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Thermoacoustic Refrigerator for Space Applications

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A new spacecraft cryocooler which uses resonant high-amplitude sound waves in inert gases to pump heat is described. The phasing of the thermoacoustic cycle is provided by thermal conduction. This "natural" phasing allows the entire refrigerator to operate with only one moving part (the loudspeaker diaphragm). A space-qualified thermoacoustic refrigerator was flown on the Space Shuttle *Discovery* (STS-42) in January, 1992. It was entirely autonomous, had no sliding seals, required no lubrication, used mostly low-tolerance machined parts, and contained no expensive components. Thermoacoustics is shown to be a competitive candidate for food refrigerator/freezers and commercial/residential air conditioners. The design and performance of the Space ThermoAcoustic Refrigerator (STAR) is described.

Nomenclature

a = speed of sound
c_p = isobaric specific heat
f = frequency
M = molecular weight
p = pressure
Q = heat
R = universal gas constant
T = absolute temperature
x = mean position of stack within resonator
Γ = ratio of mean temperature gradient to critical temperature gradient
γ = polytropic coefficient
δ_κ = thermal penetration depth
κ = thermal conductivity
λ = acoustic wavelength
Π = stack perimeter
ρ = mass density

Subscripts
m = mean (ambient) value
1 = linear acoustic variable
2 = second-order acoustical quantity

Introduction

THE ubiquity of reliable and inexpensive refrigerators, freezers, and air conditioners in our daily experience produces a general complacency with regard to cooling engines that is responsible for some unpleasant consequences in both commercial and military applications. Recent ratification of the Montreal Protocols, an international agreement to ban the worldwide production of chlorofluorocarbons (CFCs),[1] and the failure of the chemical companies to produce stable, nontoxic, energy efficient, and environmentally safe CFC replacements for Rankine cycle coolers,[2] could have significant economic and social consequences.

One of the most obvious military problems generated by "refrigeration complacency" is the failure to develop reliable spacecraft cryocoolers with low vibration levels for use with infrared (IR) imaging. A manifestation of this bias against advanced refrigeration research and development was the creation, at great expense, of the sensors to be used by the Strategic Defense Initiative Organization (SDIO), all of which require active cooling, without a parallel development program for long-life, low vibration, space-based cryocoolers. The discovery of Hi-T_c superconductors, which may find use in high-speed electronics and signal conditioners, SQUID sensors, and computers, will only compound the military problems generated by this lack of inexpensive, long-lived, and compact cryocooler technology.

Since 1986, the Naval Postgraduate School has been attempting to alter this situation by an extensive research and development program based on the recently discovered[3] thermoacoustic heat pumping cycle. These efforts have included basic research into the nonlinear acoustics of both the thermoacoustic heat pumping process[4] and the generation of high-amplitude sound by temperature gradients,[5,6] as well as the development of an autonomous, space-qualified thermoacoustic refrigerator.

The Space ThermoAcoustic Refrigerator (STAR) was launched as a get away special (GAS) on the Space Shuttle *Discovery* (STS-42) in January, 1992, and will be the focus of this article. Before presenting a description of its most unique subsystems, we will present a simpified discussion of the principles of thermoacoustic refrigeration.

Thermoacoustic Heat Pumping Cycle

The interaction between acoustics and thermodynamics has been recognized ever since the dispute between Newton and Laplace over whether the speed of sound was determined by the adiabatic or isothermal compressibility of air. Although today there are probably many physicists who might still make the wrong choice (as Newton did!) most physicists have at least been exposed to a lecture demonstration such as the Rijke Tube[7] or have cursed Taconis oscillations in liquid helium,[8] so they are not surprised that temperature gradients can lead to the production of sound. The reverse process—thermoacoustic heat pumping—is far less well known, and was the first intentional demonstration of a new class of intrinsically irreversible heat engines.

