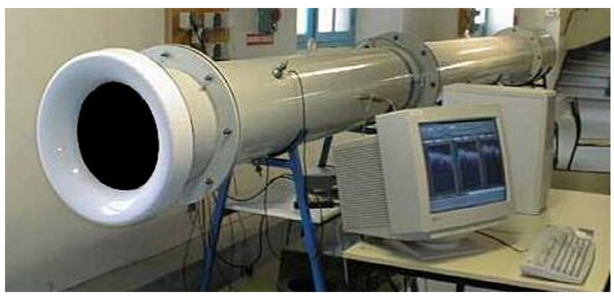

Figure 6: Aeroacoustic test bench of the axial fan.

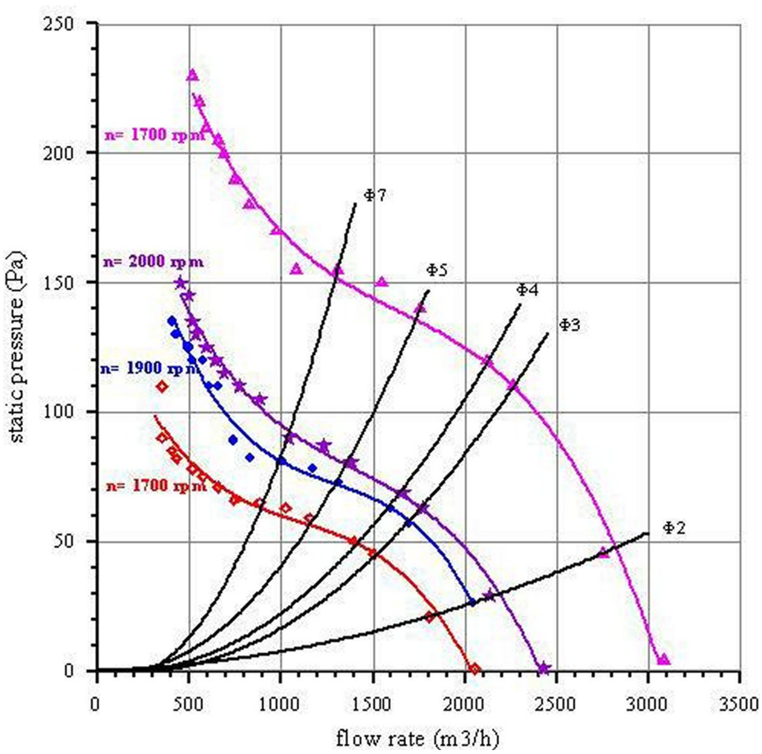

Figure 7: Global aeraulic performances.

