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Philip Shien Spoor
Innovation Consortium

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PHILIP S. SPOOR

CFIC-QDrive,
Troy, NY, USA
Phone: (518) 272-3565 x208 Fax: (518) 272-3582 email: pspoor@qdrive.com

ABSTRACT
In 2006, CFIC Inc. completed a prototype thermoacoustic-Stirling refrigerator for the U.S. Army. Like most thermoacoustic devices, this one is “green,” i.e., CFC-free and oil-free with inert gas as the working fluid. Unlike other thermoacoustic prototypes known to us, this device is configured and used as an appliance, with the thermoacoustic hardware confined to a relatively small enclosure on top of a large cabinet (17 internal cubic feet) and the operation controlled by a single on/off switch and a thermostat. There are no secondary fluid loops for heat exchange, only heat pipes and fans. In early 2007, this fridge was put on long-term test as a beverage cooler, running continuously. As of this writing (7 May 2008) the fridge has surpassed 11,000 hours of continuous operation with no observed degradation of performance. Here we present some details of construction, test data, performance history, and paths for improving efficiency.

1. INTRODUCTION
Mounting concern about global warming has spurred increasing interest, both social and commercial, in alternative refrigeration methods. Thermoacoustic cooling, or the use of sound waves in an inert gas to pump heat, is attractive because it is environmentally benign and mechanically simple, and more efficient than thermoelectric refrigeration. Early work in this area was based on the “standing-wave” thermoacoustic effect, which is intrinsically irreversible—hence the early designs were fundamentally limited to an efficiency below Carnot’s ideal. However, Kees de Blok and Greg Swift independently realized that an acoustic refrigerator with slightly more complexity could be an embodiment of the Stirling cycle, without the mechanical displacer. A refrigerator based on this principle could in theory exceed the efficiency of a Rankine cycle or VC system, and with no mechanical displacer, would eliminate the technically most challenging and failure-prone element in more conventional Stirling systems. Our device is a realization of this concept, adapted to match the needs of a food storage refrigerator.

1.1 Army mobile kitchens create opportunity
Unlike the old days of K-rations, today’s U.S. Army troops in active combat get their meals from well-equipped mobile kitchens. Hot meals and refrigerated items are routine. This has highlighted one of the major drawbacks of VC refrigerators in severe service: they are not particularly robust against shock and vibration, being easily damaged in shipping or transport (due, in part, to the many unsupported braze joints in all the serpentine tubing). Recharging in the field requires special tools, training, certifications, and a supply of the proper refrigerant (there are many different refrigerants in use and in general they are not interchangeable.) The Army supported CFIC’s development of a more combat-hardy refrigerator prototype that uses inert gas, with the idea that a fleet of such refrigerators would be more reliable and cause fewer logistical difficulties than the commercially available VC fridges.

2. ACOUSTIC-STIRLING BASICS

2.1 Stirling Fundamentals
The Stirling cycle, named after Scottish inventor and pastor the Reverend Robert Stirling, has been studied and embodied in various forms since the early 1800’s. It is the basis of nearly all tactical cryocoolers, and Stirling devices are making inroads into power generation and CHP (combined heat-and power) applications. Nonetheless, the Stirling cycle is nowhere near as ubiquitous as the Otto, Rankine, or Brayton cycles. Therefore, many engineers have little or no familiarity with it from their coursework. If they have encountered it, it is usually in the form of a “pedagogical” Stirling cycle diagram showing the idealized cycle steps, such as Figure 1. The ideal Stirling cycle is comprised of two constant-temperature (isothermal) processes and two constant-volume (isochoric) processes. The processes are all reversible, so this cycle has the same fundamental efficiency as the Carnot cycle.

