Senior Design Team 20

Solar-Powered Phase Change Compressor

Final Design Report – Fall Semester

Team Members:
Addison Bender
Jesse Diaz
Emmanuel Ferdinand

Sponsor: Grant Peacock
Faculty Advisor: Dr. Juan Ordonez

December 11, 2012

Contents

Executive Summary

This report describes the design of a compressor for an air conditioning system that is capable of being powered by solar generated steam. The design concept is based on the idea provided in the project sponsor's patent (Peacock). The patent describes a process that allows steam to act on a diaphragm, which ultimately compresses the refrigerant through an air conditioning unit.

Understanding the air conditioning unit and selecting correct materials are major tasks that were faced. First, the group members were challenged with the idea of modeling the system based on a refrigerant, which needed to be chosen. Once it was understood that a low vapor pressure refrigerant was needed, R-134a was selected. The material selection process for an elastic diaphragm is then described. After calculating its deflection, temperatures it would undergo, and fluids it would be exposed to; silicone rubber was selected. Then, check valves were selected to control the refrigerant and high frequency solenoid valves were selected to control the steam flow. The precise control of these valves will allow the diaphragm to reciprocate and continuously compress the refrigerant through the system. The initial design stage of the project has been completed. Parts will be ordered, the prototype will be assembled, and testing will take place in the upcoming spring semester.

Background

Air conditioning accounts for a large fraction of household energy costs. The compressor unit is a major component of an HVAC system, which consumes a high amount of power. If a compressor could be developed that is powered by a readily available sustainable energy source, then energy costs could be reduced.

The project sponsor has conceptualized a compressor that is driven by high-pressure steam. The function of a compressor in a refrigeration system is to raise the pressure and temperature of a refrigerant fluid. The concept for the compressor is to transfer energy from steam, which can be produced using a concentrating solar collector, to a refrigerant fluid.

The invention consists of a pressure vessel with two chambers that are separated by an elastic membrane. In one chamber steam is to be admitted from the source. This will fill the chamber and cause the membrane to expand. The opposite chamber will contain the refrigerant fluid. As the membrane expands, the volume on the refrigerant side will decrease, and it will be compressed.

Using check valves, the flow of refrigerant will be constrained to one direction through the system. The steam being admitted to and vented from the chamber will be controlled by solenoid valves. In this way, the pressure on the steam side can be regulated and the device made to reciprocate at a specified frequency (Peacock).

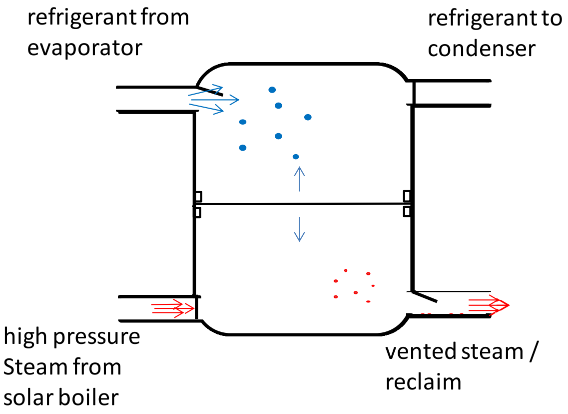


Figure 1: Concept for two-chamber compressor

Project Scope

The compressor is being designed for a system that would use a concentrating solar collector to focus energy onto a water supply and generate steam in a boiler. This supply of steam drives the compressor, and the compressor is integrated into a vapor-compression refrigeration system. In such a system, the high-pressure, high-temperature steam generated in the compressor is circulated though a condenser, expansion valve, and evaporator before returning to the compressor at lower pressure and temperature.

The scope of the project is limited by resources available, a budget of $2,000, and the eight-month time frame. After conferring with the project sponsor, it was decided to focus on the design of a prototype compressor. The compressor is being designed to meet the system requirements of a small window-unit air conditioner. The most modest commercially available air conditioners have a cooling rating of 5,000 btu/hr, which is equivalent to 1465 W. (Home Depot)

Since the project scope is concerned mainly with the compressor, an electrically powered steam supply will be used. This will simulate the best-case operating conditions and will allow for consistent test conditions to refine the prototype once it is constructed.

Concept Selection

Several different design concepts were evaluated in order to determine which would be feasible to design and build and accomplish the stated goals. As a first basis for comparison, a cost analysis of a solar-photovoltaic system was conducted. It was found that the PV panels required to power a small air conditioner day round would cost approximately $13,000. A solar thermal system would potentially much less expensive. (Bender et al.)

One concept that was proposed was to modify a two-cylinder air compressor. One cylinder would be controlled by the steam flow. The second cylinder, connected by a crankshaft would compress the refrigerant. The primary advantage of this design would be that the valves would be coordinated with the reciprocation of the cylinders, without any computer control. The disadvantage is that the only parameter that will be able to be varied is the flow of steam.

