

A Note on the Applicability of Thermo-Acoustic Engines for Automotive Waste Heat Recovery

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ABSTRACT

A thermo-acoustic engine is a device converting thermal energy into high amplitude acoustic waves that can be harvested, for example, to obtain electricity. The core of the device is a stack/regenerator along which a temperature gradient is created using one hot and one cold heat exchanger. Correctly designed, the thermal interaction between the working fluid and the regenerator assists in amplifying incident acoustic waves. Previous studies have indicated good efficiency obtained with a system of low geometrical complexity. However, for the practical application of this technique it is vital to understand and identify critical design parameters and operating conditions. This is of special interest in automotive applications where the operating conditions vary significantly over a drive cycle. This work aims at providing a framework for studying the net power generation over a drive cycle. First, an engineering non-linear model for the efficiency of the thermo-acoustic engine is established. It is based on low-order acoustic networks that have the advantage of being computationally effective and allows for individual optimization of components. This model is then used to loop over a drive cycle of a typical commercial vehicle, while also accounting for pump losses given by the heat exchange processes. Although an engine not optimized for the present problem was used, promising performance, with a thermal efficiency of 7%, was found. Also the importance of adapting the engine to the varying operating conditions over the drive cycle was illustrated.

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INTRODUCTION

Waste heat recovery has been identified as one of the promising techniques to further reduce the fuel consumption of vehicles equipped with an internal combustion (IC) engine [1]. For example, in the Super Truck program [2], all participants have investigated this route. The most common technique is Rankine cycle machinery; good efficiency has been proven but the equipment is bulky and complex. An alternative approach is the use of thermo-electric generators. They have the advantage of not having any moving parts; however, at present their practical use is limited by the low efficiency of the materials available commercially.

A third alternative sporting a combination of high efficiency and low mechanical complexity is thermo-acoustic (TA) engines. In a thermoacoustic engine, heat is converted to high intensity sound waves that can be harvested effectively into electricity. While the phenomenon can be traced back to the singing tubes of Sondhaus or the often used laboratory example of the Rijke tube (all described by

Rayleigh [3]), the more fundamental understanding of the theory was first derived by Rott in a series of papers [4]. The core of the engine is a stack/regenerator along which there is a temperature gradient. The temperature gradient is created using a hot and a cold heat exchanger on respective sides. Properly designed, the stack/regenerator acts as an amplifier for the incident sound (see theory section). The engine could be driven in different modes (different phase relations between pressure and particle velocity), most commonly either as standing wave or travelling wave. Travelling wave machines are the most appealing since it makes the best use of the heat and-as pointed out by Ceperley [5] - the process then resembles a Stirling engine where the piston is replaced by the acoustic wave.

From these first ideas a massive contribution towards both the theory and the practical implementation came from Swift and coworkers, for example, see [6]. They demonstrated an engine with a thermal efficiency of 0.30 (or 0.41 Carnot efficiency). Published results [7] from this engine will be used to validate the model developed in this work.

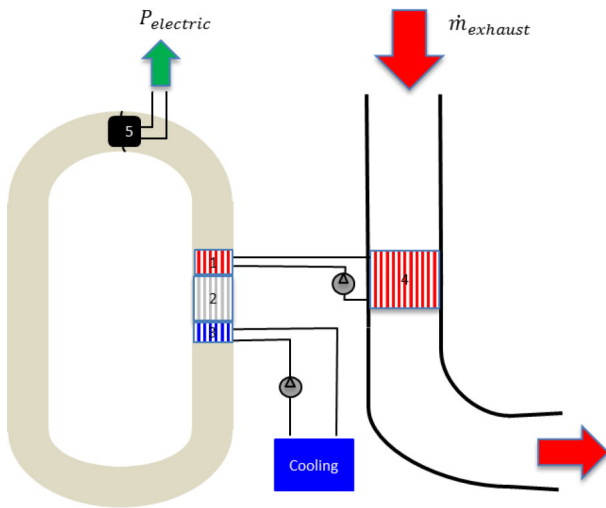


Figure 1. Schematic of the TA engine installation. Exhaust pipe downstream of the exhaust after-treatment system. 1. Hot side heat exchanger, 2. Regenerator, 3. Cold side heat exchanger, 4. Exhaust heat exchanger, 5. Generator (harvesting unit).

The purpose of this work is to provide an engineering tool to study the operation of a thermo-acoustic engine over an automotive drive cycle. Most previous studies are aiming at stationary applications, but to be useful in an automotive application it has to cover a wide range of inlet temperatures on the hot side. However, the results in the studies by Keolian [8] and Fritzsche et al [9] are promising and encourage us to evaluate this technique. In Keolian [8], a setup for a heavy duty truck was studied. The final design was a closed loop containing two cores and two generator units. The system had the envelope of $0.86 \times 0.71 \times 0.64$ m and weighed 163 kg. Rather large but still possible to fit on the truck frame. The system was designed to deliver an average of 4 kW over a long haul cycle. Fritzsche et al. [9] discusses an installation that fits in the middle tunnel of a passenger car. Over the NEDC and WLTC, they simulate a reduction of 2-4 and 3-5 g CO₂/km respectively.

