

A Study on Acoustical Time-Domain Two-Ports Based on Digital Filters with Application to Automotive Air Intake Systems

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ABSTRACT

Analysis of pressure pulsations in ducts is an active research field within the automotive industry. The fluid dynamics and the wave-transmission properties of internal combustion (IC) engine intake and exhaust systems contribute to the energy efficiency of the engines and are hence important for the final amount of CO₂ that is emitted from the vehicles. Sound waves, originating from the pressure pulses caused by the in- and outflow at the engine valves, are transmitted through the intake and exhaust system and are an important cause of noise pollution from road traffic at low speeds. Reliable prediction methods are of major importance to enable effective optimization of gas exchange systems. The use of nonlinear one-dimensional (1D) gas dynamics simulation software packages is widespread within the automotive industry. These time-domain codes are mainly used to predict engine performance parameters such as output torque and power but can also give estimates of radiated orifice noise. However, components with large cross-dimensions, fluid-structural interaction, frequency-dependent damping and boundary conditions are difficult to describe analytically in 1D in the time domain. Since a frequency-domain description in the form of a two-port is normally straightforward to obtain analytically, numerically or experimentally it is of interest to introduce these in time-domain calculations as black box models. This paper suggests the use of Finite Impulse Response (FIR) filters as a method to achieve this improvement. An initial study is presented where tabulated frequency-domain two-port data representing an air cleaner unit on the impedance form is inversely transformed to the

time domain and used as FIR filters in nonlinear time-domain 1D calculations with good accuracy. Favourable attenuation, achieved from the filter paper itself, is demonstrated experimentally as well as by the calculations.

INTRODUCTION

The theory of sound in ducts is important for a wide range of applications in the modern society. Technical systems that transport some kind of fluid or gas and which are connected to a fluid machine often exhibit pressure oscillations in the connected pipes as a result from the characteristics of both the source and the connected pipe system. Important examples are the noise that is produced in internal combustion (IC) engines and contributes severely to the noise pollution in populated areas and the pulsations that might cause fatigue damage to industrial pipe-networks if left unattended. The design of such systems is often initially carried out by simulations and reliable prediction of in-duct fluctuating pressures in a wide frequency band is therefore of large interest. In principle there are two approaches to treat the problem; time-domain solutions and frequency-domain solutions. Within the automotive industry the use of nonlinear, one dimensional (1D) time-domain solution strategies to analyse the dynamics of the gas exchange system is widespread. Commercially available, highly specialized software packages are used to predict engine performance parameters such as torque and power. These codes also give time-varying pressures and velocities within the exhaust and intake system. However, the inherent limitation of 1D modelling makes the predictions somewhat unreliable for components where the cross-sections are comparable to or

larger than the wave length. Effects of vibrating walls as well as frequency-dependent damping and boundary conditions are also difficult to include in the analyses. Frequency-domain descriptions based on the two-port method (or four-pole) [1] are generally much more computationally efficient. Components with large dimensions, effects of vibrating walls and frequency-dependent damping can be included without too much effort, especially if experimental data is available. Procedures to obtain experimental two-port data are well documented see e.g. the work by Åbom and Bodén [2] and Åbom [3]. Major disadvantages concerning two-port frequency-domain modelling include the requirement for linearity, that a linear source has to be specified and that a predetermined cyclic frequency is required. Research about hybrid methods, where the benefits from the time-domain and frequency-domain modelling are combined, is therefore an active field. Albertson *et al.* [4] has e.g. investigated different methods to couple nonlinear source descriptions in the time domain to linear system descriptions in the frequency domain. Payri *et al.* [5] and Chiavola [6] studied aspects of using acoustic boundary conditions together with nonlinear 1D calculations based on iterations between time- and frequency-domain solutions. Several efforts to introduce acoustic boundary conditions, i.e. local impedance or one-port models, in computational aero acoustics (CAA) have been reported with application to acoustic liners in aircraft engines. A technique that has been reported to be promising is the impulse response convolution technique as a way to introduce locally reacting impedance boundary conditions which was proposed by Özyörük *et al.* [7]. A method to create causal impedance boundary conditions for Helmholtz resonators in the time-domain was suggested by Rienstra [8]. The use of the two-port method is also widespread within the electronic community; in this case for design of electrical networks. An interesting method to achieve causal frequency response functions from tabulated finite-bandwidth data with this application was recently reported by Triverio and Grivet-Talocia [9].