Traditional heat engine cycles, such as the Carnot cycle typically studied in elementary thermodynamics courses, assume that the individual steps in the cycle can be made reversibly. Such analyses, which invoke the first and second laws of thermodynamics, lead to the limiting values for the efficiencies of prime movers and the coefficients-of-performance (COP) of refrigerators. These limiting values are never realized in practical heat engines and refrigerators due to the unavoidable irreversibilities, such as thermal diffusion, the thermophysical properties of real working fluids, and viscous

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dissipation, which always reduce performance below the ideal Carnot values. Reversible engines also require various mechanical devices (e.g., valves, cams, pushrods, linkages, timing chains, etc.) in order to execute the proper phasing of various cyclic processes (e.g., compressions, expansions, displacements, regeneration, etc.). In thermoacoustic engines, the irreversibility due to the imperfect (diffusive) thermal contact between the acoustically oscillating working fluid and a stationary second thermodynamic medium (the "stack") provides the required phasing. This "natural phasing" has produced heat engines which require no moving parts other than the self-maintained oscillations of the working fluid.

Simple, Inviscid, Lagrangian Model of the Heat Pumping Process

Although a complete and detailed analysis of the thermoacoustic heat pumping process is well beyond the scope of this article, the following simple, inviscid, Lagrangian representation of the cycle contains the essence of the process. A complete analysis[9] would necessarily include the gas viscosity, finite wavelength effects, longitudinal thermal conduction along the stationary second thermodynamic medium and through the oscillating gas, and the ratios of the heat capacities of the gas and second medium.

A schematic diagram of a simple, one-quarter wavelength, $\lambda/4$, thermoacoustic refrigerator is shown in Fig. 1. The loudspeaker at the left maintains the standing wave within the gas-filled tube. Its frequency is chosen so that the loudspeaker excites the fundamental ($\lambda/4$) resonance of the tube. The termination at the right end of the tube is rigid so that the longitudinal particle velocity at this end is zero (a velocity node) and the acoustical pressure variations are maximum (a pressure antinode). We have arbitrarily assumed that the loudspeaker is "ultracompliant,"[10] so that the left end of the tube is an acoustic pressure node and a particle velocity antinode. (If the loudspeaker was assumed to be noncompliant, then the tube would be a half-wavelength long.) To the left of the rigid termination is a stack of plates (the stack) whose spacing is chosen to be a few thermal penetration depths.

δ_κ represents the distance over which heat will diffuse during a time which is on the order of an acoustic period $T = 1/f$, where f is the acoustic frequency. It is defined[11] in terms of the thermal conductivity of the gas κ, the gas density ρ, and its isobaric specific heat (per unit mass) c_p:

$$\delta_\kappa = \sqrt{(\kappa/\pi f \rho c_p)} \tag{1}$$

This length scale is crucial to understanding the performance of the thermoacoustic cycle since the diffusive heat transport between the gas and the stack is only significant within this region. It is for that reason that the stack and the spacing between its plates are central to the thermoacoustic cycle.

For this analysis we will focus our attention on a small portion of the stack and the adjacent gas which is undergoing acoustic oscillations at a distance from the solid stack material which is small enough that a substantial amount of thermal

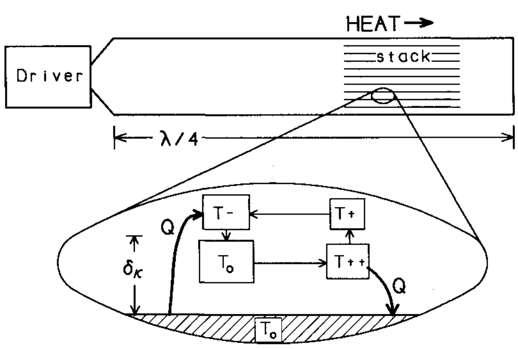

Fig. 1 Schematic diagram of a one-quarter wavelength thermoacoustic refrigerator shown in cross section.

