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THE REALITY OF A SMALL HOUSEHOLD THERMOACOUSTIC REFRIGERATOR

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ABSTRACT

This paper discusses the thermoacoustic refrigeration cycle and how it can be applied to real world uses, particularly its use in household refrigeration systems. The commercial viability of this technology is determined by comparing it to a vapour compression system. The paper has two goals: to determine practical applications where thermoacoustic refrigeration may prove a strong rival to current methods, and to determine what future developments are required for this technology to be of commercial value.

INTRODUCTION

The thermoacoustic effect was first discovered in the 19th century when heat driven acoustic oscillations were observed in open-ended glass tubes [1]. These devices were the first thermoacoustic engines, consisting of a bulb attached to a long narrow tube (see Figure 1). Lord Rayleigh made a qualitative explanation of the effect in 1896 [2], however a quantitatively accurate theory of the phenomena was not established until the 1970's [3]. It was in the 1980's that thermoacoustic refrigeration was first developed, when a research group at the Los Alamos National Laboratory [4] showed that the effect could be used to pump heat. The technology has seen rapid growth since then, much of this being attributed to the Naval Postgraduate School in Monterey, California, which carried out the development of a reliable spacecraft cryocooler [5].

What thermoacoustic refrigerators offer is both simplicity and reliability. Unlike current commercial devices that require crank shafts and pistons, these devices use only a single moving part - the diaphragm of a loudspeaker. What currently makes them very attractive as an alternative to other approaches is their use of an inert gas as the working fluid, making them environmentally clean.

In order for the thermoacoustic refrigerator to become a viable commercial alternative their efficiency has to be competitive when compared to currently used systems. This paper compares the COP of a thermoacoustic refrigerator to that of a vapour compression based system for a variety of heat loads and temperature spans to investigate under what conditions a thermoacoustic refrigerator might be competitive with current refrigeration systems.

The basics of thermoacoustic refrigeration

A basic thermoacoustic refrigerator consists of a stack of thin parallel plates housed within a resonator, as shown schematically in Figure 2(a). Heat can be pumped from the cold to warm end of the stack by setting up a standing wave within the resonator. This effect, where heat is pumped up a temperature gradient by the use of sound, may be explained by considering an element of fluid as it oscillates back and forth along the stack, as shown in Figure 2(b).

The element experiences a cyclic temperature oscillation about its mean temperature, due to adiabatic compression and expansion of the gas. Irreversibilities caused by a temperature difference between the oscillating working fluid and the stack result in the correct phasing between the pressure and temperature oscillations. The phasing is such that when

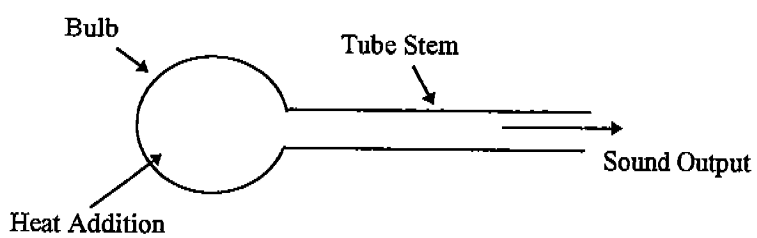
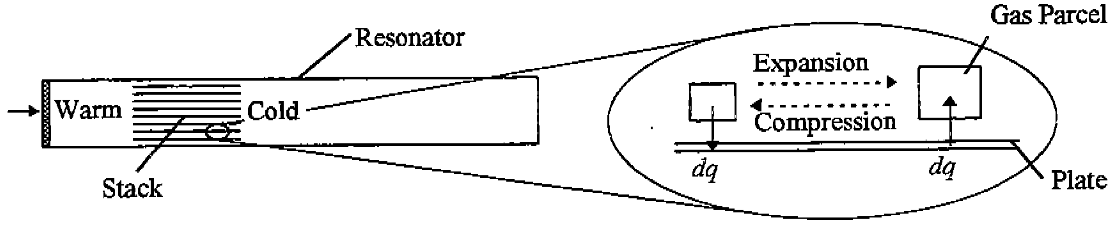


Figure 1. The Sondhauss tube. Heat is applied to the bulb resulting in the production of sound.



