

# Final Report

Team #2

Decreasing Electricity Usage at Cummins Technical Center



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## **ABSTRACT**

Team 2 has written the following report to summarize the complete energy savings plan to reduce the energy consumption at the Cummins Technical Center (CTC) in Columbus, Indiana. The team has proposed a variety of ideas, and has primarily focused on a detailed design of a waste heat recovery system and a simulation for this system, which are presented within the report. In order to validate the waste heat recovery selection, the team has created a design and simulation of the results for an Organic Rankine Cycle (ORC) with solar thermal collector pre-heaters. This simulation uses a Guided User Interface (GUI) integrated with the MATLAB analysis for the system that allows the user to alter different inputs and outputs of the system in order to see how the system performs throughout the year. From this simulation, Cummins, Inc. can view the cost and efficiency of the system, as well as the potential amount of savings by implementing Team 2's design in the facility. According to the energy analysis conducted, harnessing these exhaust gases will save the CTC just under 1.15 million kWh of heat energy savings and equates to a net savings of \$97,519 in annual savings after accounting for the annual cost of maintenance. The ORC system design is estimated to save the facility 1-2% of the total energy consumed. The payoff factor was calculated to be 18.9 years, with the designed ORC system efficiency ranging between 8-10%. Additionally, through the implementation of all the ideas proposed, the CTC has a projected \$466,401 of annual savings per year, resulting in a reduction of the total energy consumed by 4.35%. Conclusively, Team 2 proposes that Cummins, Inc. should greatly consider creating an ORC waste heat recovery system after additional developments in the technology in the next 3-5 years help reduce the costs of implementation, and harness more of the energy from the exhaust gases.

## **ACKNOWLEDGEMENTS**

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# 1. Introduction

Energy efficiency is a major area of concern for many businesses. Whether it is inefficient light bulbs, poor insulation, or heat leaks from piping; wasteful energy practices cost companies hundreds of thousands of dollars each year. Companies have implemented different energy savings techniques that help reduce their energy consumption. Some techniques are replacing out dated light bulbs with more efficient ones, installing motion sensor triggers onto a lighting system and upgrading major energy consumption units to more efficient units. These practices are also known as an energy audit where a company's energy usage is analyzed and energy savings ideas are developed. Cummins has made great strides to become an industry leader in recyclable energy practices that minimize impact on the environment. In 2011, Cummins began to supplement some of its power sources in certain locations with solar panels and rerouting energy developed from their test cell dynamometers back into the grid. Also, Cummins has had an energy audit conducted to help assist their initial energy usage.

Cummins Inc. wanted a further reduction in their energy consumption and has chosen to sponsor a senior design team from Florida State University to reduce their energy consumption by 10%. In order to reduce the CTC energy, team 2 has been assembled to generate the necessary ideas, conduct the required analysis and produce an energy savings plan. Team 2 focused in multiple energy reduction areas for the CTC and created different ideas. Through the use of a decision matrix, the main areas of improvement that team 2 has decided are as followed: installing a solar panel design, installing absorption chillers (AVS), rerouting engine exhaust into a power generation cycle, improving dynamometer efficiency and applying a modern building insulation. These are ideas are analyzed in terms of energy saved per year and total annual dollars saved per year.

After Team 2 visited the CTC in Columbus Indiana in November 14<sup>th</sup> 2014, the realization that exhaust gases from the testing engines were used for emission testing only; team 2 has converted their focus to utilize the hot temperature of the exhaust gas (~400°F) for power generation.

Since the beginning of the spring semester team 2 has focused on designing an organic Rankine cycle powered by the exhaust gases. The purpose of focusing on one idea was to get a maximum amount of energy from a well-designed system. While team 2 hit several roadblocks through the

semester they were able to code a simulation that demonstrated the amount of power that can be generated from an organic Rankine cycle.

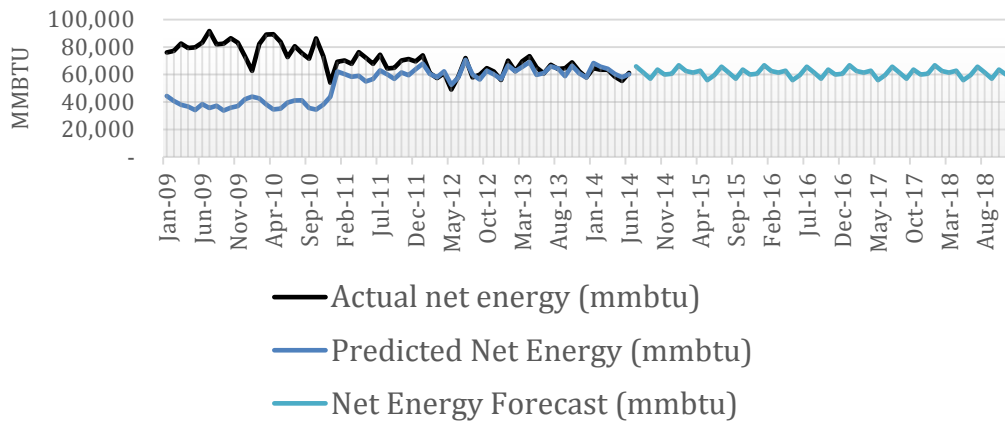
## **2. Background Research**

Cummins Inc. has embraced through the years “green” practices, such as, remanufacturing operations that keep fifty millions pounds of Cummins products out of landfills. Cummins Inc. has been leading the path to more fuel efficient engines for commercial trucks by helping to develop the first greenhouse gas emission regulations for commercial trucks in the United States. Cummins Inc. has also been awarded the first engine that meets these regulations [1]. Team 2 helped Cummins achieve their goal to become “greener” and reduce their environmental footprint. The first semester of the project, the team conducted an energy audit on the Cummins Technical Center (CTC) building in Columbus, IN. The energy audit was based on practices in the industry such as changing insulation in the roof and in piping, review lighting practices, implementing solar panels in the roof, and changing chillers in the building.

### *2.1. Background Research*

Since 2011, Cummins has implemented solar cells as an alternative means of power in several of their facilities. Presently, the company has five solar arrays capable of generating up to 230kW. While this is an impressive feat, Cummins is looking to further capitalize on cleaner, more efficient sources of energy. For clarification, many companies and engineers use the notation MMBTU to represent one million BTU (British Thermal Units), and it should be noted that 1 BTU is about 1,055 Joules of energy. Cummins has provided the team with the data shown below in Figure 1 to depict the total energy use at the Cummins Technical Center (CTC), which shows that they have already made significant strides in lowering their total energy use, and this value is predicted to be around 60,000 MMBTU for the next four years [2].

## CTC Total Energy Use and Forecast



**Figure 1:** CTC Total Energy Usage and Predicated Usage (2009-2018)

Team 2 is fortunate to be from Florida State University (FSU) because of the FSU’s Off-Grid Zero Emission Building (OGZEB). The OGZEB is a prototype used to test the feasibility of solar power in residential buildings. It consists of several rooms that are powered through solar energy, with the use of hydrogen powered appliances, high efficiency lighting such as sky lights, and recycled material insulation practices. The OGZEB building also has several power and temperature monitoring systems that analyze the conditions inside and outside the building. Though this is a residential application, the theory and techniques can be applied to an industrial setting. By utilizing the OGZEB, Team 2 will have the ability to do the necessary research and have experience with alternative energy power generation methods [3]. These methods proved to be useful in the design of the solar panel design which is presented later in the report.

The next subsections of the background research delve into aspects of energy audits, solar panels, insulation, engine testing, absorption chillers (AVS) and exhaust waste heat recovery.

### 2.1.1. Energy Audit

An energy audit is the practice of surveying a facility in order to identify different opportunities to reduce energy consumption and optimize the efficiency of energy usage [4]. Energy management is critical and consists of organizing financial and technical resources and personnel to maximize energy efficiency. Energy management also consists in keeping and maintaining records of energy usage and the performance of all machinery or appliances. An appropriate record keeping method can help to determine the areas in which the efficiency is under the standards of the facility. Depending on the building, the approaches to improve the

efficiency of the energy usage can vary, and are often subject to the areas that consume the most energy. In most cases, the HVAC (heating, ventilating and air conditioning) is the source of major energy consumption in manufacturing facilities mainly due to heat generation from machinery and processes.

### *2.1.2. Solar Panels/Solar Thermal Collectors*

A solar panel, or module, is a series of interconnected silicon cells joined together to form a circuit. In large numbers, the amount of power produced by these interconnected cells can be increased and used as an electricity production system [5]. For this project, the main focus will be analyzing the feasibility of implementing photovoltaic cells on the roof of the technical center for energy generation. Photovoltaics cells are designed to supply energy to commercial or residential buildings. The efficiency of these cells can go up to 19% and have a life time of 25 years. These cells need to be in arrays because each cell can only produce a limited amount of energy. Each package of solar panels may include the cell, a DC/AC converter, a solar tracker, and a battery. The theory behind the energy generation of these cells is the photovoltaic effect which consists of converting the energy of the Sun coming to Earth in the form of photons to electrical energy. This is done when the photons interact with the material's surface on the cells and excite the valance electrons, breaking them free and making them jump to the conduction band enabling the use of the electrical energy. The movement of electrons will produce a current and through ohm's law, if the current is supplied across a resistor, a voltage can be developed.

