ACOUSTIC CHARACTERIZATION OF A CENTRIFUGAL FAN IN AN AUTOMOTIVE VENTILATION UNIT SOUND PREDICTING MODEL

Ali AL MEZZAWI[1, 2], Yvon GOTH[1], Xavier CARNIEL[1],
Olivier CHERIAUX[3]

[1] *CETIM, Noise and Vibration Engineering Group, 52 avenue Félix-Louat – BP 80064, 60304 Senlis Cedex, France*

[2] *Roberval Laboratory UMR 6253, Industrial Acoustic and Vibration Division, Centre de Recherches Royallieu – BP 20529, 60205 Compiègne Cedex, France*

[3] *Valeo Thermal Systems, Climate Control Product Group, 8 rue Louis Lormand ZA de l'Agiot, Le Mesnil Saint Denis Cedex, 78321 La Verrière, France*

SUMMARY

The present paper describes the acoustic model used to characterize the centrifugal fan of an automotive air conditioning ventilation unit. A sound prediction approach based on virtual prototyping and noise synthesis is carried out to estimate the sound power of a prototype of an automotive ventilation unit. The modelled components, considered as main sound generation sources, are: the forward curved blade centrifugal fan, the heat exchanger and the butterfly flap. Several figures illustrate the contribution to the overall sound power of these sources. Sound power measurements performed on the prototype showed that when the fan is the dominant source, the predicted and the measured values are very similar.

INTRODUCTION

The progress in engine noise reduction made in recent years, along with the development of hybrid and electric vehicles make the sound control of automotive air conditioning units an essential issue in automobile manufacturing. The demand for more compact and aero-thermally performing systems is growing, considering the contribution of the climate comfort on interior vehicle comfort which represents an essential passenger demand on the car selling market.

Several investigations related to this subject have been published. De Guillebon *& al* [1] present a hybrid approach to model the noise of automotive ventilation units based on a measured fan source model and aero-acoustic singularities inducing noise models built from computational fluid dynamics (CFD) data. They report satisfactory results with major discrepancies on peaks around the resonant acoustic frequencies. Humbad [2] gives empirical formulas for single components, subsystems and whole systems, relating the generated noise to pressure drop and/or air flow rate. This method aims to predict the overall sound pressure level only. The frequency spectrum of the sound pressure level cannot be obtained. Ayar *& al* [3] investigate a computational fluid dynamics (CFD) and computational aero-acoustics (CAA) approach to predict the sound generated by automotive air conditioning systems. Sound pressure level spectra at the microphone positions were obtained using a boundary element method (BEM). Although good correlations were obtained between numerical results and experimental ones, further analyses are needed to investigate more closely the chosen boundary conditions.

In this context the REVA-CESAM-SCA project was launched to conduct measurements and develop numerical models that allow the automotive air conditioning system manufacturer to understand and master the mechanisms of sound generation, absorption and distribution along its circuit.

This paper presents a sound prediction model for air conditioning ventilation unit. The first part presents the sound prediction model. Then it outlines its limits and introduces the modeled prototype of a ventilation unit and its aero-acoustic scheme. The second part describes the characterization of each component. The third part shows the results obtained from calculations, and compared to measured data. The last part evaluates the most important aspects of the model and gives the model enhancing perspectives.

MODEL DESCRIPTION

The acoustic synthesis method used to predicted the sound power level produced by a ventilation unit consists in sub-structuring the ventilation unit into its physical components which are characterized as sound generating sources or/and sound transmitting paths. Their aero-acoustic behaviour is modeled using specific measurements and empirical expressions. A sound synthesis software provides a graphical user interface that facilitate the insertion of the components models and performs the desired computations leading to the overall sound level. The figure below is a representative scheme of this method.

