Design, Construction and Operation of a Traveling-Wave Pulse Tube Refrigerator

S. Rotundo, G. Hughel, A. Rebarchak, Y. Lin and B. Farouk

Mechanical Engineering and Mechanics Department
Drexel University
Philadelphia, PA 19104

ABSTRACT

A low cost traveling-wave pulse tube refrigerator was designed, built, and successfully operated. The main goal of the project was to understand the working principles of such a device and attempt to build and operate the system with atmospheric air as the working medium. We used the thermoacoustic modeling software *DeltaE* effectively to design the air operated system with well defined constraints and performance parameters. The designed pulse tube components were fabricated based upon the *DeltaE* results and solid modeling layouts. A jig-saw was reconfigured to serve as the compressor to provide oscillating gas flow. The refrigerator was found to cool room temperature air down to about -45°C a difference of about 75°C between hot and cold regions of the pulse tube. This result is quite remarkable considering the facts that an ingenious drive for the compressor was used and that atmospheric air was used as the working medium (instead of high pressure helium). Additional tests were done using a unique regenerator comprised of stainless steel and nylon screens to help reduce the axial conduction within the regenerator. The refrigerator was then able to reach a temperature of about -60°C. This is an improvement over traditional regenerators that only use stainless screens. For the attainment of cryogenic temperatures, high power density in the pulse tube is essential. Our system does however, operate along the same principles and will exhibit similar behavior, thus allowing extrapolation of what effects or improvements might be expected in a high pressure helium system. Future tests are planned with pressurized helium as the working medium.

INTRODUCTION

The objective of the project was to build a functional low cost pulse tube refrigerator for the purpose of conducting research and giving demonstrations. In a pulse tube refrigerator, an acoustic driver (essentially a reciprocating piston) produces low frequency pressure oscillations that travel through three heat exchangers, a regenerator, a pulse tube, and a device to tune flow impedance. These pressure oscillations cause a reciprocating mass flow and together they cause an energy separation across the pulse tube and regenerator. By taking advantage of this energy separation with heat exchangers, such a system can be used in single-stage form to pump heat across very large temperature gradients thereby reaching even cryogenic temperatures. The compact size, simplicity, lack of harmful materials, and reliability have made pulse tube refrigerators indispensable for many applications including cell tower superconductor cooling and oxygen liquefaction in space. The

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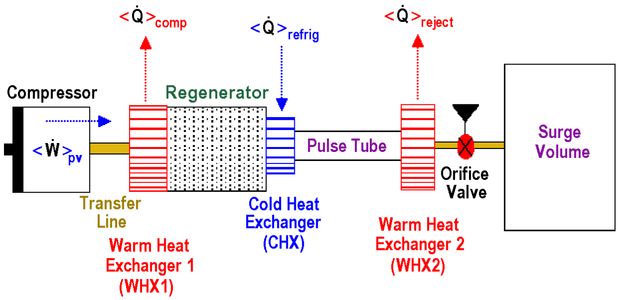

Figure 1. Schematic configuration of the orifice pulse tube refrigerator (OPTC) [1]

pulse tube, which is nothing more than a hollow tube, replaces the displacer found in a Stirling engine. Instead of a solid displacer, the volume of gas in the center of the pulse tube must oscillate back and fourth like a plug without ever contacting the hot and cold heat exchangers located on the pulse tube's ends.

Numerical simulations were conducted to aid the design and construction of a working pulse tube refrigerator complete with the necessary equipment for data acquisition and collection. For safety reasons, as well as to maintain a degree of simplicity, the design is constrained to using lower average system pressures and use only air as the working fluid. These two constraints keep our system from achieving the power density of some commercial units.

DESIGN APPROACH

The initial stages of our design phase focused on identifying and individually addressing all of the elements that would be necessary in order to successfully complete our objective. Based on our research it was clear that our design (for an orifice pulse tube refrigerator) would need to include a pressure wave generator, several heat exchangers for transferring heat into and out of the working fluid, a regenerator to temporarily store heat between the pressure oscillations, and a pulse tube in which energy separation within the working fluid could occur (see Figure 1).

