SIZE CONSIDERATIONS IN INTERFACING THERMOACOUSTIC COOLERS WITH ELECTRONICS.

E. Abdel-Rahman, N.C. Azenui, I. Korovyanko, and O.G. Symko
Department of Physics
University of Utah
Salt Lake City, Utah 84112-0830
Phone: (801) 581-6132
Fax: (801) 581-4801
Email: orest@physics.utah.edu

ABSTRACT

Small thermoacoustic coolers/heat pumps show much promise for heat management in microcircuits, especially since they can be miniaturized for interfacing with circuits. Usually they are operated in the resonant mode with sound pumping heat; the device dimensions are reduced when the acoustic pump frequency is raised. Results on the development of this technology will be presented for devices operating in the frequency range of 4 kHz to 25 kHz. Since the efficiency and cooling power depend on geometric factors, scaling in the miniaturization plays an important role. The cooling power varies with the total effective cross-sectional area of stack relative to sound field and hence on the cross-sectional area at stack position.

We have developed and tested devices which vary in length from 4 cm down to 0.8 cm, the cross-sectional area being fixed by available acoustic drivers. An important issue is that of direct interfacing of single units with a microcircuit versus an array of smaller basic thermoacoustic units. Advantages of such arrays range from lifetime of drivers, long-term reliability of the device, to closer interfacing with chip on microcircuits. Results show that indeed the cooling power depends on the cross-sectional area of stack in a single unit. Also, the cooling power depends on level of acoustic drive; in the ultrasonic range it can be raised since the power density of such units is typically high.

Keywords: Thermoacoustics, cooler, microcircuits.

INTRODUCTION

The development and miniaturization of thermoacoustic coolers for heat management on microcircuits require scaling and optimization of the main parameters that are usually used in large-scale devices [1]. Moreover suitable components have to be identified for the miniature coolers. They consist of an acoustic driver, a resonator, heat exchangers, and a stack, an energy storage element. The working gas is air, but for improved performance it can be a gas mixture [2], which could be pressurized. The miniaturization of this type of device raises interesting questions dealing with efficiency, performance and limitations as well as fabrication problems when the device size is reduced by almost an order of magnitude or even more. This paper deals with such miniaturization issues and it presents some of the achievements in this type of device. As the cooler size is reduced, the operating frequency is raised covering here the range of 4 kHz – 24 kHz; in that case cooler dimensions vary from 4 cm to 0.8 cm.

Although the field of thermoacoustics has a long history, it is only in recent years that important applications have started to be developed. Essentially there are two effects: heat can generate sound in a suitable resonator, and conversely, sound can pump heat. It is the basis of a refrigerator or a heat pump. Merkli and Thomann were the first to show that sound in a resonant tube produces cooling [3]. Wheatley and collaborators [4] built the first acoustic refrigerator and demonstrated its potentials; the acoustic frequency was around 500 Hz. This was followed by a variety of large scale devices [5]. On the theoretical side there has been very strong support by Rott and collaborators [6]; their approach was to linearize the Navier-Stokes equations in the limit of low amplitude. Here we present the development of small scale acoustic coolers.

A thermoacoustic refrigerator works on the principle that sound can pump heat up a temperature gradient, and the temperature difference ΔT that is then achieved depends on the sound intensity used to pump heat. The element across which heat is pumped is called a stack; it has a heat exchanger at each end in order to maintain the heat flow. A high sound intensity of order of 160 dB, is needed to produce a ΔT of ~10°C or more; this can be achieved with a powerful driver and a high-Q resonator. Problems of accessing the high temperature and low temperature heat exchangers will be presented and the thermal interfacing with the circuit where the device will be used will be discussed.

HIGH FREQUENCY ACOUSTIC COOLER

The basic unit is shown in Fig. 1. It consists of a sealed ½-wave resonator with the hot heat exchanger near the driver which is thermally anchored to ambient temperature. The cold heat exchanger is thermally attached to a copper flange which makes contact to the circuit that needs to be cooled. Depending on its size it can operate in the frequency range

4 kHz – 25 kHz. The rate at which heat is pumped acoustically along a stack, neglecting viscous dissipation for simplicity, is given by [1]:

$$\dot{Q} = -\tfrac{1}{4}(\Pi\,\delta_\kappa)(\mathrm{T}_m\beta)\,p_1 u_1\,(\Gamma - 1) \qquad (1)$$

The rate at which acoustic work $\dot{W}$ is absorbed is given by:

$$\dot{W} = \frac{1}{4}\Pi\,\delta_\kappa \Delta x \frac{T_m \beta^2\,\omega}{\rho_m\,c_p}\,p_1^2\,(\Gamma - 1) \qquad (2)$$

where Δx is the stack length, ρ_m is the gas density and c_p its specific heat.

