Dual-Driver Standing Wave Tube: Acoustic Impedance Matching with Robust Repetitive Control

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Abstract
In many applications of acoustic standing wave tubes, for instance thermoacoustic heat pumping systems, it is desirable to make a shorter tube operate like a longer standing wave tube at the same driving frequency. The basic idea here is to reduce the physical length of the tube, and replace the removed section with a secondary driver. The problem is then to match the acoustic impedance at the boundary where the secondary driver is installed to that of the original system. Two control formulations were investigated: a two-input-two-output (TIITO) and a single-input-single-output (SISO) formulation. The TIITO formulation directly tracks the two acoustic variables related to the impedance, while the SISO formulation minimizes the impedance matching error. The desired impedance containing a very lightly damped mode is embedded in the augmented plant for feedback control design. In addition to the balance realization method, the Schur method was used in model reduction for the high-order non-minimum phase plants. Since the standing wave tubes are driven by tonal signals, repetitive control was incorporated into the control frameworks to achieve the desired performance. Good impedance matching performance was obtained for both formulations. The SISO formulation yielded slightly wider bandwidth of good impedance matching than the TIITO. The TIITO formulation offered additional control to individual signals related to the acoustic impedance of interest.

1. Introduction
Ideal velocity-driven, closed standing wave tubes have a length equal to one half of the acoustic wavelength at their fundamental resonance frequency [1]. The length of such tubes for operation near resonance must therefore be inversely proportional to the driving frequency. For some applications, space is at a premium and the resonator creates a large “dead” volume. Of particular interest in this study, a thermoacoustic refrigerator prototype consists of a half-wavelength standing wave tube containing a pressurized inert gas mixture [2-4]. A stack and heat exchangers are porous components that realize the heat pumping. For a half wavelength standing wave tube (Figure 1), they must be installed close to the driver, i.e. near the pressure antinode. These components occupy only a small fraction of the total length. The rest of the tube is a “dead volume” since it does not contribute to heat pumping except for forming the standing wave. The “dead volume” increases heat losses and may induce acoustic streaming. Both these effects are detrimental to the thermal performance of the system.

Under these circumstances, it is desirable to create compact tubes with an acoustic behavior similar to that of a longer standing wave tube at the same driving frequency. The basic idea here is to reduce the length of the tube and to replace the passive resonator by a secondary driver (Figure 2). This is possible if the dual-driver system is properly controlled to match the boundary conditions, i.e. the impedance at one location. Such a controlled dual-driver system can also provide the additional benefit of emulating standing wave tube of different lengths without any change of hardware. This facilitates the study, for the case of thermoacoustic devices, of the optimal stack position in the acoustic field. In the thermoacoustic device of interest, the temperature of the system (and thus the speed of sound) may change continuously during operation requiring frequent adjustments in the operating frequency in order to maintain the system at resonance. For this reason, mechanical resonator terminations with fixed parameters do not allow continuous operation at resonance without simultaneously modifying the position of the stack in the standing wave.

Figure 1. Diagram of a single-driver thermoacoustic cooler

Figure 2. Diagram of proposed dual driver thermoacoustic cooler

Impedance matching problems have been investigated in the past. Most thermoacoustic prototypes to date have used a Helmholtz resonator at the end. A typical configuration is the so-called “Hofler’s configuration” that was developed in Hofler’s thesis work in 1986 [5]. Grant [6] followed a passive impedance-matching approach by replacing the long tube with a mass element. The mass element consisted of a disk bonded to a servo-meter, electro-formed, single convolution nickel bellow. However, adjusting the parameters of the mass element was difficult. Passive elements also suffer from the inconvenience as mentioned above, i.e. they cannot adapt to varying gas mixture properties.

