On the Simulation of ElectroPneumatic Transducers

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Introduction

The aim of this paper is to find a model for the sound source of an electropneumatic transducer (EPT) — sometimes called an electropneumatic sound source or compressed-air loudspeaker — suitable for fitting into a feedforward active noise control scheme. That is, a time-discrete input-output model from which the output of the secondary source can be directly calculated in every time step by current and past input signal values and current and past output or internal signal values. Nevertheless, once the nonlinear models are constructed they cannot simply be inserted into the filtered-x LMS algorithm used with linear models, e.g. FIR filter models. Instead new algorithms must be developed which consider the nonlinear behavior of the nonlinear secondary path models. But this is not the topic of this paper.

Electropneumatic sound sources modulate the airflow through small orifices (Figure 1) in order to generate tones corresponding to the frequency of the modulation. Unfortunately, in practical devices some nonlinearities occur and higher harmonics are excited along with the fundamental frequency. In Figure 2 a time-discrete block diagram of a model of the sound source is shown. The electromechanical part from the driving voltage U_e to the displacement of the slider x is currently well modelled including the nonlinear friction curve [1].

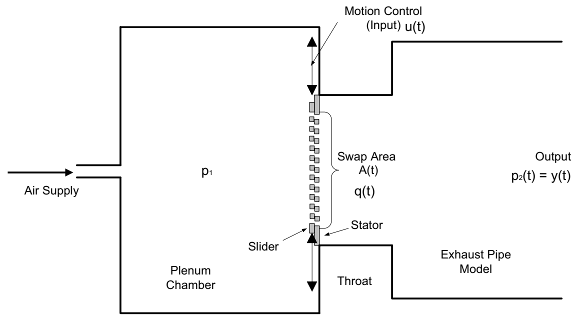

Figure 1: Sketch of the electro pneumatic transducer.

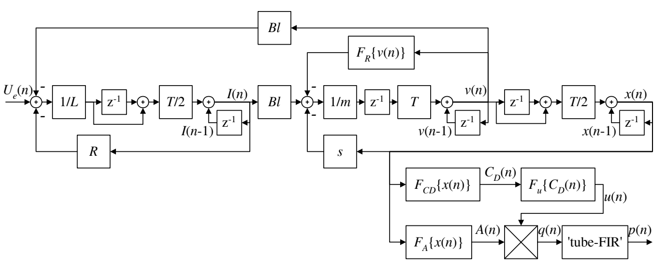

Figure 2: Time-discrete block diagram.

Modelling of the aeroacoustical part from the displacement x of the slider to the sound pressure p' is the topic of this paper. For simplicity the block called 'tube-FIR' in Figure 2 is neglected here. It is also assumed that the EPT radiates sound directly into open space and is not connected to an exhaust pipe.

Acoustic Model

The model for the acoustic source was initially taken to be a simple monopole due to the displacement of air by the jets leaving the holes [2]:

$$p'(\mathbf{x}, t) = \frac{Q(t - r/c)}{4\pi r}, \qquad (1)$$

where $r = |\mathbf{x}|$. The source strength Q is assumed to be given by the rate of change of the mass flow rate with time. The mass flow rate $\dot{m}$ is assumed to be quasi-steady and given by:

$$\dot{m} = C_D A \sqrt{2\varrho(P_p - p_a)}, \qquad (2)$$

where P_p and p_a are the upstream total pressure in the plenum and the downstream static pressure respectively, ϱ is the density of air and A is the area of the holes. The discharge coefficient C_D was determined as a function of relative displacement of the overlapping holes from a set of CFD simulations assuming an unsteady incompressible fluid [3].

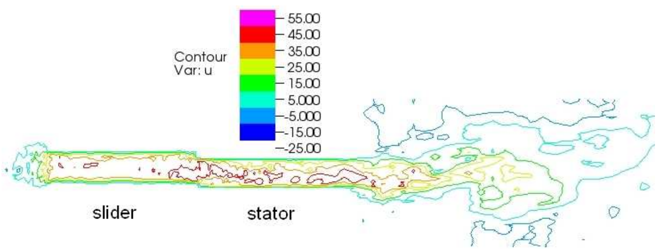

Figure 3: Axial velocity contours from CFD simulation.

Results

Figure 4 shows two typical simulated pressure signals using the acoustic model and Figure 5 shows the equivalent signal measured in an anechoic chamber.

The symmetry of the hole movement in the model means that no fundamental or odd harmonics can be predicted. However, a small offset of 5% of the hole diameter is a mechanism for introducing the absent components as shown in Figure 4.

The largest discrepancy between measurements and predictions is the level which is over an order of magnitude smaller in the predictions than the measurements even when assuming a lossless model.

