PREDICTION OF FAN NOISE
BY AN INVERSE METHOD

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SUMMARY

An inverse method is developed to carry out a noise prediction tool of a fan placed in a complex environment. Based on a theoretical approach from the equations of acoustic propagation, the fan is represented by a source term reflecting its ability to produce a sound. The value of the source term is experimentally quantified using a test bench to determine the fan's intrinsic capacity to produce the sound measured, regardless of the measurement environment. Then, one may calculate the noise produced when the fan is integrated inside a complex system.

INTRODUCTION

Due to multiple complex aero-acoustics phenomena, fan noise can be predicted with very different tools. The easiest way is to use the logarithmic law ASHRAE [1] despite the fact that no link exists between the physical sources and the acoustic calculation and that the prediction is only possible if the fan is integrated into its shroud.

Other computational methods have been developed, based on the Lighthill's analogy for instance [2]. Those methods, coupling a CFD (Computational Fluid Dynamics) result with a calculation of acoustic propagation, are very demanding in terms of computing resource.

In this paper, an inverse method is developed based on experimental data in order to provide a prediction tool having a high calculation speed.

PART ONE: PRINCIPLE OF THE METHOD

Any component placed in a complex acoustic environment has two different acoustic specifications. A source term represents its intrinsic capacity to produce a noise while an attenuation term is required to translate its intrinsic capacitance to lessen the sound (such as the transmission loss).

The prediction method of the noise produced by a fan is presented on the Figure 1. This method is performed in two steps: thanks to an acoustic measure performed on an appropriate bench (1.c), one may get the intrinsic source and attenuation terms (a) after determining the transfer function of the bench (1.b). Then, the terms are propagated (2.b) through any system to predict the noise generated by the component placed in a confined environment (2.c).

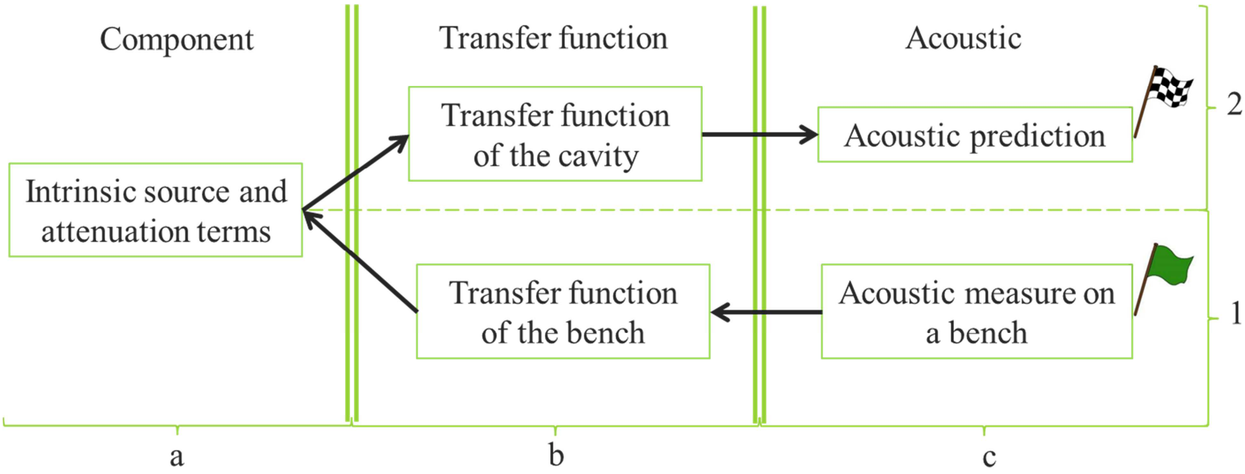

Figure 1: Inverse method principle

The first step (1.c, 1.b, 1.a) is qualified as "inverse": an acoustic result is used to get the terms characterizing the fan. The second step (2.a, 2.b, 2.c) is qualified as "direct": These terms are propagated through any complex acoustic system called "cavity" in order to predict the acoustic situation.

It has been chosen to compute the transfer function with the intention of model any acoustic cavity. Therefore, the intrinsic terms has to be taken into account in the computation. The modified equation of acoustic propagation – commonly called the Helmholtz equation [3] – is presented in the APPENDIX A and used for the calculation of the transfer functions.

In this paper, one considers that the fan is only represented by an intrinsic source term while the attenuation term is now neglected.

