Experiments with an Intrinsically Irreversible Acoustic Heat Engine

John Wheatley, T. Hofler,[a] G. W. Swift, and A. Migliori
Los Alamos National Laboratory, University of California, Los Alamos, New Mexico 87545
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The general qualities of a type of thermodynamic engine that depends intrinsically for its operation on irreversible processes are set forth and demonstrated experimentally in the context of a thermoacoustic heat-pumping engine.

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When a stack of plates as in Fig. 1 is placed near the end of a closed tube containing ^{4}He gas and an oscillating pressure is applied to the gas with period comparable to the thermal relaxation time between gas and plates, starting with a uniform temperature throughout, the end C of the stack cools while the end H heats. This is a consequence of an acoustically stimulated average hydrodynamic flow of entropy in the gas from the end C to the end H of the stack. If the stack is weakly thermally coupled to its surroundings at C and H and otherwise thermally insulated, a limiting temperature distribution along the stack quickly results, roughly independent of the static and dynamic pressure of the gas and of the frequency. This temperature distribution can be stabilized by manipulating the thermal contacts at the ends. The temperature difference thus obtained is geometry dependent but independent of other engine parameters and can be very large compared with the adiabatic gas temperature oscillations resulting from the pressure oscillations. For a given temperature T_H at H, the limiting temperature distribution for zero heat transferred to a thermal reservoir at H bounds a region of generally lower temperature gradient in temperature-position space where work is used to effect the transfer of heat. There is another limiting temperature distribution, for the same T_H, corresponding to a rather larger temperature difference imposed on the stack which bounds a region of generally higher temperature gradient where heat is absorbed at H and work may be produced in the form of spontaneous oscillations. Related phenomena were investigated as early as the last century by Sondhauss[1] and discussed qualitatively by Lord Rayleigh[2] in the case of the spontaneous oscillations where heat is converted to work, and more recently by Gifford and Longsworth[3] with their "pulse-tube refrigerator," in the case where work is used to pump heat. Further background has been given by Rott,[4] whose thermoacoustic theory facilitates a quantitative interpretation of these phenomena, in particular the recent quantitative work on the spontaneous oscillations in ^{4}He gas by Yazaki, Tominaga, and Narahara.[5] In this Letter we present some of the central experimental characteristics and calculated properties of our acoustic heat-pumping engine. Although this engine is thermoacoustical in nature, we believe that it is just one manifestation of a quite general class of intrinsically irreversible thermodynamic engines which need bear no relation to acoustics and which can use working substances other than gases. We also propose some of the general features of such intrinsically irreversible engines.

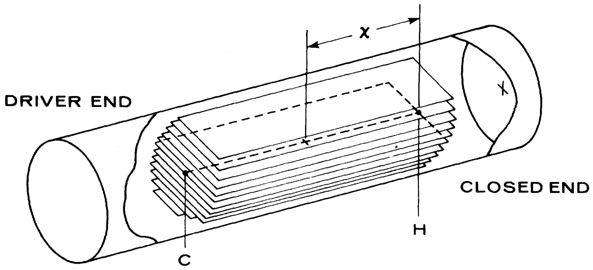

FIG. 1. Schematic of an intrinsically irreversible thermoacoustic engine.

Both the entropy flow central to the operation of this engine and the relationship to acoustics are demonstrated very nicely by a simple instrument which we call a thermoacoustic couple, shown in the inset to Fig. 2. It is just a short "engine" of length Δx very much less than the radian length λ of the sound and consisting of one or more solid plates. The central plate of the couple is fitted with a thermopile to measure $\Delta T = T_C - T_H$. This temperature difference can be used to measure entropy flow in the gas provided the longitudinal thermal conductance Σ of the couple is so high that it dominates all diffusive heat flows. In the presence of acoustic power, entropy flows out of one end of the plates, down along the plates in the gas, and back into the

