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ACOUSTIC MODELING OF FAN NOISE GENERATION AND SCATTERING IN A MODULAR DUCT SYSTEM

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Abstract

Fan noise is an important noise source in computers. The noise spectrum of fans contains tonal noise, found at the so-called Blade Passing Frequency (BPF) and its higher harmonics, that plays an important role in the perceived sound quality. An acoustic resonator integrated in the duct of an in-duct axial fan causes an impedance change in the duct and, depending on the dimensions and location, the resonator acts as an acoustic mirror reflecting the noise back to the fan. By using a resonator on both the inlet and outlet side of the fan the emitted noise can be contained between the resonators thus reducing the noise radiated to the surroundings. In previous publications by the authors, a model was outlined describing viscothermal wave propagation in the duct and in different resonator geometries. A model of the complete resonator setup can be constructed by coupling the solutions for wave propagation in the different elements of the setup. Such a modular model can be used to determine the resonator dimensions and position to optimally reflect noise near the BPF. An important factor in this model is the element that describes noise generation and scattering of incident sound waves by the fan. The description of such an element was not yet available and is presented in this paper. Furthermore, an experimental setup is presented that was built to obtain the different model parameters and to experimentally validate the theory.

INTRODUCTION

One of the main noise sources in computers are the cooling fans. Tonal noise at the rotational frequency of the fan, the so-called blade passing frequency (BPF) and its higher harmonics are important in fan noise. A well known solution to reduce the sound level at specific frequencies in a ducted system is the application of so-called ‘side-resonators’. The side-resonator, a cavity of air connected to the circumferences of the duct, housing an in-duct axial fan, causes an impedance change in the duct. Having the proper dimensions and correct position, the side-resonator acts as an acoustic mirror reflecting the noise back to the fan. When a side-resonator is applied at both sides of the fan, the noise is contained between the resonators and the level of the emitted noise can be reduced (see Figure 1).

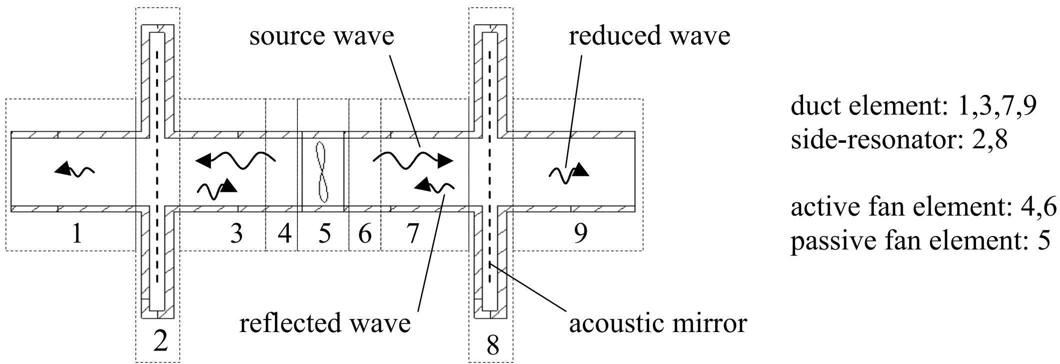


Figure 1 - Wave propagation in a system of coupled acoustical elements

A modular model consisting of multiple acoustic ‘elements’ that describe viscothermal wave propagation in the duct and side-resonators was described in previous publications [1,2,3]. The model can be used to determine the resonator dimensions and position to optimally reflect noise near the BPF.

An important factor in the optimization of the different dimensions is the acoustic properties of the fan. In order to model the characteristics of the fan properly, the model of the fan is split up into two sub-models describing the active and passive contribution, respectively. The active model describes a piece of duct in which sound waves are induced in a single direction (traveling outwards from the fan). The passive model describes the propagation of sound waves that travel through the fan. Both the active and passive model are presented in this paper.

The analytical solutions of the models describing wave propagation through the duct and resonators and the passive and active contribution of the fan are combined in a modular (FEM-like) model. Figure 1 depicts how a coupled system comprised of a ducted fan with multiple resonators might look. A similar system without resonators was built to determine the parameters used in the passive and active models of the fan. A slight modification of this test setup allowed it to be used as a test case to validate the theory of the fan elements.

