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THERMOACOUSTIC REFRIGERATOR

by

Nathan A. Hammond

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of

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Thermoacoustic Refrigerator

Final Report
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Left to right: Glenn Roth, Nathan Hammond, Nathan Holyoake, David Harris, Christine Merrill, Marriner Merrill, Matthew Morgan

Abstract

Thermoacoustic refrigerators have typically been designed to cool an isolated working fluid, necessitating a heat exchanging device to draw heat from the outside target fluid into the cooled working fluid. For example, a thermoacoustic refrigerator designed to chill air might utilize argon as an isolated working fluid, necessitating a heat exchanger to draw heat from the air to the argon. A second heat exchanger would be required to draw heat from the argon to a cold sink.

A design team at Utah State University has created a thermoacoustic refrigerator which uses air at atmospheric pressure as its working fluid. By selecting an open-open tube design with pressure nodes at both ends, the design team was able to impose airflow directly through the thermoacoustic stack. In this way, atmospheric air is cooled directly, obviating the need for a heat exchanger to operate between the air and an isolated working fluid.

1. Introduction to thermoacoustics

This thesis details an exploration into the relatively new field of thermoacoustics. This is a burgeoning field which exploits the natural relationship between acoustics and thermodynamics. As a sound wave propagates through a gas, each small parcel of gas experiences the same fluctuation in pressure that it might experience under a piston with sinusoidal motion. This change in pressure induces changes in the volume and temperature of the parcel. Additionally, the change in pressure across space causes motion of the parcel. Given a sound wave, a temperature and displacement cycle occurs naturally and may serve as the basis for a thermoacoustic refrigerator.

While thermoacoustic refrigerators have not yet begun to compete with conventional systems on the market, the future of thermoacoustics holds much promise due to the near absence of moving parts and the low tolerances required for manufacture. Also, thermoacoustic devices have the theoretical potential to reach Carnot Efficiency. Let us explore some of the basics of this promising field.

A sound wave is composed of velocity and pressure components, which operate in a manner analogous to the voltage and current behavior in an AC circuit. These velocity and pressure components oscillate either in-phase or out-of-phase at the same frequency and possess individual amplitudes. They are coupled traits, so both pressure and velocity are functions of each other, reacting to each other and readjusting rapidly to new conditions.

These velocity and pressure qualities make it possible to use sound waves to transport heat energy through a given working fluid, such as air. As oscillation occurs, the velocity portion of the wave carries with it a small parcel of the air. The position of the parcel of air is merely cycled between two locations. The distance between these stations is most strongly governed by the amplitude of the velocity component.

The pressure portion of the wave facilitates heat transfer to and from the parcel of air according to the ideal gas law. As the pressure increases, the air packet is rapidly compressed and its temperature rises. This temperature increase promotes heat transfer from the air parcel to its surroundings. Conversely, when the pressure drops, the air parcel expands, which lowers its temperature and promotes heat absorption. This process is repeated with each pressure oscillation.

It is possible to coordinate these velocity and pressure oscillations so that the parcel of air consistently absorbs heat at one location and then deposits it at the other. This results in a microscopic heat pump. To create a significant energy transfer, it is necessary to coordinate the effects of a great many of these air parcels. If these parcels are not combined into a common effort the vast majority will only transfer heat to each other. These parcels would continue to oscillate back and forth exchanging heat with each other, but never depositing it away from their system. This action results in no net heat transfer and is the reason why typical sound-bearing air does not suddenly change temperature. For heat transfer to occur, it is necessary to divide the air parcels into separate regions and provide them each with access to a surface area where they can exchange heat with an external medium. This is accomplished with a device known as a stack, one of the essential components of a thermoacoustic refrigerator or thermoacoustic generator.

A thermoacoustic refrigerator is a device which utilizes the properties of an acoustic wave to cool a target fluid. It typically comprises the following components:

Stack: The stack is a type of thermal regenerator. It has some degree of thermal capacity, and it has any number of surfaces oriented so that air can flow across them with relatively low impedance. Picture a series of sheets stacked parallel to one another, with gaps between each sheet to allow the passage of air. This arrangement keeps the heat moved by the packets of air from dissipating by holding the heat long enough to dump it to the next packet of air, facilitating a ‘bucket brigade’ effect. So a stack has the effect of generating a temperature gradient in the presence of a standing acoustic wave. Alternate arrangements for a stack include a section of honeycomb material, or a number of cylindrical shells, such as straws, stacked concentrically.

Speaker: The speaker is a device used to convert electrical power into acoustic power. It is through the speaker that energy is introduced into a thermoacoustic system.

Chamber: The chamber is an area of space with desirable acoustic properties. It has intrinsic natural frequencies, and sound waves at these frequencies will be amplified. The chamber provides an acoustic environment in which the stack may function to create a temperature gradient from acoustic power.

Heat exchangers: As a temperature gradient forms across the stack, one side of the stack will become warmer than ambient temperature while the other side drops below ambient temperature. Typically, two heat exchangers are used. The first transfers heat from the hot side of the stack into the environment. This indirectly lowers the temperature of the entire stack, including the cold side. A second heat exchanger transfers heat energy from the target fluid to the cool end of the stack.

Figures 1.a through 1.d illustrate how a thermoacoustic stack works. Figure 1.a shows the stack positioned in a sound chamber. In this instance, picture the chamber as a tube whose axis runs laterally across the page. A small parcel of air near the center of the stack is delineated.

Figure 1.b shows a close-up view of this air parcel. The lines above and below the parcel represent layers of the stack. Envision a sheet of paper viewed from the edge.

In the presence of a sound wave, the volume, pressure, and position of the parcel all change. In the case of a standing wave, the maximum and minimum pressures occur while the air parcel is displaced farthest from its original location. Figure 1.c shows the air parcel displaced a maximum distance to the right while pressure is at a minimum and volume is at a maximum. In this low-pressure state, the air parcel drops in temperature, and thermal energy is transferred from the stack into the air parcel.

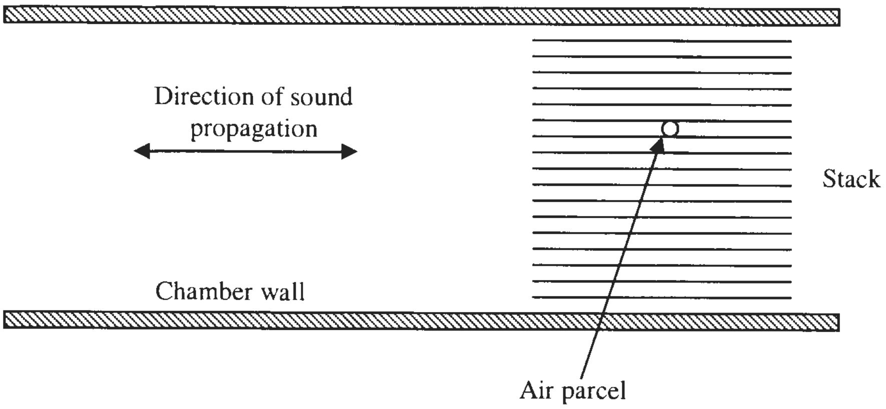

Figure 1.a: Parcel of in a stack positioned within a sound chamber.