The two-chamber membrane concept has the advantage of multiple parameters that can be varied if the prototype requires tuning to produce the required flow of refrigerant. A second advantage is that the challenges of manufacturing and assembling a high-precision piston and cylinder can be avoided. Moreover, the project sponsor would like to see his general concept, which is outlined in the patent, engineered and brought to the prototype stage.

Determining Compressor Parameters

The primary function of the compressor is to increase the pressure of the refrigerant fluid that enters it. As a result of increasing the pressure, the temperature will also rise. Modeling the cycle as an ideal vapor-compression refrigeration cycle allows the properties of the fluid entering and leaving the compressor to be determined, in as well as the properties at the other components in the cycle. Using this model, the necessary flow rate can also be determined.

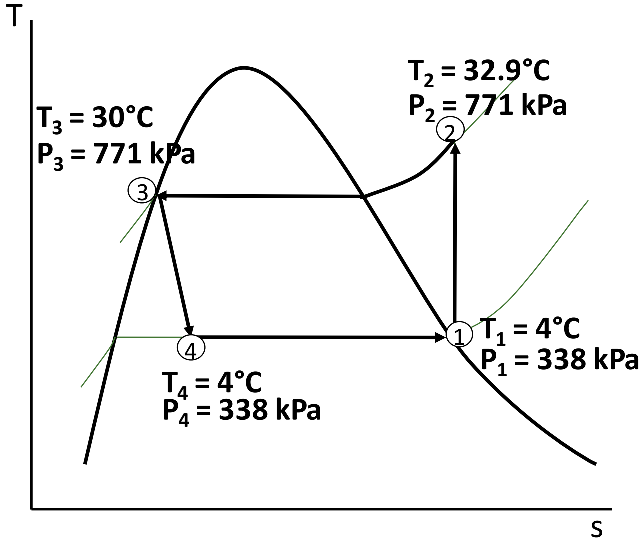


Figure 2: Ideal vapor-compression refrigeration cycle R134a

A vapor-compression refrigeration cycle contains four major components: a compressor, condenser, expansion valve, and evaporator. Within these components the following processes take place: isentropic compression (1-2), isobaric heat rejection (2-3), adiabatic expansion (3-4), and isobaric heat absorption (4-1). These four processes are superimposed onto the saturation vapor curve for R134a in the temperature-entropy graph shown above (Cengel). The reasons for the choice of R134a as the refrigerant fluid are described more in detail below (p 13).

The properties at each state can be determined beginning with the temperature at the condenser outlet (3). The function of the condenser is to exchange heat from the refrigerant fluid to the outside air. Therefore, the temperature at (3) can be no less than the outside temperature; 30°C is chosen as a reasonable daytime high temperature. Because the fluid at this point is a saturated liquid, the pressure can be determined based on the temperature, P_3 = 771 kPa. Neglecting pressure losses in the condenser, process 2-3 is isobaric; so the pressure at (2) is P_2 = 771 kPa. This is the high pressure that the compressor must produce.

The bottom temperature limit should be as low as possible in order to maximize heat transfer from the evaporator. If 0°C was chosen, there would be a risk of water vapor from the ambient air freezing on the evaporator. A bottom limit of $T_4 = T_1 = 4$°C is chosen to mitigate this risk. Because the refrigerant entering the compressor at (1) is saturated vapor, the pressure at this point can be determined, P_1 = 338 kPa. Now it is specified that the compressor must raise the pressure of the refrigerant from 338 kPa to 771 kPa.

Following the process assumptions described previously, the remainder of the properties can be determined at each state. Once the enthalpies of 1 through 4 are known, the required mass flow rate can be determined. The project objective is to design a compressor for a system that can produce 1465 W of cooling. This is the heat transferred to the evaporator in process (4-1). Using $\dot{Q} = 1465$ W, h_1 = 253 kJ/kg.K, h_4 = 94 kJ/kg.K, and equation I below, the necessary mass flow rate is determined to be $\dot{m} = 0.009\ \frac{kg}{s}$.

$$\dot{Q} = \dot{m}\,\Delta h \qquad (I)$$

Design and Selection of Components:

Membrane Design

After the function of the compressor has been quantified, the membrane that will separate the steam from the refrigerant, and do work on the refrigerant can be designed. The theory behind the membrane is that its deflection can be predicted as a function of the properties of the material it is made from, its thickness, and the differential pressure acting on it. The formula describing the deflection of a thin, circular, elastic disk, loaded with a uniform pressure is given below (Ashby).

$$\delta = \frac{3}{16}(1-\nu^2)\frac{P\,R^4}{E\,t^3} \qquad (II)$$

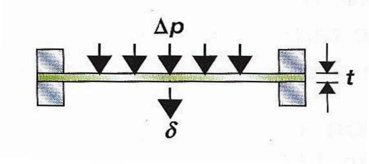


Figure 3: Circular elastic disk clamped

The approach used involved first determining the volume required for each stroke, translating this to a linear displacement δ, applying the properties of the chosen material, and solving for the required thickness. Because the displacement required to produce a specific volume depends on the radius of the membrane (a small radius requires a larger δ to produce the same volume as a large radius) the process was iterated using different values for the radius until a reasonable thickness was found.