### *2.1.3. Insulation*

Thermal insulation refers to the reduction of heat transfer between objects that are in thermal interaction. This can be accomplished by different ways from a specific process to the implementation of a different material. The capacity to insulate a material is measured in thermal conductivity (k). This property is proportional to the heat transfer. In other words the lower the thermal conductivity, the lower the heat transfer will be across the material. Most insulators work by slowing down the heat transfer from the exterior to the building or vice versa. This can be done by increasing the thickness of the insulator, changing the insulator material, or adding reflective barriers to decrease the radiation coming from the outside to the building. The thermal conductivity (k) is measured in watts per meter kelvin. As the thickness of the insulator material increases, the heat transfer decreases because thermal resistance increases. The main application for insulators is present in buildings. In order to maintain an acceptable temperature in a building

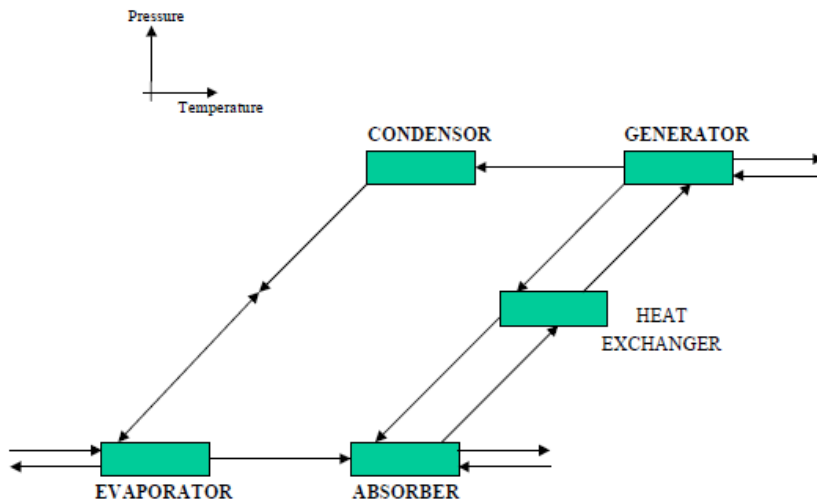
by heating or cooling, a large portion of the total energy consumption is needed. When a building is well insulated, the owner can see major savings in energy usage. Good insulation helps to provide a more uniform temperature throughout the building reducing the temperature gradient in all directions [6].

#### *2.1.4. Engine Testing*

To measure the power of the Cummins QSK23 and other tested engines Cummins implements dynamometers. Dynamometers or “dyno’s” for short can be categorized into 2 types, inertial dyno’s and absorption dyno’s. An absorption dyno directly absorbs power from the prime mover (usually driveshaft) and converts it to heat which is generally dissipated into the ambient air [7]. Here is an area where Team 2 plans to improve overall plant efficiency by harnessing this heat or possibly the energy before it is converted to heat and use a DC to AC converter to run this power back into the grid or use the heat if it will be more cost effective. In most dynamometers power is not measured directly and must be calculated from the product of either torque and angular velocity or force and linear velocity. Conversion factors must be implemented to get the result in relatable units.

#### *2.1.5. CVS Chillers and Chilled Water*

Chillers are used to cool water that is used for air conditioning and cooling machinery. In any industry, chillers are always a major power consumer. In order to reduce energy consumed by the chillers it is recommended that Cummins use absorption chillers. There are three types of absorption chillers: single effect, double effect, and triple effect chillers. In the case of the Technical Center, it will be more prudent to use a single effect chiller as it will be able to use low pressure waste heat to power the chiller efficiently. A double or triple effect chiller will require too much pressure and heat to be a feasible energy saver. Figure 2 depicts a flow chart of a single effect absorption chiller with temperature increasing in the horizontal and pressure increasing in the vertical. The flow of the water begins in the evaporator on the bottom left corner of the figure. Once heated the vapor is sent to the absorber where the heat is absorbed as the vapor enters the heat exchanger. The vapor runs a turbine attached to a generator to power the chiller. The vapor is sent to the condenser where cooled and returned to the evaporator. Using an absorption chiller as opposed to a mechanical chiller is beneficial because absorption chillers require low maintenance; they have a high reliability, and are relatively quiet and vibration free [8].



**Figure 2:** Absorption Chiller Cycle (Closed Cycle Beginning at Heat Exchanger)

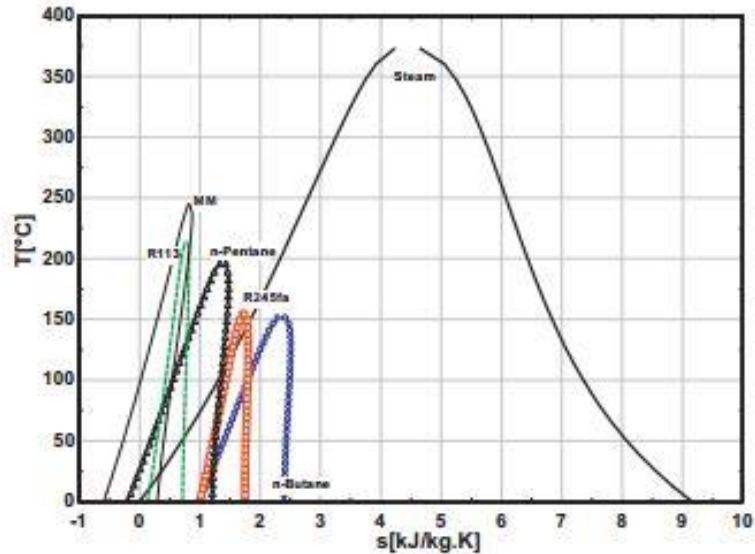
#### 2.1.6. Organic Rankine Cycles

An Organic Rankine cycle is a Rankine cycle that uses an organic material as a working fluid. Generally organic fluids are fluids that have high molecular mass meaning that organic fluids are capable of experiencing a phase change at lower temperatures, compared to a water-steam phase change. The organic fluid allows heat recovery from lower energy such as; exhaust waste heat, geothermal heat, or solar [9].

There are a wide variety of working fluids that can successfully operate in a Rankine Cycle. Water is the preferred working fluid for large scale systems that work with fossil fuel. Water is suited for this high temperature application because of water's high specific heat capacity. The main difference between water and an organic fluid is their behavior after an isentropic expansion in a turbine or expander. In the absence of super heat expansion, the exhaust steam from the turbine and compressor will consist of a low quality mixture. Low quality mixture is undesirable because it can result in severe damage of the equipment [10]. Most organic fluids such as benzene, toluene, pentane, R123, and n-butane show varying degrees of dry behavior resulting in super heat vapor upon isentropic expansion. The degree of dryness depends on the molecule complexity of the organic fluid. This dry behavior makes the working fluid superior to water in low temperature heat sources [11].

The different working fluids can be classified between isentropic, dry, and wet [12]; their classifications depend on the slope of the entropy to temperature curve. The presence of a

hydrogen bond in molecules like ammonia or water results in a larger vaporizing enthalpy therefore making it wet and inappropriate for ORC systems [10]. For an organic Rankine Cycle dry or isentropic fluids are more appropriate because they are super-heated after isentropic expansion eliminating any concerns of liquid droplets on the turbine blades. The thermal efficiency of working fluids have a weak correlation to the critical temperature of the working fluid.



**Figure 3:** T-s Diagram of Water and Few Organic Fluids

In Figure 3 it shows the temperature to entropy diagram for water and many common organic fluids used in organic Rankine cycles. As shown in the graphs, organic fluids can work in a lower temperature than a steam power generation cycle.

Organic fluids also present a variety of practical issues such as: toxicity, flammability, thermal stability, and cost [13]. After an in-depth examination of the organic working fluids available for an ORC and taking into account cost, flammability, and critical temperature of the fluids, the team realized that the working fluid that fits the system designed by the team will be n-butane with the highest thermal and economic efficiency [10].

The team decided that the system will have an increase in efficiency by using solar panels to heat the water before going to the heat exchanger. There are research papers that have been published in recent years talking about using solar panels for energy generation such as McMahan [14] or Tchanché [15].

## 3. Project Approach

### *3.1. Project Scope*

Since August the scope of Team 2's project has been to reduce the energy consumption of the Cummins Technical Center by any means the team has proposed. Before Team 2 began work on generating ideas it was discovered the CTC had a previous energy audit performed. The previous audit eliminated conventional energy saving ideas, i.e. low flow water systems, fluorescent lightbulbs, timers, insulation improvement, etc. This required Team 2 to generate higher tier energy saving ideas such as absorption chillers, solar panels, wind turbines, and an organic Rankine cycle.