Component Selection and Mechanical Design

Several approaches were considered for the design of each component. The approaches were then reviewed based on their ability to perform their intended role in the system, their ability to replicate our numerical simulation, and also for factors such as ease of fabrication, cost, and likelihood of complications. DeltaE [2] simulation results were used for the sizing of each component. In addition to geometric constraints, our theoretical model also provided us with ideas for material specification.

Heat Exchangers

By experimenting with different heat exchanger designs in DeltaE, we found it was beneficial to maximize radial heat conduction between the exchanger center and periphery while minimizing gas volume contributions. In addition to that, it was important to minimize the pressure drop across them due to flow restriction. Ideal values of conductivity and porosity were then used as targets which led to the selection of stacked 100 tpi woven copper screen disks with an opening percentage of 30%. The disks were stacked in a copper tube of inner diameter smaller than that of the disks. Disc press-in was chosen over solder-in because of its ease as well as the tendency for solder to be drawn into the center of the screen thus clogging it. In the case of the hot heat exchangers, heat transferred from the working fluid to the screen, then to the copper wall, and was then transferred to a soldered-on coil of 1/8” copper tube fed by a constant supply of water. In the case of the cold heat exchanger, the outer surface of the tube was left bare so that it could be insulated for “no-load” testing, or heated for “known load” testing, or used to conduct heat from and thereby cool an object.

Regenerator

An approach similar to that taken with the design of heat exchangers was used to package a regenerator that would most closely mimic that which optimized our DeltaE simulation. The regenerator was found to be most effective when its thermal capacity was maximized, and its ability to conduct heat to and from the working fluid was maximized, its ability to axially conduct heat within itself or radially conduct heat to or from its confines was minimized, and the resistance it imposed on fluid flow was minimized. Woven stainless screen disks (100 tpi) were selected for the regenerator matrix. These disks were of greater number and larger diameter than the heat exchanger disks and were pressed into a Delrin tube rather than a metallic one. The relatively low thermal conductivity of Delrin served to prevent radial heat conduction between the matrix and the outside, as well as additional axial conduction along the length of the matrix.

A second regenerator with alternating stainless and nylon screens rather than all stainless ones was also tested. The addition of the nylon screens further reduced axial conduction making larger temperature gradients possible from one end of the regenerator to the other.

Pressure Wave Generator

A unique piston powered by a jig saw was designed and built. A quick market analysis turned up about 15 models, all of which had roughly five amp motors, 26mm peak to peak strokes, and 1000-3000 strokes/minute operating frequencies. A RIDGID model was selected based on its easily adaptable round shaft. A 30 mm piston diameter was then dictated by the predetermined optimum gas displacement volume and fixed stroke of the saw motor.

The driven piston assembly was designed to minimize friction and heat by utilizing a clearance seal of about 10 μm to 20 μm. The pressure of piston seals on the cylinder walls would produce friction, and as a result, too much heat which would then have to be dissipated by a larger aftercooler with more dead volume. While not leak proof over extended time, the amount of gas forced through the gap was calculated and is entirely negligible for the operating pressure amplitude. The gas that does flow through the gap acts as a hydrostatic bearing which dramatically reduces friction and heat generation. Since there could be significant movement of gas though the clearance seal over time, the cylinder case was designed with a latex diaphragm rearward of the piston through which the drive rod is passed. This fixed seal between the piston and the general atmosphere allowed gas exchange between the fore and aft sides of the piston without compromising the composition of the gas within the system. Aluminum was specified for the cylinder casing to facilitate the removal of any heat generated by the piston as well as to ease fabrication. A purchased shaft misalignment coupler was used to take up the inevitable parallel or angular misalignment between the saw shaft and piston. A simple water jacket around the exterior of the cylinder was added to carry away any heat produced by the sliding of the piston or its compression of the local working fluid.

