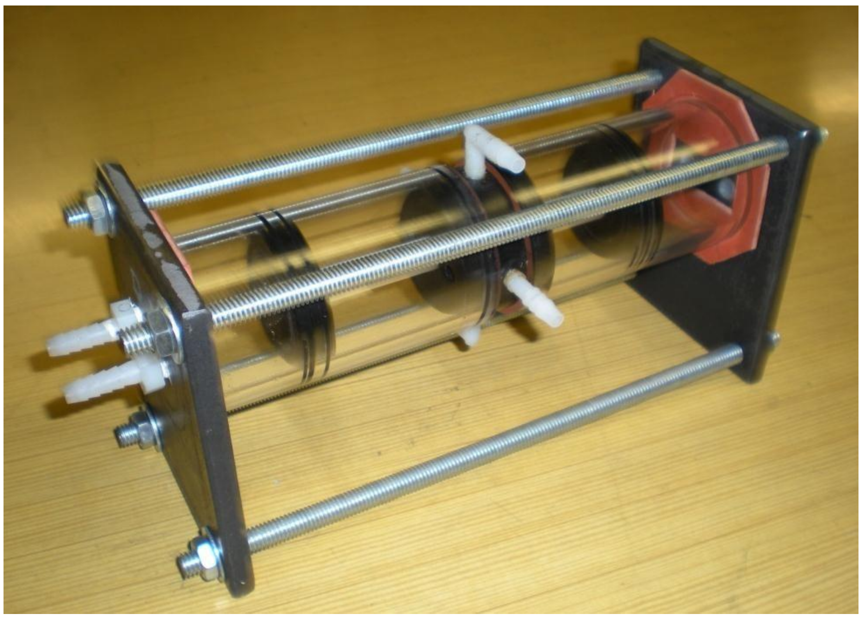
Design Report: Solar Powered Refrigeration Compressor

To: Mark Hall



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May 4, 2012

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May 4, 2012

Mark Hall

Attention: Mark Hall
Subject: Solar Powered Refrigeration Compressor

We are submitting a final report for a proof of concept for a solar powered refrigeration compressor and thermodynamic system.

The report contains material on two major portions of our efforts: thermodynamic modeling and compressor construction. For both of these areas, we have included a discussion of concepts considered and reasoning for major design decisions. Our recommendations for future work are also addressed in this document.

Please contact us at UofISPR@gmail.com with questions, comments, or concerns. We have enjoyed the challenges the project has given us and hope to see the project continuing in future semesters. Thank you for your time and the opportunity to work on this project.

Sincerely,

Solar Powered Refrigeration Team

Table of Contents

1. Executive Summary

The goal of this project was to provide a proof of concept for a solar powered refrigeration compressor designed to operate off of a pressure difference created by solar thermal energy. This included validating that the thermodynamic cycle is feasible and fabricating of a prototype, driven by compressed air, to acquire data on the design's mechanical functionality. With this system, an adequate cooling effect is produced with minimal electrical energy input, allowing small standalone units to operate almost entirely off solar thermal energy. This type of vapor compression refrigeration (VCR) is vastly different than what is on the market today. Current solar refrigeration technology involves ammonia evaporation, which is highly inefficient and bulky in comparison. Typical evaporation refrigeration devices are in the range of thirty to forty percent efficient [1]. Needless to say, it's time for a change. This new VCR cycle could serve many markets, reducing the use of non-renewable energy sources and moving towards a sustainable future. Markets such as produce transportation, biomedical refrigeration, commercial and residential air conditioning, and even the familiar drink cooler could benefit from this technology. To turn this idea into a marketable, economically feasible, mechanical device would forever change the way we use our energy.

2. Background

Commercial refrigeration and air conditioning account for over 17% of the nation's energy consumption, there is a clear opportunity to increase the efficiency of refrigeration cycles. Today, most solar refrigeration devices on the market use absorption refrigeration to achieve the cooling effect. These absorption devices are typically in the range of 30-40% thermal efficiency and require massive heat transfer rates at high temperatures [1].

The use of vapor compression refrigeration systems (VCR) in the solar thermal sector is a new idea with minimal available research. Our team researched and developed a prototype of a dual piston compressor to implement in VCR. We wanted to determine the thermodynamic efficiency of the system and the economic feasibility of creating a VCR cycle using our compressor. Our motivation was to create an economical refrigeration system using a thermal vapor compression instead of the traditional thermal absorption. We needed to build a VCR system prototype that could have a cooling effect equal to the cooling capacity of a mini-fridge. Our expected benefits of a thermal compression system were the reduction of electrical input and non-renewable energy use. This system could be used for air conditioning or refrigeration, so there is a large market for widespread use of this sustainable system.