The volume that each stroke must produce can be expressed as a function of the required mass flow rate, the density of the fluid being compressed, and the frequency of oscillation. It is proposed that a frequency, f = 2Hz, will be used. A higher frequency would decrease the volume required for each stroke, and this would require the membrane to be unreasonably thick. The target mass flow rate for the refrigerant is $\dot{m} = 0.009\ \mathrm{kg/s}$, and the average density of the fluid is $\rho = 27.03\ \mathrm{kg/m^3}$.

$$V = \frac{\dot{m}}{\rho \cdot f} \qquad (III)$$

This yields a volume per stroke of $V = 1.67\mathrm{x}10^{-4}\mathrm{m}^3$.

While the solenoid valves that have been selected are capable of frequencies up to 200Hz, the response of the system will depend on the flow rate of steam that is available. The system that will be used to simulate solar steam production is capable of supplying 0.022kg/s of steam. In order to check whether the steam supply will be adequate, the same equation above is applied to the steam and solved for mass flow rate. Substituting the average density of steam $\rho = 4.16\mathrm{kg/m^3}$, f = 2 Hz, and $V = 1.67\mathrm{x}10^{4}\ \mathrm{m}^3$, the mass flow rate is 0.0013kg/s, which is less than the maximum available.

The per-stroke volume is translated into a linear displacement, δ, which describes the change in position of a point on the center of the membrane. To accomplish this, the deflection of the membrane is modeled as a spherical cap, as shown below.

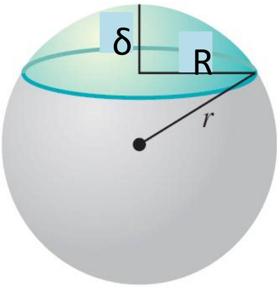


Figure 4: Volume of spherical cap

$$V_{cap} = \frac{1}{6}\pi\delta(3R^2 + \delta^2) \qquad (IV)$$

The radius has been chosen at R = 0.06 m. This is the result of iterating the calculation until a reasonable thickness was arrived at. Equation IV above is solved for δ, using the per-stroke volume and specified radius. The maximum displacement of the center of the membrane is δ = 2.7 cm.

The final missing piece for determining the membrane thickness by equation II is Young's Modulus, E, which describes the thickness of the material. The class of materials that have been surveyed are elastomers. These were chosen first because they have a low stiffness compared to other materials such as metals. If a metal were used for the application, its yield strength would be exceeded before the required displacement was reached. Elatomers can undergo large elastic deformation prior to yielding.

The constraint which limits the selection of a material is the high temperature environment that it must withstand. Information on the maximum temperature that various elastomers can be exposed to without the degrading the yield strength has been gathered, and is summarized below.

Table 1: Maximum operating temperature of common elastomers

Material	**Max T (°C)**
Butyl rubber	150
Isoprene rubber	150
Neoprene	115
Polyurethane elastomer	115
Silicone elastomer	260

Silicone has a much greater temperature tolerance than the other common elastomer materials. The properties for silicone elastomer are Young's Modulus, E = 12.5 MPa, and Poisson's ratio ν = 0.5. Solving equation I for thickness yields, t = 1.3 cm.

Determining Pressure Vessel Thickness

To determine the thickness of the 1018 Mild Steel pressure vessel, the yield strength of the material must be related to the maximum hoop stress. The axial stress on the cylinder is not considered because it is half of the hoop stress. The following equations determine the minimum thickness required for the vessel not to yield. σ_θ is the yield stress of the 1018 mild steel, P is the maximum pressure that the vessel will experience, r is the radius of the thin walled vessel, and t is thickness of the shell.

$$Hoop\ Stress: \sigma_\theta = \frac{P * r}{t} \qquad (V)$$

$$370\ x\ 10^6\ Pa = \frac{770\ x\ 10^3\ Pa * 6\ x\ 10^{-2}\ m}{t}$$

$$\boldsymbol{t} > 0.13\ mm$$

The thickness of the vessel must be greater than 0.13 mm for it not to yield under the specified pressure.

In the design of the pressure vessel, a thickness of 0.5 cm has been specified. Because the pressure in the system is not extremely high, the limiting factor for the thickness is not the pressure, but manufacturing choices. The vessel must be thick enough to allow threaded holes for the valve ports to be created. Because of the small size of the compressor, compared to a pressure vessel such as a propane tank, the component will be machined from a solid steel rod, rather than welded together from sheet metal. Therefore, no cost is being saved by reducing the wall thickness.