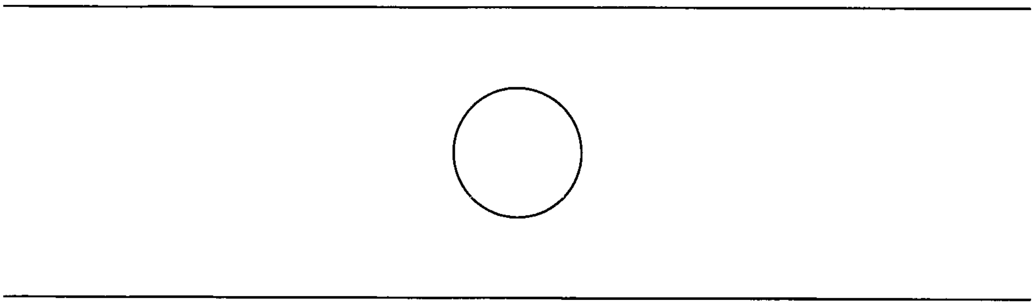

Figure 1.b: Close-up view of air parcel

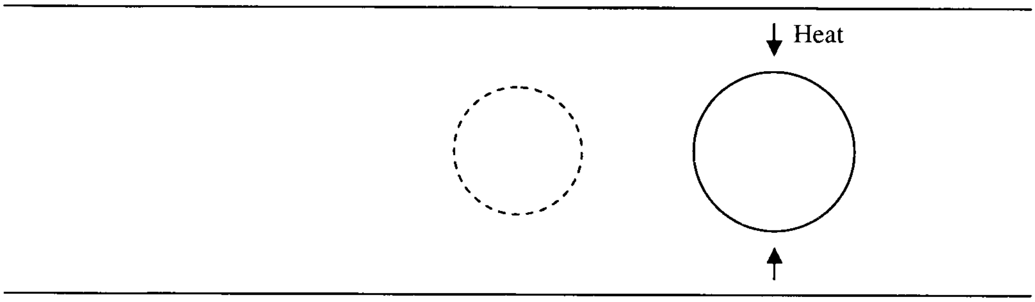

Figure 1.c: Expanded, displaced air parcel

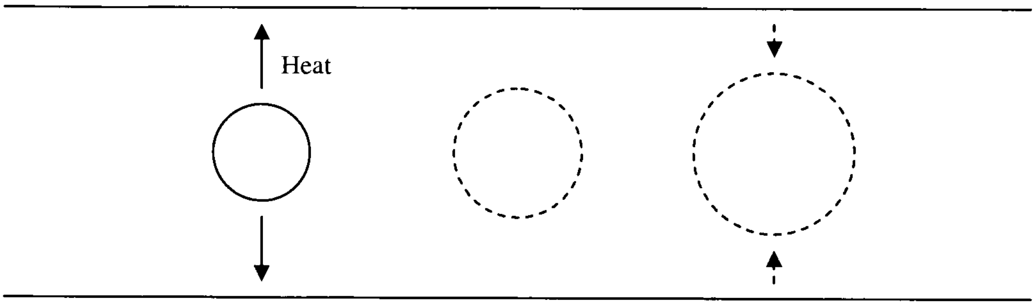

Figure 1.d: Compressed, displaced air parcel

Figure 1.d shows the parcel displaced a maximum distance to the left while pressure is at a maximum and volume is at a minimum. In this high-pressure state, the temperature of the air parcel is increased, and thermal energy is transferred from the air parcel to the stack.

This process is repeated as long as a standing wave is sustained. The net effect is an increase in temperature on one side of the stack and a decrease in temperature on the other side. In the case depicted, the right side of the stack will be relatively cool and the left side relatively hot.

Thermoacoustic refrigerators have typically been designed to cool an isolated working fluid, necessitating a heat exchanging device to draw heat from the outside target fluid into the cooled working fluid. For example, a thermoacoustic refrigerator designed to chill air might utilize argon as an isolated working fluid, necessitating a heat exchanger to draw heat from the air to the argon. A second heat exchanger would be required to draw heat from the argon to a cold sink.

A design team at Utah State University sought to design and build a thermoacoustic refrigerator which uses air at atmospheric pressure as its working fluid. In this way, atmospheric air would be cooled directly, obviating the need for a heat exchanger to operate between the air and an isolated working fluid. This report documents the team's design process, describes the refrigerator constructed by the team, and presents conclusions drawn from analysis and experimentation.

2. Instrumentation

In order to successfully characterize any system, key variables have to be identified and accurately quantified. In building a thermoacoustic refrigerator, one such variable is pressure, since accurate pressure measurements are necessary to determine wave magnitude and identify pressure nodes and antinodes. Another key measurement is temperature, since system performance is dependent on the formation of a temperature gradient across the stack. It was necessary, therefore, to find or develop appropriate instrumentation for the measurement of pressure and temperature at a point. Other variables were less challenging to accurately measure and were dealt with on an ad hoc basis (e.g. a simple ruler could be used to measure the placement of the stack within the chamber.

Thermocouples, when used properly, can provide an accurate temperature measurement. In addition, the low profile of the thermocouple wire allows temperature measurement with minimal disruption of system performance.

Type T thermocouples were selected because of their accuracy (approximately 1 degree Celsius) over the range of temperatures that were measured. The wire was cut to specified lengths, stripped, and twisted together. The twisted junction was placed in a welder for a few seconds until a spherical junction resulted. The wires were tested against existing thermocouples for consistency.

For pressure measurement, an active pressure sensor was used. This came in the economically viable form of a small Radio Shack microphone. Somewhat bulkier than the thermocouples, the microphone offered the benefits of accuracy and minimal system disruption to a lesser extent. However, this sensor was deemed the best option, due to reasonable accuracy and low cost.

3. Conceptual design

As previously mentioned, the primary challenge of the Utah State University engineering team was to design and build a thermoacoustic refrigerator which would cool atmospheric air directly, without the aid of a cold heat exchanger. This required that air be allowed to flow through the chamber, eliminating the option of a traditional sealed-chamber design. The team had to find a way to place openings in the chamber while still maintaining the desired acoustic properties.

To meet this requirement, the team opted for a chamber which was open at both ends. Resonance could be achieved by placing the pressure nodes of the wave exactly at the open ends of the chamber.

Since the chamber would be roughly symmetrical, it would be possible to place a pressure node at each end of the chamber if the speakers were located near the center. Some compensation would have to be made since the impedance of the stack and the heat exchanger would appear only on one side of the chamber, but this would have to be dealt with later in the design process.

Figure 2 shows the basic refrigerator design. Air would be forced into the chamber at one end (left). In the opposite side of the chamber (right-center) the stack would create a temperature gradient which was hot on one side (left), cold on the other (right). A heat exchanger would be placed on the hot side, where it would use an ambient-temperature medium to cool the warm side of the stack. On the cool side of the stack, no heat exchanger would be necessary, since the air cooled by the stack would exit the chamber as the end product of refrigeration.

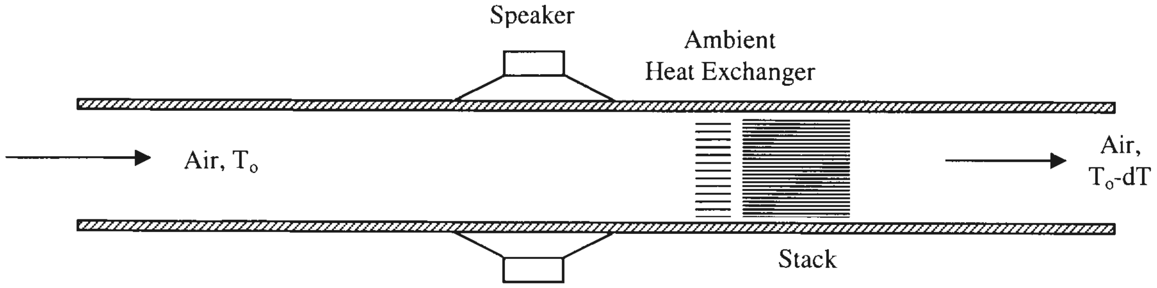

Figure 2: High-level design of a direct-cooling thermoacoustic refrigerator

With a broad concept of the design in mind, components of the system could be engineered more or less individually. Each component would be tested and optimized until the entire thermoacoustic refrigerator was assembled. At that time, further testing could be conducted on the system as a whole.

4. Analysis tool: DeltaE

Thermoacoustic analysis can be quite complicated. Systems with any degree of complexity are poorly represented by explicitly solved models, and all useful analysis involves the implicit solution of the differential equations.

One tool which helps perform this analysis is DeltaE, a program written by the Los Alamos National Laboratory engineer Greg Swift. This program allows the user to enter thermoacoustic systems in terms of simple components (round ducts, branches, heat exchangers, etc.). DeltaE then uses an indirect solution technique to find steady-stead properties of the system.

Throughout the design process, DeltaE was like the elusive pot of gold at the end of the rainbow, whose properties were desperately desired by the team of engineers. In the end, however, this great hope turned out to be merely a mirage. In spite of considerable effort, no reasonable solution was ever obtained for the system being analyzed.

The failure to achieve meaningful results from DeltaE placed an even higher emphasis on experimental results. Key system parameters could be optimized only with considerable testing.

5. Speakers

Speaker selection was probably the single biggest key to any success that was gained. In thermoacoustic cooling systems that have already been created, speakers have been the source of a lot of work and frustration. The difficulty is that the speakers have to be cooled. In a conventional audio system the speakers are self-cooling. They have vents in the back, and as the speaker cone moves in and out a jet is formed that circulates air past the electric coil (the part that tends to heat up the most), keeping the system cool.