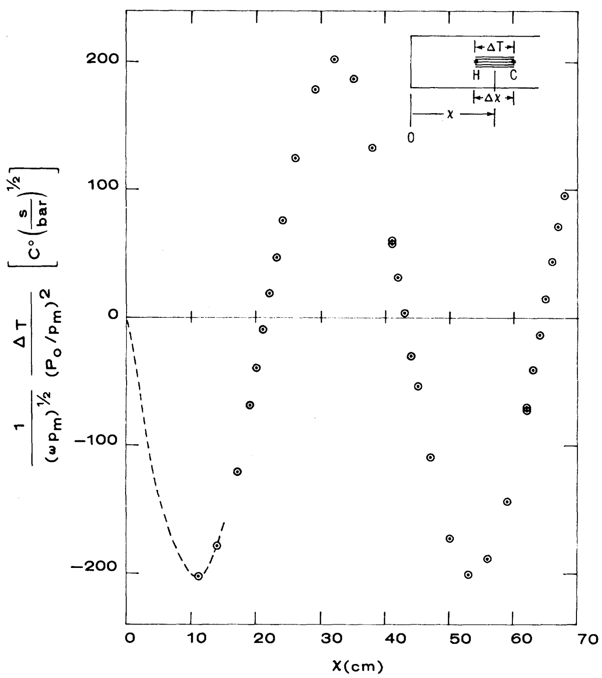

FIG. 2. Normalized thermoacoustic couple response $(p_m\omega)^{-1/2}\Delta T/(P_0/p_m)^2$ as a function of the average distance x to the couple from the closed end of a tube filled with ^{4}He gas at 2.55 bars and 22 °C and resonant at 1163 Hz. The normalization of ΔT gives a quantity which is independent of static and dynamic pressure and frequency and depends only on the geometry and constitution of the couple. The schematic geometry is shown in the inset: This couple had five parallel plates 2.0 cm long and 1.9 cm wide with a six-couple thermoelectric pile measuring ΔT across the central plate. Each plate was a composite of two 0.0125-cm-thick stainless-steel sheets with a 0.0125-cm-thick fiberglass sheet sandwiched between them. The plate separation is 0.114 cm while $\delta_\kappa = 0.014$ cm. The data were obtained with $P_0/p_m = 0.0025$.

other end. The corresponding second-order energy flow is, following Rott,[4]

$$\langle \Delta \dot{H} \rangle = \Pi \int dy\, \rho_m T_m \langle s_1 u_1 \rangle = \Pi \int dy\, \langle (\rho_m c_p T_1 - p_1) u_1 \rangle, \tag{1}$$

where Π is the surface area per unit length, or perimeter; y is the direction perpendicular to the plates; ρ_m and T_m are mean mass density and mean temperature; s_1, u_1, T_1, and p_1 are, respectively, the first-order entropy per unit mass, velocity, temperature, and pressure; c_p is the constant-pressure specific heat; and the angular brackets indicate a time average. In the boundary-layer approximation $(\rho_m c_p T_1 - p_1)$ is zero except in the thermal boundary layer of characteristic dimension $\delta_\kappa = (2\kappa/\omega)^{1/2}$ next to the plates, where κ is thermal diffusivity and ω is angular frequency. In the absence of the couple, $\langle \Delta \dot{H} \rangle$ would be zero in the space occupied by it. Evaluation of (1) for a large enough Σ that temperature gradients are adequately small gives, in the boundary-layer approximation and with the assumption that the phase between p_1 and u_1 is $\pi/2$ far from the plates,

$$\langle \Delta \dot{H} \rangle = -\tfrac{1}{4}\Pi p_a v_a \delta_\kappa \left(\frac{1+\sqrt{\sigma}}{1+\sigma}\right) = -\frac{\Pi}{4}\frac{\omega \delta_\kappa P_0^2}{\gamma p_m}\left(\frac{1+\sqrt{\sigma}}{1+\sigma}\right)\left(\frac{a}{2\omega}\right)\sin\left(\frac{2\omega}{a}x\right). \tag{2}$$

Here p_a and v_a are the pressure and velocity amplitudes far away from the plates, γ is the ratio c_p/c_v of specific heats, σ is the Prandtl number, p_m is the mean pressure, P_0 is the amplitude of the dynamic pressure measured at the closed end of the tube, a is the sound velocity, and x is the mean distance of the couple from the closed end of the tube. We assume that the response ΔT of the couple may be determined from

$$-\Sigma \Delta T + \langle \Delta \dot{H} \rangle = 0, \tag{3}$$

where $\langle \Delta \dot{H} \rangle$ is evaluated in the center of the couple. This assumes that the average energy flow down the couple in the gas in the thermal boundary layer is balanced by diffusive heat flow in the plates. An example of normalized thermoacoustic couple response is shown in Fig. 2, the couple support structure having prevented measurements near the closed end. If enough plates giving a high enough Σ are used there is quantitative agreement between the observations and Eqs. (2) and (3) for a broad range of P_0 and p_m, provided P_0/p_m is sufficiently small. The response is zero at both pressure and velocity antinodes; the hot end of the couple is always toward the nearest pressure antinode.

