An Illinois Institute of Technology Interprofessional Project (IPRO)

IPRO 357
Heat-Driven Refrigeration for Developing Nations

Team Members:
Tom Alworth, Alex Callow, Keon Woo Kim, Donghoon Lee, Stefan Vogel, Leonardo Nortes, Anel Medrano, Andrew Keen, Alec Frost, Aron Ahmadia

Faculty Advisors:
Dr. Francisco Ruiz, Dr. Richard Holbrook

Sponsors:
ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers)
NCIIA (National Collegiate Inventors & Innovators Association)

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Introduction

With the coming of the 21st century, mankind is seeing the dawn of a new millennium with a certain confidence and optimism. Mankind has advanced more technologically in the past two hundred years than ever before. Technological advancements have made the current standard of living possible.

These advancements, however, have not come without a price. Technological and industrial developments require large amounts of continuous, reliable energy. The emissions resulting from large-scale energy consumption contribute to many adverse environmental effects such as depletion of the ozone layer, and air and water pollution. Furthermore, for the majority of the world's developing countries, energy is at a premium and is not very easily available. The extremities of tropical or arid climates often compound energy-dependency problems.

Faced with such a scenario, this project addresses these problems in order to find a solution that is economically and socially feasible.

The solution will take the form of a novel system based on a refrigeration cycle driven by heat. The cycle will be built around a "pressure exchanger" consisting of a piston, along with solenoid valves controlled by a microprocessor. Heat will most likely be obtained through the combustion of biomass such as plant by-products and animal wastes, but will be flexible enough to allow for more conventional fuel types such as kerosene and coal.

Areas that would benefit greatest from this type of project are developing regions such as those found within the South American, Asian, and African continents where the daytime temperature reaches an average of 40 degrees C or more in the summer months. Due to the extreme heat, the local population must face several problems including the wastage of agricultural produce, the outburst of epidemics due to unhygienic living conditions, and spoiled food. Electricity is often unreliable, limited to certain hours, or altogether nonexistent. As a consequence conventional refrigeration or air-conditioning systems, which require a substantially higher and uninterrupted supply of power, cannot be relied upon.

If a system that is inexpensive, reliable under local conditions, and operates without electrical power could be introduced, however, it would be a tremendous asset and an immensely desirable alternative to the current system in existence.

This project aims to develop a refrigeration system that uses a heat source instead of electricity as its main source of power, for developing countries in hotter parts of the world, like Central America, South America, and Africa, where electrical power is often unreliable or nonexistent. This refrigeration machine, in addition, is likely to be widely demanded in the developed countries. Due to the fact that this is a novel device, the prototype to be built will use electricity instead of different energy sources, but its thermodynamic cycle and its main components are conceived so that the accomplishment of the refrigeration effect can be easily achieved with few modifications.

Project Background

The Heat Driven Refrigeration Cycle was used in an IPRO in previous semesters a few years ago. The first group's main focus was research. They examined the refrigeration cycle being proposed, the possible heat sources, and the different types of refrigerant available. With the theoretical groundwork completed the second group began construction on the prototype of the refrigerator to serve as a proof of concept.

The first IPRO began with the idea of a different type of heat driven refrigeration cycle than what is currently used in absorption cycle refrigeration. This novel design utilizes a pressure exchanger composed of a piston and electronically controlled valves to replace the traditional compressor found in vapor-compression cycle machines used in the industrialized world. This then would allow a refrigerator to be constructed that does not need electricity but instead heat. Possible sources for the heat included a coal, kerosene, propane, natural gas, agricultural waste, firewood and some currently less viable alternatives like locally produced methane and aquatic biomass. The IPRO did not reach a conclusion on which exactly was the best. The geographical location of the refrigerator would determine the quantity and cost of fuel. They did find that the fuel needed to be low in moisture content, less than 60% moisture, and long burning, four hours or more so as to not need constant attention. After evaluating the thermodynamic cycle for multiple refrigerants the group decided on R134a for several reasons. First, it is relatively inexpensive and easy to obtain throughout the world. Also, it is environmentally friendly and would be able of producing an estimated Coefficient of Performance (COP) of 1.5.

The second IPRO team began building a prototype and divided the work into the mechanical subsystems and electrical subsystems. For the mechanical part they built the piston-cylinder apparatus composed of a high and low pressure cylinder and a piston between them, four solenoid valves to control the movement, and magnetic sensors to track the motion of the piston. This was connected to a standard window unit air conditioner that supplies the evaporator, condenser, and expansion coil. Also they began constructing the boiler system, but they did not finish. The electrical subsystem was a circuit constructed on a breadboard that took input from the magnetic sensors and controlled the solenoid valves. By the end of their portion of the project the piston-cylinder assembly along with the control circuit could be operated with compressed air. The boilers, however, needed a good deal of work.

Project Purpose

The purpose of this semester's IPRO was to start where the last group left off and finish the construction of the prototype and begin testing. This goal consisted of many smaller construction and testing goals because of the complexity of the apparatus. More precisely, some components had to be replaced, purchased and tested. Therefore, the IPRO group divided up into four smaller groups to work on the boilers, piping, control system, and analysis. Another objective is to begin developing a marketing strategy for this product. This will require research to identify possible markets and to understand the range of needs and requirement the system can meet.

The boilers will be tested separately and in tandem, in order to prove they are sealed properly. This involves reviewing the way they work, as well as the conditions of all their components. The piping needs many modifications and improvements. Some of the pressure gauges need to be replaced. A way to fill the system with refrigerant will be needed as well. Moreover, the fluids used for the tests and the refrigerant fluid will need to be acquired. The existing electronic system will be checked to make sure it still operates and if not repair or replace it. Data acquisition will use a computer to take data readings from the apparatus while it operates and record them for analysis. Also a program will be designed to control the prototype from the computer and effectively eliminate the previously existing control circuit.

In addition to objectives directly related to the project the group also will complete all the deliverables required by the IPRO office including the project plan, mid-semester report, meeting minutes, abstract, group presentation, poster, and final report.

Project Research Methodology

In order to achieve the project goals as stated earlier, the team will take the following steps:

Technical Aspects

- Repair leaks present in boiler system
- Amplify thermocouple signals and feed them into computer controller
- Interface pressure exchanger with computer controller
- Devise and implement a way to fill and purge system of refrigerant
- Analyze system, devise and implement ways to optimize the system

Business Aspects

- Describe the problem that we are trying to solve, how we solve it, and the advantages and limitations of our solution
- Determine who the customer will be, what the barriers will be to introducing this product, and what the competition is for this product
- Determine the cost to start production and what the profit margins will be

Team Organization and Individual Assignments

To complete the construction of the refrigerator, we divided the group into four teams: The boiler team, the piping team, the control system team, and analysis team.

Boiler Team

Stefan Vogel, Alec Frost, Alex Callow

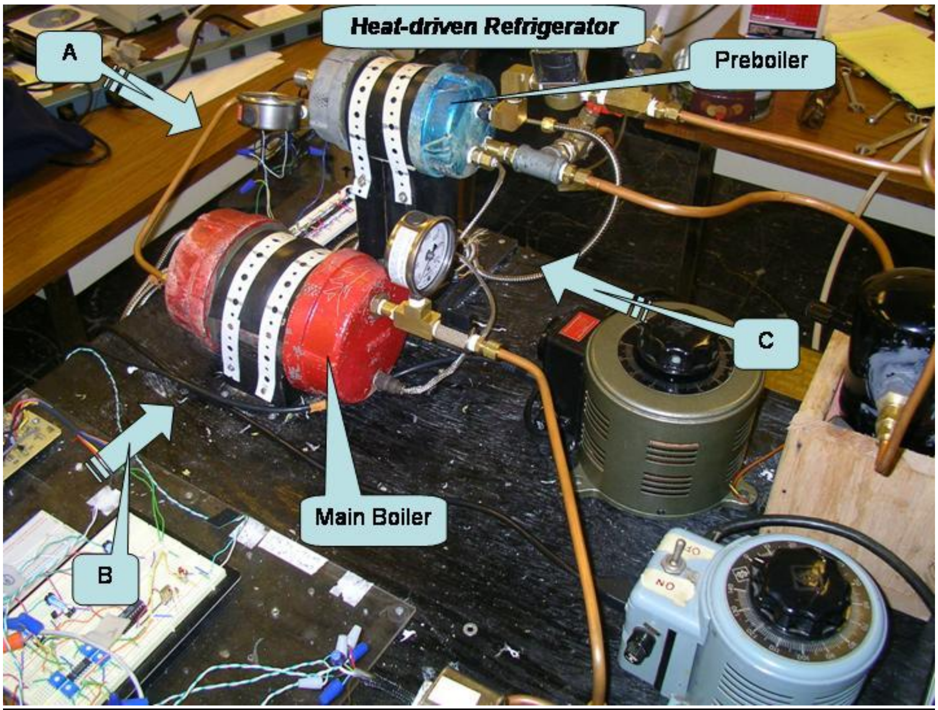


Figure 1: Boiler Subsystem—Complete View

Status at the beginning of the semester

At the beginning of the semester, the boiler subsystem was assembled and the control circuit was working. This meant that the pre-boiler would properly charge and discharge. A problem, however, was that the boiler system leaked at about 40psi, while the target is 400psi. The main problem seems to be the large threads of the pipe end caps. These caps form the pressure tanks by sealing off a piece of pipe, but the threads do not seal well with Teflon tape. Another problem is the position of the threaded holes and sensors. The position of both the floating sensors and inlets/outlets relative to the pressure tank have to be very accurate, otherwise charging and discharging may not operate properly. As the small threads for the sensors and connections are already drilled and alignment/position of these sensors is critical, one cannot screw on the caps onto the pipe all the way which may lead to improper tightness and leakage. Also, some check-valves seem to be missing or do not operate. It is believed that proof-testing the system with tap water during the last semester may have obstructed the valve mechanism with lime.

Accomplishments to date

The first action we took was to disassemble the boiler subsystem and clean all the parts with brake cleaner. To do this all of the tubing, sensors and heaters were disconnected. We marked the correct position of float sensors and correct position of the two caps on each boiler relative to each other before disassembling it all. Then we fixed the leaky boilers by using existing caps and tightening them so the position of the sensors and connections was correct. The threads were then sealed with JB Weld. We cleaned the small threads in the caps of all the glue that was stuck on them and sealed each boiler and tested each by itself with distilled water. This test proved that the main-boiler was without leaks up to 300psi and the pre-boiler was without leaks up to 200psi. This was a great improvement from the previous semester.

The next thing done was buying and intalling new check-valves and tube-fitting adaptors. With this completed we assembled the whole Boiler-Subsystem and proof-tested it with distilled water under operating conditions (but disconnected from the rest of the refrigeration system). The system worked properly over 100psi without leaking and we achieved two proper charging-discharging cycles of pre and main boilers.

Then we fixed a second power supply unit so the two heaters for the pre- and main boiler could be run independently. We installed pressure gauges capable of reading pressure up to 600psi on the main boiler and the pre-boiler as well as a new heater in the pre-boiler so that both boilers now have 1kW heaters. Following this the pre-boiler and main-boilers were tested independently up to 450psi and then together up to 450psi with Ethanol rather than distilled water. In order to do all this many piping provisions (elbows and valves) were made to hook up the vacuum pump for evacuating air and filling system with refrigerant. Minor smoke did come from the rubber bands holding the boilers and this can be avoided by using heat-resistant material.

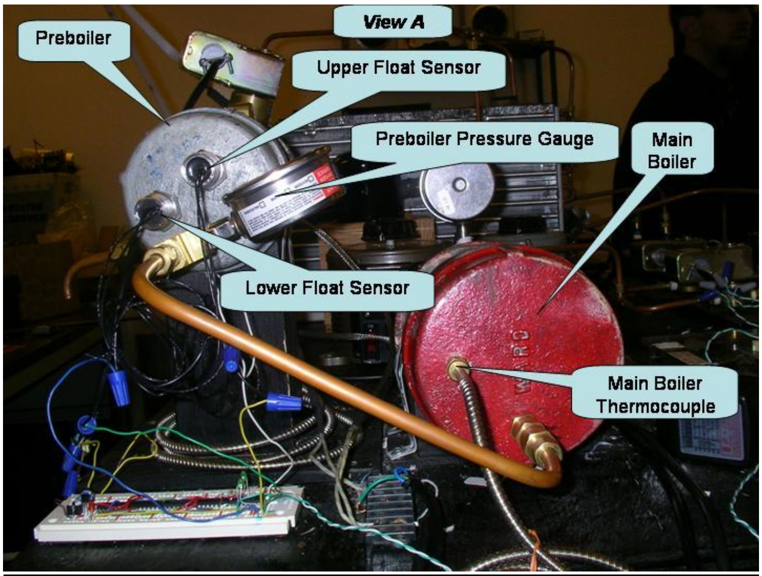


Figure 2: Boiler Subsystem—Side View A

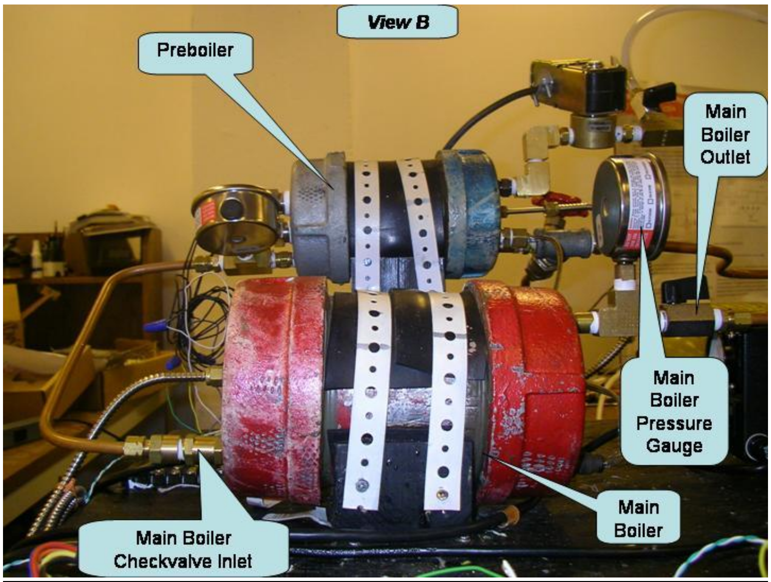


Figure 3: Boiler Subsystem—Side View B

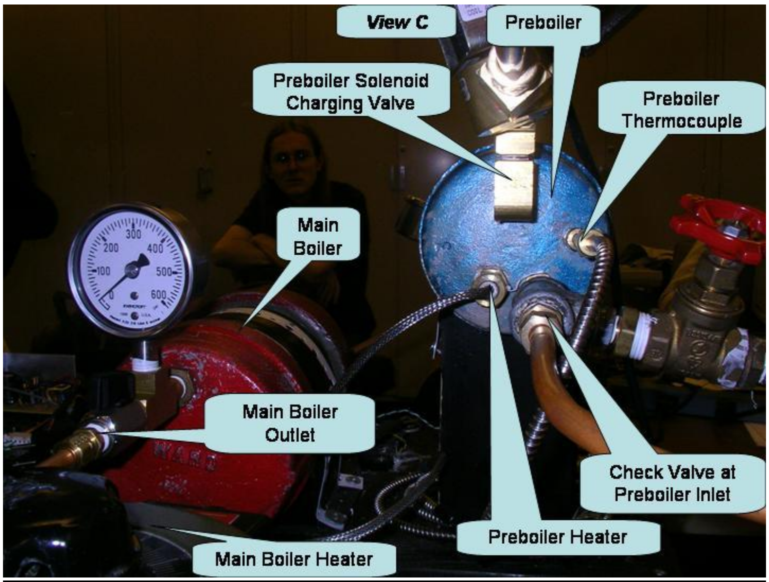


Figure 4: Boiler Subsystem—Side View C

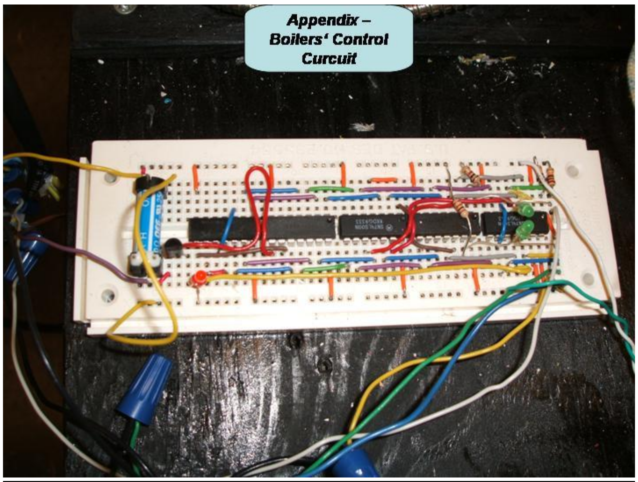


Figure 5: Boiler Subsystem Control Circuit

Analysis Team

Leonardo Nortes Planas

Status at the beginning of the semester

At the beginning of the semester the IPRO team had to face a very important problem: the complete lack of information. There was no information about why the previous IPRO groups had chosen the considered working conditions for the refrigeration machine, nor about why refrigerant R134a had been chosen as the refrigerant fluid. Some parts of the equipment were not suitable for the proper performance of the device. There was no reliable data in regards to the coefficient of performance of the machine, nor was there significant data on the impact of heat loss in the boiler and pre-boiler.