However, in most thermoacoustic applications, the effect of having a constant frequency generated is that the speakers tend to heat up past their capacity to cool. A second problem is that conventional thermoacoustic systems use a gas at a pressure much higher than atmospheric, so the speakers must be enclosed in the pressurized system to provide the cone with equal pressure on both sides. This rules out unassisted convection cooling. The final problem is that even if the working fluid is not kept at a high pressure, most speakers have porous cones, and large amounts of energy are lost through the speakers when they are playing through an impeding circuit. Since the sound waves and energy follow the path of least resistance they will tend to go back through the cone rather than through the stack and heat exchangers. This also necessitates some sort of casing which negates the normal cooling mechanism.

Clearly, speaker cooling can become a very difficult problem. In one thermoacoustic cooler built at Los Alamos, first air cooling, than liquid cooling and finally a combination were used to cool the speakers. In fact, the cooling apparatus became much larger and more complex than the actual thermoacoustic device!

Since the goal of this project was to form a flow-through thermoacoustic cooler, there were no pressure constraints on the speaker. Therefore, the goal was to find a speaker with a non-porous cone. A set of Alpine speakers was found that not only had a non-porous cone, but also a very open body to enhance cooling on the back. Tests bore out the effectiveness of using this simpler speaker system. The speakers are probably one of the most effective parts of the system.

6. Chamber

Much of the research and analysis performed by the design team involved chamber properties. The most important area of research was open-ended tube acoustics. This included the prediction of the chamber's resonant frequency (this was a key parameter for building a stack) and the characterization of acoustic impedance in an open-end tube (an understanding of this was prerequisite to making an accurate model in DeltaE).

The earliest acoustic study was really a feasibility search. For the refrigerator's stack to function as intended, a standing wave had to be formed. Elementary physics dictates that a standing wave can be formed in a pipe which is open at both sides. However, a question arose as to whether or not a standing wave could be maintained while air flow was superimposed. This problem was resolved by observing the example of an organ pipe. As air is pumped through the pipe and past a vane, the resulting oscillatory disturbances are distilled into a standing wave which forms at the pipe's resonant frequency. All this occurs while air continuously flows up through the pipe. This example was accepted as proof that a standing wave can exist with a superimposed air flow.

Some explanation needs to be made about some of the acoustic principles that were learned. This is best done as a sort of glossary of terms. First, a "standing wave" is a wave in which pressure and velocity are 90 degrees out of phase with each other. This forces displacements to be in phase or 180 degrees out of phase with pressure. Also, a standing wave exists naturally inside pipes where it bounces off either closed or open ends without losing any energy. The mechanical equivalent of this wave is a perfect spring.

If a standing wave is a spring, a traveling wave is a box sliding across a frictionless surface with no forces acting on it. A traveling wave is the sound wave that occurs when a sound is made at one location and propagates away from the source with no losses or reflection. This could be shown by an infinite pipe with a piston oscillating at one end. In a traveling wave the pressure and velocity oscillations are in phase, so areas of maximum pressure oscillation are also areas of maximum velocity oscillation. At the same time the displacements are 90 degrees out of phase with these two. Basically, this means that where pressure oscillates the most, the gas displaces the least. This means that instead of using a stack, it is better to use a regenerator (which has smaller, more tortuous passages than a stack) which has better thermal properties, but would be impractical in a standing wave due to viscous losses. When a regenerator is used, the thermoacoustic system goes through the same processes as the Stirling cycle, and therefore could reach Carnot Efficiency.

While there are processes which utilize a standing wave design and processes that utilize a traveling wave design, in a real system there is no such thing as a pure "standing" or "traveling" wave condition, there is always a mix. The design team's goal was to approximately form a standing wave. In the models developed at Los Alamos National Laboratory, a standing wave allowed for a simpler model and worked with a stack.

Another term is "resonance." Resonance occurs when sound waves are reflected within a definite space. The result is twofold. First, the system can be driven so that the sound waves have a wavelength that matches the dimensions of the space in which the waves are reflected. These waves build on each other and are amplified. Second, these resonated waves form into standing waves with pressure and velocity more or less 90 degrees out of phase.

In a perfect open-open pipe with a perfect sound wave, only standing waves would endure, due to the boundary conditions. However, in a driven system, any frequency can be played through an open-open pipe. There are two reasons that not every frequency will work for a thermoacoustic device. The first is that a stack can only function if the air oscillations and pressure variations occur in phase. This occurs only in a standing wave. The second is that at resonance, more sound is reflected back into the pipe at the open end so losses are lower.

Assume a packet of sound travels from the speaker towards the open end. At the open end, the packet of sound is reflected back up the pipe toward the speakers. At some point, it will meet the next packet of sound from the speakers. Under the most ideal conditions, these two packets will be of the same sign and magnitude and will therefore add to each other. That is a "resonant" condition. The mechanical parallel would be that the base of an ideal spring can be vibrated in such a way as to amplify existing vibrations or to damp them out. In acoustics, the goal is the amplified state if resonance is desired.

The two previous definitions become very complex very quickly. The two-speaker, open-open system contains a very mixed traveling/standing wave. It also has many resonant frequencies. For the design team, there was no real way to further predict exact acoustic results. This is primarily because there is no way to get a closed form solution of the effect caused by the speakers. In testing, standing wave conditions were assumed to be at resonance, and resonance was assumed to be a local maximum in the pressure oscillations while doing a sweep of usable frequencies.

The chamber dimensions were actually the first decisions made in the project. Since all of the various length and frequency scales were interrelated, at some point, one had to be chosen, and then the others designed to fit it. Two-inch pipe was chosen because it was easily accessible, and the length was chosen to be 1m because that would give some room to work with and yet not be too unwieldy. As the design progressed, the device gradually became shorter. This was done to try to match the most effective frequencies for the speaker with the lowest resonant frequency of a certain length of pipe. Then, in the final round of testing, it was found that as the pipe was shortened, efficiency increased. Due to a lack of time, these iterations of shortening the pipe were stopped before the maximally efficient length was found.

7. Stack

In order to predict an approximate stack spacing an acoustic guess was needed. To find the resonant frequency, it is known that:

$$f = \frac{c}{\lambda} \quad \textbf{(1)}$$

λ is the wavelength, f is frequency, and c is the speed of sound. The speed of sound c is determined by the equation:

$$c = \sqrt{kRT} \quad \textbf{(2)}$$

The variable k is the ratio of specific heat at constant pressure C_p to specific heat at constant velocity C_v; R is the gas constant, and T is temperature. Assuming room temperature and a medium of air, the speed of sound c is approximately 343 m/s.

In order to find frequency, an estimation of the wavelength in the open-ended chamber was still needed. At the most simplified level, an open end provides a boundary condition of zero pressure variation and maximum velocity variation. This is most easily explained by looking at the logical counterpart: the closed end. At the closed end of a pipe, the velocity of the oscillating air is forced to go to zero. At the same time, the pressure first increases to a maximum and then decreases to a minimum as the air oscillates toward and away from the closed end. At the open end, the opposite course is true. The open end provides compliance to the sound wave, keeping the pressure constant while allowing velocity to fluctuate. This means that for an open-ended pipe there is a pressure node and a velocity antinode at both ends.

A standing wave can be approximated as a sine wave. As with a sine wave, one complete wave contains 2 antinodes and two nodes. So, it is generally assumed that the lowest harmonic of a standing acoustic wave in a pipe contains only half of a wavelength.

Some approximations were made to the above model to get a prediction of the pipe's resonant frequency. First, an effective length was used. While in the ideal case a pipe has length ½ λ, impedance at the end of an open-ended pipe distorts the relation, and an actual pipe strikes resonance with a length a little less than ½ λ. So, on the suggestion of *Fundamentals of Acoustics* (Coppens et al 2000, p. 174), a length of 0.6r (where r is the pipe radius) was added on to the pipe length for each open end, giving an effective length to be used in calculating the resonant wavelength. The other approximation made concerning length was that the branches from the speakers would be treated as though they were linear with the rest of the pipe. With these assumptions,

$$\lambda = 2\, L_{effective} \tag{3}$$

$$L_{effective} = L_{original} + L_{branch} + 2(0.6\, r) \tag{4}$$

The final large approximation was that the speakers would force a pressure node to occur in the center of the pipe. So, instead of λ being two times the length, λ was set equal to the length. With all of this,

$$f = \frac{c}{\lambda} = \frac{c}{L_{effective}} \tag{5}$$

Assuming a meter long pipe, $r = 2.54$ cm, $L_{original} = 1$ m, and $L_{branch} = 10$ cm, the frequency was predicted to be 303 Hz.