The present paper presents an initial study on representing two-ports in time-domain nonlinear 1D gas exchange calculations by using Finite Impulse Response (FIR) filters. This can be considered as a simple form of system identification [10]. Experimentally obtained two-ports for a box with and without a filter paper mounted inside are presented and the interesting sound attenuation capabilities of the filter paper are highlighted. The two-ports, in impedance-matrix form, are inversely transformed to the time domain and are thereafter used as FIR filters in time-domain finite volume calculations of a simple duct network. The gas exchange calculations and filtering operations are carried out using the commercial software packages GT-POWER [11] and Simulink [12], respectively. This new approach enables more accurate modelling of losses, higher-order modes and flexible walls but does not suffer from the need of a

predetermined cyclic frequency that is required in frequency-domain two-port modelling.

MODELLING

PLANE WAVES IN DUCTS

Sound waves travelling in a duct system can for sufficiently low frequencies be approximated as being constant throughout the duct cross-section. The limit for this approximation is set by the appearance of cross-modes present for frequencies larger than their corresponding cut-on frequencies. For a circular duct with radius R the cut-on frequency for the first cross-mode can be calculated as, see Ref [1],

$$f_{\text{cut-on}} = \frac{\alpha_{10} c_0}{2\pi R} \quad (1)$$

where $\alpha_{10} = 1.841$ is the first zero of the Bessel function J'_1 , $c_0 = (\gamma p_0 / \rho_0)^{1/2}$ the isentropic speed of sound, γ the specific heat ratio and p_0 and ρ_0 are the pressure and density of the undisturbed gas, respectively. The approximation of plane waves is of large importance for reducing sound in ducts into a 1D problem that is much simpler to solve than the full 3D case. For automotive gas exchange applications in the passenger car segment the diameters of the ducts are typically less than 0.1 m which yields an upper limit for the plane wave assumption of about 2000 Hz. In components with larger cross-dimensions such as silencers, after-treatment devices, charger air coolers and air cleaner units cross-modes will appear for much lower frequencies and more advanced techniques will have to be used. Several such techniques exist, e.g. the null-field approach [13], the point-matching method [14], the finite element method (FEM) or the boundary element method (BEM). For irregularly shaped components such as an air cleaner unit for which the design often is based on the available space in the engine compartment and a requirement for a certain volume, most likely no analytical solution will exist. In order to predict sound transmission through such devices a numerical method has to be used. Unfortunately the computational effort will be much larger than for a 1D solution. However, as long as the component is connected to ducts with cut-on frequencies below the frequency of interest and that linear theory is valid it is possible to reduce the description of the complete system by using the two-port method [1]. When the two-port for a component is obtained, the frequency-domain representation consists just of a $[2 \times 2]$ -matrix for each frequency component.

LINEAR FREQUENCY-DOMAIN TWO-PORTS

The acoustic two-port method has been known for a long time and its use is, due to its simplicity, widespread. Basically, a two-port describes a linear and time-invariant system with an input and an output port. The relations between two state variables at each port are, in the frequency domain, described by a $[2 \times 2]$ -matrix. Depending on how the state variables are chosen the two-port becomes suitable for different problems. In acoustics a common choice is to use the acoustic pressure $\hat{p}$ and the acoustic volume velocity $\hat{q}$ to describe both the input and output state which leads to the transfer-matrix relation

$$\begin{bmatrix} \hat{p}_{\text{IN}} \\ \hat{q}_{\text{IN}} \end{bmatrix} = [\hat{\mathbf{T}}] \begin{bmatrix} \hat{p}_{\text{OUT}} \\ \hat{q}_{\text{OUT}} \end{bmatrix}, \quad (2)$$

where $\hat{}$ denotes a Fourier-transformed quantity and the indices IN and OUT refer to the name of the port as is shown in Fig. 1.

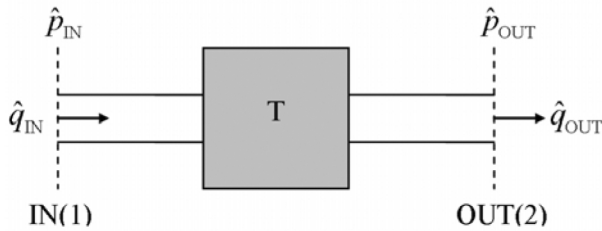


Figure 1. Acoustical two-port.