Accomplishments to date

A complete analysis of the overall performance of the machine has been made. An exhaustive study about the different refrigerant fluids available has been completed. The suitability of refrigerant R134a has been proven. Given the choice of refrigerant R134a, the optimum conditions for the thermodynamic cycle have been searched. By means of using a computer program, several working conditions have been compared. Thus, new estimations of the coefficient of performance of the machine have been obtained.

The importance of the heat loss in the boiler and the pre-boiler has been studied. Numerical results have been obtained, proving the convenience of insulating both the main boiler and the pre-boiler. The refrigeration power that the device can provide was calculated. Since the power of the electrical heater in one of the boilers was found to be too low, a new and more powerful heater has been installed.

Data Acquisition Team

Donghoon Lee, KeonWoo Kim, Tom Alworth

Status at the beginning of the semester

At the beginning of the semester, the goal of the Data Acquisition Subgroup was to gain computer control of four solenoid valves in refrigeration system by measuring four temperatures using thermocouples and using this information to help control the speed of the pressure exchanger.

Accomplishments to Date

The first problem we encountered was that we were unsure whether the data acquisition system in place from the previous semester. We attempted to test the IOTech Data Acquisition (DAQ) card and at first we had very little success getting the computer to recognize the card at all. Finally, we determined that only the LabVIEW program written last semester recognized the DAQ card. The other companion programs that work with LabVIEW did not recognize the card and neither did any other program in LabVIEW. This provided a major problem because the LabVIEW program written the previous semester was not able to control the valves in the refrigeration system at all, it only measured the temperatures at the four locations.

We also tested the existing program using a thermocouple and found that the resolution was only to 20mV. The needed resolution for accurate temperature reading in our application is around 0.2mV. We were unable to change the resolution and therefore decided that a solution to this would be to build some amplification circuits for the thermocouple signals. We contacted National Instruments to obtain a sample of some high precision operation amplifiers (op amps) to build these circuits. We also built a trial amplification circuit using standard op amps (LM 324) to test our theory that this would solve the resolution problem. Unfortunately, this did not solve the resolution problem, either. We also determined that the program written the previous semester was insufficient and could not be used.

After much time and energy unsuccessfully trying to get the IOTech DAQ card to interface properly with LabVIEW, we decided that a better solution would be to obtain a DAQ card and terminal board that would definitely interface properly with LabVIEW. After some research we found and ordered a DAQ card, terminal board, and cable from National Instruments (NI) that would satisfy our needs and that we were certain would interface with LabVIEW. While waiting for this new equipment to arrive, we worked on writing a new LabVIEW program to better accommodate our data processing needs. We talked with Dr. Ruiz multiple times to gain a fuller understanding of how the magnetic sensors on the pressure exchanger and the pressures (calculated from the temperatures obtained from the thermocouples) will work together in this program to control the pressure exchanger and the rest of the refrigeration system.

The new NI DAQ card finally arrived after a few weeks of delay and we set out to see if the LabVIEW program we had written would work. The picture in below indicates the DAQ card and terminal board we ordered. The first problem we encountered was that the new card only works with LabVIEW Version 7 or higher, while we had written the program using Version 6. As a result we had to rewrite the program. This was another problem because LabVIEW Version 7 is significantly different from Version 6 (i.e. most of the functions were changed) and it took a lot of time to figure out how to use Version 7.

Figure 7: NI PCI-6220 M Series Multifunction DAQ

Figure 8: NI Terminal board

We tried to control the valves in the system by using switches in the LabVIEW program, but when the wire was connected from the terminal board to the circuit from the last semester we observed that the circuit did not work properly. The DAQ card may drain the current from the circuit when connected in this way, thus prevent the circuit working. In the end, we decided that we did not have sufficient time to figure out how to use LabVIEW Version 7 well enough to operate the refrigeration system properly. Therefore, we decided to use both the circuit from the last semester for controlling the solenoid valves *and* the LabVIEW program for measuring temperature. Currently, we have been successful in measuring the temperature with the LabVIEW program to a high resolution and the circuit controlling the valves is working properly. The picture in below indicates the front panel and block diagram in LabVIEW program we made for measuring temperature and controlling four valves.

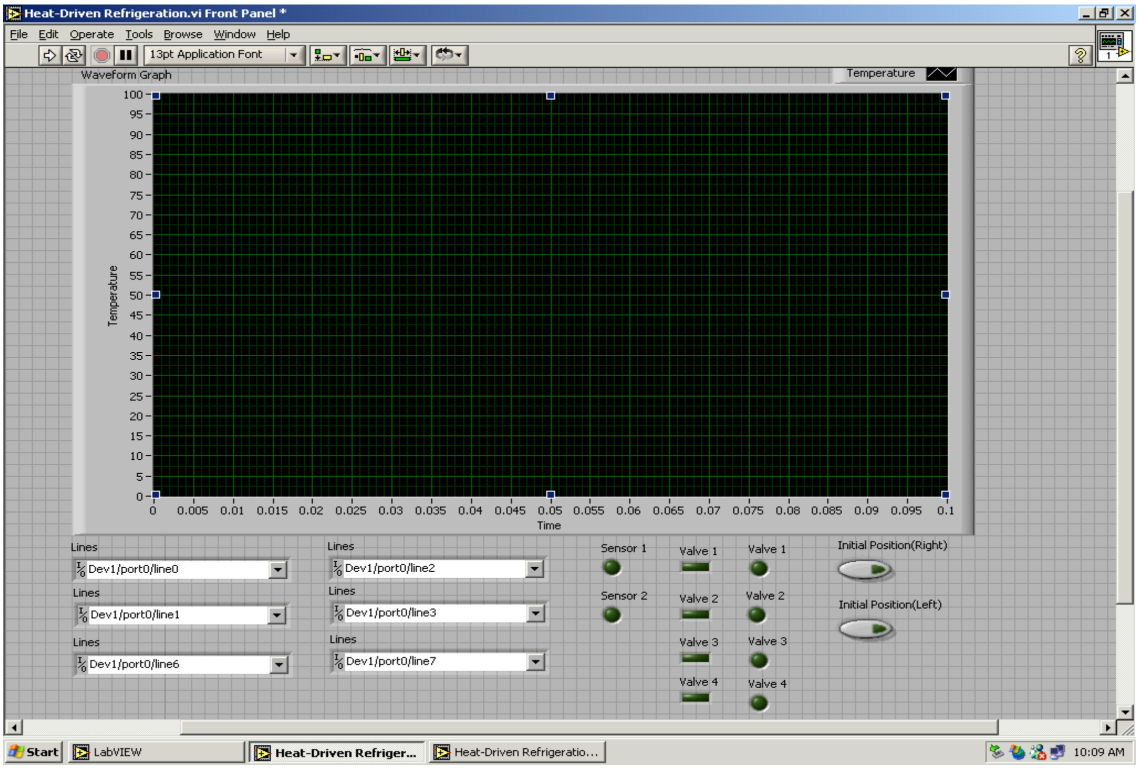


Figure 9: Front panel of LabVIEW program

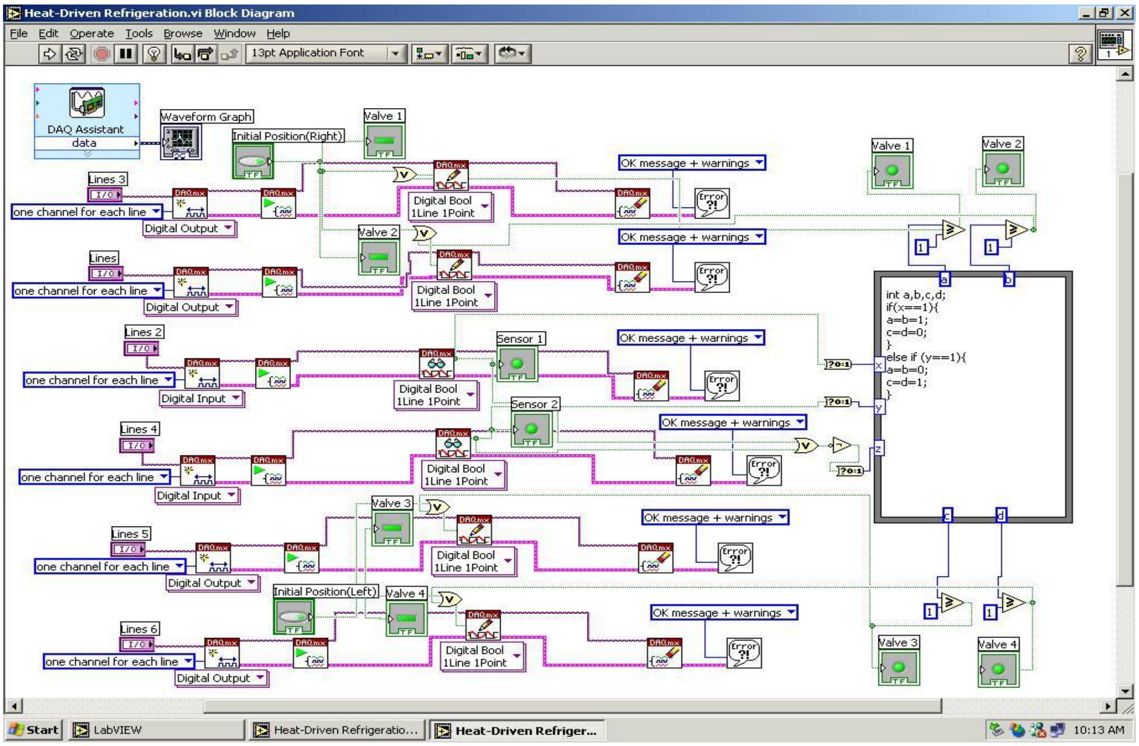


Figure 10: Block Diagram of LabVIEW program

Refrigerant Transportation and Control Team

Anel Medrano, Andrew Keen

Accomplishments to Date:

Unlike a store bought air conditioning unit, the filling of a refrigerant is an added procedure for the initial operation of our prototype. Since water has been the testing fluid so far, it needs to be purged from the system. To do so, a vacuum may be used to expunge the water to make way for the planned R-134a. The testing of the data acquisition portion of our project currently postpones the use of the planned R-134a. Instead, when the apparatus is in working order, the fluid that will be used to test the system will first be a non-toxic chemical prior to testing with the refrigerant. Currently the prototype has been fitted with a valve for the purpose of filling the system with ethanol for preliminary tests.

To fill the system with refrigerant, we followed recommendations of the previous semester by using a Schrader type valve similar to one found on a bicycle tire. Schrader valve is to be used to provide a one-way directional flow into the piping system. Due to prior existence of automotive air-conditioning systems utilizing R-134a systems, an accumulator from an automotive system (a 1996 Ford Crown Victoria to be exact) was installed to provide a filling point for the system. The choice was made for its two-fold benefit, not only does the accumulator contain a Schrader valve designed for R-134a type refrigerant, but also contains a desiccant to remove any excess water in the system to prevent formation of ice crystals that could damage the system. In addition, a fueling gun designed for filling automotive air-conditioning systems and the appropriate refrigerant have been purchased.

Our machine shop was able to custom-make some fittings to adapt the accumulator over to fit our system. In order to prevent leakage, these fittings were JB Welded into place. The placement of the accumulator is just before the pre-boiler of the system, which by pressure lies in the midpoint of the cycle. Because this point receives pressure in excess of 100 PSI, the accumulator was pressure tested up to 160 PSI for safety reasons. There were no leaks.

Product Team

Tom Alworth, Alec Frost, Stefan Vogel

Status at the beginning of the semester

No previous work had been performed on the economic side of the product at the start of the semester. The main goals this semester were then to state the problem, find the significance of the problem, and identify possible barriers that could hinder progress. At which point the method of solving the problem needed to be identified in terms of our specific product and the business type that would break the product into the market. In order to accomplish these tasks a detailed description of the product itself needed to be made.

Accomplishments to date

The main problem identified is that currently there are few efficient refrigeration systems available that do not depend on a large consistent electricity supply. Because of the limited availability of refrigerators independent of electricity the quality of life in areas without electricity suffers. Proper refrigeration is needed for the preservation of food products, availability of vaccinations, and to guarantee the potency of certain medications. Despite the seemingly open market for a product such as the current prototype, the path is not without barriers. Currently the technology is unproven, and uncertainty could hinder the marketability of the product. Although there are not many other electricity dependant systems on the market, there is an absorption cycle system on the market that could be potential competition. Our device can potentially solve the problem identified by being heat-driven and hence independent of electricity, thus improving the quality of life.

Our product specifically is a refrigeration cycle that consists of a pressure changer an evaporator, a condenser and capillary tubing (also called an expansion coil). First, the refrigerant in the evaporator removes heat from the space to be cooled at low pressure. Second, the pressure is increased to the medium pressure in the pressure exchanger. Third, the refrigerant flows in the condenser where it releases into the outside environment. Then the pressure is reduced back to the evaporator pressure by the capillary tubing.

The power cycle consists of the condenser, the pre-boiler, the main-boiler, the pressure exchanger and a low pressure suction valve. The medium pressure refrigerant in the condenser, in addition to flowing to the capillary tubing as part of the refrigeration cycle, is sucked into the pre-boiler when the low pressure suction valve is open. This low pressure suction comes directly from the low pressure line between the evaporator and pressure exchanger. The pre-boiler heats the refrigerant until it begins to vaporize. The vaporization of the refrigerant causes the pressure to begin to rise and this increase in pressure forces the refrigerant into the main-boiler. In the main boiler vaporization is completed and the refrigerant pressure is further increased to the highest pressure in the system. The refrigerant then exits the boiler into the pressure exchanger. The pressure exchanger uses this high pressure refrigerant from the power cycle to increase the pressure of the refrigeration cycle from low to medium.

Our particular product has the advantage of being mobile, easy to use, and requires minimal electrical input. Because it is heat-driven it has the potential to run on multiple heat sources.

Marketing Team

Keon Kim, Donghoon Lee, Anel Medrano, Leo Nortes

Status at the beginning of the semester

No previous market research had been conducted in previous semesters. Goals for this semester were to identify possible consumer market for our particular product, analyze the market size and our ability to capture that market, and our marketing strategy. It is also essential to determine possible competition in the designated market to estimate how successful our product will be in the marketplace.

Accomplishments to date

The refrigeration machine this project deals with might be of interest to many relief organizations or government agencies. Some relief organizations have been contacted, with the purpose of getting to know how they get the refrigeration equipment they need, and in order to try to get to know how much money they expend on that equipment. No important results have been obtained from this research. The only noticeable results that have been obtained are related to the kind of refrigerant equipment used for some relief organizations. Potential relief organizations such United Nations International Children's Emergency Fund (UNICEF), the World Health Organization (WHO), and Doctors Without Borders have been identified but future research is needed. Information is available in the Refrigeration Equipment for Relief Organizations Annexe. On the other hand, very useful information has been obtained from government agencies, as the Supply to Federal Agencies Annexe reports. The Defense Logistics Agency (DLA) uses its government position as a large volume buyer to negotiate favorable procurement for commonly purchased items. Through this federal agency we can reach the government, its agencies and also many civilian relief organizations. This seems the most sensible way to make our product reach the market.

Other areas in which this refrigeration system might be useful are in developing countries and in places where electrical supply is not reliable. Hence, a possible market may be for recreation, but further research is needed.

Because substantial capital is needed to begin manufacturing, licensing for our product is the ideal marketing strategy. Four major refrigerator companies have been found. These companies—Amana, Whirlpool, Kenmore, and Frigidaire—are experienced in the field of refrigeration, mainly vapor-compression refrigeration. Since vapor-compression is one half of our double-cycle the only completely new technology to these companies is the rankine power cycle which is the other half of the double cycle. Here there may be some design adoptions for large scale manufacturing necessary. As soon as the prototype is set-up as a whole unit and first optimization iterations have been performed by the next IPRO team, these companies may be contacted as potential licensees.

In regards to competition, most companies in the market currently use the two known thermodynamic cycles – vapor-compression or absorption. There are several companies which have specialized on absorption-run refrigerators which use mainly propane gas as a heat source. These refrigerators are for recreational use. There seem to be a few manufacturers which produce same refrigerator for many companies. Our research found six specific competitors including Danby Products Inc. , Sunfrost, Dometic, Atlantic Mini-Fridge Company Ltd., Equator Advanced Supplies, and Crystal Cold (see Appendix B). Moreover there are a small number of companies which offer highly specialized refrigerators on the basis of the vapor-compression cycle. These are designed in such a way that they can be powered by solar energy through photovoltaic panels. Research showed two companies offering such devices. Current analysis shows that there is no device using the Double-Cycle developed in this IPRO. This may be a niche in the refrigeration market.

Financial Team

Andrew Keen & Alex Callow

Status at the beginning of the semester

No financial research or calculations had previously been performed. Goals this semester were to estimate profit margins, project sales, project costs and income, and estimate financing requirements.