3. Problem Definition

Our goal for this project is to develop a proof of concept for a compressor which is powered by thermal energy. This study will include a mechanical proof of concept and a proof of concept for the thermal system.

The client needs and specifications are defined as follows:

3.1 Needs and Specifications

Need	**Specification**
Cooling Effect	¼ ton cooling capacity (mini-fridge)
Minimal Energy Input (non-renewable)	Electric input to pump acceptable
Range of Operation	Boiler Temperature: 75 º to 140º F Condenser Temperature: 50 º to 75 º F
Mechanically Driven Compressor	Driven by pressure differences
Computer Modeling	Predict system behavior with thermodynamic model
Prototype Data Acquisition	Measure outlet pressures

4. Project Plan

4.1 Schedule

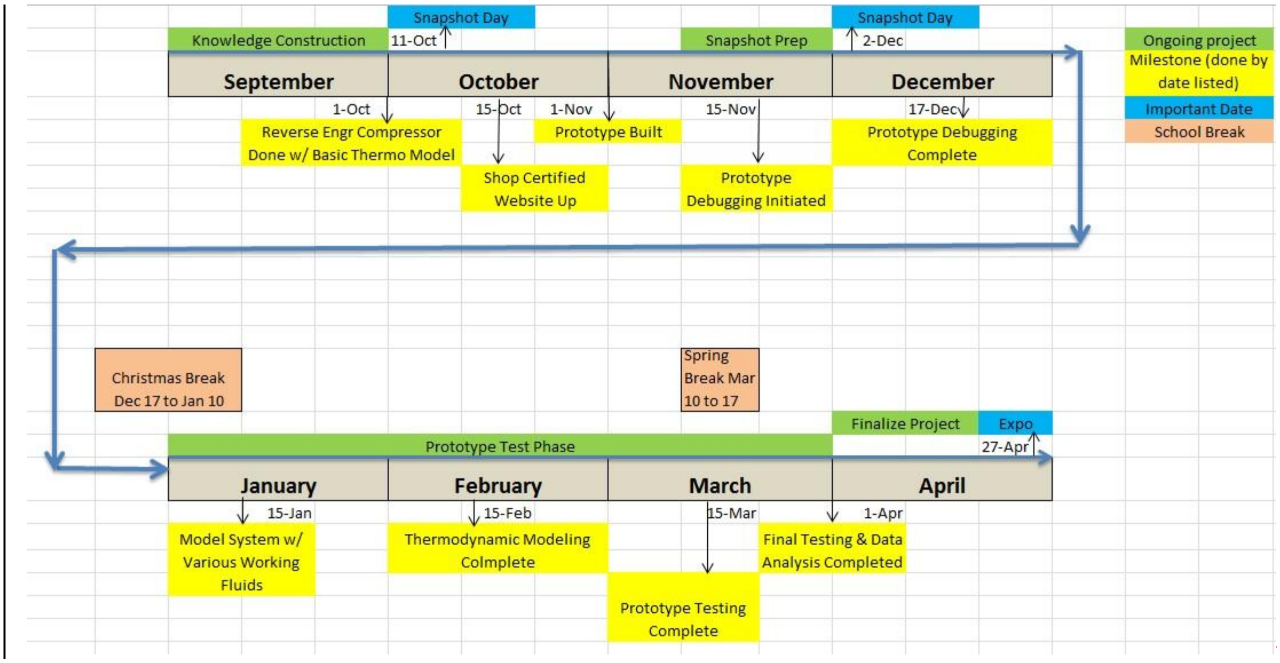

Figure 1: Project schedule for August 2011 – April 2012

To design this refrigeration compressor, we broke the research and development into two parts; thermodynamic modeling and physical prototyping. Part one involved determining the necessary cycle configuration for the compressor to operate in and thermodynamically modeling the system in Engineering Equation Solver (EES). Part two involved fabricating and acquiring data from a compressor prototype driven by compressed air.

4.2 Team Task/Responsibilities

[redacted] Thermal designer of the system within EES, researcher, designer

[redacted] Liaison to client for task updates, project organizer, researcher, designer

[redacted] Financial officer, prototype fabricator, researcher, product developer

[redacted] Internet guru, researcher, product developer, designer

[redacted] Thermal designer of the system within EES, designer, researcher, product developer

5. Concepts Considered

Once the general concept of the thermally driven compressor unit was understood, the proof of concept was divided into two studies:

- Mechanical prototype of compressor unit
- Thermodynamic cycle configuration

The concepts for both of these categories are outlined below:

5.1 <u>Mechanical Prototype</u>

The original diagram for our client's dual-piston compressor can be seen below in Figure 2. This original idea contains 8 valves that are placed at the inlets and outlets of each of their respective chambers.

