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Experimental and theoretical analysis of a small scale thermoacoustic cooler driven by two sources

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Being concern by scaling down thermoacoustic coolers to provide practical solutions for thermal heat management, especially in microcircuits, a current architecture has been proposed recently. A non resonant small cavity fitting the stack dimensions is driven by two loudspeakers coupled through the stack. One of them creates the acoustic pressure field inside the stack while the other one creates the particle velocity field. This cooler has both advantages of being compact and flexible, as the acoustic field in the stack can be controlled to access the optimal field which optimizes thermoacoustic effects. Moreover, the working frequency is not related to resonance conditions, therefore either a quasi-isothermal stack (regenerator) or a quasi-adiabatic stack can be used. Experimental results, which validate theoretical ones, are presented to illustrate the thermal behaviour of a stack and a regenerator in this device. The performances compared with those of classical devices having equivalent stack (standing wave or coaxial devices) show the potentiality of this compact thermoacoustic cooler.

1 Introduction

The paper aims at providing results obtained from both an analytical model presented in a previous paper [1] and an experimental setup installed for the purpose, in order to describe the advantages of new thermoacoustic refrigerators which involve compactness, having in mind to propose devices which could be of practical interest.

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Being concerned herein by the efficiency of these kind of devices, several experimental results, yet typical of applications, are presented below and compared with both some analytical results obtained from the model mentioned above and results on classical devices having equivalent stack (standing wave devices or travelling wave coaxial devices) available in the literature, showing the potentiality of this compact thermoacoustic cooler and its interest from a practical point of view.

2 Experimental setup

A schematic drawing of the compact thermoacoustic cooler under test is shown in Fig. 1. It consists of a small cylindrical cavity of length L = 10.2 cm and of internal diameter d = 9.4 cm. A thermoacoustic core (stack or regenerator, without any heat exchangers) is set in the cavity. Its length is ℓ_{st} = 4 cm and its diameter is d_{st} = 4 cm. The thermoacoustic core is surrounded by a 0.7 cm thick peripheral channel. The acoustic field in the thermoacoustic core is controlled by two electrodynamic loudspeakers. The loudspeaker 1 (Visaton SC 8N), placed inside the cavity, creates the particle velocity field in the stack, while the loudspeaker 2 (PHL audio 1590), placed at one extremity of the cavity, creates the acoustic pressure field in the stack. The gas filling the cavity is air at atmospheric pressure and room temperature.

The experiments are conducted with either a quasi-adiabatic stack or a near-isothermal regenerator. The quasi-adiabatic stack consists of a ceramic porous material with 600 cells per square inch and a porosity of 80 %. Each channel has a cross-section of 600×600 μm^2. The regenerator is composed of stainless steel meshes with a hydraulic radius r_h = 183 μm and exhibits a porosity of 85 %. The working frequencies are chosen in such a way that the ratio between the hydraulic radius r_h and the thermal boundary layer δ_h is around 3 for the stack (i.e. f = 200 Hz) and around 0.5 for the regenerator (i.e. f = 50 Hz). These working frequencies are such that the wavelength is much greater than the dimensions of the compact thermoacoustic cavity.

The instrumentation of the device is schematically presented in Fig. 1. A microphone flush mounted at 1.2 cm from the loudspeaker 2 measures the acoustic pressure in the cavity. A laser vibrometer gives access to the membrane velocity of the loudspeaker 1. To that purpose, a hole is drilled in the magnetic motor of the loudspeaker to allow laser beam to access to the rear of the membrane. The amplitude p of the acoustic pressure, the amplitude u of the acoustic velocity and the phase difference between both (pressure and velocity) $\varphi = \varphi_u - \varphi_p$ in the stack can be deduced from the measured acoustic pressure in the cavity and the measured membrane velocity of the speaker 1, by using an equivalent electroacoustic model [1] or by modeling the device with DeltaEC (which is a specific numeric code to design thermoacoustic devices [2]). The temperature difference $\Delta T = T_2 - T_1$ between the stack ends is measured using two T-type thermocouples.

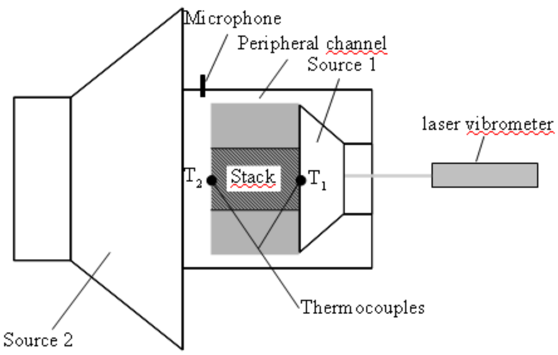

Figure 1: Schematic drawing of the compact thermoacoustic device.