Valves

Valves are mechanical devices used to regulate and control the flow of fluids. To regulate the different types of fluids in the compressor, two different types of valves will be utilized. To control the refrigerant, R-134a, check valves will be implemented. Check valves allow flow in only direction, which is necessary for the closed loop refrigerant. They are purely mechanical and do not require any external controller. The check valves operate at a minimum cracking pressure, which is the minimum upstream pressure at which the valve will operate. To control the flow of steam, electrically operated solenoid valves will be used. These valves will allow the flow of steam to enter and exit the system at 2 Hz. The solenoid valves are electromechanical; they are normally closed, until an electric current is supplied. This solenoid actuates a built in coil to open a port by an external controller. Overall, proper valve control is necessary to cause the diaphragm to reciprocate in the desired manner. Valves that are selected must be able to handle the various operating conditions of the fluids.

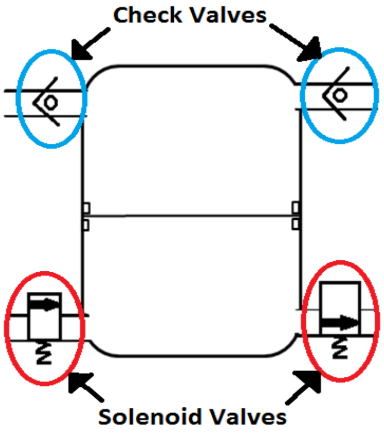


Figure 5: Use of valves on compressor

Check Valve Selection

The selection of the check valves were limited to devices that could control and operate with R-134a refrigerant as its medium. The valve that was selected was a PARKER Copper Check Valve. Displayed below, is a table portraying the technical specifications of the refrigerant check valve.

PARKER Copper Check Valve	
Item	Copper Check Valve
Type	Ball
Description	Copper Body Check Valve, Spring Loaded Ball
Connection	3/8 Fe Solder
For Use With	All CFC, HCFC, and HFC Refrigerants
Image	
Price	$38.05 each

Solenoid Valve Selection

The selection of the correct solenoid valve must withstand the high temperatures and high pressures the steam will produce, as well as operate with the correct frequency to compress the diaphragm. An ASCO Solenoid Valve was selected to ensure accurate operation of the steam. Below is a table which displays the technical specifications of the valve.

ASCO Solenoid Valve, 2 Way	
Item	Solenoid Valve
Type	Next Generation
Configuration	Normally Closed
Max Operating Pressure	1,034 kPa (150 psi)
Max Fluid Temperature	82°C (180°F)
Power	2 W
Voltage	100-240 VAC
Body Material	Brass

Image	
Price	$194.25 each

Steam Generation

To compress the diaphragm, steam must be generated. The project budget does not allow for the team to create an entire steam generation system based off a concentrating solar dish. Instead, an electric boiler that generates steam will simulate the system. This electric boiler, located at the Energy & Sustainability Center at Florida State University, supplies steam at a mass flow rate of 80 kg/hr at a maximum temperature of 650°C. Also, the outlet of the boiler can be adjusted to multiple fittings; therefore selection of the size of the valve is a free variable rather than a constraint.

Figure 6: Simulated steam supply

Theoretical Solar Collector

Even though the team will be using an electric boiler, a theoretical steam generation system utilizing a solar collector must be calculated in order to determine the correct size of the dish that would be implemented if the budget could be expanded. The calculations and efficiencies are based off a thesis, *Low-Cost Concentrating Solar Collector for Steam Generation* by John Dascomb. The overall system concentrates light, on a parabolic dish, into a receiver that has water pumped through its insulated copper pipes. The heated water is then flashed into a chamber, generating a steam pressure of 550 kPa. To actuate the diaphragm at 2 Hz, a pressure of 770 kPa is desired. To calculate the size of the dish needed, the previous dish size and efficiencies must be understood. The following equation was used to find the overall thermal efficiency of the system:

$$\eta_{thermal} = \eta_{insulation} * \eta_{optical} * \eta_{absorber} \qquad (VI)$$

The efficiencies will remain constant because the only thing will be altered is the size of the dish. In the thesis the insulation of 83% was found at an average of 946 W/m^2, which will remain consistent due to the fact it respects area. The size of the solar collector used for the steam generation, in the thesis, was 14 m^2, with respect to the overall thermal efficiency of 39%. Directly correlating the receiver area with pressure achieved, a solar concentrator, using same devices and components as in thesis, will be 19.6 m^2 to achieve the needed 770 kPa.