Testing later revealed the assumption of a forced pressure node at the center of the chamber to be completely erroneous. The true boundary conditions for a speaker are very complex, and solutions have the greatest chance of being accurate a long ways away from the cylinder face. Still that guess was made, and it was experimental testing that ultimately set the numbers straight. Figure 3 shows the anticipated waveform and the wave that was actually measured. The x-axis represents displacement along the length of the tube while the y-axis represents maximum pressure amplitude at a given point.

Actual observations were made of an 80-cm pipe with $r = 2.54$ cm and $L_{branch} = 3$ cm. The fundamental frequency seems to have been 238-Hz. The primary reason for the discrepancy was that the speakers did not create a pressure node in the pipe, but a local antinode.

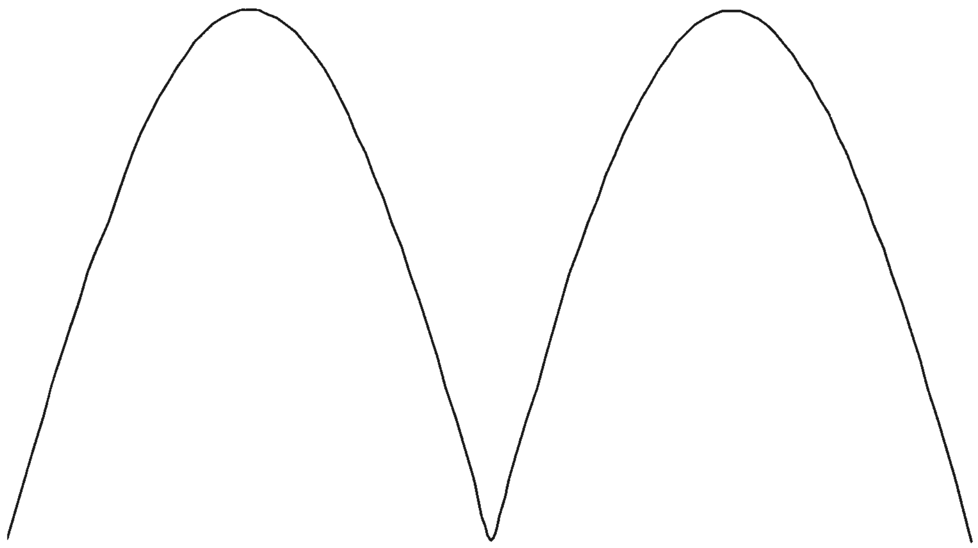

Figure 3.a: Anticipated wave form

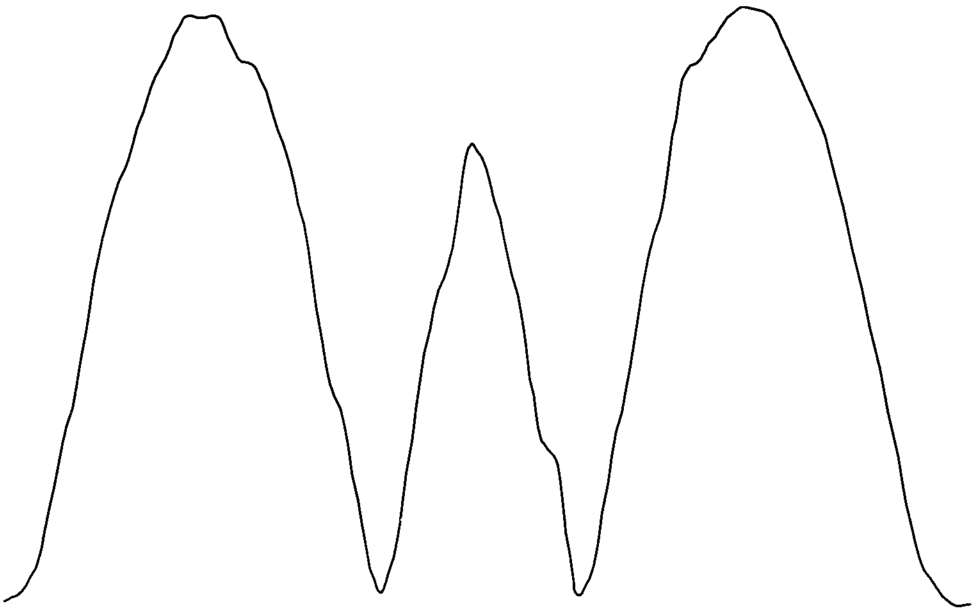

Figure 3.b: Measured wave form

Since parts were being built in parallel, the initial calculation of 300-Hz was used to size the stack. Swift states that the optimum spacing in the stack is related to the thermal boundary layer (2002). The thermal boundary layer is:

$$\delta_k = \sqrt{\frac{2k}{2\pi f}} \qquad \textbf{(6)}$$

where k is the thermal diffusivity. For air at room conditions, this gives δ_k as 0.142-mm. Swift recommends that the stack spacing be a few times larger than the thermal boundary layer, so the approximate goal for stack spacing was 0.5-mm.

The stack is the heart of a thermoacoustic refrigerator. It is there that the temperature gradient is formed, allowing the medium to drop below room conditions. In order for the stack to function, it must have several key parameters. First, the number of plates in the stack increases the efficiency. It is possible to have a stack with only one plate; however, it would be very inefficient. Second, the plates cannot be too close together. If there is perfect thermal contact between the plate and the oscillating gas, the gas will constantly exchange heat with the walls of the stack, and so never be hotter than the stack or cooler than it. These temperature variations at the extremes of the displacements are vital to creating a temperature gradient. A less important quality is that the stack has to have sufficient thermal properties to be able to receive the heat from the air and dump it back to the next packet of air. Given the importance of the stack, perhaps it is not surprising that it turned out to be difficult to build as well. The final product was the result of dozens of hours of hand crafting with specially made tooling.

This posed a severe challenge for optimizing the design. While the Swift suggested an appropriate spacing for the stack, it was really intended as an initial guess to be refined through testing (2001). Swift also suggested a stack length on the order of the tube diameter, but again, this guideline was intended to be used as a starting point for experimentation. Given the difficulty of constructing a stack, it was not feasible to create ten stacks of progressively increasing length, or another ten stacks with varying distance between layers. A compromise had to be reached.

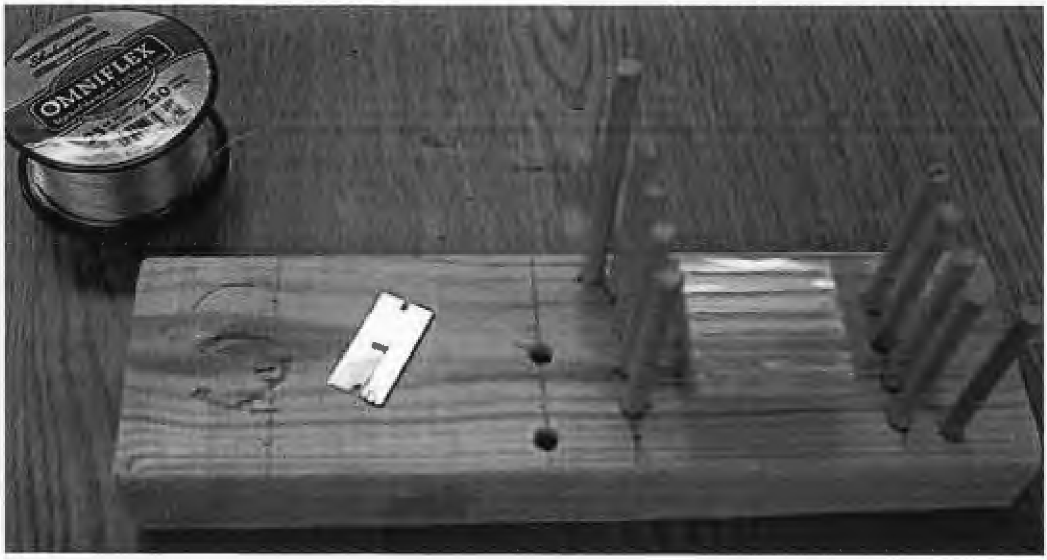

Figure 4: Structure used to maintain strings parallel during stack construction.

It was decided that the stack would be built with a spacing of approximately 0.5 mm, and if acceptable results were achieved, this spacing would be kept. As for stack length, an easier-to-build counterpart was found for testing. By bundling coffee straws together, the inner and outer surfaces of the straws would provide a surface across which the thermal gradient could form.