Orifice and Surge Volume

The manipulation of phasing between pressure and mass transfer within the system was accomplished using a purchased orifice positioned after the hot heat exchanger. An orifice with a needle valve was chosen so that fine adjustment of the flow impedance could easily be accomplished. Our simulations (DeltaE) indicated that our surge volume be a minimum of 3×10^{-4} m^3 in order to keep pressure oscillations within a negligible amplitude. A 3.5" diameter (3.7×10^{-4} m^3) spherical copper tank float was thus chosen based on its availability and price.

NUMERICAL SIMULATIONS

The computer simulations of the pulse tube refrigerator were performed using the thermoacoustic modeling program DeltaE [2]. DeltaE is a design environment for low-amplitude thermoacoustic engines. Through our discussions with Qdrive Inc. [3] and R. Radebaugh [4], design specs were developed that reflected a practical system that would be effective in producing a prototype of a pulse tube refrigerator. We designed the system to operate with a mean pressure of one atmosphere to help eliminate the need to pressurize the device. The pressure amplitude of the driver would be on the order of 13% of the mean or 13,000 Pa for an atmospheric mean pressure [4]. Most pulse tube systems have an operating frequency of 60 Hz or less. Better refrigeration can be produced when the system runs at a lower frequency.

A pressure amplitude from 8% - 20% of the mean pressure was considered for our pulse tube design. It is known that helium is the desired gas for pulse tube coolers. However, we designed and built our system using air at atmospheric pressure. The heat transfer in the regenerator is a key factor in any pulse tube design. The regenerator divides the hot and cold heat exchangers by removing the heat from the gas caused by the driver and storing it for a half cycle to be used to reheat the gas when it is being pulled back through the hot heat exchanger [1].

It seems that using a low pressure (atmospheric pressure) for the system actually helps the air perform better as the working fluid. The density of air is higher than helium, and at these low pressures the power density produced by the system is higher for air than helium. With this increase in power density, the system produces more refrigeration. To confirm the theory, simulations were performed using air and helium at five atmospheres. The results showed helium having double the cooling of air, thus showing its higher performance in higher-pressure situations.

A design of a completely uniform diameter through all components of the pulse tube was initially decided on to reduce pressure losses between the components. The first hot heat exchanger known as the aftercooler is 1.25 cm long. This size was decided on as being completely acceptable, as it allows a large enough exposed area to the environment to properly cool a component. The cold heat exchanger present next in the system is 2 mm long. Comparing DeltaE results for this heat exchanger size, and wanting to create a longer heat exchanger, a length of 1 cm was chosen. The final hot heat exchanger was 5 mm long and this was also changed to 1 cm for reasons similar to the cold heat exchanger. There was a series of nine total simulations performed in DeltaE in order to produce the best system.

The plan of using a uniform diameter for the entire pulse tube refrigerator was unsuccessful due to the fact that it could not achieve the desired performance. Figure 2 shows that, at a diameter of about 1.5 cm, the pulse tube has the maximum cooling power. Figure 3 shows a maximum amount of cooling power when the pulse tube has a length between 5 and 10 cm. The pulse tube is supposed to be divided into three sections of air: a cold region, a buffer region, and a hot region. Pulse tubes with low aspect ratios are more susceptible to gas mixing, therefore the final design needed a pulse tube with a high aspect ratio. For this reason a length of 10 cm was chosen for the pulse tube, as this length maximized the aspect ratio. After the design model had been completed, a driver was chosen with a variable frequency that ranged from 50 Hz to 20 Hz.

Figure 4 helps to illustrate the improvement in cooling power as the frequency of the driver is decreased. The maximum amount of cooling occurred at 15 Hz, which is close to the driver's lowest operating frequency of 20 Hz. The final step of the simulation is the adjustment of the orifice valve to change the impedance of the system. Improvement in cooling power was noticed when the impedance of the system is adjusted. According to our DeltaE simulations, the designed pulse tube refrig-

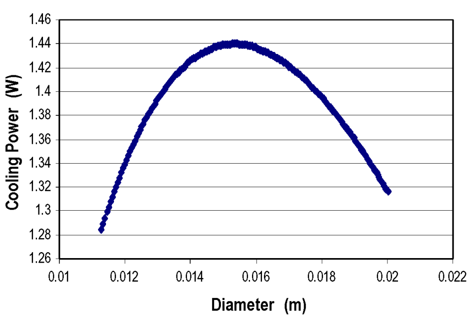

Figure 2. Cooling power vs. pulse tube diameter.

