Analysis, Testing, and Redesign of an Environmental Control System for an Engine Dynamometer

by
Taylor H. Morris

A THESIS

submitted to

Oregon State University

University Honors College

in partial fulfillment of
the requirements for the
degree of

Honors Baccalaureate of Science in Mechanical Engineering
(Honors Associate)

Presented March 8, 2016
Commencement June 2016

AN ABSTRACT OF THE THESIS OF

Taylor H. Morris for the degree of Honors Baccalaureate of Science in Mechanical Engineering presented on March 8, 2016. Title: Analysis, Testing, and Redesign of an Environmental Control System for an Engine Dynamometer
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Abstract approved: _____

Robert K. Paasch

This project analyzes, tests, and redesigns an environmental control system for Global Formula Racing's (GFR) dynamometer. It seeks to develop a working system to test GFR's car engine under air conditions similar to those faced during competition. A model is developed to predict the performance of the environmental control system, and the model is validated through comparison to physical testing. Based on the model and testing results, a new vapor compression system is recommended to replace the existing unit for future testing.

Key Words: Environmental Controls, Test Cell, Dynamometer Testing

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I understand that my project will become part of the permanent collection of Oregon State University, University Honors College. My signature below authorizes release of my project to any reader upon request.

Taylor H. Morris, Author

Contents

1. Project Description

1.1 Introduction

The Global Formula Racing Team (GFR) is a collaboration between Oregon State University (OSU) and Duale Hochschule Baden-Württemberg - Ravensburg (DHBW), which competes in student Formula Society of Automotive Engineers (FSAE) competitions in the United States and Europe [1].

Competition Details
Formula SAE is an international student competition for universities to design, manufacture, and race formula-style racecars. The teams are given the task of developing a prototype of a racecar for non-professional autocross racers. Each year, FSAE limits these designs with a set of rules, many of which focus on safety [1].

Every year approximately 13 competitions are held worldwide where teams meet to race their cars and compete in a variety of events. These competitions are engineering competitions, not racing competitions. To this end, different amounts of points are awarded at each event based on car design and performance rather than being based purely on which car is the fastest. These events are divided into Static and Dynamic events [1].

Static Events
There are three point-based static events: engineering design; cost report; and business presentation [2].

The engineering design event requires teams to defend their design decisions to a panel of judges. Prior to competition, teams submit a design report for the judges to review. At the design event, the team must be prepared to speak and answer questions on fundamental knowledge of the vehicle, details of each subsystem on the car, and how the subsystems relate to each other and the overall vehicle. Scoring for this event is between 0 - 150 points [3].

The cost event is divided into three portions. First, like the design event, a cost report must be submitted in advance of the competition. This report comes in the form of a complete Bill of Materials and Manufacturing Plan for the vehicle. Points are awarded based on how cheap the car is compared to all the other cars in competition. Second, at the cost discussion, judges examine the car to validate the accuracy of the cost report, and teams are allowed to defend the report. Finally, the teams will present a plan to alter the vehicle design or manufacturing plan in order to reduce the cost of the vehicle in accordance with a scenario that is released two weeks prior to the competition. Scoring for this event is between 0 - 100 points [4].

The business presentation event requires that the team give a presentation to fictitious investors to demonstrate that the vehicle design will be attractive to amateur racers and that the car can be manufactured, marketed, and sold in a profitable way for the investors. Scoring for this event is between 0 - 75 points [5].

Dynamic Events
There are four dynamic events: endurance; autocross; skidpad; and acceleration [6].

The endurance and fuel economy event is designed to measure the reliability, fuel economy, and performance of the vehicle. Each team has one chance to complete the course, which is usually about 22 kilometers long. During the event, no work may be done on the car, and only one driver change is allowed. Points are awarded based on fuel used and time to complete endurance. Fuel economy score is calculated based on the volume of fuel used compared to the minimum amount of fuel used at the event. The endurance score is calculated based on time to complete compared to the minimum time at the event.

The maximum score for these events varies from competition to competition. However, scoring for this event is usually between 0 - 400 or 0 - 425 points [7].

The autocross event is designed to test the vehicle's performance in acceleration, braking, and cornering. Each team gets four runs on the course, which contains a series of straightaways, turns, and slaloms. The best time is used for scoring, which is based on the time to complete the course relative to the fastest and slowest times at the event. The maximum score for this event varies from competition to competition. However, scoring for this event is usually between 0 - 150 or 0 - 100 points [8].

The skidpad event is designed to test the lateral acceleration capability of the vehicle. Like the autocross event, each team is given four runs on the course. The skidpad course is a figure-8, and two runs are taken to the right, while two runs are taken to the left. The best time across all runs is used for scoring, which is based on the time to complete the course relative to the fastest time at the event. Scoring for this event is between 0 - 50 points [9].

The acceleration event is designed to test the straight-line acceleration of the vehicle. Four runs are conducted on the 75 meter straightaway course. The best run is used for scoring, which is based on the time to complete the course relative to the fastest time at the event. Scoring for this event is between 0 - 75 points [10].

GFR's goal is to win every competition they enter by earning more points than all other teams, and they strive to associate all design decisions with competition points. When this is not possible, GFR adheres to a strict philosophy of "simplicity, reliability, and simulation validated by physical testing" to guide design decisions.

GFR focuses on designing a system for scoring the most points at competition rather than merely designing a car. This system includes everything including the people, the organizational structure, and project management. Testing the subsystems of the car is important, not only to remain true to the team philosophy, but also as a component in the system that has proved successful for the team [11].

One subsystem that must be tested is the engine. Because GFR designs and builds a new car every year, it is beneficial to be able to begin engine testing before the rest of the vehicle is complete. The ability to run the engine before the rest of the car is complete is valuable because it helps break-in the engine, and it allows time to test the engine and generate the tunings which will be used at competition.

In order to do this, GFR makes use of a SuperFlow SF-901 engine dynamometer (Dyno) for engine testing [12]. There are several types of dynamometers, but the SuperFlow SF-901 Dynamometer is a hydrokinetic, or hydraulic dynamometer [13]. In this type of dynamometer a shaft with a cylindrical rotor revolves in a watertight casing. As the rotor is driven, centrifugal forces set up circulation within the casing, and a torque is generated to resist the rotation of the shaft. In this manner, the work from the motor is converted into heat which is rejected to the water [14].

This is valuable, because the engine is controlled by a MoTeC Engine Control Unit (ECU) which handles fuel injection and spark timing. The ECU is tuned during dynamometer testing to generate different maps specific to different events at competition [15]. The ability to run the engine on the dynamometer before the rest of the car is complete is valuable because it helps break-in the engine, and it allows time to test the engine and generate the tunings which will be used at competition. Currently, these tunings are based purely on engine torque and RPMs [16].

One limitation on the current dynamometer is that it can only be tested with the air available in the engine room, while the car will be required to run with air at varying temperature, humidity, and pressure, depending on the current conditions at the time and location of competition.

The idea that dynamometer testing should be conducted under various conditions is not new. Many companies already use some form of environmental control during dynamometer testing. For example, Mitsubishi Electric Automotive America created an 8,500 sq. ft. all-weather test chamber for their

dynamometer testing. The system is capable of temperatures from -40 to +251 Fahrenheit, can range humidity from 0-90 percent, and can simulate wind speeds and UV radiation from the sun [17]

During the 2014-2015 design cycle, a GFR project was started to create an environmental control system in the engine room to control the air temperature, humidity, and pressure which is fed into the engine. The result of that project is a four-stage system to control the given parameters. The first stage controls the pressure, the second stage cools and dehumidifies the air, the third stage reheats the air, and the final stage re-humidifies the air. This system is contained within a ducting system shown in Figure 1.1.1.

Figure 1.1.1: Environmental Control System Ducting [12]

The system was designed and built during the 2014-2015 design cycle, but it was never used or tested. The purpose of this project is to test the current system, repair and redesign the system as needed, and to use the system to design and run tests on the engine to determine the need for additional MoTeC tunings to account for variations in temperature, humidity, and pressure.

In order to assess the current system, it is first critical to know what parameters are important for engine testing. The 2014-2015 design selects air temperature, pressure, and humidity as important parameters for engine tuning and performance.

Effect of Temperature on Engine Performance

In racing conditions, air temperature can vary throughout the day, and is dependent on the location and time of year. It has been shown that as inlet air temperature to an engine increases, the power output of the engine will decrease [18]. MoTeC advises that the standard fuel injection temperature compensation tables will usually be sufficient as long as tuning occurs in moderate conditions [19].

Effect of Pressure on Engine Performance

Location of competition can affect air pressure. Changes in altitude in particular can alter the air pressure. In turn, inlet air pressure can have significant effects on engine performance because of a reduction in air density and available mass for combustion [20], and increasing altitude and decreasing air pressure have a negative influence on engine performance [21]. However, the change in pressure due to altitude between the testing location and competition venue is likely to be small. This will be explored in further depth during requirement evaluation.

Effect of Humidity on Engine Performance

The effects of humidity on engine performance have been noticed for almost as long as engine testing has been done. In a study on humidity and engine performance, Gardiner noted that it is important for humidity to be considered when doing testing on automobile engines [22]. Higher humidity has some positive effects in the form of lowering emissions and reducing knock risk, but engine efficiency is reduced [21]. However, MoTeC suggests that a humidity sensor is not necessary with their ECU because humidity affects air density less than temperature and pressure, and can be accounted for through feedback control [23].

The significance of this project to the team is that it will provide research into the importance of alternate ECU tunings. The tests designed by this project will enable the team to test the engine under a variety of environmental conditions in order to develop alternate engine tunings as necessary. Because it would not be possible to test the engine at every possible combination of temperature and pressure, it is important to be able to test the engine under a limited number of conditions and use that information to develop appropriate engine maps for different scenarios.

Tests should be able to be conducted under a variety of air conditions to assess and improve the reliability of the engine under different environmental conditions. Additionally, the system should be able to consistently reproduce the same conditions so that the engine can be tested under a consistent set of average conditions.

The goal of this project is to take GFR's current environmental control system and analyze, test, and redesign it such that it can be used successfully in the future to improve engine performance under competition conditions.

1.2 Rules and Constraint Analysis

Rule Constraints

This project is not limited by the competition rules because no part of the environmental control system will be attached to the car or used in competition. The environmental control system is merely to be used as a design tool to inform design decisions, improve MoTeC tunings, and to justify design decisions at competition.

Point Gains

While the environmental control system is not restricted by the rules, it can benefit overall point total at competition.

Design Event

The environmental controls for the dynamometer will benefit the team in the design event, regardless of the findings during testing. Data taken will be able to inform and justify design decisions related to MoTeC tuning and usage under various environmental competition conditions, which will enhance the design report and allow the team to better defend choices at the design event.

Business Presentation and Cost Event

The environmental controls will likely have no effect, positive or negative, on the business presentation or cost event.

Endurance and Fuel Economy, Autocross, and Acceleration

The environmental control system has the potential to increase points earned at these dynamic events by improving engine performance and increasing engine torque. Based on Trevor Takaro's "Equating design parameters to points," simulations have shown that additional engine torque results in an increase in points in these events [24].

Skidpad

However, the environmental control system will likely have no effect, positive or negative, on the skidpad event based on the lap simulations. Takaro's simulations show that additional engine torque does not result in additional points in this event.

1.3 Requirements

In order for the environmental control system to be useful, it must be able to simulate the conditions that the car will face at competition. The exact temperature, humidity, and pressure that will be present at competition are unknown, but certain ranges can be assumed based on competition locations and historical data.

Historical weather data for Michigan, Germany, and Austria indicate that in order to simulate average conditions, the environmental controls must be able to create conditions between 10 and 30°C, with relative humidity of between 60% and 75%, and air pressure (to account for altitude changes) of 92 kPa to 101 kPa. More extreme temperature ranges could be useful from -3°C to 37°C, but the extremes fall outside of typical race conditions [25][26][27]. The data is summarized in Table 1.3.1.

Table 1.3.1: Average Environmental Conditions at Competitions

Location	Avg. Low T (C)	Avg. High T (C)	Record Low T (C)	Record High T (C)	Avg. Relative Humidity (%)	Altitude (m)	Effective Pressure (kPa)
Michigan	8	22	-3	34	67	300	98
Germany	16	26	9	37	63	105	100
Austria	10	23	2	35	73	700	94
Extremes	8	26	-3	37	63 - 73	700	94

Additionally, the project must take into account how tuning with the environmental control will affect powertrain reliability and how the dynamometer readings can be corrected to account for environmental conditions.

The environmental control system is limited physically by available space in the engine room and by necessary interface with the dynamometer. It is a requirement that the environmental control system be designed in such a way that the engine intake pulls the controlled air into the engine during testing, rather than the building air. However, this must be done in such a way that the environmental control outlet can be positioned out of the way during other dynamometer testing and maintenance.

In order for team members to be successful in this project: the current state must be adequately assessed so that the required modifications can be made; calculations must be done to determine what conditions the engine must be tested under; and appropriate tests must be developed to test the engine under these conditions.

As noted in Section 1.2, the environmental control system will not be physically attached to the car or present at competition. However, the ways in which the system will affect each competition event were detailed in Section 1.2. In addition to the effect on points, the system has potential to affect the way that tunings are used during competition if findings show that significant gains could be made by using alternate tunings to account for differing environmental conditions.

1.4 Literature Review

2008 (Moran [28])

There are two main types of internal combustion engines: spark-ignition and compression-ignition. In compression-ignition systems, air is compressed until the pressure and temperature become high enough for combustion to occur immediately once fuel is injected. However, internal combustion engines in automobiles typical operate with a spark-ignition. In these engines, a spark ignites the air-fuel mixture.

A reciprocating internal combustion engine consists of a piston-cylinder system. Most engines are either a four-stroke or a two-stroke engine. This refers to the number of unique strokes the piston makes during one cycle. In a four-stroke spark-ignition engine, the piston expands to draw the air-fuel mixture into the chamber. Then, work is done on the piston; the piston compresses the mixture, which raises both the temperature and the pressure.

Combustion is then initiated by a spark. This forces the piston back, which generates work. Finally, the piston forces the exhaust out of the chamber, and the cycle is ready to begin again. The entire process takes two revolutions of the engine. In a two-stroke engine, this process is all done with one revolution of the engine.

2013 (Husaboe [20])

In this study, Husaboe built on previous research to analyze the effects of temperature and pressure variation on a two-stroke internal combustion engine. The study used a test chamber capable of imitating atmospheric conditions to model the engine performance at high altitudes.

The test chamber had been previously used to control pressure for automotive engine testing, but was not designed for temperature control. An air to liquid nitrogen heat exchanger was used to cool the air in the test chamber to mimic the lower air temperatures at high altitudes. However, no humidity control was used during testing.

Once the flow passed through the heat exchanger, the cooled air flowed directly into the engine intake through 1.3 cm diameter tubes. Additional cool air was pumped into the test chamber to cool the engine.