Progress to date

Using the tools found on the internet a compressor system was tailored to a system load of about 24000 BTU's. Prices were quoted for compressor-based systems at four different efficiency (SEER) ratings ranging from 10 to 14. The coefficient of performance (COP) was calculated for each as a means of comparison for each of the compressor systems, and the predicted COP of our prototype was converted to a SEER rating for the opposite comparison. Other products found on the same website that could be used to construct a large-scale version of our own heat-driven prototype were noted as well. Comparisons were done for a split system—one in which the condenser is separate from the evaporator—and one for a self-contained system—the condenser and evaporator being in the same unit. Fuel costs were noted as well for means of comparing the cost-effectiveness of the two systems for comparison. Figures for the prices during the months of January through July 2004 were averaged in order to calculate a standard price. Because the SEER efficiency ratings are according to the units BTU/kwh, the price of all fuels were converted to the units $/kwh. Using the component prices as a base price and the fuel costs and efficiency ratings figures, the price of fuel consumption was processed and charted according to the BTU output from 2000 to the maximum capacity of the compressor system, 24000 BTU.

Although our system has a higher cost as far as base price, it is significantly cheaper for fuel consumption. The main reason why the base price is so high is the non-existence of a boiler system that is designed for our process. Most boilers available are used for the purpose of heating water or industrial chemicals on a large scale. Thus the industrial boilers are significantly larger than would be needed for our purposes while the residential water-boilers are not pressure tested beyond 250 PSI, significantly less than the required 400 PSI operating pressure used for our process. Existing boilers that can withstand such a pressure tend to be expensive. Also, the use of two boilers makes the system quite costly when using existing products because they are designed to work individually. Should an existing 2-stage boiler be adapted over for our purposes, and properly pressure rated, the cost could potentially drop to a range closer to that of a compressor system. As far as the efficiency of the system, it seems our process has a lower rating than the compressor-based systems. This is mainly because of the use of the Refrigerant R-134a, which was chosen for environmental concerns. The compressor systems noted for comparison use the refrigerant R-22. To observe the significance of changing refrigerants on the COP ratings consult Appendix A. The relatively low cost of fuel used to power our system does tend to make up the difference in the long run. Future groups should contact some boiler companies in order to attain a more accurate cost for production of a boiler useable by our system.

Barriers and Obstacles:

Of course, when contsructing an untested prototype system of any type, problems are going to come up. This section highlights the barriers and obstacles that were faced.

Critical Barriers

One of the ongoing problems with the float sensor circuit was that it had intermittent problems that caused difficulty closing the vacuum solenoid. This was probably a problem of actuating the upper float sensor with the liquid level.

Regarding the analysis of the system the main barrier to make a more accurate and exhaustive analysis of the way this device works was the lack of information about the different components that are part of the prototype. Since many of them have been obtained from old devices, it is very difficult to get precise technical information about them. This information could be very useful in order to get to know if the prototype has been built using the most appropriate components.

Our preliminary test with ethanol found that getting the piston moving at the start of the cycle was more difficult than expected. Pressure seemed to be equalizing throughout the system and piston movement was not working properly. Testing of the system with ethanol was attempted, but due to leakage and a few other problems, the test was unsuccessful.

The biggest barrier with the data acquisition and control of the system was that none of us had enough experience with LabVIEW to know how to control the system completely. We were able to measure temperatures, but when the DAQ card was connected it seemed to drain the current from the circuit that operates the pressure exchanger. This problem was never resolved

Major Obstacles

One of the major setbacks of this project was finding that the DAQ card and LabVIEW program used in the previous semester were not going to work for what we were trying to do. We were forced to obtain a new DAQ card, which involved weeds of waiting, and write a LabVIEW program from scratch without prior knowledge of how to program in LabVIEW. Though we did get the card and were able to write a program, much more could have been done without this delay.

Another significant obstacle was the placement of the float sensors in the pre-boiler. The placement from the previous semester did not allow the precise movements that were needed from optimum operation and for that reason new endcaps for the pre-boiler had to be designed and constructed.

Results and Conclusions:

On the technical side much progress was made over the course of the semester. The boiler system has been pressure tested and does not show any leaks up to well past 600psi in the main boiler and up to 450psi in the pre-boiler. The float circuitry does work but has had intermittent problems with shutting the solenoid at the appropriate time so further investigation may be necessary. All check valves and pressure gages are working properly and there are no signs of leaking in the pipe connections between the boilers.

A complete theoretical analysis of how the refrigeration device is supposed to work has been done. Some numerical results have also been obtained. Therefore, the value of the coefficient of performance the prototype is supposed to accomplish during its performance is already known. Some other technical features, like the suitability of insulating the boilers, are also known. All the information collected has been compiled into a report, which could be of good help to next groups working on this project.

A method of filling the system with refrigerant has been completed—namely the accumulator—and a supply of refrigerant 134a has been obtained. The air and moisture in the system can be evacuated using the vacuum pump and then the refrigerant can be added through the Schrader valve on the accumulator using the adaptor which connects directly to the refrigerant canisters.

Currently the pressure exchanger cannot be controlled by the LabVIEW program, as was a goal at the beginning. This can be achieved, though, with a bit more time and knowledge of LabVIEW Version 7. Despite this the device still is able to operate using the control circuit from the previous semester—modified slightly.

The conclusion that can be reached regarding the technical aspects of this project is that the prototype is completed, but work still remains in testing and optimizing it, which will involve gaining computer control of the pressure exchanger.

Recommendations and Next Steps:

Based on the work completed and experience gained on this project, we are able to make some recommendations for the future. In the optimization process it seems best to explore the option of using circuitry to control the heaters rather then by hand control. Also, insulating the pipes and boilers would greatly decrease the heat loss and thus decrease the power required by heaters, making the cycle more effective. Also, it is recommended that the LabVIEW program be used to control the solenoid valves in the next semester. By measuring the temperature in the refrigeration system, the valves can be controlled more effectively and make it possible to achieve a higher efficiency in the refrigeration cycle. In order to use LabVIEW to control the system, the problem with the connection between the circuit and the DAQ card needs to be solved

A practical step that should be taken is to modify the float sensors so they are easier to switch in a light fluid such as ethanol or refrigerant such as 134a. A more sensitive float sensor or a new sensor position relative to the vacuum solenoid outlet may ensure that the solenoid is activated properly. Also, the magnet sensors in the circuit are still a little too sensitive. To correct this problem, it is recommended that a more accurate amplifier be constructed. Further tests with ethanol or similar solution should be conducted to find any leaks before testing the system with R-134a. This would also be useful to ensure that the control system is working properly once it is modified. It will also be necessary to monitor the pressure exchanger solenoid valves opening and closing during the filling process, or perhaps design a manual start-up process for the cycle.

One recommendation simply for safety is to install a pressure relief valve to prevent the possibility of damage to system components and increase overall safety of the device. The proposed location of such a valve is at the main boiler outlet.

Since the prototype is ready for testing, all the theoretical results that have been obtained should be compared to the actual values yielded by the machine in its performance. The actual values of the coefficient of performance should be calculated, and some variations in the performance conditions of the machine should be done in order to see if the theoretically optimum conditions are really the best for the machine to work. Also, the efficiency of the prototype should be compared to that of regular refrigeration devices, and some feasible ways to improve the efficiency of our machine, like working beyond the critical point – which right now has been disregarded because of safety reasons – should be studied.

APPENDIX A1 - Used software.

In order to make the calculations more easily, the computer program EES (Engineering Equation Solver) have been used. This is an equation solver that can be used so as to simulate thermodynamic cycles, due to the fact that it has some thermophysical functions implemented.

The next EES program was used to simulate the actual thermodynamic cycle:

```
{HEAT DRIVEN REFRIGERATION CYCLE, USING BATCH PRESSURE
EXCHANGER}
{after January 21st, 1992, enhanced August 11th, Sept 3rd}
{new version July 20, 1999}
{brand new version September 24, 2004}
{The cycle expands saturated vapor at the top temperature, using an isentropic
batch pressure exchanger, all the way to the bottom temperature.
No regenerative heating. The efficiency of  the pressure exchanger is taken into
account}

{data}

T1=196  {F top temperature}
T6=98,6    {F middle temperature}
T9=35,2  {F bottom temperature}
Presseff=0,9   {isentropic efficiency of the pressure exchanger}

V1=VOLUME(R134a;T=T1;X=1)
V10=VOLUME(R134a;T=T9;X=1)
V6=VOLUME(R134a;T=T6;X=1)

p1=PRESSURE(R134a;T=T1;x=0)
p6=PRESSURE(R134a;T=T6;x=0)
p9=PRESSURE(R134a;T=T9;x=0)

{main states}

h1=ENTHALPY(R134a;T=T1;x=1)
s1=ENTROPY(R134a;T=T1;x=1)

h2s=ENTHALPY(R134a;p=p6;s=s1)  {isentropic case}
h2= h1-Presseff*(h1-h2s)       {real}
T2=TEMPERATURE(R134a;p=p6;h=h2)

h3=ENTHALPY(R134a;T=T9;x=1)
s3=ENTROPY(R134a;T=T9;x=1)

h4s=ENTHALPY(R134a;p=p6;s=s3)  {isentropic case}
h4=h3+(h4s-h3)/Presseff        {real}
T4=TEMPERATURE(R134a;p=p6;h=h4)

y=1/((h1-h2)/(h4-h3)+1)     {first law in pressure exchanger}
{if y > 1 or y < 0, the cycle is not possible}

h6=ENTHALPY(R134a;T=T6;x=0)

Wp=VOLUME(R134a;T=T6;x=0)*(p1-p6)*144/778,169  {the factor converts psi*ft3/lb
into BTU/lb}    {no factor: in kJ/kg if p in kPa}
h7=h6+Wp
```

{efficiency calculation}

Qhot=y*(h1-h7)
Qmid=h4-h6
Qcold=(1-y)*(h3-h6)

COP=Qcold/Qhot
COPp=Qcold/(Qhot+Wp*y)   {with the pump}

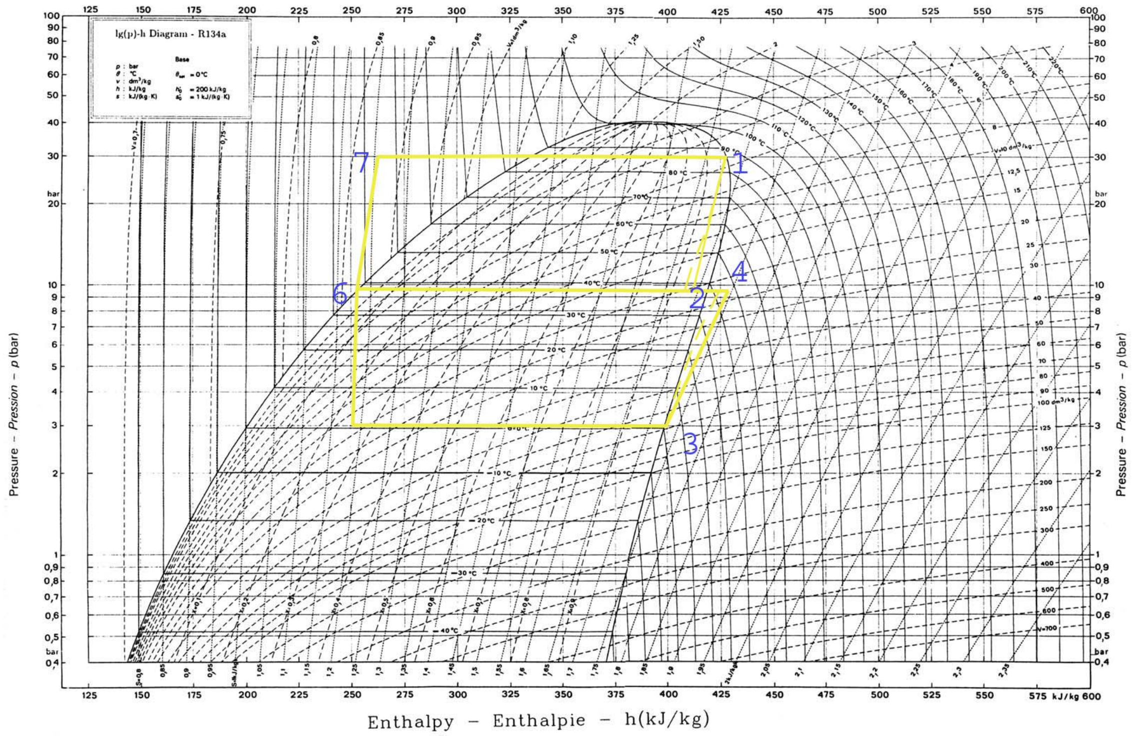
COPideal=(1/(T6+460)-1/(T1+460))/(1/(T9+460)-1/(T6+460)) {Carnot, no pump}

Eff2=COP/COPideal {2nd law efficiency}

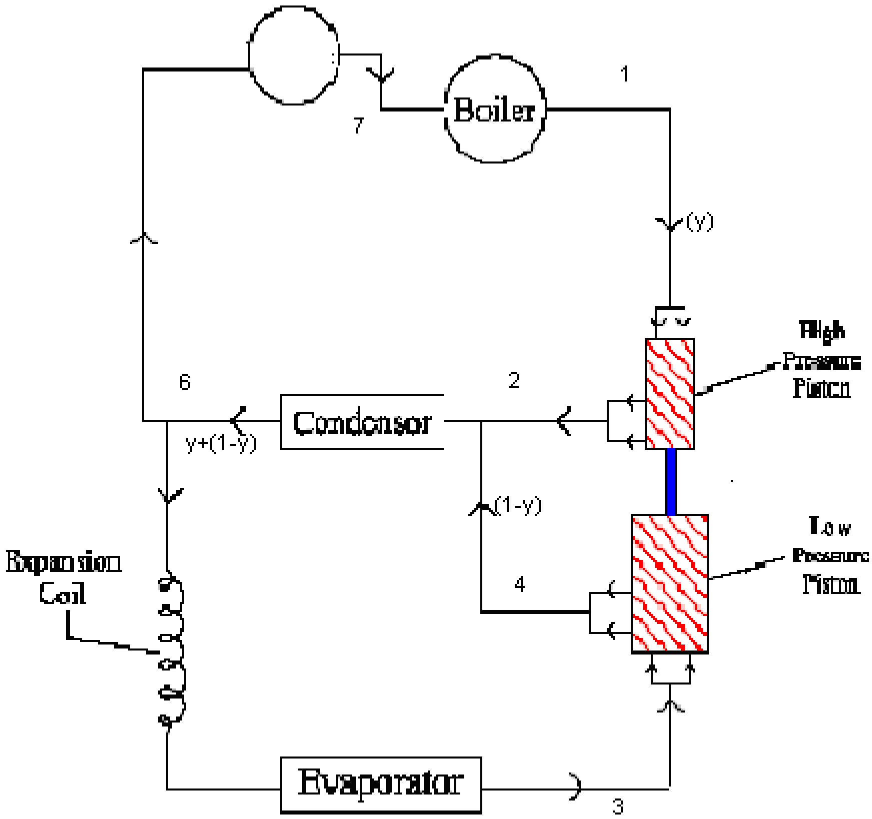
Qvratio=V10/Qcold

{Options: English units, F, psia, mass basis}

The thermodynamic cycle corresponding to this program is shown in the next pressure-enthalpy diagram:



The device that could run this cycle is shown in the next figure:



The parameter "y" represents the fraction of the total mass flow of refrigerant that flows through the expander.

In this cycle, the pre-boiler has been substituted by a pump. This does not affect the final results as long as the pump work (Wp) is regarded as the heat supplied to the fluid in the pre-boiler.

One of the main differences between this and the real cycle in that in the actual cycle the process between points 6 and 7 takes place in the pre-boiler, instead than in a pump (as the simulation implies). Anyway, this difference is very little, since in both cases the fluid is compressed maintaining its volume constant. Another difference is the fact that the simulation cycle does not consider the flow of refrigerant that takes place from the pre-boiler to the evaporator outlet when the vacuum valve in the pre-boiler opens.

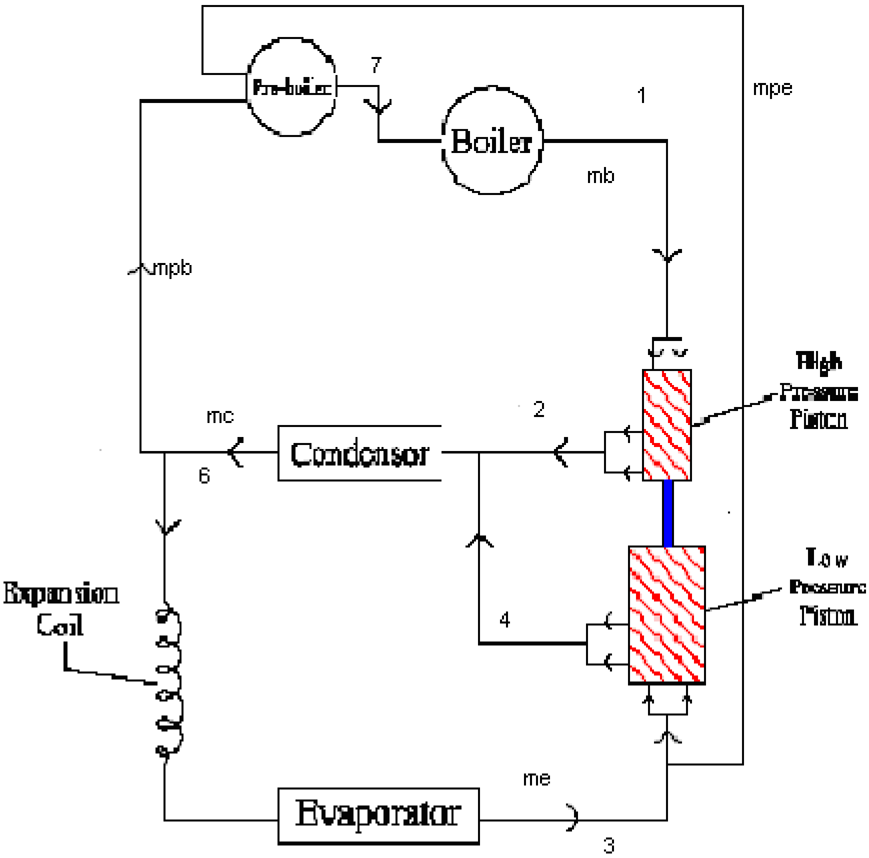
A -. 1. Accuracy of the simulation.