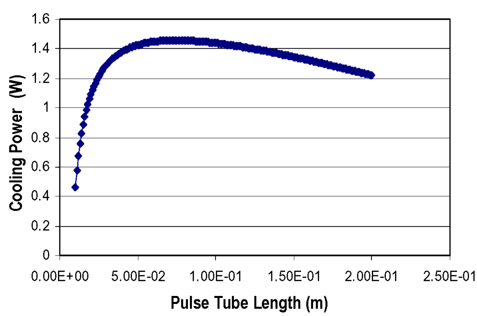

Figure 3. Cooling power vs. pulse tube length.

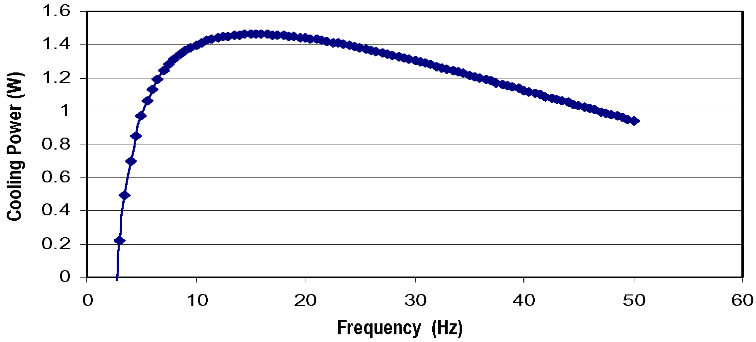

Figure 4. Cooling power vs. frequency.

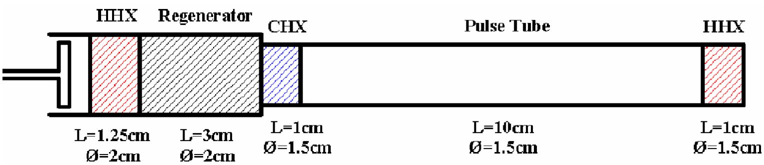

Figure 5. Designed system showing component dimensions for length (L) and radius (Ø).

erator can produce 1.44 W of cooling power at 285 K after the orifice is optimized. When the pulse tube is producing no cooling power, a minimum temperature of 210 K was predicted. The final design of the system with dimensions is shown in Figure 5.

SYSTEM HARDWARE FABRICATION AND ASSEMBLY

Solid, rather than hollow alloy-145 copper, 6061 aluminum, white Delrin, and white glass-filled Teflon round stock were purchased. Cylindrical and tubular elements were then precision turned from stock using toolmaker's lathes. All of the fabricated cylindrical components were press-fit together. The press-fit mating of soft Delrin and hard copper sections resulted in a self sealing connection. The purchased components including orifice valve and surge tank mount to the hot heat exchanger via threaded connections. Small sealable ports in the system components allowed insertion of thermocouples for temperature measurement and thus performance data acquisition. Precision punch and die sets were fabricated from tool steel to punch 50 µm oversize discs from stainless and copper screens so that they could be pressed into the nominal tube diameters. Each screen in the regenerator was individually oriented 45° from its closest neighbor to make the passage as tortuous as possible and to minimize axial heat transfer due to contact between them. A base plate and stand were fabricated from MIC-6 aluminum plate for the purpose of supporting the unit during testing. The stand acts as the clamp plate for the fixed outer edge of the latex diaphragm. In addition, it acts as the first of two plates attached by threaded rods which keep the press-fit parts firmly clamped together to avoid loosening due to vibration. The most difficult and time consuming part of fabrication was the modification of the jig-saw. It had to be completely disassembled. Both of the castings that contain its drive mechanism had to be machined so that they could carry new shaft bearings and support the rest of the system in the proper orientation. Also, a new, longer shaft had to be made to work with the original drive. Once completed (see Figure 6), our system was ready for testing.