The engine was tested at 268 K, 278K, and 295K. The high temperature, 295 K, represented room temperature in the test chamber. This test was conducted without temperature controls. The two lower temperatures were meant to represent temperatures at increased altitudes of 1.5 km and 3 km.

Husaboe noted that the effect of inlet air pressure on the engine performance was substantial. Lower pressure means lower density of the air. This means that less air by mass is present in the cylinder for the combustion cycle. Increased pressure has the opposite effect, increasing the mass of air in the engine.

Prior to testing temperature variations, tests were conducted at various pressures to establish a baseline. Once the effects from pressure were measured, the temperature was varied to assess the additional effects of cold air on engine performance. During these tests, the pressure was held constant (within the limits of the pressure control system).

Prior to the testing, Husaboe calculated expected decreases based on changes of density due to the ideal gas law in order to compare the results with measurements taken during testing. As Husaboe expected, the power produced by the engine decreased as the pressure increased. The tests found that simulating 1.5 km altitude, the power dropped to 70%. At 3 km, the power dropped to nearly 50%.

Husaboe then conducted the tests of temperature changes at constant pressure. At 84 kPa and 278K, conditions at 1.5 km, the pressure was at 85% of the room condition, whereas the temperature was at 94% of its room temperature condition. This indicates that increases in altitude result in a more

significant effect from pressure than from temperature.

Because of this disparity, pressure is much more significant to engine performance at altitude than temperature difference and results in a net decrease in engine performance at altitude.

2012 (Martyr and Plint [29])

The condition of the air being used for combustion can have significant effects on the performance of the engine. Because of this, it matters under what conditions an engine is tested. There have been efforts to standardize the results of engine tests across differing air conditions. In an ideal case, all engines could be tested at the same standard air condition. However, this is not possible in the real world.

For most commercial purposes, it is sufficient for the engine to be tested with the available air. The results are then standardized with a correction factor to allow for comparison between different engines tested under different conditions.

However, while this method is acceptable for most engine specification and benchmarking, in research and development settings it may be imperative to control the air to constant conditions. In general, the three important parameters are temperature, humidity, and pressure. However, depending on the application, polluting particles in the air may also be of concern.

In large facilities, central air-conditioning may be used to keep multiple test cells at the same standard condition. However, in the case where each individual test cell has specific needs, individual air-conditioning units will be required for each cell.

Specialized companies develop and sell units to control air for engine testing. The simplest of these systems control only temperature and are “flooded inlet” units. These units supply excess air to the test cell to ensure that more conditioned air is available for the engine than is required.

Though not the primary purpose of these systems, some humidity control is possible. By running the air through chiller coils and then reheating the air to the desired temperature, it is possible to somewhat standardize the humidity of the air throughout a day of testing without installing specific humidity controls.

The next level of system directly controls humidity as well as temperature. This system is similar to the temperature control but with an added humidifier. As in the temperature control system, some moisture will condense out of the system during cooling. After the air is reheated to the desired temperature, the humidifier then sprays water into the air stream in order to raise the humidity of the air back up to desired levels.

Finally, the most complex systems control temperature, humidity, and pressure. In this case, the air outlet must be fixed to the engine intake to create a seal. Additionally, the air duct must be sufficiently large to prevent unintended pressure drop during engine acceleration. For accurate results, the engine exhaust must also be pressure controlled to fully simulate altitude conditions.

In addition to the air control systems, full environmental chambers are sometimes created for cold start testing. However, cold start systems exclusively for powertrain testing is not common. More often cold chambers are used for work on the complete vehicle.

2015 (Leskinen et al. [30])

Leskinen et al. used a similar chamber for testing aerosol combustion. The environmental chamber, located at the University of Eastern Finland was previously used for small-scale combustion and vehicle exhaust experiments with a dynamometer.

This chamber had temperature, humidity, and pressure control. The chamber was roughly 30 cubic

meters with a surface area of 58 square meters. The chamber was essentially a collapsible box made of uniform cross section aluminum frames. The pressure in the chamber was controlled by adding or removing weight from the top frame.

The temperature was controlled by an air conditioner capable of 6.5 kW of cooling. The temperature could be controlled between 16°C and 25°C, and the temperature control could maintain temperature in the operating range within 1°C.

This system also ran a humidifier in conjunction with an air purifier to remove contaminants and add moisture to raise the humidity of the air.

2015 (Stalp [12])

The 2015 GFR project at OSU took a different approach. The environmental control designed by Stalp relied on a freezer as a cold reservoir for cooling the air down. Ethylene-glycol was pumped through the freezer to cool it, then passed through a cross flow heat exchanger with the air for the engine intake. This served to reduce the humidity of the air and lower its temperature.

Then the air was reheated by a second ethylene-glycol cycle. In the second cycle the ethylene-glycol passed through a series of tube heaters to preheat the fluid before it passed through a second heat exchanger. This process reheated the air which would then be at a lower humidity.

The temperature could be controlled by these two cycles, and a series of ultrasonic humidifiers was placed after the second heat exchanger to boost the humidity of the air up to the desired levels.

This system was specifically designed to be used by GFR for dynamometer testing of the powertrain.

2013 (Porges [31])

An alternative to the above mentioned cooling cycle is the vapor compression cycle.

The vapor compression cycle works by moving heated, compressed vapor refrigerant through a heat exchanger with a cooling fluid. The refrigerant is then expanded through a valve and moved through a second heat exchanger. The expanded refrigerant becomes a low pressure liquid and absorbs heat through the heat exchanger. As it absorbs heat, it vaporizes and enters the compressor. From there the cycle repeats.

The vapor compression cycle takes advantage of latent heat to transfer large amounts of heat without large temperature gradients. As a liquid begins to evaporate, it will absorb heat while remaining at a constant temperature. This gives the fluid an extremely high specific heat during the evaporation process. Once all of the liquid has turned to vapor, the temperature will begin to rise.

Vapor compression cycles are highly dependent on the refrigerant used. Air and water were two of the earliest fluids used for heat transfer, but other refrigerants were developed to better suit the needs of specific applications.

2008 (Hundy [32])

Many different types of refrigerants have been used throughout the years. Ideal properties for refrigerants include: high latent heat of vaporization; reasonable, positive pressures; critical temperatures well above the high temperature of the system; non-toxic; environmentally friendly; non-flammable; non-corrosive; and inexpensive.

From an environmental perspective, major changes came about in the 1930s with the development of chlorofluorocarbons (CFCs). These refrigerants seemed perfect at the time; they had very good thermodynamic properties, they were non-flammable and non-toxic, and they quickly became common

refrigerants.

Unfortunately, these refrigerants turned out to be catastrophic for the Ozone Layer. As CFCs were released into the atmosphere, chlorine atoms would split off, break up ozone, and bond with the remaining oxygen atoms.

These reactions caused a hole in the ozone layer, and the issue drew international attention. The Montreal Protocol was an international agreement in the late 1980s, which was ratified by all members of the United Nations, and began the phase out of the ozone destroying CFCs.

2011 (Daly [33])

Another environmental concern is global warming potential (GWP). R134a was developed to replace harmful CFCs, and is now the most common refrigerant used in automotive vapor compression cycles. However, it is worth noting that R134a is slowly being phased out due to its high GWP.

Global warming potential is the measure of the impact of a refrigerant on the greenhouse effect relative to the effect of carbon dioxide. R134a has a GWP of 1300. This means that it has 1300 times the impact on the atmosphere as carbon dioxide. In other words, 1 kg of R134a released into the atmosphere is equivalent to 1.3 tons of carbon dioxide.

Legislation has passed in the European Union phasing out R134a and other refrigerants with high GWP over the next few years. Now similar measures are being considered in the United States to reduce the greenhouse gas emissions that result from leaked refrigerant.

The two leading alternatives to the common vapor compression cycles are Trans-critical CO2 systems and vapor compression cycles using drop-in replacement refrigerants which have similar operating conditions as R134a.

2012 (Yan et al. [34])

One alternative to vapor compression systems is Ejector-cooling systems (ECSs). ECSs have been a focus of several studies in the literature because they are cheap and easy to install. The coefficient of performance (COP) of ECSs remains low compared to vapor compression cycles.

One way to improve the COP of ECSs is to combine them with vapor compression cycles rather than completely replace them. These ejector-vapor compression cycles (EVCCs), improve the COP of ECSs.

Yan et al. developed an EVCC using refrigerant R134a. This system used an ECS connected to a vapor compression cycle through a subcooler. By using this combination of system, a higher level of subcooling was achieved.

2015 (Yang et al. [35])

The theoretical efficiency of refrigeration cycles can be compared to the efficiency of the Carnot cycle. While it is impossible for real cycles to match the Carnot cycle, much effort is expended in the literature to improve the efficiency of refrigeration cycles.

This study, by Yang et al., attempted to design a more efficient refrigeration cycle for domestic use. To improve the efficiency, four potential cycles were examined. Each cycle had two evaporators and two compressors. The components were connected either in series or in parallel, and each of the four unique combinations were considered.

The study found that the most efficient method for connecting evaporators and compressors is to link the evaporators in series and the compressors in parallel. This method results in the highest coefficient of performance (COP) and the lowest mass flow rate.

Using simulations, Yang et al. discovered that this configuration resulted in a larger change in enthalpy for a smaller mass flow rate. Lowering the mass flow rate in turn lowers the power required by the compressors, which results in a higher COP.

However, this configuration has disadvantages. Most importantly, in reality it was difficult to separate the liquid and vapor streams between the two evaporators due to the small mass flow rate.

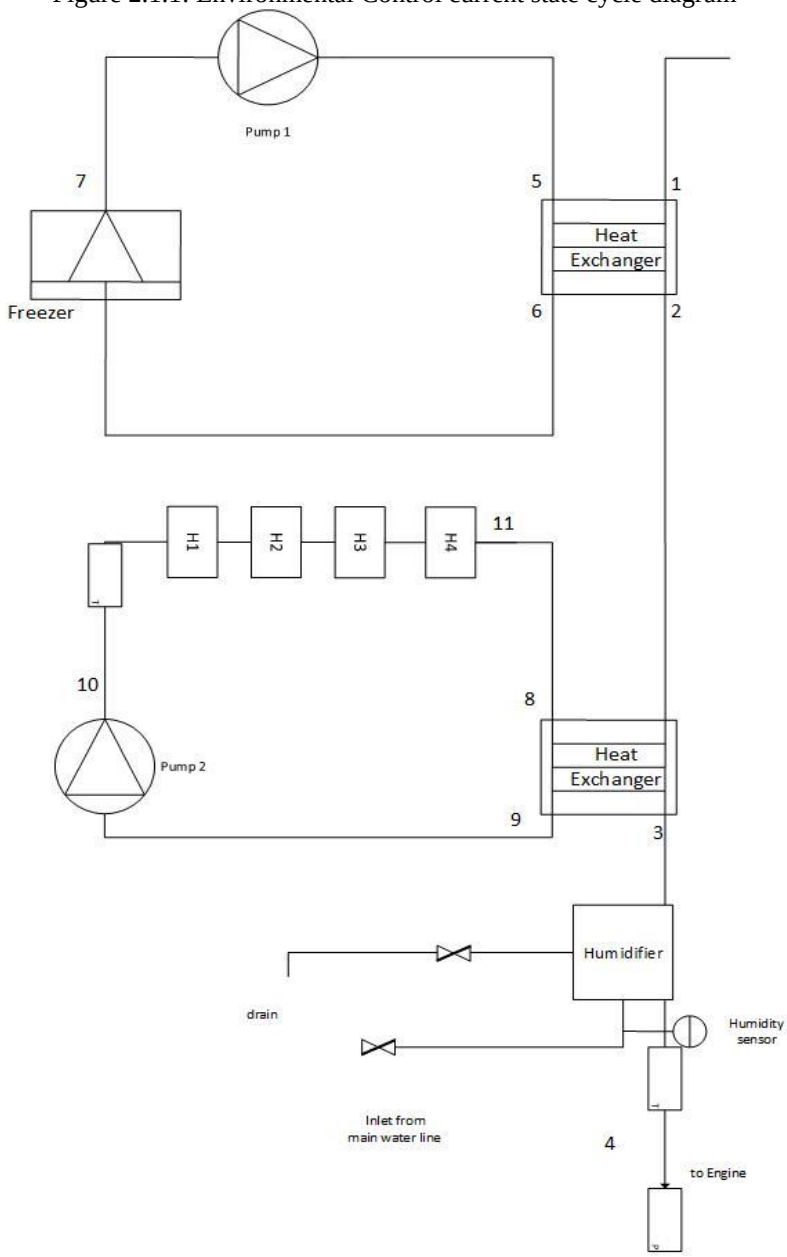
To address this problem, Yang et al. developed a new cycle which used a "two-circuit cycle with evaporative subcooler" to move the evaporators back into parallel configuration. The addition of the subcooler allows the second evaporator to reach colder temperatures. The mass flow rate in this configuration was the same as in the series configuration, but the efficiency did decrease somewhat. However, the COP of new cycle was still higher than the alternate combinations of compressors and evaporators

2. Current State Analysis and Benchmarking

2.1 Current State Analysis

The overall system is designed such that the engine will pull air from the ducting of the environmental control system. The system relies on the engine to pull the air through the environmental controls. The air travels through the ducting, passing through four stages. Figure 2.1.1 shows the cycle diagram for the current state.

Figure 2.1.1: Environmental Control current state cycle diagram



The first stage of the system is designed to control the air pressure. The design relies on a sliding metal sheet to restrict airflow into the ducting. Figure 2.1.2 shows the system completely open and Figure 2.1.3 shows it completely closed [12].

Figure 2.1.2: pressure control open [12].

Figure 2.1.3: Pressure control closed [12].

The second stage, from State 1 to State 2, cools and dehumidifies the air using a radiator. The working fluid of the radiator was unspecified, and it is cooled by running the fluid through an antifreeze bath in an external chest freezer before being pumped through the radiator. Figure 2.1.4 shows the freezer with insulated copper pipe carrying the working fluid from the freezer [12].

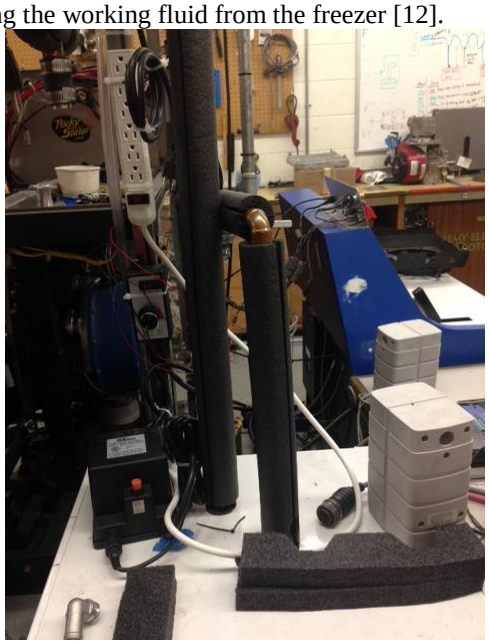


Figure 2.1.4: Freezer with attached pipes [12].