One advantage of using the transfer-matrix formalism is that a cascade-coupled chain of n elements easily can be combined to yield a representation of a complete system. Mathematically this corresponds to a series of matrix multiplications as

$$[\hat{\mathbf{T}}] = [\hat{\mathbf{T}}_1] \cdot [\hat{\mathbf{T}}_2] \cdot \dots \cdot [\hat{\mathbf{T}}_n]. \quad (3)$$

A formalism that has been shown to be useful when a system is coupled in parallel is based on mobility-matrices whose state variables are related as

$$\begin{bmatrix} \hat{q}_{\text{IN}} \\ \hat{q}_{\text{OUT}} \end{bmatrix} = [\hat{\mathbf{M}}] \begin{bmatrix} \hat{p}_{\text{IN}} \\ \hat{p}_{\text{OUT}} \end{bmatrix}. \quad (4)$$

The scattering-matrix formalism, which is based on travelling pressure waves as state variables, is defined as

$$\begin{bmatrix} \hat{p}_{\text{IN-}} \\ \hat{p}_{\text{OUT+}} \end{bmatrix} = [\hat{\mathbf{S}}] \begin{bmatrix} \hat{p}_{\text{IN+}} \\ \hat{p}_{\text{OUT-}} \end{bmatrix}, \quad (5)$$

where the sign indicates in what direction a wave propagates with reference to the arrows in Fig. 1. In Ref [15] Glav and Åbom argued that this formalism is the most general and can be used to describe complex systems with different types of connections. In the present work the impedance-matrix formalism, where the acoustic pressures and volume velocities at the ports are related as

$$\begin{bmatrix} \hat{p}_{\text{IN}} \\ \hat{p}_{\text{OUT}} \end{bmatrix} = [\hat{\mathbf{Z}}] \begin{bmatrix} \hat{q}_{\text{IN}} \\ \hat{q}_{\text{OUT}} \end{bmatrix}, \quad (6)$$

is used. This formulation is chosen as it appears to be the most straightforward approach for implementation in GT-POWER.

If the transfer-matrix properties are known the impedance-matrix components can easily be calculated as

$$\begin{bmatrix} \hat{Z}_{11} & \hat{Z}_{12} \\ \hat{Z}_{21} & \hat{Z}_{22} \end{bmatrix} = \begin{bmatrix} \hat{T}_{11}/\hat{T}_{21} & \hat{T}_{12} - \hat{T}_{11}\hat{T}_{22}/\hat{T}_{21} \\ 1/\hat{T}_{21} & -\hat{T}_{22}/\hat{T}_{21} \end{bmatrix}. \quad (7)$$

The transfer-matrix for a straight duct element of length L without any mean flow and without any losses is [1]

$$[\hat{\mathbf{T}}] = \begin{bmatrix} \cos(kL) & iZ_0 \sin(kL) \\ i \sin(kL)/Z_0 & \cos(kL) \end{bmatrix}, \quad (8)$$

where i is the complex operator, k the wave number, $Z_0 = \rho_0 c_0 / S$ the characteristic impedance for propagating waves and S the duct cross-sectional area. When the transfer-matrix of a system is known the transmission loss can easily be calculated using the equation [1]

$$TL = 10 \cdot \log \left[\frac{Z_{\text{OUT}}}{4Z_{\text{IN}}} \left| \hat{T}_{11} + \frac{\hat{T}_{12}}{Z_{\text{OUT}}} + Z_{\text{IN}} \hat{T}_{21} + \frac{Z_{\text{IN}} \hat{T}_{22}}{Z_{\text{OUT}}} \right|^2 \right], \quad (9)$$

where Z_{IN} and Z_{OUT} denote the characteristic impedances for propagating waves at the in- and outlet respectively and $\hat{T}_{11}$, $\hat{T}_{12}$, $\hat{T}_{21}$ and $\hat{T}_{22}$ are the components of the transfer-matrix. The effect of mean flow is not considered in the present work and all provided measurements are carried out without flow. In order to calculate the fluctuating pressure at any specific internal position or at the orifice of the system the characteristics of the source has to be known.