The comparison between the results of the actual cycle and those of the simulation has been done for the optimun working conditions.

- Pressure at point 1: 435.11 psia (30 bar).
- Pressure at point 6: 138.9 psia (9.6 bar).
- Pressure at point 9: 45 psia (3.1 bar).

The features of the points of the thermodynamic cycle are shown next (the enthalpies have been calculated according to the diagrams enclosed):

- Enthalpy at point 1: 430 kJ/kg.
- Enthalpy at point 2: 418 kJ/kg.
- Enthalpy at point 3: 390 kJ/kg.
- Enthalpy at point 4: 428 kJ/kg.
- Enthalpy at point 6: 245 kJ/kg.
- Enthalpy at point 7: 260 kJ/kg.
- Mass flows are shown in the figures bellow.

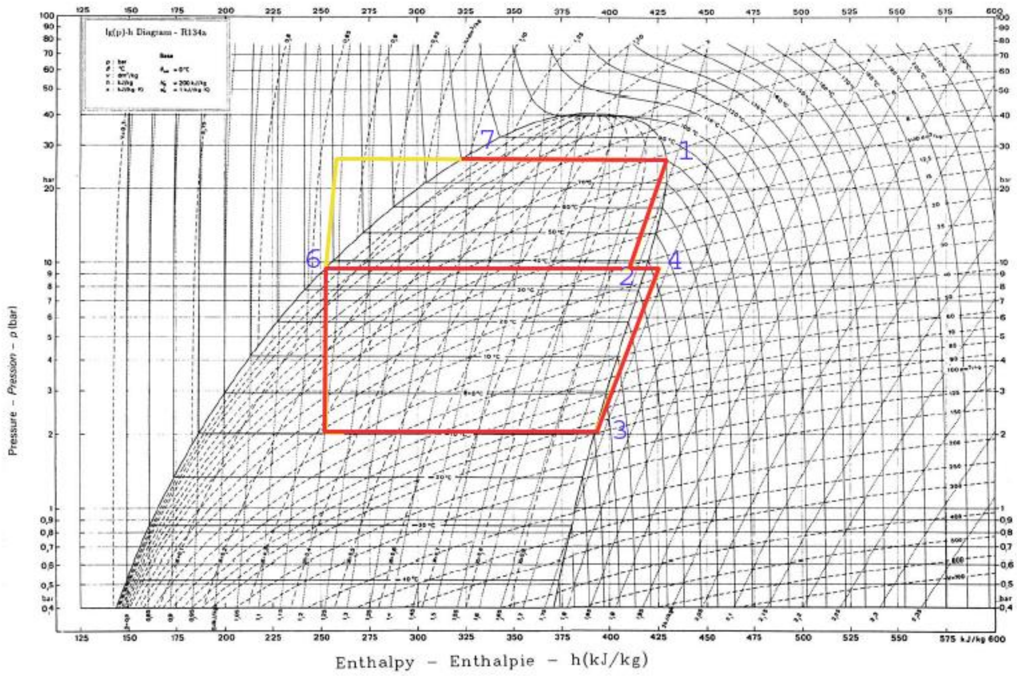


The main differences between this cycle and the simulation related to the pre-boiler and the refrigerant flow from it to the evaporator outlet.

As far as the pre-boiler is concerned, it does not work continuously. The pre-boiler absorbs refrigerant – thank to the solenoid valve with which it is equipped – and, once is full with refrigerant, turns into a closed system – as all its valves close –. Then, the fluid confined within the pre-boiler is heated at a constant volume, and the outlet valve do not open until the pressure inside the pre-boiler rises to the maximum pressure value in the cycle (around 400 psia). When the outlet valve of the pre-boiler opens, the refrigerant, as saturated liquid, flows from the preboiler to the main boiler. The flow from the pre-boiler to the main boiler, and the flow through the condenser and the evaporator can be considered as continuous flows (the higher the frequency af the expander and compressor, the more accurate this statement is); the calculations of the heat rates in the evaporator and in the condenser, as well as the energy balance in the pressure-exchanger can be done, therefore, as if the processes were continuous.

The pre-boiler´s capacity is 42.41 in^3 (695 cm^3). When it is full of liquid refrigerant coming from the condenser, this liquid is at 138.9 psia (9.6 bar). At that pressure, the density of the refrigerant is $9.3 \cdot 10^{-4}$ kg/cm^3. Therefore, the quantity of fluid refrigerant contained in the pre-boiler is 0.647 kg/cycle. When the pre-boiler runs out of liquid refrigerant, it only contains saturated vapor at a pressure slightly bellow 400 psia. The density of this vapor is $1.52 \cdot 10^{-4}$ kg/cm^3. Therefore, the quantity of refrigerant remaining in the pre-boiler right before the solenoid valve´s opening is 0.105 kg/cycle. This vapor is the refrigerant that flows to the evaporator outlet when the solenoid valve opens. Thus, the amount of refrigerant that in a cycle flows from the pre-boiler to the main boiler is 5.16 times bigger than that which flows from the pre-boiler to the evaporator outlet. In addition, the flow from the pre-boiler to the evaporator outlet occurs only in one particular and short moment in each cycle (when the solenoid valve opens) whereas the flow from the pre-boiler to the main boiler occurs during all the cycle except that moment in which there is a flow through the solenoid valve.

Therefore, during most of the time, there is not flow from the pre-boiler to the evaporator outlet and, thus, the actual cycle is this:



log p,h-Diagramm für R134a ($C_2H_2F_4$)

The processes in red in the figure are those that occur almost continuously, whereas those colored in yellow (corresponding to the processes that take place within the pre-boiler) only happen once in a cycle.

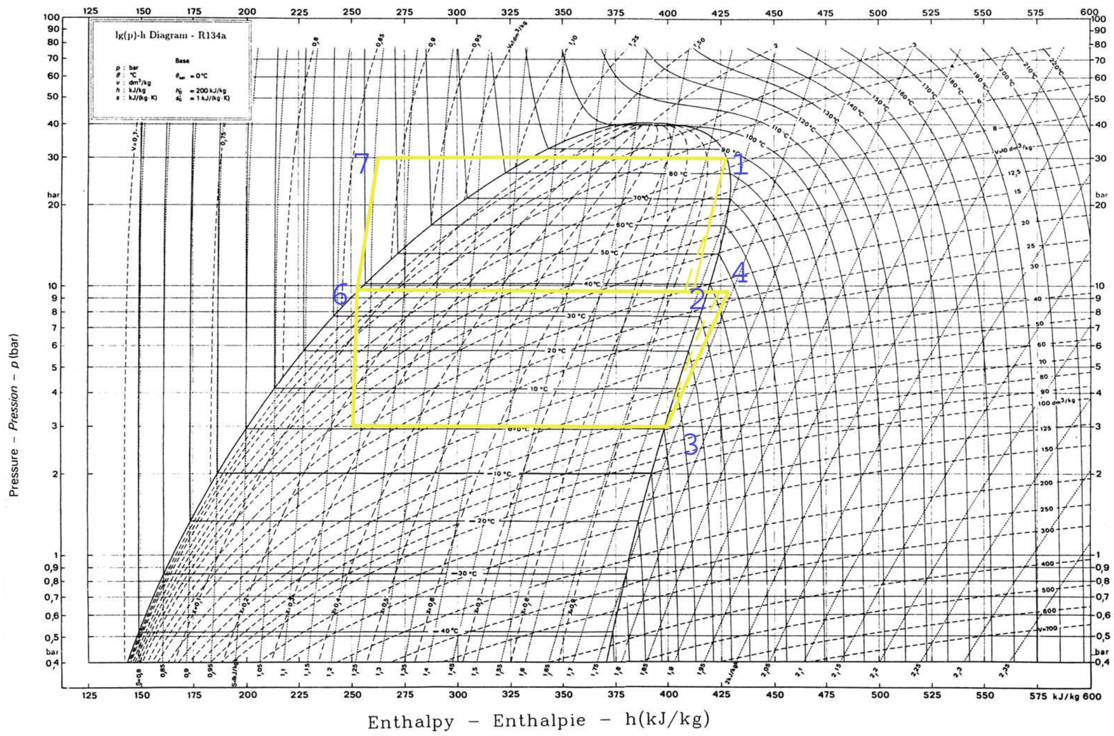
It should be noted that this cycle is exactly the same, in terms of power balances, as the one modelized by the computer program.

Only once in each cycle, and during a very short period of time, there is a refrigerant flow from the pre-boiler to the evaporator outlet. Then, the cycle can result slightly modified, due to the fact that the refrigerant may be superheated vapor, instead of saturated vapor, at the compressor inlet. The superheating is caused by the mixture of saturated vapor coming from the evaporator (at 45.1 psia) with saturated vapor coming from the pre-boiler (at 400 psia). Anyway, the enthalpy of the two blended flows is almost the same (the difference is around 40 kJ/kg) and, therefore, the cycle is barely affected.

As a conclusion, it can be stated that the computer simulation of the actual cycle is accurate enough.

A-. 1. 2. Computer analysis.

The program used to simulate the real thermodynamic cycle is shown next:



The computer model used for simulating the operation of the refrigeration cycle differs from the actual cycle in the fact that the former does not consider the refrigerant flow from the pre-boiler to the evaporator outlet.

The computer model also assumes that all the parts of the device work at steady state, which is not true but simplifies the analysis. Neither the compressor, nor the expander nor the pre-boiler work at steady state.

The compressor and the expander work the same way. Since they are reciproticating machines, the flows throgh them are not continuous. However, due to the high frequency with which they should move, the steady state assumption can be regarded as valid.

The results obtained using EES are shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,667	COPideal = 1,160	COPp = 0,659
Eff2 = 0,575	h1 = 118,795 [Btu/lb$_m$]	h2 = 110,397
h2s = 109,464 [Btu/lb$_m$]	h3 = 108,119 [Btu/lb$_m$]	h4 = 119,026
h4s = 117,935 [Btu/lb$_m$]	h6 = 44,629 [Btu/lb$_m$]	h7 = 45,513
p1 = 481,738 [psia]	p6 = 136,009 [psia]	p9 = 45,303 [psia]
Presseff = 0,900	Qcold = 27,618	Qhot = 41,403
Qmid = 74,397	Qvratio = 0,03776	s1 = 0,207 [Btu/lb$_m$-R]
s3 = 0,222 [Btu/lb$_m$-R]	T1 = 196,000	T2 = 98,6 [F]
T4 = 109,929 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,071 [ft^3/lb$_m$]	V10 = 1,043 [ft^3/lb$_m$]	V6 = 0,348 [ft^3/lb$_m$]
Wp = 0,884	y = 0,565 [R^{134a}]	

Calculation time = ,0 sec

APPENDIX A2- Comparisons between refrigerants

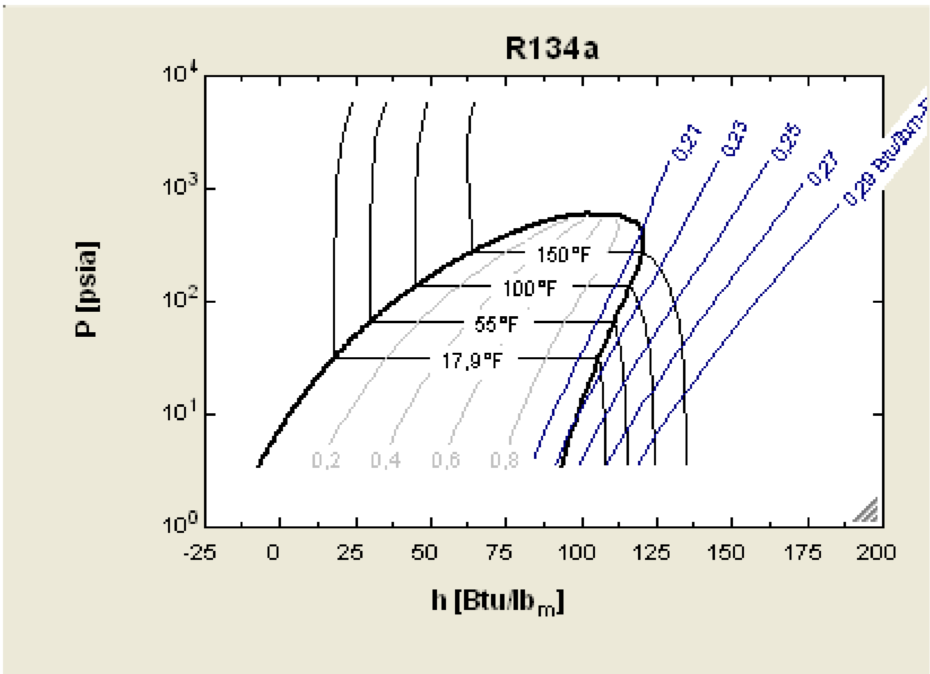
The choice of the refrigerant to be used is of paramount importance for the coefficient of performance of the refrigerating machine is strongly dependant on the properties of the refrigerant fluid.

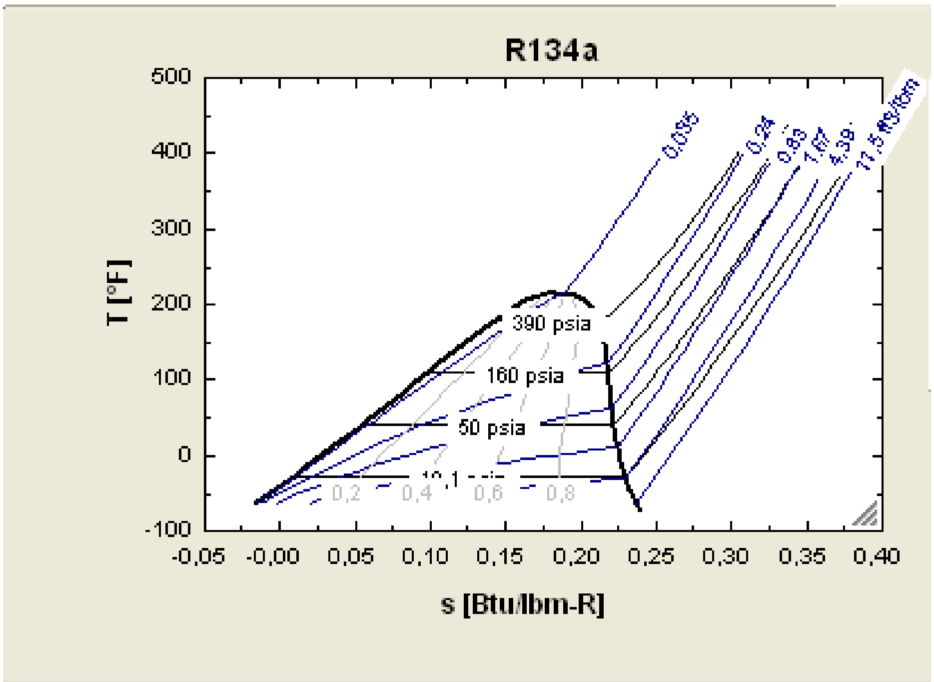
Next, various of the refrigerants mentioned in chapter 3. 4 are considered. The EES software has been used in order to fulfill the analysis. The temperatures in the evaporator and in the condenser are regarded as constant (35.2 F and 98.6 F, respectively) in all of them. The parameter whose value has been considered as variable is the temperature in the boiler (T1).

Notice: in the next analysis, there are three different COP values, whose meanings are explained next:

- COP: this parameter does not include the heat contribution due to the pre-boiler.
- COPideal: this is the coefficient of performance of a reversible Carnot cycle. It is the upper value the coefficient of performance can reach given the rest of the conditions.
- COPp: this is the coefficient of performance of the device, considering the heat contribution due to the pre-boiler. This is the proper parameter to be considered.

R134a.