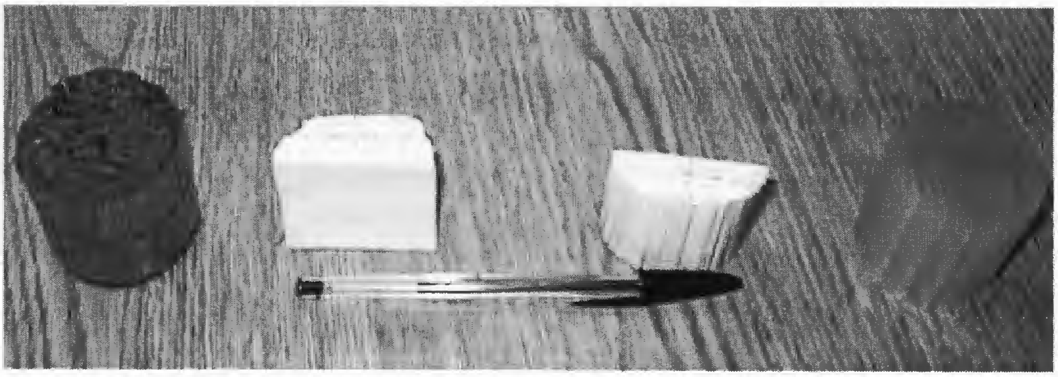

Figure 5: Stacks made of straws, paper (disassembled), and transparency sheets

These straws could be trimmed, allowing testing across a range of lengths. The most effective of these lengths would be used in the final design.

Cutting the straw stacks to different lengths proved practicable. However, the temperature gradient was so small that no reliable data could be collected. Then the first paper stack was built using two inch paper strips spaced by fishing line. Since this stack was very efficient, the same length was used for all future stacks.

Three parallel-sheet stacks were built. The stack which worked best, and which was ultimately used, was constructed from photocopier transparencies and fishing line. Not only did this prove to be more effective than paper, but it also was in keeping with the design team's goal of utilizing transparent components to allow maximum visibility of the finished product.

8. Heat exchanger

The heat exchanger went through several iterations. In the earliest phase of the design, different ideas were brought up, including using a cooled stack approach where the stack is directly cooled without the use of an extra heat exchanger. This was ruled out due to difficulty of construction. Then, according to the step-by-step plan of development, heat exchanger design was postponed until there was a working stack. When it was time for the heat exchanger to be built, the requirements were that the device be easy to build, be air cooled, and be easy to add into the currently working thermoacoustic pipe design.

The first real heat exchanger was designed to be a group of fins stretching horizontally through the pipe, sticking out the walls and spreading out into the air. A simplified diagram is shown in Figure 6.

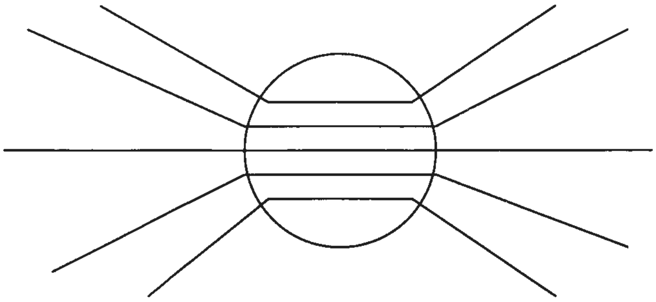

Figure 6: Early heat exchanger design

This design was seemed to offer great simplicity of construction, since it only needed thin fins of aluminum that went all the way through the pipe. It also would be air cooled and seemed to promise an efficient way to get the heat from the pipe center to the ambient air. In practice, this became much more difficult. First, the number of fins was severely limited by the ability to cut slots into the pipe for the ½ inch fins. The best that could be done only used 12 fins. Second, the fins didn't seem to pick up heat from the hot side of the stack. This process is supposed to occur by the oscillations of the sound waves causing lumps of air to pick up heat from the hot side of the stack and transfer it to the heat exchanger. It may be that the amplitude of the oscillations was too low due to the higher operating frequency and lower pressure used by this thermoacoustic device in comparison to others in operation.

The final design of the heat exchanger involved increasing the surface area inside the pipe and increasing the cooling abilities of the fins. This meant that one of the earlier design criteria was revoked and water was used. This last heat exchanger was made purely out of copper allow the fixture to be brazed. Brazing was used to hold the heat exchanger together, eliminate air leaks, and provide greater heat conduction between the corrugated metal plates and the copper tubes carrying the room temperature water. A piece of an old car radiator that was about 1/2 inch thick provided the increased surface area on the inside of the pipe. Then there was a long section of 1/8 inch copper tubing that was wrapped around the cylinder and inserted through to cross 4 times through the cylinder. These tubes were then brazed to the radiator surface. The body of the heat exchanger was a copper cylinder that fit over the pipe. Because the side of the heat exchanger with the water running through it was the cooler side, this is the side that was put next to the hot side of the stack. This heat exchanger was far more efficient than the earlier designs, however, in the final tests it is clear that the internal temperature of the pipe was steadily rising, so further refinements should be made.

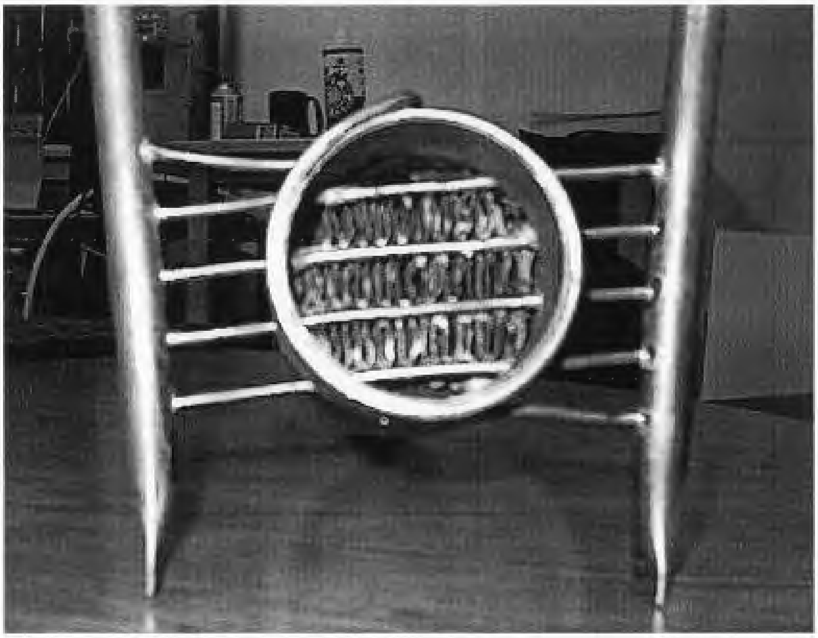

Figure 7: Final heat exchanger

9. Sound cancellation

Very early into the project, it was apparent that the sound wave amplitude would have to be very high in order to produce sufficient cooling, yet a sound level unsafe for human ears could be produced with just a small amount of power. Therefore, it became clear that some sort of sound dampening would be required for an effective system that could be safely operated.

There are four generally used methods to reduce sound levels: sound absorption, sealing of speakers, active sound cancellation, and Helmholtz resonators. All of these ideas were considered in an effort to reduce sound levels while maintaining cooling power.

Sound absorption occurs when a sound wave is caused to pass through a thick, porous medium, such as foam. As the wave passes through the material, energy is lost through friction. This type of sound reduction lends itself to higher frequencies, because of their short wavelengths. It is very poor for frequencies under 300 Hz. Some companies have designed and manufactured foams, called bass traps, that are effective at low frequencies. This type of foam is much more expensive than high frequency absorbing foam

Sealing the speakers is an effective method for all frequencies. This eliminates wave propagation through the back of the speakers. For this particular project, however, this method presents severe disadvantages. If the speakers are sealed, heat generated by the speakers has no way to escape. Because this would effect considerable heating of the air in a system designed for cooling, this was not considered a viable option.

Active sound cancellation is a fairly new method for eliminating sound. This results when the exact anti-wave is produced, and the two waves annihilate each other through interference. The anti-wave is fairly easy to produce. A speaker is simply wired opposite to another speaker. This method has been effective in many small applications.