Acoustic measure on the bench (1.c)

When the acoustic measure is performed, a link has to exist between the fan's operating points and its acoustic characteristics. The operating point, represented by the rotational speed N, the pressure rise and the flow rate, depicts the fan performance under charged conditions. Following the aero-acoustic standards ISO 5801 [4] and NF S 31-063 [5], an upstream Sound Power Level (SWL) measure is performed at every operating points. The SWL value on the bench is obtained with the equation (1):

$$\mathrm{SWL} = 10\log_{10}\left(\sum_{i=1}^{n} 10^{\frac{SPL_i}{10}}\right) - C + 10\log_{10}\left(\frac{S}{S_0}\right). \qquad (1)$$

with S the area of the measuring conduit, S_0 equal to one meter square and C a correction term which consider various measure's errors.

The acoustic power associated to (1) is noted $\mathrm{W}_{\mathrm{measure}}$.

Fan modeling

The source term is modeled as a force per unit volume, noted f, wrapping the space occupied by the blades. The various noise sources of a fan (trailing edge, leading edge and so on) are taken into account and included to obtain a "global source".

When rotating, the fan has as many incoherent sources as blades. Considering a coherent source is irrelevant. To represent the broadband noise, a randomly selected group of adjacent blades is considered (Figure 2-right). Then, the noise power produced is adjusted to take into account all of the blades.

However, at the blade passage frequency and its harmonics, all sources are coherent. A phase shift must be incorporated in order to translate the blades' delay against each other. The tonal noise is then represented by a global phased force.

$$f = f.\sum_{\ell} e^{\frac{-j\ell F_{pp}}{N}\theta} \qquad (2)$$

with N the rotating speed, θ the phase shift (Figure 2-left) and ℓ the nth harmonic of F_{pp}, the blade passage frequency.

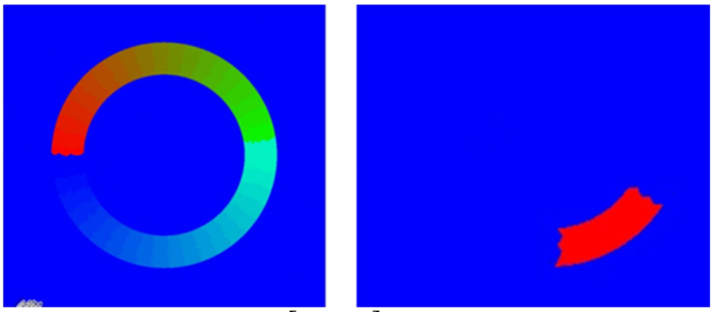

Figure 2: Left, value of the phase shift $[-\pi, \pi]$.*Right, 6 blades out of 41 randomly selected*

Computation of the transfer functions (1.b and 2.b)

The determination of transfer function using finite-elements approaches is done by modeling the fan integrated in the two different acoustic environments. The modified equation of propagation used in the finite-elements computation is the equation (3) of the APPENDIX A. Even if the

- The computation of the bench transfer function (1.b) gives the acoustic power W_{bench_unit}.
- The computation of the cavity transfer function (2.b) gives the acoustic power W_{cavity_unit}.

Determination of the intrinsic source term (a) and acoustic prediction (2.c)

No more calculations are needed once the transfer functions (TF) are characterized. Only the measure made on the bench changes the acoustic prediction. The force per unit volume value is acquired using the relation (3):

$$f = \sqrt{\frac{W_{measure}}{W_{bench\_unit}}}. \qquad (3)$$

The final result is carried out by the equation (4) which is straightforward to compute:

$$W_{prediction} = f^2.W_{cavity\_unit}. \qquad (4)$$

PART TWO: VALIDATION TEST

The component used to validate the method is a tangential fan used in a simplified automotive air cooling system (HVAC) from which all flaps and heat exchangers have been removed as shown in Figure 3. The method unfolds in three acts:

- The first one follows the method presented: the fan's intrinsic acoustic term is extracted once the acoustic measure is done on the bench and the transfer functions calculated (Figure 4). This term is propagated through the acoustic cavity in order to get the outlet's SWL.
- A calibration test is settled in order to have a reference measurement.
- A results analysis is performed to retune the model and identify errors' sources.