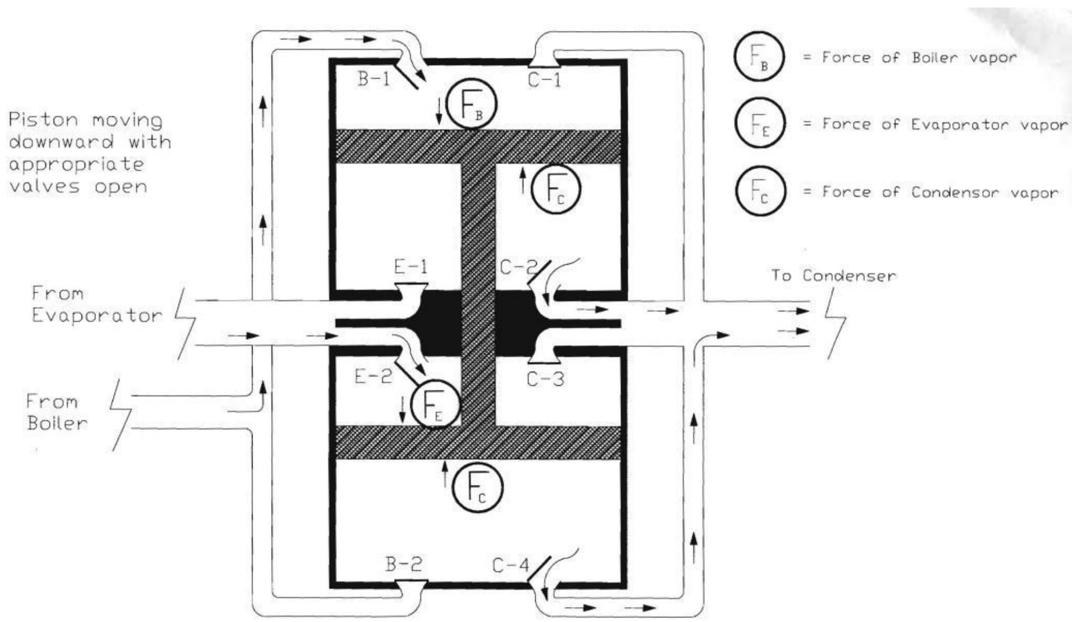

Figure 2. Original compressor configuration

In the first iteration of the prototype we decided to externalize these valves to be in-line, allowing us to reduce the amount of valves to 4 three-way valves as shown in Figure 3. A consequence of this externalizing the valves was an introduction of some dead volume in the lines between the valves and the chambers. Externalizing the valves also allowed us to create a two-bar link switching mechanism, letting all valves be switched simultaneously.

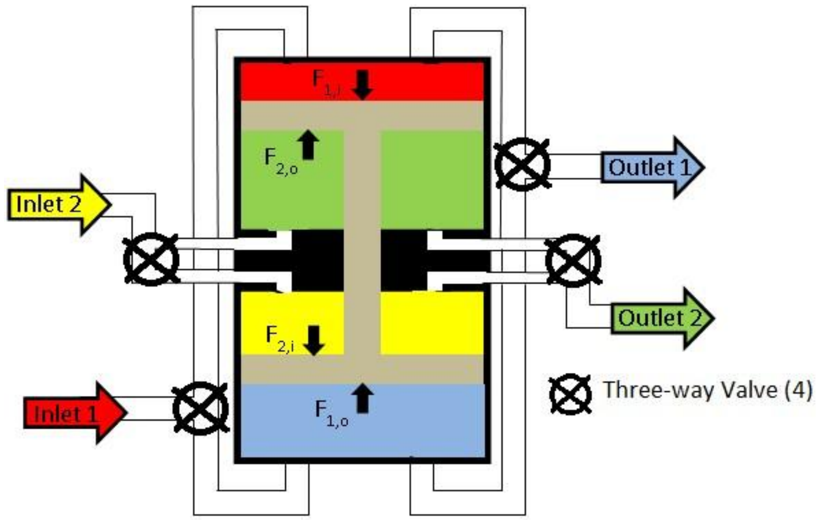

Figure 3: Valve schema

With the use of rapid prototyping, we built an inexpensive prototype. We built this prototype out of easy to machine ABS plastic and scrap steel plates that we found in the UI machine shop, rubber O-rings for all of the seals and Lucas Oil as the lubrication. Cast acrylic was used for the chambers so that we could see the pistons move and get a general idea about the speed of the apparatus. Raw materials can be seen in Figure 4 below.