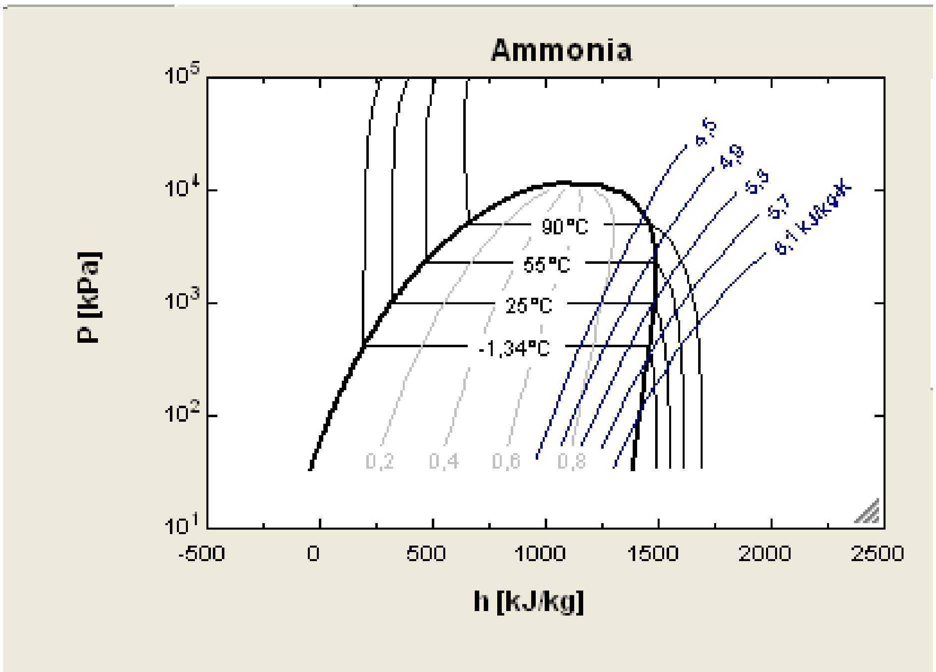
Unit Settings: [F]/[psia]/[lbm]/[degrees]

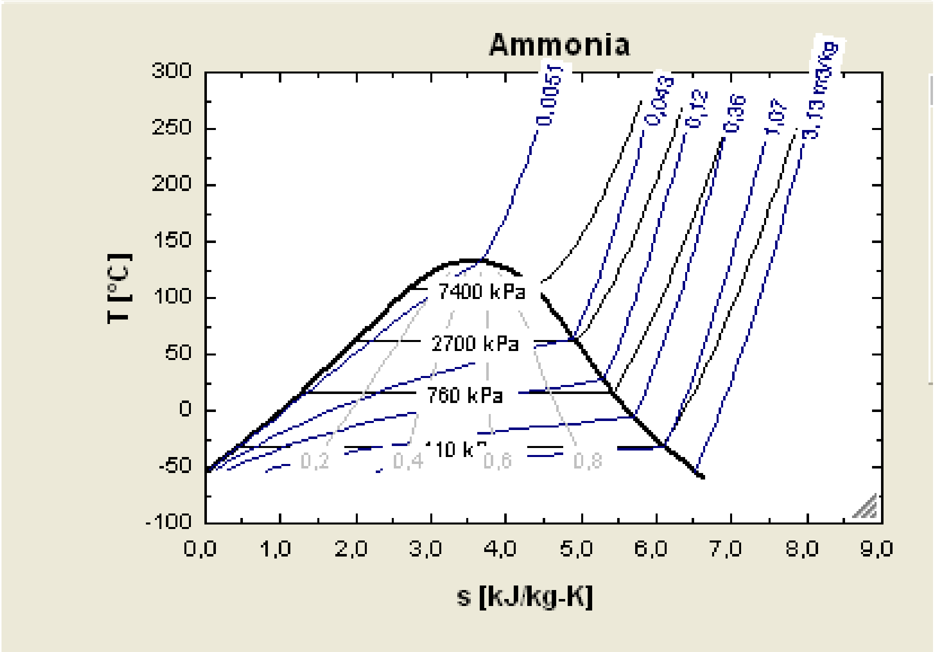
COP = 0,667	COPideal = 1,160	COPp = 0,659
Eff2 = 0,575	h1 = 118,795 [Btu/lb$_m$]	h2 = 110,397
h2s = 109,464 [Btu/lb$_m$]	h3 = 108,119 [Btu/lb$_m$]	h4 = 119,026
h4s = 117,935 [Btu/lb$_m$]	h6 = 44,629 [Btu/lb$_m$]	h7 = 45,513
p1 = 481,738 [psia]	p6 = 136,009 [psia]	p9 = 45,303 [psia]
Presseff = 0,900	Qcold = 27,618	Qhot = 41,403
Qmid = 74,397	Qvratio = 0,03776	s1 = 0,207 [Btu/lb$_m$-R]
s3 = 0,222 [Btu/lb$_m$-R]	T1 = 196,000	T2 = 98,6 [F]
T4 = 109,929 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,071 [ft^3/lb$_m$]	V10 = 1,043 [ft^3/lb$_m$]	V6 = 0,348 [ft^3/lb$_m$]
Wp = 0,884	y = 0,565 [R^{134a}]	

Calculation time = ,0 sec

Ammonia.

Ammonia is a very efficient refrigerant fluid, whose diagrams are shown next.





Its foremost features are summarized in the next table:

Molecular Weight

- Molecular weight : 17.03 g/mol

Solid phase

- Melting point : -78 °C
- Latent heat of fusion (1,013 bar, at triple point) : 331.37 kJ/kg

Liquid phase

- Liquid density (1.013 bar at boiling point) : 682

kg/m^3

- Liquid/gas equivalent (1.013 bar and 15 °C (59 °F)) : 947 vol/vol
- Boiling point (1.013 bar) : -33.5 °C
- Latent heat of vaporization (1.013 bar at boiling point) : 1371.2 kJ/kg
- Vapor pressure (at 21 °C or 70 °F) : 8.88 bar

Critical point

- Critical temperature : 132.4 °C
- Critical pressure : 112.8 bar

Gaseous phase

- Gas density (1.013 bar at boiling point) : 0.86 kg/m^3
- Gas density (1.013 bar and 15 °C (59 °F)) : 0.73 kg/m^3
- Compressibility Factor (Z) (1.013 bar and 15 °C (59 °F)) : 0.9929
- Specific gravity (air = 1) (1.013 bar and 21 °C (70 °F)) : 0.597
- Specific volume (1.013 bar and 21 °C (70 °F)) : 1.411 m^3/kg
- Heat capacity at constant pressure (Cp) (1.013 bar and 15 °C (59 °F)) : 0.037 kJ/(mol.K)
- Heat capacity at constant volume (Cv) (1.013 bar and 15 °C (59 °F)) : 0.028 kJ/(mol.K)
- Ratio of specific heats (Gamma:Cp/Cv) (1.013 bar and 15 °C (59 °F)) : 1.309623
- Viscosity (1.013 bar and 0 °C (32 °F)) : 0.000098 Poise
- Thermal conductivity (1.013 bar and 0 °C (32 °F)) : 22.19 mW/(m.K)

Miscellaneous

- Solubility in water (1.013 bar and 0 °C (32 °F)) : 862 vol/vol
- Autoignition temperature : 630 °C

The critical temperature of ammonia is quite high (132.4 ºC = 279.32 F). Since the coefficient of performance of the cycle is higher as the maximum pressure is higher, using ammonia could imply a higher coefficient of performance than using other refrigerants.

The results that using ammonia instead of R134a, at the same temperature levels at which R134a is used (T1 = 196 F), implies are shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,751	COPideal = 1,160	COPp = 0,746
Eff2 = 0,647	h1 = 626,449 [Btu/lb$_m$]	h2 = 569,353
h2s = 563,009 [Btu/lb$_m$]	h3 = 629,455 [Btu/lb$_m$]	h4 = 706,459
h4s = 698,759 [Btu/lb$_m$]	h6 = 161,595 [Btu/lb$_m$]	h7 = 164,394
p1 = 759,214 [psia]	p6 = 207,406 [psia]	p9 = 66,554 [psia]
Presseff = 0,900	Qcold = 199,201	Qhot = 265,326
Qmid = 544,864	Qvratio = 0,02184	s1 = 1,101 [Btu/lb$_m$-R]
s3 = 1,337 [Btu/lb$_m$-R]	T1 = 196,000	T2 = 98,6 [F]
T4 = 193,915 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,358 [ft^3/lb$_m$]	V10 = 4,351 [ft^3/lb$_m$]	V6 = 1,448 [ft^3/lb$_m$]
Wp = 2,800	y = 0,574 [R^{134a}]	

Calculation time = ,0 sec

T1= 196 F

The higher the pressure in the boiler, the higher the coefficient of performance is. By means of accomplishing a higher temperature in the boiler, a higher coefficient of performance can be, therefore, achieved. In case of working with 250 F in the boiler, the results would be those shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

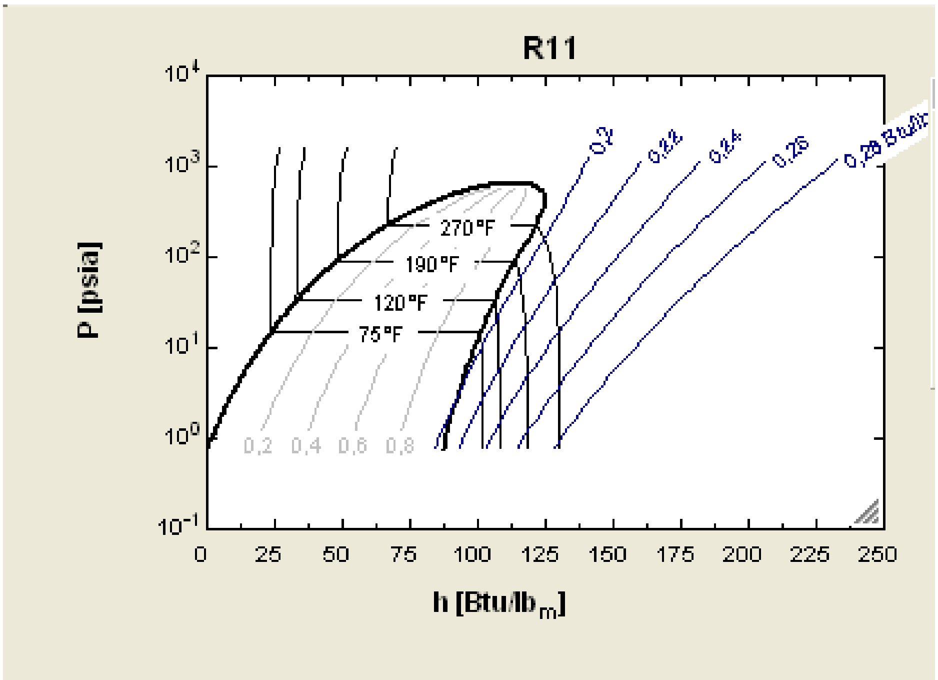
COP = 1,009	COPideal = 1,666	COPp = 0,995
Eff2 = 0,606	h1 = 576,885 [Btu/lb$_m$]	h2 = 508,907
h2s = 501,354 [Btu/lb$_m$]	h3 = 629,455 [Btu/lb$_m$]	h4 = 706,459
h4s = 698,759 [Btu/lb$_m$]	h6 = 161,595 [Btu/lb$_m$]	h7 = 167,387
p1 = 1348,975 [psia]	p6 = 207,406 [psia]	p9 = 66,554 [psia]
Presseff = 0,900	Qcold = 219,365	Qhot = 217,497
Qmid = 544,864	Qvratio = 0,01983	s1 = 0,990 [Btu/lb$_m$-R]
s3 = 1,337 [Btu/lb$_m$-R]	T1 = 250,000	T2 = 98,6 [F]
T4 = 193,915 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,154 [ft^3/lb$_m$]	V10 = 4,351 [ft^3/lb$_m$]	V6 = 1,448 [ft^3/lb$_m$]
Wp = 5,792	y = 0,531 [R^{134a}]	

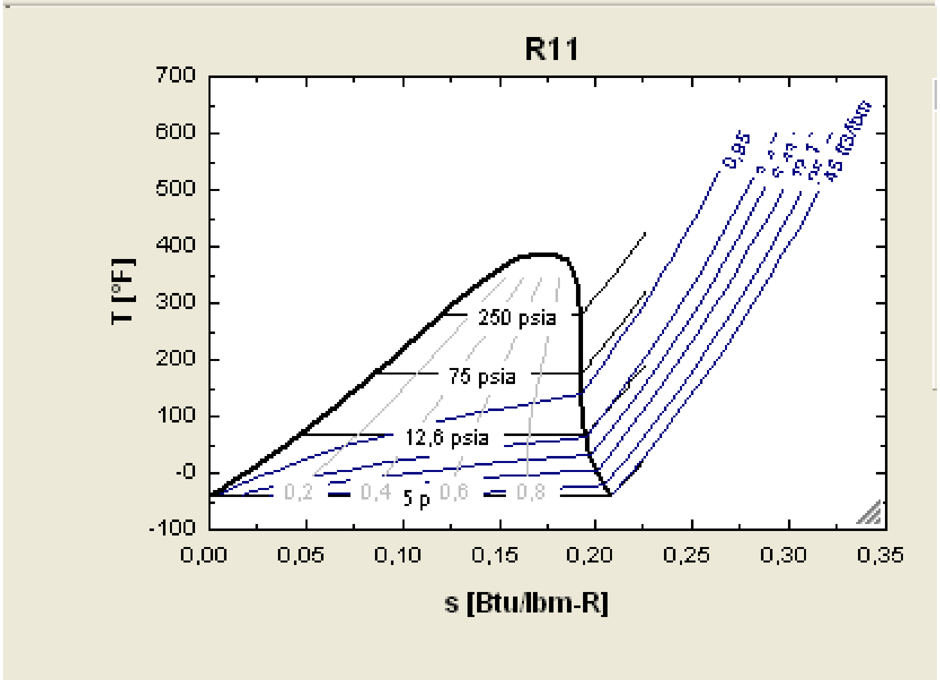
Calculation time = ,0 sec

T! = 250 F

Although using ammonia as a refrigerant implies high coefficient of performance values it has many important drawbacks that make it no appropiate. Ammonia is highly explosive and toxic and, therefore it requires very extrict caution measures. In addition, it is not compatible with many materials.

R11.





This refrigerant fluid has a very high critic temperature and its critic pressure is not excesively high; thus very high temperatures can be achieved in the boiler in order to enhance the coefficient of performance of the cycle.

The results yielded by refrigerant R11 for a cycle whose highest temperature is 196 F are the next ones:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,762	COPideal = 1,160	COPp = 0,761
Eff2 = 0,657	h1 = 114,577 [Btu/lb$_m$]	h2 = 104,232
h2s = 103,083 [Btu/lb$_m$]	h3 = 96,216 [Btu/lb$_m$]	h4 = 106,911
h4s = 105,841 [Btu/lb$_m$]	h6 = 28,407 [Btu/lb$_m$]	h7 = 28,560
p1 = 97,542 [psia]	p6 = 22,881 [psia]	p9 = 6,284 [psia]
Presseff = 0,900	Qcold = 33,339	Qhot = 43,725
Qmid = 78,504	Qvratio = 0,1806	s1 = 0,192 [Btu/lb$_m$-R]
s3 = 0,197 [Btu/lb$_m$-R]	T1 = 196,000	T2 = 101,7 [F]
T4 = 120,136 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,453 [ft^3/lb$_m$]	V10 = 6,020 [ft^3/lb$_m$]	V6 = 1,806 [ft^3/lb$_m$]
Wp = 0,153	y = 0,508 [R^{134a}]	

Calculation time = ,0 sec

T1 = 196 F

This results are similar to those corresponding to R134a.

If the the temperature T1 was increased up to 360 F, the results would be those shown next.