3 Experimental and theoretical results

In this section, the influence of the three acoustic parameters (the acoustic pressure amplitude p, the particle velocity amplitude u and the relative phase $\varphi = \varphi_u - \varphi_p$) on the compact system performance is presented. The cases of both a stack and a regenerator are considered and the study is limited to the temperature difference between the stack ends as the small cavity has no heat exchanger. The effect of each of the three acoustic parameters is studied independently by fixing the two others parameters at their theoretical optimal value. These theoretical optimal values are given in Table 1 for both the quasi adiabatic stack and the isothermal regenerator. These values have been analytically obtained in a previous paper [3]. The optimal values of particle velocity amplitude and relative phase depend on the frequency, on the shape and the dimensions of the stack/regenerator, and on the thermo-physical properties of the fluid and the stack/regenerator. Besides, it has been shown that the optimal value of the acoustic pressure is the maximum pressure level which can be reached in the cavity. In the present setup, the maximum pressure level p = 1000 Pa corresponds to the theoretical maximum linear excursion of the membrane of the loudspeaker 2 controlling the acoustic pressure in the cavity.

Table 1: Theoretical values of the optimal acoustic field in the stack and theoretical temperature difference.

Quasi adiabatic stack	Isothermal regenerator
$p = 1000$ pa	$p = 1000$ pa
$u_{opt} = 1.43$ m.s^{-1}	$u_{opt} = 1.26$ m.s^{-1}
$\varphi_{opt} = 3\pi/4$ rad	$\varphi_{opt} = 2.9$ rad
$\Delta T_{max,th} = 15.8$ K	$\Delta T_{max,th} = 16.6$ K

3.1 Device with quasi adiabatic stack

Figure 2.a shows the evolution of the temperature difference ΔT normalized by its maximum value ΔT_{max} as a function of the acoustic pressure p when $u = u_{opt}$ and $\varphi = \varphi_{opt}$. The experimental results obtained (crosses) are compared with the theoretical predictions (solid line) given by the linear steady state theory [4]. As predicted by the linear theory, the measured temperature difference ΔT increases linearly with the acoustic pressure. However, for an acoustic pressure of 1000 Pa, the temperature difference at the stack ends reaches $\Delta T_{max,exp}$ =10.3 K, which is smaller than the theoretical one $\Delta T_{max,th}$ = 15.8 K. This discrepancy arises from neglecting, in the linear theory [5], all complex thermal mechanisms.

Figure 2.b shows the evolution of the temperature difference ΔT normalized by its maximum value ΔT_{max} as a function of the velocity amplitude u (when $p = p_{max}$ and $\varphi = \varphi_{opt}$). A good agreement is obtained between the theoretical predictions (solid line) and the experimental results (crosses). For both cases, the value of the optimal velocity is $u_{opt} = 1.4$ m.s^{-1}.

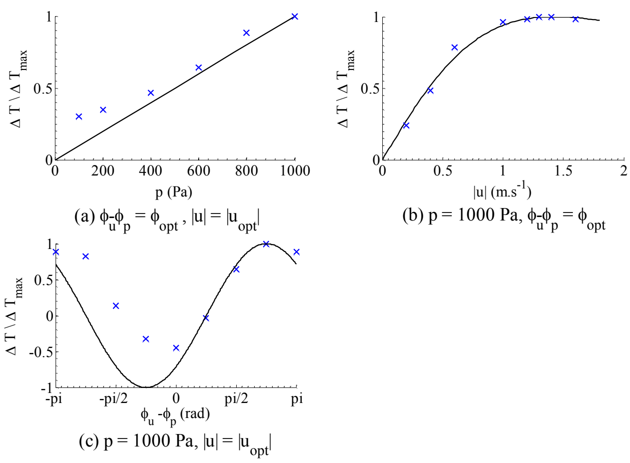

Figure 2: Normalized temperature difference $\Delta T/\Delta T_{max}$ between the stack ends measured (×) and calculated (solid line) as a function of (a) the acoustic pressure p, (b) the particle velocity amplitude u and (c) the phase φ between the particle velocity and acoustic pressure.