(a) (b)

Figure 2. A simple thermoacoustic refrigerator showing a magnified view of a gas parcel as it transports a small amount of heat dq along the stack.

the element is in its right-most position it has been expanded to a temperature that is colder than the local stack temperature, and so absorbs heat from the stack, and when the gas parcel is displaced up the plate to its left-most position it is compressed to a temperature that is hotter than the stack, thereby rejecting heat to the stack. As all gas elements within the stack behave in a similar manner the net result is the transport of heat up a temperature gradient (from the cold to warm end of the stack). This heat transport between the gas and the stack only occurs within a region close to the stack known as the thermal penetration depth (δ_K).

Work is absorbed by the gas element as the thermal expansion occurs during the low pressure phase and the thermal contraction during the high pressure phase of the acoustic cycle. If a temperature gradient is imposed along the stack and the temperature gradient is large enough, the device ceases to be a thermoacoustic refrigerator and starts producing work - it becomes a thermoacoustic engine. This is because after the gas parcel has been adiabatically compressed it will no longer be hotter than the stack and will absorb heat (instead of rejecting it) at high pressure and expand, thereby doing work. For a review of the theory behind thermoacoustic engines see [6].

A THERMOACOUSTIC REFRIGERATOR

A thermoacoustic refrigerator has three other major components apart from the stack and resonator. They are the driver, and hot and cold heat exchangers. The driver is responsible for producing the standing wave. Typically this is an electromagnetic device, but a thermoacoustic engine coupled to the hot end of the stack has been used. The heat exchangers are responsible for getting the desired heat into and out of the stack.

There is a variety of configurations for these components, but a device similar to the design shown in Figure 3 is considered in this paper, as this has been shown to be an efficient design. The device consists of a spirally wound stack which has a spacing of about four times the thermal penetration depth. The heat exchangers are parallel copper strips extending across the length of the tube. This is a quarter wavelength design and can be pressurised allowing higher power densities. The operating frequency of the system is determined by the length of the resonator and the speed of sound in the gas, and typically ranges from 50 to 1000 Hz. The working fluid is typically helium or a mixture of helium and another inert gas. Gas mixtures that reduce the Prandtl number increase the efficiency, but unfortunately also reduce the heat load. Current gas mixtures have Prandtl numbers significantly below 2/3.

The capacity of a thermoacoustic refrigerator is determined primarily by the cross sectional area of the stack and the driving ratio P_{osc}/P_m. This is the ratio of the peak oscillating pressure P_{osc} to the mean pressure P_m. An increase in either of these factors results in an increase in the heat load.

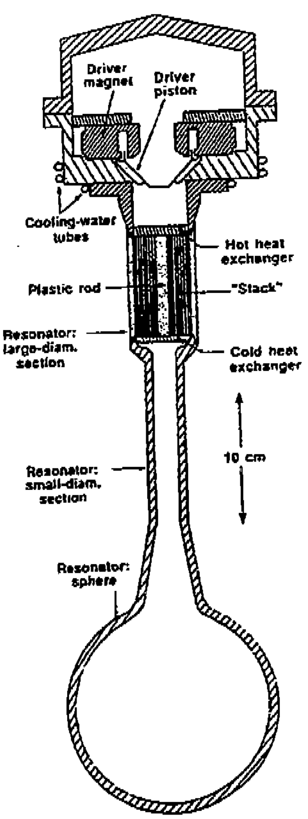


Figure 3. Hofler design thermoacoustic refrigerator

VAPOUR COMPRESSION REFRIGERATION SYSTEM

A basic vapour compression refrigeration system consists of four major components: a compressor, condenser, capillary tube, and evaporator as shown in Figure 4(b). Heat is absorbed at low temperature in the evaporator and rejected at the condenser. Work is supplied through the compressor. Non-adiabatic capillary tubes are normally used in household refrigerators where the capillary is soldered to the suction line or run inside the suction side to superheat the suction gas to ambient temperatures. In the future compressors are likely to have variable speed drives, enabling them to run continuously to meet lower load situations but at higher compressor efficiencies.