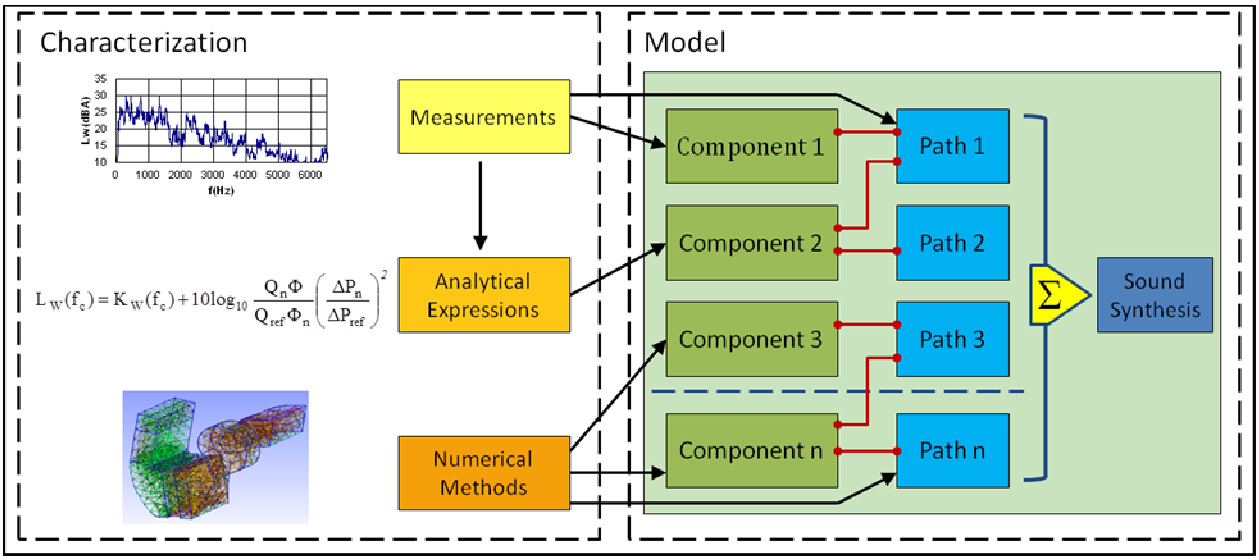

Figure 1: Diagram of the sound synthesis method

The studied system is a prototype of a ventilation unit represented in figure 2a. Compared to an actual ventilation unit which is a very compact system composed of a complex network of air guiding paths the prototype has only one path. Therefore it is much more simple and easier to model. Nevertheless the modeled components and the sound synthesis approach are the same. Figure 2b presents the virtual pattern of the prototype of the ventilation unit created under the sound synthesis software. Each box corresponding to a component is characterized as a source and/or transmission path model. The solid line connecting the boxes represents the propagation path of the sound.

The function to be calculated for each source model is the sound power level and for each path model is the transmission loss, both expressed in one third octave bands. The overall sound power level of the ventilation unit model is obtained by summing the individual contribution of each source and transmission path. We note that the present model predicts only the aero-acoustic sound

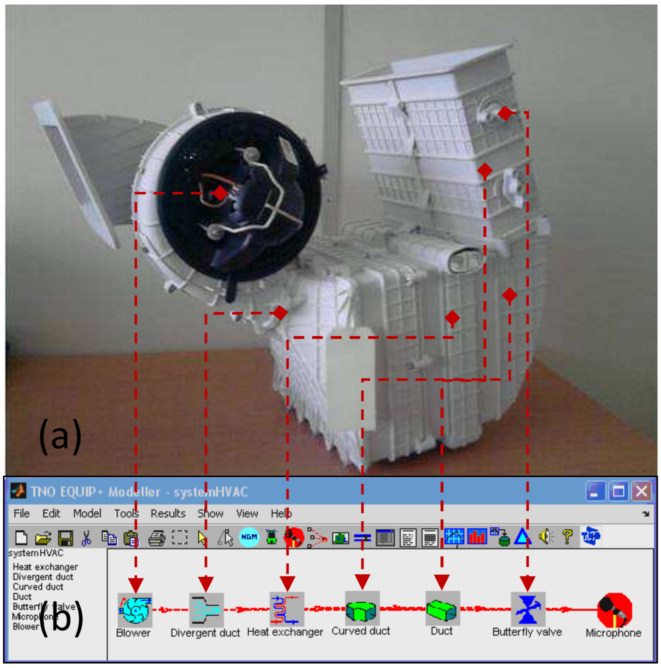

Figure 2: (a) Prototype of the ventilation unit, (b) virtual pattern of the prototype

i.e. resulting from the interaction of air flow with obstacles. As it is based on empirical models certain points are not fully investigated, thus:

- Air flow speed is considered as uniform in a section. It is calculated for every component using the mass flow rate and cross section values at its inlet.
- Sound sources are considered to be uncorrelated, which means that no relation of level or phase exists between the acoustic pressures emitted by these sources.
- Only the sound transmitted by the component is calculated. The reflected sound is not taken into account.
- Openings are anechoic; hence wave reflection at the inlet or at the outlet is not taken into account.
- The effects of integrating a component in the assembly are not treated i.e. the acoustic interaction between elements is not modeled. The sound power of each component is calculated as if it is taken apart

COMPONENTS CHARACTERIZATION

Centrifugal fan characterization

Lots of efforts have been and still are being made by acousticians to understand and master the aerodynamic sound generation mechanisms in industrial fans and turbomachines, and as a consequence predict their acoustic behaviour. Neise [4] proposes a diagram classification of these mechanisms that distinguishes three types of source radiation: monopole, dipole and quadrupole. Many authors [4][5] report that monopole and quadrupole radiations are neglected at low Mach number, which is the case for most fan working conditions. The primary causes of sound generation in fans are the aerodynamic forces induced by the flow on the blades and on the stator.

Various sound prediction correlations for fans have been proposed and used in the industrial domain. Bouquet [6] presents a non exhaustive list of sound prediction correlations commonly found in the literature. He sorted it in three categories, depending on the complexity of the input parameters: The first category correlations are simple algebraic formulas that make use of the aeraulic or/and the geometrical parameters of the fan to calculate the global sound level. This is the case of ACMA [5] and the ASHRAE [7] formulas. The second category correlations calculate the acoustic pressure emitted by a sound generation mechanism throw an aero-acoustic analysis of the mechanism in question [8][9]. The third category correlations give detailed information of the acoustic pressure field created by the fan but require a full knowledge of the flow morphology through it which is possible by solving the Navier-Stokes equations [10]. Therefore they involve the use of numerical simulation methods.