The simple embodiment shown on the right-hand side of Figure 1 helps to visualize the cycle. From step 1 to step 2, the gas is compressed; if we suppose that the first heat exchanger is in intimate enough thermal contact with the gas, then we can imagine that the heat of compression is removed isothermally. From step 2 to step 3 the gas is translated at constant volume through the regenerator (a fine porous structure, typically a stack of finely woven metal screens). The regenerator material has much higher thermal conductivity and heat capacity than the gas, so it can absorb the sensible heat of the gas nearly isothermally, and the gas comes out the cold end at $T_C$, the cold HX temperature. From steps 3 to 4 the gas is expanded, and its tendency to cool when expanded will allow it to accept heat from the cold HX. In the final steps, the gas is translated back through the regenerator, where its sensible heat is “regenerated” and it comes out the warm end at $T_H$. If the pistons are imagined moving sinusoidally rather than in a stepwise fashion, it is apparent that the Stirling cycle results from approximately 90 degrees of phase-shift between the two pistons.

From the point of view of the fluid, the means of enforcing the Stirling cycle on the thermal core (the two heat exchangers plus regenerator) is irrelevant. The pedagogical form shown in Figure 1 is a simple example of a so-called “alpha” configuration, where the two pistons share the functions of gas compression/rarefaction and displacement. It is equally feasible to configure a device with one piston that provides the compression/rarefaction (the “power piston”) and another that is connected via fluid paths to either side of the thermal core, and provides the displacement (the “displacer”). This is generally referred to as a “beta” configuration.

### 2.2 Stirling vs. Acoustic-Stirling

One may note that the motion of the fluid on either side of the thermal core (e.g. the motion of the pistons in the “alpha” configuration) corresponds to the relative motion of adjacent fluid particles in a traveling, or propagating, wave, e.g. a wave that carries momentum and energy. Thus one of the salient features of a Stirling cycle is that power flows into the thermal core (in the case of a cooler), some is used to drive the cycle, and the rest propagates onward. This outgoing work must be recycled in order to achieve high efficiency. In a “kinematic” Stirling machine, this feedback is achieved with mechanical linkages; in a “free-piston” cooler, i.e. a beta-style cooler with a free displacer, this work is fed back through the displacer path. A “slug” of gas in a duct can behave like a piston as well, so in principle, the displacer and the fluid paths connecting it to either side of the thermal core can be replaced by a simple duct. This duct serves as displacer and power feedback path, in an acoustically displaced Stirling, or acoustic-Stirling, cooler.
Figure 2 shows a schematic free-piston Stirling cooler, and its relationship to an acoustic-Stirling. Figure 2(a) shows the free-piston version, with a solid displacer in the fluid branch connecting the ends of the thermal core. The displacer has some intrinsic mass, and is typically connected to a spring (a mechanical spring or a gas spring). The combination is tuned to have a slightly higher resonance frequency than the operating frequency of the cooler, which results in a (tuneable) phase difference between the power piston motion and the displacer motion. In (b), the displacer is replaced by a plain duct (which may not be the same dimension as the displacer housing, as the figure suggests) and the movement of fluid in the duct is tuned, by choice of proper length and diameter, to enforce the Stirling cycle in the thermal core. The mass of gas in the duct, however, cannot be tuned to resonate above the operating frequency, as can the mechanical displacer. Instead of leading the motion of the power piston, the acoustic displacer will lag. The core efficiency can be the same either way, but this subtle phasing difference means that power flow through the core is reversed, and the cold end now appears at the end close to the power piston(s). Hence a fridge with purely acoustic displacement has to be topologically modified to create a salient cold zone, rather than an embedded one that would be hard to adapt to a refrigerator cabinet, as shown in (c).

Figure 2(c) is schematically very similar to the refrigerator we built for the Army; the cold end penetrates the roof of a cold storage cabinet, and finned heat-pipes with fans emanate from both heat exchangers. It also has pistons of relatively large diameter, compared to the pistons that might be used for an acoustic-Stirling cryocooler that required the same amount of input $pV$ work. This is another indirect consequence of acoustic displacement; the acoustic volume fluxes at the junction between the acoustic displacer / feedback path and the tube down the middle of the thermal core tend to be more additive than with mechanical displacement. Therefore, more flow, and more swept volume at the pistons, is required to support the cycle.