COMPARISON OF THERMOACOUSTIC AND VAPOUR COMPRESSION REFRIGERATION SYSTEMS

The coefficient of performance (COP) is the standard measure of the efficiency of refrigeration systems. The COP of the thermoacoustic and vapour compression refrigeration systems shown in Figure 4 are respectively

$$\mathrm{COP} = \frac{Q_c}{W_a} \quad ; \quad \mathrm{COP} = \frac{Q_{evap}}{W_{Comp}} \tag{1}$$

where, for the thermoacoustic system, Q_c is the heat load into the cold end of the stack and W_a is the acoustic power into the resonator. For the vapour compression system Q_{evap} represents the evaporator capacity (i.e. the heat load that the refrigerant in the evaporator has to remove to keep the fridge air at the desired temperature; normally 3 C) and W_{Comp} the mechanical work into the compressor. For a refrigeration cycle the COP is bounded by the Carnot efficiency which is given by

$$\mathrm{COP}_{\mathrm{Carnot}} = \frac{T_{evap}}{T_{cond} - T_{evap}} \tag{2}$$

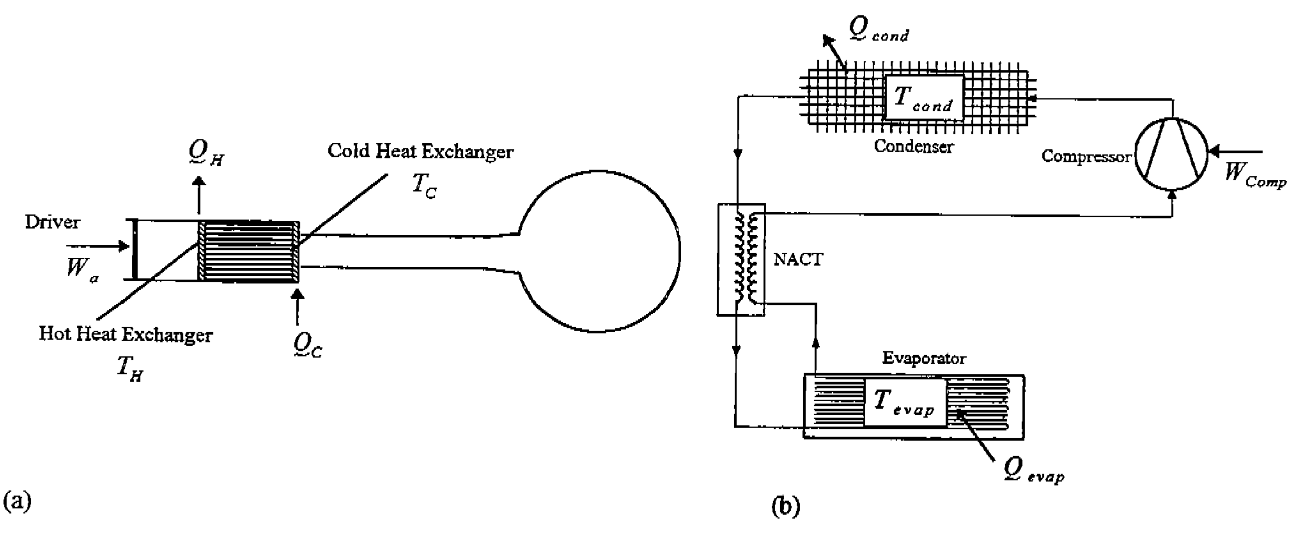


Figure 4. (a) A simple thermoacoustic refrigerator. (b) Component diagram of a vapour compression household refrigerator/freezer with Non-Adiabatic Capillary Tubes (NACT) as expansion valves.