Figure 4: Prototype parts

Compressed air is to be our working fluid because of availability and cost. We used ¼" pneumatic lines for the lines between our air tanks and the compressor. The inlet and outlet pressures will be measured with Bourdon tube pressure gauges. The compressor can be seen in Figure 5. A drawing package of this prototype can be seen in Appendix A.

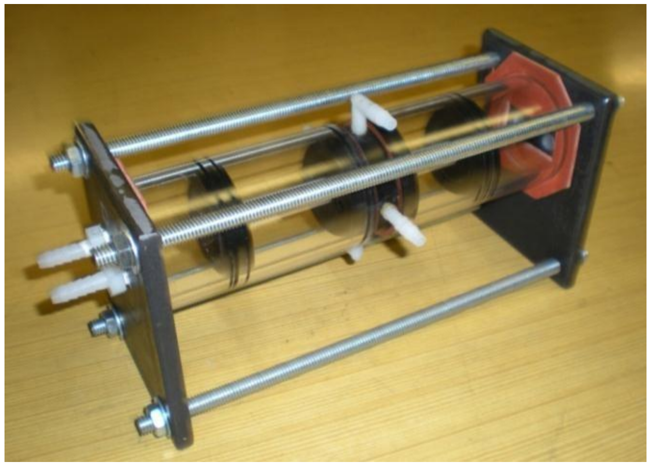

Figure 5: Compressor prototype without plumbing

Our prototype's valves could handle a maximum working pressure of about 165 psig, which meant the sealing elements between chambers must also be able to withstand this pressure. On our first iteration we employed a single O-ring to seal between chambers. However, some leakage was noticed, and our design was revised to include two sealing elements between each chamber. The pistons are cut with grooves for two O-rings which protrude about .010" from the piston surface. The centerpiece also employs two internal O-rings. These are about .25" from the edge with .010" overlap to provide an adequate seal. The fully assembled compressor is shown below in Figure 6.

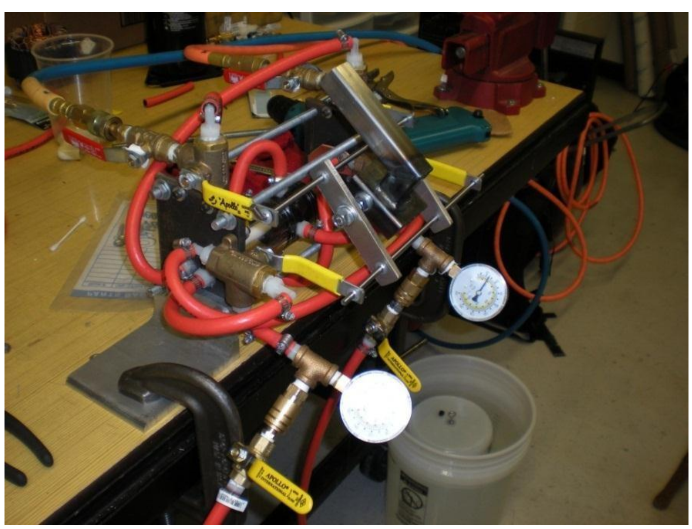

Figure 6: Fully assembled compressor prototype

5.2 Thermodynamic Cycle Configuration

With the given compressor configuration, there are many options for fluid placement, line combinations, and working pressures within the bounds of our specifications. The configuration of the compressor requires the combined pressure forces of two inlet chambers to overcome the combined pressure forces in two outlet chambers, causing the working fluid to be compressed to the condenser pressure of the refrigeration cycle as shown in Figure 7.