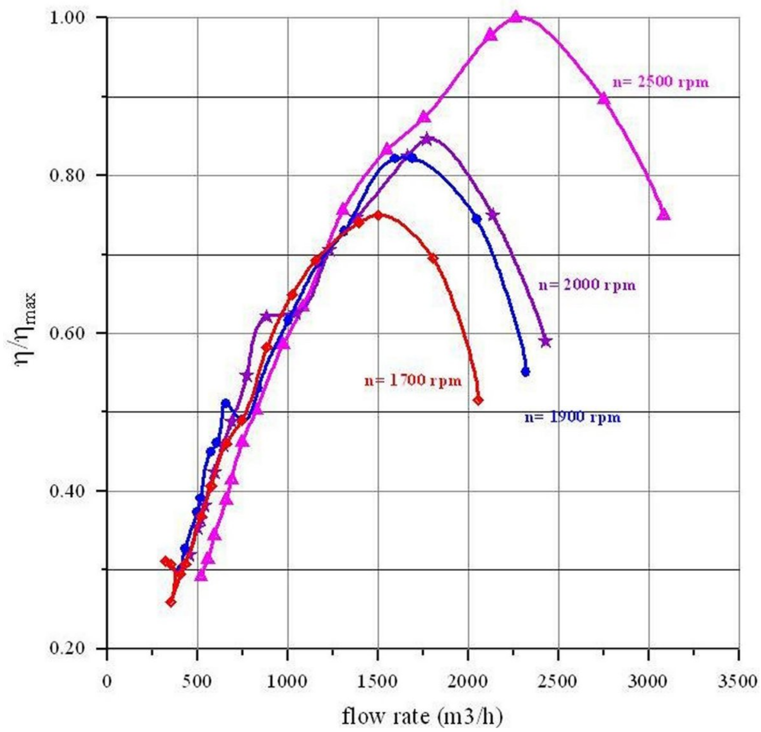

Figure 8: Global efficiency and noise of fan vs the flow rate for various velocities of rotation.

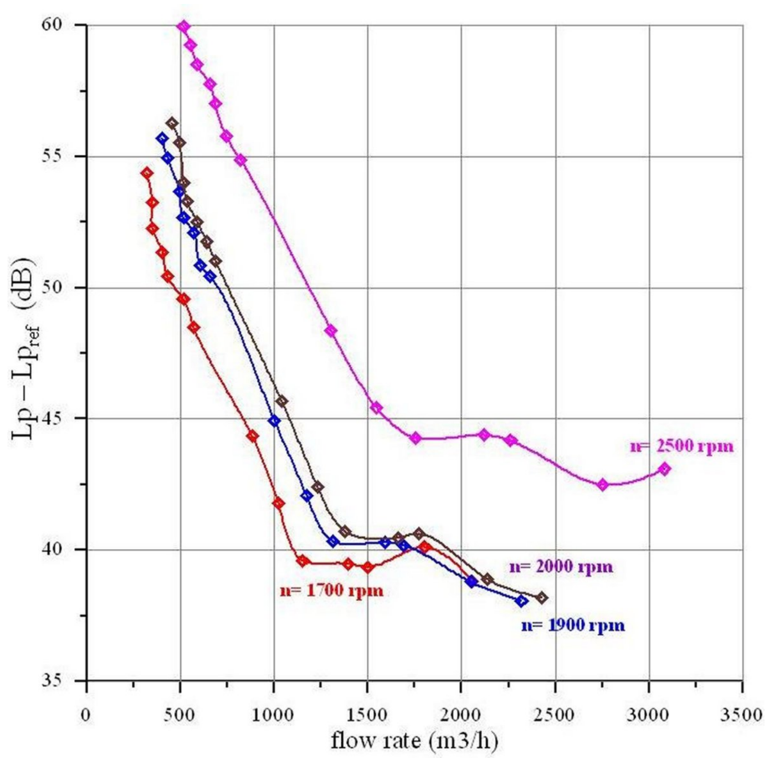

Figure 9: Global noise of fan vs the flow rate for various velocities of rotation.