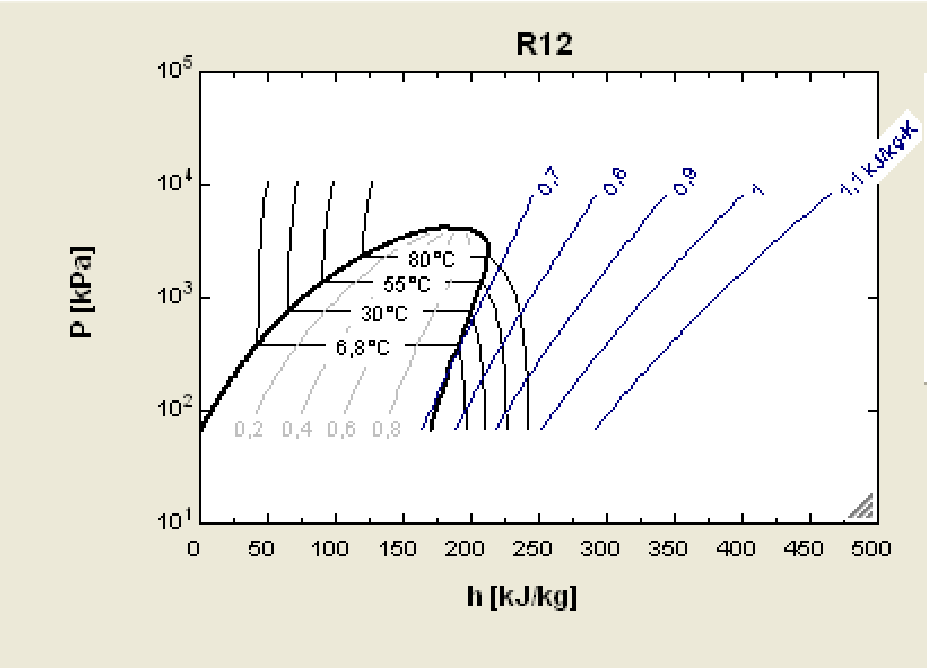
Unit Settings: [F]/[psia]/[lbm]/[degrees]

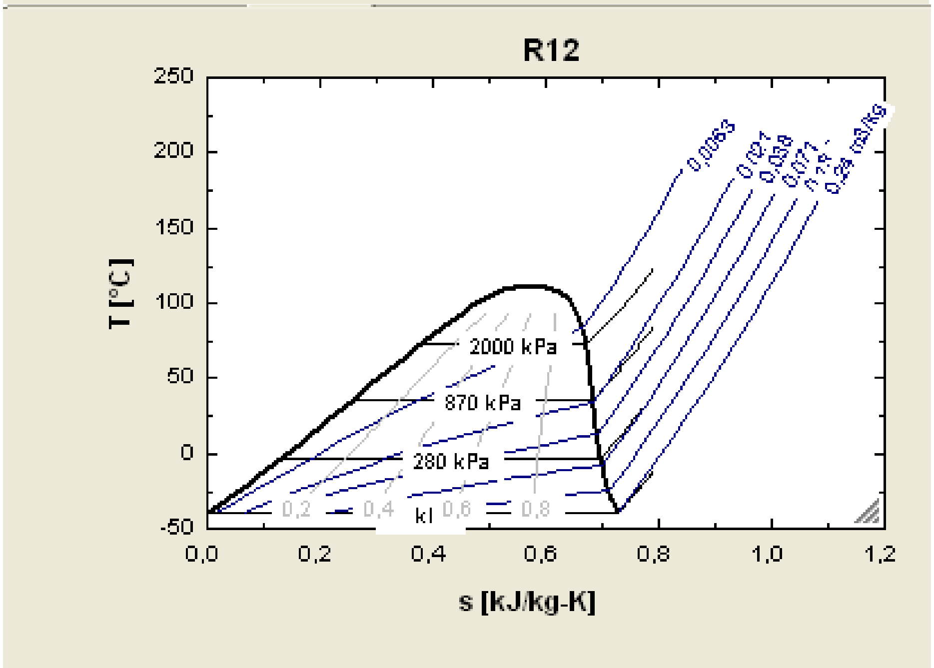
COP = 1,407	COPideal = 2,490	COPp = 1,393
Eff2 = 0,565	h1 = 124,249 [Btu/lb$_m$]	h2 = 103,196
h2s = 100,857 [Btu/lb$_m$]	h3 = 96,216 [Btu/lb$_m$]	h4 = 106,911
h4s = 105,841 [Btu/lb$_m$]	h6 = 28,407 [Btu/lb$_m$]	h7 = 29,402
p1 = 508,845 [psia]	p6 = 22,881 [psia]	p9 = 6,284 [psia]
Presseff = 0,900	Qcold = 44,966	Qhot = 31,952
Qmid = 78,504	Qvratio = 0,1339	s1 = 0,188 [Btu/lb$_m$-R]
s3 = 0,197 [Btu/lb$_m$-R]	T1 = 360,000	T2 = 98,6 [F]
T4 = 120,136 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,067 [ft^3/lb$_m$]	V10 = 6,020 [ft^3/lb$_m$]	V6 = 1,806 [ft^3/lb$_m$]
Wp = 0,996	y = 0,337 [R^{134a}]	

Calculation time = ,0 sec

T1= 360 F

R12.





Molecular Weight

- Molecular weight : 120.93 g/mol

Liquid phase

- Liquid density (1.013 bar at boiling point) : 1486 kg/m^3
- Liquid/gas equivalent (1.013 bar and 15 °C (59 °F)) : 292 vol/vol
- Boiling point (1.013 bar) : -29.8 °C
- Latent heat of vaporization (1.013 bar at boiling point) : 166.95 kJ/kg

Critical point

- Critical temperature : 112 °C
- Critical pressure : 41.15 bar

Gaseous phase

- Gas density (1.013 bar at boiling point) : 6.25 kg/m^3
- Gas density (1.013 bar and 15 °C (59 °F)) : 5.11 kg/m^3
- Compressibility Factor (Z) (1.013 bar and 15 °C (59 °F)) : 0.995
- Specific gravity (air = 1) (1.013 bar and 21 °C (70 °F)) : 4.2
- Specific volume (1.013 bar and 21 °C (70 °F)) : 0.195 m^3/kg
- Heat capacity at constant pressure (Cp) (1.013 bar and 30 °C (86 °F)) : 0.074 kJ/(mol.K)
- Heat capacity at constant volume (Cv) (1.013 bar and 30 °C (86 °F)) : 0.065 kJ/(mol.K)
- Ratio of specific heats (Gamma:Cp/Cv) (1.013 bar and 30 °C (86 °F)) : 1.138889
- Viscosity (1.013 bar and 0 °C (32 °F)) : 0.0001168 Poise
- Thermal conductivity (1.013 bar and 0 °C (32 °F)) : 9.46 mW/(m.K)

This refrigerant too has a higher critic temperature than R134a, but the differences between R12 and R134a are not so big as those between ammonia and R134a.

Using R12 instead of R134a, at the foreseen highest temperature level in the actual cicle (196 F), implies the next results:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,704	COPideal = 1,160	COPp = 0,696
Eff2 = 0,607	h1 = 91,434 [Btu/lb_m]	h2 = 84,310
h2s = 83,518 [Btu/lb_m]	h3 = 80,947 [Btu/lb_m]	h4 = 89,413
h4s = 88,566 [Btu/lb_m]	h6 = 30,756 [Btu/lb_m]	h7 = 31,420
p1 = 412,731 [psia]	p6 = 129,268 [psia]	p9 = 47,408 [psia]
Presseff = 0,900	Qcold = 22,936	Qhot = 32,589
Qmid = 58,657	Qvratio = 0,03662	s1 = 0,157 [Btu/lb_m-R]
s3 = 0,166 [Btu/lb_m-R]	T1 = 196,000	T2 = 98,6 [F]
T4 = 112,281 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,082 [ft^3/lb_m]	V10 = 0,840 [ft^3/lb_m]	V6 = 0,314 [ft^3/lb_m]
Wp = 0,664	y = 0,543 [R^{134a}]	

Calculation time = ,0 sec

T1= 196 F

In case of working with a higher temperature level in the boiler (210 F), the results would be:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

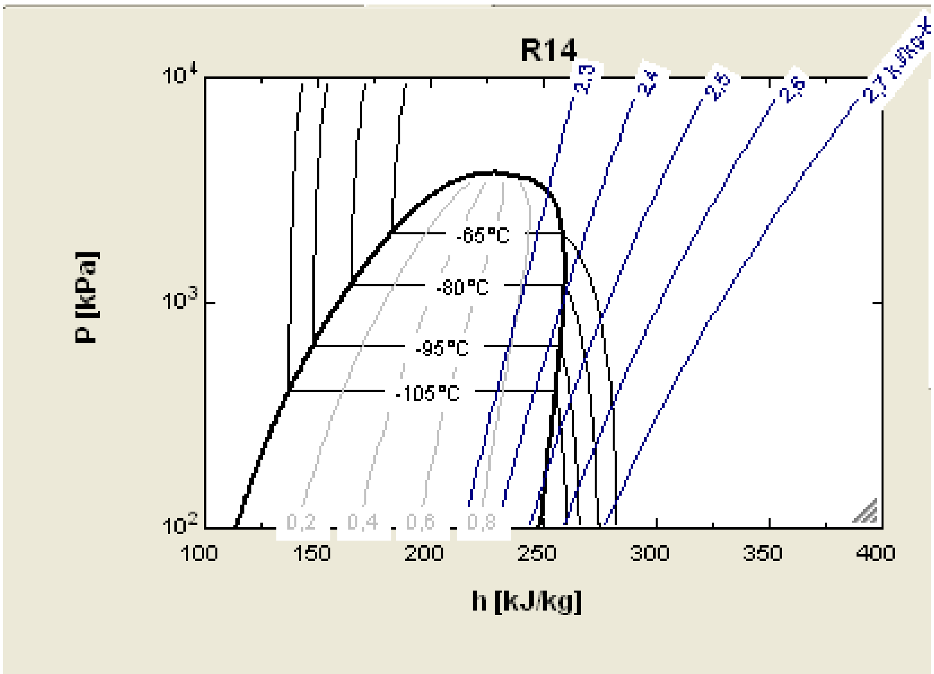
COP = 0,768	COPideal = 1,299	COPp = 0,758
Eff2 = 0,591	h1 = 90,561 [Btu/lb_m]	h2 = 82,920
h2s = 82,071 [Btu/lb_m]	h3 = 80,947 [Btu/lb_m]	h4 = 89,413
h4s = 88,566 [Btu/lb_m]	h6 = 30,756 [Btu/lb_m]	h7 = 31,567
p1 = 475,376 [psia]	p6 = 129,268 [psia]	p9 = 47,408 [psia]
Presseff = 0,900	Qcold = 23,811	Qhot = 31,007
Qmid = 58,657	Qvratio = 0,03527	s1 = 0,155 [Btu/lb_m-R]
s3 = 0,166 [Btu/lb_m-R]	T1 = 210,000	T2 = 98,6 [F]
T4 = 112,281 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,065 [ft^3/lb_m]	V10 = 0,840 [ft^3/lb_m]	V6 = 0,314 [ft^3/lb_m]
Wp = 0,811	y = 0,526 [R^{134a}]	

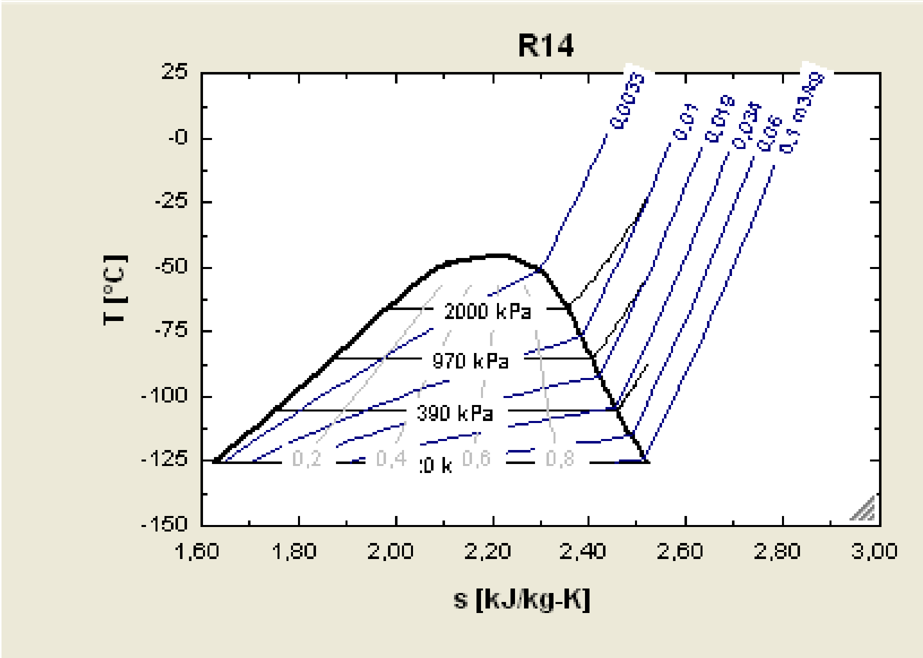
Calculation time = ,0 sec

T1= 210 F

The results that could be accomplished using R12 are better than those corresponding to R134a. Though, due to its ozone-depletion effect, R12 is being substituted by different refrigerants (specially, R134a).

R 14.





The features of this refrigerant are shown next.

Molecular Weight

- Molecular weight : 86.48 g/mol

Liquid phase

- Liquid density (1.013 bar at boiling point) : 1413 kg/m^3
- Liquid/gas equivalent (1.013 bar and 15 °C (59 °F)) : 385 vol/vol
- Boiling point (1.013 bar) : -40.8 °C
- Latent heat of vaporization (1.013 bar at boiling

point) : 233.95 kJ/kg

Critical point

- Critical temperature : 96 °C
- Critical pressure : 49.36 bar

Gaseous phase

- Gas density (1.013 bar at boiling point) : 4.706 kg/m^3
- Gas density (1.013 bar and 15 °C (59 °F)) : 3.66 kg/m^3
- Compressibility Factor (Z) (1.013 bar and 15 °C (59 °F)) : 0.9831
- Specific gravity (air = 1) (1.013 bar and 21 °C (70 °F)) : 3.08
- Specific volume (1.013 bar and 21 °C (70 °F)) : 0.275 m^3/kg
- Heat capacity at constant pressure (Cp) (1.013 bar and 30 °C (86 °F)) : 0.057 kJ/(mol.K)
- Heat capacity at constant volume (Cv) (1.013 bar and 30 °C (86 °F)) : 0.048 kJ/(mol.K)
- Ratio of specific heats (Gamma:Cp/Cv) (1.013 bar and 30 °C (86 °F)) : 1.178253
- Viscosity (1.013 bar and 0 °C (32 °F)) : 0.0001256 Poise

Miscellaneous

- Solubility in water (1 bar and 25 °C (77 °F)) : 0.7799 vol/vol

This refrigerant has a relatively low critic temperature (204.8 F = 96 ºC). Because of that, the highest temperature in the cycle can not be, in this case, so high as in the previous cases.

In case of working at 185 F as the temperature in the boiler, the results yielded by the cycle would be those shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

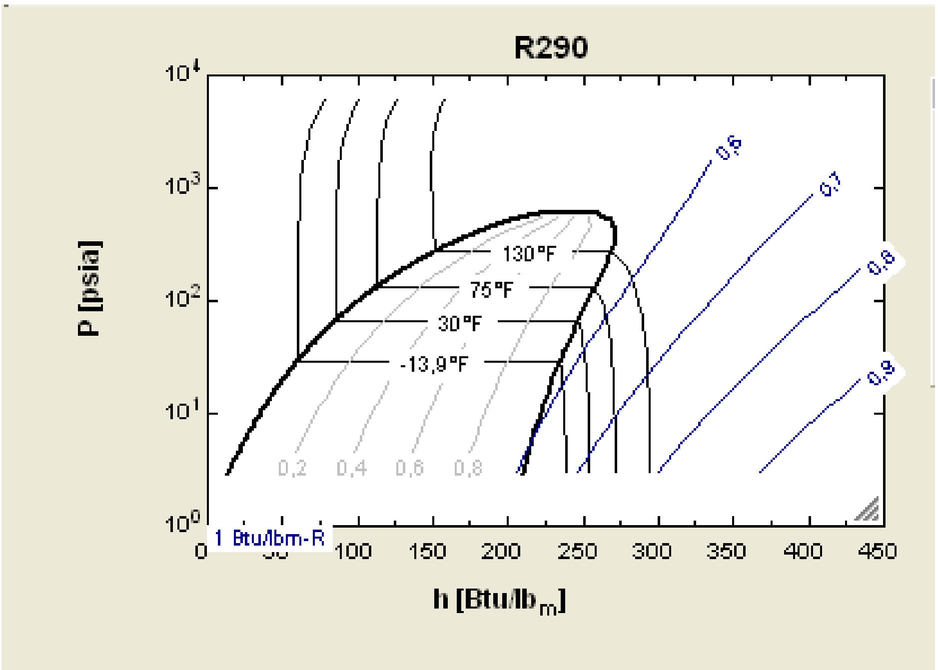
COP = 0,650	COPideal = 1,046	COPp = 0,641
Eff2 = 0,622	h1 = 175,492 [Btu/lb$_m$]	h2 = 167,840
h2s = 166,990 [Btu/lb$_m$]	h3 = 174,393 [Btu/lb$_m$]	h4 = 186,136
h4s = 184,961 [Btu/lb$_m$]	h6 = 105,690 [Btu/lb$_m$]	h7 = 106,673
p1 = 585,516 [psia]	p6 = 206,659 [psia]	p9 = 76,520 [psia]
Presseff = 0,900	Qcold = 27,105	Qhot = 41,668
Qmid = 80,446	Qvratio = 0,02632	s1 = 0,386 [Btu/lb$_m$-R]
s3 = 0,417 [Btu/lb$_m$-R]	T1 = 185,000	T2 = 98,6 [F]
T4 = 132,587 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,069 [ft^3/lb$_m$]	V10 = 0,713 [ft^3/lb$_m$]	V6 = 0,262 [ft^3/lb$_m$]
Wp = 0,983	y = 0,605 [R^{134a}]	

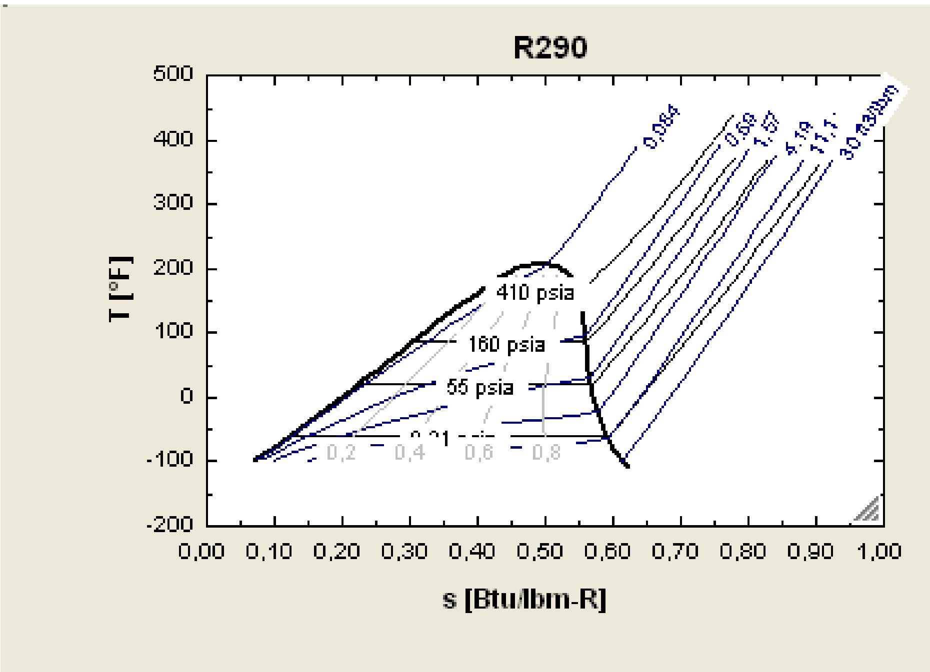
Calculation time = ,0 sec

T= 185 F

The results yielded by R134a in its optimal conditions are better than those yielded by the use of R22. In addition, due to its environmental effects, the use of refrigerant R22 is limited, and it will no longer be manufactured since 2010. Therefore, to use R22 is not a good option.