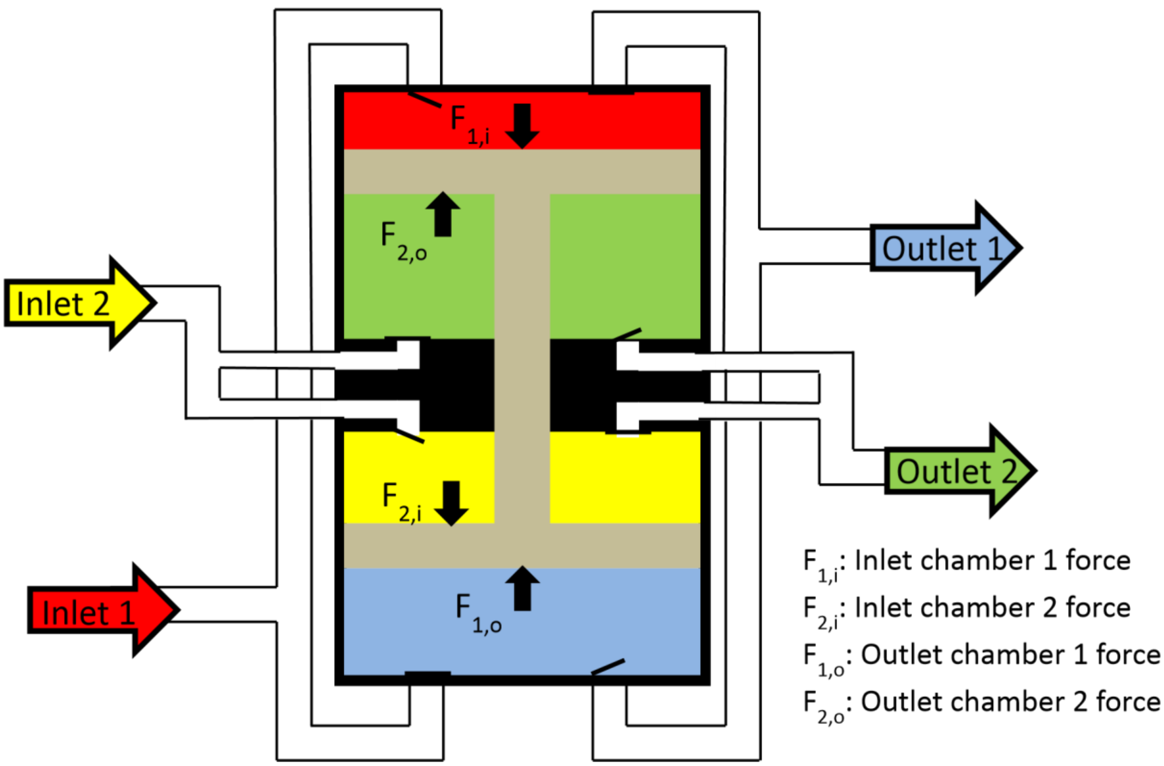

Figure 7: Compressor unit on a downward stroke

Using this compressor in a VCR cycle requires additional components to create the energy input that will allow the compression to take place. To obtain this input, a vapor power cycle has been added to the VCR cycle. In this configuration, the work input to the compressor of the VCR cycle is replaced by the vapor power cycle. The basic idea is shown in Figure 8.