There have been many studies of active acoustic-impedance control, in particular for noise control and sound adsorption problems. In the 1950’s Olson and May [7] first proposed the concept of “electronic sound absorber” in which a secondary loudspeaker and an error microphone formed a feedback system to reduce the sound pressure in the vicinity of the microphone. This idea is essentially a pressure-release scheme. Guicking and his colleagues [8, 9] proposed an active impedance matching scheme in the early 80’s. In their approach, the reflected wave in a Kundt tube was obtained using an analogue circuit to separate the incident and reflected acoustic waves using two microphone measurements. Ideal sound absorption requires zero-amplitude reflected waves. A hybrid active/passive sound absorption system was investigated by Beyene and Burdisso [10], combining active impedance control for lower frequencies and passive sound absorption using foam materials for higher frequencies. Numerical and experimental studies were carried out for individual frequencies. Further work for broadband sound absorption was done by Smith et al. [11]. The control scheme was a SISO adaptive feed-forward method using filtered-X LMS (least mean
square) algorithm. Furstoss et al. [12] investigated the optimal impedance-matching problem at the back surface of a porous layer to ensure no reflection at the front surface for both oblique and normal incidences. They studied two control approaches: 1) a direct method; and 2) a hybrid approach that combines both active and passive methods. For the direct impedance control, the impedance was matched at the loudspeaker end. The sound pressure and cone velocity at the controlled speaker was measured. The control design objective was to match the ratio of these two signals, i.e. the effective acoustic impedance, to the desired impedance. The impedance-matching error was expressed as the difference between the sound pressure and the product of the particle velocity and the acoustic impedance. A feed-forward control scheme based on filtered-X LMS algorithm was applied and tested for both normal and oblique incidence of sound waves. For the hybrid method, the sound pressure was minimized at the back surface of the porous layer to control the impedance. An active feed-forward control scheme was implemented for the single-driver-single-microphone case, while the passive-control based attenuation was provided with an absorption layer. A diagonal feedback controller was implemented using analog circuits for a 3-by-3 loudspeaker-array. The coupling within this multi-input-multi-output problem was not considered.

The impedance to be matched in a dual-driver standing wave tube is the impedance of the longer tube at the location where the second driver is to be located. In contrast with the sound absorption problem, the impedance in this problem usually includes a lightly damped mode in the operating frequency range. In order to maintain consistent performance despite parameter variations during the operation of the device, feedback control is preferred over feedforward control. To make the standing wave tube work properly, the requirement for the performance of impedance matching is relatively high. Therefore, the feedback control system for this problem is expected to provide good performance in matching lightly damped mode, but yet needs to be robust. As band-pass actuators, loudspeakers usually have large uncertainties at frequencies below and above the pass band in addition to having non-minimum phase zeros. These constraints need to be considered when designing a high-performance acoustical control system.

In this paper, two impedance matching formulations are proposed for the dual-driver impedance matching problem. One is similar to the direct matching method described by Furstoss et al. [12], i.e. using the impedance matching error, in which only the secondary driver is controlled. This is similar to the so-called master-slave control. The resulting control structure is a single-input-single-output (SISO) system. The other formulation is to directly control the two acoustical variables related to the acoustic impedance, i.e. to use the desired impedance model to generate two reference signals and force the two outputs to follow the reference signals. This approach requires active control of both speakers, and it is realized as a two-input-two-output (TITO) system. Similar to many other standing wave tube applications, thermoacoustic systems are driven using a tonal excitation. Impedance matching for a tonal excitation is equivalent to tracking periodic reference signals. It is well known that repetitive control can achieve good performance for tracking or rejecting periodic signals. Hu [13] demonstrated the effectiveness of using repetitive control for duct noise cancellation. In this study, repetitive control was applied to the two different impedance matching control schemes mentioned in the previous paragraphs.

The robustness of the closed-loop system was addressed by formulating the repetitive controller into an augmented $H_\infty$ framework. The robust performance of the closed-loop systems was verified using a structured singular value analysis.

The remaining of the paper is organized as follows. The two impedance matching control formulations are presented in the next section. Key repetitive control results are summarized in Section III, followed by the discussion of the robust repetitive control design. The experimental setup and the control implementation are described in Section V, together with experimental results.