The third stage, from State 2 to State 3, reheats the air using a second radiator. The working fluid for this portion of the cycle was not identified in the previous report. Additionally, the working fluid level was too low to run the heating-side pump. This could be because not enough fluid was added to the system, or the fluid could have undergone a significant pressure change and escaped through the included safety valve. However, the intended design has the working fluid pumped through the pipe and heated by a series of four tube heaters before entering the radiator. According to the 2015 report, the tube heaters are Kat brand radiator heater [12]. Figure 2.1.5 shows the four tube heaters.

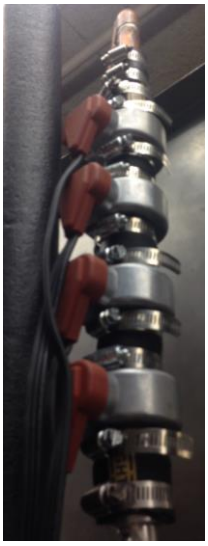


Figure 2.1.5: Tube heaters for heating cycle.

The final stage, from State 3 to State 4, re-humidifies the air using an ultrasonic humidifier system. The water for the humidifier was originally taken off the mainline which supplies water to the dynamometer [12]. However, the water line was disconnected from the humidifier. The humidifier is controlled by a voltage controller (Figure 2.1.6) and is contained beneath the ducting, as shown in Figure 2.1.7.

Figure 2.1.6: Voltage controller for humidifier.

Figure 2.1.7: Humidifier Container

SWOT Analysis

Table 2.1.1: SWOT Analysis

Strengths	Weaknesses
● Simple design ● Simple operation	● Not currently operational ● Limited control over experimental variables
Opportunities	**Threats**
● Interest in acquiring data on engine performance in different conditions	● Lower priority, because environmental controls are not required for car to run

2.2 Benchmarking

The current state of the environmental control system is nonfunctional. Benchmarking the current system as it performs is not extremely useful. Currently the system is unable to cool the air, heat the air, humidify the air, or change the air pressure.

Benchmarking the current components and the intended design is a more difficult but potentially useful effort.

1. **Cooling cycle**
   For the cooling cycle, the pumps used are Chugger Pump CPP 115V Inline Pumps. For these pumps the max head is 4.1m, the max flow rate is 26.5 LPM, and the max temperature is 121°C [36]. The freezer used to cool the working fluid is an Igloo FRF434 chest freezer and can produce temperatures between 0 and -18°C, according to the specifications sheet [37].

   Based on a model created in Engineering Equation Solver (EES), under ideal conditions this cycle would be able to lower the air temperature to 10°C. The system is not currently operational, so it is not possible to experimentally determine a true value. Once physical testing becomes possible, the model will be updated.

2. **Heating Cycle**
   For the heating cycle, four 600W heaters are used to heat the working fluid. The pump used is the same as in the cooling cycle. The EES model indicates that under ideal conditions with 100% efficiency, the cycle can heat the air temperature to 37.7°C. Like with the cooling cycle, this model will be updated as better information becomes available [38].

3. **Humidifying**
   For the current state, the humidifier was not connected to a water supply, which prevented it from humidifying the air. However, based on the specifications of the House of Hydro ultrasonic Mist Maker, it can output 6000 mL of water per hour. This can be tested further once the system is operational [39].

The primary tasks required in order to get the current design working are:

- Reconnecting water to the humidifier
- An appropriate working fluid needs to be added to the heating-side cycle.
- The ducts need to be properly sealed

Additionally, it would be beneficial to the design to replace the copper pipes that carry the working fluid in the heating and cooling cycles. Currently the pipes are unsupported connections between the pumps and the ducting. It falls outside of the scope of this project, but vibrations could be of concern given that the piping is located right next to the dynamometer. Additionally, the pipes on the cooling side loop through the freezer, entering through holes in the top. This makes it impossible to move the freezer without dismantling sections of the copper piping.

One solution would be to use PEX tubes instead of copper. PEX tubes would be cheaper than copper, and these tubes could be disconnected and reconnected more easily. Another benefit is that PEX tubing is more flexible, and alleviates the needs for multiple connections and bends in the pipe.

3. Design Analysis

Because the initial purpose of this project is to finish and repair the existing system, the design phase focused primarily on areas of the environmental control system that needed to be completed or fixed rather than areas that could be improved and redesigned.

Three areas were identified from the current state and benchmarking analyses:

1. A water line needs to be connected to the humidifier
2. An appropriate working fluid needs to be added to the heating side.
3. The ducts need to be properly sealed

Water Line

Reconnecting the waterline to the humidifier was a straightforward task. It required cutting new tube and adding a split to the dynamometer main water line, but no design analysis was required.
The new waterline was built to connect the humidifier to the water supply, and the humidity sensor was reconnected to monitor the humidity in the air. However, data cannot be collected on how well the humidifier is working until the full system is operational.

Working Fluid

Concept 1: For the working fluid in the heating cycle, one option is a 70% (volume) ethylene glycol, 30% (volume) water mixture. This will raise the boiling point to roughly 118°C [40]. The 2015 Report states that the temperature of the fluid going into the radiator should not exceed 100°C. No justification is given based on the working fluid, but using a working fluid with a higher boiling point would give the option of a higher temperature fluid to pass through the radiator [12].

Concept 2: The other option for working fluid in the heating cycle is a 60% (volume) propylene glycol, 40% (volume) water mixture. This would raise the boiling point to approximately 107°C. This boiling point is not as high as the ethylene glycol-water mixture, but it is still higher than the stated 100 degree limit [41].

Table 3.1: Working Fluid Concepts

Concept	Boiling Temp (C)	Freezing Temp (C)	Est. Cost ($/gal)
Ethylene glycol	118	-60	12
Propylene glycol	107	-48	56

Seal Ducting
Concept 1: Use duct mastic to seal any gaps. Mastic is a non-hardening putty like substance which is used to seal ducting [42].

Concept 2: Alternatively, cloth backed adhesive duct tape can be used to seal gaps. The benefit of this concept is that it is a quick and easy solution. However, it is more of a temporary solution than mastic, as such tapings typically do not last long [42].

4. Design Selected

As indicated in Section 3, the selected design will largely use the existing system. However, changes are necessary to the heating cycle, pressure control, and overall ducting in order to make the system operational and useful to the team.

The selected design will continue to use the cycle shown above in Figure 2.1.1. The chest freezer will be resealed to prevent leaks, but the freezer cycle will be used as previously designed. The heating cycle needs a replacement working fluid. A water mix of 70% ethylene glycol will be used to ensure a sufficiently high boiling point while keeping costs down.

The installed humidity controller will continue to be used in the new design. As with the heating and freezing cycles, changes to the humidifying controls would add unnecessary expense to the project before determining if the current system works.

4.1 Rationale for Selection

By using the current freezer cycle, the system is able to achieve a higher cooling capacity by offsetting the peak load of the system. Because the freezer can be run independently of testing to maintain an ice block, a large compressor/expansion refrigeration cycle can be avoided. Additionally, the chest freezer and necessary components are already available, which saves additional unnecessary cost.

Using the current heating cycle, with replacement working fluid, will allow the system to meet requirements. Alternate heating cycles could improve controllability of the system, but would ultimately take time and resources away from a functional system. The downside of the system is the need to manually control the individual heaters. However, an electronic control could be designed to interface the system with MoTeC at a later date. Additionally, a new internal heating system would require dismantling the current ducting and adding unnecessary time and cost to the project when the current system can be used to raise the air temperature within the ducting.

The specifications of the humidity system indicate that it should meet requirements as designed. Therefore it would be costly, in terms of both time and money, to remanufacture a new system. As with the heating cycle, replacement of the humidifying system would require significant dismantling of the ducting, which would take valuable time away from repairing the system, running tests, and tuning the engine, which is the primary goal. Instead, the current system will be reconnected and tested as is.

4.2 Technical Specification

The EES model and simulation created and referenced above were used to specify the future state. The technical specifications were determined by running the model with the future state values. Below are included the base calculations that were used to relate the heating and refrigeration cycles to the eventual air outlet temperature and to fully develop the model.

Sample equations for refrigeration cycle:

$0 = Q \quad + \Sigma m_i h_i - \Sigma m_e h_e$	Steady State energy balance for heat exchanger
$Q_R = m_1(h_2 - h_1)$	Heat leaving the air duct
$Q_R = m_R(h_5 - h_6)$	Heat entering the refrigeration cycle
$m_1(h_2 - h_1) = m_R(h_5 - h_6)$	Relationship between air and the working fluid

Where,
Q_R = the heat transferred between hot and cold sides of heat exchanger
m_R = the mass flow rate through the refrigeration cycle.
m_1 = the mass flow rate through duct.

Sample equations for heating cycle:

$0 = Q \quad + \Sigma m_i h_i - \Sigma m_e h_e$	Steady State energy balance for heat exchanger
$Q_H = m_1(h_3 - h_2)$	Heat entering the air duct
$Q_H = m_H(h_8 - h_9)$	Heat leaving the heater cycle
$m_1(h_3 - h_2) = m_H(h_8 - h_9)$	Relationship between air and the working fluid

Where,
Q_H = the heat transferred between hot and cold sides of heat exchanger
m_H = the mass flow rate through the heating cycle.
m_1 = the mass flow rate through duct.

Model Results

Inlet Air (T)	25 C
Inlet Air (P)	101 kPa
Max Cooling (T)	10.5 C
Max Heating (T)	57.5 C
Outlet Air (T)	50.2 C
Outlet Air (P)	101 kPa

The calculations show that the system will be able to lower the air temperature to nearly 10°C, and raise the temperature to above 57°C.

These values are determined using EES. Thermodynamic properties of air and water-ethylene glycol mixtures can be determined using the above equations and given conditions. At room temperature in the engine room, the system should be able to output air between 10 and 50°C.

4.3 Manufacturing Plan

The physical components for the heating, cooling, humidifying, and pressure stages of the system are all currently available from the current system. An electronic throttle valve was not used in the current system, however one is available from current stock in the engine room if one is added to this system or used in future systems. Therefore, vendors are not required for materials or parts to be manufactured.

No additional piping or connectors should be necessary. However, in the event of part failures or unforeseen problems, PEX will be purchased from Home Depot and used in the heating and cooling cycles, as discussed above.

The sealing of the duct requires additional supplies. However, the sealant for the duct will be Duct Mastic

purchased from Home Depot to remove lead time. The Mastic will be applied to all openings and along the corners where the separate pieces of ducting meet. This step is necessary to ensure a pressure seal so that air will be pulled in from the duct- not from the room.

4.4 Testing Plan

4.1.1 Environmental Control Testing

Before any testing is done with the engine, it is necessary to test the capabilities of the environmental control system itself. The purpose of the system is to simulate competition conditions in the engine test room. Therefore, the ultimate test for the environmental controls is how well it is able to reach and maintain the temperatures and conditions laid out in Table 1.3.1.

The first test will be to run the cooling cycle at its maximum pump power and determine the lowest temperature achievable with the system. This step is important, not only to assess what is possible with the current environmental controls, but also in order to validate the model that was generated to analyze the system.

Likewise, the heating cycle must be run with maximum pump power and full use of the heaters to determine the highest temperature achievable. In regular operation the heating cycle will only be run if the cooling cycle is also being run. However, for preliminary testing, the heating cycle will be run both independently and with the cooling cycle to determine both maximum temperatures.

After the minimum and maximum temperature tests, testing will be done to assess the system's ability to reach and maintain key temperatures, as specified in Table 1.3.1. These target temperatures are 8, 26, and 37°C. 8°C falls outside of the expected operation of the system based on the mathematical model. However, assuming that the environmental controls will be unable to reach this temperature, the test will measure how long the system can hold at its minimum temperature.

26 and 37°C both fall within the range of expected performance. In order to hold the temperature near these targets, the heaters can be cycled on and off. Any combination of the four heaters may be used. Additionally, the cooling and heating side pumps may each be turned on and off independently.

In order to maintain the desired air temperature, the goal will be to maintain a constant temperature in the working fluid on the heating cycle. The model was generated with the expectation that the heaters would be used to regulate the working fluid temperature.

In addition to the temperature testing, humidity testing will be conducted. Humidity testing relies completely on the functioning ultrasonic humidifiers. The first test will be to assess their performance by running them at full power. The humidifiers were selected by Stalp [12] to increase the relative humidity to 100% at 45°C. However, the humidifiers were never tested.

After testing the humidifiers for the maximum humidity, the voltage controller will be used to attempt to hold the humidity at 75%.
Finally, the above data will be used to assess the overall performance of the environmental control system. In addition to the system's purpose of reaching and maintaining target temperatures and humidity, the system should be controllable enough to be used during engine testing. In its current state, it is expected that it will be difficult to manually control the system while running actual engine tests.

The testing results will be used to develop improvements for the control of the system. More precise manual control, or automatic electronic control, would be ideal for use during engine testing. Recommendations will be made to improve controllability of the system.

Additionally, if the system fails to perform sufficiently for engine testing, then the test data will be useful in redesigning faulty components. Recommendations will be made to increase the performance of the system or, in the extreme case, to redesign the overall system to better suit GFR's needs.

4.1.2 Engine Testing Plan

The goal of the environmental control system is to improve reliability and performance at competition by testing the engine under different conditions before the rest of the car is even finished. In order to do this, it is necessary to determine what conditions are important to the performance and reliability of the car and to perform the necessary engine tunings.

It would not be practical or feasible to run the engine at every possible combination of pressure, temperature, and humidity. Therefore an effective strategy is necessary to use the environmental control system to generate useful engine maps.

In order to effectively use the environmental control system, it must be determined:

1. whether the differing air conditions have a significant impact on the engine maps and performance.
2. what conditions or range of conditions have the most significant impact.
3. if it is valuable to generate maps from an average set of conditions
4. and/or if it is valuable to generate maps at a few key condition sets

First, the engine will be tested across the range of possible pressures by running the engine and environmental control system with the cooling, heating, and humidifying sections turned off and slowly adjusting the throttle valve. This will provide information on pressure ranges under which the engine can run, and it will also help to determine how strong an effect the air pressure at intake is across the likely pressure ranges.

Second, the engine will be tested across the range of possible temperatures by running the system at a constant pressure (101 kPa) and beginning with the coldest temperatures that the system can achieve. Then the heating cycle will be engaged in stages to monitor the engine throughout the temperature range. Similar to the pressure tests, this will help to determine the importance of the temperature variations, and it will also provide information to compare the effect from temperature to the effect from pressure.

Third, the engine will be tested at room temperature and pressure with varying levels of humidity. This will demonstrate the possible importance of humidity testing, and provide a benchmark to compare to the temperature and pressure data.

Using the data collected from the above tests, the first two items will be determined: how the system is affected by the changing environmental conditions, and which condition has the most significant impact. If any of the environmental conditions are determined to be unimportant, then they will be excluded from further steps. However, the testing plan assumes that all factors will be important.

Based on the collected data, a set of "average" conditions can be determined which will generate maps that best address the entire range of conditions. In other words, if engine maps will only be generated at one set of temperature, pressure, and humidity, then they should be generated under the conditions that will be most versatile to the entire range.