The normalized temperature difference $\Delta T/\Delta T_{max}$ versus the relative phase φ (when $p = p_{max}$ and $u = u_{opt}$) is represented in Figure 2.c. When the phase φ varies between ($-3\pi/4$) and ($\pi/4$), the temperature difference $\Delta T/\Delta T_{max}$ is positive and the cold-side stack end is near the loudspeaker 2 controlling the velocity. Whereas for a value of the phase φ between ($\pi/4$) and ($5\pi/4$), the temperature difference is negative and the cold-side stack end is located near the loudspeaker 1 controlling the pressure. Thus the cold-side stack end location can be controlled by the phase φ. This is important to note as this adds the possibility to choose the position of the cold heat exchanger in the system. From the experimental results presented in Figure 2.c, it can be noticed that there is an optimal phase $\varphi_{opt,exp} = 3\pi/4$ rad which corresponds to the theoretical optimal phase. However, the evolution of the experimental normalized temperature difference does not fit completely the theoretical one. This difference is due to the heating of the loudspeaker voice-coil controlling the velocity. This heating is added to the thermoacoustic heat flux and leads to an increase of the stack end temperature near the loudspeaker 1.

The behavior of the small cavity cooler is now theoretically compared with the behavior of a standing wave cooler. The standing wave cooler considered for the comparison consists of a half wavelength resonator driven by an acoustic source. The fluid filling the resonator is air at atmospheric pressure and room temperature. The source is chosen to be the same electrodynamic loudspeaker which controls the acoustic pressure field in the small cavity cooler. The resonator length L_s = 0.65 m is adjusted in such a way that the resonance frequency of the system is the working frequency of the compact device, i.e. f=200 Hz. The same stack is used for both the compact device and the standing wave cooler.

To provide a fair comparison of the two devices, the best performance in terms of temperature difference of each one is compared when the electric power provided to the sources is given (P_{el} = 7.7 W). In the case of the compact device, P_{el} represents the total electric power provided to the two loudspeakers. The maximal temperature difference in the small cavity is obtained when the acoustic field is the optimal one (given in table 1). In the standing wave cooler, the maximal temperature difference is obtained when the stack is set at its better location along the resonator for which the temperature difference is maximal (i.e. 0.55 cm away from the loudspeaker in our case). The acoustic fields which lead to the maximal temperature difference as well as the theoretical performances are indicated for both systems in the Table 2.

Table 2: Theoretical comparison between the behaviour of a small cavity cooler and the behaviour of a standing wave cooler.

Small cavity cooler	Standing wave cooler
Acoustic field in the stack	
$p = 1000$ pa	$p = 1335$ pa
$u_{opt} = 1.43$ m.s^{-1}	$u_s = 1.27$ m.s^{-1}
$\varphi_{opt} = 3\pi/4$ rad	$\varphi_s = \pi$ rad
Performances	
$\Delta T_{max} = 15.8$ K	$\Delta T_s = 13.8$ K
$Q_{max} = 0.17$ W	$Q_s = 0.15$ W

P_{el} = 7.7 W	$P_{el,s}$ = 7.7 W
$\eta_{h/el}$ = 2.14%	$\eta_{h/el,s}$ = 1.88%

As indicated in Table 2, the temperature difference in the compact system is greater than the one in the standing wave cooler. Indeed, like already mentioned [3], the operating point in a resonant system imposes a compromise between acoustic pressure and particle velocity in the stack. Hence, even if the acoustic pressure is higher in the resonant system, the velocity amplitude and the relative phase are not the optimal ones for the temperature difference.

The global efficiency $\eta_{h/el}$, defined as the ratio between the thermoacoustic heat flux Q and the electric power P_{el}, is higher for the small cavity despite the use of two sources. Therefore, the small cavity cooler operating with a stack has both advantages of being compact and being more efficient than a classical standing wave thermoacoustic cooler.

3.2 Device with isothermal regenerator

Figure 3.a shows the temperature difference ΔT versus the acoustic pressure when $\varphi = \varphi_{opt}$ and u = 0.8 m.s^{-1}. Note that the amplitude of the particle velocity is not set to its optimal value u_{opt} = 1.26 m.s^{-1}. This optimal value cannot be reached experimentally because the maximum linear excursion of the velocity source membrane limits the experimental velocity to u = 0.8 m.s^{-1}. However, the conclusions would not change since the optimal phase and the optimal acoustic pressure are independent of the value of the particle velocity u. As expected, both the measured and predicted temperature differences increase with the acoustic pressure. Note that the theoretical values of ΔT_{th} are calculated by adjusting the regenerator thermal conductivity in order to fit the experimental temperature difference value ΔT_{exp} = 14.1 K measured when p = 1000 Pa. Indeed, the effective thermal conductivity of the regenerator is lower than the thermal conductivity of the regenerator material due to the poor contact between adjacent screens in the regenerator. This reduction is usually taken into account in models by an empirical degradation factor between 0.1 and 0.2 [6]. In our case, this factor is set to 0.19.