#### *3.1.1. Need Statement*

The Cummins Technical Center (CTC) in Columbus, Indiana is looking to reduce their power consumption by 10%. While Cummins has already made great strides in reducing power consumption in their company, they believe there is still more work to be done. Cummins has asked Team 2 to develop multiple energy saving ideas and designs that can help contribute to this power reduction. A heavy emphasis was placed on investigation into the feasibility and cost benefit analysis of the proposed designs. **“Cummins needs to reduce their energy usage in order to save money and reduce their environmental impact.”**

#### *3.1.2. Goal Statement and Objectives*

The goal of the project is to, “Review current Cummins Technical Center (CTC) electrical usage and devise a plan to decrease it by 10%” and deliver it to Roger England by April 17<sup>th</sup>, 2015. The project scope is broken up into an overall goal being the reduction of the facility power consumption and the subsequent objectives that will help the team to accomplish this goal.

- 1) Analyze feasibility of implementing solar panels for energy generation based on weather, altitude, and longitude conditions
- 2) Create a Pareto chart to show opportunities to reduce electrical consumption based on capital invested
- 3) Come up with a variety of creative ideas for potential areas of improvement



- 4) Investigate the potential application of absorption style chillers and how much energy would be saved with respect to initial and annual costs
- 5) Determine the insulation practices at the CTC and ways to improve the building insulation
- 6) Design a way to harness the energy from the engine exhaust emissions before they are released to the outside atmosphere
- 7) Research the feasibility of increasing the efficiency of the dynamometers used in the actual engine testing
- 8) Conduct a cost-benefit analysis on the rate of return for each of the suggested ideas

### *3.1.3. Project Constraints*

The project must be completed under the following constraints listed below. These constraints were developed in order to create a feasible system.

- 1) Any methods applied for reduction in power consumption must be cost effective for capital invested
- 2) Any power saving ideas (e.g., Solar Panels) should apply to weather conditions for Columbus, IN
- 3) Solar Panels must be able to fit on the roof of Cummins Technical Center
- 4) Insulation must meet federal and state regulations
- 5) Exhaust gases must first be routed through the emissions lab for testing before they can be harnessed
- 6) Emissions to the atmosphere must comply with environmental regulations

### *3.2. Previous Work*

The first semester of the project involved gathering information about the CTC and generating a plethora of ideas to reduce energy consumption. The most feasible of these ideas were installing solar panels and updating from compression chillers to absorption chillers. The team also began to investigate uses for exhaust gases in order to generate electricity. Different applications such as trigeneration, powering the absorption chillers, or an organic Rankine cycle were investigated. It was determined at the end of the fall semester to focus the remainder of the school year on designing an organic Rankine cycle.

The other major task in spring was organizing a trip to the CTC. Due to scheduling conflicts and miscommunication between ourselves, professors, advisors and sponsors the trip was delayed until mid-November. The trip to the technical center was Team 2's main source of information as to what was feasible to accomplish and what was unreasonable or had been implemented already. Previous to the trip Team 2 had been focusing time on updating the insulation in the CTC however upon arrival it was discovered the insulation had recently been updated. The sections below explain the design and cost analysis of all of the first semester energy saving ideas.

### 3.2.1. Solar Panels

From the average insolation calculated, solar panel dimensions were derived from the constraint of the roof in order to calculate the total area of the solar panels allowed. As shown in Table 1, the dimensions of both roofs and usable area are presented. The amount of useable roof area was found to be  $4,825 \text{ m}^2$  and is sectioned into 3 parts as can be seen in Figure 3 in the design specification section of the report [16].

Table 1 - CTC Roof Dimensions

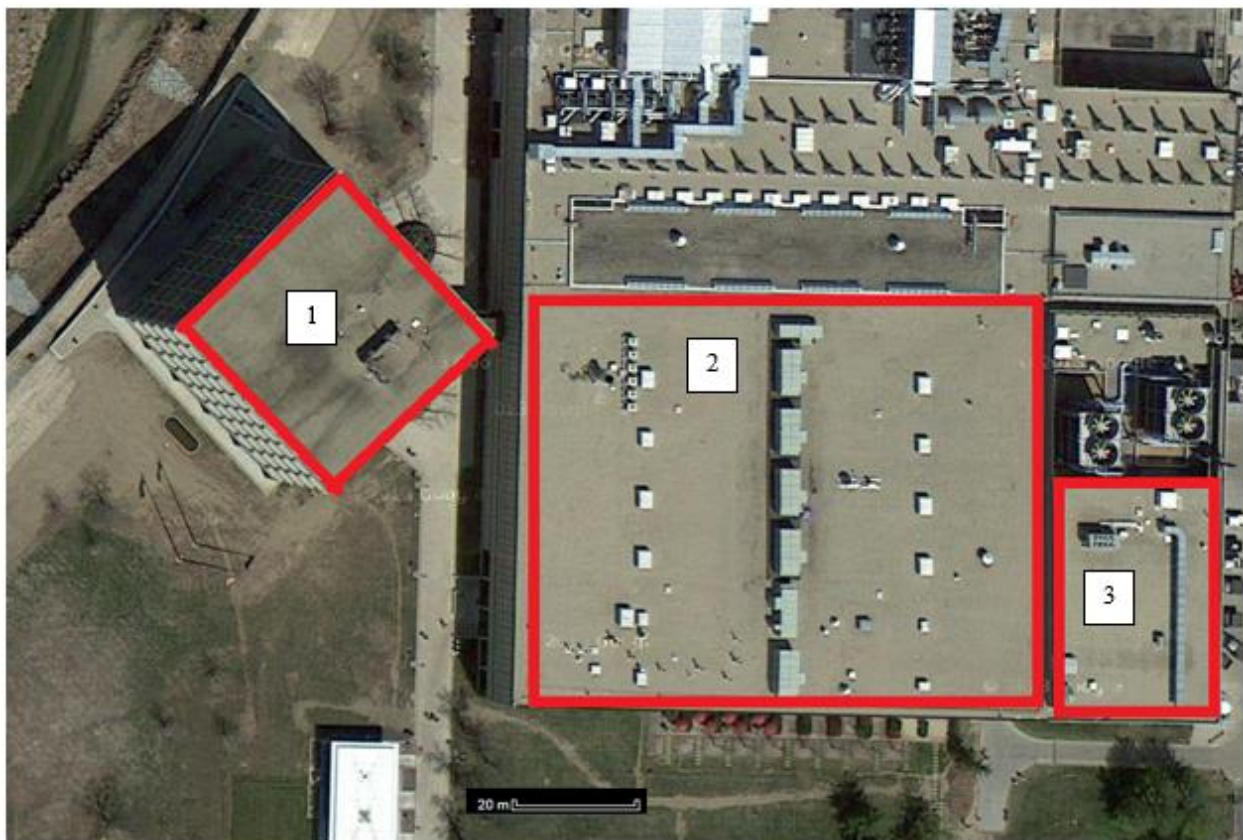
	<b>Width (m)</b>	<b>Length (m)</b>	<b>Area (<math>\text{m}^2</math>)</b>	<b>% Useable</b>	<b>Useable solar area (<math>\text{m}^2</math>)</b>
Roof 1	35	35	1,225	100	1,225
Roof 2	90	100	9,000	40	3,600

#### 3.2.1.1. Design Specifications

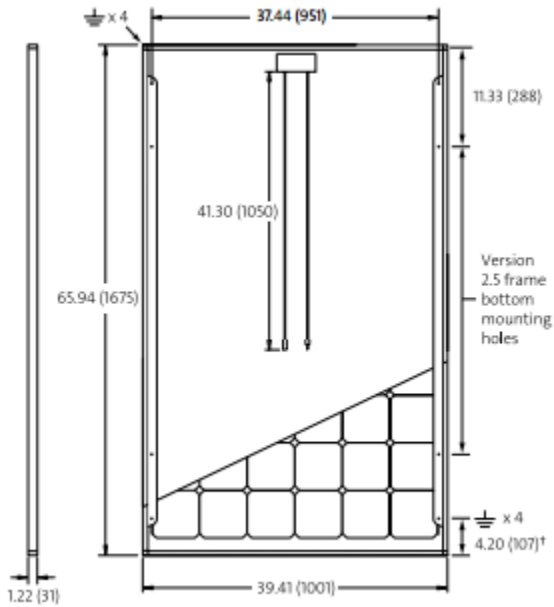
The design specifications for the required solar panels are described in the following section. These design specifications were calculated with the average insolation of  $556 \frac{W}{\text{m}^2}$ . A more accurate insolation reading for the CTC can be developed with the implementation of a STR-21G Sun Tracker. The average insolation was calculated by a MATLAB program Team 2 developed, which can be seen in the appendix of this report. The MATLAB code takes into account equations that were derived from Vieira [53] and computes the necessary values used to estimate the average insolation that the CTC is exposed to throughout the year. This code can be seen in the appendix. [5]

In order to transform the solar irradiance into electrical power solar modules were chosen and multiple modules were placed in an array. There will be six different solar arrays, all varying in size depending on the location of the roof which they are placed on. Figure 4 is an image with the three different sections and is labeled as follows: section 1 is the roof for the office building with dimensions of 35m x 35m, section 2 is the 40m x 80m section of the CTC roof and section 3 is a 20m x 20m section of the CTC roof. For section 1, the total area was assumed to be usable for the implementation of solar panels because there are hardly any obstructions. The CTC roof was divided into 2 sections and as stated previously, labeled section 2 and 3. Section 2 and 3 of the roof are the areas with the least amount of obstructions so the maximum number of solar panels can be used. The rest of the CTC roof as displayed is obstructed and cannot be used for solar panel location.