The ASHRAE formula reported by Graham [11] is chosen to predict the sound power level of the centrifugal fan of the ventilation unit. This empirical formula serving initially to model the acoustic behavior at nominal working conditions is modified (see appendix 1) in order to give the sound power level at the actual working point as follows:

$$L_W(f_c) = K_W(f_c) + 10\log_{10}\frac{Q_n}{Q_{ref}} + 20\log_{10}\frac{\Delta P_n}{\Delta P_{ref}} + 10\log_{10}\frac{\Phi}{\Phi_n} \quad (1)$$

Where:

- K_W is the specific sound power level, given for octave bands in the table below.

Table 1: Values of the specific sound power level

f_c (Hz)	63	125	250	500	1k	2k	4k	8k	FPP
K_W (dB)	98	98	88	81	81	76	71	66	2

- FPP is the value to be added to the sound power level at the blade passing frequency.

- Q_n & ΔP_n are respectively the volumetric flow rate and the total pressure rise of the centrifugal fan at nominal working point.

- Q & ΔP are respectively the volumetric flow rate and the total pressure rise of the centrifugal fan at actual working point.

- Q_{ref}= 1 m^3s^{-1}

- ΔP_{ref}=1 kPa

- Φ is the aeraulic reduced opening given by the formula below:

$$\Phi = \frac{Q}{\pi DL}\sqrt{\frac{\rho_0}{2\Delta P}} \quad (2)$$

- D is the fan's wheel diameter (m).
- L is the fan's blade width (m).
- ρ_0 is the air density at working conditions (kg/ m^3).
- f_c is the centre frequency of the octave band (Hz).

At a given rotation speed, the nominal flow rate and the pressure rise of the fan are respectively determined from the graphs of figure 3a and figures 3b. They illustrate the results of measurements performed on the centrifugal fan of the prototype. Each rhombus represents a working point. The dashed lines are the regression curves that fit best the plotted data.

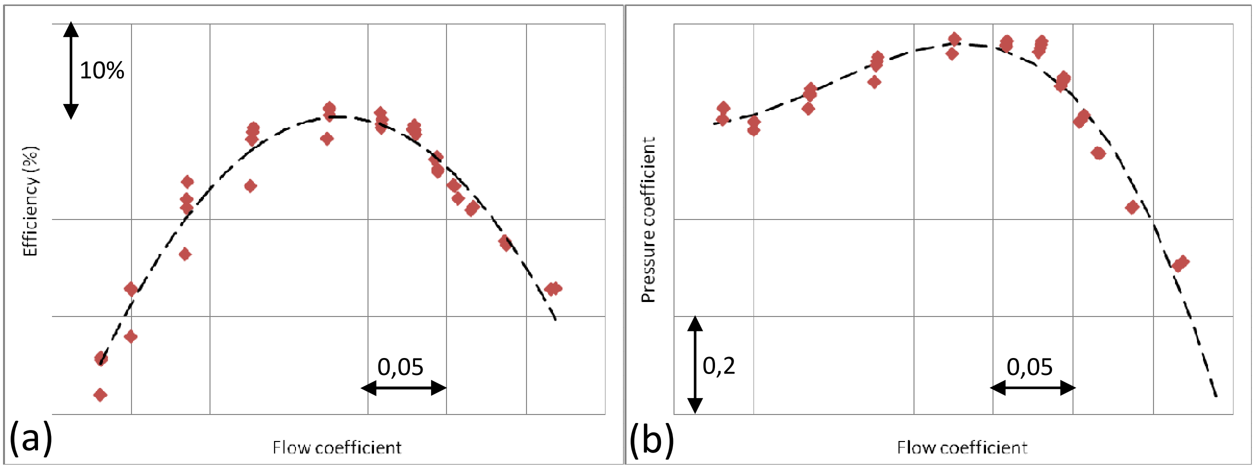

Figure 3: Global efficiency (a) and pressure coefficient (b) plotted as functions of the flow coefficient (μ)

Other components characterization

Just like the centrifugal fan, the heat exchanger and the butterfly flap are characterized as sound generating sources. They are also characterized as transmitting paths along with the divergent and rectangular ducts. Concerning the heat exchanger, its sound power level is calculated using an empirical equation similar to the one given by Oldham and Waddington [12]. It is written:

$$L_W(f_c) = \Gamma_W(f_c) + 10\log_{10}A + 80\log_{10}U \tag{3}$$

Where:

- Γ_W is the specific sound power level generated by a heat exchanger having a cross section area of 1 m^2, and placed in an air flow with mean velocity equal to1 m/s.
- A is the cross section area of the heat exchanger found in the ventilation unit.
- U is the mean velocity of the air flow just upstream of the heat exchanger.

The transmission loss of the heat exchanger is also calculated using a similarity equation. This equation which transposes a measured transmission loss to the prototype working conditions is expressed as follows:

$$TL(f_c) = TL_0(f_c) + \alpha\log_{10}U \tag{4}$$

Where:

- TL_0 is the transmission loss at an air flow velocity equal to zero.
- α is an empirical constant value which was found to be approximately equal to 1,7 for heat exchangers having the same internal geometry of the one found in the prototype of the ventilation unit.