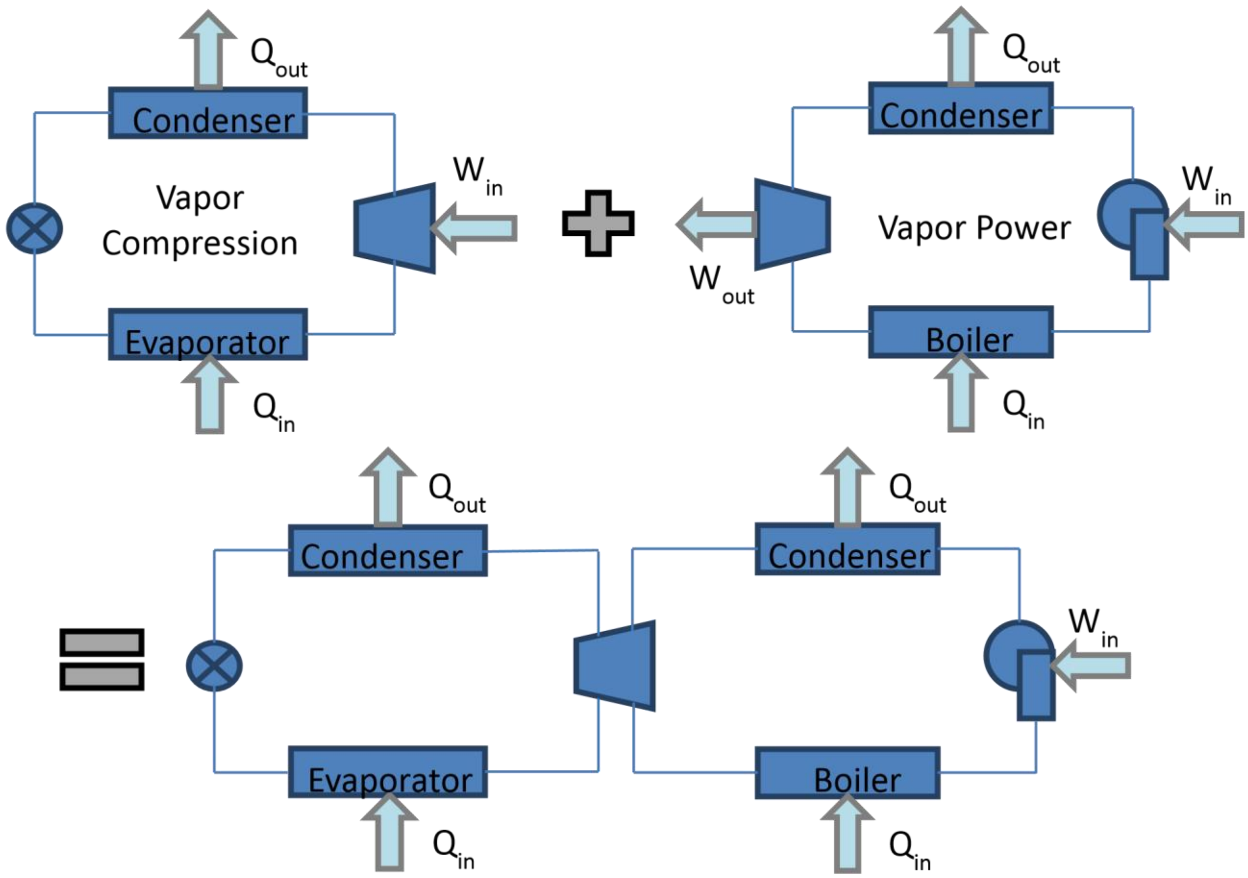

Figure 8: Combination of the VCR and vapor power cycles

When choosing the layout of the cycle, there were three major areas of concern: 1) compressor inlet and outlet pressures, 2) development of condensation in the compressor unit, and 3) cycle temperatures. These areas are discussed in detail below.

Compressor Pressures

The forces caused by the pressure in the inlet chambers must overcome the forces caused by the pressure in the outlet chambers to achieve the needed compression. In reference to Figure 5, this can be stated mathematically as,

$$F_{1,i} + F_{2,i} > F_{1,o} + F_{2,o}$$

The forces are determined by the chamber pressures and areas. For these thermodynamic calculations, an area reduction of fifteen percent has been assumed to account for the area of the connecting rod in the two inner chambers.

Condensation Development in the Compressor

A major concern for this compressor is the development of condensation within the unit. As with any compressor unit, liquid within this compressor can cause serious problems. Slugging, a common failure mode of many refrigeration compressors, occurs when pure liquid exits the compressor. However, with this compressor configuration, any phase change within the unit could be detrimental to the process. When a fluid transitions from a vapor to a liquid, there is a drastic increase in density and thus a decrease in the volume occupied by the fluid. This decrease in occupied volume will then cause a sudden change in pressure, not allowing the compression to occur.

Cycle Temperatures

Cycle temperatures must be carefully selected for use in a domestic environment. The range of desired boiler temperatures is 70°F to 140°F. This range will be achieved with the use of a solar thermal collector. The desired condenser temperature range is 50°F to 75°F. The lower temperatures of the range can be achieved with the use of well water supply and the higher temperatures of the range can be achieved by ambient air. The desired temperature for the evaporator is 10°F or lower, because 10°F is the maximum temperature that will allow for an effective refrigerator. Along with these requirements for cycle temperatures, the temperatures must also be selected so that the required forces are achievable for compression and there is no development of condensation inside the compressor.

6. Experimentation and Modeling

Experimentation for the mechanical prototype of the compressor unit is outlined in section 6.1. Section 6.2 contains modeling of the thermodynamic cycle configuration.

6.1 Prototype Experimentation

The goal of our experiments was to be able to understand the relationships of our inlet and outlet pressures as well as fluid flow rates into and out of our compressor for a given cycle time. However, measuring the mass flow rates proved to be difficult in this static system. We also had to ensure our system was leak free when pressurized, to keep our data consistent.

Instead of measuring mass flow rates, we measured the volume displaced per cycle in order to back out volumetric flow rates. To do this, we used a volume measurement device called an old spirometer, which was originally used to measure lung capacity. This device employs a counterweight and a floating drum to ensure the gasses are at atmospheric pressure, while providing an accurate means of measuring displaced volume. A schematic can be seen in Figure 9 below.