WAVE PROPAGATION IN A DUCT WITH AIRFLOW

The solution for viscothermal wave propagation in a duct and the description of the corresponding acoustic element that were published previously are valid for quiescent fluids [1]. A simple modification allows the model to be used in case of steady uniform fluid flow with velocity v_{flow}. A schematic representation of the duct element is shown in Figure 2.

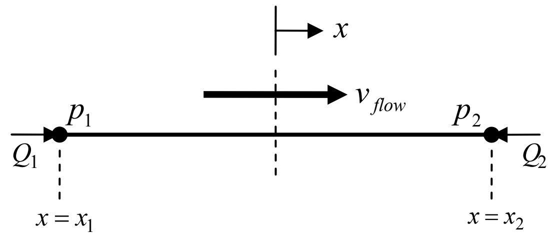


Figure 2 – FEM-like duct element including a steady uniform fluid flow

When the airflow is relatively small, the viscous effects resulting from the airflow can be neglected in the model and the expression for the mass inflows Q_1 and Q_2 can be written in terms of the complex pressure amplitudes p_1 and p_2:

$$\begin{Bmatrix} Q_1 \\ Q_2 \end{Bmatrix} = -\frac{\Gamma B(s)}{\gamma}\rho \begin{bmatrix} A & 0 \\ 0 & -A \end{bmatrix} \begin{bmatrix} e^{\Gamma k_A x_1} & -e^{-\Gamma k_B x_1} \\ e^{\Gamma k_A x_2} & -e^{-\Gamma k_B x_2} \end{bmatrix} \begin{bmatrix} e^{\Gamma k_A x_1} & e^{-\Gamma k_B x_1} \\ e^{\Gamma k_A x_2} & e^{-\Gamma k_B x_2} \end{bmatrix}^{-1} \begin{Bmatrix} p_1 \\ p_2 \end{Bmatrix} \tag{1}$$

In which Γ is the propagation coefficient for viscothermal wave propagation and $B(s)$ a function of the shear wave number s. See [2,4] for a description of Γ, $B(s)$ and s according to the Low Reduced Frequency (LRF) model. The cross-sectional surface of the duct is defined as A, the ratio of specific heats is denoted as γ, and k_A and k_B are the wave numbers for the negative and positive *x*-direction that account for an airflow with velocity v_{flow} [5].

MODELING A FAN AS A TWO-PORT

The different sound waves that determine the sound field on both sides of a ducted axial fan (in operation) are sketched in Figure 3. The open duct ends are located in different rooms A and B separated by an acoustically impenetrable wall.

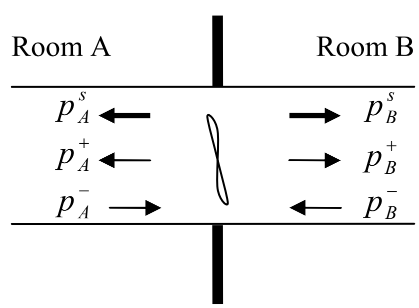


Figure 3 - Sound waves that determine the sound field in a duct with an axial fan

The fan emits sound waves with complex amplitudes p_A^s and p_B^s, towards the individual duct ends. The frequency dependent parameters p_A^s and p_B^s are the so-called source strength components and together they form the source. Depending on the acoustic loads, there are sound waves p_A^- and p_B^- traveling towards the fan as well. These sound waves will be partly reflected by, and transmitted through the fan. The sums of the outgoing sound waves are denoted by p_A^+ and p_B^+. The relation between the different amplitudes is described by the following system of equations:

$$\begin{Bmatrix} p_A^+ \\ p_B^+ \end{Bmatrix} = \begin{bmatrix} \rho_A & \tau_B \\ \tau_A & \rho_B \end{bmatrix} \begin{Bmatrix} p_A^- \\ p_B^- \end{Bmatrix} + \begin{Bmatrix} p_A^S \\ p_B^S \end{Bmatrix} \tag{2}$$

The matrix that relates the amplitudes of the inward traveling waves (p_A^- and p_B^-) to the amplitudes of the outward traveling waves (p_A^+ and p_B^+) is the so called scattering matrix. Its coefficients determine to what extend sound waves are reflected by, or transmitted through the resonator.