3. THE ARMY FRIDGE

3.1 Requirements
The Army specified a thermoacoustic refrigerator capable of cooling the interior of an Army-supplied 17 ft³ cabinet; the details were left largely to our discretion. Since we intended this prototype to resemble and operate like a product, we put a number of restrictions on our own design:

1. No CFC’s, HCFC’s, or hazardous substances anywhere (e.g. no fair using heat pipes filled with R134a!)
2. Operation at 60 Hz; no variable frequency drive (VFD) allowed.
3. No secondary fluid loops. (Originally we disallowed heat pipes as well, but the helium-to-air heat exchange was very challenging without them).
4. Compact size.
5. Reasonable efficiency.
6. Pushbutton operation; no maintenance required.
7. No intrinsically expensive components.
Because this was a first prototype, requirements 1 and 3 made the heat exchanger construction somewhat challenging. We chose finned heat pipes to carry heat from the internal helium cycle to the air; but finned heat-pipe assemblies are not easily purchased items, especially with custom fluids. We eventually chose to purchase bare heat pipes that were filled with water for the warm side and ethanol for the cold side, had a second vendor shrink-fit aluminum fins to the pipes, and then we soldered the pipes into the heat-exchanger blocks ourselves with low-melt solder. In a production situation, the heat exchanger blocks, with finned heat pipes, would be produced as single brazements; the heat pipes would be filled afterwards. But in our case, we were saddled with some extra temperature defects between the fins, pipes, and blocks that compromised performance.

Figure 3 shows a cutaway solid model of the final configuration of the Army fridge (minus cabinet and control module). Some of the named components can be identified with their counterparts in the schematic shown in Figure 2(c). The regenerator is made of stacked plastic screens; the heat exchangers are copper screens brazed to copper blocks, to which the heat pipes are soldered. There is an additional component not shown in the schematic, which is of particular interest in our work. The creation of a work loop to recirculate power also allows circulating steady flow, which carries warm helium to the cold end or vice versa. In a conventional Stirling, the displacer would provide a barrier to this flow, but here, we must create an alternative. It does not have to be a precision component like the displacer in a Stirling cooler; it merely has to allow the passage of acoustic power while suppressing steady flow. We have chosen a latex membrane sandwiched between two steel shim washers, with a fillet of flexible adhesive around the edges of the latex. There are no data available for the lifetime of such an object, so the long-term testing of this fridge is of particular interest. However, the rubber is in a dark, oxygen-free, inert gas atmosphere, so it should not experience degradation.
3.2 Bench test results
Before we arrived at the configuration in Figure 3, we tested an earlier configuration on the bench, without heat pipes, with the cold end in a smaller, insulated container. Electric heaters were applied directly to the cold end, and the warm end had a water jacket. This version was much easier to instrument than the final version. While the bench test configuration was not the most efficient, it allowed us to validate our thermacoustic simulation.

As Figure 4 shows, this bench-test unit had no trouble reaching an internal temperature span of over 80 C; at the highest ΔT, the cold helium was below –60 C. It could have reached lower temperatures, except the fluoropolymer seals in the cold end began to fail below –50 C. In the version installed in the cabinet, a lot of this temperature difference is used overcoming temperature defects in the heat exchangers; but nonetheless, it shows the ability of a fridge designed for a modest temperature difference to span a large one if desired, a feature of acoustic coolers. A “single-stage” acoustic cooler can easily reach temperatures below the coldest cascaded “ultra-low” vapor-compression refrigerator, for instance. The other interesting result is that the total heat load (applied heat load plus the heat leak into the insulated box) almost perfectly matches the simulated results. This says that we understand what we’re building, and can use the simulation tools to push the design in directions that improve efficiency.

3.3 Performance in Cabinet
After making some efficiency improvements as suggested by (DELTAE) simulations, and installing the fridge into the cabinet with the heat pipes, we took some more performance data and compared again with our simulations. The results are shown in Figure 5. The efficiency was distinctly improved (note that the power input in Figure 4 does not include any fan power, since it was a bench test with water cooling and no cabinet). However, the data do not agree as well with the simulation, perhaps reflecting the additional challenges of accurate calorimetry in the real-world installations, and perhaps some compromised internal performance due to uneven heat transfer in the heat-pipe heat exchangers. Our low-melt soldering left something to be desired, and some pipes were better bonded to the copper than others.