The thermoacoustic system was compared to a vapour compression system for two evaporator temperatures, T_{evap}, of -15 C and -25 C, corresponding to a refrigerator and freezer configuration respectively. The evaporator temperature corresponds to the cold heat exchanger temperature T_C in the thermoacoustic system. These temperatures were chosen in order to keep the internal air temperature at 3 C for a refrigerator and -15 C for a freezer as required by the New Zealand Standard. The condenser temperature, T_{cond}, which corresponds to the hot heat exchanger temperature T_H of the thermoacoustic system, was held at 43 C for both of these evaporator temperatures. From equation (2) the Carnot efficiency for the fridge and freezer are respectively 4.45 and 3.65.

Thermoacoustic refrigerator model

The thermoacoustic refrigerator was modelled using DeltaE [7] (Design Environment for Low amplitude ThemoAcoustic Engines). DeltaE is a program which solves the one dimensional wave equation for acoustic and thermoacoustic elements based on a low-amplitude acoustic approximation. Within the stack the wave equation [6] is solved simultaneously with the enthalpy equation [6] in order to find both the temperature and pressure profiles. For other components in the system, the appropriate wave equation is used with continuity of pressure and volumetric velocity applied at the intersection of each section. The program does not take account of non-linear effects. (For a detailed explanation of the program see the thermoacoustic home page at http://rott.esa.lanl.gov/)

Losses generated within the stack are accounted for as well as losses from the resonator and rough estimates for heat exchanger losses. Losses not included were those from the driver plus those incurred by the need for some external heat exchanger loop to get the heat from the cold refrigerated compartment into the stack and reject the waste heat at the other end of the stack.

Before using DeltaE an initial design for a 200 W thermoacoustic refrigerator was obtained using closed form solutions of the short engine equations [6]. These equations do not provide an accurate enough estimate to predict the actual performance, but do give an initial design that can be used by DeltaE.

Vapour compression refrigerator model

It is assumed that future refrigerators will be designed for a maximum capacity (say, Q_{evap} = 200 W) and will have a variable speed control mechanism on the compressor. This will allow them to run continuously (rather than the current 'on - off' approach) at lower speeds to meet lower load conditions. For such a mechanism it is further assumed that the compressor efficiency at the design point (Q_{evap}=200 W) will have a volumetric efficiency, say of about 60% and an isentropic efficiency, say of about 55%, and that these efficiencies will increase in steps of about 10 % and 1% respectively as the cabinet load reduces by 50 W, as shown in Figure 5.

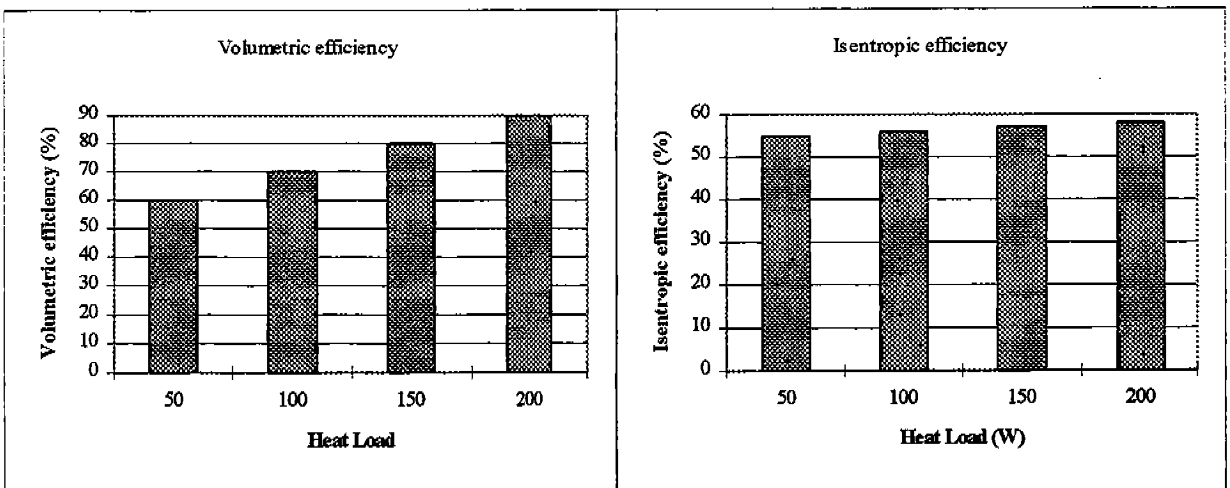


Figure 5. Changes in the volumetric and isentropic efficiency with heat load of a variable speed compressor.