## 2. Impedance Matching Control for Dual-Driver Standing-Wave Tubes

![Figure 3. Impedance matching for dual-driver standing wave tube](image)

Figure 3 illustrates the problem of using a dual driver short tube to match its acoustic field to that of a longer half wavelength standing wave tube, where $x$ denotes the coordinate along the axial direction of the tube. Let $x_1$ denote the location of loudspeaker #1 (the primary driver) for both the long and the short tube, and $x_2$ denote the location of loudspeaker #2 (the secondary driver) for the short tube. The acoustic impedances at $x_2$, $Z_{2S}(j\omega)$ for the short tube and $Z_{2L}(j\omega)$ for the long tube, are defined by

$$Z_{2S}(j\omega) = \frac{P_{2S}(j\omega)}{V_{1S}(j\omega)} \quad \text{and} \quad Z_{2L}(j\omega) = \frac{P_{2L}(j\omega)}{V_{1L}(j\omega)},$$

where $P_{2S}(j\omega)$ and $P_{2L}(j\omega)$ are the Fourier transforms of the sound pressures at $x_2$ for the short and the long tubes, respectively. Similarly, $V_{2S}(j\omega)$ and $V_{2L}(j\omega)$ are the Fourier transforms of the particle velocities at $x_2$ for the short and the long tube, respectively. The objective for impedance matching is to equate $Z_{2S}$ and $Z_{2L}$, i.e. $Z_{2S}(j\omega) = Z_{2L}(j\omega)$ for $\omega \in \Omega$, where $\Omega$ represents the frequency range where impedance matching is desired. Based on a traveling wave decomposition of the sound field, the distributions of the sound pressure and the particle velocity are given by the following equations [1]:

\begin{align}
P(x) &= A e^{j(k_2 x - \omega t)} + B e^{j(k_3 x + \omega t)} \\
V(x) &= \frac{1}{\rho c} \left[ A e^{j(k_2 x - \omega t)} - B e^{j(k_3 x + \omega t)} \right]
\end{align}

where $A$ and $B$ are the complex amplitudes of outward and inward sound waves, and $L$ is the length of the duct. The mean density, the speed of sound and the wave number are denoted by $\rho_0$, $c$, and $k$, respectively. Since Equations (1a) and (1b) contain two unknowns $A$ and $B$, only two boundary conditions are needed to uniquely define the sound field, i.e. controlling any two variables at any location within the duct is sufficient to controlling the corresponding sound field. This provides some flexibility in the selection of controlled variables based on sensors availability. For the impedance matching problem in Figure 3, apart from the pair $(P_2, V_2)$, the controlled variables can be $(V_1, V_3)$, $(P_1, P_3)$, or $(P_1, V_3)$, as long as the appropriate relationship between the properties is matched. In this study, the pair $(V_1, V_3)$ was chosen as the controlled variables. The choice of velocity measurements is based on trade-off between signal-to-noise ratio (SNR) and model uncertainties. The response of the transfer function from the input voltage to the piston velocity.
output rolls off for higher frequencies, at a much faster rate than the response for the sound pressure. This means that if the sound pressures are chosen as the controlled variables, there will be higher uncertainties in the plant for control design, which lead to higher order plant model.

Figure 4 illustrates the TITO formulation to achieve impedance matching using velocity feedback. To simplify the notation, the Laplace variable $s$ will replace $j\omega$. $G_1(s) = V_1(s)/U_1(s)$ is the desired relationship between $V_{1S}$ and $V_{2S}$. In this formulation, $U_1$ and $U_2$ are the input voltages of drivers 1 and 2, respectively. $V_{1S}$ and $V_{2S}$ are the outputs of the plant model and $V_{1L}$ and $V_{2L}$ are the references for $V_{1S}$ and $V_{2S}$ to follow. $G_{1U1}(s)$ and $G_{1U2}(s)$ are the transfer functions from $U_1$ to the two velocity outputs in the short tube, and $G_{2U1}(s)$ and $G_{2U2}(s)$ are the transfer functions from $U_2$ to the two outputs. The impedance matching problem is then a two-input-two-output reference-tracking problem.

Figure 4. TITO impedance matching control formulation

Another formulation for impedance matching is to use the impedance-matching error as the feedback signal. This approach is similar to the direct method proposed by Furstoss et al. [12]. Figure 5(a) shows the block diagrams for calculating the impedance matching error $E_t(s) = V_{2S}(s) - G_s(s)W(s)$. With simple block diagram manipulation, Figure 5(a) can be simplified as Figure 5(b). In this realization, the control input is $V_{1S}$, i.e. the second loudspeaker in Figure 3. The plant for this control system design is $P(s) = G_{1U1}(s) - G_{2U2}(s)G_s(s)$. Notice that in this realization, loudspeaker #1 is not being controlled, i.e. $U_1$ in Figure 5(c) can be considered as an exogenous disturbance. The transfer function $G_{2U2}(s)$ is viewed as a disturbance filter. The impedance-matching problem is converted into a single-input-single-output disturbance rejection problem.