Figure 7: Solar concentrating dish used to generate steam

Refrigerant Selection

The air conditioning unit's working fluid is a refrigerant, and there are many refrigerants that are commonly used refrigeration systems. In home a/c units, currently there are three refrigerants in use: R22 is the oldest, it is being phased out by the U.S. government to reduce the impact it has on the environment; R410a replaces R22 for having less of an environmental impact, and R134a which is primarily used in vehicles and rarely in home air conditioning.

After having been compressed, the refrigerant enters a condenser to reject heat to the outside environment. The compressor, which is being designed to replace a 5000 BTU air conditioning unit's compressor, needs to replicate the same function as the electric compressor. Having contacted air conditioning repair companies, they informed us that the pressure gauged after the compressor reads between 190 psi (1310 kPa) and 250 psi (1724 kPa) on R22 home units. The higher the temperature outside, the higher the pressure needed; these values correspond to temperatures between 98 ^{o}F and 117 ^{o}F in a pressure-temperature chart of R22. Looking at the pressure values and having seen the data of a solar collector, we see that that amount of pressure may be unachievable. We then looked at the other two refrigerants to see what pressures would be needed to achieve the same refrigerant temperature range. R410a needs to be between 275 psi (1896 kPa) and 364 (2510 kPa) psi, which is even more than R22. R134a needs to be between 104 (717 kPa) psi and 146 (1007) psi, which is achievable. The compressor is being designed for a system that would use R134a as the refrigerant.

Bill of Materials

Table 2: Cost of major components

Item	Dimensions	Use	Quantity	Price
Silicone Rubber Sheet	0.5" x 12" x 12"	Diaphragm	1	$123.01
1018 Mild Steel Rod	D = 5" , L = 6"	Pressure Vessel	1	$124.12
Copper Check Valve	D = 3/8"	Control Refrigerant	2	$76.10
Solenoid Valve	D = 0.5"	Control Steam	2	$388.25
			Total	**$711.48**

Safety and environmental concerns

There are two primary safety concerns. The first is that our compressor is a pressure vessel, and pressure vessels fail by crack propagation when the material's failure strength is exceeded. If this occurs, and the vessel is ruptured, then a high pressure stream of fluid would be released that could injure a person who is near it. There are a few ways to tackle this problem. A factor of safety could be applied. If the material needed is too expensive we can use a weaker/cheaper material but increase the thickness of the vessel. To make the compressor extra safe, if it is deemed necessary after we start testing, we may add a casing to contain any unexpected failure.

The second safety concern involves the refrigerant. Corrosiveness, toxicity, and flammability properties of refrigerant fluids were researched. The chemical name for R-134a is tetrafluorethane; it is a compound of carbon, hydrogen, and fluorine. The safety rating of R134a is A1. The ASHRAE (American Society of Heating, Refrigeration, and Air-conditioning Engineers) rating of A1 means that it is nontoxic and non-flammable, although if air is mixed in it becomes combustible under pressure and large quantities, though higher than what's used in a/c units. It can be harmful if inhaled. It has no corrosive potential and would have no interaction with the silicone membrane. Our compressor will need to have leak prevention; it needs to be sealed between the two halves of the body and also between the interfaces of the tubing.

Environmental concerns include the ozone depletion and global warming potential of the refrigerant. These effects are quantified as ODP (ozone depleting potential) and GWP (global warming potential). The ODP is the potential of a refrigerant to deplete the ozone layer compared to the first generation refrigerants which are set at 1. R134a has an ODP of 0 because it does not break down into compounds which bound with oxygen, unlike its chlorofluorocabon compressors. The GWP is the measure of how much heat is trapped in the atmosphere due to greenhouse effect of released refrigerant in the environment; it is compared to CO_2 which has a GWP of 1. Over a 14 year period, R134 has a GWP of 1300. Compared to R410a, GWP = 1725, and R22, GWP = 1810, it is safer to the environment. Even so, any unnecessary release should be avoided.

A second environmental concern is the responsible use of fresh water resources. The steam generated will require a continuous fresh water supply. This could potentially be supplied by a rainwater collection system or tap water. Rather than venting the steam to the atmosphere, it would be more responsible to attach a simple heat exchanger that would allow the steam to condense and be collected as water. Depending on the quality of the input water, this could then be used as drinking water. It could also be recirculated to the steam generation system.

Future Plans

After determining the correct design through the decision matrix, choosing the correct diaphragm by material selection, and selecting suitable valves for the system, the initial design concept is complete. The table and chart below depict the future work and schedule that will be accomplished over the spring 2013 semester. The team will order the materials necessary for the compressor, machine parts to specification, correctly assemble the entire system, program the solenoid valves for precise operation, test the system, and make any necessary revisions in order to produce a fully functional device.

Over the break the team will search for the exact parts and suppliers in order to prepare to order the parts the first week of the semester. Machining the materials will take place in the College of Engineering and will be conducted by one of the teammates after undertaking a 2-week machining course. Programming the solenoid valves will have to be conducted with supervision from a faculty member or specialist that has knowledge in the field. The team will commence testing and revising as soon as it can in order to leave enough time if replacement parts have to be machined or changes to original design must be made.