R290.





This refrigerant has a critic pressure which is slightly bellow that of R134a. Therefore, no excesively high temperatures can be achieved with R290. In case of establishing 190 F as the highest temperature in the cycle, the results yielded by it would be the next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,647	COPideal = 1,098	COPp = 0,637
Eff2 = 0,589	h1 = 267,554 [Btu/lb$_m$]	h2 = 252,074
h2s = 250,354 [Btu/lb$_m$]	h3 = 247,903 [Btu/lb$_m$]	h4 = 268,768
h4s = 266,682 [Btu/lb$_m$]	h6 = 128,609 [Btu/lb$_m$]	h7 = 130,739
p1 = 524,520 [psia]	p6 = 185,216 [psia]	p9 = 72,642 [psia]
Presseff = 0,900	Qcold = 50,809	Qhot = 78,543
Qmid = 140,159	Qvratio = 0,02888	s1 = 0,537 [Btu/lb$_m$-R]
s3 = 0,566 [Btu/lb$_m$-R]	T1 = 190,000	T2 = 98,6 [F]
T4 = 109,412 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,147 [ft^3/lb$_m$]	V10 = 1,467 [ft^3/lb$_m$]	V6 = 0,572 [ft^3/lb$_m$]
Wp = 2,129	y = 0,574 [R^{134a}]	

Calculation time = ,0 sec

T1= 190 F

This results are slightly worse than those corresponding to refrigerant R134a.

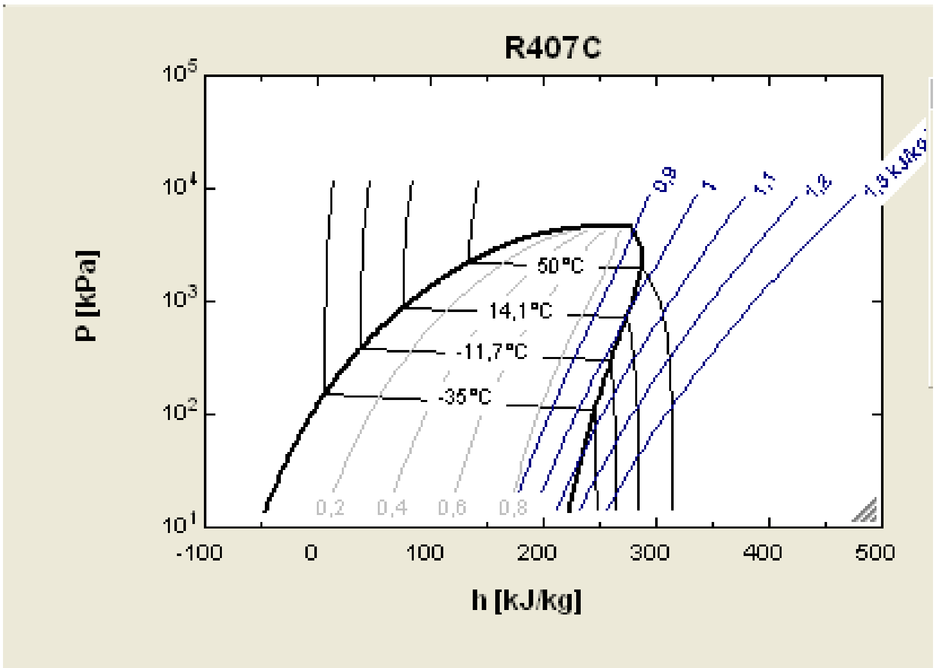
R407C.

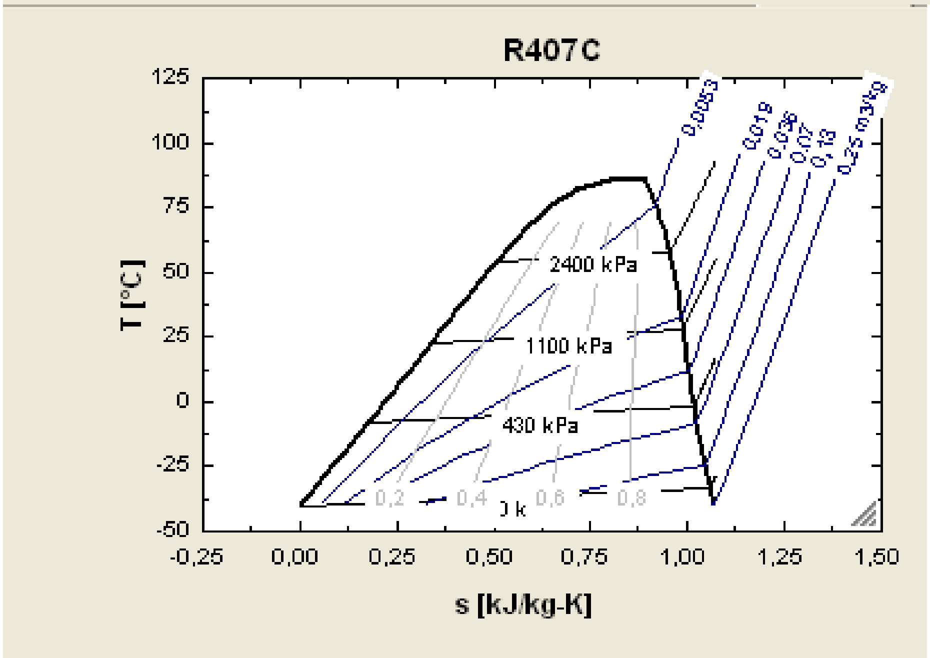
With the phase-out of R-22 in the EU, R-407C has emerged as the preferred working fluid for theb majority of comfort cooling applications. R-407C, while similar to R-22 in many of its physical properties, is a zeotropic mixture of

HFCs which does impose certain application restrictions and specific handling and equipment design requirements.

R407C has properties very similar to those of R-22 (which it has replaced in air conditioning applications) in terms of both its operating pressures and its performance in dry expansion air conditioning systems. R407C is very similar, but not identical to R22 in its performance as an air-conditioning refrigerant. For example, system condensing pressures will be somewhat higher (around 1 bar higher at 45°C condensing temperature) than for R22.

R407C is a mixed, zeotropic, refrigerant consisting of 3 HFC components: R32, R125 and R134a in the proportions 23%/25%/52% by weight (a ± 2% tolerance is allowed for each of the components).





As it can be noted, the critic temperature of R407C is around 90 ºC (194 F). Therefore, the highest temperatures that could be achieved in the cicle if the refrigerant was R407C are slightly lower than those achieved using R134a.

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,471	COPideal = 0,993	COPp = 0,464
Eff2 = 0,474	h1 = 120,466 [Btu/lb$_m$]	h2 = 113,295
h2s = 112,499 [Btu/lb$_m$]	h3 = 114,985 [Btu/lb$_m$]	h4 = 129,275
h4s = 127,846 [Btu/lb$_m$]	h6 = 47,882 [Btu/lb$_m$]	h7 = 48,941
p1 = 621,575 [psia]	p6 = 234,128 [psia]	p9 = 87,047 [psia]
Presseff = 0,900	Qcold = 22,421	Qhot = 47,626
Qmid = 81,393	Qvratio = 0,03545	s1 = 0,215 [Btu/lb$_m$-R]
s3 = 0,242 [Btu/lb$_m$-R]	T1 = 180,000	T2 = 106,9 [F]
T4 = 134,649 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,070 [ft^3/lb$_m$]	V10 = 0,795 [ft^3/lb$_m$]	V6 = 0,272 [ft^3/lb$_m$]
Wp = 1,059	y = 0,666 [R^{134a}]	

Calculation time = ,0 sec

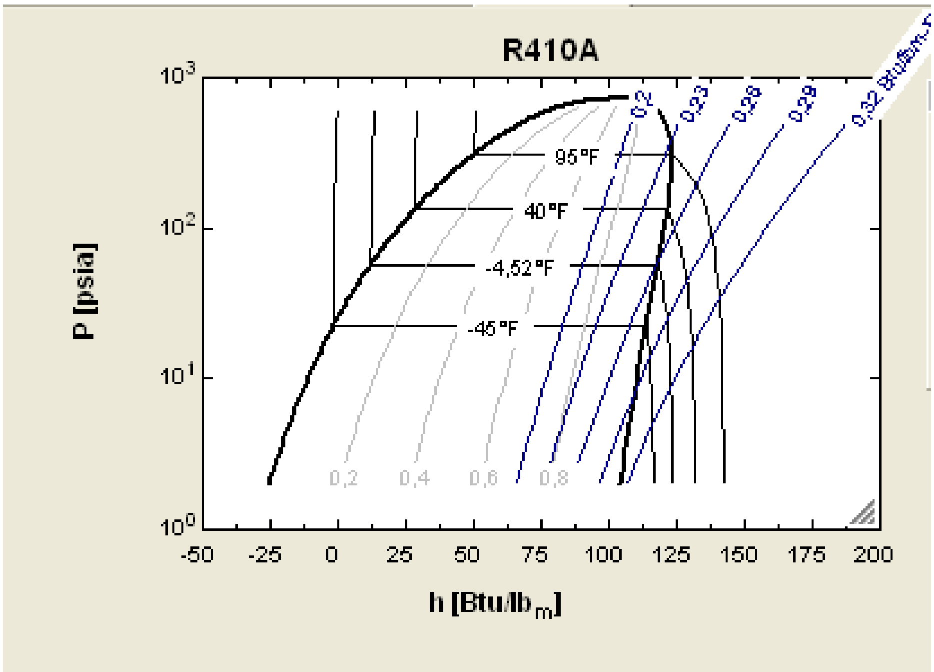
T1= 180 F

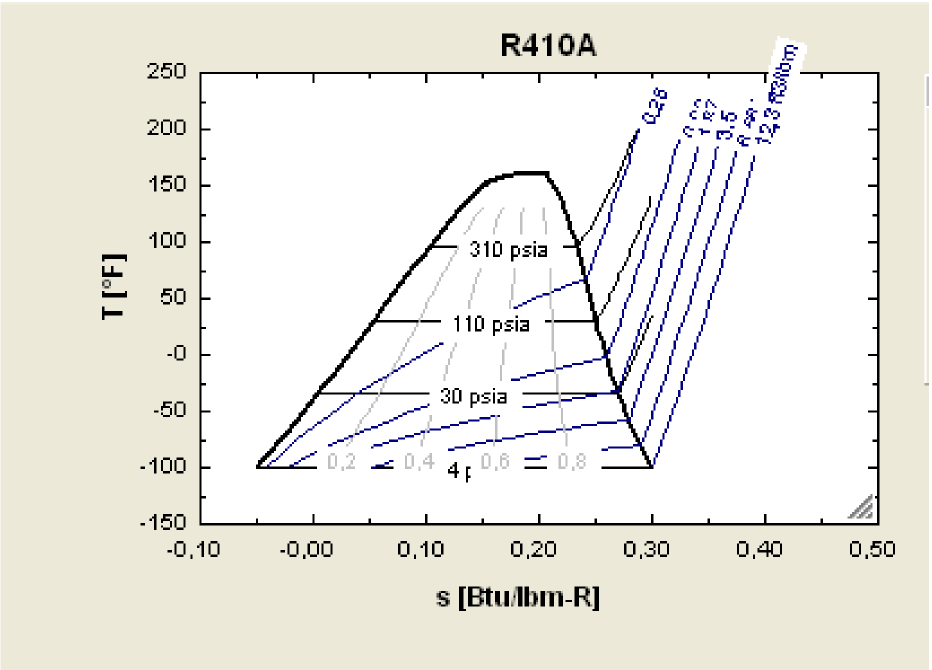
The results yielded by this refrigerant are worse than those obtained with R134a.

R410a.

R410a operates at a significantly higher pressure than R22 and for this reason special systems have to be designed to utilise it. It has the potential to outperform R22 because the effect of pressure drop is reduced and because it has good heat transfer properties. It is limited at high condensing conditions, not because of the high pressures, but due to the effect of its relatively low critical temperature, below 75°C.

The vapour compression cycle has a very low efficiency when condensation is close to the critical temperature because the amount of latent heat, which produces the refrigeration effect, becomes small.





If the used refrigerant was R410a and the highest temperature achieved in the cycle was 135 F, the results of the cycle would be those shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

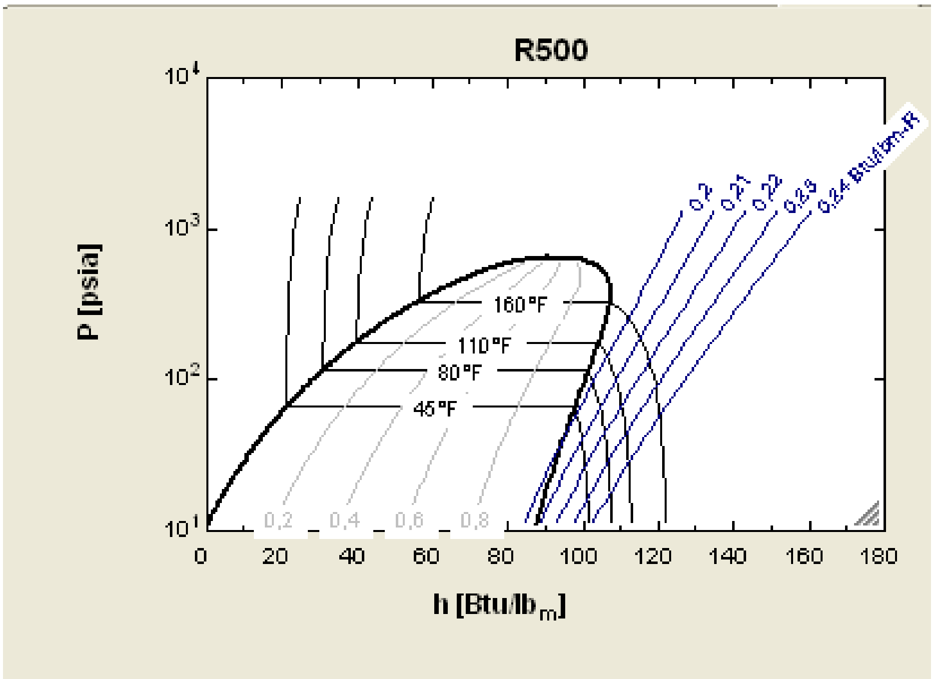
COP = 0,312	COPideal = 0,478	COPp = 0,310
Eff2 = 0,654	h1 = 120,317 [Btu/lb$_m$]	h2 = 116,451
h2s = 116,021 [Btu/lb$_m$]	h3 = 121,196 [Btu/lb$_m$]	h4 = 133,838
h4s = 132,573 [Btu/lb$_m$]	h6 = 51,637 [Btu/lb$_m$]	h7 = 52,223
p1 = 522,720 [psia]	p6 = 325,732 [psia]	p9 = 122,536 [psia]
Presseff = 0,900	Qcold = 16,293	Qhot = 52,145
Qmid = 82,201	Qvratio = 0,03031	s1 = 0,220 [Btu/lb$_m$-R]
s3 = 0,250 [Btu/lb$_m$-R]	T1 = 135,000	T2 = 98,79 [F]
T4 = 128,853 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,091 [ft^3/lb$_m$]	V10 = 0,494 [ft^3/lb$_m$]	V6 = 0,171 [ft^3/lb$_m$]
Wp = 0,586	y = 0,766 [R^{134a}]	

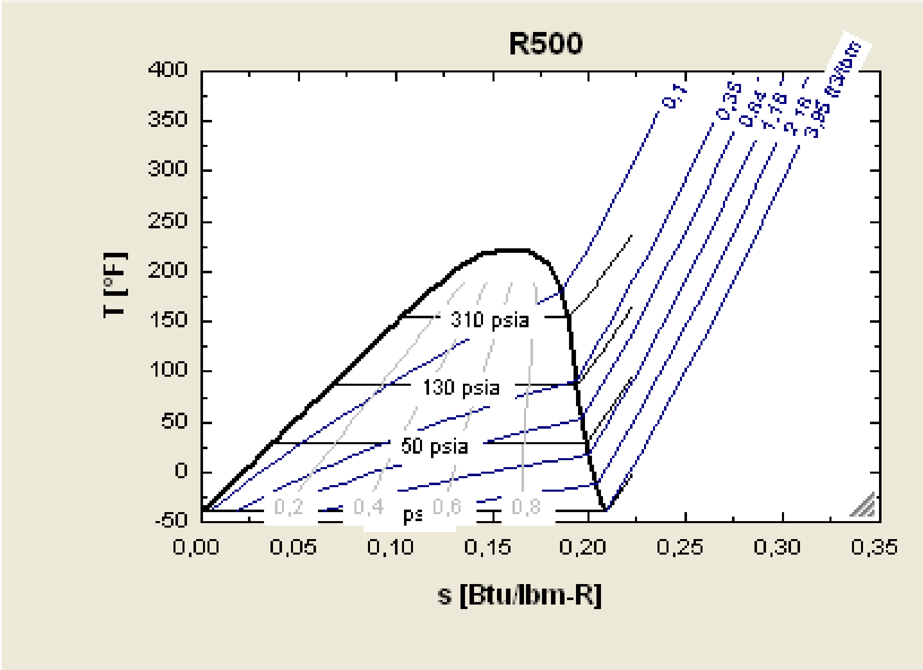
Calculation time = ,0 sec

T1 = 135 F

As it can be noted, the low critic temperature of this refrigerant makes the coefficient of performance of the device rather low.

