13th Australasian Fluid Mechanics Conference
Monash University, Melbourne, Australia
13-18 December 1998

NOISE REDUCTION APPROACH OF AN AIR CONDITIONING TURBOMACHINE

Jean-Christophe LEGROS and Mireille LEMASSON
Advanced Design Group Engineer
Liebherr Aerospace Toulouse, Toulouse, FRANCE

Simone PAUZIN
ONERA-CERT, Toulouse, FRANCE

ABSTRACT

The present work is devoted to the noise reduction of an air conditioning system. This kind of system is mainly composed of a turbine, a compressor and a fan. This paper deals with the method we used to compute the noise of the high speed fan. The aim of the study is to develop an aeroacoustic code which will be able to design a fan with the best compromise between acoustic and aerodynamic matter. First of all, acoustic measurements of a serial air conditioning system are presented. As the fan noise has been found to be the main acoustic problem, a test bench was developed. Acoustic measurements are presented and analyzed. Supposedly the entrance of the fan is important for noise radiation. Hence, aerodynamic measurements are achieved. Blade passing frequency noise is assumed to be the main source of noise. Therefore, a Ffowcs Williams and Hawkings formulation is used in order to do a simulation of this acoustic source. Good agreement with measurements are obtained.

INTRODUCTION

Despite the fact that there is no real acoustic standards concerning plane's external noise due to air conditioning system, airlines companies and manufacturers have to respect acoustic limitations. The problem that we have to deal with, occurs during the ground period when the plane is in check between two flights. At this time, there are only the APU and the air conditioning system on. We are in charge of reducing the noise of the air conditioning system and whatever its characteristics are (geometry, speed of rotation, flow,...). We have to reduce the noise annoyance outside the plane at services points.
The general purpose of an air conditioning system is to provide the right pressure and temperature in the plane whatever the working conditions are (hot or cold day, altitude, number of passengers,...). It is mainly constituted of a turbomachine including a fan, a centrifugal compressor and a radial turbine. Also, there is an heat exchanger, a plenum and several ducts and bends. The acoustic study of an air conditioning system can be simplified by the study of the turbomachine. The total system is named a pack of air conditioning system. The function of the compressor and the turbine is to transform the high temperature and pressure, coming from the engine, into temperature and pressure adapted to the cabin conditions. This can be performed thanks to an heat exchanger. The purpose of the fan is to cool the heat exchanger with air coming from outside. Figure 1 presents a schematic of the pack function. The pack is a compact structure, examples dimensions are 1m20 in length, 1m in width and 0m60 in height. In this study, blade numbers are respectively 13 for the fan, 12 for turbine and 14 for compressor. All of them are in the same shaft and have a rotational speed of 55000 RPM.

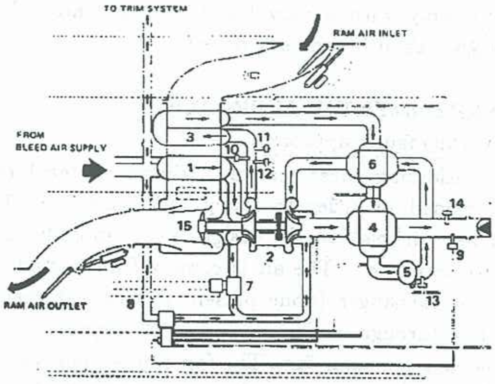

Figure 1: The function of the pack

Acoustic measurements on a pack are first presented. The high speed fan is the principal source of the noise generation. The blade passing frequency of the fan is defined as the main noise source. Aerodynamic and acoustics measurements are carried out on an experimental machine in order to study more specifically the fan noise. Finally, a blade passing frequency noise simulation is presented.

SOUND INTENSITY MEASUREMENTS

Measurements of the global power spectrum of the machine are performed in order to know more about this noise radiation. For this kind of machine (high pressure and temperature in entrance) we decided to use sound intensity measurements because it isn't possible for us to move the machine inside an anechoic chamber. These acoustic measurements are made between 200 Hz and 20 kHz band. First of all, a calibration of a special intensity probe was achieved (3mm spacing microphone), for the 1 kHz-20 kHz band. For the 200 Hz-1 kHz band a classical 50 mm commercial probe was used.