conduction can take place in an amount of time which is on the order of the acoustic period. In the lower half of Fig. 1, a small portion of the stack has been magnified and a parcel of gas undergoing an acoustic oscillation is shown schematically. The four steps in the cycle are represented by the four boxes which are shown as moving in a rectangular path for clarity, although in reality, they simply oscillate back and forth. (In a more complete analysis, there is some mass flux normal to the plate due to the coefficient of thermal expansion of the gas.[12]) As the fluid oscillates back and forth along the plate it undergoes changes in temperature due to the adiabatic compression and expansion resulting from the pressure variations which accompany the standing sound wave. The compressions and expansions of the gas which constitute the sound wave are adiabatic if they occur farther than a few δ_κ from the surface of the plate. The relation between the change in gas pressure due to the sound wave p_1 relative to the mean (ambient) pressure p_m, and the adiabatic temperature change of the gas T_1 due to the acoustic pressure change, relative to the mean absolute (Kelvin) temperature T_m, is derived from the adiabatic equation of state for an ideal gas and is given below in Eq. (2):

$$(T_1/T_m) = [(\gamma - 1)/\gamma](p_1/p_m) \tag{2}$$

γ is equal to the ratio of the specific heat of the gas at constant pressure to the specific heat at constant volume, and is exactly $\frac{5}{3}$ for inert gases at the pressures of interest. It is smaller for all other gases, but is always greater than one.

Although the oscillations in an acoustic heat pump are sinusoidal functions of time, Fig. 1 depicts the motion as articulated (a square wave) in order to simplify the explanation. The thermodynamic cycle can be considered as consisting of two reversible adiabatic steps and two irreversible isobaric (constant pressure) steps. The plate is assumed to have a mean temperature T_m and a temperature gradient ∇T referenced to the mean position $x = 0$. The temperature of the plate at the left-most position of the gas parcel's excursion is, therefore, $T_m - x_1\nabla T$. At the right-most excursion it is $T_m + x_1\nabla T$.

In the first step of this four-step cycle, the fluid is transported along the plate by a distance $2x_1$ (the peak-to-peak acoustic displacement) and is heated by adiabatic compression from a temperature of $T_m - x_1\nabla T$ to $T_m - x_1\nabla T + 2T_1$. The adiabatic gas law provides the relationship between the change in gas pressure p_1 and the associated change in temperature T_1, as described in Eq. (2). Since we are considering a heat pump, $2T_1 > 2x_1\nabla T$, and work, in the form of sound, is done on the gas parcel. The gas parcel is therefore at a temperature which is higher than that of the plate at its present location.

In the second step, the warmer gas parcel transfers an amount of heat, dQ_{hot}, to the plate by thermal conduction at constant pressure and its temperature decreases to that of the plate, $T_m + x_1\nabla T$. In the third step, the fluid is transported back along the plate to position $-x_1$ and is cooled by adiabatic expansion to a temperature $T_m + x_1\nabla T - 2T_1$. This temperature is lower than the original temperature at location $-x_1$, so that in the fourth step the gas parcel adsorbs an amount of heat, dQ_{cold}, from the plate, thereby raising its temperature back to its original value, $T_m - x_1\nabla T$.

The net effect of this process is that the system has completed a cycle which has returned it to its original state, and an amount of heat, dQ_{cold}, has been transported up a temperature gradient by work done in the form of sound. It should be stressed again at this point that no mechanical devices were used to provide the proper phasing between the mechanical motion of the gas parcel and the thermal effects.

If we now consider the full length of the stack as shown in the upper portion of Fig. 1, the overall heat pumping process is analogous to a "bucket brigade" in which each set of gas parcels picks up heat from its neighbor to the left at a lower temperature and hands off the heat to its neighbor to the right at a higher temperature. Heat exchangers are placed at the ends of the stack to absorb the useful heat load at the left (cold) end of the stack, and exhaust the heat plus work (enthalpy) at the right (hot) end of the stack. The fact that the individual gas parcels actually move a distance which has typically been on the order of several millimeters means that good physical contact between the heat exchangers and the stack is not crucial since the moving gas can provide the requisite thermal contact.