As will be shown later, we take on a slightly different approach to modeling where we represent the engine as a low-order network in the frequency domain. This is in line with recent experimental [10] and numerical [11] work. In the linear regime, this yields a good opportunity for studying the potential of the system and performing various parameter sweeps. However, it will not yield the saturation amplitude of the engine that is determined by non-linear loss mechanisms. Our contribution is to extend the model to include these losses and hence being able to actually predict the power produced.

First, the basic theory is introduced and then the modeling framework is discussed. The model of the engine is then validated against the experimental data of Backhaus and coworkers [7]. Finally, this engine is run over a generic long haul drive cycle for a commercial vehicle where the losses associated with the heat exchange processes are subtracted from the power produced by the thermo-acoustic engine to produce a net gain.

THEORY AND MODELING

Basic Thermo-Acoustics

In ducts with no mean flow but a finite temperature gradient the basic 1D (plane wave) thermo-acoustic equations are:

$$\frac{dp}{dx} = \frac{i\omega\rho_m}{(1-f_v)A} U \quad (1)$$

$$\frac{dU}{dx} = -\frac{i\omega A}{\gamma p_m} \left(1 + \frac{\gamma-1}{1+\epsilon_x} f_k\right) p + \beta \underbrace{\frac{f_k-f_v}{(1-f_v)(1-\sigma)(1+\epsilon_x)} \frac{dT_m}{dx}}_G U, \quad (2)$$

where it is clear that the thermo-acoustic gain factor G depends on the temperature gradient. Rewriting Equation 1 using the classic electroacoustic analogy:

$$\frac{dp}{dx} = -(r_v + i\omega l)U, \quad (3)$$

where r_v is the viscous resistance and l the inductance. They simply correspond to the real and imaginary part of $-\frac{i\omega\rho_m}{(1-f_v)A}$. The non-linear losses in the regenerator and heat exchangers are introduced adding a “turbulent” term to the viscous resistance:

$$r_{v,turbulent} = \frac{64\rho_m|U|}{3\pi^3 D^5} \left[f_M - \left(1 - \frac{9\pi}{32} |N_{R,1}| \frac{df_M}{dN_{R,1}}\right) \right] \quad (4)$$

where the friction factor f_M and its derivative with respect to the peak Reynolds number $N_{R,1}$ can be found in any standard handbook.

Modeling Approach

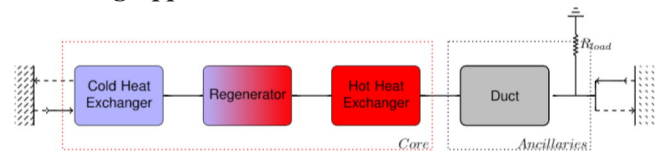


Figure 2. Model to represent the thermo-acoustic setup

The thermo-acoustic engine is represented by a low-order acoustic network, as illustrated in Figure 2. The benefit of using low-order linear acoustic networks is that each component could be determined individually. That is, one could for example make experimental characterization or in depth studies of the core itself without considering the rest of the system. Since the elements in this example are coupled in series the transfer matrix representation is chosen; each component is then of the form:

$$\begin{bmatrix} p_d \\ U_d \end{bmatrix} = \mathbf{T} \begin{bmatrix} p_u \\ U_u \end{bmatrix} \quad (5)$$

Here the subscripts d and u denote the downstream and upstream side of a block. The core of the engine (heat exchangers and the regenerator) yields the amplification. The basic thermo-acoustic equations (Eqs. 1-2) express the derivatives of the pressure and volume velocity along

to arrive at the transfer matrices we solve for the eigenvalues of the differential matrix. Once the core is known different types of engines (e.g. travelling or standing wave machines) can be represented by simply setting the phase of the state variables upstream.

The remaining duct work of the engine (whose only purpose is to provide the desired phase at the core) is represented as a lumped loss. This might seem abrupt, but one lesson learned from previous work [7, 10] is that the losses in the process are complex and difficult to model in detail anyway. We assume a first order non-linearity where the loss of acoustic energy in to the harmonics is neglected. It then takes the form:

$$\begin{bmatrix} p_d \\ U_d \end{bmatrix} = \begin{bmatrix} 1 & -(k_1 + k_2 U) \\ 0 & 1 \end{bmatrix} \begin{bmatrix} p_u \\ U_u \end{bmatrix} \quad (6)$$

where one needs either detailed models or an experimental data set to obtain the coefficients k_1 and k_2 . Here the data from Backhaus et al [7] is used. If more detailed models are available / necessary, it is easily implemented in the linear acoustic network.