The reference sound power spectrum and the transmission loss used in the similarity equations were obtained from a series of measurements conducted on a heat exchanger which is homothetic to the one found in the prototype. A more detailed description of the experimental setup and the theoretical basis behind these measurements is given in [16] and [17].

As for the butterfly flap, a correlation established by Nelson and Morfey [13] to predict the sound power level of flow spoilers is chosen. This correlation is described by the equation (5) and (6) given below.

$$f_c < f_0 \qquad L_W(f_c) = K^2(St) + 10\log_{10}\frac{A\Delta P^2}{4\rho_0 c_0} \tag{5}$$

$$f_c > f_0 \qquad L_W(f_c) = K^2(St) + 10\log_{10}\frac{\pi A[8Af_c + 3c_0(h+b)]f_c \Delta P^2}{48\rho_0 c_0^3} \tag{6}$$

Where:

- $K^2(St)$ is a Strouhal number depending factor obtained from the collapse of butterfly data measured by Oldham and Ukpoho [14].
- ΔP is pressure drop across the butterfly valve.
- b is the width of the duct enclosing the butterfly valve.
- h is the height of the duct enclosing the butterfly valve.
- $A=b\times h$
- c_0 is the velocity of sound in air.

The transmission loss across the butterfly valve is assimilated to be equal to that of a transition between two cross sections of different areas i.e. a sudden duct contraction followed by a sudden duct expansion. It is calculated using the formula introduced by Munjal [15] and rearranged in the following form:

$$TL = 10\text{Log}\left(\left|\frac{T_{11} + (AT_{12}/c_0) + (c_0 T_{21}/A) + T_{22}}{2}\right|^2\right) \tag{7}$$

T_{ij} are the elements of the transfer matrix for plan wave propagation of the "sudden contraction – sudden expansion" duct represented below.

$$\begin{bmatrix} 1 & \dfrac{Mc_0(3-5\sigma+2\sigma^2)}{2A\sigma^2} \\ 0 & 1 \end{bmatrix} \tag{7}$$

Where:

σ is the open area ration.

M is the Mach number.

Finally, no source or transmission path models are affected to the curved duct.

RESULT ANALYSIS

The overall sound power level of the prototype of a ventilation unit is measured (see appendix 2) for many working points. Each one is identified by the pressure rise and the flow rate of the fan which corresponds to a fan wheel rotation speed and butterfly flap open area ratio. The results are plotted in figures 4 to 8 along with the overall sound power level predicted by the ventilation unit model and the sound power level of the three sound generating components. As mentioned earlier the sound power of a component is calculated as if it is taken apart, whereas overall sound power of the ventilation unit model is the sum of the individual sound powers and transmission losses. This implies that the components sound power level curves could be higher than the overall predicted or/and measured ones.

Figure 4 shows that, for a relatively opened flap the curves of the measured and the predicted values are almost overlapping, especially for medium and high flow rates. A small difference is detected at low flow rates. The fan is the main contributing source in these conditions. The heat exchanger's contribution to the overall sound increases with the flow rate and reaches the fan contribution at high flow rates. The same observation is made for the flap, but its contribution remains insignificant compared to the fan.

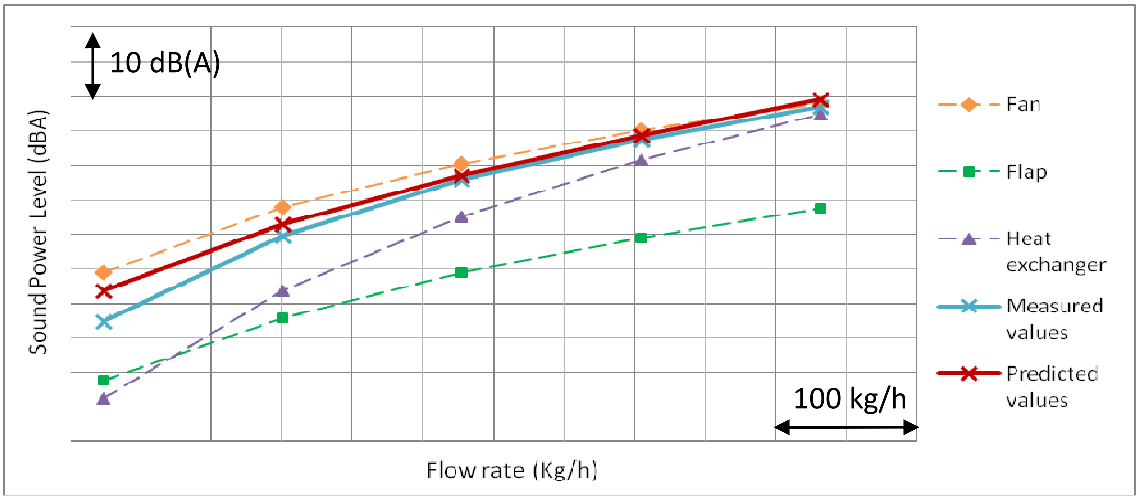

Figure 4: Sound power level plotted against the flow rate (open area ratio=60%)

In figure 5 it is noticed that flap contribution to the overall one increases considerably, while the heat exchanger contribution decreases a little. However the fan remains the main sound generating source with a sound power level slightly higher than the one of the flap. The difference between the measured values and the predicted ones increases for all flow rates.