The different methods available to determine the frequency dependent parameters in the scattering matrix and the source vector are described by Lavrentjev [6]. The distinction is made between techniques with and without external sources. It is argued that the use of external sources yields the best results in case the correlation between the source strength components is unknown. In this case there is no prior knowledge about the source strength components and the external source method is adopted.

MODULAR APPROACH FOR MODELING A FAN AS A TWO-PORT

The scattering matrix and source term have to be rewritten such that they can be applied in a modular acoustic model that links the analytical solution for wave propagation in ducts and resonators.

The fan is composed of three coupled elements (see figure 1). The center element describes the passive behavior of the fan and determines how incident waves are reflected by and transmitted trough the fan. This passive fan element is linked on both sides to an active fan element. This second type behaves like a duct element in which a sound wave can be induced in a single direction.

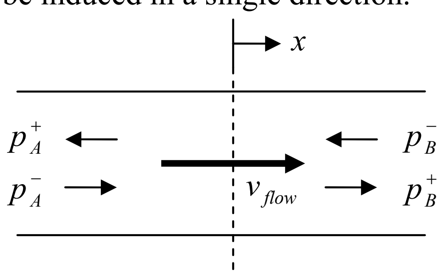


Figure 4 – Passive fan element; a representation in terms of sound waves

The passive fan element behaves like a normal duct element (see figure 2) with a center plane that partially reflects and transmits incident sound waves. A schematic representation of the sound waves in the passive fan element is given in Figure 4.

Adopting the Low Reduced Frequency description (see [1]) for sound waves traveling in a duct, we find the following expression for the pressures p_1 and p_2, and mass flows Q_1 and Q_2 in both nodal points of the element (see figure 2):

$$p_1 = \begin{Bmatrix} e^{\Gamma k_A x_1} & e^{-\Gamma k_B x_1} \end{Bmatrix} \begin{Bmatrix} p_A^+ \\ p_A^- \end{Bmatrix}, \quad p_2 = \begin{Bmatrix} e^{-\Gamma k_B x_2} & e^{\Gamma k_A x_2} \end{Bmatrix} \begin{Bmatrix} p_B^+ \\ p_B^- \end{Bmatrix} \tag{3}$$

$$Q_1 = -\frac{\Gamma B(s)}{\gamma} \rho A \begin{Bmatrix} e^{\Gamma k_A x_1} & -e^{-\Gamma k_B x_1} \end{Bmatrix} \begin{Bmatrix} p_A^+ \\ p_A^- \end{Bmatrix}, \quad Q_2 = -\frac{\Gamma B(s)}{\gamma} \rho A \begin{Bmatrix} -e^{-\Gamma k_B x_2} & e^{\Gamma k_A x_2} \end{Bmatrix} \begin{Bmatrix} p_B^+ \\ p_B^- \end{Bmatrix} \tag{4}$$

When the source term is dropped, equation (2) expresses the relationship between the ingoing and outgoing waves of a two port system. Rewriting this equation makes it possible to relate the backward and forward traveling wave pairs on both sides of the two-port:

$$\begin{Bmatrix} p_A^+ \\ p_A^- \end{Bmatrix} = \frac{1}{\tau_A} \begin{bmatrix} \rho_A & \tau_B \tau_A - \rho_A \rho_B \\ 1 & -\rho_B \end{bmatrix} \begin{Bmatrix} p_B^+ \\ p_B^- \end{Bmatrix} \quad or \quad \begin{Bmatrix} p_B^+ \\ p_B^- \end{Bmatrix} = \frac{1}{\tau_B} \begin{bmatrix} \rho_B & \tau_B \tau_A - \rho_A \rho_B \\ 1 & -\rho_A \end{bmatrix} \begin{Bmatrix} p_A^+ \\ p_A^- \end{Bmatrix} \tag{5}$$

With equations (3) and (5) the complex pressure amplitudes on both sides of the fan are rewritten in terms of the pressure in points 1 and 2.