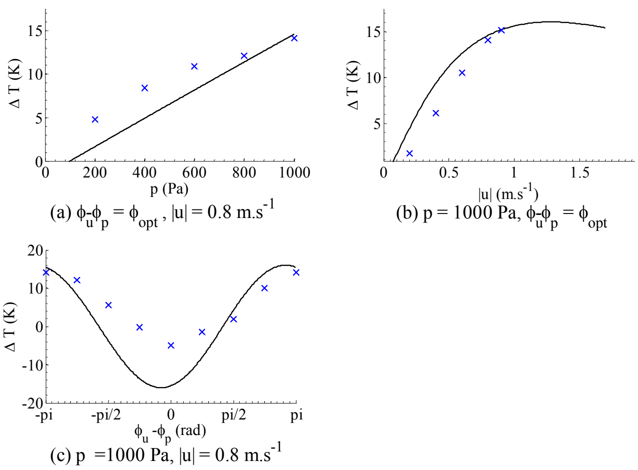

Figure 3: Temperature difference ΔT between the regenerator extremities measured (×) and calculated (solid line) as a function of (a) the acoustic pressure p, (b) the particle velocity amplitude u, (c) the phase φ between the particle velocity and acoustic pressure.

Figure 3.b represents the temperature difference ΔT as a function of the velocity amplitude u (when $p = p_{max}$ and $\varphi = \varphi_{opt}$). The theoretical curve (solid line) exhibits the same trend than experimental one (crosses), even if the optimal value of the particle velocity u_{opt} = 1.26 m.s^{-1} cannot be reached. The experimental determination of the optimal velocity amplitude would required changing the velocity loudspeaker or to design a new setup with appropriate geometrical characteristics.

Finally, Figure 3.c represents the influence of the relative phase φ on the temperature difference when u = 0.8 m.s^{-1} and $p = p_{max}$. The presence of an experimental optimal value for the relative phase φ is highlighted. This value corresponds to the theoretical one φ_{opt} = 2.9 rad.

In the following, the performance of the small cavity cooler working with a regenerator is compared theoretically with the performance of a travelling wave thermoacoustic refrigerator with a coaxial configuration. This refrigerator is composed of a linear motor coupled to a resonator in which a regenerator unit is placed. The regenerator unit section is smaller than the resonator section to form an acoustic feedback path around the regenerator. Hence, this geometry allows to create a local travelling wave phasing and to control the impedance in the regenerator by playing on the dimension of the regenerator unit and on the position of the unit in the resonator [7].

The fluid is air at atmospheric pressure. The source characteristic is the same as those of the electrodynamic loudspeaker which controls the acoustic pressure field in the small cavity cooler. The resonator length L_t = 3.39 m is adjusted in such a way that the resonance frequency of the system is the working frequency of the small cavity i.e. f=50 Hz. The thermo-physical and geometrical characteristics of the regenerator are those of the small cavity.

Here, the temperature difference in the travelling wave device is 20% greater than the one obtained in the compact device. Nevertheless, small cavity cooler remains interesting when compactness is needed.

Table 3: Theoretical comparison between the behaviour of a small cavity cooler and the behaviour of a coaxial travelling wave cooler.

Small cavity cooler	coaxial cooler
Acoustic field in the stack	
p = 1000 pa	p = 1166 pa
u_{opt} = 1.26 m.s^{-1}	u_t = 1.26 m.s^{-1}
φ_{opt} = 2.9 rad	φ_t = 2.9 rad
Performances	
ΔT_{max} = 20 K	ΔT_t = 23.9 K
P_{el} = 0.96 W	$P_{el,t}$ = 0.96 W

5 Conclusion

The experimental results presented here illustrate the thermal behaviour of a stack and a regenerator in devices which involve the optimal compactness. They validate theoretical ones given in a previous paper [1]. The performances of these compact devices compared with those of classical devices having equivalent stack (standing wave devices or travelling wave coaxial devices) show the potentiality of this compact thermoacoustic cooler. They enlighten a new feature, namely the way of adjusting the available parameters in order to achieve optimal heat transfer or temperature gradient, even COP. In summary, beyond this flexibility, the efficiency of the proposed device is greater than or equivalent to the classical devices having equivalent stacks although it is much smaller.

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