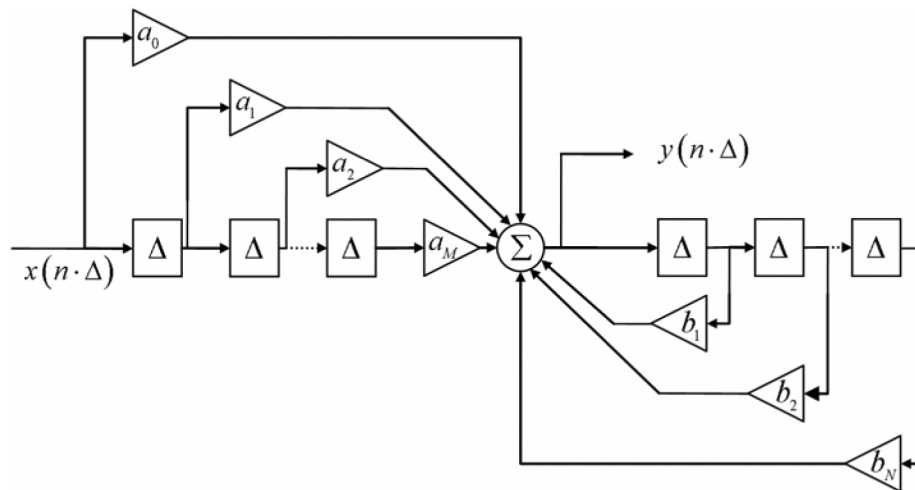


Figure 2. Block diagram of an IIR filter.

NONLINEAR TIME-DOMAIN GAS DYNAMICS SIMULATIONS

Situations where linear methods are not feasible for description of gas dynamics arise for example when pressure pulsations with very large amplitudes are present. This is the case in those ducts that are located close to the inlet and exhaust valves in the gas exchange system of an IC-engine. Due to the cancelling effect present on multi-cylinder engines with reasonably symmetric manifolds the amplitude of the pressure pulsations becomes much more moderate upstream the intake manifold and is typically less than 150 dB. They can therefore normally be described, with good accuracy, by using linear theory. However, if the complete gas exchange system, from air intake to exhaust tailpipe, is to be treated using the same solution strategy a nonlinear tool will have to be used. Several highly specialized commercial software packages are used within the automotive industry for this purpose e.g. GT-POWER, Ricardo WAVE and AVL BOOST. In order to get reasonable calculation times plane waves are assumed and hence a 1D problem is achieved. In this study the commercial software GT-POWER from Gamma Technology is used. This nonlinear code uses the finite volume approach to solve the 1D compressible gas dynamics equations for mass, energy and momentum. The solver that is used for the present calculations is explicit in the time domain. Flow quantities are calculated at the finite volume boundaries, while scalar properties such as pressure and temperature are predicted at the volume centres. For more information about the software and how the equations are solved the reader is referred to [11].

The strategy in the present investigation is to use a gas dynamics model that is split into two parts and connected with a time-domain two-port represented by FIR filters. The discrete fluctuating velocities at the interfacing positions are extracted and communicated to the external software

Simulink [12] where the filtering operations are carried out. The resulting fluctuating pressures at the interfaces are thereafter reintroduced to the gas dynamics model. This procedure implies the use of the impedance-matrix for the two-port, see Eq. (6).

DIGITAL FILTERS

A digital filter is a linear time-invariant discrete time system that performs mathematical operations on a sampled signal $x(n\Delta)$ with the purpose of attenuating or transmitting certain frequency bands. The operations that take place within a digital filter can be described by a linear difference equation on the form [16]

$$y(n\Delta) = \sum_{m=0}^M a_m x(n\Delta - m\Delta) - \sum_{j=1}^N b_j y(n\Delta - j\Delta), \quad (10)$$

where $y(n\Delta)$ is the output signal for sample n and Δ the sampling interval. M and N are the total number of filter coefficients on the input and output side, respectively. A filter where the coefficients $a_m \neq 0$ and $b_j \neq 0$ is called an Infinite Impulse Response filter (IIR). An IIR filter depends on previous values of both the input and the output, i.e. it is recursive. As a consequence it can become unstable because of the feedback loop for the output signal. A schematic illustration of an IIR filter is shown in Fig. 2.

A filter where the coefficients $b_j = 0$ is called a Finite Impulse Response filter (FIR) and is non-recursive, i.e. it only depends on the input signal. Normally a FIR filter requires more coefficients than an IIR filter to obtain the same amplitude characteristics. Important advantages of FIR filters include that they are stable and simple to implement. The computationally faster IIR filters are popular in situations where distributed boundary conditions are used, such as analyses of air craft engine liners. In this initial study, which is in 1D, the calculations are very fast despite only FIR filters