Thermoacoustic Energy Transport

If there were no external (i.e., useful!) heat load applied to the stack and no longitudinal heat conduction along the stack, then eventually the temperature gradient in the plate would approach that caused by the adiabatic processes in the gas. In the absence of gas viscosity, this critical temperature gradient ∇T_{crit} is a function only of the gas thermophysical properties, the wavelength of the sound λ, and the mean position of the stack x within the standing wave field.

$$(\nabla T_{crit}/T_m) = [2\pi(\gamma - 1)/\lambda]\tan(2\pi x/\lambda) \tag{3}$$

The ratio between the temperature gradient in the stack and the critical gradient, $\Gamma = \nabla T_m/\nabla T_{crit}$, plays an important role in the performance of the stack, as will be explained below.

The rate of heat transport or heat pumping power Q_2 within the stack can be expressed in a simple form if we assume that the stack is much shorter than the wavelength of the sound and we again neglect viscosity:

$$Q_2 = -(\Pi\delta_\kappa/4)p_1u_1(\Gamma - 1) \tag{4}$$

The prefactor in Eq. (4) is simply one-quarter of the thermally effective cross-sectional area of the stack, where Π is the stack perimeter and δ_κ is again the thermal penetration depth. The heat pumping power is also proportional to the product of the acoustic pressure p_1 and the acoustic particle velocity u_1 within the stack. If the stack were located at a pressure or velocity node, no heat pumping would take place since either the pressure variations which cause the adiabatic temperature changes would be absent or there would be no motion of the gas parcels. Since pressure and particle velocity are proportional (the proportionality constant depends upon the location of the stack within the standing wave field), a doubling of the acoustic pressure would quadruple the heat transport. This is the origin of the subscript 2 on the heat transport symbol, Q_2, which emphasizes the fact that the magnitude of the thermoacoustic heat transport depends upon the square of the linear acoustic field variable.

The parenthetical factor in Eq. (4) is a measure of how close the system is to the limiting temperature gradient. As mentioned before, when $\Gamma = 1$, the gas parcels "see" their adiabatic temperature span on the stack so that no heat transfer from the gas to the plate takes place. When $\Gamma = 0$, the temperature of the plate is uniform and the greatest quantity of heat is pumped by the oscillating gas parcels.

The stack also absorbs work W_2 at a rate proportional to $(\Gamma - 1)$. The following simple expression for the work absorbed by the stack of length Δx, in the absence of viscosity, can be written if one assumes that the heat capacity of the stack is much greater than that of the gas:

$$W_2 = (\Pi\delta_\kappa/4)\Delta x[(\gamma - 1)/\gamma p_m]2\pi f(p_1)^2(\Gamma - 1) \tag{5}$$

This work represents the acoustical energy dissipated due to irreversible thermal conduction between the gas and the plate.

The ratio of the heat pumped Q_2 to the work done W_2 to pump that heat, is defined as the COP of the refrigerator. Since the temperature which is spanned by the stack is $\Delta T = \Gamma\nabla T_{crit}\Delta x$, one can show that the thermoacoustic COP $= \Gamma\ \mathrm{COP}_{Carnot}$, where $\mathrm{COP}_{Carnot} = T_{Cold}/\Delta T$. The Carnot coefficient-of-performance is dictated by the first and second laws of thermodynamics. Since for a heat pump, $\Gamma < 1$, we see that the thermoacoustic COP is always less than that of Carnot. We also see that with the thermoacoustic heat pump, there will be the same competition between power density and efficiency which exists in all heat engines. As we pointed out earlier, there is no useful heat pumped when $\Gamma = 1$, which is the point where the efficiency is at its maximum, and when $\Gamma = 0$, the heat pumping is greatest and the COP $= 0$.