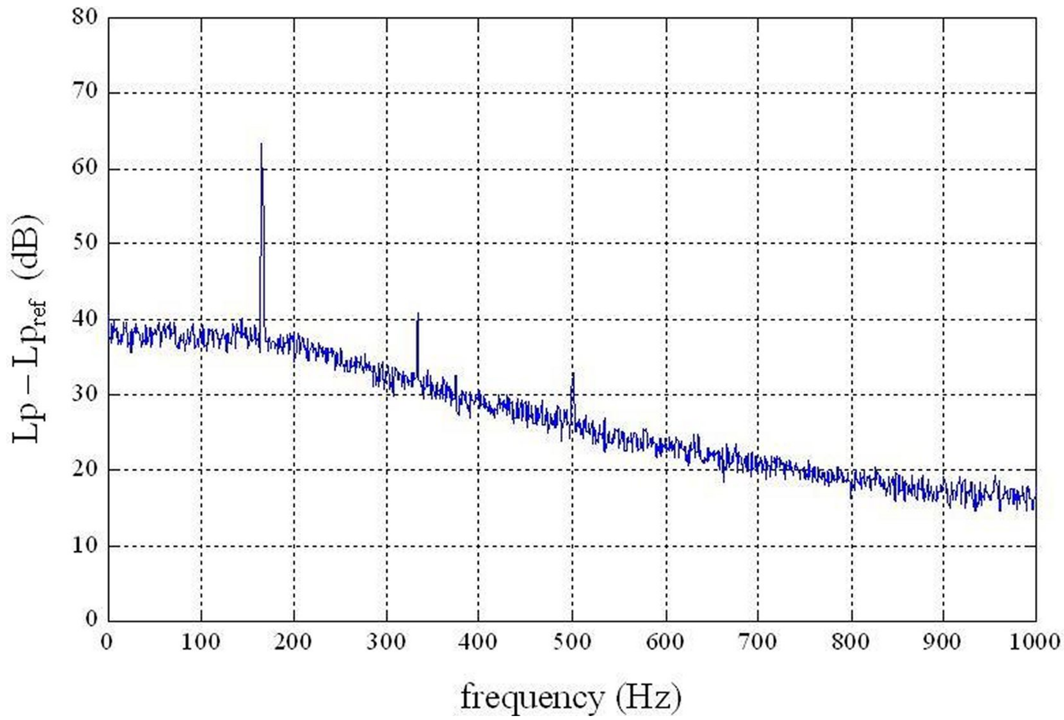

Figure 10: Spectrum of the sound pressure level of the fan at nominal point.

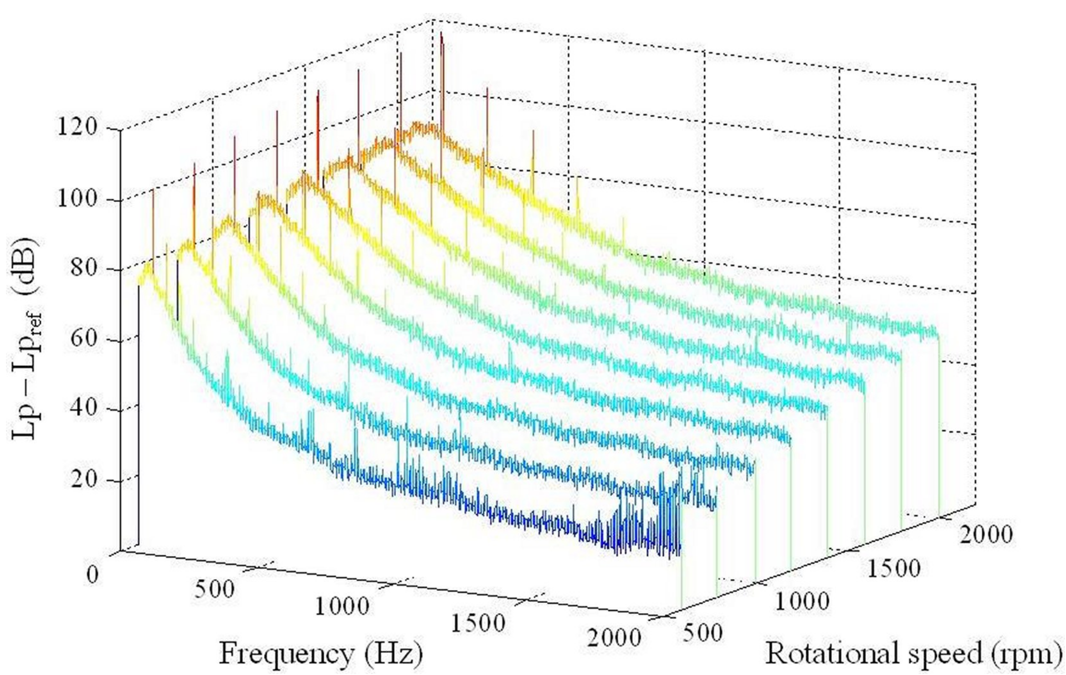

Figure 11: Spectrum of sound pressure level according to the velocity of rotation.