R500.





This refrigerant has a higher critic temperature than refrigerant R134a; therefore, a higher temperature can be achieved in the boiler. In case of working at 210 F as the highest temperature in the cycle, the results obtained would be the next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,760	COPideal = 1,299	COPp = 0,748
Eff2 = 0,585	h1 = 103,971 [Btu/lbm]	h2 = 95,363
h2s = 94,407 [Btu/lbm]	h3 = 96,605 [Btu/lbm]	h4 = 106,867
h4s = 105,841 [Btu/lbm]	h6 = 36,217 [Btu/lbm]	h7 = 37,323
p1 = 568,166 [psia]	p6 = 152,774 [psia]	p9 = 55,724 [psia]
Presseff = 0,900	Qcold = 27,547	Qhot = 36,246
Qmid = 70,650	Qvratio = 0,03123	s1 = 0,178 [Btu/lbm-R]
s3 = 0,198 [Btu/lbm-R]	T1 = 210,000	T2 = 98,6 [F]
T4 = 115,167 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,058 [ft3/lbm]	V10 = 0,860 [ft3/lbm]	V6 = 0,317 [ft3/lbm]
Wp = 1,106	y = 0,544 [R134a]	

Calculation time = ,0 sec

T1 = 210 F

Even though the highest temperature in this case is considerabily higher than that of the cycle run with R134a, the coefficient of performance of the former cycle is only slightly higher (some 10%) than that of the latter. In case that the highest temperature in the cycle run with R500 was the same as that of the cycle run with R134a (196 F), the results accomplished would be those shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

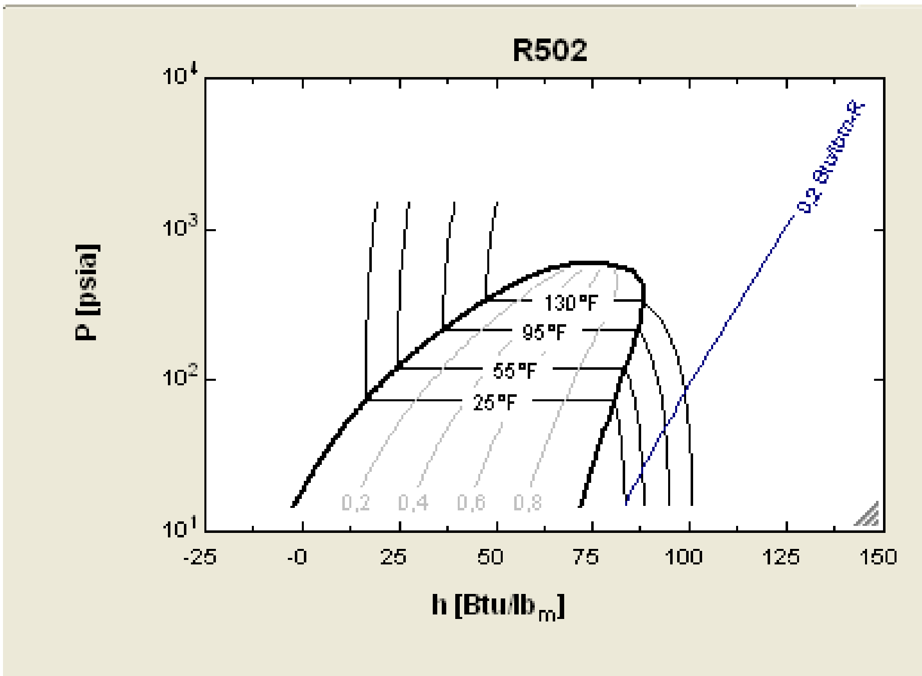
COP = 0,699	COPideal = 1,160	COPp = 0,690
Eff2 = 0,602	h1 = 106,348 [Btu/lb$_m$]	h2 = 98,129
h2s = 97,216 [Btu/lb$_m$]	h3 = 96,605 [Btu/lb$_m$]	h4 = 106,867
h4s = 105,841 [Btu/lb$_m$]	h6 = 36,217 [Btu/lb$_m$]	h7 = 37,119
p1 = 491,444 [psia]	p6 = 152,774 [psia]	p9 = 55,724 [psia]
Presseff = 0,900	Qcold = 26,856	Qhot = 38,442
Qmid = 70,650	Qvratio = 0,03204	s1 = 0,183 [Btu/lb$_m$-R]
s3 = 0,198 [Btu/lb$_m$-R]	T1 = 196,000	T2 = 98,6 [F]
T4 = 115,167 [F]	T6 = 98,600	T9 = 35,200
V1 = 0,077 [ft^3/lb$_m$]	V10 = 0,860 [ft^3/lb$_m$]	V6 = 0,317 [ft^3/lb$_m$]
Wp = 0,902	y = 0,555 [R^{134a}]	

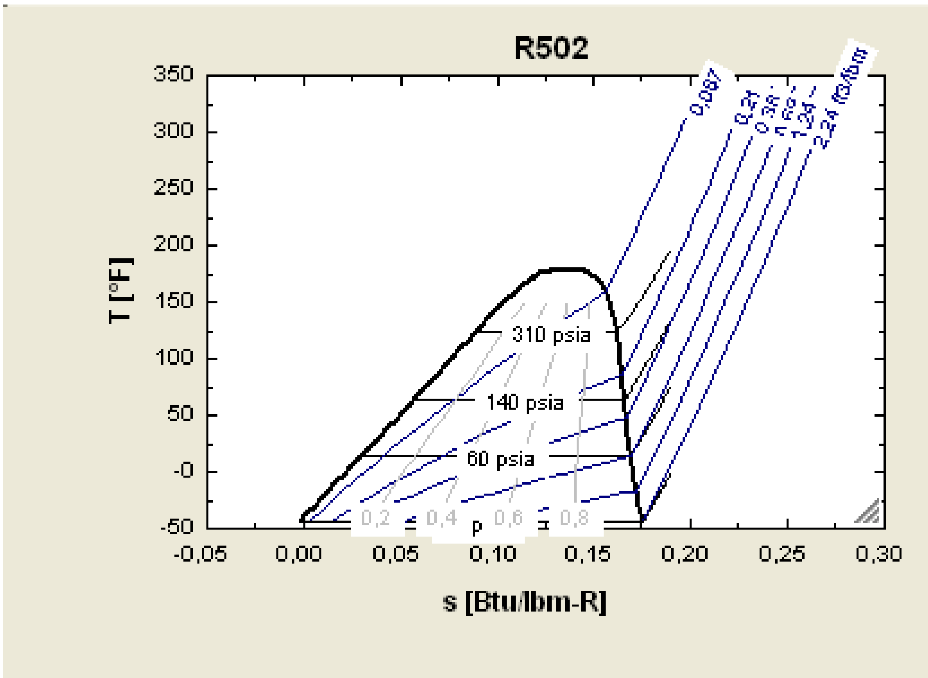
Calculation time = ,0 sec

T1= 196 F

The coefficient of performance obtained this way is only some 2.4% higher than that achieved using R134a.

R502.





This refrigerant has the drawback of its low critic temperature. Because of that, the highest temperature in the cycle can not be very high, and this implies a low coefficient of performance. In case of using R502 at 160 F as the highest temperature in the cycle, the achieved results would be those shown next:

Unit Settings: [F]/[psia]/[lbm]/[degrees]

COP = 0,468	COPideal = 0,774	COPp = 0,462	Eff2 = 0,605
h1 = 87,117 [Btu/lbm]	h2 = 82,913	h2s = 82,445 [Btu/lbm]	h3 = 81,510 [Btu/lbm]
h4 = 89,593	h4s = 88,785 [Btu/lbm]	h6 = 37,253 [Btu/lbm]	h7 = 37,885
p1 = 473,389 [psia]	p6 = 226,688 [psia]	p9 = 87,822 [psia]	Presseff = 0,900
Qcold = 15,145	Qhot = 32,385	Qmid = 52,340	Qvratio = 0,03089
s1 = 0,156 [Btu/lbm-R]	s3 = 0,167 [Btu/lbm-R]	T1 = 160,000	T2 = 98,6 [F]
T4 = 111,311 [F]	T6 = 98,600	T9 = 35,200	V1 = 0,066 [ft3/lbm]
V10 = 0,468 [ft3/lbm]	V6 = 0,174 [ft3/lbm]	Wp = 0,632	y = 0,658 [R134a]

Calculation time = ,0 sec

T!= 160 F

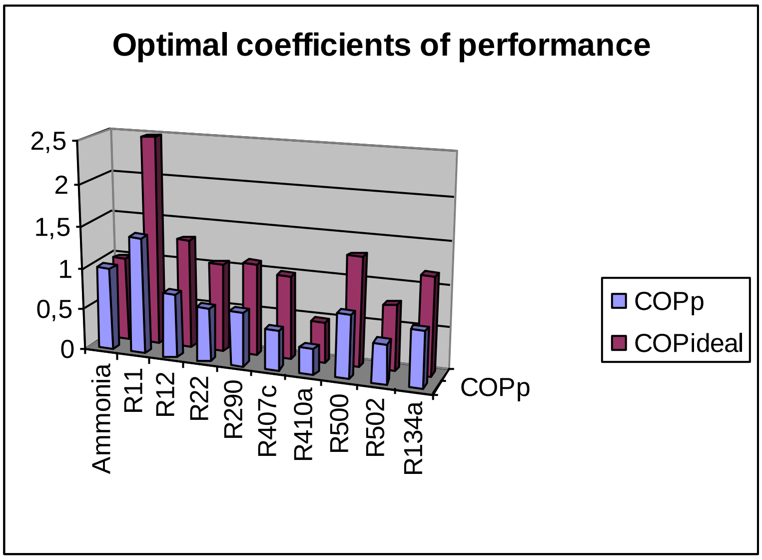
Some other refrigerants.

CO_2.

Carbon dioxide (CO2) is a potential substitute for HCFC refrigerants with favorable environmental properties compared to other HCFC alternatives. However, carbon dioxide has a very low critic temperature,

Conclusions.

So far, the coefficients of performance corresponding to the use of different refrigerants have been compared. For those refrigerants whose critic temperatures are quite higher than that of refrigerant R134a, two different conditions have been considered: the optimal ones (with T1 at a close to its critic temperature value) and those in which T1 is the same as that regarded as optimal for R134a (196 F). The next figure compares the optimal values obtained for each refrigerant.



Refrigerant R134a is the refrigerant to be used though there are some other that could provide higher coefficients of performance. The reason why this election has been made are explained next:

Ammonia could yield more energy efficient results than R134a. The coefficient of performance that could be achieved using ammonia as the refrigerant fluid is around 0.995 (quite close to the coefficient of performance that a Carnot cycle could yield in the considered conditions). The efficiency of a cycle run with ammonia is very high, and the coefficient of performance enhancement that using ammonia would imply is very important (the coefficient of performance corresponding to the use of ammonia is some 51.44% higher than that related to R134a). Thus, ammonia seems to yield gorgeous results for this thermodynamic cycle. However, ammonia is not a suitable refrigerant at all, due to the fact that it is extremely toxic and explosive. Therefore, its use could only be considered in industrial facilities equipped with very strict safety systems.

Regarding R11, it could yield redults even better than those yielded by ammonia. The coefficient of performance corresponding to the use of R11 is around 1.393 (some 40% higher than that corresponding to ammonia and some 111.4% higher than that corresponding to the use of refrigerant R134a). However, refrigerant R11 is banned due to its environmental effects.

As far as refrigerant R12 is concerned, its results are slightly better than those of refrigerant R134a. The coefficient of performance obtained using R12 is some

15% higher than that corresponding to R134a. Nevertheless, refrigerant R12 can not be used either, due to the fact that it has some adverse environmental effects. Its production in the United States ended in 1995.

As for refrigerant R500, the results that it provides are slightly better than those of refrigerant R134a. The coefficient of performance attained using refrigerant R500 is some 13.5% higher than that corresponding to refrigerant R134a. Nevertheless, this refrigerant is not appropiate either because its use is banned in many countries due to the environmental effects that it has.

R134a is regarded as one of the safest refrigerants. In evaluations done by the chemical industry R134a has been found to be safe and to pose no cancer or birth defect hazards. R134a has been chosen by auto manufactures to be the replacement for R12. In fact it is the only new refrigerant that is recommended by the OEM and after market manufacturing companies. R134a is not corrosive on standard steel, aluminum and copper samples. R134a is not flammable at ambient temperatures and atmospheric pressures. However R134a systems should not be pressure tested with air, because mixtures of air and R134a have been shown to be dangerous. In addition the results obtained using this refrigerant are quite good.

APPENDIX B- Competitor Analysis:

1) Danby Products Inc.

This company is a manufacture of household appliances such as ovens, dishwashers, freezers, microwaves, and of course refrigerators (regular and heat-driven). They sell several heat-driven refrigerators which incorporate the absorption cycle and are powered by propane, mainly. Further research suggests that their refrigerators are manufactured by a company in Brazil named Consul. Possibly these units are made to Danby's specifications by Consul but this is not a fact.

Danby Products Inc.
P.O. Box 669
Findlay, Ohio 45839-0669
1-419-425-5052
www.danby.com
http://danby.com/en/printFriendly.asp?model_no=DPR2262WCD&dept=0009

2) Sunfrost

This company manufactures both refrigerators and freezers which appear to be designed to be the most efficient as possible. They use a vapor-compression cycle yet but are designed to be highly efficient (thermally) and even use solar power to offset the dependency on electrical power from the outlet. The website mentions this particular product being used in locations where power is "off-grid".

Sunfrost
824 "L" Street
Arcata, CA 75521
1-707-822-9095
www.sunfrost.com

3) Dometic

Swedish manufacture of refrigerators and freezers using the absorption cycle. They appear to market their products to recreational use such as in RV's or a boat. They also sell products to air condition semi trucks. Products from Dometic are seen under the Servel product name.

Dometic Corporation
P.O. Box 490
Elkhart, IN 46516
1-574-294-2511
www.dometicusa.com

4) Atlantic Mini-Fridge Company Ltd.

Primarily they are a manufacture of mini-bars and chest freezers which operate with the absorption cycle. From information from the website, it appears that their products will operate when powered from propane, butane, kerosene, or natural gas. Products are sold under the name Frostek.

Atlantic Mini-Fridge Company Ltd.
1-877-426-3646
www.amf-bartener.com

5) Equator Advanced Appliances

This company based out of Houston Texas seems to be a North American manufacture which sells high efficiency refrigerators under the brand name Conserv. Further investigation proved that these are actually made by a Scandinavian company by the name of Vestfrost. They appear to use a normal vapor-compression cycle but are supposedly of a higher efficiency.

Equator Advanced Appliances
Equator Plaza
2801 W. Sam Houston Pkwy North
Houston, TX 77043-1611
1-713-464-3422
www.equatoronline.com

6) Crystal Cold

This is a LP refrigerator that is seen for sale from several online stores but it has been difficult to locate the actual manufacture of the product. One website suggests that the refrigerators are made by the Amish and is located in central Illinois but it is most likely that this company is selling the refrigerators as their own when and are just buying them from another source.

Possible Buyers/ Liscensees

1) Amana

A division of the Maytag Corporation

Amana (Maytag Corporation)
403 W. 4th Street N
Newton, Iowa 50208
www.amana.com

2) Whirlpool

Whirlpool Corporation
2000 N. M-63
Benton Harbor, MI 49022-2692
1-269-923500
1-800-253-1301
www.whirlpool.com

3) Kenmore

A Sears, Roebuck and Company Brand

Kenmore (Sears, Roebuck and Co.)
3333 Beverly Rd.
Hoffman Estates, IL 60179
1-847-286-2500
www.kenmore.com

4) Frigidaire

Frigidaire (Electrolux Home Products)
P.O Box 212378
Martinez, GA 30917
www.frigidaire.com