The general results derived above from the simple, inviscid picture are essentially preserved when the viscosity of the gas is included, but the detailed mathematical descriptions are substantially more complicated. Discussion of those equations is well beyond the scope of this brief introduction to thermoacoustics. The reader is referred to the excellent review article by Swift[9] for a detailed derivation and discussion of more complete results.

Space ThermoAcoustic Refrigerator

STAR is the first attempt to demonstrate the advantages of the thermoacoustic heat pumping cycle for cryocooler applications in space. It is intended to operate autonomously in low Earth orbit aboard the Space Shuttle in a GAS canister. It derives its power from an internal battery power source (700 W-h) and is optimized for a modest temperature span ($\Delta T \leq 80$ K) and small heat loads ($Q_{cold} \leq 5$ W). Due to requirements of small size and light weight imposed by the GAS envelope, it operates at a frequency of about 400 Hz and is driven by an electrodynamic loudspeaker.[13] Its frequency of operation is adjusted automatically[14] to keep the system on resonance. The resonator length is approximately equal to a quarter wavelength of sound in a mixture of helium and argon or helium and xenon gas[15] maintained at a mean pressure of 10 atm (1.0 MPa). It has a single stack with a uniform spacing made of polyester film (Mylar™) and fishing line[15,16] which is rolled up similar to a "jelly roll." The stack used in STAR is 7.9 cm in length and 3.8 cm in diameter. Copper-fin, parallel plate heat exchangers are located at either end of the stack. Figure 2 is a scale drawing of the acoustical subsystems.

Acoustical Subsystems

The driver housing does more than simply hold the electrodynamic driver. The considerable mass and size of the

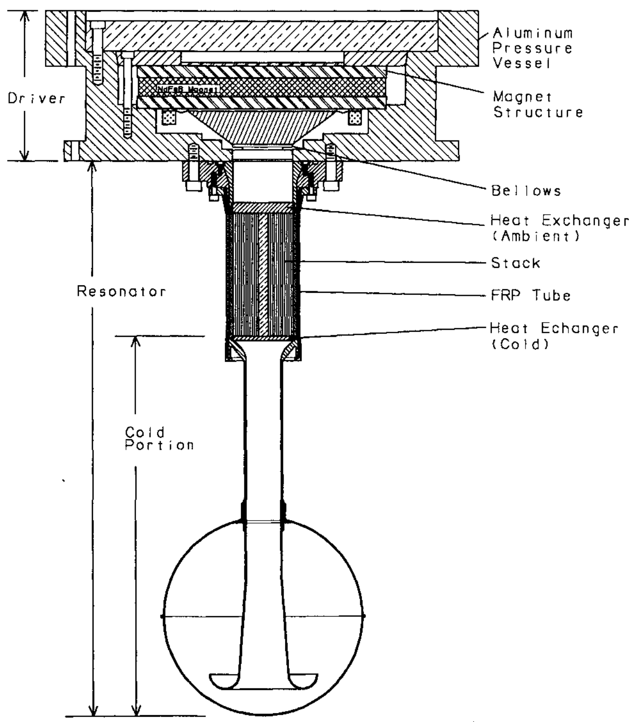

Fig. 2 Scale drawing of the STAR acoustical subsystems.

housing is a consequence of the fact that a commercially available loudspeaker was modified for this application. The driver housing serves as a heat sink for the heat generated by the resistive losses in the driver voice coil and the heat pumped away from the cold end of the resonator. It also contains the 10 atm of the helium/xenon gas mixture which is the "working fluid" within the refrigerator. The housing is bolted to the standard 12-in. bolt circle provided on the GAS canister lid, which acts as the heat radiator while on orbit.