Since the standing wave tube is driven by tonal excitation, repetitive control can be used to improve system performance. In the following section, a robust repetitive control framework is introduced.

Figure 5. Formulation of impedance matching error

### 3. Robust Repetitive Control

Repetitive control is a special realization of the internal model principle control [14]. It provides an excellent solution for tracking (and/or rejecting) periodic signals (and/or disturbances). The general scheme of continuous-time MIMO repetitive control was given in the classic paper by Hara et al. [15]. The basic structure is illustrated in a typical tracking problem as shown in Figure 6. Disturbance rejection can be formulated in a similar block diagram. In Figure 6, $R(s)$ is the periodic reference input with period $\tau$, $Y(s)$ is the system output, and $E(s)$ is the tracking error. The stable transfer matrix $G(s) \in \mathbb{R}^{pp \times pp}$ is considered as the concatenation of the actual plant $P(s)$ and some forward-loop controller $K(s)$.

Theorem 1: For stable $G(s)$, the repetitive control system in Figure 6 will be stable if

$$\left\|G_1(s) + G_3(s)\right\|_{\psi} < 1$$

where $\left\|\cdot\right\|_{\psi}$ denotes the $L_\psi$-induced norm of the transfer matrix.

Figure 6. General framework of repetitive control

Theorem 1 can be cast into the following control design problem as illustrated in Figure 7: Find a real-rational stable controller $K(s)$, such that $\left\|W_p S\right\|_{\psi} < 1$ with $S = \left(I + PK\right)^{-1}$ the sensitivity function, and $W_p(s) = q(s)$ is the performance weight matrix. In this formulation, the repetitive control design problem is equivalent to a standard $H_\infty$ formulation, in which the $q$-filter acts as the performance weighting. Based on this framework, efforts were made to design a robust controller using the various optimal robust control techniques. Weiss and Hafele [16] provided the theoretical basis for designing MIMO repetitive controller using $H_\infty$ design. In particular, the measurement was divided into two sets: one set was for general feedback control, and the other set for repetitive feedback control.

The effect of plant uncertainty on system stability needs to be considered. The $q$-filter in repetitive control design provides some robustness especially against the plant uncertainty outside the frequency range of interest. To investigate robust performance, Langari and Francis [17] provided a theoretical framework for robust sampled-data repetitive control considering plant uncertainty. Later, Li and Tsao [18] and Li et al. [19] formulated the robust performance repetitive control problem in discrete-time domain. In the continuous-time domain, their framework can be illustrated in Figure 7, where $A_1(s)$ and $W(s)$ denote the model uncertainty and the corresponding
uncertainty weighting, respectively. Normally, $\Delta(s) \in \mathcal{RH}_\infty$ and $\|\Delta(s)\|_{\infty} \leq 1$.

The filter $q(s)$ is the performance weight in the equivalent $H^\infty$ robust performance design problem. Note that the criterion for choosing performance weight in a general $H^\infty$ design is different from that for choosing $q$-filter in a repetitive control design. Designing a performance weight for a typical $H^\infty$ control design requires consideration of the magnitude response only, while designing a $q$-filter for repetitive control requires the consideration of both magnitude and phase responses. The frequency response of $q(s)$ needs to be unity for the frequency band within which good performance is expected. For the acoustic systems, there are factors that limit the bandwidth of the $q$-filter, such as non-minimum phase zeros and plant uncertainties at both low and high frequencies. With limited bandwidth, the $q$-filter inevitably has a phase distortion that moves the low-frequency poles of the signal generator in the repetitive block away from the imaginary axis. Therefore, phase compensation is necessary. Phase compensation can be made by adding or reducing the time delay (or advance) that is in the prototype repetitive controller, which corresponds to adjusting the phase response of the $q$-filter. If we denote $\phi_g$ the phase response of $q(s)$ in degree, the time delay (or advance) to compensate is $\zeta = \phi_g / (2f_0)$, where $f_0$ is the operating frequency. In practice, the controller is required to work continuously across a range of frequencies. For example in thermoacoustic applications, adaptive frequency tuning was used to lock the driving frequency to obtain the best driver efficiency [24, 25]. It is preferable to have one controller working within a given frequency band with guaranteed stability and performance. A repetitive controller meets the requirement. For different driving frequencies, the parameters to adjust are the period and phase in the delay block.