For the final criterion, the engine can be tested at a few key combinations. For example, the engine can be tested with cold, dry air, and separately with hot, humid air. While it would be impractical to change engine maps for every combination of humidity and temperature, it may be worthwhile to change maps for certain extreme conditions. By running tests at these levels and comparing the data with the "average" conditions and the results of tests outside of the environmental control system, it can be determined whether these maps may provide significant benefit to powertrain reliability and performance.

5. Implementation

5.1 Modelling

Engineering Equation Solver (EES) was used to generate a working model of the system. The system was modeled using information provided in the documentation of the current system and the component specification sheets, sensor readings in the dynamometer room, and measurements taken during current state analysis.

Several assumptions were made throughout the modeling process. It was assumed that all pumps added no heat to the system, that pressure drops across the heat exchangers could be neglected, and that minor losses could be neglected in the piping networks. Further necessary assumptions are detailed below in the specific sections to which they pertain.

The volumetric flow rate of the air through the duct was calculated based on the maximum expected RPM of the 450cc engine. The piston takes in air every other revolution of the engine. Therefore, the volumetric flow rate can be calculated as follows:

$$\dot{V} = Volume_{piston} * \frac{revolution}{minute} * \frac{1}{2\ revolutions} * \frac{minute}{60\ seconds} \quad \text{(eq 5.1.1)}$$

$$\dot{V} = 450cm^3 \frac{m^3}{10^6 cm^3} \frac{12000\ rev}{min} \frac{1}{2\ rev} \frac{min}{60\ sec} = 0.045\ m^3/s \quad \text{(eq 5.1.2)}$$

5.1.1 Cooling Mode

The model for the cooling mode is based on an assumed constant temperature reservoir of -13°C in the freezer. During engine testing, the freezer will be filled with a pre-frozen ethylene glycol (EG)-water mixture, and the cooling cycle will reject heat to this solid block. This model assumes sufficient mixing to maintain an isothermal reservoir as the EG-water mixture melts. This assumption is not completely accurate, but should be reasonable over the short duration of each individual test.

Pump Model

The model for the hydraulic loop of the cooling mode models the EG-water mixture as incompressible and assumes a constant viscosity of the fluid. The model assumes constant diameter pipe, constant mass and volumetric flow, and therefore a constant velocity and Reynolds number.

Friction in the piping network is calculated using Churchill's Equations [43].

$$f_{churchill} = 8 * \left(\left(\left(\frac{8}{Re}\right)^{12}\right) + \left(\frac{1}{(B+C)^{1.5}}\right)\right)^{1/12} \quad \text{(eq 5.1.3)}$$

$$B = 2.457 * ln\left(\frac{1}{\left(\frac{7}{Re}\right)^{0.9} + \frac{0.27\epsilon}{D}}\right) \quad \text{(eq 5.1.4)}$$

$$C = \left(\frac{37530}{Re}\right)^{16} \quad \text{(eq 5.1.5)}$$

Minor losses are neglected in the loop because they should be small in comparison to the major frictional losses.

The performance of the pump itself can be determined by the pump's performance map [44]. For the model, the pump was specified using the manufacturer's pump curve. The efficiency of the pump can

then be determined by comparing the power delivered to the fluid, or the water horsepower, with the power required to run the pump, or the brake horsepower. The following equations, taken from White, were used to calculate the pump efficiency [45].

$$P_w = \rho * g * \dot{V} * h_{pump} \quad \text{(eq 5.1.6)}$$

$$\eta = \frac{P_w}{P_b} \quad \text{(eq 5.1.7)}$$

Heat Exchanger Model

Because the heat exchanger is enclosed in the ducting, it was not possible to physically examine it to develop a model. However, it is known that the heat exchangers are cross flow, finned heat exchangers. This is a common configuration for heat exchangers which prevents mixing of either the interior or exterior fluid [46]. An average effectiveness was assumed, but the model does not take into account any damage to the heat exchanger or additional resistance due to fouling. The effectiveness-NTU method was used to model the heat transfer through the heat exchange and to determine the rate of heat transfer from the air to the cool EG-water mixture.

Results

Max Cooling Model

The model for maximum cooling assumes inlet conditions of 25.0°C and 101 kPa for the indoor air. Using the above assumptions for the pump, piping network, and heat exchangers, the model predicts a temperature drop of 14.5°C for a steady state outlet temperature of 10.5°C. This temperature matches closely to the design parameters selected by Stalp [12].

Holding at 8 degrees

As noted above, if the system performance matches the model, the air temperature will never decrease to 8°C. Given that this is still above the desired minimum temperature for average competition conditions in Michigan, there is no model for holding the air temperature at 8 degrees. Instead, the system will be run with no heating, and if the model is accurate, the system will hold at roughly 11°C. While this is higher than desired, one advantage is that this will not require manual cycling of the pump to maintain the temperature.

5.1.2 Heating Mode

The model for the heating mode is based on an assumed constant temperature for the working fluid in the heating pipes. During engine testing, the heaters can be cycled on and off to maintain a desired temperature within the working fluid. In reality, the temperature will fluctuate during testing, but with careful heater cycling, the temperature should be maintained within 3°C. Therefore, this assumption should be acceptable during testing.

Pump Model

The pump for the heating mode is the same model as for the cooling mode. Therefore the same model was used for both. Changes were made to account for slight differences in the lengths of the piping, but the piping networks were the same for the most part.

Heat Exchanger Model

The heat exchanger for the heating mode is the same model as for the cooling mode. For this reason, the model for the heating side once again matches the model for the cooling side. The inlet and outlet temperatures on the air and liquid sides differ from the cooling mode, but the respective flow rates remain the same.

Results

Max Heating

The model for maximum heating assumes inlet conditions of 25.0°C and 101 kPa for the indoor air. The model also assumes that the cooling side will be running, lowering the intermediate temperature to 10.5°C. Using the other above assumptions for the pump, piping network, and heat exchanger, the model predicts a temperature rise of 39.7°C for a steady state outlet temperature of 50.2°C. This temperature exceeds the design parameter, of 45°C selected by Stalp [12].

If the system performance matches the model, this is well above the desired maximum temperature for average competition conditions.

Holding at 37 degrees

The model for holding the temperature at 37°C uses the same assumptions and procedure as the maximum heating simulation. However, the working fluid on the heating cycle is held to a constant temperature. The real world conditions to create this assumption would be to cycle the heaters on and off as necessary to maintain a near constant temperature. In reality, it would not be possible to maintain a constant temperature. However, with sufficiently rapid cycling, it should be possible to maintain a temperature within 2-3 degrees of the desired temperature.

In order to maintain an air temperature of 37°C at the outlet, it was found that the working fluid should be held at 64°C. A temperature of 66.0°C would result in an air temperature of 38.3°C, and a temperature of 62.0°C would result in an air temperature of 36.2°C. Therefore, a discrepancy of 2 degrees in the working fluid would still result in less than 2 degrees difference in air temperature.

Holding at 26 degrees

The model for holding the temperature at 26°C uses the same assumptions and procedure as the previous simulation. However, the working fluid on the heating cycle is held to a lower constant temperature. The target temperature is barely above room temperature conditions, so it may seem unnecessary to run the cooling and heating cycles. However, for humidity testing, it may be necessary to run the cooling cycle to condense water out of the air and lower the absolute humidity. Then the air could be reheated to room temperature at a lower relative humidity.

In order to maintain an air temperature of 26°C at the outlet, it was found that the working fluid should be held at 42°C. A temperature of 44.0°C would result in an air temperature of 27.3°C at the outlet, and a temperature of 40.0°C would result in an air temperature of 25.1°C at the outlet. Therefore, as before, a discrepancy of 2 degrees in the working fluid would still result in less than 2 degrees difference in air temperature.

Any adjustments to the engine tunings would not be able to account for such subtle temperature changes, so there is little need for more precision in the environmental control system.

6. Testing

6.1 Tests Complete to Date

The environmental control system was tested independently from the engine in order to assess the capabilities of the system. A fan was used to force air through the ducting, simulating the flow that would normally be generated by the engine intake.
The required air velocity was calculated based on the volumetric flow produced by the engine (calculated in eq 5.1.2) and the cross sectional area of duct.

$$v = \frac{\dot{V}}{A} = \frac{0.0485}{0.186} = 0.261\ m/s \qquad \text{(eq 6.1.1)}$$

The air velocity of the fan was then measured with an anemometer to determine the average velocity to ensure an equivalent airflow through the system.

$$v_{fan} = 0.715\ m/s \qquad \text{(eq 6.1.1)}$$

The fan generated a flow nearly 3 times faster than the calculated engine draw. However, due to the specific heat and heat transfer coefficient of the air compared to working fluid properties, this change had negligible effects on the model. In fact, air speeds within an order of magnitude of the engine draw resulted in changes of outlet temperature of less than 1°C.

6.1.1 Cooling Mode

Pure Cooling Test

The cooling performance of the environmental controls was tested by running the cooling cycle and forcing air through the duct with the fan, as described above. The effect of the cooling mode was minimal in comparison with the predicted results from the model. The results of one test are shown below in Figure 6.1.1.

The room temperature at the time of testing was 24.7°C. The pump was turned on to begin circulating the working fluid through the pipe network. Then the fan was turned on to force air through the duct. The temperature of the air began dropping. After seven minutes, the air temperature reached its minimum of 21.3°C. The temperature held steady for 3 minutes, and then the pump was turned off. The air temperature began to rise. Finally, the fan was turned off and the test was completed.

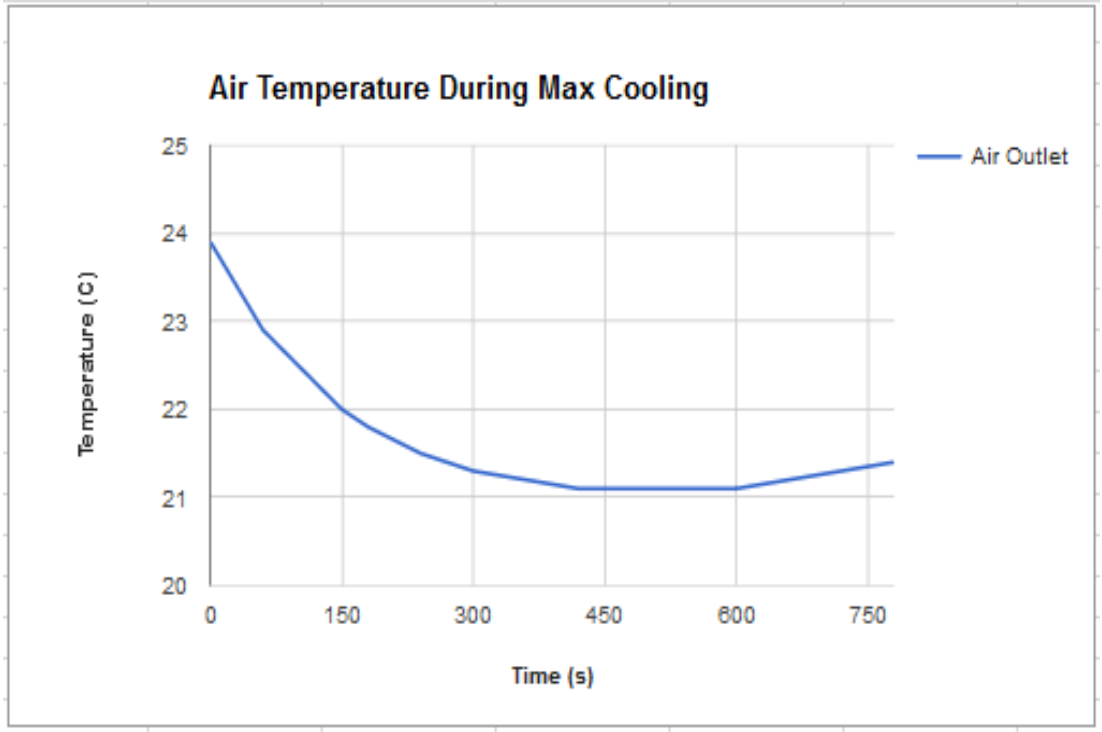


Figure 6.1.1: Air Temperature at Environmental Control Outlet

The ratio of actual cooling to predicted cooling was approximately 15%. Given that the system did not perform as expected, additional testing was conducted to attempt to diagnose the problem.

Modified Cooling Test

Because of the layout of the system, the cooled air must pass through the heating side heat exchanger, regardless of if heating is being attempted. Even with the heaters off, the air will be losing some heat to the room temperature fluid as it passes through the heat exchanger. The integrated EES model shows that the air may rise as much as 7°C through the second heat exchanger. This does not explain why only 3.4 degrees of cooling were achieved, but it could partially explain the underperformance of the system.

To assess the effect of the second heat exchanger, the cooling tests were performed again, measuring the change in temperature across the first heat exchanger. The results are shown below in Figure 6.1.2.

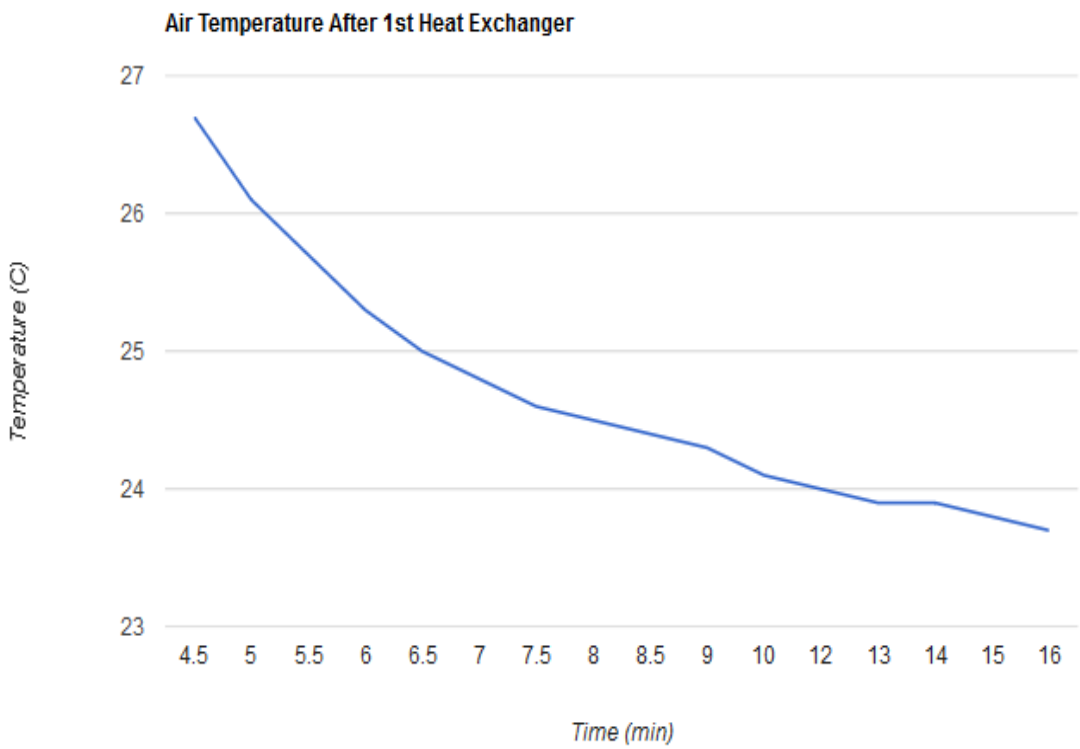


Figure 6.1.2: Air Temperature at Heat Exchanger Outlet

The room temperature at the time of testing was 26.6°C. As with previous tests, the pump was turned on to begin circulating the working fluid through the pipe network, and then the fan was turned on to force air through the duct. The temperature of the air began dropping at four and a half minutes. After fifteen minutes, the air temperature reached its minimum of 23.7°C.