For the system analysed no account was taken of subcooling or superheating, however a rise of 27 C in the gas suction line was assumed, with the capillary exposed to ambient air (about 32 C) so that the return gas temperature to the compressor is 32 C. It is also assumed that the pressure losses in both the heat exchangers (the evaporator and condenser) are about 10 %. The computations were made using BICYCLE [8] for both the refrigerator and freezer.

COP Comparison

A comparison of the COP of each system was made for both the refrigerator and freezer at a variety of heat loads, namely Q_c and Q_{evap} of 50, 100, 150, and 200 W. The results are shown in Figure 6.

The amount of heat that the thermoacoustic system could pump was altered by changing the driving ratio as shown in Figure 7(a). The higher the driving ratio, the more heat that could be pumped, but at a lower efficiency. The highest driving ratio obtained was 0.049, which corresponds to a freezer with a 200 W heat load. Here, non-linear effects are becoming significant, but linear theory still gives reasonably accurate results. As already stated, the heat load in the vapour compression system was altered by varying the speed of the compressor, the appropriate compressor size as calculated by BICYCLE for each heat load is given in Figure 7(b).

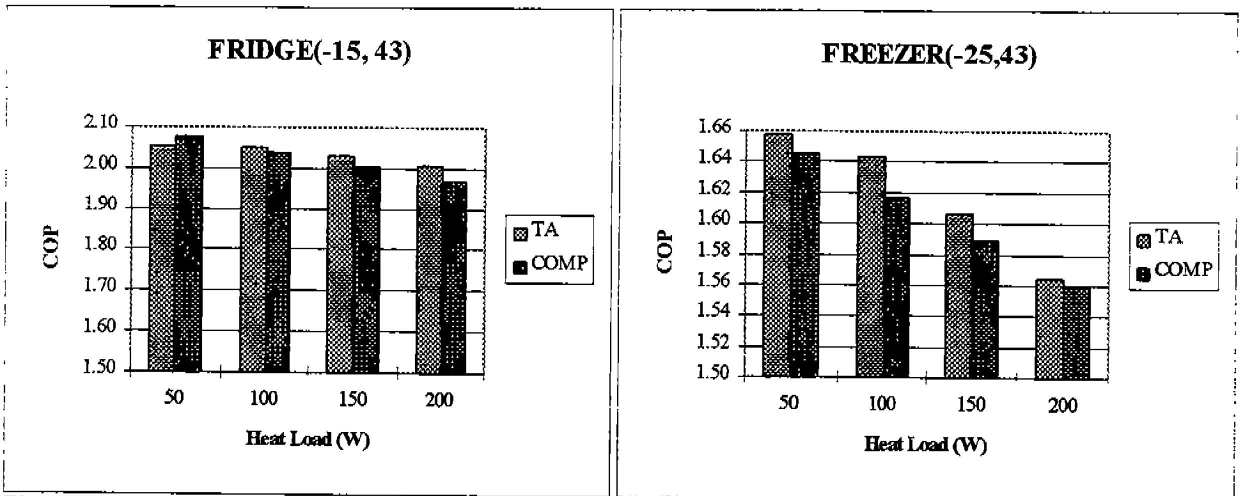
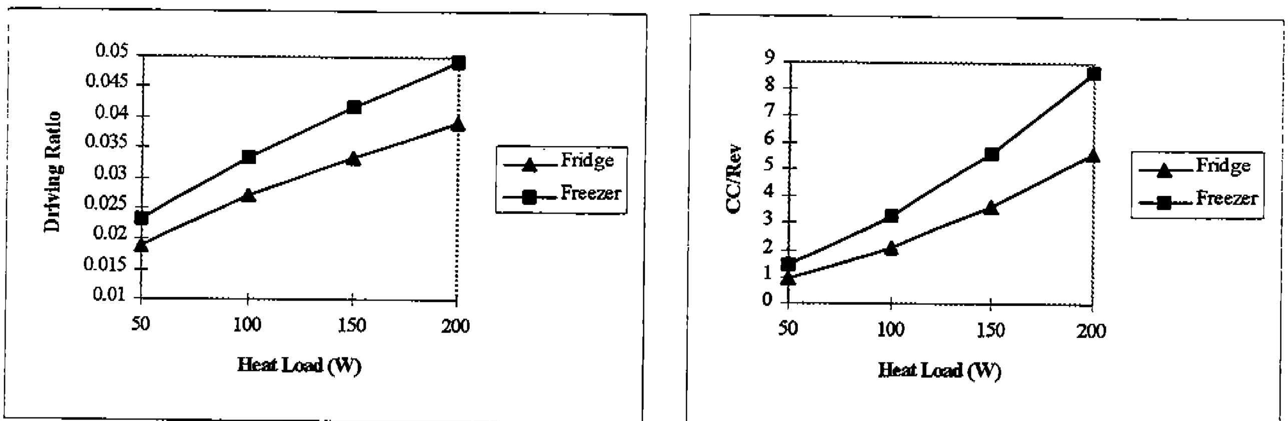