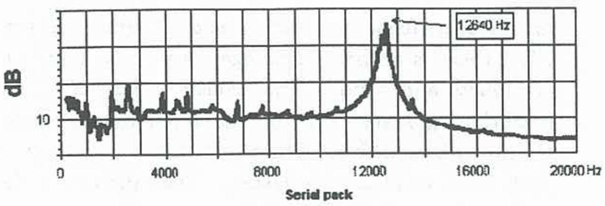

Figure 2: Sound intensity at the pack entrance.

Measurements were made at the entrance and the exit of the fan cooling system because both of them are connected with the outside which is the place of the acoustic problems. The results, presented on Figure 2, show that studying the high speed fan noise is a priority. The blade passing frequency of the fan is higher than broadband noise (around 25 dB higher), at least for this machine. The blade passing frequency of the turbine does not appear on the spectrum. The blade passing frequency of the compressor appears but only with a very low level (only around 2 dB higher than broadband noise).

EXPERIMENTAL APPROACH

Aerodynamic approach

Considering these results, a simplified machine was designed in order to study more precisely the fan noise. In this case the design of the fan is the same as the serial one. The air is coming from outside to an heat exchanger (none present in this test machine), then through a plenum and a 180 degrees bend upstream the axial fan. The fan exit is connected with the outside by a duct. The aim of the study on this simplified machine is to investigate the blade passing frequency and harmonics of the fan and more specifically the non uniform inlet condition. This kind of noise is mainly due to the blade loading. In our case we have a potential, high flow distortion elements (180 degrees bend with a plenum). Therefore we started this part of the study with an exploration of the aerodynamic conditions in front of the fan. An important distortion on the front of the fan increase the unsteady loading blade and so the acoustic level.

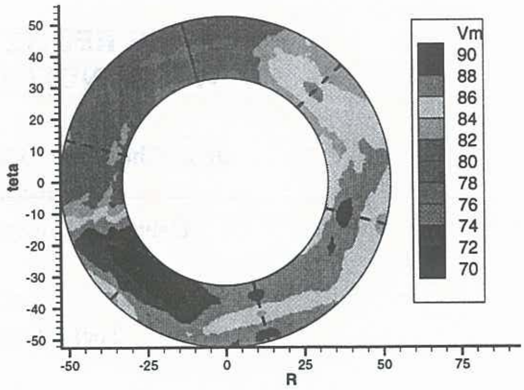

Figure 3: with plenum

Three holes pressure probes are put in six different azimuths and move along the respective radius.

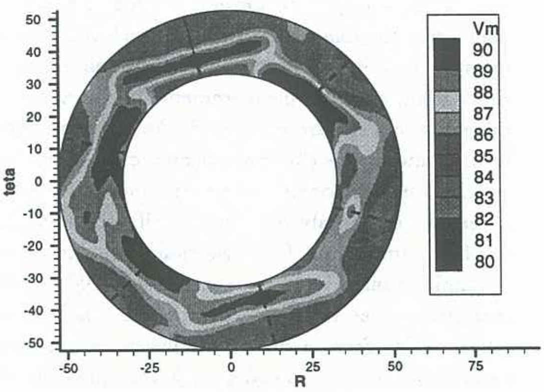

Figure 4: without plenum

We assumed that the flow is mono-dimensional and isentropic. The gas is ideal. Then the Mach number can be evaluated with the following relation :

$$\frac{P_t}{P_s} = \left(1 + \frac{\gamma_0 - 1}{2} M^2\right)^{\frac{\gamma_0}{\gamma_0 - 1}} \qquad (1)$$

To apply this relation it is necessary to measure the dynamic and static pressures at the same point. We can then deduce the absolute speed formulation :

$$M = \frac{V}{c_0} \quad \text{with } c_0 = \sqrt{\gamma_0 r_0 T_s} \qquad (2)$$

The dynamic pressure, static pressure, dynamic temperature and the relative angle of the flow are measured in order to calculate the relative speed. Measurements with and without plenum are carried out to evaluate its influence. The results are shown on Figure 3 and Figure 4, assumed that the plenum is on the top of the circle (on figure 3). The resulting radial distortion is quite the same in both configurations (with and without plenum) but the azimuthal distortion is really different. The influence of the plenum is clearly demonstrated. Because of the unsteady blade loading generated by this distortion, the blade passing frequency level should be higher with a plenum than without.