Across all tests, the air temperature was only lowered by roughly 3°C. This indicates that the second heat exchanger is not the primary problem with the cooling mode. Despite expectations that the air may heat back up through the second heat exchanger, the second set of tests yielded similar results to measurements at the outlet of the duct.

Maintain Temperature at 8C

Because the cooling mode was not functioning, the tests to maintain temperature at 8°C was cancelled. To run this test without proper cooling would be meaningless as the system cannot approach anywhere close to 8°C.

6.1.2 Heating Mode

Pure Heating Test

The heating performance of the environmental controls was tested by running all four heaters and the heating cycle pump while forcing air through the duct with the fan, as described above. The cooling cycle was not run during the pure heating test.

The effect of the heating mode performed above expectations of the model and of the original design expectations. The results of testing are shown below in Figure 6.1.3.

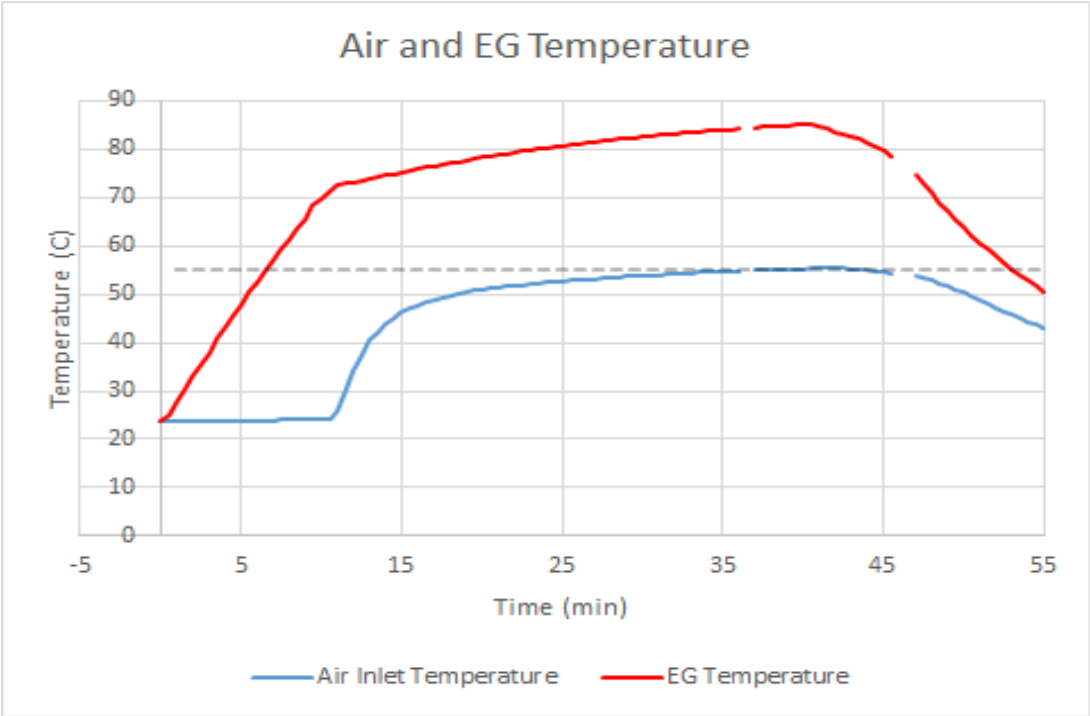


Figure 6.1.3: Pure Heating Test Results.

The temperature of the working fluid is shown in red, and the temperature of the air is shown in blue.

The room temperature at the time of testing was 23.6°C. All four tube heaters were turned on to begin heating the working fluid. The pump was then turned on to begin circulating the working fluid through the pipe network. The working fluid was allowed to heat for 10 minutes before the fan was turned on to force air through the duct.

The temperature of the air began rising immediately. As can be seen in the figure, the rate of increase began to slow down, and after an additional 31 minutes, the heaters were turned off the prevent the working fluid from exceeding 90°C. The air reached its maximum temperature of 55.4°C. Then the temperature of both the air and the working fluid began to decrease as expected.

The maximum temperature of 55.4°C was higher than specified by Stalp [12] and higher than calculated by the model. This temperature is well above what could be expected at competition, which indicates that the heating cycle could be used to sufficiently expose the engine to higher temperature air.

However, the rise time to get to the maximum temperature after the fan was turned on was 41 minutes. This means that the system would need to be turned on nearly three-quarters of an hour before engine testing at this maximum temperature. This is longer than ideal for convenient testing.

Nonetheless, as noted above, testing at this temperature would occur rarely if at all. Reaching 37°C only took 13 minutes, which is more reasonable lead time.

Maintain Temperature at 37C

The ability of the heating mode to maintain a temperature of 37°C was tested by turning on all four heaters until the air temperature reached 37°C. Then the heaters were shut off. As the air temperature cooled, two heaters were turned back on. These heaters were then cycled on and off to maintain the temperature close the 37°C. After 30 minutes of testing, the temperature of the air remained between 37 and 38°C for the remaining 28 minutes of testing. This range falls within the goal of plus or minus 2 degrees. Figure 6.1.4 shows the temperature over time during this test.

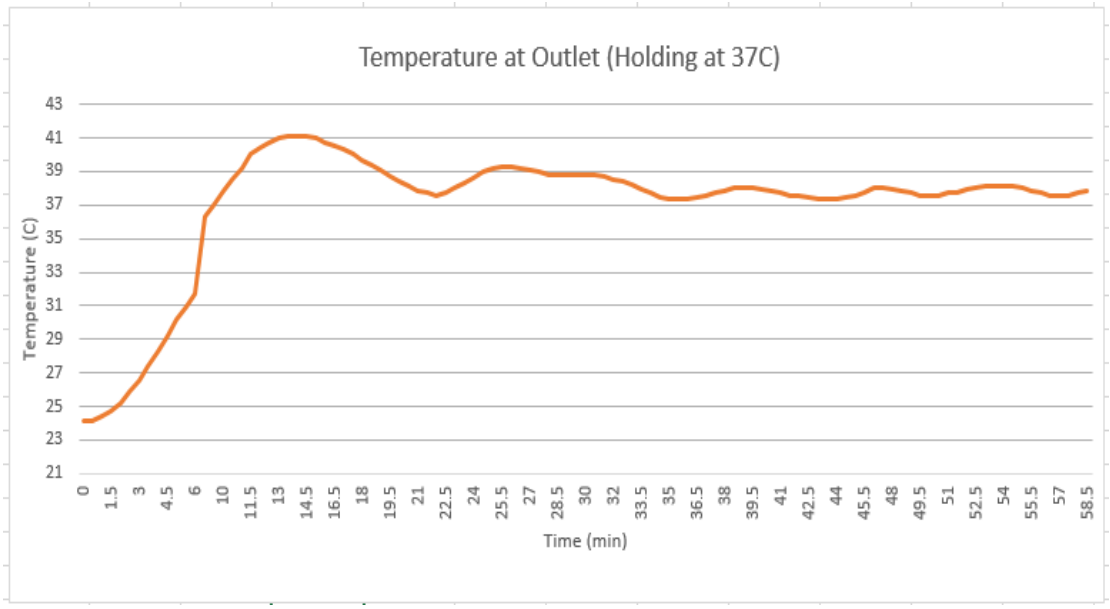


Figure 6.1.4: Temperature at outlet while holding at 37°C

This pattern was maintained by leaving two of the heaters on for 3.5 minutes and heating the ethylene-glycol mixture up to 51°. Then the two heaters were turned off for 2 minutes or until the temperature fell to approximately 49°.

The environmental control system's ability to maintain the temperature at 37°C exceeded expectation. By following a simple algorithm the temperature could be maintained within 1 degree of the target. The downside of this method is that it requires frequent cycling of the heaters. During engine testing, this manual cycling of the heaters would be extremely distracting.

However, this problem could be remedied with an electronic control that cycles the heaters on and off as the temperature of the working fluid reaches key specified values.

Maintain Temperature at 26°C

Because the cooling mode was not functioning, the test to maintain temperature at 26°C was cancelled. To run this test without cooling would not be valuable, as the room temperature air remains between 24° and 26°C naturally.

6.1.2 Humidifying Mode

Pure Humidifying Test
The system was tested with the humidifiers running at full power. The humidifiers were run for 10 minutes before the fan was turned on and air was forced through the duct. However, no change was detected in the humidity of the system.

Visual inspection indicated that the humidifiers were functioning to some degree, and water was being sprayed into the air above the humidifier chamber. However, it appeared to hover just above the waterline. Water did not appear to enter the duct itself. As previously noted, no change was detected in the humidity sensors at the air outlet.

Given the lack of results, additional testing was conducted with the ultrasonic humidifiers. The humidifiers were left on during heating tests to see if any measured change could be detected. Because the humidity sensor measures relative humidity, the readings changed as the temperature was heated up.

However, it can be shown that the humidity of the air remained unaffected by the ultrasonic humidifiers. Figure 6.1.5 shows the change in relative humidity recorded by the sensor alongside the calculated change in relative humidity assuming a constant humidity ratio.

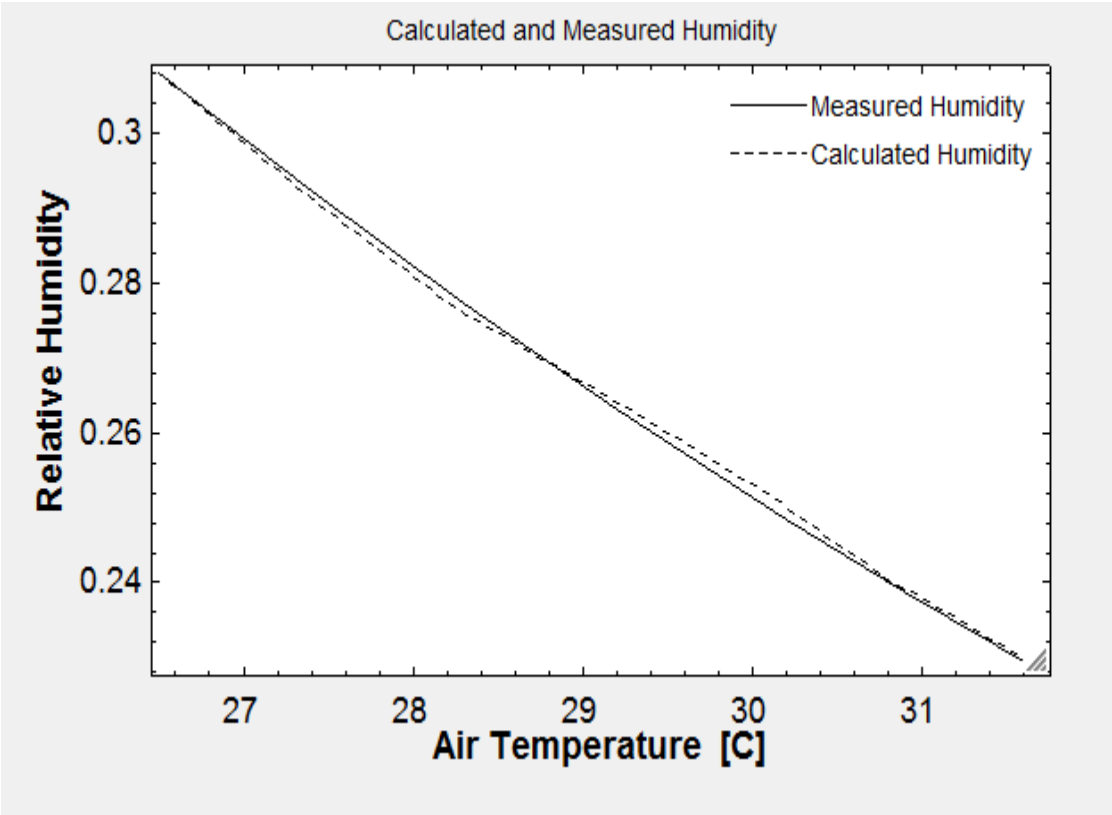


Figure 6.1.5: Calculated and Measured Humidity during humidifier testing

The lines nearly overlap, which makes it difficult to see. This indicates that the humidifier was not humidifying the air. If the humidifier was working properly, then the relative humidity of the measured air would be higher than the calculated value (which assumes a constant humidity ratio). The average difference between the two values was 0.00006436. This is an extremely small difference given the possible error due to precision of the humidity sensor. For clarity, the values were plotted against each other in Figure 6.1.6 to show how closely the values match.

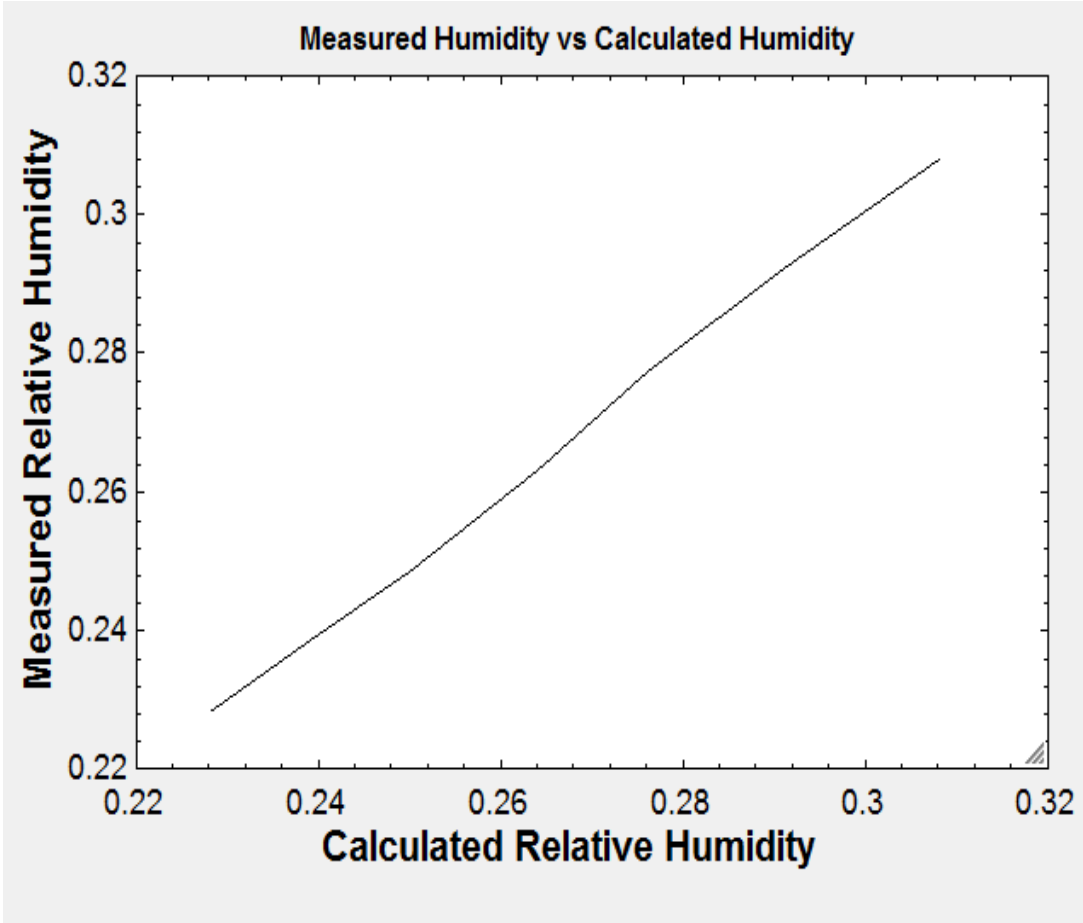


Figure 6.1.6: Measured Humidity vs. Calculated Humidity

As expected based on the above analysis, the result is nearly a linear relationship.