Helmholtz resonators are often used for sound amplification in acoustic settings, but may also be used to reduce noise. For example, automobile mufflers are based on the principle of the Helmholtz resonator. The resonator is a cavity, build with a size appropriate to the frequency of interest. The frequency is governed by the following equation (Coppens 2000, p. 185):

$$\omega = c\sqrt{\frac{S}{L_{eff}V}} \tag{7}$$

where c is the speed of sound, S is the area of the neck, L_{eff} is $L_{measured} + 1.7a$, where a is the radius of the pipe, and V is the volume (ie physical size) of the resonator.

As sound passes through a properly tuned resonator, waves of the desired frequency will become trapped inside. By placing a sound absorbing material in the resonator, the sound can be eliminated.

The major obstacle faced in building a Helmholtz resonator or sound absorption device was the absorptivity of available materials. Most commonly available materials have low acoustic absorptivity, and even commonly available foams are poor absorbers at low frequencies. A Helmholtz resonator was tested with absorbing materials of low and high density Styrofoam, quilt batting, egg carton foam, and house insulation. Of these, the most effective was the high density Styrofoam, followed by the house insulation. However, these only resulted in a drop from 85 dB to 80 dB. Since this drop was too small to be noticed by people in the area, the resonator was not considered successful.

Active sound cancellation was tested from two different approaches. In one approach, the resonator tube was bent into a U-shape, so that the sound was emanating from both ends at approximately the same location. This resulted in sound cancellation in the region very close to the tube ends, but had no positive effect farther away from the tube. A conclusion made from testing this approach was that the geometry of the tube has little effect in the efficiency or operation of the cooler. In the other approach, the anti-wave was produced with two speakers separate from those attached to the cooler, and these anti-wave speakers were positioned at the tube endings or behind the main speakers. These had the most successful effect of any sound reduction method, with sound levels around the room decreasing about 7 dB. However, this was not adequate to meet the needs of the project.

The major obstacle to active sound cancellation is the open nature of the cooler. Because the speakers and the tube are all open to the atmosphere, there are 4 locations from which sound is emanating. Because the ends of the tube or the back of the speakers could not be sealed or blocked, the sound permeated the room very quickly. The best solution to this problem would be to build the room as a low frequency sound absorber, or a Helmholtz resonator, thus reducing sound in the room without restricting the airflow to the cooler. This approach would be beyond the scope of the project.

10. Final testing

After the design and development of the system, extensive testing was done to optimize system components and try to superimpose an airflow. Over the Christmas holidays, the system was tested to find optimum stack position and superimpose an airflow. However, the testing proved to be very difficult. It was particularly difficult to get consistent readings from the thermocouples. The result was that qualitative results were found for chamber length and the heat exchanger was integrated into the system. Also, any airflow that was superimposed threw the temperature measurements off to such a degree that this idea was abandoned for the project.

In January, it was decided that one more set of data needed to be gathered in a more accurate manner. This meant that there had to be 6 simultaneous temperature readings that were all referenced to a steady reference. To solve this problem, a computer data acquisition program was written using Labview. It was possible to write a program that allowed six thermocouples to simultaneously transmit data to the computer. This was referenced to room temperature with a built in temperature reference in the National Instruments PCB-100 junction box that was used.

With this setup, the final tests were run. Due to time constraints in the lab, they were all run in one day and there was no effort to optimize. The purpose of these tests was to collect data that could be trusted. The results of the three tests are summarized on Table 1. Equations for calculating the coefficient of performance are included as Appendix C.

A chart of test two is included as Figure 8. The flow of water through the heat exchanger was not started until partly through the test. This can be seen by the sudden cooling in all values.

Table 1: Results of the final tests.

Maximum Temperature Difference	18.35C (33.03F) Test 3, January 15, 2003
Lowest temperature (Room temperature = 22C, 72F)	14.81C (58.66F) Test 3, January 15, 2003
Maximum Cooling	7.4C (13.32F) Test 3, January 15, 2003
Heat Out	25.24W
Work In (excluding the pump)	15.16W
Efficiency (COP)	0.67

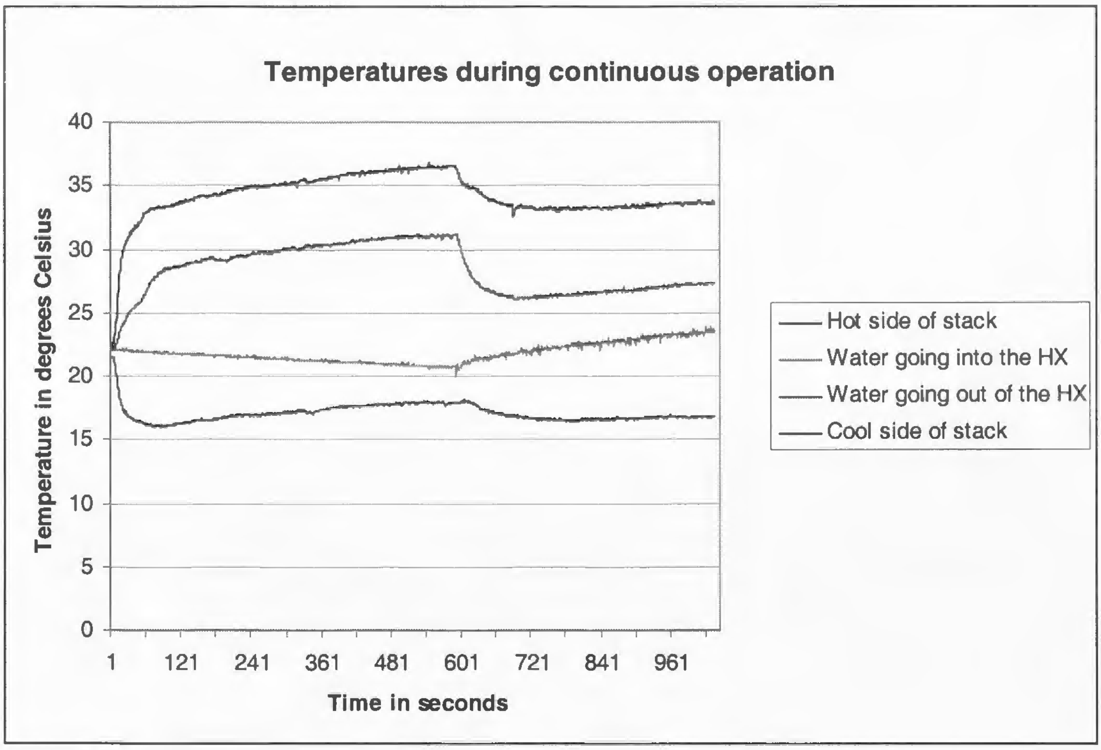

Figure 8: System temperatures

These tests were also not perfect. The high degree of complexity in the system made such a goal unrealistic. However, these results did yield quantitative data about the efficiency of the stack, the heat exchanger, and the system as a whole. If anything, the data collected is a slight under-representation of the efficiency of the system.

11. Conclusion

A quick retrospection of the past five months of research reveals a number of highlights: The elation when a bundle of coffee straws, a set of stereo speakers, and some PVC pipes produced a refrigeration effect, the astonishment when a higher-quality stack raised that effect to 40 degrees Fahrenheit, the pride of presenting this project at the State Capital, the sense of accomplishment when final data were collected and the project neared conclusion.

The project was successful in many ways. A working thermoacoustic refrigerator was built for future students to learn from and to improve upon. All seven members of the team now have a working understanding with the field of thermoacoustics. A benchmark has been set for future projects in this area, and with the work of future students this standard will surely continue to rise.