The overall COP of this fridge was about 0.5, at 0C in the cabinet, including fan and controller power. While this particular prototype will not be competitive with existing vapor-compression refrigerators, there are very clear paths to improving efficiency which will make future version competitive. Even more promising are the possibilities for acoustic-Stirling ultra-cold freezers, which promise to be more efficient than their cascaded VC counterparts, and can reach colder temperatures.
One of the key paths to higher efficiency can be seen from Figure 6, which shows sample temperatures taken in the helium, the HX block, the heat pipe surface, the heat pipe fin, and the air, for both warm and hot sides. The temperature defect between the helium and the air is on order of 15°C on the warm side, and over 25°C on the cold side for some points. This underscores the challenge of adapting Stirling-type devices to applications well-suited to vapor-compression—in a Stirling device, one does not have a circulating, high-enthalpy refrigerant, but localized hot and cold zones, and the heat must be spread over a larger surface area and transported into the air. Nonetheless, we believe we can do a lot better than Figure 6. The biggest single defect on the cold side is between the pipe and the block, which suggests that the pipe is just not well bonded to the block. This would be corrected by having the pipes brazed to the blocks at the same time as the internal screens. The biggest defect on the warm side is between the helium and the block, which suggests that there isn’t enough screen cross section bonded to the block to handle the heat flux. This will be improved by having a longer screen bed, or a different type of warm heat exchanger with more copper mass (like a stack of perforated copper plates brazed together.)

One advantage of an acoustic system is that the linear motors are inherently easy to modulate with voltage (like a loudspeaker) rather than with frequency (like a rotary motor). It is quite simple to install an inexpensive load-following circuit, like a dimmer switch with feedback, to make the fridge run at a constant maintenance level rather
than turn on and off in a duty cycle. This itself confers a time-averaged efficiency benefit not captured by a measurement at a single point. This will be explored further in the following section.

Figure 7. Views of the whole refrigerator, the thermoacoustic components on the top, and the interior of the cabinet.
4. LONG-TERM LIFE TEST

After performance tests were complete, the refrigerator (as shown in Figure 7) was stocked with beer and soda, and placed on long term testing, with a power meter and an hour meter. As mentioned in the previous section, this refrigerator had a load-following controller, via temperature feedback to an SCR, which allowed it to run at constant maintenance level, with small corrections rather than on/off duty cycling. As such, it consumed only twice as much electric power in a time-averaged way than our standard refrigerator in the break room upstairs, which has a much smaller cabinet.

Figure 8. Long-term test data of the Army acoustic-Stirling refrigerator.

Figure 8 shows the results of the long-term test, during which no maintenance was performed other than an occasional helium recharge, every 2 or 3 months (this prototype has many elastomer seals, and we suspect that the heat exchanger blocks themselves have some fissures in them.) We didn’t even perform a defrost, (although we probably should have.) Nonetheless, no change in performance is observed. When the building temperature goes down in December, dipping briefly to 60 F, the power follows, dropping to near 50 W. When the building temperature comes back up in the spring, the power follows, rising to an average of 75 watts. Note that the electric power numbers given are instantaneous values, not time averages, so there is some scatter—but the overall average is stationary. The temperature of the air in the cabinet (also instantaneous) varies by less than one degree F over the entire year, and usually by no more than 0.1 F for days at a time. This kind of temperature stability is a secondary effect of having nearly constant power, and makes this technology attractive for applications that demand consistent temperature.
5. CONCLUSIONS

Acoustic-Stirling cooling has been demonstrated in a configuration resembling a standard refrigerator, in everyday use for over a year with no maintenance other than topping off the helium charge. The flexing membrane element used to suppress Gedeon streaming has survived nearly 2½ billion cycles, which shows that properly designed, these elements are not prone to fatigue. This should encourage the pursuit of acoustic-Stirling products for the commercial marketplace, with the confidence that the technology is fundamentally sound. There are still challenges in meeting or exceeding the efficiency of conventional vapor-compression equipment, and in becoming cost-competitive. However, since this technology can easily reach –80°C and beyond without cascading, it may be that “ultra-low” freezers and similar near-cryogenic specialty products may be the best entry point, as the cost targets are higher and the acoustic coolers are fundamentally more efficient than existing vapor-compression in this temperature regime.

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