7. Results

7.1 Current State Testing Results

The biggest issue with the current state is the underperformance of the cooling mode. The model indicated that the system would not perform as desired, but the physical testing did not come close to the modelled results.

This is most likely due to the assumption that the cold side heat exchanger located in the freezer would remain at a constant temperature. Because the heat exchanger is located in a block of frozen ethylene-glycol, heat transfer will be reduced within the freezer. The model did not account for a lack of mixing within the freezer. If the ethylene-glycol remained in liquid form and was forced across the heat exchanger, then the heat transfer would likely be improved. While this would decrease the ultimate cooling capacity of the cycle, it would improve the temperature difference. In the current state, the cooling cycle is not useful for engine testing.

The heating cycle performs largely as expected. The temperature rose higher than predicted by the model, but this can be explained by the lack of cooling. The model assumed that adequate fooling would

be provided by the cooling cycle. Instead, the heating cycle was heating room temperature air. Regardless, the heating cycle performed acceptably and could nearly be used for engine testing.

The largest drawback of the heating cycle is the effort required to maintain a constant temperature. If an automatic control were used to monitor the working fluid temperature and cycle the heaters on and off appropriately, then the heating cycle could be used for engine testing at higher temperatures.

Additionally, while the heating cycle works well enough, if the system is to be completely redesigned, it may be beneficial to use electrical resistance heating directly in the ducting to eliminate the need for a second cycle with an additional heat exchanger, pump, and pipe network.

This method would remove the potential for reheating the cooled air during cold tests, and would allow for simpler on/off control of the heater based on outlet air temperature.

Finally, the humidifying stage of the environment control system was nonfunctional. While less important than the cooling stage, the humidifier is necessary for testing the engine at specified humidity. In all humidity testing, no detectable change in humidity was recorded. If humidity testing is to be conducted on the engine, this system would need a complete redesign.

7.2 Future State Concepts

The current state model was unable to meet expectations. In order for the system to be used for engine testing, significant improvements need to be made. Two potential paths forward were identified. The current system can be modified to make improvements in system performance, or the current system can be replaced with a more effective system. Recommendations are made for how to proceed in either direction.

7.2.1 Concept Selection

Cooling Mode

Freezer Modification

The heat exchanger inside the freezer could be replaced with a shell-and-tube heat exchanger, and the ethylene-glycol-mixture could be pumped through alongside a cold liquid-phase mixture within the freezer. This would improve heat transfer from the working fluid with minimal changes to the overall system.

The main advantage of this approach is that it would require few changes to the cooling cycle and avoid the purchase of many new components. However, the improvements to cooling capacity would likely be slight. Additionally, the freezer takes up significant space in the engine room and is prone to leaking.

Vapor Compression Refrigeration

Alternatively, the vapor compression cycle is a common form of refrigeration cycle which could be used to eliminate the need for the chest freezer as a thermal reservoir and improve the cooling performance of the environmental control system. Additionally, the vapor compression cycle is one of the most common refrigeration systems in use [28].

The existing system could be converted to a vapor compression cycle by removing the pump and freezer from the loop, replacing them with a condenser and compressor, and inserting an expansion valve into the line.

One big advantage to this approach is that future incoming GFR members would more easily understand the cycle because it is covered in required engineering courses. If any problems occurred, or adjustments

were required, there would be easy access to plenty of information about these types of cycles. Another advantage would be the removal of the freezer loop and increased control over the cooling cycle.

The disadvantage of this approach is that it would require the purchase of new components. In addition to the condenser, compressor, and expansion valve, the in duct heat exchanger may need to be replaced if the new system cannot be specified to use the current size heat exchanger.

Another potential concern is that the heat that is removed from the air in the duct would be dumped into the engine room. The duct would in turn pull that heated air inside for cooling. This would cause a loop of attempting to re-cool the heated air. However, during engine testing, the engine room exhaust system is usually powered on, which helps to remove excess heat and refill the room with air-conditioned air.

Heating Mode

Current State

Because the current heating cycle performed above requirements, it could be kept in the new system. This would save time and money. However, there are some disadvantages to retaining the old system.

First, as previously stated, the old system will partially reheat the air during cooling tests, even if the heating element is off. During long tests this effect should be neutralized as the fluid in the heat exchanger is cooled, but it could pose significant problems during maximum cooling tests and other short tests.

Secondly, it adds additional complexity to the system which makes it harder to control. Heating the fluid to heat the air adds an additional layer which makes electronic control of the system more difficult.

Finally, it is an inefficient method for heating the air, as unnecessary waste heat is generated and dumped into the engine room.

Resistive Heating

Alternatively, the current system could be removed, and resistance heating strips could be installed inside the ducting. By placing heating elements directly in the ducts, it would ensure that the majority of the heat goes directly to the air. Additionally, the heating elements could be cycled on and off remotely based on temperature sensors at the air outlet.

This would be an improvement on the current system, because it would remove the excess piping, fluid, and pump required to run the heating cycle; it would be more easily controlled during testing; and it would reject less waste heat to the room.

One major disadvantage to this approach is the need to open the ducting to remove and replace components. This would be a time consuming task. However, the duct will likely need to be opened to adjust the cooling cycle, regardless of changes to the heating cycle. There would also be added cost from the purchase of new heating elements.

Pressure Control

Throttle Valve

Use of an electronic throttling valve at the opening of the duct can be used to control the pressure drop through the ducting. The valve can be set to automatically open or close more as the pressure drops or rises at the air outlet. The throttle valve can access pressure readings from a sensor placed at the air outlet to determine when to make adjustments.

The throttling valve will reduce pressure downstream without significantly altering the specific kinetic energy of the flow [47]. This means that the throttling valve can be used to control pressure without significantly changing the flow or thermodynamics through the remainder of the ducting.

This will be a significant improvement on the current system's "throttle door" which was not useable.

7.3 Vapor Compression Design Specification

Proceeding with the vapor compression cycle, it is necessary to fully specify the system. Figure 7.3.1 shows the system schematic with all components.

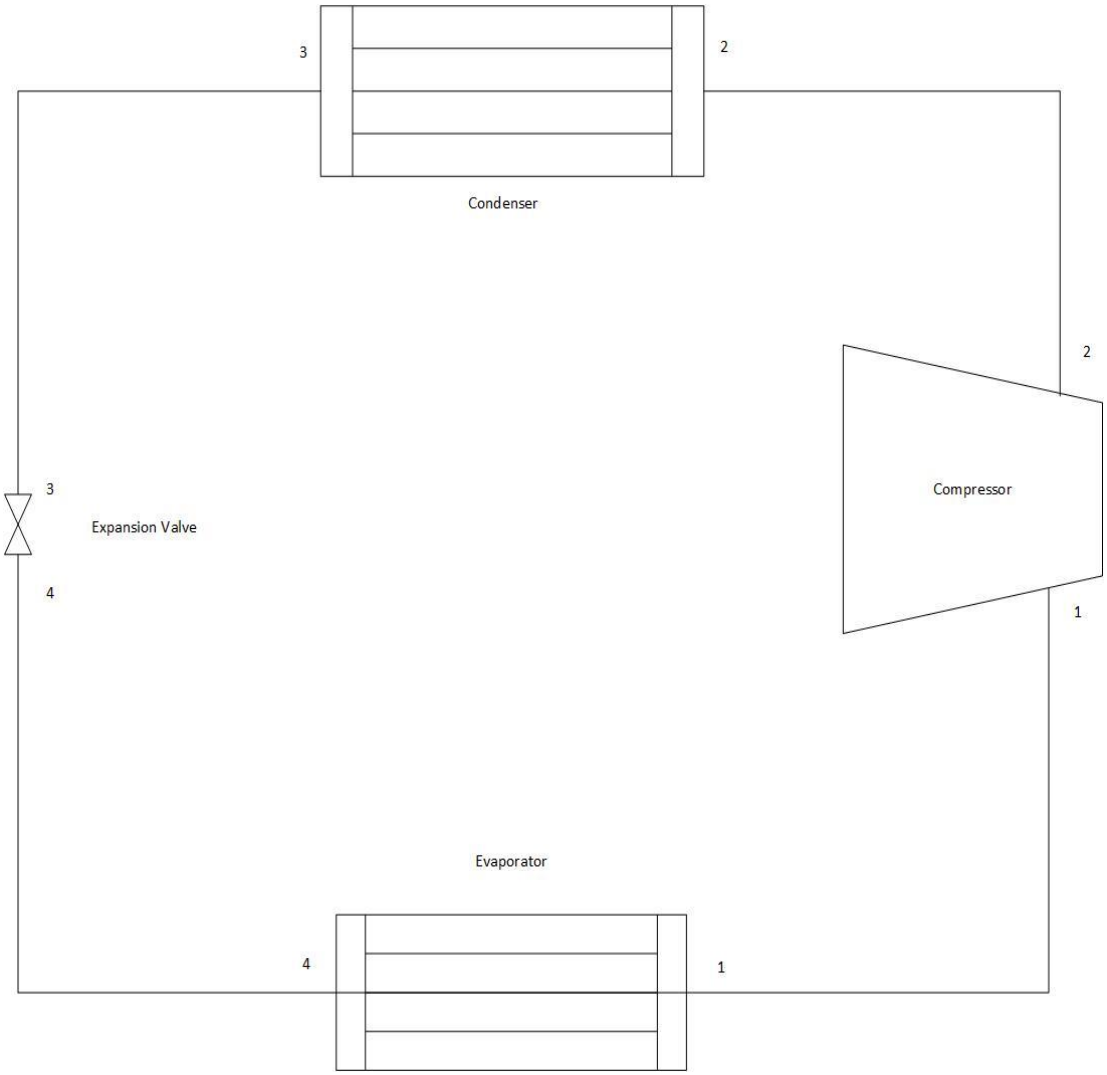


Figure 7.3.1: System Schematic of Vapor Compression Cycle

As Figure 7.3.1 indicates, the system will be comprised of a compressor, condenser, expansion valve, and an evaporator. State point 1 is the compressor inlet, state point 2 is the condenser inlet, state point 3 is the expansion valve inlet, and state point 4 is the evaporator inlet.

First, the system requirements were identified. They are included below in Table 7.3.1.

Table 7.3.1: Design Requirements

Air Temperature		Mass Flow	Pressure
Inlet	Outlet		
25 [C]	5[C]	0.05733 [kg/s]	101 [kPa]

The following parameters need to be determined in order to select components:

- Required Compressor Power
- Required Heat Exchanger Area
- Refrigerant Used
- Pipe Diameter
- Pipe Material

7.3.1 Thermodynamic Cycle Analysis

Procedure

The thermodynamic cycle was analyzed using EES and the system requirements listed above. Additionally, a low side evaporator temperature of 3°C was chosen. The high side condenser temperature was set to vary based on the working fluid to ensure that it remained lower than the working fluid's critical temperature.

The rate of heat leaving the air was calculated from the steady state energy equation.

$$0 = \dot{Q} + \Sigma m_i h_i - \Sigma m_e h_e \qquad \text{(eq 7.3.1)}$$

Thermodynamic properties were determined at each state, assuming an adiabatic compressor, ideal heat exchangers, and an isenthalpic expansion valve. The properties are listed by state point in Table 7.3.1.

Table 7.3.1: State Point Table

State Point	T [C]	P [kPa]	h [kJ/kg]	s [kJ/kg-K]	x
1	3	326.2	252.2	0.9298	1
2	76.04	1806	296.5	0.9554	100
3	63.01	1806	144.3	0.5036	0
4	3	326.2	144.3	0.5391	0.4504

Once the thermodynamic properties were determined at each state, the mass flow rate of the refrigerant was calculated using the steady state energy equation and the calculated heat transfer rate.

A parametric table was then generated to compare different potential refrigerants. The following refrigerants were considered:

- R22
- R290
- R134a
- Butene
- Water

Results

The selection of refrigerant was based on maximum pressure, maximum temperature, and required mass flow rate. Additionally, cost, safety, and availability were considered. Figure 7.3.2 shows the pressure comparison for the five fluids.