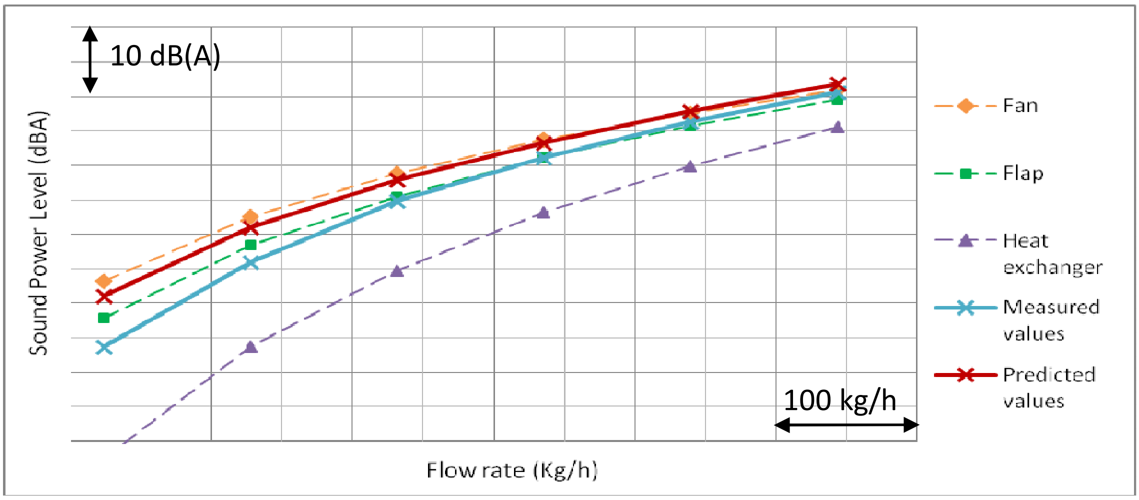

Figure 5: Sound power level plotted against the flow rate (open area ratio=35%)

In figure 6 the nearly closed butterfly flap becomes the main sound generating source. Its sound power level exceeds the one of the fan whose contribution to the overall sound remains high, contrary to the heat exchanger contribution that becomes negligible. The error between the measured and the predicted values increases once again, meaning that the model overestimates the sound power level of the ventilation unit.

The three figures illustrate the evolution of the components sound source models toward the working point of the system. The centrifugal fan sound power, function of the pressure rise and the flow rate, remains significant for all working points, whilst the flap sound power increases with the increase of pressure drop and the heat exchanger one increases with the increase of the flow rate. It is important to note that the prediction model is more accurate for medium and high flow rates which correspond to an open flap.

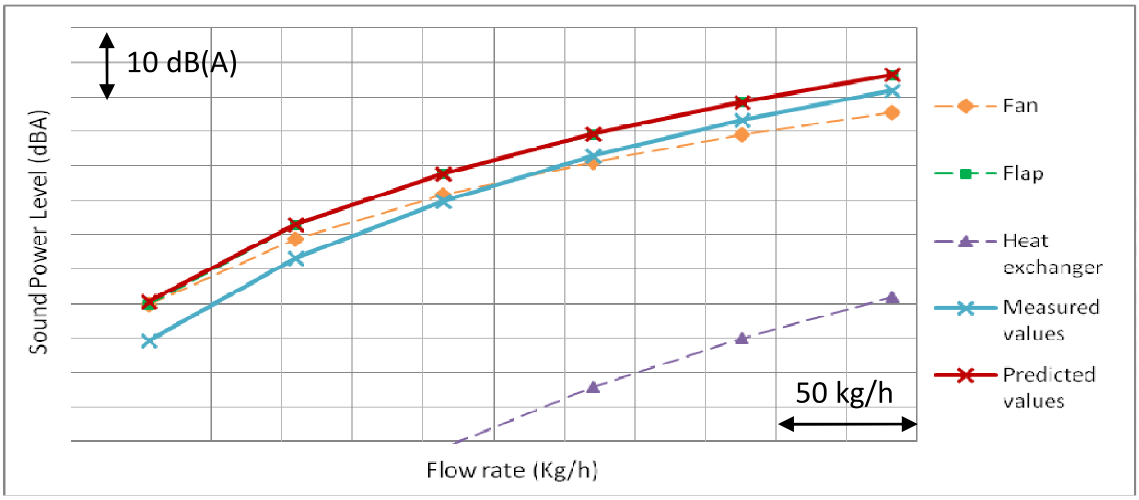

Figure 6: Sound power level plotted against the flow rate (open area ratio=15%)

A more detailed investigation of the data is possible with figures 7 and 8 which present the sound power spectra respectively in the cases of a widely open flap and a fairly closed one.