Finally there is an electricity extraction element of the same type. Here the electric resistance has to be found (when tuning the model). This is the parameter one can change to vary the operating conditions of the engine at stationary use. Or alternatively, it could be changed to adapt the TA engine to varying input conditions.

Apart from losses in the system, the actual amplification over the regenerator depends on the temperature profile. Ideally one has a linear temperature gradient, but that is of course seldom achieved in reality. Again the real temperature gradient then has to be modeled or experimentally determined.

Finally knowing the gain and losses in the different component, one can calculate the heat input necessary to drive the system. Depending on the configuration, this could be a restriction; if the heat exchanger from the exhaust cannot deliver the heat required it would of course limit the output of the engine. One can also calculate the cost of delivering the required heat.

Acoustic Amplification Potential

The phasing of the acoustic waves is most important. Following the procedure derived by Auréan [12], the optimum input states for a given engine can be found. Here it is easier to work with the scattering matrix, $\mathbf{S}$, formulation of the acoustic components. That is, the studied block is represented in terms of traveling wave amplitudes-of some state variable x -in the positive and negative direction on each side of it:

$$\begin{bmatrix} \hat{x}_u^- \\ \hat{x}_d^+ \end{bmatrix} = \mathbf{S} \begin{bmatrix} \hat{x}_u^+ \\ \hat{x}_d^- \end{bmatrix} \quad (7)$$

Since the aim is to maximize the acoustic power, x is chosen so that $\langle P_{inc} \rangle = \mathbf{a}^* \mathbf{a}$ and $\langle P_{refl} \rangle = \mathbf{b}^* \mathbf{b}$. This could be expressed in terms of the travelling pressure wave amplitudes (for convenience) using first the convention derived by Holzinger [8],

$$[\hat{p}_i^j] = \left[\frac{p_i^j}{\rho c} \right]. \quad (8)$$

Here, ρ denotes the density of the working fluid and c denotes the speed of sound. And, then introducing the scaling factor $\mathbf{V}$:

$$\mathbf{V} = \begin{bmatrix} \sqrt{\rho_u c_u \frac{A_u}{2}} & 0 \\ 0 & \sqrt{\rho_d c_d \frac{A_d}{2}} \end{bmatrix} \quad (9)$$

$$\underbrace{\mathbf{V} \begin{bmatrix} \hat{p}_u^- \\ \hat{p}_d^+ \end{bmatrix}}_{\mathbf{b}} = \underbrace{(\mathbf{V} \mathbf{S} \mathbf{V}^{-1})}_{\mathbf{S}_p} \underbrace{\mathbf{V} \begin{bmatrix} \hat{p}_u^+ \\ \hat{p}_d^- \end{bmatrix}}_{\mathbf{a}} \quad (10)$$

The generated (or lost) acoustic power is now simply given by:

$$\langle P_{gen} \rangle = \mathbf{b}^* \mathbf{b} - \mathbf{a}^* \mathbf{a} = \mathbf{a}^* (\mathbf{S}_p^* \mathbf{S}_p) \mathbf{a} - \mathbf{a}^* \mathbf{a} \quad (11)$$

This equation is positive for a system that generates sound. Now, normalizing the incident acoustic power to unity-that is, setting $\mathbf{a}^* \mathbf{a} = 1$, then the reflected acoustic power is given by the value of the first part of Eq. (11). This quadratic form is symmetric and positive definite, and thus it can be transformed into a diagonal form:

$$\mathbf{a}^* (\mathbf{S}_p^* \mathbf{S}_p) \mathbf{a} = \sum_n \lambda_n |a'_n|^2 \quad (12)$$

where λ_n are the real and positive eigenvalues of $\mathbf{S}_p^* \mathbf{S}_p$ and the vector $\mathbf{a}'$ is given by $\mathbf{a} = \mathbf{T} \mathbf{a}'$. The transformation matrix $\mathbf{T}$ is a unitary matrix where the columns represent the eigenvectors of the Hermitian matrix $\mathbf{S}_p^* \mathbf{S}_p$. Now by studying the eigenvalues and eigenvector the optimal phasing and amplitude for the component is found.

Saturation Pressure

To find the saturation pressure at a given setup, the following procedure is used. Initially a low amplitude of the incoming states $\hat{p}_u^+$ and $\hat{p}_d^-$ is chosen. From Eq. (7), the outgoing states $\hat{p}_u^-$ and $\hat{p}_d^+$ are determined. The upstream side states are combined to generate the cold side pressure p_{CHX} and volume flow U_{CHX} . That is, we go from scattering matrix (used in component view) to transfer matrix (used in system view). Using these, the pressure and volume flow at each point in the system is defined, as the corresponding transfer matrices defining each element are known. The obtained transfer matrix of the system is then again converted to a scattering matrix and the eigenvalue and vectors for maximization of $\mathbf{S}_p^* \mathbf{S}_p$ are evaluated.