Section 2 will be split into 4 equally spaced sections. The assumption of evenly spaced arrays made the calculations easier and also presented the maximum area per section. The assumed sections will have dimensions of 40m x 20m. Section 1 and section 3 will have one solar module array.



**Figure 4:** Satellite Image of CTC Roof



**Figure 5:** Dimensions of SW 265W Sunmodule

Figure 5 shows the dimensions for the solar cell as 1.675m x 1.001m x 0.031m. From the dimensions of both solar panels and the allowable roof space for each section, there will be a total of 1,415 module cells used for the solar system. Section 1 has 360 solar modules, section 2 has 940 solar modules and section 3 has 115 modules. Table 2 summarizes these sections [5].

Table 2 - Number of Panels per section of Roof

Section	Dimensions (m)	# of Panels
Office Building (1)	35 x 35	360
CTC Roof (2)	40 x 80	940
CTC Roof (3)	20 x 20	115

### 3.2.1.2. Performance Specifications

The performance specifications are limited by the solar modules chosen for this design. The SW 265 W Sunmodule was chosen and the maximum power out voltage and current are 27.6 V and 5.02 A, respectively. The maximum voltage and current were chosen from the maximum power outputted of 138.55 per module [17]. The solar panels will not continuously operate at maximum values because the maximum amount of solar insolation will not reach the CTC 365 days of the year. This was taken into account by taking the average sunshine hours of  $4.21 \frac{hr}{day}$ . The assumption was made that a solar panel will receive the

maximum insolation from the sun for only 4.21 hours of the day. So, the solar panels were assumed to operate at maximum values for only  $4.21 \frac{hr}{day}$  throughout the entire year [6]. Also, the specifications of the solar panels were rated at  $800 \frac{W}{m^2}$  and since the CTC is in a location where the insolation of the sun on average is only  $556 \frac{W}{m^2}$ , the current and voltage were adjusted accordingly through a ratio comparison. Each module was joined into an array and depending on the section on the roof, will have different size and power outputs. Table 3 summarizes these performance specifications for the proposed solar arrays in their associated sections.

Table 3 - Performance Specifications for Solar Panel Arrays

Section	Voltage (V)	Current (A)	Power (kW)	Energy ( $\frac{kWhr}{year}$ )
Office Building (1)	437.8	119.6	52.38	80,483
CTC Roof (2)	501.2	67.62	142.7	208,330
CTC Roof (3)	247.4	67.62	16.73	25,708

The values for current, voltage, power and energy were calculated with the developed MATLAB and knowledge mentioned in the introduction about the behavior of solar panels depending on orientation. The total energy generated from the proposed solar panel system is an addition of the three different sections and equals  $314,521 \frac{kWhr}{year}$ .

### 3.2.1.3. Cost Analysis

The follow section will describe the initial cost to install the solar panels onto the CTC roof. The SW 265 Sunmodule from solar world are listed at \$270.00 a piece [17]. These solar panels were found to be the best on the market with the maximum power output and size. The cost to purchase 1,415 SW 265 Sunmodule's is \$382,050. The average cost of labor for a solar technician is  $\frac{\$19.53}{hr}$  and an estimated 400 hours of labor was used to calculate the labor cost of \$7,812. Other cost assumptions were used and can be seen in the following Table 4.

Table 4 - Assumptions Made for Solar Panel Cost

Item	Cost
SW 265 Sunmodule	\$382,050
Labor	\$7,812
Battery System	\$15,000

The battery system is the system needed to store the voltage that the solar panels create. This storage system will be to store any voltage generated from the solar panels that were not converted to power. The battery system costs include; cables, batteries, a DC/AC convertor, installation costs, and a monitoring system. These cost estimates are all relative to the contractor chosen to perform the installation, but, the total cost is \$404,862 for the proposed solar panel design. Once solar panels are installed there will be a payback period of 18 years.

*3.2.2. Insulation*

The following section is an analysis done for replacing the insulation at the CTC and this section consists of design specifications, performance specifications and cost analysis.

*3.2.2.1. Design Specifications*

Table 5 lists several alternative insulation options for Cummins. Table 5 shows the thermal resistance, what type of insulation, how ‘Green’ each material is, and whether or not it is fire resistant. These categories were all characteristics used to describe modern day insulation [6].

Table 5 - Insulation Design Specifications

<b>Material</b>	<b>Thermal Resistance</b>	<b>Types</b>	<b>Green</b>	<b>Fire Resistant</b>
Fiber Glass	2.2 to 2.7	High, Medium, Low Density	20% to 30% Recycled	Yes
Mineral Wool	3.7	Blanket and loose fill	75% post-industrial recycled	Yes
Cellulose	3.2 to 3.8	loose fill or spray	82% to 85% recycled	No
Plastic Fiber	3.8 to 4.3	High, Low Density	----	Yes
Closed Cell Foam	5.6 to 8	Spray, Foam board	----	No
Closed Cell	9	Foil	----	Yes

### 3.2.2.2. Performance Specifications

The reason for updating insulation is based mainly on the thermal resistance of the current insulation installed at the facility. A higher thermal resistance means a greater savings in heat because less is transferred to the outside. The different types of insulation listed give an idea of how feasible it will be to install in the facility. This is based on work involved as well as cost of each material. A “green” material may result in tax reductions due to the environmentally friendly nature of the material. This could be done through the ‘green point’ system or following government standards for green initiatives. The next area of design consideration is engine testing.

### 3.2.2.3. Cost Analysis

The total cost of implementing insulation is given by a fixed installation cost of \$50,000. It will take Cummins about two and a half years to make a return. This was calculated by calculating the heat flux of each insulation material. To calculate the heat flux the difference between the outside and the inside temperature of the building was calculated and then divided by the thermal resistance values of the material given in Table 5. Multiplying the heat flux by the average energy cost for industrial facilities for each material will provide the savings generated by each material. The total savings will be equal to the difference between loss in the insulating material Cummins is currently using (asphalt) and the new materials shown in Table 6.

Table 6 - Cost Analysis for Insulation

<b>Material</b>	<b>Cost (per ft<sup>2</sup>)</b>	<b>Total Cost for the roof</b>	<b>Total Saving per year</b>
Fiber Glass	0.42	\$59,606.66	\$18,331.76
Mineral Wool	0.625	\$64,295.62	\$18,352.97
Cellulose	1.25	\$78,591.25	\$18,448.09
Plastic Fiber	1.5	\$84,309.50	\$18,596.58
Closed Cell Foam	2.2	\$100,320.60	\$18,955.43
Closed Cell	2.3	\$102,607.90	\$19,068.17

### 3.2.3. Engine Testing

The CTC currently has 88 main test cells with an additional 8 auxiliary cells. The CTC is currently testing the Cummins ISX and ISB engines. For ease of calculations, the assumption was made that all test cells contain the ISX15 600 due to the fact that specifications for this engine are readily available. This assumption has been preapproved by Dr. Hays. These engine specs are listed below and from this data performance specifications were made for both the mechanical energy generated by the dynamometers and the heat captured from exhaust gasses [7].

#### 3.2.3.1. Design Specifications

The design specifications are limited to constraints of the engines shown in Table 7.

Table 7 - Engine Specifications

<b>Engine Type</b>	<b>In-line, 6-cylinder</b>
Advertised Horsepower	447 kW
Peak Torque	2,779 Nm
Governor Speed	2,000 rpm
Certification Level	EPA 2013
T <sub>out</sub>	400°C
Test Cell Temperature Range	(-20 – 150) degrees C

With these engine specifications, performance specifications are generated to harness this available energy. With ninety-six engines, each generating up to 447 kW of power there is a huge amount of energy available for capture. Also all ninety-six engines are emitting exhaust gasses at upwards of 400 degrees Celsius.