Acoustic measurements

The influence of the plenum on noise radiation is now studied. We can assume that, with this kind of distortion the noise must be higher with than without plenum. Sound intensity measurements were performed at the exit of the machine. Results of the measurement without plenum are presented on Figure 5.

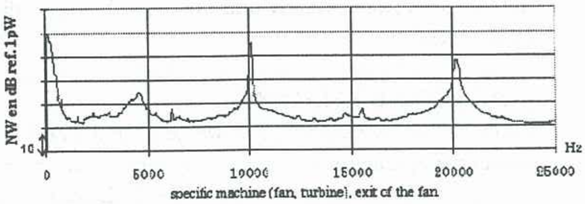

Figure 5: sound intensity measurements

Some points on the global power spectrum of this machine are out lighted:

- Rotational frequency and harmonics noise are observed. The origin of this noise can be aerodynamic (angular variation of the wedging, geometrical difference between blades,...) or mechanical (ball bearings, unbalanced, ...). A formulation is proposed by M.J.Benzakein for the simulation of this kind of noise generated in a duct. In our configuration, this is not the most part of the noise.

- Blade passing frequency noise can have several possible origins. The noise comes from the interaction between blades and flow. The flow can be uniform steady or non uniform steady. A formulation is proposed by Ffowcs Williams J.E., Hawkings D.L.. We decided to use and adapt this formulation to our case.

- Important level round 4 kHz frequency is observed. Studying the work of Kameier and Neise W. we are thinking that this probably comes from the rotating instability. We can, like Kameier and Neise W. did, observe that this appears at half time and one and half time the blade passing frequency. There is no formulation proposed.

- broadband noise has lower important level that blade passing frequency noise but it is anyway important to reduce it. This study is in progress at this time.

Unfortunately, the acoustic measurements with plenum are not available. Then, the only way for us to know the importance of the plenum is to do an acoustic numerical simulation. After all of these conclusions the blade passing frequency simulation study is presented.

BLADE PASSING FREQUENCY SIMULATION In order to optimize the fan on the aeroacoustic point of view, the discrete frequency noise at the blade passing frequency is simulated by a Ffowcs Williams and Hawkings formulation. The input data of this code can be the steady or unsteady blade loading obtained with an aerodynamic code.

Aerodynamic codes used A 3D steady Navier Stokes code and a 2D unsteady Euler code are used. The first one is used to validate the acoustic code and 2D Euler code in steady conditions. We can also evaluate the importance of the steady part of the blade loading on the noise. The second one is used to evaluate the unsteady loading blade effect with the aerodynamic distortion measurements performed in entrance of the fan.

Acoustic code The primary work of Sir M.J. Lightill performed in the fifties to tackle the problem of jet noise, is the starting point of the aerodynamic noise. This work was next extended by J.E. Ffowcs Williams and D.L. Hawkings at the end of the sixties to be applied to rotating machines too. The basic idea is to define an aeroacoustic analogy, by which the real problem of the noise radiated by a rotor and/or a highly disturbed flow is replaced by the problem of the classical acoustic radiation in a medium at rest with equivalent acoustic sources.

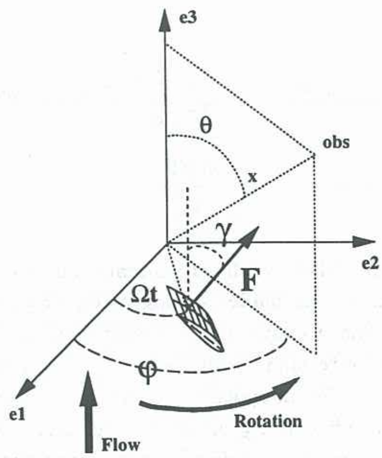

Figure 6: Frame of reference.