Table 3: Remaining project tasks

Tasks	Start Date	Duration (Days)	End Date
Ordering Materials	10-Jan	19	29-Jan
Machining	31-Jan	15	15-Feb
Assembly	27-Feb	9	8-Mar
Programming	5-Mar	14	19-Mar
Testing	18-Mar	17	4-Apr
Revising	22-Mar	25	16-Apr

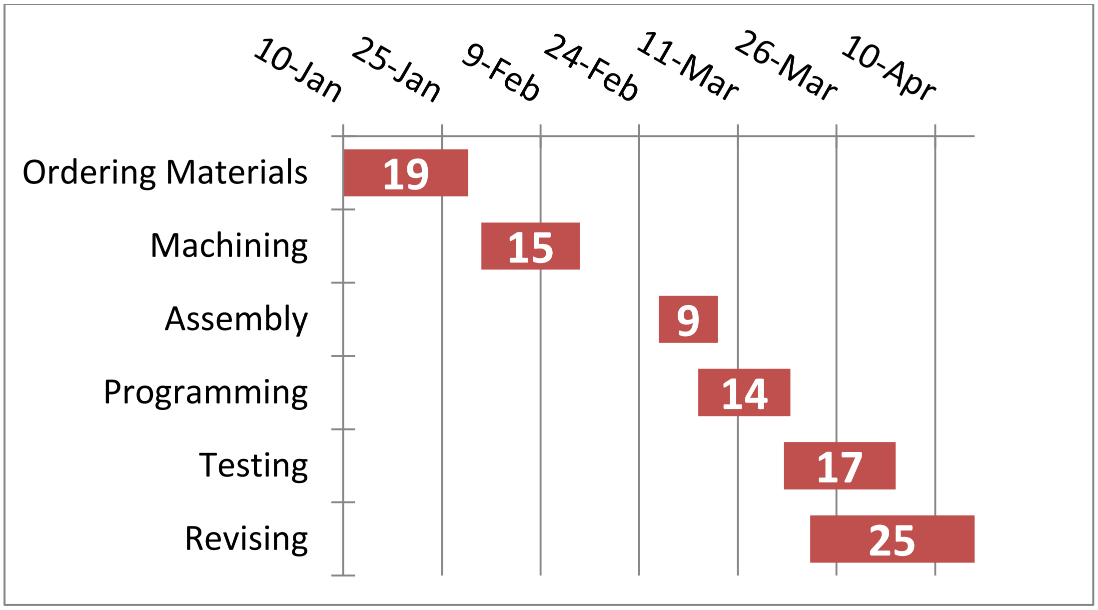


Figure 8: Project timeline of remaining tasks

References

Ashby, Michael. *Material Selection in Mechanical Design*. 4th ed. Butterworth-Heinemann, 2011.

Bender, Addison; Diaz, Jesse; Ferdinand, Emmanuel. Conceptual Design Review. 2012. http://eng.fsu.edu/me/senior_design/2013/team20/concept%20report.pdf

Cengel, Yunnus; Cimbala, John; Turner, Robert. Fundamentals of Thermal-Fluid Sciences. 4th ed. New York: McGraw-Hill. 2012.