The $H^\infty$ controller design will result in a full order controller, i.e. the controller has the same order as the generalized plant. The order of the generalized plant is the sum of the orders of the plant and the related weighting functions. The acoustic system is a distributed-parameter system with a number of lightly damped modes, and a finite-dimensional model is often used as the approximation within the bandwidth of interest. This tends to result in high order plant model. Due to the lightly damped modes at high frequencies, and the uncertainty at low frequencies, the order of the uncertainty weight is also relatively high. All these factors result in a high order generalized plant. Furthermore, the system order increases rapidly as the dimensions of the inputs and outputs of the system increase. A generalized plant with very high order leads to numerical difficulty in optimal controller design, and a high order controller is difficult and expensive to implement. Model reduction may be required for both the plant model and the controller. Since model reduction implies the removal of the system information, the procedure must always be applied rather conservatively. Stability and performance are normally checked after performing model reduction and controller design, to verify that the reduced-order controller is good enough.

A standard approach to model/controller reduction is through balance realization [21, 22]. However, this approach tends to be poorly conditioned for non-minimal models, and failed in model reduction for the plant and controller in TITO impedance matching.
control problem. The Schur Method proposed by Safanov and Chiang [23] provided an alternative. In this method, a non-balanced state-space realization of the Moore’s reduced model can be computed directly without balancing, via projections defined in terms of arbitrary bases for the left and right eigenspaces associated with the large eigenvalues of the product of the controllability and observability Gramians.

5. Experimental Results and Discussion

An experimental dual-driver standing wave tube was built using a 48cm (18.9") long 15cm (6")-diameter PVC tube. Two 15cm (6")-diameter paper-diaphragm loudspeakers were installed at both ends of the tube. The speakers were driven by a JBL-6230 dual-channel power amplifier. The target impedance that was to be matched is that of a 1.28m long tube, which is a half-wavelength standing wave tube operating at 123 Hz at ambient pressure. Two PCB-353-B17 accelerometers were installed on the diaphragms of the loudspeakers. The accelerometers outputs were fed to PCB-480D06 power units for amplification. Velocity information is obtained by integrating the acceleration signals, after filtered through 10 Hz high-pass filters. The accelerometer outputs were fed to PCB-480D06 power units for amplification. Velocity information is obtained by integrating the acceleration signals, after filtered through 10 Hz high-pass filters.

The desired impedance that is to be matched, $G_d(s) = V_1/V_s$, was calculated based on Kinsler et al. [1]. Good impedance matching in the frequency range of 100 ~ 200Hz is needed. A lower order approximation of the desired impedance up to 200Hz is

$$G_d(s) = \frac{6.615 \times 10^4 (s^2 + 83.91s + 1.956 \times 10^6)}{(s^2 + 36.66s + 694100)(s^2 + 334.4s + 2795000)}$$

Theoretically, the damping ratio for the mode at 123 Hz is 0.007. Actual measurements showed that the damping ratio for the experimental system at this mode is 0.02. As to be shown in later discussion, tracking a mode with a small damping, e.g. 0.02 in this case, is a challenging objective for controller design. The $q$-filter for the repetitive controller was selected as

$$q(s) = \frac{1210000 (s^2 + 16.34s + 1580000)}{(s^2 + 402.12s + 252700)(s^2 + 1105.8s + 1910800)}$$

Its frequency response has a flat magnitude response between 100 ~ 200 Hz, but the phase change of the $q$-filter is substantial.