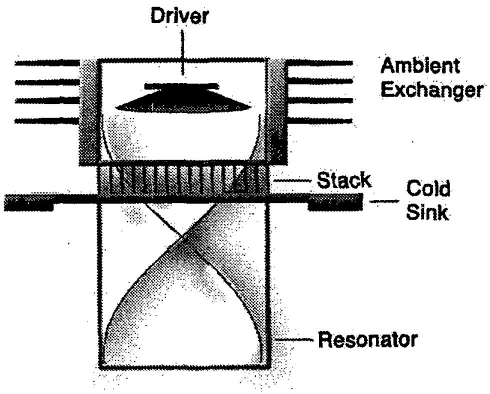

Figure 1. Thermoacoustic Cooler with hot heat exchanger at ambient temperature and the cold one used for cooling sample.

- $\Pi\delta_k$ is the heat contact area between the sound field and the stack elements (here, random fibers); it is perpendicular to the axial sound field.
- δ_k is the thermal penetration depth (average distance over which sound field interacts thermally with a body); it is 44 μm at 5000 Hz with air at one atmosphere.
- $(\mathrm{T}_m\beta)$ are gas parameters of working fluid, β being the thermal expansion coefficient, and T_m the average temperature.
- $p_1 u_1$ is the acoustic intensity of the standing wave at the stack produced by the driver and amplified by the quality factor Q of the resonator; p_1 is the acoustic pressure and u_1 is the particle velocity.
- Γ is the ratio of achieved thermal gradient across the stack normalized to a critical temperature gradient; when $\Gamma = 1$ or larger the refrigerator stops working as a cooler and the engine starts operating backward, becoming a prime mover.

For N stack elements $\dot{Q}$ will be that many times larger.

The above parameters need to be maximized, in particular when the operating angular frequency ω is raised to make the devices smaller.

The total heat flow area $\Pi\delta_k$ for the whole stack is essentially $\mathrm{N}(\pi\delta_\kappa^2)$ where N is the number of fibers. Its value is raised by using many fibers in parallel, i.e. by increasing the overall stack cross-sectional area πR^2, R being the stack radius. Since the thermal penetration depth δ_κ decreases as the frequency goes up, it will be necessary to have more fibers in the stack when the operating frequency is raised;

$$\delta_\kappa = \left(\frac{2\kappa}{\omega}\right)^{\frac{1}{2}} \qquad (3)$$

κ is the thermal diffusivity. Fig. 2 shows the κ cross-sectional area of the stack of fibers.

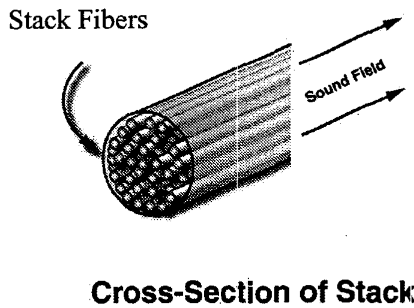

Figure 2. Cross section of stack with fiber elements.

In practice the stack consists of a random arrangement of fibers (such as cotton or glass wool) in thermal contact at each end to a copper fine mesh heat exchanger, capable of transferring heat radially and yet open to sound field.

The acoustic intensity $p_1 u_1$ used to drive the cooler is a dominant factor. It can be raised to a high level by an efficient driver and high Q resonator.

The quality factor Q of the resonator plays a major role here. It is defined as:

$$Q = \frac{\omega m}{r_s} \qquad (4)$$

with m = mass of gas in resonator, and r_s = resonator dissipation; the Q can be maximized by adjusting geometric factors, i.e. Q varies as R/δ_υ, where δ_υ is the viscous penetration depth [1]. This favors a large resonator radius R, assuming that the axial mode remains dominant up to the point where R ~ L, where L is the length of the resonator. The Q can also be raised by increasing the gas density, thus requiring high pressure operation.

The driver contribution to the sound level is limited by its sensitivity, acoustic load, and frequency range. A piezoelectric driver in the bimorph configuration loaded with a cone was chosen for this task because of its high sensitivity, efficiency, and lightweight [7]. A typical electro-acoustic sensitivity is ~ 105 dB/W at 1 m. At the driver cone surface the sound level can reach 140 dB for a cone of 4 cm diameter; moreover the Q of the resonator will raise the acoustic level substantially. Exceeding the maximum drive to the device can lead to cracks in the thin film piezoelectric elements.

Eq. 1 shows that for $\Gamma < 1$ heat will flow from cold to hot when pumped acoustically. Maximum rate of heat pumping will occur when the temperature gradient across the stack is zero. The heat flow will be toward the nearest pressure antinode. Between the pressure antinode and node the product $p_1 u_1$ will be the largest, and that is an optimal location for the stack-heat exchanger assembly.