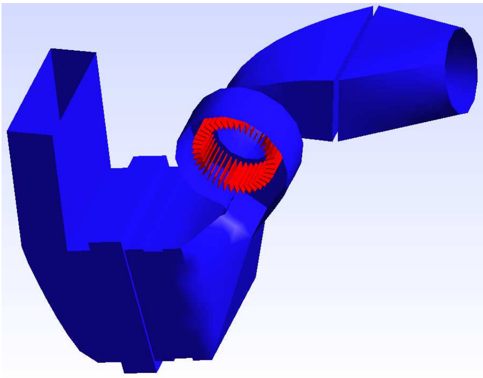

Figure 3: Acoustic cavity and the fan represented by its blades

Application of the method

For this purpose, FreeFem++ [6] is used to perform the finite-elements resolution while the bench and the cavity are modeled by using the Gmsh mesh generator [7]. FreeFem++ allows great flexibility since the mathematical definition of the weak form can be written. The weak form is the equation implemented and solved by any finite elements software.

The figure presented on the APPENDIX B plots the results of the method at the different steps.

The time needed to calculate the two acoustic TF (up 20 to 4000 Hz with 995 frequencies) is about 2 hours using a standard laptop which corresponds to the desired preparation time. Acoustic prediction is then almost instantaneous using relations (3) and (4).

Calibration test

Since all the components responsible of the pressure loss have been removed from the simplified HVAC, it is mandatory to adjust the pressure loss. Two different obstructions have been placed upstream of the fan in order to reproduce the fan's nominal operating conditions. Eight operating points are chosen by crossing the flow rate-pressure loss curves of the created circuits with those of the fan. The sizes of the obstructions have been defined from a pressure loss calculation using the CFD software Code_Saturne [8].

The method is tested through the comparison of the outlet's SWL whereas a duct placed on the air inlet is connected to the outside of the test room (Figure 4). The air outlet is baffled so the impedance is easily computed into FreeFem++.

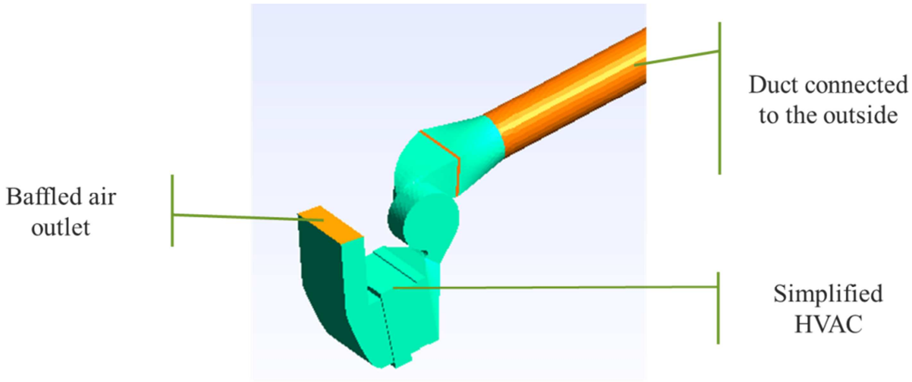

Figure 4: Representation of the calibration test

From the database of the bench, eight measures close to the operating points of the fan placed in the acoustic cavity are identified. Therefore, one may compare the acoustic results between two operating points having the same rotating speed or between two points of the same circuit.

Results analysis and errors identification

The Figure 5 is representative of the eight predictions compared to the acoustic SWLs measured. Even if the overall SWL are close to one another (80 and 82 dBA), the third octave and narrow bands charts show that the method has some shortcomings.

Q=500 kg/h
dp=850 Pa
N=3300 rpm

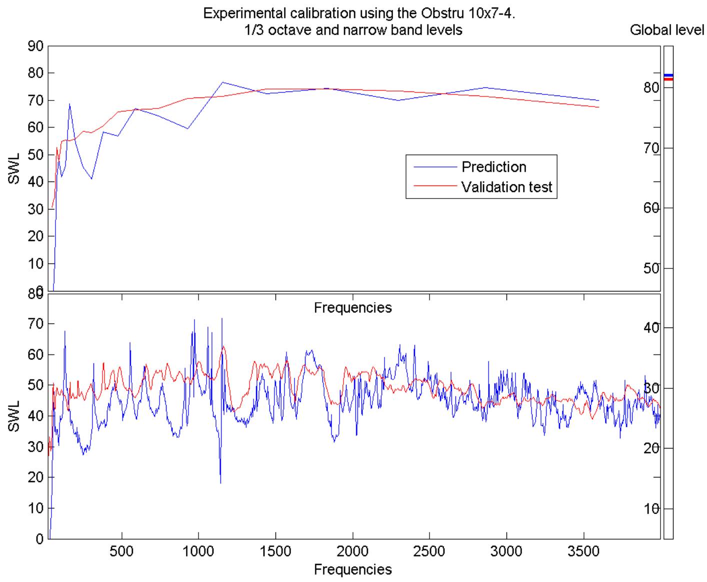

Figure 5: Comparison between the prediction (blue) and the experimental test (red)

The measure shows that the few acoustic modes visible are not responsible for the noise level. Therefore, the tonal noise is negligible with respect to the broadband noise throughout the frequency spectrum.