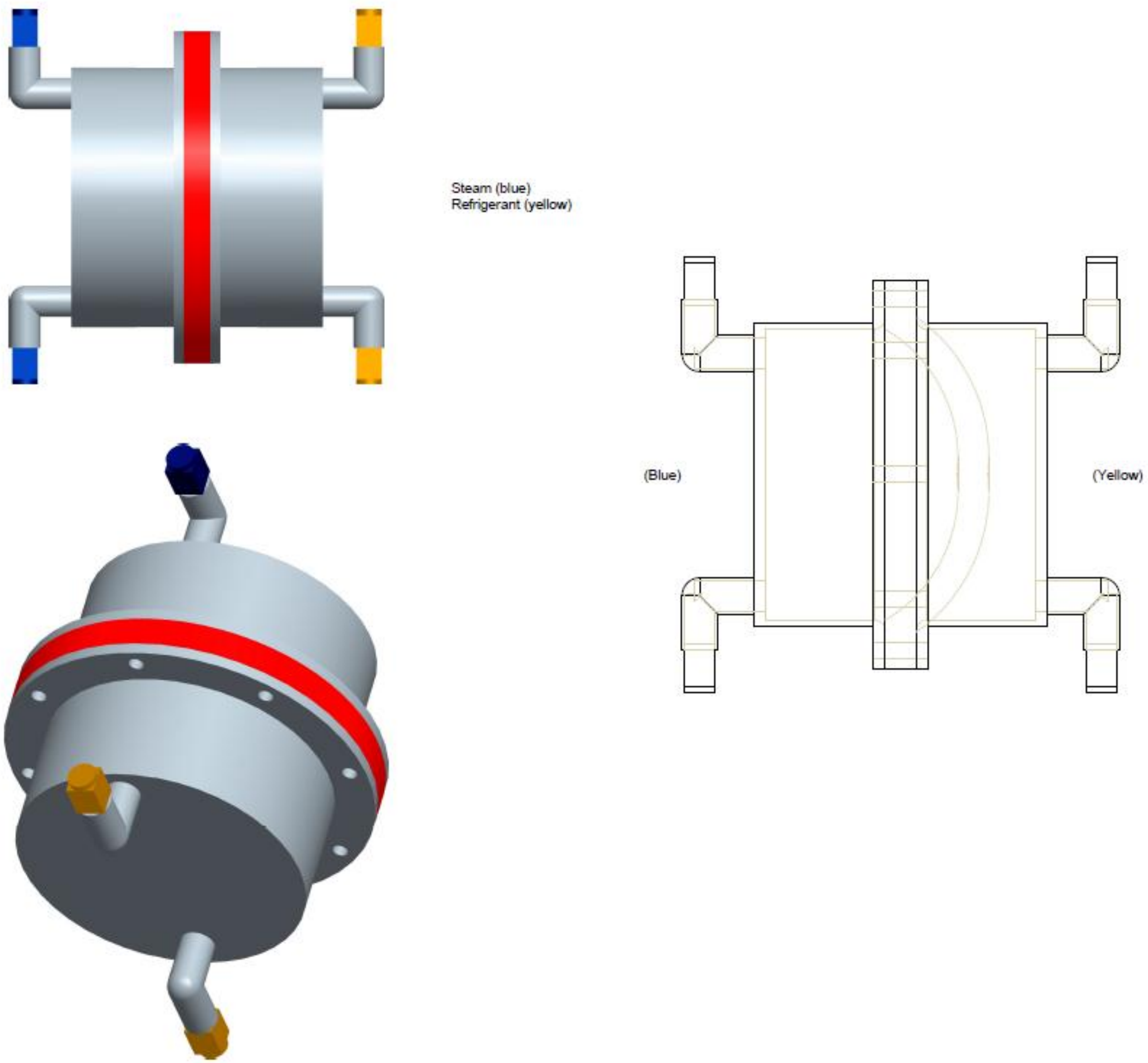
Dascomb, John, "Low-Cost Concentrating Solar Collector for Steam Generation" (2009).*Electronic Theses, Treatises and Dissertations.* Paper 833. http://diginole.lib.fsu.edu/etd/833

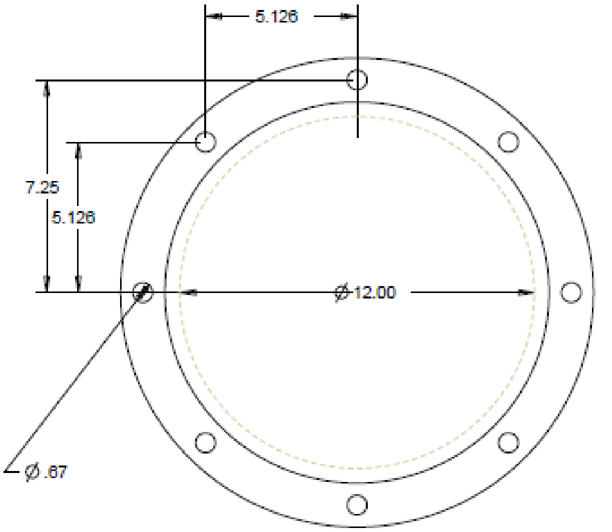
Home Depot. GE 5,000 BTU 115 Volt Window Air Conditioner. 2 Dec 2012. http://www.homedepot.com/Appliances-Heating-Cooling-Air-Quality-Air-Conditioners-Window-Air-Conditioners/h_d1/N-5yc1vZbv6l/R-203188059/h_d2/ProductDisplay?catalogId=10053&langId=-1&storeId=10051#specifications

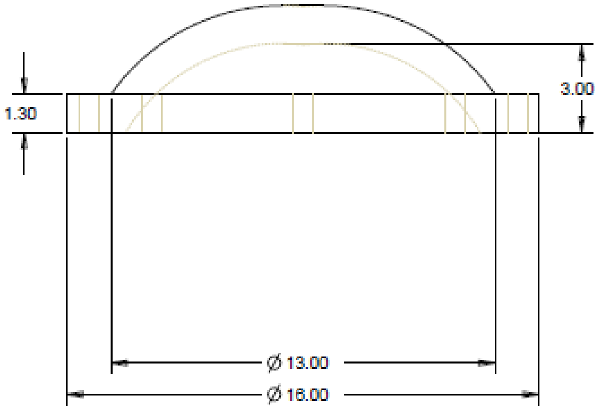
Peacock, Grant. “Phase Change Compressor”. Patent 20,100,192,568. 5 Aug 2010. http://www.google.com/patents/US20100192568?pg=PA1&dq=grant+peacock&hl=en&sa=X&ei=jViIUOT4H4PY8gT2i4CoBQ&ved=0CDEQ6AEwAQ#v=onepage&q=grant%20peacock&f=false