A condition that needs to be met by the stack is that the specific heat of the stack fibers remains larger than that of the gas surrounding it. This is determined by the parameter ε [1] which is the ratio of gas specific heat per unit area to stack specific heat per unit area and is given by:

$$\varepsilon = \frac{\rho_m c_p \delta_\kappa}{\rho_s c_s \delta_s} \tag{5}$$

it needs to be less than 1. Here δ_s is the solid thermal penetration depth of the stack material. In the case of air and fiber glass at 5,000 Hz $\varepsilon = 0.027$.

In scaling down the device as the frequency is raised, it is possible to use similitude principles [8] to help optimize the relevant parameters. As a start, a direct application of Equation 1 can help in scaling. For example the thermal penetration depth will decrease with frequency reducing $\Pi\delta_\kappa$. Going from 4 kHz to 25 kHz will cause a reduction of δ_κ by ~ 2.2 and hence the number of stack fibers will have to be increased by that factor in order to keep the cooling power constant as the frequency is raised, for a constant stack cross-section. It is assumed that the acoustic intensity factor $p_1 u_1$ at the stack is fixed, otherwise changes in resonator dimensions will affect the results. There is a direct frequency dependence for the intensity factor $p_1 u_1$ since the resonator quality factor Q has a $\sqrt{\omega}$ dependence. This makes the device attractive for the high frequency range including the ultrasonic range, where acoustic cooler arrays can be developed.

RESULTS

In dealing with down scaling of thermoacoustic devices for small scale applications, we have made use of a computer program DELTAE, which has been developed for modeling thermoacoustic refrigerator performance [9]. Application of this software to the problem presented here, that of increasing the operating frequency and hence reducing the size of the cooler, has provided interesting results on the frequency dependence of the cooling power and the Coefficient of Performance (C.O.P.) defined as rate of heat pumped divided by rate of acoustic work. In order to get an estimate of the cooling power we assumed a refrigerator as described in the software program, a ¼-wave resonator terminated with a Helmboltz resonator and with parallel plates as stack; we studied the effect of raising the pump frequency. The results for the C.O.P. as a function of frequency are shown in Fig. 3 for a refrigerator of fixed stack cross-sectional area; its length is reduced as the frequency is increased. Stack plate thickness and acoustic intensity are kept constant. Results show a slight drop in the C.O.P. as the operating frequency f is increased from 4 kHz to 25 kHz. A few experimental points are included. The very large experimental value of the C.O.P. on that graph is 2.4 for a device at 500 Hz and it is due to high pressure operation and use of Xe-He mixture as the working gas.[10].

As discussed previously the rate of heat pumping depends on frequency, according to Eq. 1. The rate of work absorption $\dot{W}$ by the stack also depends on frequency [1]. Consequently the coefficient of performance $\dot{Q}/\dot{W}$ should be independent of frequency, at least within the approximations presented in reference 1. The slight departure from no frequency dependence can be attributed to inherent losses in the device as the operating frequency is raised, in particular the heat conduction losses across the stack as the stack length Δx is decreased in the size reduction.

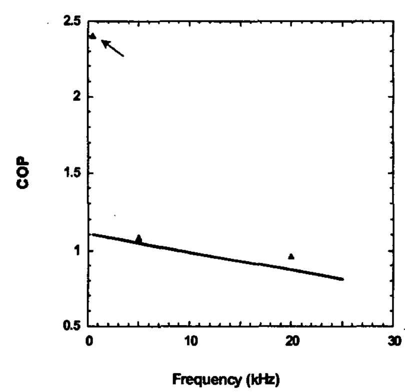

Figure 3. Coefficient of Performance of a Thermoacoustic Refrigerator as a function of operating frequency based on modeling and a few experimental points. The solid line is DELTAE calculations and the triangle represents experimental data.

The above results show that the coefficient of performance of a thermoacoustic refrigerator at constant acoustic intensity is essentially independent of operating frequency well into the ultrasonic range, even though engine performance depends on geometric factors.

We have developed coolers of the type shown in Fig. 1, which so far have operated at frequencies from 4 kHz to 25 kHz, the smallest one being 8 mm long. The diameter varied from 4.1 cm for the 4 kHz device to 6 mm for the 24 kHz unit.

The 5kHz refrigerator has achieved cooling power levels of 0.5-1 watt and a C.O.P. of 1.2. Much of the performance depends on the sensitivity of the driver and the amount of acoustic power that it can produce. Temperature differences of 38-40°C have been reached at sound intensity levels of 165-170 dB. The ultrasonic unit has achieved ΔT of 5°C in preliminary measurements, a limitation being the precision in its fabrication and in tuning of the resonator to the driver.