On the contrary, the prediction gives a significant presence of acoustic modes responsible for the SWL in some third octave bands such as 1000 Hz. As shown in APPENDIX B, the transfer function of the cavity is clearly accountable for the apparition of tonal noise while the bench’s TF varies slowly between 25 and 35 dB. This artificial tonal noise appears below 1500 Hz and causes the greatest differences between the prediction and the measure.

Even if the cavity is composed of absorbing materials, no damping is taken into account during the cavity’s TF calculation. The damping doesn’t get rid of the acoustic modes but their amplitudes can be reduced.

One has been able to quantify the requisite damping based on an experimental characterization of the acoustic cavity of the simplified HVAC. To do so, one has measured the outlet acoustic power produced by a loudspeaker – whose original spectrum is handled – placed at the fan's location in the simplified HVAC.

The second acoustic prediction is presented in the Figure 6.

Q=500 kg/h
dp=850 Pa
N=3300 rpm

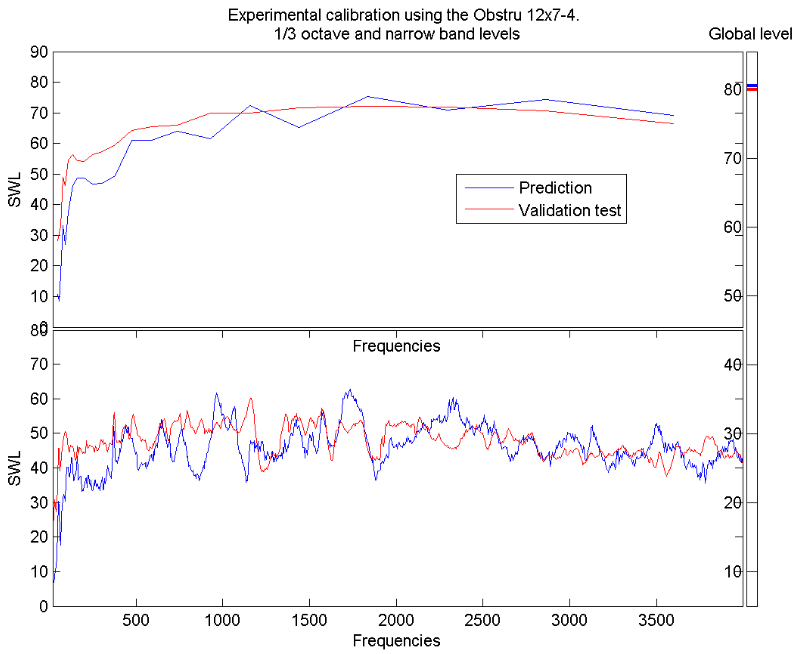

Figure 6: Second result of the acoustic prediction of the noise produced

The acoustic modes are now less visible in the acoustic prediction. Nonetheless, the acoustic prediction is not inadequate below 1200 Hz. One may think that one can't get the whole intrinsic source term from the bench used. In the following months, a bench providing a source term based on a downstream and upstream measure [10] will be used in order to extend the work and determine the suitability of this method.

CONCLUSION

The inverse method relies on the propagation of the fan's intrinsic acoustic term through a complex acoustic medium such as the cavity of an automotive HVAC. The addition of source term in the Helmholtz equation has to be considered in order to modify the weak form used in finite-elements softwares. Demanding low numerical resources and having an acceptable calculation time, the method has advantages in terms of implementation.

In the first version, the acoustic prediction is close to reference measure but some flaws need to be corrected such as the appearance of unobserved acoustic resonances. A suggestion for improvement has been identified but didn't provide the expected betterment.

ACKNOWLEDGEMENTS

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APPENDIX A

The modified Helmholtz is obtained considering the term represented by a force per unit volume noted f and the attenuation term translated by an acoustical damping factor α.