Figure 6. Comparison between the COP of a thermoacoustic (TA) and vapour compression (COMP) refrigerator and freezer for heat loads of 50, 100, 150, and 200 W.



(a) (b)

Figure 7. (a) Variation of the driving ratio with heat load for a thermoacoustic refrigerator and freezer, and (b) The CC rating of a vapour compression refrigerator and freezer at different heat loads.

DISCUSSION AND CONCLUSIONS

For the relatively small heat loads of household refrigerators and freezers, thermoacoustic refrigeration compares well with the vapour compression system. However, they are not competitive when the application requires large heat loads. This is a result of the inherent irreversibility of the process (the transfer of heat across a non-zero temperature difference) which causes a conflict between the COP and cooling capacity. Only in the limiting case of zero heat load can the process become reversible and reach the theoretical maximum efficiency. They are therefore ideal for applications that require low heat loads, such as the cooling of electronic equipment.

While in theory thermoacoustic refrigerators offer competitive efficiencies for the household refrigeration market, practical applications require that efficient means of getting heat into and out of the stack be developed. This will most likely require some external loop which connects the cold end of the stack to the space that needs to be cooled and the hot end to the environment. This loop will of course add additional losses and a further complication to the system. Only by considering these heat exchanger losses can a realistic comparison be made with vapour compression systems.

One feature that may prove useful is that thermoacoustic refrigerators can be driven by a thermoacoustic engine. This would mean that such a refrigerator would have no moving parts and could use other forms of energy directly, e.g. natural gas, instead of electricity. In areas where electrical energy may not be readily available, either due to isolation or being expensive, thermoacoustic refrigeration may offer a competitive solution.

There is still much that can be done to improve the performance of thermoacoustic engines. One area where the efficiency of these devices can be increased is the reduction of the viscous loss within the stack. The viscous loss is a result of work being required to overcome the viscous shear force as the gas oscillates. Due to the viscous and thermal penetration depths being comparable most of the area within the stack experiences viscous shear. This loss may be reduced by using alternative stack geometries. A general formulation for channel stacks of arbitrary geometry [9] concludes that parallel plate channels are the most efficient. It has been further identified that pin stacks offer even greater improvements in efficiency [10]. Another area of current interest is to extend the theory to include non-linear effects. This is because the assumption of small oscillations made in the linear theory becomes inaccurate for the high drive ratios which allow higher heat loads.

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