An interesting problem in this type of device is that of heat transfer, internally and externally. Heat transfer from and to the stack is maintained by a high level of sound that moves heat in small displacement steps of u_1/ω just as it did along the stack; this is especially important at the heat exchanger-stack interfacing which is a loose mechanical contact. The hot heat exchanger handles the heat pumped from the cold heat exchanger, plus the acoustic power and the heat generated by the driver. The latter is due to the extensive bending of the piezo element, electric dipole relaxation in the piezo material, and Ohmic contacts. In the devices presented here heat at the hot exchanger is dissipated to room temperature by convection and radiation by parallel plate fins.

Performance inefficiencies arise from heat transfer problems and viscous losses within a viscous penetration depth δ_υ from stack filaments and from resonator walls. Stack resistance to sound wave causes intensity attenuation and introduces non-linearties [11]; this places a limit on the amount of stack material [12].

Size reduction of the thermoacoustic cooler so that it can operate in the ultrasonic range, as presented here, provides an opportunity for combining such units into an array with increased cooling power. Such approach will make it easier to interface with the electronic components that need cooling. Further reduction in size of the thermoacoustic cooler will lead eventually to the limit where deviations from the continuum model will occur, when the mean free path in the gas starts to become comparable to a characteristic flow dimension such as the gradient of a macroscopic quantity.

The heat transfer problem with this type of device is an interesting one and some parts of it are peculiar to the thermoacoustic device. The steps in the overall heat transfer process consist of: (i) heat absorption from circuit by direct metallic contact with cold heat exchanger; (ii) heat transfer from cold exchanger to stack elements by the pumping action of the sound field in steps of u_1/ω; this is unique to thermoacoustics, (iii) heat pumping along stack elements by sound field in steps of u_1/ω, (iv) heat transfer across stack-hot heat exchanger interface enhanced by acoustic pumping, (v) heat dissipation by hot heat exchanger by conduction to thermal fins and air convection. It would seem that the highest thermal resistance occurs at the interfacing of heat exchangers and stack. Computer modeling of this part of the refrigerator shows some hydrodynamic turbulences which also help in the heat transfer process between stack elements and the heat exchangers. The large area structure of the heat exchangers lends itself well to heat transfer from the circuit that needs to be cooled and for heat dissipation in air at the hot exchanger.

The devices presented here are attractive for electronics heat management because they are simple, they have essentially no moving parts, they use environmentally safe working fluids, and they are quite efficient.

Supported by DARPA-HERETIC Program and the Office of Naval Research.

REFERENCES

[1] Swift, G.W., "Thermoacoustic Engines," J. Acoust. Soc. Am. 84, 1145 (1988).

[2] Hofler, T.J., Proc. 5^{th} Int. Cryocoolers Conf., P. Lindquist, ed., Wright-Patterson Air Force Base, Ohio (1988), P. 93.

[3] Merkli, P. and Thomann, H., "Thermoacoustic Effects in a Resonance Tube." J. Fluid Mech. 70, 161 (1975).

[4] Wheatley, J., Hofler, T., Swift, G., and Migliori, A., "Experiments with an Intrinsically Irreversible Acoustic Heat Engine," Phys. Rev. Lett. 50, 499 (1983).

[5] Garrett, S.L. and Hofler, T.J., "Thermoacoustic Refrigeration," Am. Soc. of Heating, Refrigerating and Air-Conditioning Engineers Journal (1992).

[6] Roth, N., "Thermoacoustics," Advances in Applied Mechanics, 20, 135 (1980).

[7] Bost, J.R., "A New Type of Tweeter Horn Employing a Piezoelectric Driver," J. of the Audio Eng. Soc., 23, 796 (1975).

[8] Olson, J.R. and Swift, G.W., "Similitude in Thermoacoustics," J. Acoust. Soc. Am., 95, 1405 (1994).

[9] Ward, W.C., and Swift, G.W., J. Acoust. Soc. Am. 95, 3671 (1994); DELTAE program.

[10] Tijani, M.E.H., "Loudspeaker Driven Thermoacoustic Refrigerator," Ph.D. Thesis, Sept. 2001, Technische Universiteit Eindhoven.

[11] Kuntz, H.L. and Blackstock, D.T., "Attenuation of Intense Sinusoidal Waves," J. Acoust. Soc. Am. 81, 1723 (1987).

[12] Atchley, A.A., Hofler, T.J., Muzzerall, M.L., Kite, D., and Ao, C., "Acoustically Generated Temperature Gradients in Short Plates," J. Acoust. Soc. Am. 88, 251 (1990).