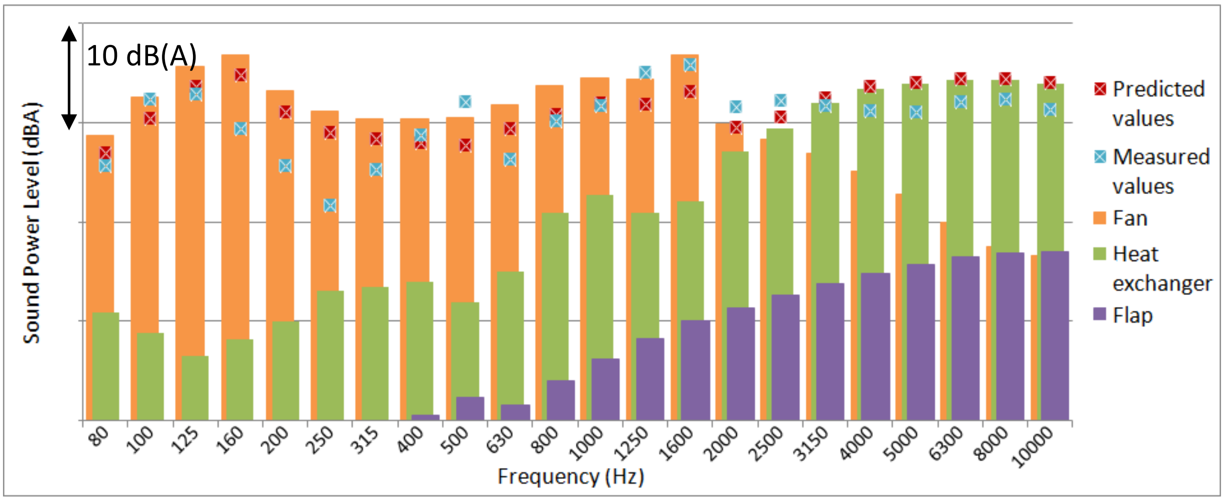

Figure 7: Sound power spectrum plotted in one third octave bands (rotation speed=2500rpm, open area ratio=99%)

In figure 7, the predicted spectrum follows well the outline of the measured one. Up to the 2000 Hz one third octave band, the fan is by far the dominant source. From the 2500 Hz one third octave band the heat exchanger becomes the dominant source. The flap has a very low sound power level at low frequencies. It increases with increasing frequencies to attend that of the fan at 10000 Hz, but remains a lot smaller than the higher one.

Figure 8 shows that, when the flap is partially closed, the measured and predicted spectra have the same frequency evolutions except for the [160 Hz 400 Hz] interval. Their values are much closer at high frequencies than at lower ones. The fan is the main source in the whole frequency range, despite a flap sound power emergence predicted between 250 Hz and 500 Hz. The heat exchanger sound power is insignificant compared to the other sources, except for very high frequencies.

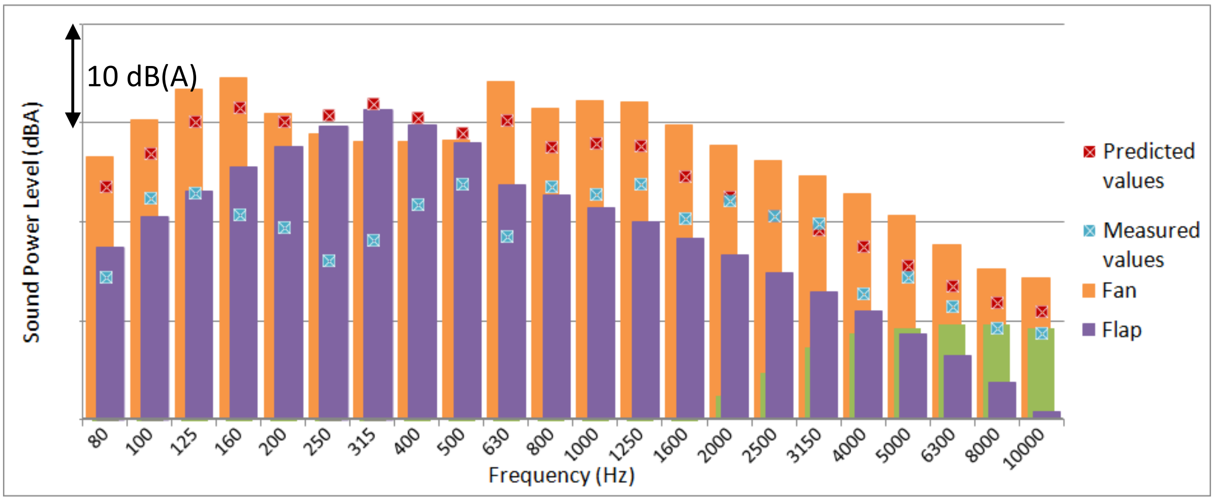

Figure 8: Sound power spectrum plotted in one third octave bands (rotation speed=1000rpm, open area ratio=35%)

CONCLUSION

Sound power level of automotive air conditioning ventilation unit is predicted using an acoustic synthesis method. A prototype of a ventilation unit is studied in this paper. Its components are characterized as sound generating sources or/and sound transmitting paths. The overall sound power of the prototype is measured and compared to the predicted one at different working points. For a sixty percent open aero ratio the measured and predicted values are very close. The error increases as the flap is closed and the model tends to overestimate the actual sound power. Results show that the fan is the major sound source of a ventilation unit despite the strong contribution of the heat exchanger at high flow rates. The flap model shows some inconsistencies, especially when the flap is closed.