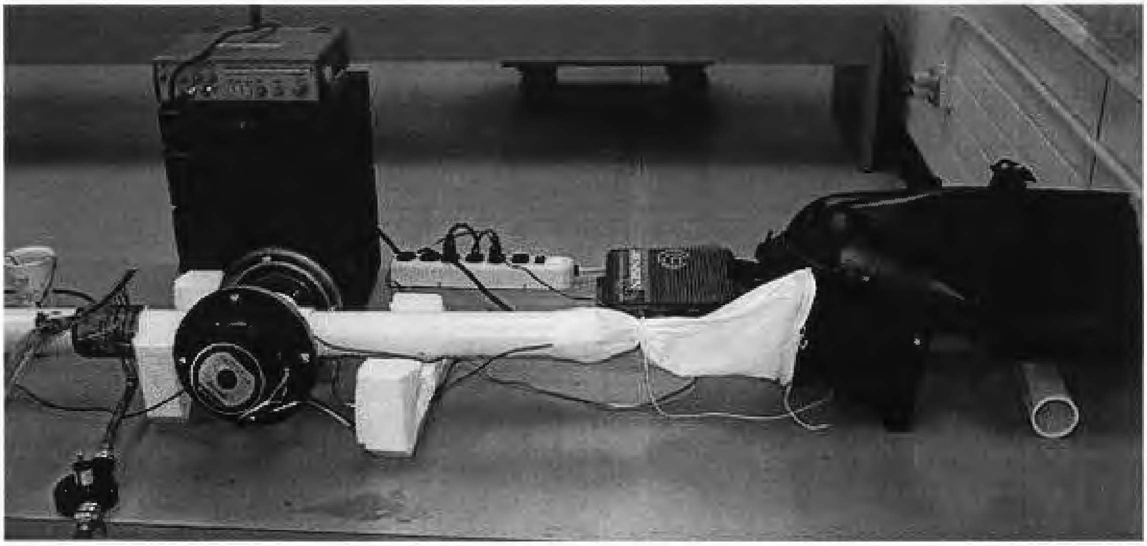

Figure 9: Final thermoacoustic refrigerator and auxiliary systems

A significant failure accompanied the project as well. While a temperature drop of 40 degrees Fahrenheit was achieved for static air, the introduction of even a slight airflow drops this gradient to nearly zero. The refrigerating power of the system which was built is too small for most practical applications.

It is possible that the use of atmospheric-pressure air limited the success of our project, since pressure fluctuations accompanying an acoustic wave are much smaller when the average temperature is low. It is thought less likely that turbulent flow was the source of this problem, but this, too, is a possibility. Another area in which much work is needed is that of sound cancellation. Before practical application of thermoacoustic products can be a reality, the sound must be reduced to an acceptable level.

It is the sincere hope of the design team that students who wish to further research in this field will build on our successes and overcome our failures. The field of thermoacoustics holds amazing promise. Devices utilizing this technology will be durable and low-maintenance, with no moving parts. Low tolerances will drive the cost of manufacture far below that of comparable products. With theoretical efficiencies approaching Carnot, only the lack of practical knowledge stands in the way of achieving incredible performance. Continued research in this field is sure to drive it to the forefront of industrial and scientific applications.

Works Cited

Swift, Greg. *Thermoacoustics: A unifying perspective for some engines and refrigerators.* Retrieved August 26, 2001, from: http://www.lanl.gov/projects/thermoacoustics/Book/index.html.

Coppens, A.B.; Frey, A.R.; Kinsler, L.E.; Sanders, J.V.. *Fundamentals of Acoustics*, Fourth Edition. NewYork, 2000.

Appendix A: Personnel

Nathan Hammond

Work Experience

- Biomechanical Engineer Intern, Medicine Lodge, Logan, UT. I aided research and development engineers in orthopedic product development, especially dealing with minimally invasive joint replacement techniques.

- Manufacturing/Mechanical Engineer Intern, Edwards Research Medical. Worked in manufacture of polyurethane balloons. Developed process improvements estimated to save $1.2 M over the next five years.

- Calculus Tutor, University Math Lab. Apart from solidifying my math skills, working as a tutor was helpful in teaching me to convey complex ideas in a way that can be understood by anyone.

Course Work

Advanced Dynamics
Fluid Mechanics
Vibrations
Persuasive Writing
Thermodynamics
Manufacturing Processes
Machine Design
Numerical Methods
Instrumentation
Material Science

Education

- BS in Mechanical Engineering, minor in Mathematics, Utah State University, May 2003. GPA 3.80

Skills

- Drafting experience in ProEngineer, AutoCAD, and EasyCAD.
- Speak, read, and write Spanish, with six months foreign residency.

David Harris

Work Experience

- General Atomics, San Diego, CA. Assigned to work in the Advanced Division, on electromagnetic projects. Development of a linear motor for launching aircraft

- Teaching Assistant, IDEAS Engineering Graphics, Utah State University. Instructing students in IDEAS and related research. Programming and design of graphic models, prototypes and unique designs.

- Visteon Corporation, Dearborn, MI. Worked on a forward model vehicle with systems engineers in Climate Control. Analyzed system variables and compared designs with test data. Evaluated as consistently demonstrating leadership, surpassing expectations and was deemed a role model for integrity. Granted University credit for my technical report and presentation on climate systems.

- Gilson Engineering, Riverdale, UT. Acted as on-site Construction Manager/Inspector on a $6M bridge construction project

Education

- BS in Mechanical and Aerospace Engineering, Utah State University, May 2003.

Course Work

Material Science
Strength of Materials
Manufacturing Processes
Fluid Dynamics
Statics
Mechanics of Solids
Machine Design
Thermodynamics
Dynamics
IDEAS
Fortran 90/95
Numerical

Projects

- Modeled gravity-controlled flow through a pipe with considerations of minor losses, friction, and pressure drop. Calculated the velocity/time distribution within the pipe using a discretized form of the energy equation.

- Composed a Fortran90 program to model and analyze the temperature distribution on a fin by using the Gauss-Seidel iterative method.

- Modeled temperature distribution versus time on a vehicle window cross-section by calculating and performing a Fortran90 algorithm.

- Managed a four-member team on a reverse engineering project, based on a cassette tape player.

Skills

- Proficient in IDEAS, Microsoft Office, Fortran, Mathcad, UNIX, etc.
- Extensive experience in automotive mechanic and electronic work
- In depth understanding of automotive design and racecar technology
- Familiar with A/C System modeling
- Background in fluid analysis (incl. Pipe flow)
- Familiar with system and cycle modeling
- Background in physics

Honors

- Dean's List
- University Honor Roll
- Presidential Scholarship Recipient
- Awarded for Academic Excellence in MAE
- Awarded the "A" pin for academic achievement
- Golden Key National Honor Society
- National Society of Collegiate Scholars

Nathan Holyoak

Work Experience

- NK Repair. Owner of Automotive repair facility. Performed repairs to collision-damaged vehicles including such aspects as frame damage analysis and working with specialized materials such as composite body panels and high strength steel.

- N.P. Auto Inc. President and 55 % owner of Automotive repair facility. Repaired import and domestic makes and models. Specialized in on board diagnostics and computer controlled systems.

- State Farm Insurance. Appraised damaged vehicles to determine if repairable or total loss. Prepared written estimate for repairable vehicles. Worked closely with collision repair facilities throughout the repair process. Resolved conflicts with claimants and insured's.

Education

- MS in Mechanical Engineering, Utah State University, May 2004.
- BS in Mechanical Engineering, Utah State University, May 2003. GPA 3.45
- Salt Lake Community College, graduate in Auto Collision Repair and paint. GPA 3.65
- Nextlevel Business development graduate
- I-CAR certified (in all eight areas as well as plastic and aluminum welding)
- ASE certified for HVAC (including R-12)
- Graduate of Academy of Finance
- State certified in Accounting and Data Processing

Course Work

Advanced Dynamics
Fluid Mechanics
Vibrations
Thermodynamics
Manufacturing Processes
Machine Design
Numerical Methods
Instrumentation
Material Science

Skills

- Self-motivated
  - Job experience–NK Repair and NP Auto; I started both businesses (I sold NP Auto for profit after 6 months)
  - Education–Utah State University
  - Personal Accomplishments–Explorer Scouting; Organizing the first leadership training program for adults that explorer scouting had.

- Understanding of Mechanical Engineering techniques and practices
  - Education–BS from USU in May 2003

- Working with people and projects:
  - Job Experience–State Farm Insurance (I worked on an estimatics team of 4 people)
  - Personal Accomplishments–Unity Four rocket, ASME member

- Finance
  - Job Experience–NK Repair, N.P. Auto
  - Education–Academy of Finance, State Certificates, Nextlevel (Academy of Finance accepts 90 out of 1200 applicants)

Christine Merrill

Work Experience

- Research Assistant, Department of Mechanical Engineering, Utah State University. Conducted research in the field of microscopic fluid flow/PIV. Designed flow measurement apparatus, adapted flow analysis code, analyzed data.

- Teacher's Assistant, Department of Mechanical Engineering, Utah State University. Was responsible for grading papers, recording and computing grades, and assisting student in Thermodynamics. Developed professionalism and dependability; used self-motivation to perform work.

- Packager, Mity-Lite Chairs, Orem, Utah. Cleaned, packed, and prepared chairs for shipping; checked for quality. Developed team and leadership skills. Used creative ability to assist engineers with product improvement.