For a stack substantially longer than a thermoacoustic couple but still shorter than λ, the entropy flow in the gas will rapidly lead to a large temperature difference between H and C if longitudinal heat flow is small and thermal coupling at the ends is weak. We have studied this quality extensively; details will be presented elsewhere. Briefly, in the experiments the temperature at the closed end is maintained near ambient while the temperature at some distance from C toward the driver is controlled independently to be some value T_C'. Under these conditions acoustic power produces a temperature distribution in the stack which is "rigid" in the sense that, provided the longitudinal heat transfer is small, it does not depend strongly on the temperature of either the closed end or the controlled region. This temperature distribution changes by only a few percent as p_m and P_0^2 are changed by an order of magnitude and ω by a factor of 3. The controlled temperature T_C' can be adjusted to make T_H approximately ambient. For that condition we find that the normalized temperature distribution T/T_H along the stack depends primarily on the geometry, specifically on the ratio of the distance along the stack to the effective length of the open space between H and the closed end. It depends only weakly on changing the composition of the stack from longitudinal fiberglass parallel plates to longitudinally insulated *transverse* copper screens. Using a computational method based on the theory of Rott,[6,7] we find that the actual temperature distribution lies between those calculated for the two limiting cases $\langle \dot{Q}_H \rangle = 0$ and $\langle \dot{Q}_C \rangle = 0$, where these are the average heat transfer rates at H and C, respectively. The smallest value of T_C/T_H thus far observed for these conditions is about 0.53, corresponding to relative acoustic pressure amplitudes P_0/p_m of order 0.02 to 0.04. Such a large temperature difference suggests that useful devices might someday be built based on these principles.

We have not yet made measurements of the efficiency η of the acoustic engine. To develop intuition on this problem it is useful to consider the inviscid case $\sigma = 0$. For such an ideal intrinsically irreversible engine using an ideal gas, we find the result that η depends primarily on geometry and on γ rather than on absolute temperature. For an ideal irreversible magnetic engine of the present type and employing Curie-law materials, we find that η depends only on the configuration of magnetic materials in magnetic fields and the field themselves. If these ideal engines are operated near the limiting temperature distribution so that even at the nonzero frequency of the engine the heat-transfer rates approach zero, η can be shown by calculation to approach the Carnot value as an upper bound.

A few general features of this type of engine can be learned from the thermoacoustic engine. Referring to Fig. 1 our engine has a primary thermodynamic medium, ^{4}He gas, enclosed in a tube also containing a secondary thermodynamic medium, a stack of thin fiberglass plates, whose function is to exchange heat with the gas. The secondary medium should have a low longitudinal conductance to reduce longitudinal heat flows. The primary medium has a reciprocating motion with respect to the secondary medium; attendant on that motion is a thermodynamic effect, in this case a change of temperature of the gas caused by the pressure change. The processes involved in the contact between primary and secondary media must be irreversible; in the gas engine the most important process is the exchange of heat. This intrinsic irreversibility in the engine has the purpose of providing for a suitable phasing between motion and the thermodynamic effect. In the present engine there is no useful average energy flow either in the isothermal case, in which the spacing between plates is small compared with the thermal penetration depth, or in the adiabatic case, in which the spacing is large compared with the penetration depth. Finally, what is necessary to produce a cooling effect starting from zero longitudinal temperature gradient is that in some region there be an increase of energy flow in the direction of the energy flow. In the apparatus of Fig. 1 this is achieved where the plates begin on the driver side as a consequence of a rapid change there of the instantaneous heat transfer per unit length between the media. There is a corresponding heating effect at the other end of the plates. These effects are a consequence of what we call a "broken thermodynamic symmetry" between the media. If the energy flow $\langle \dot{H} \rangle$ is constant in space then we have thermodynamic symmetry. Referring for example to Eq. (2), which gives the entropy-flow part of $\langle \dot{H} \rangle$, the thermodynamic symmetry can be broken if any of the geometric quantities like Π, or thermal quantities like δ_κ, or dynamic quantities like p_a or v_a are spatially dependent. For engines in which gas is the primary medium, such as the present engine, the pulse tube of Gifford and Longsworth,[3] and the uniform-diameter resonance tube of Merkli and Thomann,[8] the above conditions are met; thermodynamic symmetry, however, is broken quite differently in each case. The traveling-wave acoustic Stirling engine of Ceperley[9] is intrinsically reversible and hence of a different type.

The possibilities for the thermodynamic media and configurations for the general type of thermodynamic engine described above are very broad. Essentially any materials for which an adequate thermodynamic effect can be achieved can in principle be used. It is not necessary for the media to be separated in space; interpenetrating media, as for example the electron and ion systems in a plasma, would be appropriate. The frequency of optimum operation is characterized by the reciprocal of the natural interaction times of the media and hence might vary from, say, 1 Hz for a magnetic engine to a very high frequency for a plasma engine.

Real irreversible heat engines have also been considered recently.[10-12] In this work the Carnot engine is central, and calculation of properties of real engines are made by explicitly introducing irreversibilities external to the Carnot engine. We would like to suggest that the ideal irreversible engine of the present type, in which the irreversibilities are an inseparable part of the engine's operation, and in which the thermodynamic cycle is determined "naturally" at a finite period, would be a suitable vehicle for theoretical investigation. We conjecture that configuration and certain physical properties of the media may be primarily important while the temperature spanned follows from the power output.

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[a]Also at: Department of Physics, University of California at San Diego, La Jolla, Cal. 92093.

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