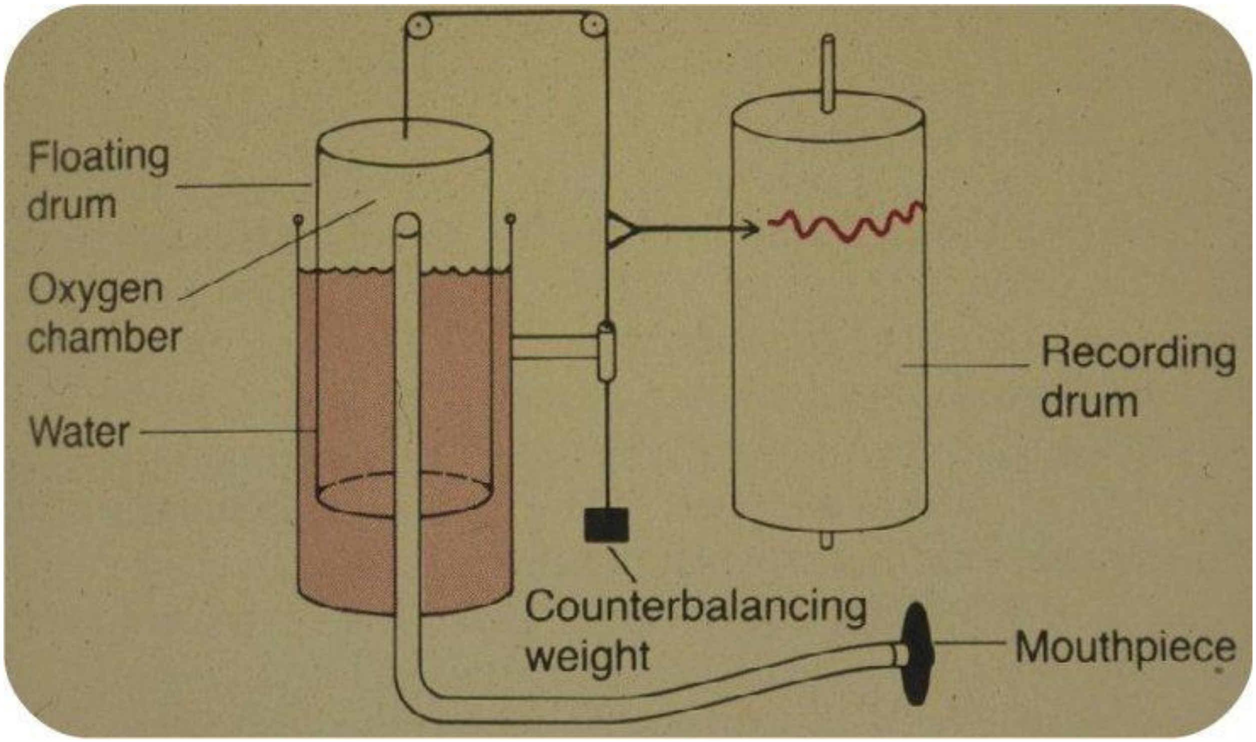

Figure 9: Old spirometer configuration

This setup is reflected in our experimental setup shown below in Figure 10.

Figure 10: Experimental setup

Our experiment was designed for 2^k fractional factorial analysis to allow us to avoid confounding variables in the experiment.   The independent variables were our high and low inlet pressures, and the dependent variables which were the outlet pressures and volumes displaced.

Inlet pressures were ranged from 25 to 50 psig, and outlet pressures and volume displacements were recorded.

ANOVA was used to statistically analyze the results of our experiment. From this statistical analysis, we were able to determine a linear correlation between inlet and outlet pressures and inlet pressures to volume displacements.

$$P_{low\ outlet} = 1.02907P_{low\ inlet} - 0.13672P_{high\ inlet}$$
$$P_{high\ outlet} = 0.09229P_{low\ inlet} + 0.8765P_{high\ inlet}$$
$$V_{low\ outlet} = 1.44910P_{low\ inlet} - 0.01143P_{high\ inlet}$$
$$V_{high\ outlet} = 0.43950P_{low\ inlet} + 1.31318P_{high\ inlet}$$

Using these equations, we can predict outlet pressures and volumes based on inlet pressures. This will be helpful in the final design of the compressor.

6.2 Thermodynamic Modeling

The two loop cycle was evaluated in EES with the following assumptions:

- 70% isentropic efficiency of pump and compressor
- No stray heat loss
- No pressure drops across heat exchangers and plumbing
- Adiabatic components

With this idealized system, we were able to obtain a working model with feasible operating temperatures and pressures. A pressure enthalpy diagram for the system using R-134a can be seen below in Figure 11.