$$\begin{Bmatrix} p_A^+ \\ p_A^- \end{Bmatrix} = [O] \begin{Bmatrix} p_1 \\ p_2 \end{Bmatrix} \quad with \quad [O] = \begin{bmatrix} \begin{Bmatrix} e^{\Gamma k_A x_1} & e^{-\Gamma k_B x_1} \end{Bmatrix} \\ \begin{Bmatrix} e^{-\Gamma k_B x_2} & e^{\Gamma k_A x_2} \end{Bmatrix} [N] \end{bmatrix} \quad in\ which \quad [N] = \frac{1}{\tau_B} \begin{bmatrix} \rho_B & \tau_A \tau_B - \rho_A \rho_B \\ 1 & -\rho_A \end{bmatrix} \tag{6}$$

$$\begin{Bmatrix} p_B^+ \\ p_B^- \end{Bmatrix} = [P] \begin{Bmatrix} p_1 \\ p_2 \end{Bmatrix} \quad with \quad [P] = \begin{bmatrix} \begin{Bmatrix} e^{\Gamma k_A x_1} & e^{-\Gamma k_B x_1} \end{Bmatrix} [M] \\ \begin{Bmatrix} e^{-\Gamma k_B x_2} & e^{\Gamma k_A x_2} \end{Bmatrix} \end{bmatrix} \quad in\ which \quad [M] = \frac{1}{\tau_A} \begin{bmatrix} \rho_A & \tau_A \tau_B - \rho_A \rho_B \\ 1 & -\rho_B \end{bmatrix} \tag{7}$$

Substitution of equations (6) and (7) in equation (4) yields the mass flows in points 1 and 2 in terms of the pressure in those points. This is the form that is required to couple the passive element with other acoustical elements.

$$\begin{Bmatrix} Q_1 \\ Q_2 \end{Bmatrix} = -\frac{\Gamma B(s)}{\gamma} \rho \begin{bmatrix} A & 0 \\ 0 & -A \end{bmatrix} \begin{bmatrix} \begin{Bmatrix} e^{\Gamma k_A x_1} & -e^{-\Gamma k_B x_1} \end{Bmatrix} [O]^{-1} \\ \begin{Bmatrix} -e^{-\Gamma k_B x_2} & e^{\Gamma k_A x_2} \end{Bmatrix} [P]^{-1} \end{bmatrix} \begin{Bmatrix} p_1 \\ p_2 \end{Bmatrix} \tag{8}$$

The active fan element behaves like a normal duct element in which a sound wave is induced in a single direction. The sound waves are induced by harmonic injection/extraction of mass at the inlet and outlet of the element. In Figure 5 a schematic representation is given that demonstrates how sound fields caused by the cyclic mass injection/extraction at the two points in the duct result in two sound waves traveling outward from the center of the active element.

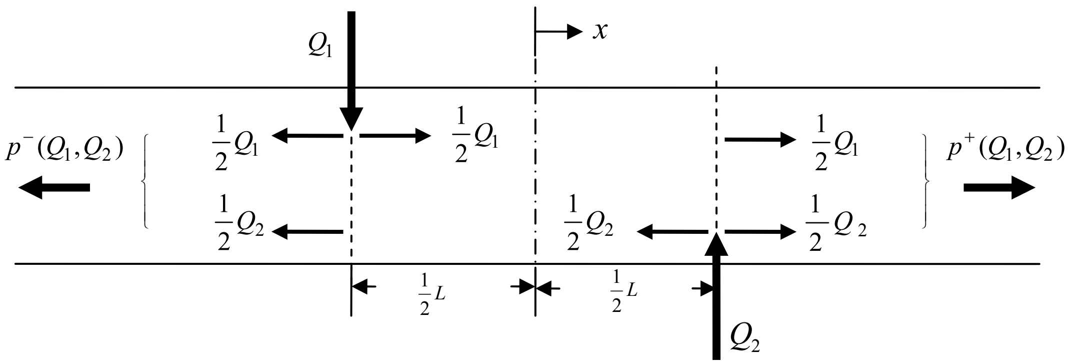


Figure 5 - Combination of sound fields resulting from two points of mass injection

Equation (4) can be used to describe the complex amplitude of a sound wave induced by a mass flow. With this equation, the relation between the complex amplitudes Q_1 and Q_2 of the mass flows and the complex amplitudes p^+ and p^- of the sound waves traveling outward from the center of the element can be defined as:

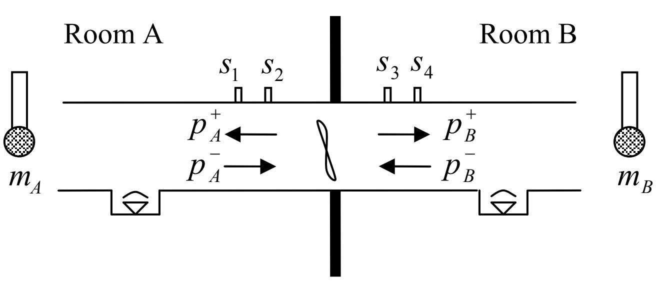
$$\begin{Bmatrix} p^- \\ p^+ \end{Bmatrix} = -\frac{\gamma}{2\Gamma B(s)} \frac{1}{\rho A} \begin{bmatrix} e^{\Gamma k_A \frac{1}{2} L} & e^{-\Gamma k_A \frac{1}{2} L} \\ e^{-\Gamma k_B \frac{1}{2} L} & e^{\Gamma k_B \frac{1}{2} L} \end{bmatrix} \begin{Bmatrix} \frac{1}{2} Q_1 \\ \frac{1}{2} Q_2 \end{Bmatrix} \tag{9}$$

Using this relation the complex amplitudes of the mass flows Q_1 and Q_2 needed to induce a sound wave p^s in the positive *x*-direction without inducing a wave in the opposite direction can be calculated:

$$\begin{Bmatrix} Q_1 \\ Q_2 \end{Bmatrix} = -\frac{2\Gamma B(s)}{\gamma} \rho A \begin{bmatrix} e^{\Gamma k_A \frac{1}{2} L} & e^{-\Gamma k_A \frac{1}{2} L} \\ e^{-\Gamma k_B \frac{1}{2} L} & e^{\Gamma k_B \frac{1}{2} L} \end{bmatrix}^{-1} \begin{Bmatrix} 0 \\ p^S \end{Bmatrix} \tag{10}$$

EXPERIMENTAL VALIDATION

The experimental setup depicted in Figure 6 was manufactured to determine the scattering matrix and source term used in the model described above. The modular design of the setup allows it to be modified easily, so it can be used for verification of the presented theory as well.



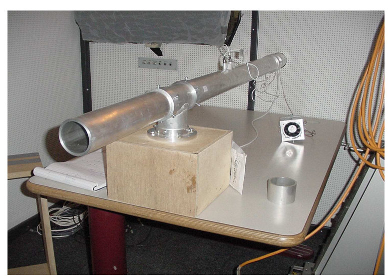


Figure 6 - Schematic representation and photo of the test setup

The setup consists of a fan placed in the center of a duct. The open duct ends are separated by a sound proof wall. Both sides of the duct are equipped with a pair of pressure sensors (s_1 to s_4) and a reference microphone m_1 and m_2 (the reference signals from the microphones are used for flow noise reduction). A speaker that is used to induce a sound field of much higher intensity then the sound field induced by the fan is connected to the duct on each side of the soundproof wall.

An example of the measured scattering matrix parameters and source strength components of a cooling fan running at 12V are plotted in figure 7. It follows from the dimensions of the test setup that the (1D) LRF model used is valid for frequencies below 3300 Hz. For higher frequencies the wave propagation in the duct can not be considered 1D and the model can no longer describe the sound field in the duct. This is observed in the plots of the parameters in the scattering matrix in figure 7.