#### 3.2.3.2. Performance Specifications – Dynamometers

As stated earlier, the CTC currently has 88 main test cells with an additional 8 auxiliary cells, each hypothetically containing an ISX engine. The ISX has an average test speed of 1200 rpm generating 268.2 kW per engine or test cell. At the CTC Cummins has implemented absorption dynamometers. A dynamometer efficiency was assumed to be 85% and the power is being added directly back to the grid. An assumption was made that the usage ratio was 10% because assuming all test cells are on all the time would cause



unnecessary error. This leads to an annual power generated from dynamometers of 7,878,643 kWh and annual savings of \$569,232 as seen in Table 8. One area that can save Cummins energy is to approve this efficiency. At the current efficiency over 1 million kilowatt hours are lost annually. If the dynamometer efficiency was increased, another 7% an additional \$50,000 in energy would be converted back to the grid [7].

Table 8 - Engine Testing Performance Specifications

<b>Variable</b>	<b>Value</b>
Test Power	268.2 kW
Power Generation	7,878,643 kWh
Power Lost	1,390,348 kWh
Annual Savings	\$569,232
Annual Savings Lost (inefficiency)	\$100,453

3.2.3.3. *Performance Specifications – Exhaust Gases*

The performance specifications for exhaust gas are a theoretical assumption and the process is explained for values determined. The CTC currently does not use exhaust gases exiting the engines. One reason is the engine exhaust gases must be analyzed prior to any type of recirculation. Cummins suggested taking into account an efficiency loss of 10-15% in this process which will reduce the total mass flow rate exiting from the emissions lab. The Next step team 2 took was to calculate the amount of energy available through exhaust gas heat. Team 2 started with the conservative assumption that fuel consumption of the ISX15 600 is 6 m/G. This consumption is at a highway velocity of 70 m/h, leading to fuel Consumption of 11.67 G/h. According to emissions data from a study conducted by Cummins and the American Transportation Research Institute during, one ISX engine running for one hour, would produce 1174 kg of CO<sub>2</sub>, 1.167 g Particulate Matter, and have 130 g of Methane Hydrocarbons. These pollutants are well under the EPA regulations and would be found analyzed and reduced before recirculating into the exhaust gas heat converter. Team 2 assumed an ambient temperature of 25°C and used the earlier stated outlet temperature of 400°F in order to calculate the temperature gradient with the system. Team 2 used a Diesel fuel density of 832 kg/m<sup>3</sup> and air density of

1.225 kg/m<sup>3</sup>. Next Volumetric flow rate of fuel in is converted to  $3.24 \times 10^{-3}$  G/s and then  $1.23 \times 10^{-5}$  m<sup>3</sup>/s and used to find mass flow rate of fuel in of 10.21 g/s. Cummins also supplied us a volumetric flow rate of air into the engines of 18.3 ft<sup>3</sup>/min which is converted to  $8.63 \times 10^{-3}$  m<sup>3</sup>/s and used to find mass flow rate in of air at 10.58 g/s. Flow rates are then balanced and using conservation of mass along with the ideal gas law to find mass flow rate refuel of 1.021 g/s and mass flow rate exhaust of 19.77 g/s. Then, with assumed Specific Heat values for fuel equal to 0.43 BTU/lb.\*F found for Petroleum Diesel, which is then converted to 1.832 kJ / kg\*K for our equation. With these values; Newton’s Law of Cooling was applied to find Heat Energy.

$$Q = m_{egas} \cdot C_v \cdot (T_{out} - T_a) \quad \text{Equation. (1)}$$

Q is the amount of heat generated from the exhaust gases, which is usable energy. Cv is the constant specific heat at constant volume which is an assumption that was made and Ta is the ambient temperature that was established. Tout is the exhaust gas temperature of 400°F. The heat energy of 345 kW generated was found through Eq. (1). An assumed heat lost in combustion was 10% and is reflected within the calculations, resulting in 319 kW heat energy available to convert. The results of these assumptions are found in Table 9.

Table 9 - Engine Testing Performance Specifications

Variable	Value
Fuel Consumption	11.67 Gallons / Hour
Mass Flow Fuel In	10.21 g/s
Mass Flow In Air	10.58 g/s
Mass Flow In Total	20.79 g/s
Mass Flow Exhaust	19.77 g/s
Specific Heat Fuel	1.832 kJ / kg*K
Heat Generated	354.1 kW
Heat Available to Convert	318.7 kW
Heat Available	1,147,294 kWh

### 3.2.4. Absorption Chillers

Average absorption and compression chiller components and power consumptions were obtained from the department of mechanical engineering at Anna University [2]. The components are listed in Table 10 and the power consumptions are listed in Table 11. Both compression (VCRS) and absorption (VARs) refrigeration systems were compared to show the potential savings of the absorption chillers. [18]

#### 3.2.4.1. Design Specifications

The components listed in Table 10 are the major power consuming parts for absorption and compression systems. The absorption system uses several pumps to run hot water, cold water, and refrigerant through the heat exchanger. The compression system uses a compressor to cool water as well as cold water pumps. These different components have a major impact on the power consumption of each design as shown in the performance specifications.

Table 10 - List of Components of VARs and VCRS

<b>Components</b>	<b>VARs</b>	<b>VCRS</b>
Refrigerant Pump	<b>X</b>	
Solution Pump	<b>X</b>	
Compressor		<b>X</b>
Chilled Water Pump	<b>X</b>	<b>X</b>
Cooling Tower Fan	<b>X</b>	<b>X</b>
LT Hot Water Pump	<b>X</b>	
HT Hot Water Pump	<b>X</b>	
Cooling Water Pump	<b>X</b>	<b>X</b>

#### 3.2.4.2. Performance Specifications

Table 11 lists the annual power consumption of VARs and VCRS components in kWh/yr. The bottom row shows the cumulative cost and it is apparent that the absorption system uses significantly less power than the compression system, about 74% less power. This is due mostly due to the removal of the compressor from the system as it used 81% of the compression systems power.

Table 11 - List of Components of VARS and VCRS

<b>Components</b>	<b>VARS</b>	<b>VCRS</b>
Refrigerant Pump	13,140	N/A
Solution Pump	61,320	N/A
Compressor Power	N/A	2,938,980
Chilled Water Pump	324,120	324,120
Cooling Tower Fan	131,400	87,600
LT Hot Water Pump	43,800	N/A
HT Hot Water Pump	43,800	N/A
Cooling Water Pump	324,120	262,800
<b>Total Annual Power Consumption (W)</b>	<b>941,700</b>	<b>3,613,500</b>

3.2.4.3. *Cost Analysis*

Table 12 shows the average initial cost of the VARS and VCRS. The absorption systems tend to be more expensive because of their machining cost and the extra pumps. However the annual operating costs shown in Table 13 shows the absorption is less expensive than the compression system. [18]

Table 12- Initial Cost of VARS AND VARS

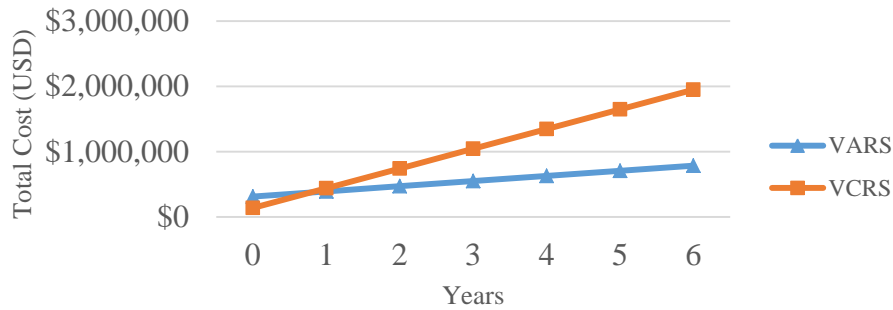
	<b>VARS</b>	<b>VCRS</b>
Machine Cost	\$278,478	\$112,041
Cooling Tower	\$22,826	\$19,565
Cooling Water Pump	\$5,435	\$4,348
Chilled Water Pump	\$3,913	\$3,913
<b>Total Initial Cost</b>	<b>\$314,348</b>	<b>\$139,868</b>

Table 13 - Annual Operating Cost of VARS and VCRS

	<b>VARS</b>	<b>VCRS</b>
Refrigerant Pump	\$1,097.19	N/A
Solution Pump	\$5,120.22	N/A
Compressor Power	N/A	\$245,404.83
Chilled Water Pump	\$27,064.02	\$27,064.02
Cooling Tower Fan	\$10,971.90	\$7,314.60
LT Hot Water Pump	\$3,657.30	N/A
HT Hot Water Pump	\$3,657.30	N/A
Cooling Water Pump	\$27,064.02	\$21,943.80
<b>Total</b>	<b>\$78,631.95</b>	<b>\$301,727.25</b>

Figure 6 shows the total cost of both absorption and a compression system. Figure 6 also shows that an absorption system will have the initial cost of the system paid back in less than a year compared to the compression system. Figure 6 can be found on the preceding page.