α may be settled equal to zero in the case of the fan which is a particular appliance of the method.

Symbols and relations

The symbols listed below are used to obtain the modified Helmholtz equation:

- c_0 : sound celerity
- ρ_0 : air density
- $\overline{u}$: stationary air velocity
- p: acoustic pressure
- φ: velocity potential
- u: variation of the air velocity
- k: acoustic wave number
- $\overline{M_i} = \frac{\overline{u_i}}{c_0}$, Mach number

The acoustic variables p, u and φ have a $e^{j\omega t}$ time dependence.

Acoustic equations

Based on the equations of momentum and mass conservation, the source term is represented by a force per unit volume noted f:

$$\rho_0 c_0 jku_i = -\frac{\partial p}{\partial x_i} - \rho_0 c_0 \alpha u_i + f_i, \quad (1.1)$$

$$jkp + \rho_0 c_0 \frac{\partial u_i}{\partial x_i} = 0. \quad (1.2)$$

If the stationary air flow has to be considered, one has to modify the equations (1.1) and (1.2):

$$\rho_0 c_0 jku_i + \rho_0 \overline{u_j} \frac{\partial u_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} - \rho_0 c_0 \alpha u_i + f_i, \quad (2.1)$$

$$jkp + \rho_0 c_0 \frac{\partial u_i}{\partial x_i} + \overline{M_i} \frac{\partial p}{\partial x_i} = 0. \quad (2.2)$$

The resolution of the equations (1) and (2) is performed in order to check whether the air flow should be taken into account.

Resolution without air flow

The modified Helmholtz equation is easily obtained by transferring the value of the velocity (1.1) in (1.2). It just remains to multiply by jk to get the modified Helmholtz equation:

$$k^2 p + \sum_{i=1}^{3} \frac{\partial}{\partial x_i}\left[\frac{jk}{jk+\alpha}\left(\frac{\partial p}{\partial x_i} - f_i\right)\right] = 0. \tag{3}$$

The weak form of the Helmholtz formulation, used by the finite-elements solvers, is obtained using the usual integration method [9]:

$$\forall q, \int_V k^2 pq dV - \int_V \frac{jk}{jk+\alpha}\frac{\partial p}{\partial x_i}\frac{\partial q}{\partial x_i} dV + \int_V \frac{jk}{jk+\alpha} f_i \frac{\partial q}{\partial x_i} dV = \int_S jk\rho_0 c_0 u_n dS. \tag{4}$$

Resolution with air flow

In the equation of the momentum conservation (2.1), the added term can be simplified in the case of the velocity deriving from a potential. Then, only the first order terms of the velocity are kept:

$$\overline{u}_j \frac{\partial u_i}{\partial x_j} = \overline{u}_j \frac{\partial}{\partial x_j}\left(\frac{\partial \varphi}{\partial x_i}\right) = \overline{u}_j \frac{\partial}{\partial x_i}\left(\frac{\partial \varphi}{\partial x_j}\right) = \overline{u}_j \frac{\partial u_j}{\partial x_i}. \tag{5}$$

From the previous equation (2.1) and using (5), one may express p with the velocity potential φ. However, this gait binds an additional hypothesis which considers the volume strength as a derivative of a potential. This is merely approximate since it requires the force to be irrotational which may be not the case.

$$p = -\rho_0 c_0 \left[(jk+\alpha)\varphi + \overline{M}_j \frac{\partial \varphi}{\partial x_j} - \frac{\Gamma}{\rho_0 c_0}\right], \tag{6}$$

The equation of mass conservation (2.2) becomes:

$$jkp + \overline{M}_i \frac{\partial p}{\partial x_i} + \rho_0 c_0 \sum_{i=1}^{3} \frac{\partial^2 \varphi}{\partial x_i^2} = 0. \tag{7}$$

Then, one may replace p using (6). The result gives an equation representing the acoustic propagation considering the velocity potential φ as variable:

$$- jk.(jk+\alpha).\varphi - (2jk+\alpha)\overline{M}_i \frac{\partial \varphi}{\partial x_i} + \sum_{i=1}^{3}\left(1-\overline{M}_i^2\right)\frac{\partial^2 \varphi}{\partial x_i^2} + \frac{1}{\rho_0 c_0}.\left(\overline{M}_i \frac{\partial \Gamma}{\partial x_i} + jk\Gamma\right) = 0. \tag{8}$$