The TITO impedance matching formulation was first studied. An $H^\infty$ controller was designed via the Riccati method. The generalized plant model was first reduced to $99^{th}$ order. After controller design, the Schur method was used to reduce the 2-by-2 controller to $10^{th}$ order. The structured singular value, $\mu$, for the closed-loop system value was computed using the original high-order plant model with the reduced-order controller. The maximum value $\mu$ is about 0.9. The impedance matching results between 90~210 Hz are shown in Figure 8. The controlled impedance (solid) matched the desired impedance (dot) between 95 and 200 Hz. The uncontrolled impedance is plotted in the dash-dotted line. The plots of magnitude responses from $V_{R,lo}^R$, the reference input for $V_1$, to the control inputs $U_1$ and $U_2$ are shown in Figure 9(a). Notice that $U_2$ has a significant peak around the resonance frequency indicating that more power was used to achieve the desired resonant impedance that does not exist in the short tube. In addition to matching the desired impedance, the TITO approach has the additional control authority to achieve one additional design objective. In this experiment, maintaining the particle velocity at $x_1$ at a constant level is chosen to be the second design objective. Figure 9(b) shows the plots of magnitude responses from $V_{R,lo}^R$ to the two velocity outputs $V_1$ and $V_2$. It can be seen that the velocity of driver 1 is kept relatively constant, while the velocity of driver 2 changes significantly around the resonant frequency range.

![Figure 8. Impedance matching result using the TITO formulation](image)

![Figure 9. Control inputs and velocity outputs of the drivers (TITO)](image)

![Figure 10. Impedance matching result using the SISO formulation](image)

![Figure 11. Control input and velocity outputs of the drivers (SISO)](image)
matching performance is shown in Figure 10. The result seems to be slightly better than that of the TITO approach. The main reason maybe that the plant of the SISO structure allows a tighter performance weighting, i.e. wider bandwidth for the $q$-filter. The SISO scheme directly controls the impedance matching error while the TITO structure is more of an indirect approach. The SISO scheme does not directly control the magnitude of the individual outputs. Therefore, the magnitude of the individual outputs are more affected by the reference input $U_r$. The magnitude response of the control input $U_2$ to the reference input $U_1$ is plotted in Figure 11(a), and those of the two velocities to the reference $U_1$ are plotted in Figure 11(b). Comparing Figure 11(b) and Figure 9(b) shows that the control over the cone velocity of certain driver is not available in the SISO formulation.

Light damped modes in the acoustic impedance implies that the velocity of the secondary driver needs to be large around the resonance frequency, and so is the required control effort. The experimental results agree with this fact. Since the cone velocity of the secondary driver needs to match the velocity at the same location for the longer tube, the position of the secondary driver will determines the of velocity magnitude for the secondary driver. Actuator saturation will be a problem for large amplitude operation. For a half wavelength standing wave tube, the secondary driver can be located as close to the pressure antinode as possible. In this sense, a smaller length of the short tube can somehow relieve the demand for the loudspeakers with large admissible displacement.

5. Conclusions & Discussion

Two control formulations for acoustic impedance matching were proposed to make a dual-driver short tube recover the acoustic characteristics of a long standing wave tube. The TITO formulation achieves impedance matching indirectly by controlling two variables in the acoustic field. The SISO formulation directly minimizes the impedance matching error. Since the application is under tonal excitation, repetitive control was formulated in to an $H^\infty$ framework for both structures and shown to be able to generate controllers that achieve robust performance under plant uncertainties. Phase compensation for the $q$-filter in the internal model was employed to further improve matching performance. The repetitive controller design was converted to an equivalent robust performance design with the $q$-filter as the performance weight. Experimental results showed that both formulations could achieve the desired level of impedance matching. Due to the high-order and non-minimum phase nature of the acoustic system, different methods of model reduction were employed in model and controller reduction. This paper provides two active-control frameworks for impedance matching in one-dimensional ducts, especially for the situations with tonal excitation. The proposed control scheme can emulate different impedance, for example different length of standing wave tube, with one setup. This would also be attractive for conducting a parametric study of the effects of tube geometry and stack placement. It can also be applied to other situations, like sound absorption problems.

Compared with the performance of impedance matching achieved by passive techniques, the proposed active approach has significantly better performance. In Grant’s work [5], the $Q$-factor achieved from the passive element based resonator was 3 times lower than the desired impedance, which is inferior to what has been achieved in this study. In addition, nonlinear behavior of the mass element introduced spurious rocking mode. The main drawbacks of the active scheme are increased cost and actuator saturation. The additional driver along with the control hardware demands more investment. Impedance matching requires relatively high cone velocity at the secondary driver, especially for thermoacoustic applications where large amplitude sound needs to be generated. As such driver is difficult to build for current technology, active control scheme is not a practical solution for thermoacoustic cooling.

References