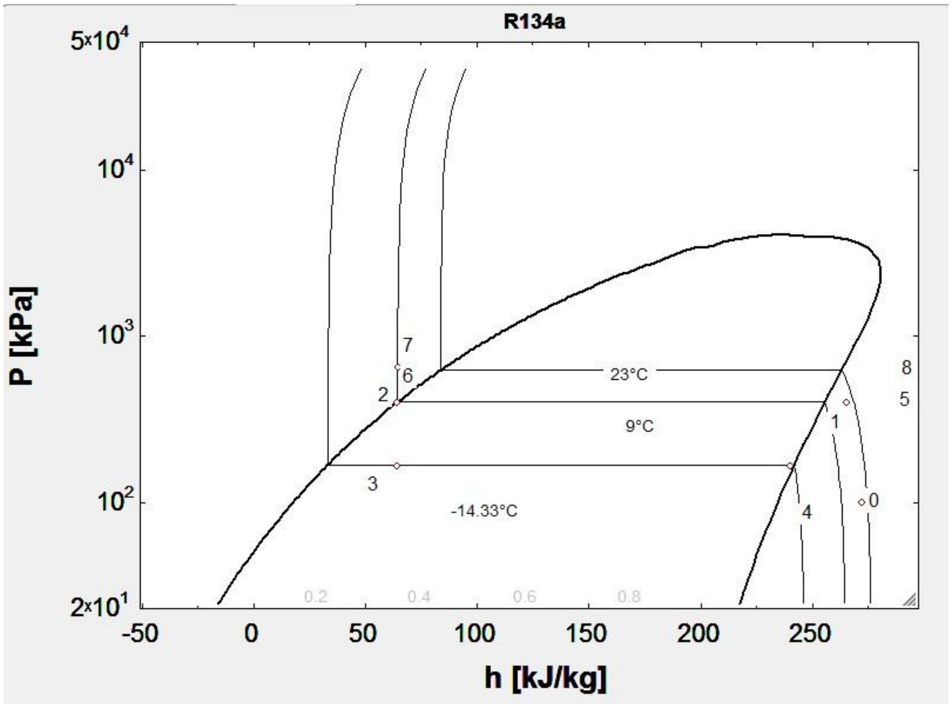

Figure 11: P-h diagram for cycle running with R-134a refrigerant

On this diagram, it is important to note that the fluid is superheated at the compressor inlets and outlets (states 1, 4, 5 and 8). Any liquid in the compressor could cause mechanical failure, so this is crucial to the thermodynamic system.

The combination of the VCR and Vapor Power Cycle has proved to be a feasible configuration for this refrigeration system. The cycle can operate within the given specifications given the assumptions stated previously in this section. Further analysis for a non-idealized system will provide more feedback for system feasibility.

7. Future Work

We have seen promising results from our research, but there is much more work to be done in order to have a complete proof of concept for this system. Thermodynamically, heat exchanger analysis and non-idealized cycle analysis will be necessary. This will help for sizing the entire system. For the compressor, valve design, valve timing, chamber sizing, chamber sealing, mechanical lubrication, and materials to be used are design issues that must be addressed for the final system which will use R134a as a working fluid. When the system is ready to be constructed, off the shelf parts should be purchased whenever possible, sized to fit compressor specifications. Testing of the whole system will provide understanding of the feasibility of this compressor configuration. As stated above, there is much more to be done, so we strongly recommend project continuation to explore this exciting new technology.

8. Budget

Figure 12 shows costs incurred during the course of the project.

Description	Material	Quantity	Unit cost ($)	Total Cost($)
Cylinder Wall	Cast Acrylic	2	40/ft	80
Plastic stock	PTFE	1	25/ft	25
Connecting Rod	Derlin	3	0	0
End Plate	Steel	2	0	0
All-thread	Steel	3	2/ ft	6
O-rings	Neoprene	7	1	7
Vinyl Tubing	Vinyl	4	1/ft	4
Vinyl Fittings	Vinyl	12	2 ea	24
Pneumatic Tubing	Rubber	4	1/ft	4
Pneumatic Fittings	Brass	9	1 ea	9
Y Valves	Brass	6	35	210
In-line Valves	Brass	2	7	14
Release Valve	Brass	1	11	11
Pressure Gauge	Brass	2	13	27
Spirometer	Plastic	2	35	70
Pneumatic Hardware	Brass	7	5	35
Fasteners	Steel	20	.5	10
			Total Cost:	536

Figure 12: Project Budget

Project costs fell well within our allotted budget due to the fact that much of our effort was focused on thermodynamic modeling. We used free materials whenever possible, and machined all components ourselves.

9. Works Cited

[1] U.S. Department of Energy, . "Energy Efficiency Trends in Residential and Commercial Buildings." 2008: n. page. Web. 9 Dec. 2011. <http://apps1.eere.energy.gov/buildings/publications/pdfs/corporate/bt_stateindustry.pdf>.