The weak form associated with (8) is written below:

$$\begin{aligned}\forall q, &\int_V - jk.(jk+\alpha).\varphi q dV - \int_V (2jk+\alpha)\overline{M}_i.\frac{\partial \varphi}{\partial x_i} q dV - \int_V \left(1-\overline{M}_i^2\right)\frac{\partial \varphi}{\partial x_i}\frac{\partial q}{\partial x_i} dV = \\ &- \int_V \frac{1}{\rho_0 c_0}.\left(\overline{M}_i \frac{\partial \Gamma}{\partial x_i} + jk\Gamma\right) q dV - jk\int_S \left(1-\overline{M}_i^2\right)\frac{\varphi}{Z+\overline{M}_n} dS\end{aligned}. \tag{9}$$

with Z the inlet and outlet surface impedance using the relation $p = Z\rho_0 c_0 u_n$.

Air flow influence

The presence of a damping factor and the source now represented by a force potential are the two notable changes between (3) and (8) in the case of α=0.

In order to determine if the air flow needs to be taken into account, a calculation is performed considering a rectangular duct baffled at its two ends (Figure A.1). The force per unit volume f inserted is unitary. The Noise Power Level (SWL) at one of the outlet is then calculated.

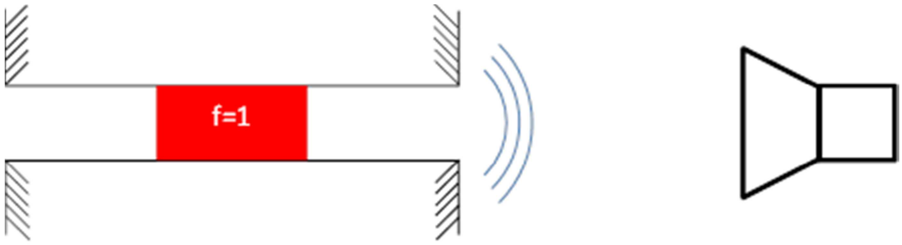

Figure A.1: Pattern of the calculation made to check the air flow influence

In the application case presented, the flows Mach number are well below 0,1. As shown in Figure A.2, the air flow doesn't influence the acoustic results below 1250 Hz. Above this frequency, one may notice a gap increasingly important up to 10 dB at the 3150 Hz band. If considered, the air flow would influence the acoustic prediction as follows:

- The force calculated by (3) would be weaker above the 1250 Hz third octave band.
- But the acoustic prediction calculated by (4) would practically be the same because the cavity transfer function would be higher above the 1250 Hz third octave band.

For now on, the modified Helmholtz equation (3) is used in the finite-elements solver even if the modeling of the air flow in a complex acoustic cavity deserves further studies.

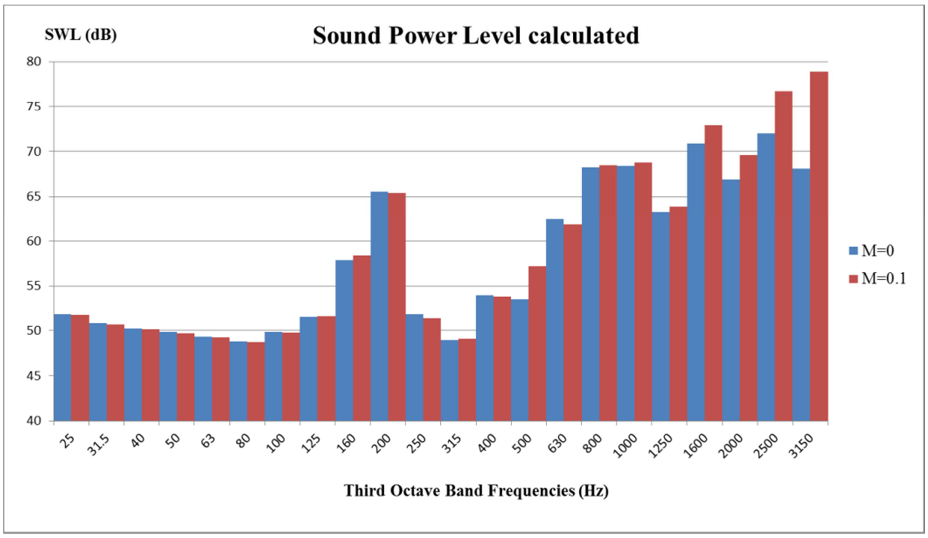

Figure A.2: Air influence on the calculated Sound Power Level (red: M=0.1; blue: M=0)