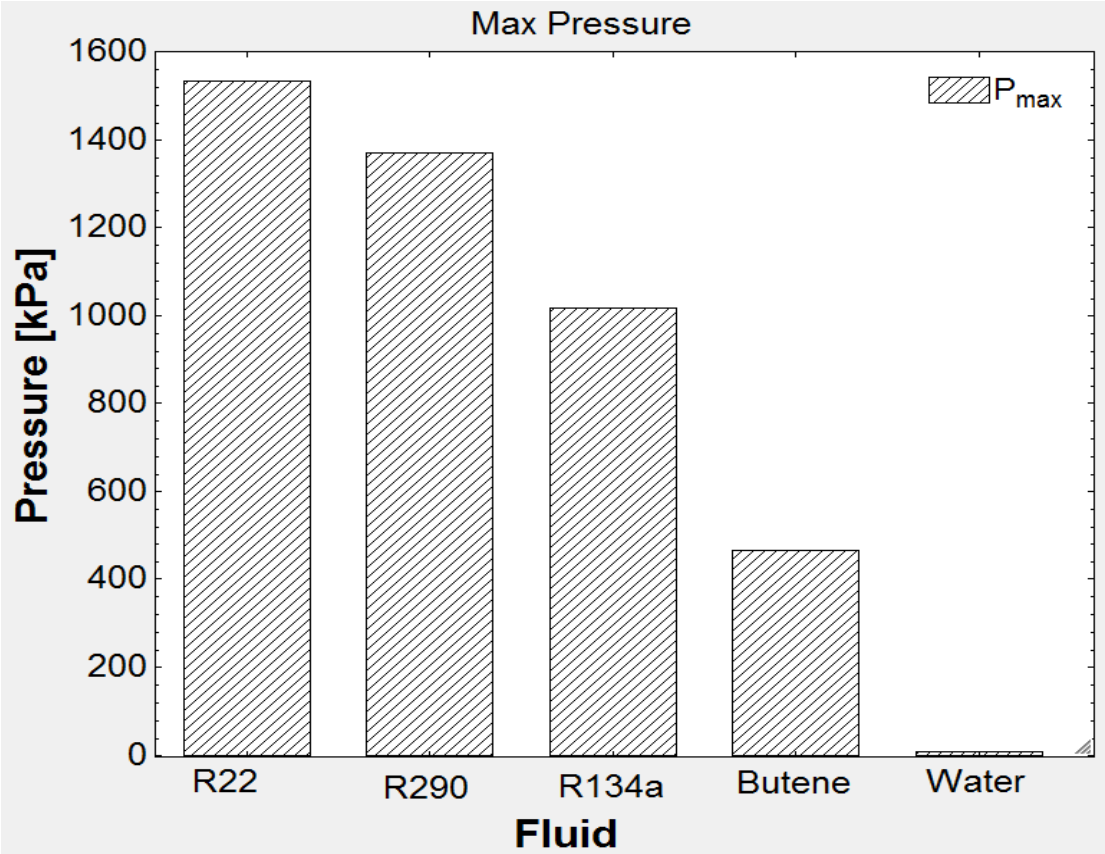


Figure 7.3.2: Maximum Pressure Comparison

The graph clearly shows that R22 results in the highest pressure difference. Water results in the lowest pressure difference of less than 10 kPa. The ambient pressure in the engine room is roughly atmospheric, which means that the water would need to be run at vacuum pressure. R134a falls in the middle and has a max pressure of about 1 MPa. Figure 7.3.3 shows the temperature comparison for the five fluids.

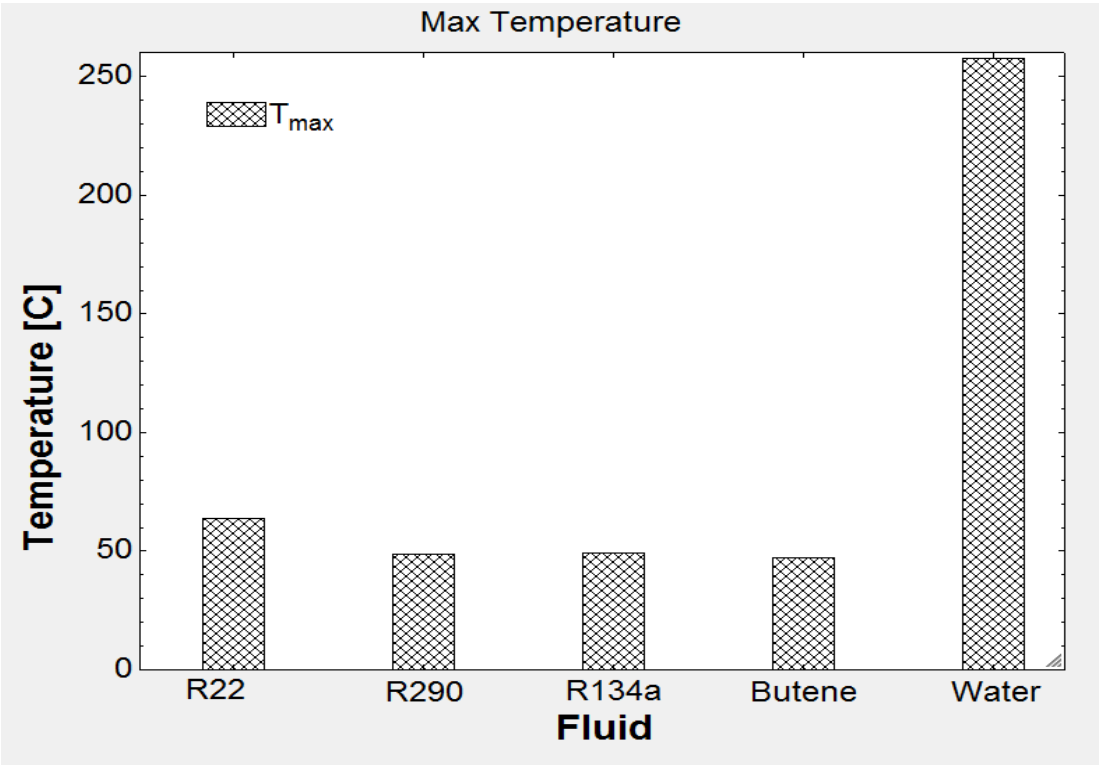


Figure 7.3.3: Maximum Temperature Comparison

As can be seen on the graph, water has a significantly higher operating temperature of over 250°C. The other four fluids are nearly 200 degrees cooler, and all are within 15 degrees of each other. R134a is again in the middle with a hot temperature of 49°C.

Finally, Figure 7.3.4 shows the mass flow rate comparison for the five fluids.

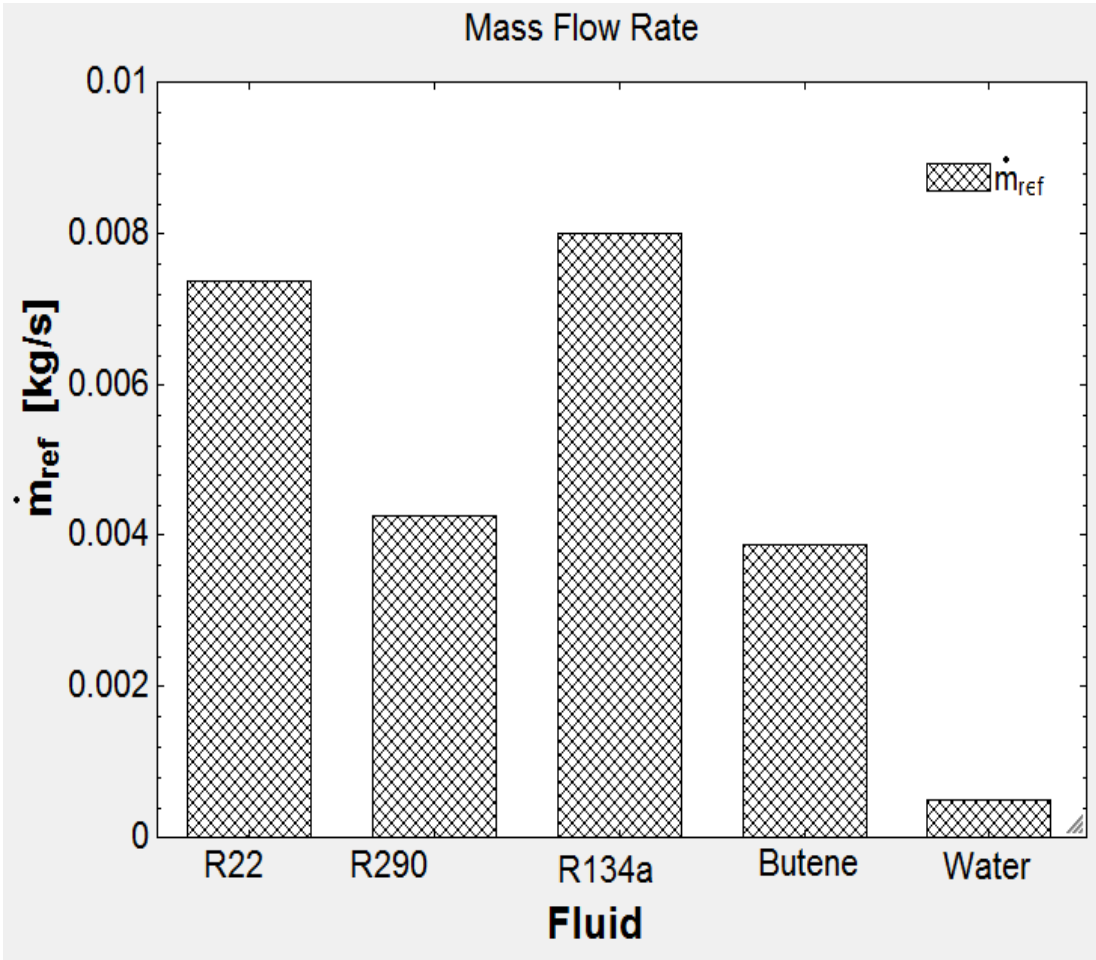


Figure 7.3.4: Mass Flow Rate Comparison

The graph shows that water has a significantly lower mass flow rate required. R134a has the highest flow rate, nearly 17 times as fast as water.

Selection

R134a was chosen as the working fluid because of its thermal properties as well as its availability. Water is even more readily available and had the lowest pressure and flow rate. However, the need vacuum pressure would prove problematic, and the high temperature would be undesirable. Butene had the next lowest flow rate and pressure. However, Butene is highly flammable. Therefore, R134a was chosen. Despite its lower performance, it is a safer option that will be adequate for this system. Additionally, it is a common refrigerant that will be available for use.

7.3.2 Hydraulic Loop

The hydraulic loop was analyzed with EES based on the calculations done in the thermodynamic cycle analysis. The analysis was conducted to determine the pipe diameter and material.

First the liquid line diameter was varied while the vapor line diameter was held constant. Figure 7.3.5 shows the results.

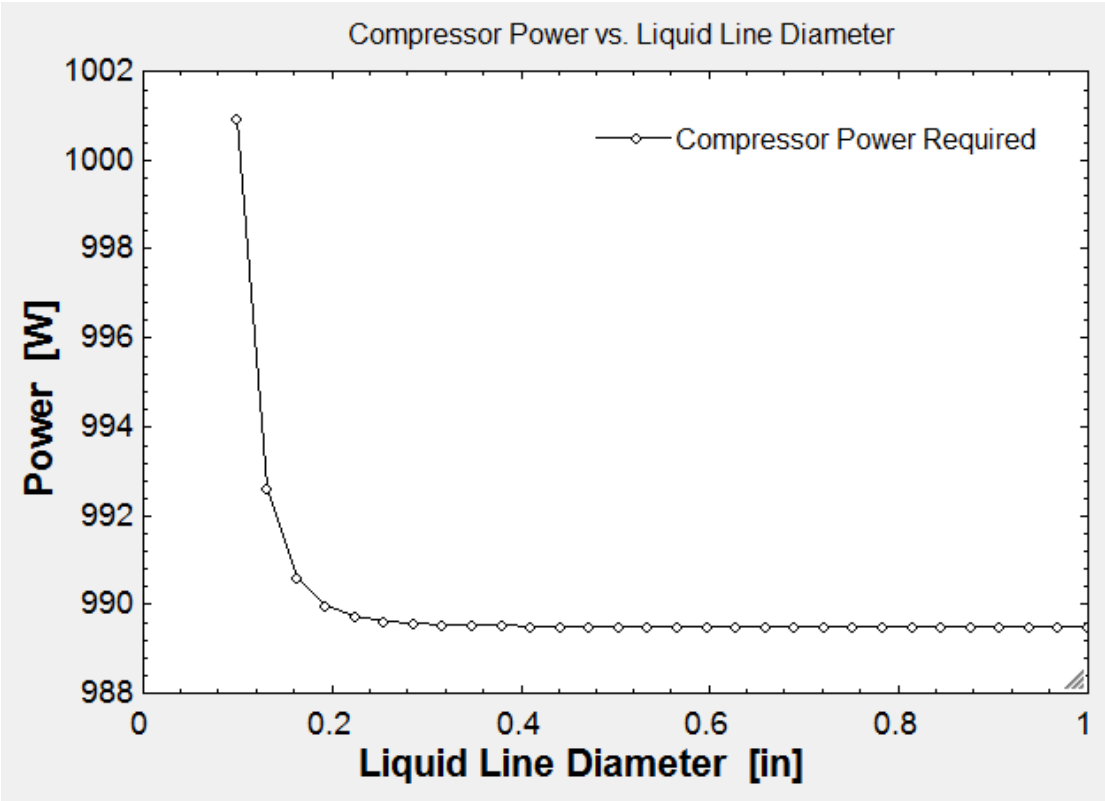


Figure 7.3.5: Compressor Power vs. Liquid Line Diameter

As the graph indicates, there is initially a sharp decrease in compressor power for increase in pipe diameter. However, after 0.2-0.3 inches, the decrease gets much smaller. Figure 7.3.6 shows a similar relationship between vapor line size and compressor power.

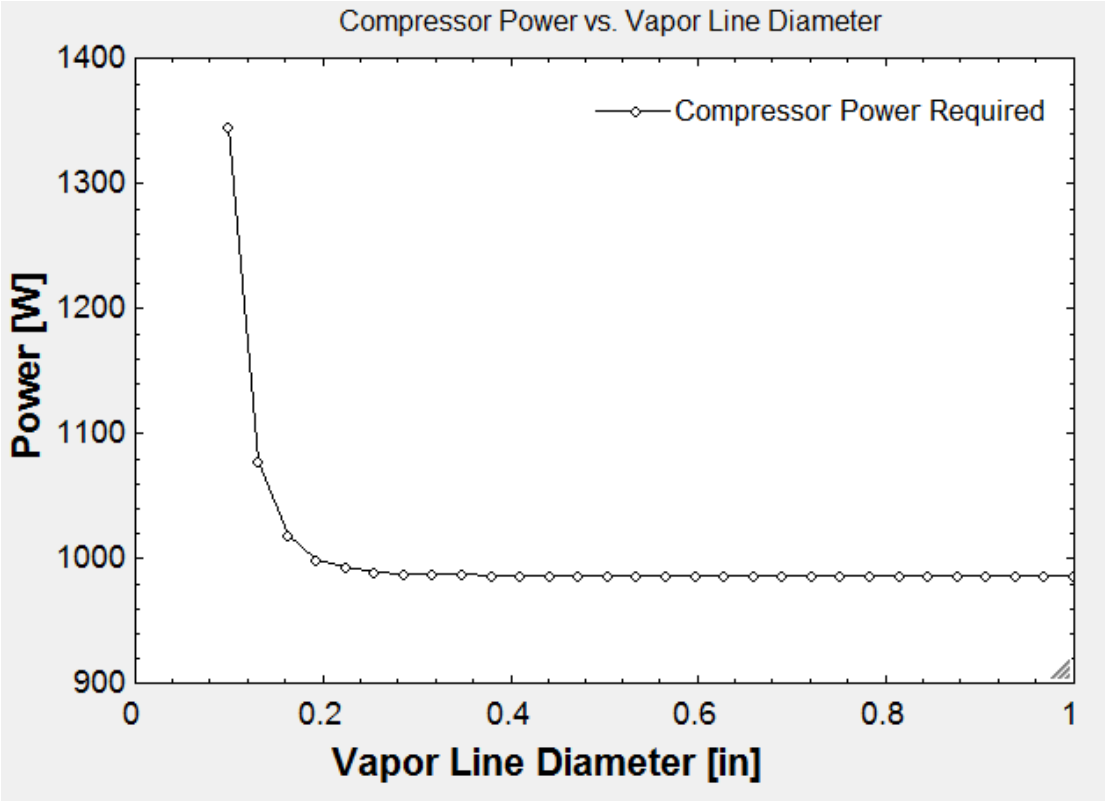


Figure 7.3.6: Compressor Power vs. Vapor Line Diameter

Again, there is a quick decline between 0.1 and 0.2 inches, but then the gains from increasing diameter become very small for changes in pipe diameter. Based on the above graphs, a uniform size of quarter inch pipe was selected for both the liquid and vapor lines.