Discussion

The results show that the current model for the acoustic source is unusable and is clearly excluding an important physical process. The strong asymmetry in the measured pressure signal dictates that the model for the acoustic source must have a dependency on something other than the symmetric motion of the holes. The asymmetry is too weak from friction and small errors in centring the hole motion.

The velocity gradients in the downstream jet lead to a relatively weak source density distributed over a fairly large volume compared to the size of the hole. On the upstream side, the velocity gradients associated with sucking the air into the hole are located very close to the hole entrance leading to a much stronger source density and a significant proportion of the acoustic energy being radiated into the hole. There will be a partial reflection at the junction between the moving and stationary holes and at the exit but this is clearly a mechanism for introducing asymmetry into the simulated pressure signal. It can also be modelled in a form simple enough to be included in the control model subject to determining some empirical information from supporting numerical simulations and measurements.

A physical mechanism for the large difference in level would appear to follow from the upstream plenum and the holes acting as a Helmholtz resonator. The equation [2]:

$$f = \frac{1}{2\pi}\sqrt{\frac{c^2 S}{Al}} \tag{3}$$

with end corrections neglected gives a resonance of 64 Hz which is well inside the intended operating range of the transducer. Again, this can probably be modelled in a form simple enough to use in the control model subject to determining supporting empirical information from numerical simulations and measurements.

Conclusions

Despite the transducer operating at less than 0.1 Mach number and the holes moving several orders of magnitude slower than the air through them, an initial acoustic model based on quasi-steady incompressible flow was inappropriate because of the presence of an acoustic resonance.

Work is currently underway to develop an alternative acoustic model for the control scheme using empirical data determined from numerical simulations using a compressible low Mach number approach.

References

[1] A. Jakob, M. Möser. Nonlinear models of electro pneumatic transducers for use in feedforward active noise control schemes, Proceedings of ACTIVE 2006, Adelaide, Australia

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[3] J. Kim, P. Moin, Application of a fractional-step method to incompressible Navier-Stokes equations, J. Comp. Phys., 59, 308-323, 1985

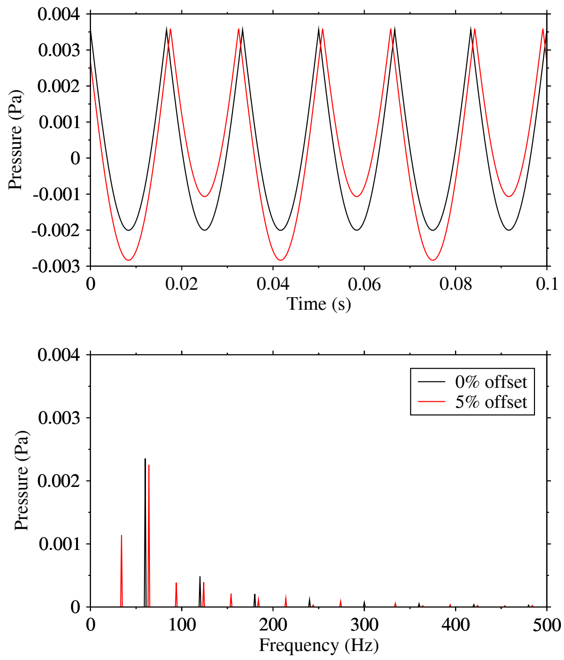

Figure 4: Predictions at 30Hz and 30% movement.

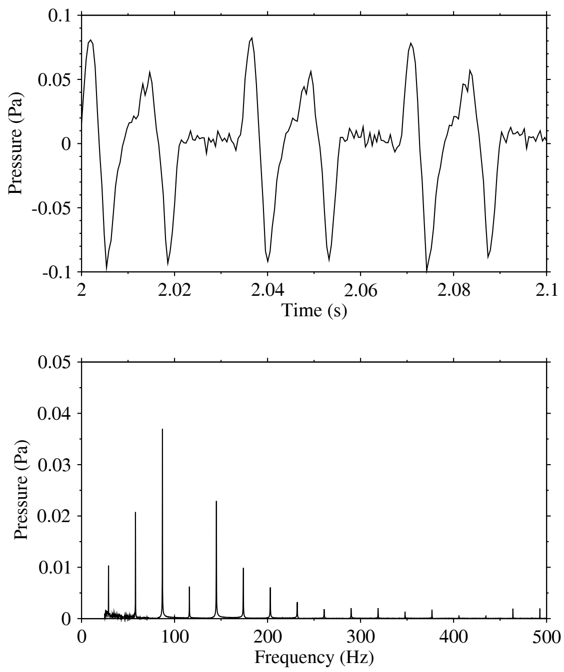

Figure 5: Measurements at 30Hz and 30% movement.