Education

- BS in Mechanical Engineering, Utah State University, May 2003. GPA 3.5

Skills

- I-DEAS
- Fortran programming.
- Numerical Methods, including finite difference techniques
- Advanced Dynamics
- Thermodynamics
- Fluid Mechanics
- Solid Mechanics
- Instrumentation and Measurement
- Machine Design
- Project Management
- Fluent in Reading, Writing, and Speaking Tagalog (Filipino)

Honors

- Presidential Scholar
- Member of Golden Key National Honor Society
- University Honors Program

Marriner Merrill

Work Experience

- Research Assistant. College of Engineering, USU. Worked with Dr. Folkman to develop pipe-testing equipment. Learned about design and instrumentation building a large, parallel-plate pipe crusher.

Projects

- Computer solution of stresses in a plate using simple Gauss-Seidel methods
- Developed an iterative simulation for pipe flow with minor losses
- A thermoacoustic Engine (primarily a research project)

Education

- B.S. in Mechanical Engineering (Solid Mechanics Emphasis), Utah State University, May 2003. GPA 3.91.

Course Work

- Plates and Shells
- Vibrations
- Advanced Dynamics
- Design I
- Material Science, Mechanics of Materials, Strengths of Materials
- Western European Politics (communication abilities and political understanding)

Skills

- Learned Finite Difference Methods in 3 Heat and Mass Transfer Projects
- Successfully completed more than 13 other projects using Fortran or Excel in Fluid Flow, 3-D Modeling and Finite Element Analysis
- Exposure to I-DEAS through a class and work experience (200 hrs)
- Able to Read, Write, and Speak Russian

Honors

- Member of Golden Key National Honor Society
- University Club Scholarship
- Academic Excellence in MAE

Matthew Morgan

Work Experience

- Computer Lab Consultant, Utah State University.. Aid students with questions and maintain lab machines.
- Customer Assistance, Convergys (Nokia), Salt Lake City, UT. Addressed consumer questions and gave technical support on cell phone usage.
- West Virginia University, Morgantown, WV, Work-study. Clerical work, including typing expenditures, filing, faxing, etc. Gave aid to MAE professors with office and technical support.

Education

- BS in Mechanical and Aerospace Engineering, Utah State University, May 2003.

Coursework

Material Science
Strengths of Materials
Manufacturing Processes
Instrumentation
Thermodynamics
Statics
Mechanics of Solids
Machine Design
Fluid Mechanics
Numerical Analysis
Dynamics
Matlab
Pro/e

Projects

- Composed a Fortran90 program to model and analyze the temperature distribution on a fin by using the Gauss-Seidel iterative method.

- Modeled temperature distribution versus time on a vehicle window cross-section by calculating and performing a Fortran90 algorithm.

- Managed a four-member team on a reverse engineering project, based on a computer mouse.

- Designed a supersonic diamond-shaped airfoil for optimum lift/drag.

Skills

- Fluent in speaking, reading and writing Spanish

- Proficient in digital cinematography, Excel, C++, HTML, word processing and management software, Office XP Suite, Matlab 6.0 and Pro/e 2000i

Honors

- Brigham Young University, Provo, UT
  - Half-tuition scholarship, 2000.
- West Virginia University, Morgantown, WV
  - NASA Space Grant Consortium, 1999.
  - Valedictorian, 1996 and 1999
  - Elks, 1996

  - McConnell, 1996 and 1999

Glenn Roth

Work Experience

- SCP Global Technologies. Provided ProEngineer Technical help for company employees. Instructed courses on ProductView.

- Summer 2000, Idaho National Engineering and Environmental Lab (INEEL). Aided in the design of a factory-assembled, transportable nuclear reactor. Performed 3-D thermal fluid flow analysis of the reactor using an analysis code. Provided 2-D drafting pictures of the reactor for concept analysis. Presented results in a paper at the 2000 RELAP5 International Users seminar.

- Summer 1999, INEEL. Conducted extensive research into the effects of micro-gravity on fluid flow. Studied needed design changes for a nuclear reactor system to be functional in space.

- Computer Consultant.

- Math and Physics tutor.

- Teacher assistant for Mechanical Engineering Department.

Education

- M.S. in Mechanical and Aerospace Engineering, Utah State University, May 2003.
- B.S. in Mechanical and Aerospace Engineering, Utah State University, expected May 2003. 3.413 GPA.

Publications

- Using RELAP5-3D to Design a Small, Factory-Built Reactor: Proceedings of International RELAP5 Users Seminar, Jackson Hole, Wyo. Sept. 12-14, 2000

Projects

- Human Powered Vehicle Competition: Tested multiple seat configurations, fairing, and frame designs for a tandem bicycle to be entered in the HPV Challenge race

- NASA Vomit Comet Experiment: Part of a student team to conduct an experiment on early planetary formations in simulated microgravity.

Skills

- Proficient in: Windows, Macintosh, FORTRAN, ProEngineer, SolidEdge, and Microsoft Office.
- Working knowledge of: Unix operating systems, C++, Mathcad, and Solid Works.

Appendix B: Expenses

Table B.1: Project expenses

Item	**Cost**	**Purpose**
Quilt Batting	$10.00	Sound damping
Mattress pad	$17.00	Sound damping
Speakers	$200	System speakers
Amplifier	Donated	Signal amplification
Fishing line	$5.00	Stack construction
Dowell	$1.00	Stack construction
Glue	$4.00	Stack construction
PVC Glue	$3.00	Chamber construction
PVC pipe	$4.58	Chamber construction
PVC Tee	$3.11	Chamber construction
10-32x1.5" nuts and bolts	$1.72	Speaker mount
#10 flat washers	$0.86	Speaker mount
2" Flexible Coupling	$3.97	Chamber construction
2" PVC Coupling	$0.69	Chamber construction
2" PVC Tee	$3.11	Chamber construction
(2) 2" PVC 45 deg Elbow	$2.54	Chamber construction
(2) 2" PVC 90 deg Elbow	$2.56	Chamber construction
2"x10' PVC Pipe	$4.36	Chamber construction
Epoxy-5 minute	$2.97	Stack construction
Plastix Bonder	$3.34	Stack construction
Plastix Bonder	$3.34	Stack construction
Epoxy-90 minute	$2.77	Stack construction
RCA Y-plug, 6'	$0.99	Signal input
BNC phono adapter	$3.99	Signal input
Electret microphone, wired	$2.99	Sensor
Electret microphone, board mount	$1.99	Sensor
Write-on Transparency Film	$3.98	Stack construction
PVC pipe glue	$2.74	Chamber construction
PVC pipe cleaner	$2.74	Chamber construction
Dowel Rod	$0.52	Stack construction
Epoxy-2 hr	$1.97	Stack construction
Epoxy-Plastic Welder	$2.27	Stack construction
General Purpose Glue	$4.97	Stack construction
Super Glue	$2.47	Stack construction
Fishing Line	$1.74	Stack construction
Epoxy-2 hr	$1.97	Stack construction
Total	**$315.25**	

Appendix C: Coefficient of performance calculations

System testing provided a value of temperature change of the water passing through the heat exchanger and the mass flow rate of the pump.

$$T_{hot}-T_{cool} = 0.1255\ \mathrm{C} \qquad \textbf{(C.1)}$$

$$\dot{m}_{pump} = 0.048\, {}^{kg}\!/_{s} \qquad \textbf{(C.2)}$$

For specific heat of water, a value of C_p = 4180-J/kg C was used. This gave the total heat removed from the system as:

$$\dot{Q} = \Delta T * \dot{m} * C_{p,water} = 25.18W \qquad \textbf{(C.3)}$$

The power into the speakers was the only power included in the overall efficiency. This is based on the hope that the pump for the heat exchanger is a temporary, and not a permanent part of air-cooling thermoacoustic devices. The current into the speakers was measured to be 16-V, 2.08-A for each each speaker. The power to each speaker is given by:

$$P_{onespea\,\mathrm{ker}} = \frac{1}{2} I^2 Z \cos(\theta) \qquad \textbf{(C.4)}$$

Using the measured values of I = 2.08-A and θ = -12°, and the manufacturer-provided value of Z=3.58-Ω, the power provided to a single speaker was calculated to be $P_{onespeaker} = 7.58$-W.

Taking into account both speakers, we find:

$$COP = \frac{Q_{out}}{W_{in}} - 1 = \frac{25.18W}{2 * 7.58W} - 1 = 0.67 \tag{C.5}$$