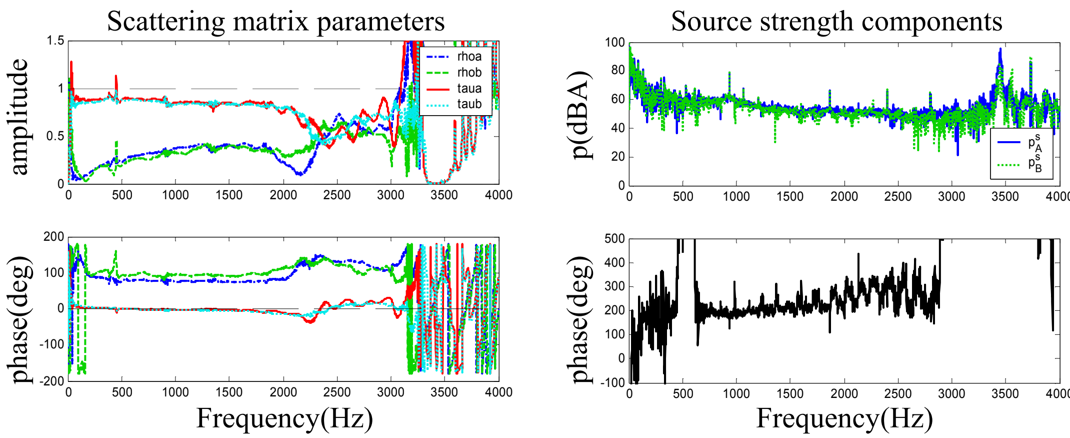


Figure 7 – Measured scattering matrix parameters and source strength components

To validate the theory of the passive and active fan elements, the scattering matrix and source vector in Figure 7 are used as input for the fan elements in a model that predicts the pressure sensor signals of an (acoustically) altered test setup; i.e. the acoustic loads on the fan are changed by shortening the ducts on both sides of the fan.

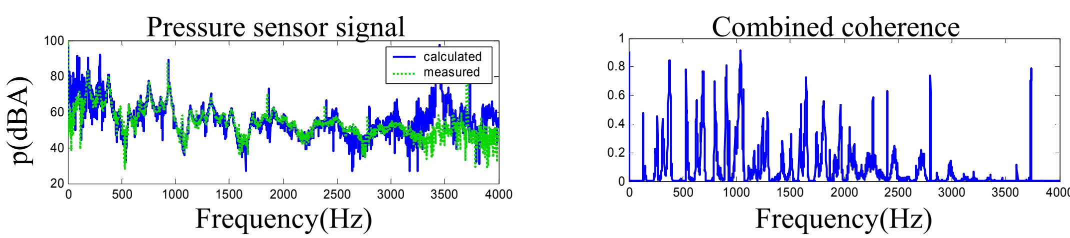


Figure 8 – Calculated and measured pressure signal and combined coherence

The measured and calculated pressure signals, presented in Figure 8, show a good correspondence between 300 and 2500 Hz, hence, the model is accurate enough to be used in optimization of a resonator design in that frequency interval. However, in two regions below the cut-off frequency, large deviations can be observed between the measured and calculated results. These regions correspond with frequencies for which the coherence of the pressure sensor signals with the signal of the reference microphones is relatively low. This is reflected by the plot of the *combined coherence* in figure 8. The combined coherence is the product of the coherence of the individual pressure signals, a quantity that is low if one or more of the pressure signals is not coherent with the sound field in the duct.

The main reason for the bad coherence of the pressure signals (besides flow noise) is the low level of the noise that is picked up by the pressure sensors. Due to the end condition of the duct, the sound field has a standing wave form for certain frequencies. In case a pressure sensor is located near a pressure node of a standing wave, the pressure fluctuation that is measured is weak and noise will dominate the sensor signal. Furthermore, the strength of the sensor signals diminishes with increasing frequencies as a result of the influence of the duct system causing the sensor signal to gradually sink into the noise floor. These effects explain the decrease of the combined coherence and the increasing discrepancies between calculated and measured pressure signal for increasing frequencies. Lastly, the noise floor of the sensors increases dramatically below 300 Hz which explains the deviations in that region. Possible solutions to these problems are decreasing the duct lengths, using sensors with a lower noise floor and the use of more sensor positions.

CONCLUSION

The theory of a modular model of a duct system including an axial fan (in operation) and fluid flow was outlined. An experimental test setup was presented that was build to obtain the different parameters used to characterize the model of the fan. The measured results from an (acoustically) different test setup were compared with results from the modular model; the measured and calculated data showed good correspondence in the 300-2500 Hz frequency range. It was found that the discrepancies outside this interval are caused by insufficient strength of the pressure fluctuations at the pressure sensors. Possible solutions to these problems are decreasing the duct lengths, using sensors with a lower noise floor and the use of more sensor positions.

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