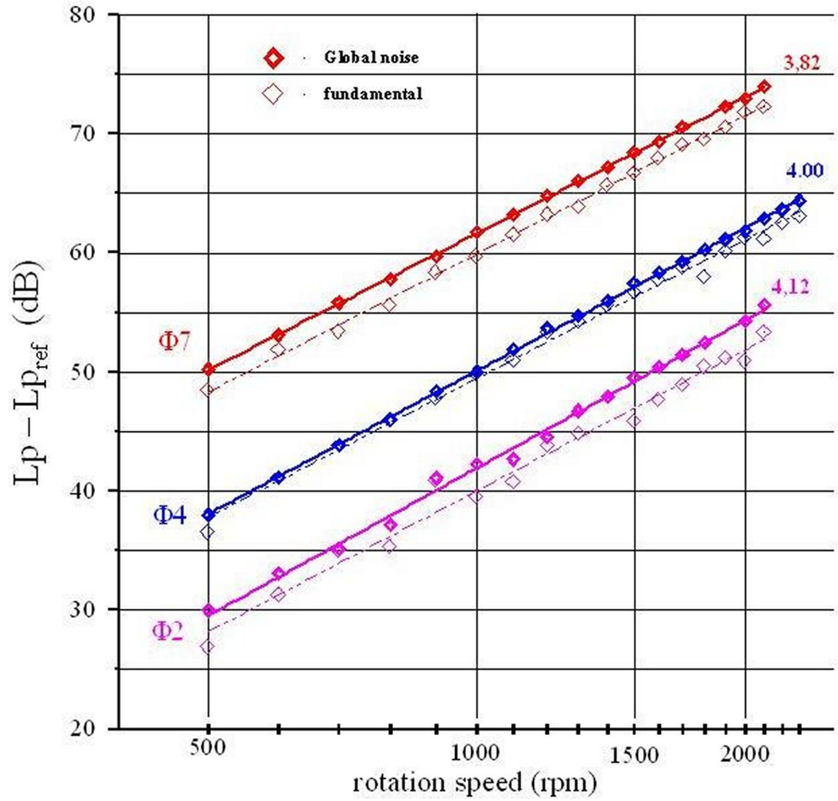

Figure 12: Influence velocity of rotation on the level of global noise.

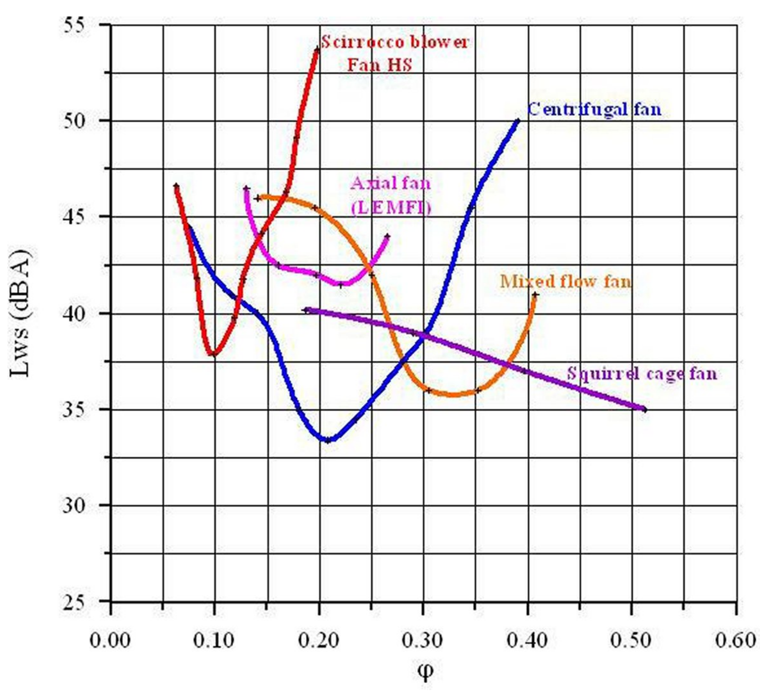

Figure 13: Dimensionless representation of the global acoustic performances.