APPENDIX B

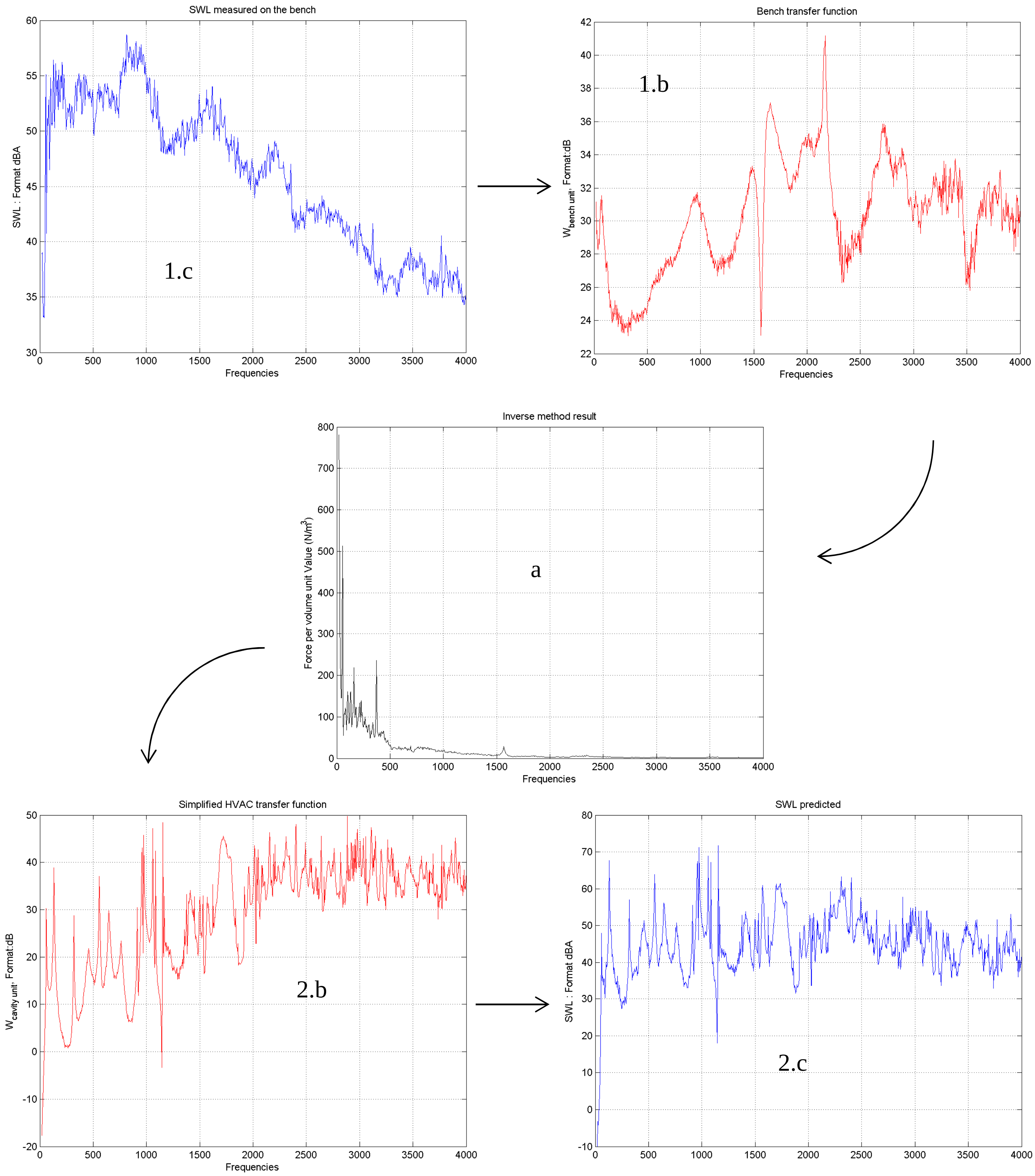

Figure B: Plots of a measure made on the bench (1.c), the transfer function of the bench (1.b), the value of the corresponding force per unit volume (a), the transfer function of the simplified HVAC cavity (2.b) and an acoustic prediction (2.c)

The HVAC transfer function plotted on the Figure B is the result of the first version (no damping added).