## Total Cost of VARS v. VCRS Over 7 Year Period



**Figure 6:** Total Costs Of VARS Vs VCRS

### 3.3.Present Work

Over the past semester Team 2 has dedicated their efforts in developing an organic Rankine cycle run off of the exhaust gases of the engine. A large portion of the first half of the semester was dedicated to designing a fabricated prototype for the technical center. After encountering several major roadblocks with designing a working fabrication that would accurately simulate the proposed system for the technical center the idea was scrapped. While this did set team 2 behind schedule for a couple weeks a lot of work was putting into catching up to produce a valuable simulation.

The remainder of the semester was focused on an in depth simulation of an organic Rankine cycle. The first step of designing a simulation involved determining the right components. Team 2's Rankine cycle consists of a heat exchanger, compressor, turbine, and solar collector. After determining and designing the appropriate components the heat transfer and energy equations were programmed into MatLab for analysis. The next sections describe the design and selection of each component.

#### 3.3.1. Heat Exchanger

The heat exchanger was designed by Daniel Baker and Marvin Fonseca. The first step was determining the temperature difference between the inlets and outlets of the exhaust gases and n-butane. The temperature of the exhaust inlet was known to be 205°C and the outlet was assumed to be 150°C. Using Equation 2 to determine the heat transfer provided by the exhaust gases it was possible to determine the temperature difference occurring in the n-butane.

Where Q is the overall heat transfer, m is the mass flow of the fluid, Cp is the specific heat of the fluid and ΔT is the temperature difference. The system was initially assumed to be perfectly insulated therefore allowing the heat transfer between the exhaust gas and n-butane to be equal. The heat transfer from the exhaust gas was calculated to be ~2500kW based on 50 ISX engines. The overall heat transfer will change with different engines due to the mass flow change however this was a good average recommended by the team’s sponsor.

$$Q_{exhaust} = Q_{n-butane} = \dot{m}c_p\Delta T \quad \text{Equation (2)}$$

In order to determine the temperature difference in the n-butane a mass flow rate of 7kg/s. This was to ensure stability through the turbine, compressor, and solar collector. The inlet temperature of the n-butane was also known because it was the outlet of the solar collector which averaged out to 80°C. Using these numbers and the specific heat of n-butane [19] the average outlet temperature of the n-butane is 133°C. Table 14 shows the inlet and outlet temperatures of the exhaust gases and n-butane.

Table 14 - Inlet and Outlet Temperatures of Exhaust Gas and n-butane

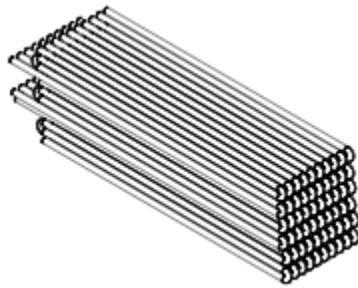
	<b>Temperature In (°C)</b>	<b>Temperature Out (°C)</b>
<i>Exhaust Gas</i>	205	150
<i>n-Butane</i>	80	133

After determining the inlet and outlet temperatures the next step was to determine the dimensions of the heat exchanger. These dimensions were assumed initially and reworked through multiple iterations to ensure the heat exchanger would be suitable. Table 15 describes the tube parameters. The number of passes describes how many times the butane will pass through the shell. To ensure a large amount of heat transfer the tubes will be made from brass, a good conductor of heat.

Table 15 - Tube Dimensions and Parameters

Parameters	Size
Tube Inner Diameter (mm)	160
Tube Outer Diameter (mm)	175
Clearance Between Tubes (mm)	25
Total Tubes	30
Number of Tubes/row	10
Number of Passes	4
Single Pass Length (m)	10
Total Tube Length (m)	40

Figure 7 shows the tube layout dimensioned to the parameters in Table 15.



**Figure 7:** Tube Layout of Heat Exchanger

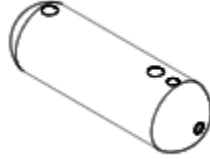
After determining tube parameters the shell was designed to fit all of the tubes comfortably and allow exhaust gases to pass through. The shell parameters are listed in Table 16. The baffles are used to direct the flow of the exhaust gases having 20 allows for a maximum heat transfer between the exhaust gas and the n-butane. In order to provide sufficient insulation for the system the shell will be fabricated from stainless steel.

Table 16 - Shell Parameters

Parameters	Size
Number of Baffles	20
Space Between Baffles (m)	0.5
Shell Inside Diameter (m)	4.25
Shell Outside Diameter (m)	4.253



Figure 8 shows the shell design of the heat exchanger. The smaller openings are 0.5m in diameter and are the inlet and outlet of the n-butane. The large openings are 0.75m in diameter and are the inlet and outlet of the exhaust gas.



**Figure 8:** Shell and Heat Exchanger

In order to calculate the velocity of the respective fluids it was necessary to first obtain the flow area. This was calculated using equation xxx for the tube and equation 3 for the shell. Where  $N_t$  is the number of tubes,  $ID_t$  is the inner diameter of the tube, and  $N_p$  is the number of passes. For equation 4 shell  $D_s$  is the inside shell diameter,  $B$  is the baffle spacing,  $C$  is the clearance between adjacent tubes, and  $PT$  is the tube pitch.

$$A_t = \frac{N_t \pi (ID_t^2)}{4N_p} \quad \text{Equation (3)}$$

$$A_s = \frac{D_s C B}{P_T} \quad \text{Equation (4)}$$

Table 17- Flow Area of Shell and Tube

	<b>Flow Areas (m<sup>2</sup>)</b>
Flow Area of Tube ( $A_t$ )	0.15079632
Flow Area of Shell ( $A_s$ )	0.265625

Table 17 shows the flow areas of both the tube and shell. These areas are directly related to the flow velocity through the heat exchanger. This relation is shown in equation 5, where  $m$  is the mass flow of the fluid,  $p$  is the density of the fluid, and the flow area of the respective fluid. The fluid velocities are shown in Table 18.

$$V = \frac{\dot{m}}{\rho A} \quad \text{Equation (5)}$$

Table 18 - Velocities of n-Butane and Exhaust Gas

	<b>Fluid Velocities (m/s)</b>
Velocity of Butane (V <sub>t</sub> )	18.7
Velocity of Exhaust (V <sub>s</sub> )	22.3

Another important parameter considered in the heat exchanger design is the Reynolds number of each fluid. The Reynolds number determines whether the flow is laminar (smooth) or turbulent (rough). In order to determine the Reynolds number of the shell it is first necessary to calculate the equivalent diameter. This is done using equation 6, where P<sub>T</sub> is the tube pitch and OD<sub>t</sub> is the outer diameter of the tube. This equation is the equivalent diameter for triangular pitch tubes. This means that the tubes are staggered creating a triangular cross section. The equivalent diameter was calculated to be 0.077m<sup>2</sup>.

$$D_e = \frac{3.46P_T^2 - \pi OD_t^2}{\pi OD_t} \quad \text{Equation (6)}$$

After the equivalent diameter of the shell was calculated it was possible to determine the Reynolds number of the respective fluids using equations 7 and 8. ρ is the density of the fluid, V is the velocity, ID is the inner diameter of the tube, D<sub>e</sub> is the equivalent diameter of the shell, and μ is the viscosity of the fluid.

$$Re_t = \frac{\rho_t V_t ID_t}{\mu_t} \quad \text{Equation(7)}$$

$$Re_s = \frac{\rho_s V_s D_e}{\mu_s} \quad \text{Equation (8)}$$

The Reynolds numbers of n-butane and the exhaust gas are given in Table xxx. Since both numbers are less than a magnitude of  $10^7$  they are feasible values for a shell and tube heat exchanger [6].

Table 19 - Reynolds Numbers of n-Butane and Exhaust Gas

<b>Reynolds Number of n-Butane</b>	<b>Reynolds number of Exhaust Gas</b>
1.06E+06	6.46E+04

The final necessity to ensure a feasible heat exchanger is to determine the pressure drop between the exhaust gas inlet and outlet and the butane inlet and outlet. The pressure drops determine whether or not the fluid has the potential to move through the system. Before the pressure drop can be calculated the friction factor of both the shell and tube must be determined. The friction factor takes into account frictional losses through the system.

Since the flow is turbulent through the system more complex formulas must be used to determine the friction factor. In order to determine the friction factor through the tubes Equation xxx also known as the Chen equation is used.  $\epsilon$  is the roughness of brass [20],  $Re$  is the Reynolds number of the n-butane, and  $ID_t$  is the inner diameter of the tube. Table 20 shows the friction factor through the shell and tube. Since both brass and stainless steel are smooth materials the friction factors for both shell and tube are low. The friction factor will have a large effect on the pressure drop as it is accounting for all frictional losses.