TESTING AND VALIDATION

The frequency of the driver was verified using a non-contact tachometer. Four thermocouples were inserted into the system (Figure 7) during assembly; these were connected to a National Instruments data acquisition card (DAC). A National Instruments LabVIEW virtual instrument (VI) pro-

Figure 6. Prototype of complete system.

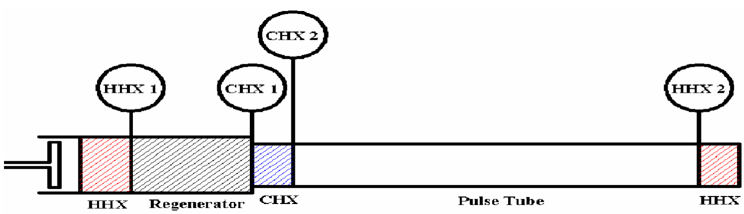

Figure 7. Thermocouple locations for testing

gram was then used to collect, condition, display, and record the output of each thermocouple over time at a user-defined sample rate. Finally, supply and return lines were run from the hot heat exchangers to a sink to carry away the heat removed from the cold portion of the system as well as the heat generated by the pressure oscillator.

The system was run at the lowest frequency the controller could support; this allowed the system to reach steady state temperature with the orifice closed, ¼ open, ½ open, ¾ open, and fully open. As the valve was opened the performance of the system continued to increase all the way to fully open. In response to the obvious inadequacy of the 1/16" orifice, we replaced it with a 1/8" one before resuming our testing. With the new orifice in place, testing was resumed but adjustment resolution was increased so that the effect of the orifice setting could more accurately be observed. The system was then run and allowed to reach steady operating conditions as the orifice was opened ½ turn (1/40th of its total range) at a time from ½ to 8 turns. The data that was colleted showed a clear peak in performance at which the no-load temperature of the air at the cold end of the pulse tube was -50°C and the temperature difference across the pulse tube was 90°C.

After establishing the optimal orifice setting, the system was shut down and allowed to return to room temperature. Figure 8 shows the early response of the four thermocouples as the system was run. The temperatures at the each end of the pulse tube (HHX1 and HHX2) climbed and the temperatures at the two ends of the cold heat exchanger (CHX1 and CHX2) registered a fall. This pattern illustrates the important role the regenerator plays in allowing the system to reach such low temperatures. If there were no regenerator the initial temperature separation magnitude seen in the early time (up to about 0.25 minutes) would be all that would ever develop. It is the temperature separation induced in the regenerator (HHX1 and CHX1) by the initial temperature separation in the pulse tube

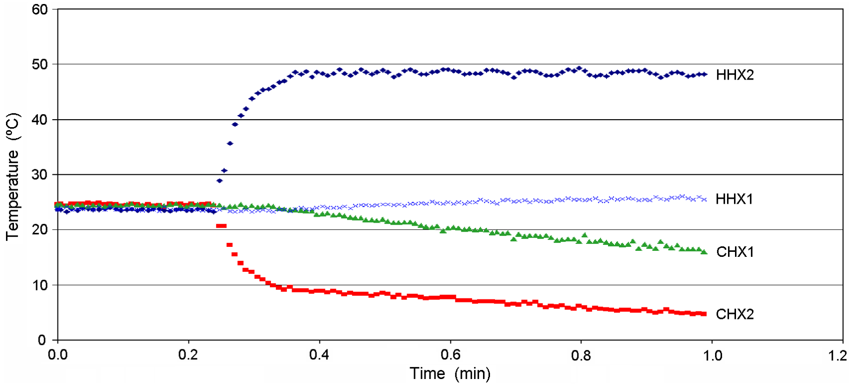

Figure 8. Temperature vs. time (first minute of operation)