The driver voice coil is attached to an aluminum reducer cone, which is in turn bonded to an electroformed nickel bellows. The bellows provides a means of transferring acoustic pressure to the resonator without the need for sliding seals. A miniature accelerometer is attached to the surface of the reducer cone opposite the bellows to monitor the displacement magnitude and its phase relative to the acoustic pressure at the face of the bellows. The acoustic pressure is monitored by a piezoelectric quartz microphone[17] followed by a MOSFET impedance converter located within the driver housing in close proximity to the microphone.

A capillary leak is provided between the driver housing internal volume and the resonator to allow for pressure equalization during operation and for the purging and filling of the entire system through a single port located at the side of the driver housing. There are electrical feed-throughs to provide access to the driver voice coil, microphone, and accelerometer. Finally, there is a piezoresistive pressure gauge mounted to the side of the housing to monitor the mean pressure of the gas mixture.

The resonator is a modified quarter wavelength tube. The open end is terminated by a tapered "trumpet" and sealed by a surrounding sphere. Thus, an "open" termination is simulated while still allowing the resonator to retain the 10 atm of gas mixture. The thermoacoustic stack and heat exchangers are located in a section of the resonator designed to reduce heat conduction back to the cold end. The resonator is instrumented with semiconductor diode thermometers at both the cold and hot ends and is wrapped with multiple layers of a superinsulation (aluminized Mylar and fiberglass) to prevent heating by thermal radiation. An electrical strip heater element is attached to the cold end of the resonator near the cold heat exchanger to permit measurement of refrigerator performance with a variable and quantifiable heat load. A thermal isolation vacuum chamber surrounds the resonator and seals against the bottom surface of the driver housing with an O-ring.

Electrical Subsystems

In order for the refrigerator to operate autonomously in space, a family of analog and digital electronic circuits are employed to monitor the "health" of the system, keep the driver running at the proper amplitude and frequency, and to acquire and store useful data for postflight analysis. The design and function of each of these electronic subsystems is documented in Ref. 14.

The circuit which is unique to this application is the resonance control board (RCB). Its function is to maintain the system at acoustical resonance and to control the amplitude of the acoustical pressure at predetermined levels dictated by the controller board. a, and therefore, the resonance frequency of the refrigerator, is a function of temperature:

$$a = \left(\frac{\partial p}{\partial \rho}\right)_s^{1/2} = \sqrt{\frac{\gamma R T}{m}} \qquad (6)$$

In Eq. (6) M is the mean molecular weight of the gas mixture. As the temperature changes, the sound speed changes, hence, for fixed resonator dimensions, the resonance frequency will be a function of temperature.

It is important that the system be maintained at resonance for two reasons. The first is that the resonance enhances the acoustic pressure generated within the resonator for a given velocity of the bellows. This resonant enhancement of the pressure is partially responsible for the fact that no sliding seals are required in the system since only small driver excursions are required to produce the necessary acoustical levels. The second reason for maintenance of the system on acoustic resonance is more subtle, but equally important. The values of the temperature span produced by the refrigerator and its heat pumping capacity are a strong function of the position of the stack with respect to the standing sound wave as demonstrated by Eq. (3). For a particular application, this stack position must remain fixed. Since the stack is not movable, the wavelength of the sound must remain constant, hence, the frequency of the sound must be varied in order to compensate for the changes in the sound speed with temperature described in Eq. (6).

The RCB maintains the resonance condition by comparing the relative phase of the microphone and accelerometer outputs. At resonance, the pressure and velocity of the gas mixture at the bellows location must be in-phase. Therefore, the pressure and acceleration are required to be in quadrature (i.e., 90-deg out-of-phase). The two signals are electronically multiplied together and the dc component of their product, which is proportional to the cosine of their phase difference, is used as an error voltage. This error voltage is electronically integrated and fed back to a voltage-controlled oscillator to close the phase-locked-loop circuit which maintains the entire system at resonance.