Next the material was analyzed. Table 7.3.2 lists the absolute roughness used in the model for both Copper and PEX pipes [48]. Figure 7.3.7 shows the respective compressor work for each.

Table 7.3.2: Absolute Roughness

Material	Absolute Roughness [mm]
Copper	0.0015
PEX	0.007

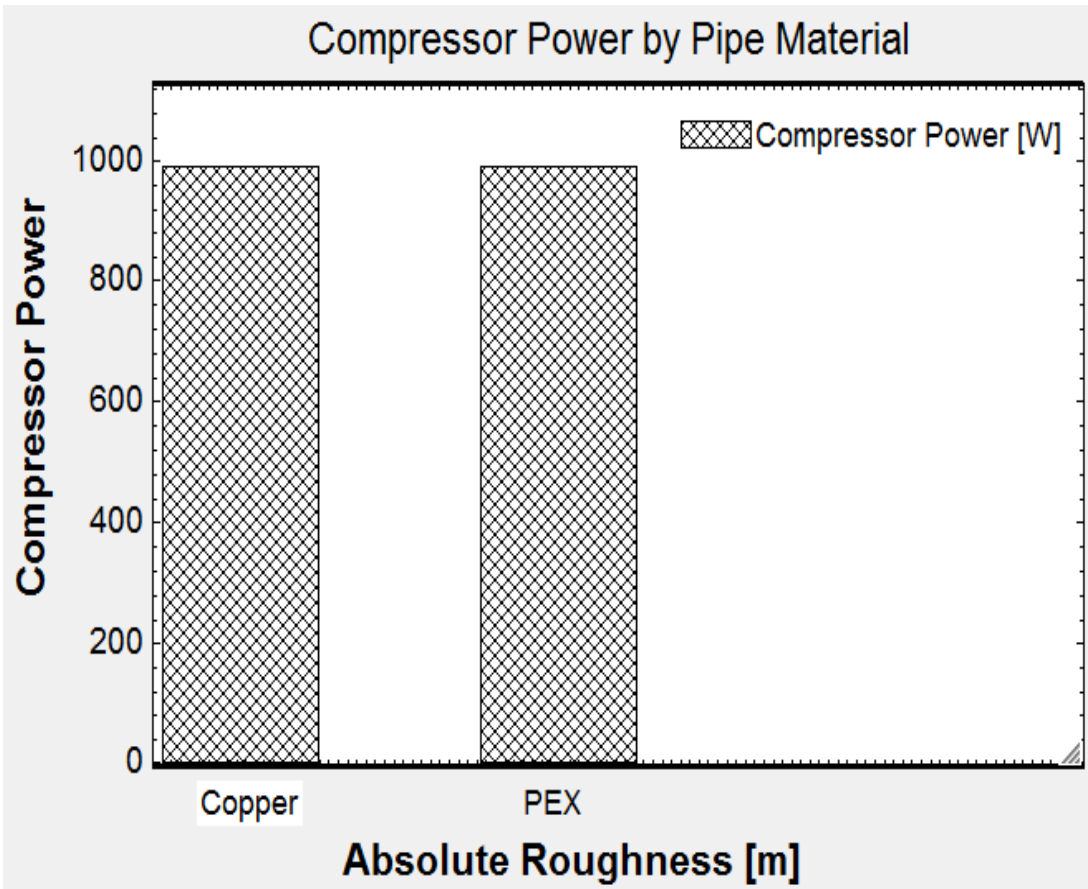


Figure 7.3.7: Compressor Power vs. Material

The two materials are very close in terms of performance, and there is no practical difference between the two materials for this system, in terms of compressor work required.

Selection

PEX was chosen to be the pipe material. While the two materials had similar roughness, PEX is cheaper and more flexible for GFR's needs, as noted in Section 2.2. Additionally, GFR team members requested that PEX be used if possible.

Quarter inch pipe diameter was selected for both the liquid and vapor lines based on the hydraulic model results. Increasing the diameter would increase the cost and do little to reduce the size of the compressor.

Based on these design decisions, the model shows that 1.326 HP of power is necessary to drive the cycle. Therefore, a 1.4 or 1.5 HP compressor will be used in the new system.

7.3.1 Heat Exchanger Sizing

The goal of the heat exchanger analysis and sizing is to determine if the existing heat exchangers could be reused in the new system. Therefore, the heat exchanger was chosen to be a cross flow heat exchanger and was modeled similarly to current system. Table 7.3.3 shows the relevant values for the heat exchanger analysis.

Table 7.3.3: Heat Exchanger Analysis

Type	Crossflow, unmixed	Crossflow, Unmixed
Inner/Outer	Inside	Outside
O.D.	0.035 [m]	0.035 [m]
Thickness	0.005 [m]	--
I.D.	0.03 [m]	--
Flow Area	0.000707 [m^2]	--
Fluid	R134a	Air
Temperature In	3 [C]	25 [C]
Pressure In	326.2 [kPa]	101 [kPa]
Mass flow rate	0.01067 [kg/s]	0.05733 [kg/s]
Volumetric Flow	0.001 [m^3/s]	0.04858 [m^3/s]
Quality In	0.4504	--
Re	344300	50870
Pr	2.567	0.7305
Nu	5689	149.5
U	105.2 [W/m^2-K]	105.2 [W/m^2-K]
UA	92.62 [W/K]	92.62 [W/K]
Q	1151 [W]	1151 [W]
LMTD	12.43 [K]	12.43 [K]
Total Area	0.8808 [m^2]	0.8808 [m^2]

The current heat exchangers have a total heat transfer surface area of 0.91 square meters [12]. Therefore, in order to save the expense of buying new heat exchangers, the current ones can be used in the new system.

7.4 Future State Recommendations

For the future system, Table 7.4.1 summarizes the recommended components to be used.

Table 7.4.1

Compressor Power	1.5 [HP]
Heat Exchanger Size	0.88 [m^2] (Use current heat exchangers)
Pipe Diameter	0.25 [in]
Pipe Material	PEX
Refrigerant	R134a
Heating Element Max Power	2 [kW]

8. Conclusion

This project started out with the goal of repairing the environmental control system, and testing the engine under different environmental conditions. However, the current environmental control system proved incapable of achieving the desired conditions. It took more time to get the system running than was expected, and the system was unable to cool effectively once it was operational.

Testing of the system occurred much later than desired, which left little time to make adjustments. The scope of the project had to be changed to focus on developing a new system rather than using the old system. The idea of engine testing had to be completely eliminated from the project.

While these setbacks were not optimal, they will ultimately be better for GFR going forward. It is important that GFR has a good system which can control the air conditions consistently and effectively.

If more time were available, the next steps would be to acquire components and build the vapor compression system. Then the unit would be tested under the same conditions as the previous system. Once the capabilities of the system are verified, the system can be used for engine testing as was previously intended.

9. Works Cited

[1] Global Formula Racing. "*Introduction Information.*" Formula Racing Wiki. Private.

[2] Global Formula Racing. "*Static Events.*" Global Formula Racing Wiki. Private.

[3] Patton, Chris. "*Design Event.*" Global Formula Racing Wiki. Private.

[4] Global Formula Racing. "*Cost Event.*" Global Formula Racing Wiki. Private.

[5] Global Formula Racing. "*Presentation Event.*" Global Formula Racing Wiki. Private.

[6] Global Formula Racing. "*Dynamic Event.*" Global Formula Racing Wiki. Private.

[7] Global Formula Racing. "*Endurance Event.*" Global Formula Racing Wiki. Private.

[8] Takaro, Trevor. "*Autocross Event.*" Global Formula Racing Wiki. Private.

[9] Takaro, Trevor. "*Skidpad Event.*" Global Formula Racing Wiki. Private.

[10] Takaro, Trevor. "*Acceleration Event.*" Global Formula Racing Wiki. Private.

[11] Global Formula Racing. *Global Formula Racing Philosophy.* Global Formula Racing Wiki. Private.

[12] Stalp, Calvin (2015). *GFR 15 Engine Dynamometer Testing Report.* Global Formula Racing Home. Private.

[13] SuperFlow. "*SF-901 Series: Engine Dynamometer Systems.*" Retrieved from http://www.dynohelp.com/resources/SF901_Brochure_Emailing.pdf

[14] Martyr, A. J.; Plint, M A. "*Engine Testing: Theory and Practice.*" 3rd ed. Kidlington: Elsevier Science, 2011.

[15] Global Formula Racing. "*EFI Basics*." Global Formula Racing Wiki. Private.

[16] Global Formula Racing. "*Sensors*." Global Formula Racing Wiki. Private.

[17] Peter, John. "*Testing Investment*." Automotive Industries. 10994130, Oct 2002, Vol. 182, Issue 10

[18] I. Watanabe and H. Kuroda, "*Effect of atmospheric temperature on the power output of a two-stroke cycle crankcase compression gasoline engine*" SAE International Congress and Exposition, 23-27 February 1981.

[19] MoTeC. "*ECU Tuning and ECU Manager Setup*." Formula Seminar Series. April 2010.

[20] Husaboe, T. "*Effects of Temperature on the Performance of a Small Internal Combustion Engine at Altitude.*" Air Force Institute of Technology. March 2013

[21] CIMAC. "*About the Influence of Ambient Conditions on Performance of Gas Engines.*" International Council on Combustion Engines. March 2009.

[22] Gardiner, A. "*Atmospheric Humidity and Engine Performance*." SAE Technical Paper 290033, 1929, doi:10.4271/290033.

[23] MoTeC. "*Engine Management & Data Acquisition Systems.*" Formula SAE Seminar Series. May 2004.

[24] Takaro, Trevor. "*Equating design parameters to points.*" Global Formula Racing Wiki. Private.

[25] Weather2. *Historical Monthly Averages for August - A1 Ring Circuit.* retrieved from http://www.myweather2.com/Motor-Racing/Austria/A1-Ring-Circuit/climate-profile.aspx?month=8

[26] Weather2. *Historical Monthly Averages for July - Hockenheimring Circuit.* retrieved from http://www.myweather2.com/Motor-Racing/Germany/ Hockenheimring-Circuit/ climate-profile.aspx?month=7

[27] Weather2. *Historical Monthly Averages for May - Michigan International Speedway.* retrieved from http://www.myweather2.com/Motor-Racing/ United-States-Of-America/Michigan-International-Speedway/ climate-profile.aspx?month=5

[28] Moran, Michael; Shapiro, Howard, *Fundamentals of Engineering Thermodynamics.* 6th ed. Hoboken, NJ: John Wiley & Sons, Inc, 2008

[29] Martyr, A. J., and M. A. Plint. *Engine Testing : The Design, Building, Modification and Use of Powertrain Test Facilities*. 4th ed. Saint Louis: Elsevier Science, 2012.

[30] Leskinen, A., P. Yli-Pirilä, K. Kuuspalo, O. Sippula, P. Jalava, M.-R. Hirvonen, J. Jokiniemi, A. Virtanen, M. Komppula, and K. E. J. Lehtinen. "Characterization and Testing of a New Environmental Chamber." *Atmospheric Measurement Techniques* 8, no. 6 (June 3, 2015): 2267–78. doi:10.5194/amt-8-2267-2015.

[31] Porges, F. *Handbook of Heating, Ventilating and Air Conditioning*. 8th ed. Kent: Elsevier Science, 2013.

[32] Hundy, G. H., A. R. Trott, and T. C. Welch. *Refrigeration and Air-Conditioning*. 4th ed. Jordan Hill: Elsevier Science, 2008.

[33] Daly, Steven. *Automotive Air Conditioning and Climate Control Systems*. Oxford: Elsevier Science, 2011.

[34] Yan, Jia, Wenjian Cai, Lei Zhao, Yanzhong Li, and Chen Lin. "Performance Evaluation of a Combined Ejector-Vapor Compression Cycle." *Renewable Energy* 55 (July 2013): 331–37. doi:10.1016/j.renene.2012.12.029.

[35] Yang, Mina, Chung Woo Jung, and Yong Tae Kang. "Development of High Efficiency Cycles for Domestic Refrigerator-Freezer Application." *Energy* 93, Part 2 (December 15, 2015): 2258–66. doi:10.1016/j.energy.2015.10.127.

[36] Chugger Pumps. "*CPPS-IN-1 General Specification*." retrieved from http://www.chuggerpumps.com/home-brewing-pumps-chugger/chugger-pump-products/item/cpps-in-1/

[37] IGLOO. *Manual Defrost Chest Freezer.* retrieved from http://www.manualslib.com/manual/282180/Igloo-Frf434.html#manual

[38] Kat's Heaters (2014). *2014 Application Guide.* retrieved from http://www.fivestarmanufacturing.com/getdoc/2c8a6911-c214-4190-9059-d6aaa10fe527/dc12

[39] House of Hydro. *Twelve Disk Mist Maker.* retrieved from http://www.thehouseofhydro.com/twelve-head-mist-maker.html

[40] Engineering Toolbox. "*Ethylene Glycol Heat-Transfer Fluid."* retrieved from http://www.engineeringtoolbox.com/ethylene-glycol-d_146.html

[41] Engineering Toolbox. "*Propylene Glycol based Heat-Transfer Fluids."* retrieved from http://www.engineeringtoolbox.com/propylene-glycol-d_363.html

[42] Green Building Advisor. *Sealing Ducts: What's Better, Tape or Mastic?* retrieved from http://www.greenbuildingadvisor.com/blogs/dept/musings/ sealing-ducts-what-s-better-tape-or-mastic

[43] Penoncello, Steven, *Thermal Energy Systems: Design and Analysis.* Boca Raton, Florida: CRC Press, 2015

[44] Janna, William. *Introduction to Fluid Mechanics.* 3rd ed. Boston: PWS-KENT, 1993

[45] White, Frank. *Fluid Mechanics*. 7th ed. New York: McGraw-Hill, 2011

[46] Bergman, T., Lavine; A. Incropera, F.; Dewitt, D., *Fundamentals of Heat and Mass Transfer.* 7th ed. Hoboken, NJ: John Wiley & Sons, Inc, 2011

[47] Kaminski, D.; Jensen, M., *Introduction to Thermal and Fluids Engineering.* Hoboken, NJ: John Wiley & Sons, Inc, 2005

[48] Engineering Toolbox. "*Ventilation Ducts - Roughness & Surface Coefficients."* retrieved from http://www.engineeringtoolbox.com/surface-roughness-ventilation-ducts-d_209.html