Appendix

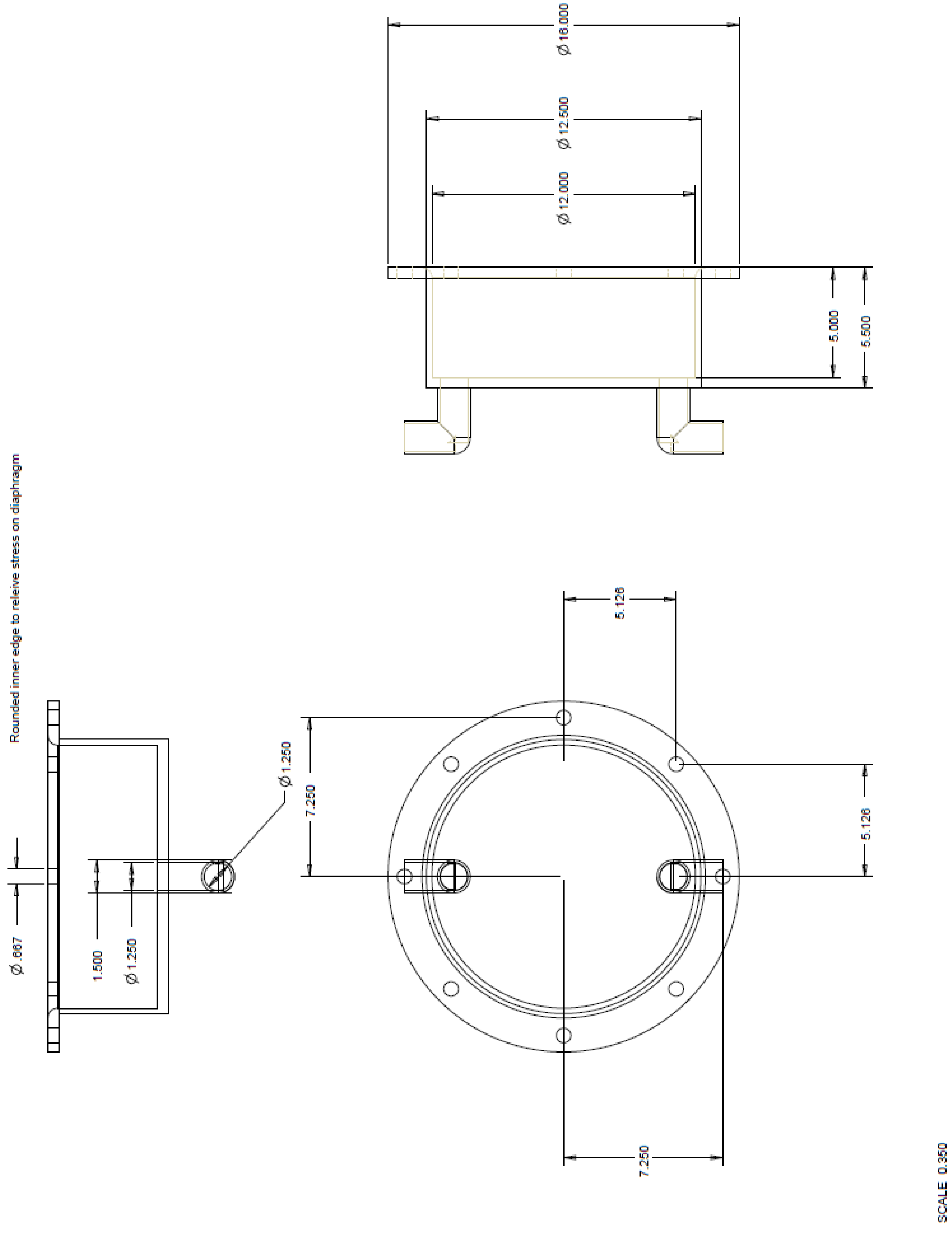
The attached drawings show the compressor and membrane. The membrane is shown at its maximum deflection.


Steam (blue)
Refrigerant (yellow)
(Blue)
(Yellow)
SCALE 0.350





SCALE 0.350


Rounded inner edge to releive stress on diaphragm
Ø16.000
Ø12.500
Ø12.000
5.000
5.500
Ø.667
1.500
Ø1.250
Ø1.250
7.250
5.126
5.126
7.250
SCALE 0.350