The acoustical amplitude is controlled by a similar feedback circuit within the RCB which compares the microphone output voltage to a set-point voltage determined by the Controller Board. The difference in the measured and set-point levels produces an error voltage which is electronically integrated and fed back to the automatic gain control input of the voltage controlled oscillator. Both of these control operations are accomplished using analog electronics.

Refrigerator Performance

The performance of STAR can be summarized by the two graphs which are presented here as Figs. 3 and 4. Those data were obtained when the refrigerator was filled with a 150 ± 1 psia mixture of helium (97.2%) and xenon (2.7%). The relative acoustic pressure amplitude, $p_1/p_m = 2.0\%$. Other measurements of the performance of the entire system, including the electrodynamic driver are included in Ref. 16. The performance shown in Figs. 3 and 4 is by no means the best which has been achieved in a thermoacoustic refrigerator of this style. Other improvements to the design, such as the use of a stack which has nonuniform spacing, has produced a single-stage, no-load temperature span of 118 K, even with-

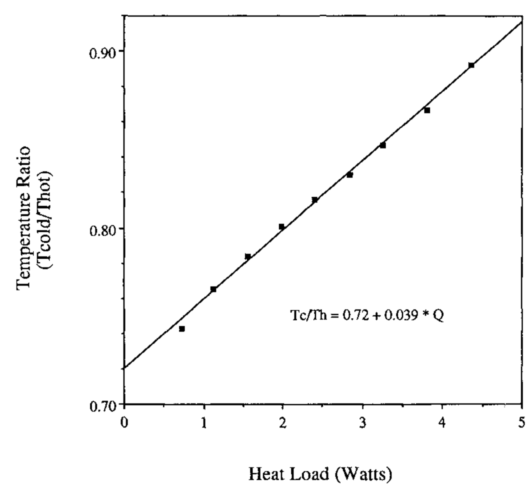

Fig. 3 Ratio of the cold and hot heat exchanger temperatures T_c/T_h as a function of heat load on the cold heat exchanger.

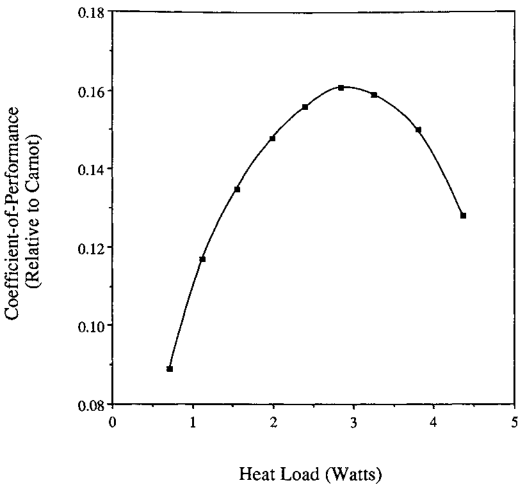

Fig. 4 Measured COP relative to the ideal Carnot COPR as a function of heat load.

out the use of gas mixtures to reduce viscous losses or improve the COP. COPR as large as 20% have been measured,[15] and calculations suggest that COPR ≈ 35% should be possible.[18]

Conclusions

This article has presented an introduction to the fundamentals of thermoacoustic heat transport and described the implementation of this new technology in an autonomous, space-qualified refrigerator. Based on the success of the STAR, two other spacecraft refrigeration systems are currently under development at the Naval Postgraduate School. One design, which should have a cooling capacity similar to a small home refrigerator/freezer[18] (200 W at 4°C and 120 W at −22°C), is intended for use on the Space Shuttle as a replacement for the existing refrigerator/freezer[19] used in life sciences experiments. The other is a space-based cryocooler which is designed to reach temperatures suitable for Hi-T_c superconductor experiments. The absence of ozone depleting chemical and greenhouse gases in conjunction with the simplicity of operation and fabrication suggest that thermoacoustics may have advantages for terrestrial commerical applications as well.

Acknowledgments

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