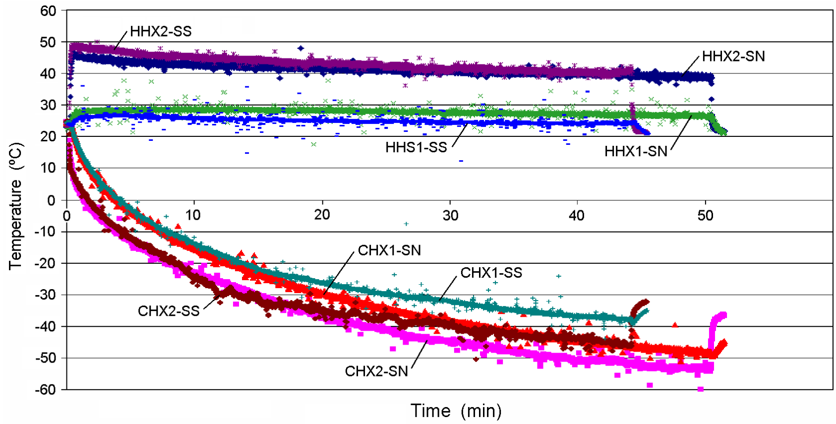

Figure 9. Temperature vs. time for extended time of operation.

(CHX2 and HHX2) that allows one moment of operation to build upon the next thus widening the attainable separation in both parts.

It was noticed that the difference in temperature across the regenerator (HHX1 and CHX1) was significantly less than that across pulse tube (CHX2 and HHX2) even as the system approached steady state. This was attributed to axial conduction through the regenerator packing and thought to be responsible for reducing overall performance by at least a few degrees.

The stack of stainless screen disks (SS) in the regenerator was thus replaced with alternating stainless and nylon screen disks (SN) with the hope that our performance would increase as a result. The unit was then run from room temperature down to near steady state just as it had been with the stack of all stainless screens so that the difference in behavior could be observed.

The results (Figure 9) show a faster temperature separation across the regenerator as well as a 10°C greater final temperature separation for the SN case compared with the SS case. When allowed to run for a longer period, the no-load temperature was measured at -60°C.

CONCLUSIONS

A low-cost pulse tube refrigerator was designed, built, and operated. The utilization of the thermoacoustic modeling software DeltaE enabled us to design a system with well defined constraints and performance expectations. The DeltaE modeling gave accurate dimensions for component lengths and diameters to base our prototype. The pulse tube system was solid-modeled and then fabricated. Once the prototype had been completed, testing was performed along with data acquisition using LabVIEW software. Four thermocouples were placed along the pulse tube refrigerator to accurately read the temperatures at all heat exchangers. By adjusting the orifice valve and frequency of the driver the system was optimized. The optimized system was able to produce a cold heat exchanger temperature of -53°C and a difference of about 90° between cold and hot regions within the pulse tube. These results were then compared to the computer modeled results that predicted a -63°C minimum temperature. The small difference can be accounted for by the fact that the prototype is not perfectly insulated and some of its cooling power is being lost to the environment. Additional tests were run using a unique regenerator comprised of stainless steel and nylon screens to help reduce axial conduction within the regenerator. When the stainless steel screen regenerator was run for 45 minutes it reached a temperature of -45°C. The new stainless and nylon screen regenerator was able to reach -55°C.

The availability of our prototype offers the opportunity for a great deal of further exploration and experimentation. We plan load testing to determine the cooling power of the unit at various temperatures for comparison to the DeltaE results. This can be done by adding a resistive heat source and a thermocouple to the outside of the cold heat exchanger and then thoroughly insulating the entire area. Varying amounts of power can then be applied to the heater as the system is operated and the resulting steady state temperatures plotted against the input power at which they were recorded will yield a load curve. The load data can then be used to calculate the efficiency of the unit as a whole or as components.

ACKNOWLEDGEMENT

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