We assume that:

- the surface of the blade is solid and rigid
- viscosity is negligible compared to the inertia
- no entropy inhomogeneity
- Mach fluctuations are small

After solving the fundamental equation for the case of a rigid solid, the Ffowcs Williams and Hawkings equation can be written :

$$c_0^2\rho(\vec{x},t) = Quad + Dip + Mono \quad (3)$$

With :

$$Quad = \frac{1}{4\pi}\frac{\partial^2}{\partial x_i \partial x_j}\int_{V_\infty}\left[\frac{T_{ij}}{r|1-M_r|}\right]d\vec{\eta} \quad (4)$$

$$Dip = \frac{1}{4\pi}\frac{\partial}{\partial x_i}\int_S\left[\frac{P_i}{r|1-M_r|}\right]dS(\vec{\eta}) \quad (5)$$

$$Mono = \frac{1}{4\pi}\frac{\partial}{\partial t}\int_S\left[\frac{\rho_0 V_n}{r|1-M_r|}\right]dS(\vec{\eta}) \quad (6)$$

The number 2 formulation has the advantage to represent sources as three differents directivities (monopolar, dipolar and quadripolar) in comparison with a Kirchoff (Lyrintzis, A.S.) which doesn't. This allowed us to give different importance to the source. We focus on the dipole sources.
After a few developments we obtain :

$$P_{nB} = i\frac{Bk_{nB}}{4\pi x}e^{ik_{nB}x}\sum_{p=-\infty}^{\infty}\sum_{r=r_0}^{r_M}J_{nB-p}(k_{nB}r sin\theta) \sum_{\beta=\beta_1}^{\beta_2}\left(cos\gamma cos\theta + \frac{nB-p}{rk_{nB}}sin\gamma\right) \times F_p(r,\beta)e^{i(nB-p)(\varphi-\varphi_0-\frac{\pi}{2})} \quad (7)$$

With :

$$\begin{cases} F_p(r,\beta) & = & \frac{\Omega}{2\pi}\int_0^{\frac{2\pi}{\Omega}} F(r,\beta,t)e^{ip\Omega t}dt \\ k_{nB} & = & \frac{nB\Omega}{c_0} \end{cases} \quad (8)$$

Periodic blade loadings generate a discrete-frequency sound at the blade passing frequency and harmonics. The acoustic field of each tone or harmonic is the infinite sum of characteristic free-field radiation modes. The magnitude of one mode is proportional to a coefficient F_p of the Fourier series $F(t)$ with a weighting factor defined by a Bessel function. The Bessel functions mean that, on the far-field point of view, rotation is responsible for a frequency modulation of the source, which is a consequence of the Doppler effect for circular motion. The complex coefficients F_p are the called blade loading harmonics.

Acoustic calculation We used an aerodynamic calculation (2D Euler unsteady) on the fan used during the measurement. Ten blade to blade sections were needed for the acoustic calculation because of the non acoustic compactness of the blade. The computational results (pressure on the blade) were used as input data to the acoustic code and first harmonics were calculated. These results show us that the steady blade loading is not negligible at this rotating speed. The unsteady blade loading is more important than steady blade loading and the calculation gives a good agreements with the measurements completed (less than 2 dB difference). We also made blade to blade (at constant radius) acoustic calculation in order to know the relative importance of the unsteady blade loading harmonics. The results shows, as we expected, that the higher is the radius, the lower is the difference between steady and unsteady blade loading harmonics.

CONCLUDING REMARKS

After several measurements we find that the blade passing frequency noise of the fan is the main contribution to the noise generation. The influence of the plenum is clearly demonstrated for the lower part of the blade (low radius). An acoustic simulation of the blade passing frequency noise is done. The results are in good agreement with measurements. The aim of this code is now to do a parametric study on the fan in order to design the optimal fan between aerodynamic and acoustic conditions. This fan will be tested to evaluate the validity of the prediction. The aim of this process has to be useful for any other kind of fan.

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