Table 20 - Friction Factors of Shell and Tube

<b>Friction Factor of Tube</b>	<b>Friction Factor of Shell</b>
0.012	0.140

With all of the necessary parameters calculated it is possible to determine the pressure drop through the system. The pressure drop through the shell and tube were calculated using equations xxx and xxx respectively. The pressure difference in the shell involves  $\rho$ , the exhaust gas density,  $V_s$ , the exhaust gas velocity,  $D_s$ , the inner diameter of the shell,  $D_e$ , the equivalent diameter of the shell,  $f_s$ , the friction factor of the shell and  $N_b$ , the number of baffles. The tube parameters are  $\rho$ , the density of n-butane,  $V_t$ , the velocity of the fluid,  $f_t$ , the friction factor of the

tubes,  $L$ , the length of the tube,  $ID_t$ , the inner diameter of the tube, and  $N_p$ , the number of passes through the tubes.

$$\Delta p_s = \frac{\rho V_s^2 D_s}{2 D_e} f_s (N_b + 1) \quad \text{Equation (9)}$$

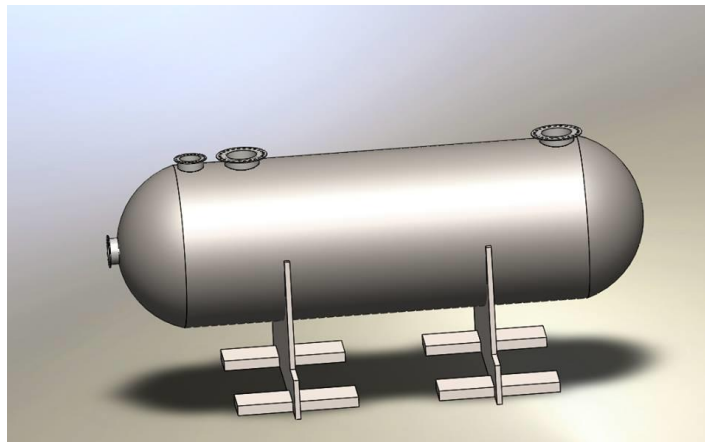
$$\Delta p_t = \frac{\rho V_t^2}{2} \left( \left( \frac{f_t L}{ID_t} \right) + 4 \right) N_p \quad \text{Equation (10)}$$

Table 21 shows the pressure drop through both the shell and tube sides of the heat exchanger. The resulting pressure drops are high enough to pull the gases through the system at the appropriate speed with minimal losses, indicating that the overall system will be feasible for Cummins.

Table 21 - Pressure Drop through Shell and Tube

Pressure Drop Through Shell (kPa)	Pressure Drop Through Tube (kPa)
30.83	12.11

After all the calculations were complete a model of the heat exchanger was drawn in SolidWorks to show a fully dimensioned model. Figure 9 shows the final heat exchanger. The total weight of the system was calculated to be 52 tons which means it would be inadvisable to install this part of the system on the roof of the CTC. While the system is very large this is justified by the fact that 44kg/s of exhaust gases are needed for the system to function properly.

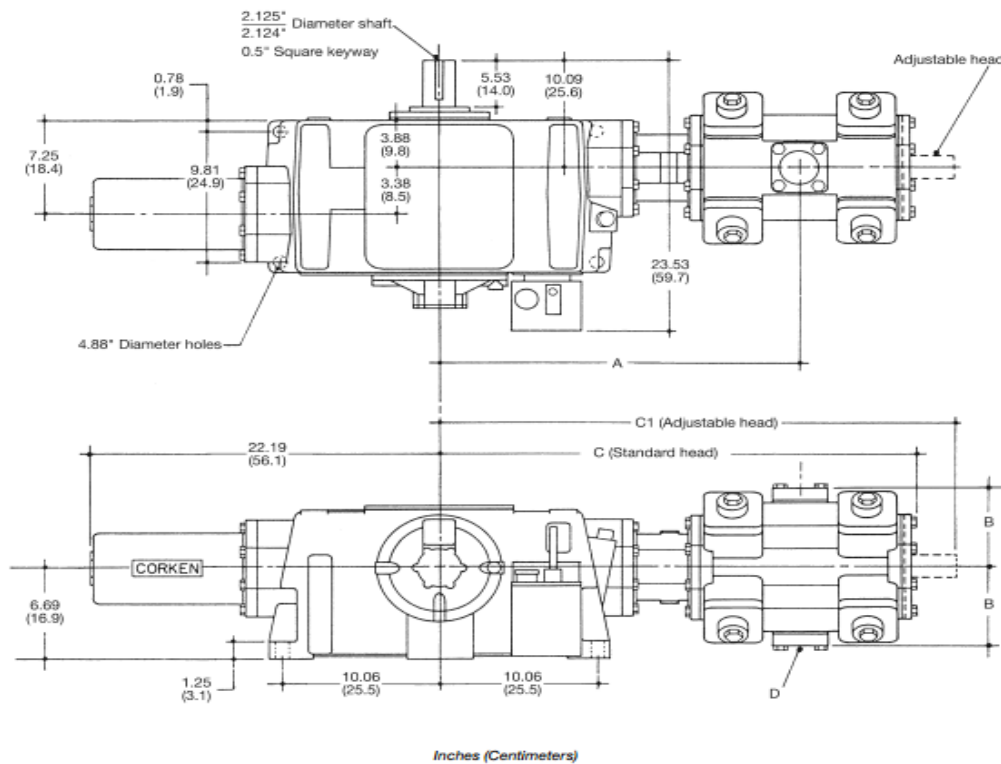


**Figure 9:** Final Heat Exchanger Design

### 3.3.2. Compressor and Turbine Selection

Designing a functional compressor and turbine was far beyond the scope of the project and capabilities of Team 2 to complete. Therefore it was decided to do research and choose the best systems need for the ORC. A Siemens HG601BX turbine was chosen as it is able to handle the high temperature butane and generate about 450kW of power. The compressor was chosen from Corken Compressors because they design compressors used for gas compression specifically for Organic Rankine Cycles.

The HG601BX Corken Compressor was selected based on its ability to compress n-butane. Also, the compressor is capable of compressing n-butane to 1.1MPa, the required pressure, which is capable of overcoming the pressure losses throughout the system. Figure 10 is a technical drawing for the HG601BX Corken Compressor [21] [52].



**Figure 10:** HG601BX Compressor

The weight of HG601BX is 318 kg, operates at 1200 rpm 55 kW, capable of compressing 115cfm of gas and has a 152.4 mm bore.

The turbine selected that would extract the maximum amount of work from the incoming n-butane is the Siemens SST-060 Series Steam Turbine. The SST-060 is capable of generating 6MW of electricity. The capable inlet pressure of the turbine is 13.1 MPa. The technical drawings could not be provided by Siemens but the dimensions of the turbine are 1.5m x 2.5m x 2.5m.

### 3.3.3. Solar Thermal Collectors

AE 3000 Solar Thermal Collectors (STC) was chosen to generate additional heat in the n-butane. The dimensions of one solar thermal collector are 3m in length, 1.2 m in width and 0.076 m in height. Given the total available area at the CTC, 500 collectors would be used for the system. The AE 3000 STCs have a life time of 30 years and are capable of containing 1.22 gals per collector. Depending on the day in the year, the solar collector efficiency range from 26% to 37%. With the varying efficiency of the solar collectors, a temperature output for n-butane range between 90°C to 130°C.

### 3.3.4. Wind Turbine Generation

Table 22: Wind Turbine Decision Matrix

option	ICC	Mass (kg)	weight(lbs)	Weight (tons)
1	1,195,322.43	433,494	953,688	476.8439015
2	1,148,624.32	489,559	1,077,031	538.5154311
3	1,132,131.08	619,748	1,363,445	681.722609
<b>4:</b>	<b>1,188,536.64</b>	<b>615,124</b>	<b>1,353,274</b>	<b>676.6369404</b>

In Table 22 above we can see the 4 options and some total results for the system. Each part is individually calculated for each option type and summed to form these criteria. Option 1 is a three-stage drive with a high speed generator. This option is the lightest available but requires 3 stages of mechanical power transfer which usually is accompanied by power losses and also has the highest initial capital cost. Option 2 is a single stage drive with medium speed, permanent magnet generator this option is relatively light as well and also relatively cheap. The problem with this design is the technology for permanent magnet generators haven't been perfected yet and mechanical energy losses still apply. Option 3 is a multi-path drive with multiple permanent magnet generators. This option has the highest weight of 682 tons once again this design is plagued by magnet generator technology and mechanical energy losses, as well as requires the implementation and upkeep for multiple generators. Option 4 is the direct drive turbine. This option doesn't need a gearbox because the generator is connected directly to the

mechanical power. This option has a slightly higher ICC (Table 22) due to the required extra support to handle the generator elevation. This option has the highest efficiency and it is found the energy saved is far greater than the cost and energy to add the extra support. This option avoids mechanical energy losses due to transferring the mechanical energy. In Table X2 below the results for the implementation of two direct drive wind turbines as proposed. With a life span of 25 years the Return on Investment for the two wind turbines is \$3,380,904.28 with only 7.26 years until profit begins.