Additional measurements are performed on each of these components to enhance their acoustic models.

ACKNOWLEDGMENTS

This work has been performed within the I-Trans project REVA-CESAM-SCA having as collaborators: CETIM, UTC, VALEO and ESI-Group.

BIBLIOGRAPHY

[1] H. De Guillebon, J. M. Auger, F. Dubief, A. Taklanti, F. Journeau, P. Blanc – *Aeroacoustic noise induced by HVAC system: experimental and modelling analysis*. Proceedings of Fan Noise 2003 Symposium, Senlis, **2003**

[2] N. Humbad – *Automotive HVAC flow noise prediction models*. 2001 SAE World Congress, Michigan, **2001**

[3] A. Ayar, R. Ambs, C. Capellmann, B. Schillemeit, M. Matthes – *Prediction of flow-induced noise in automotive HVAC systems using a combined CFD/CA approach*. 2005 SAE World Congress, Michigan, **2005**

[4] W. Neise – *Review of fan noise generation mechanisms and control methods*. Proceedings of Fan Noise 1992 Symposium, Senlis, **1992**

[5] A. Guédel – *Acoustique des ventilateurs "génération du bruit et moyens de réduction"*. 1999 éditions PYC Livres, **1999**

[6] T. Bouquet – *Etude du comportement aéraulique et acoustique des ventilateurs centrifuges à action*. Thesis, ENSAM, **2004**

[7] D. A. Bies, C. H. Hansen – *Engineering noise control "theory and practice"*. Third edition, Spon Press, **2003**

[8] H. Hayashi, Y. Kodama, T. Yamasaki, T. Ohnishi, T. Fukano – *Unsteady flow and noise characteristics of a multi-blade centrifugal fan*. Proceedings of Fan Noise 2007 Symposium, Lyon, **2007**

[9] T. Fukano, Y. Kodama – *Prediction of sound power of low pressure axial- and diagonal-flow fans*. Proceedings of Fan Noise 1992 Symposium, Senlis, **1992**

[10] Y. Cho, Y. J. Moon – *Blade tonal noise prediction of variable pitch cross flow fans by unsteady viscous flow computation*. Proceedings of Fan Noise 2003 Symposium, Senlis, **2003**

[11] J. B. Graham – *The status of fan noise measurement in North America*. Proceedings of Fan Noise 1992 Symposium, Senlis, **1992**

[12] D. J. Oldham, D. C. Waddington – *The prediction of airflow-generated noise in ducts from consideration of similarity*. Journal of Sound and Vibration, 248 (4) 780-787, **2001**

[13] P. A. Nelson, C. L. Morfey – *Aerodynamic sound production in low speed flow ducts*. Journal of Sound and Vibration, 79 (2) 263-289, **1981**

[14] D. J. Oldham, A. U. Ukpoho – *A pressure-based technique for predicting regenerated noise levels in ventilation systems*. Journal of Sound and Vibration, 140 (2) 259-272, **1990**

[15] M. L. Munjal – *Acoustics of mufflers and ducts "with application to exhaust and ventilation system design"*. John Wiley & Sons, **1987**

[16] H. Trabelsi, N. Zerbib, J.-M. Ville, F. Foucart – *Méthode de mesure de la matrice de diffusion multimodale d'obstacles complexes en présence d'écoulement uniforme*. 10ème Congrès Français d'Acoustique, Lyon, **2010**

[17] H. Trabelsi – *Banc d'essai et procédure pour la caractérisation des éléments d'un SCA par un système « 2N-ports » avec écoulement : Validation et application à des sources aéroacoustiques*. Thesis, UTC, **2011**

APPENDIX 1

The ASHRAE empirical formula is fitted to the working point using the following graph given by Guédel [5]. It represents the evolution of the specific global power level versus the aeraulic reduced opening of numerous centrifugal forward curved blade fans transposed at the same rotation speed, wheel diameter and blade width.

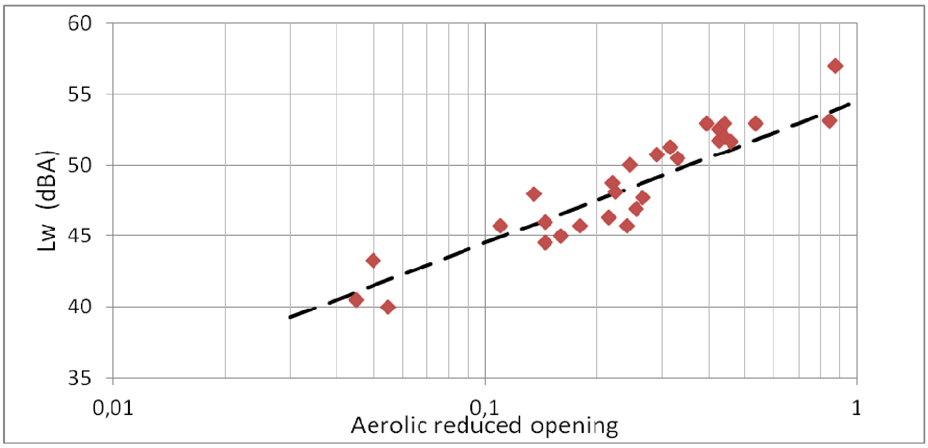

Figure A1.1: Specific power level plotted against the aeraulic reduced opening

These data show that at a given rotational speed the relation between the specific global power level of a centrifugal fan and its reduced aeraulic opening is approximated by a logarithmic regression curve defined by the following equation:

$$L_W = \alpha + 10\log_{10}\Phi \tag{A1.1}$$

At nominal working point the sound power level is written:

$$L_{Wn} = \alpha + 10\log_{10}\Phi_n \tag{A1.2}$$

Subtracting (A1.1) from (A1.2) gives:

$$L_W - L_{Wn} = 10\log_{10}\Phi - 10\log_{10}\Phi_n \tag{A1.3}$$

Now the global A-weighted overall sound power level resulting from the ASHREA formula is expressed as follows:

$$L_{Wn} = K_W + 10\log_{10}\frac{Q_n}{Q_{ref}} + 20\log_{10}\frac{\Delta P_n}{\Delta P_{ref}} \tag{A1.4}$$

Replacing L_{Wn} in (A1.3) by its corresponding expression presented in (A1.4) leads to:

$$L_W = K_W + 10\log_{10}\frac{Q_n}{Q_{ref}} + 20\log_{10}\frac{\Delta P_n}{\Delta P_{ref}} + 10\log_{10}\frac{\Phi}{\Phi_n} \tag{A1.5}$$

The sound power spectrum in octave frequency bands is then:

$$L_W(f_c) = K_W(f_c) + 10\log_{10}\frac{Q_n}{Q_{ref}} + 20\log_{10}\frac{\Delta P_n}{\Delta P_{ref}} + 10\log_{10}\frac{\Phi}{\Phi_n} \tag{A1.6}$$

In order to obtain only the inlet or outlet sound power 3dB must be subtracted from the equation (A1.6)

APPENDIX 2

The acoustic power is measured according to the method described in the international standard ISO 3746. The experimental test bench displayed in figures A2.1a & A2.1b is formed of: A stiff box filled with sound absorbing materials in which the ventilation unit will be placed, a 2m x 2m panel serving as a sound reflecting plane and a 2.5 m long flexible lined circular duct used to guide the incoming air to the inlet of the ventilation unit. The whole is placed in a room with phonics treatment.

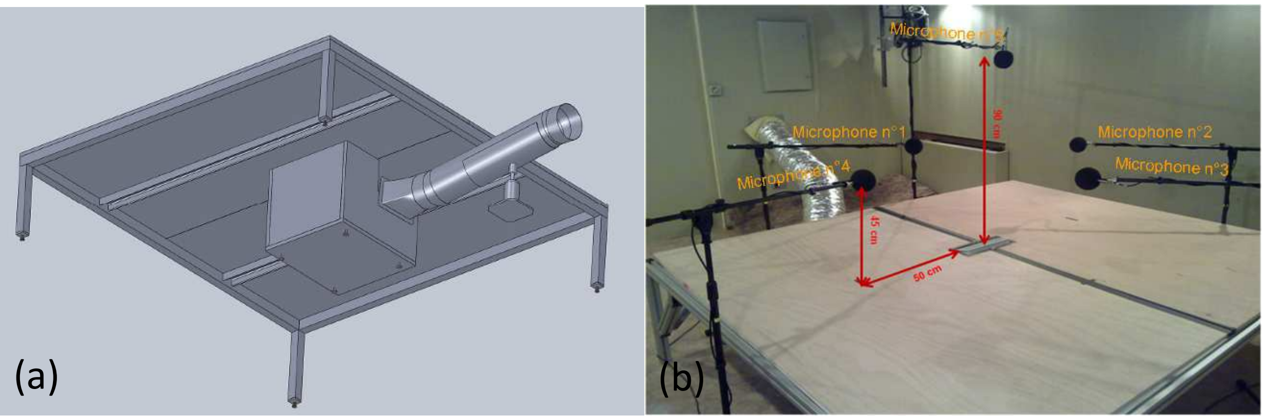

Figure A2.1: Experimental test bench for ventilation unit sound power measurements

The acoustic pressure is measured using five half inch pre-polarized microphones placed at the centers of the faces of a 1.25 m x 1.08 m x 0.9 m imaginary parallelepiped enveloping the outlet of the ventilation unit. The acoustic power level, L_w of the ventilation unit is obtained for every one third octave frequency band, using the formula below:

$$L_W = L_{pm} - K_1 - K_2 + 10 \times \log\left(\frac{S}{S_0}\right) \qquad \text{(A2.1)}$$

Where:

L_{pm} is the spatial mean of the acoustic pressure calculated by the following equation:

$$L_{pm} = 10 \times \log\left(\frac{1}{5}\right)\sum_{i=1}^{5} 10^{0.1 \times L_{pi}} \qquad \text{(A2.2)}$$

L_{pi} is the sound pressure level of the i^{th} position.

K_1 is the background noise correction.

K_2 is the reflected sound correction.

S is the area of the measurement surfaces enveloping the outlet.