Table 23: Direct Drive Turbine results

P_gen	297.7185185 kW
Egen_annual	9388.8512 GJ/year
Egen_annual	2,608,035.09 KWh/year
Revenue Annual	\$221,682.98 \$/year
25 year ROI (\$)	3,480,904.28
Years to return	7.260229215

#### **4. Methodology**

The basis of the programming works on several different assumptions. The first part of this is based on the properties of the exhaust gas produced by the engines at the facility. The team used properties consistent with air such as density, specific heat, etc. This was an assumption approved by the sponsor, because many of the properties share similar characteristics to air, and if the user would like to use different properties, it is a simple matter to replace the air properties with any other values as the user deems fit.

The team was able to determine the exhaust mass flow from several different engine families using a catalog of rated CFM rates of different HP engines within the family [22]. By storing these rates into an array and converting them to units of kg/s through unit analysis with

the density of the exhaust, the average mass flow rate of the exhaust can be calculated in respect to the number of engines running. The 5 engine families used in the simulation are: ISB, ISC, ISL, ISM, and ISX [23].

The ORC design is based on 50 ISX engines running. The average engines running are 50 engines 24 hours a day, 365 days a year, according to the sponsor, so the system must be able to accommodate this. Additionally, the properties of the working fluid n-butane are set based on properties given by the National Institute of Standards in terms of density, specific heat, and various other properties.

The heat exchanger was designed to accommodate the working fluid without the assistance of the solar thermal collectors in order to contribute an acceptable amount of energy whenever the solar collectors are not contributing at night. The inlet temperature of exhaust gas is given value of 400F, a measurement given to the team by Cummins, Inc. In order to have a feasible system size, the team had to iteratively select the output temperature of the exhaust gas from the heat exchanger to see how much exhaust gas energy is provided. This is simply provided by the thermodynamic relationship as seen from equation (1).

Since the mass flow rate is an input decided by the user and the specific heat and the inlet temperature are relatively constant, choosing an outlet temperature needed to be a reasonable number that a shell and tube heat exchanger could accommodate without being excessively large and expensive. Assuming the system is adiabatic, the heat provided would be equal to the heat gained by the n-butane working fluid.

However, the heat exchange within the heat exchanger would not be adiabatic in a true life system. In order to account for this, the team determined the efficiency of the designed heat exchanger to be around 40% through an iterative process of selecting temperatures for outlet and inlet. By solving for it iteratively, the true temperature outlet was determined, and thus the actual efficiency was determined. Since these temperatures fluctuate throughout the day based on the heat transfer from the solar thermal collectors, the efficiency of the system will slightly vary depending on the time and day of the year. This allowed the team to determine the heat energy available in the n-butane for any given day.



Next, we had to select a turbine based on the estimated power output. Based on textbook research from [24], the turbine efficiency was initially guessed to be around 50% to account for the power that would be lost to the compressor that the turbine drives. As with the other components, the design parameters for the turbine were solved for iteratively by selecting reasonable guesses that are within the range of acceptable values for a turbine-generator system [24]. Assuming that the temperature from the heat exchanger is equal to the temperature going into the turbine in an ideal system, and selecting operation pressures that are well within the thresholds of safe operating limits with good enthalpy change, the temperature of the outlet of the turbine was calculated isentropically. In turn this gave the enthalpy change value of the turbine, which when multiplied times the mass flow rate that was set at a constant 7 kg/s, gives the true power output of the turbine. In order to accommodate these operational limits, the correct turbine had to be selected. After doing research into commonly used turbines for ORC systems, the team got into touch with Siemens to request information on their turbine systems. After emailing their Germany based power generation division, we were able to come up with a few conclusions. The Siemens SST-060 Series Steam turbine is the most commonly used turbine for ORC purposes among the types of turbine [25]. As mentioned in the report for manufacturing, the dimensions of the turbine are relatively small compared to the size of the heat exchanger and solar thermal collectors. Additionally, the compressor was determined in much the same way.

The compressor was selected based on relative information close to the approximations made by Siemens for their ORC systems. Since the team had calculated the pressure drops and temperature changes throughout the system, it was simply a matter of finding a compressor that could meet those requirements and still move the same amount of mass flow rate of n-butane. The compressor temperature and pressure drops were solved in the iterative fashion assuming isentropic systems as the initial guess values, and solving for the true values. The outlet of the compressor was set to have an outlet temperature of 80C as suggested by Siemens in their module packages for ORC systems.

#### *4.1.Solar Collector Coding*

The following section is broken down into two different categories for Solar Collector system created for the power exhaust capturing system

#### 4.1.1. Solar Irradiation Code

Sunrise and sunset times for year in Columbus, Ohio tabulated in Excel using iterative process taught in power generation [32]. Other values for Solar Collector code that were important were;  $I_o = 1367 \text{ W/m}^2$ ,  $a = 0.14$ ,  $h = .192 \text{ km}$  above sea level,  $AM = 1.5$  Air Mass Index [100 units],  $aw = 0$ ,  $\rho = 0.4$ , along with latitude and longitude from online sources [2]. These values were used to determine the local irradiation, elevation at the CTC and the Air mass Index which takes into account the geographical elevation of the CTC. Before analysis solar calculations must be made shown below [53];

$$\alpha = \frac{360}{24} (t - 12) \quad \text{Equation(11)}$$

$$\cos \chi = \sin \lambda \sin \delta + \cos \delta \cos \lambda \cos \alpha \quad \text{Equation (12)}$$

$$\tan \zeta = \frac{\sin \alpha}{\sin \lambda \cos \alpha - \cos \lambda \tan \delta} \quad \text{Equation (13)}$$

Next declination, local Irradiation and direct beam irradiation are calculated throughout the day using equations equation 14, equation 15, and equation 16 in a loop for “number of points” times set to 24 setting 30 minute intervals.

$$\delta = 23.44 \sin \left[ 360 \left( \frac{d-80}{365.25} \right) \right] \quad \text{Equation (14)}$$

All equations that were used to determine the solar radiation were based off of the previous equations. For more detailed equations and explanation on how the solar code was developed refer to Appendix C. Diffuse and later Total Irradiation the average cloudiness factor of each month is set using an, if then statement, and day as the variable.

#### 4.2. Cost Analysis Coding

The numbers for the cost analysis were somewhat simple to determine based on the references for the different components that the team had chosen for the design. The team contacted Siemens for a price quote on initial costs of the turbine which ranged anywhere from 500,000-1.5 million euros, converted into dollars is \$541,500 - \$1,624,500. Likewise, the team got price quotes from the suppliers for the solar thermal collectors, storage tank, compressor, piping, and raw material needed to build the heat exchanger. The cost analysis takes into account the shipping and transportation costs of moving these components to the facility, and these costs are

fairly large due to the fact that these components are quite large and expensive in and of themselves.

For the heat exchanger, there were several resources available to calculate the initial costs along with the costs of cleaning. The initial cost is broken into 3 main components, the cost of the shell, the cost of the tubing, and the initial cleaning cost.

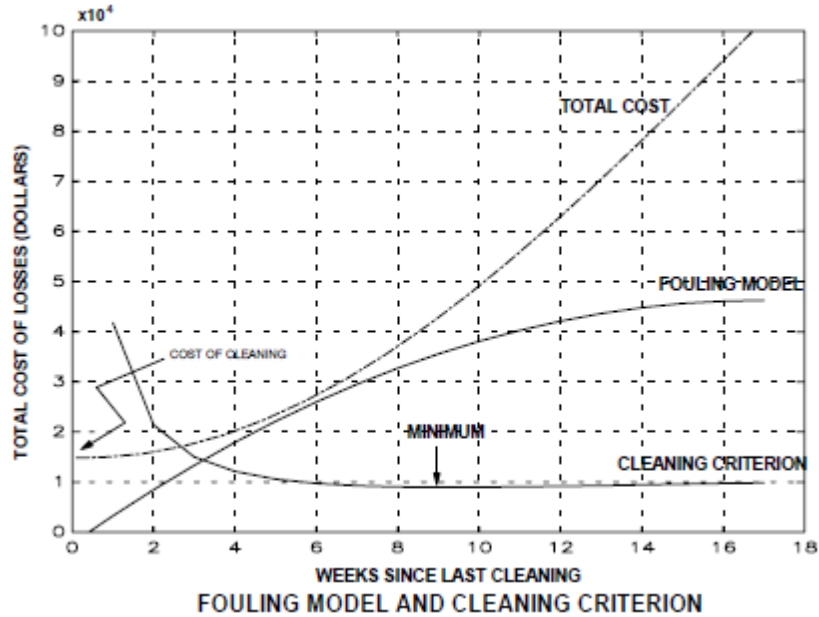
$$Cost_{Shell} = \$1800 * (D^{2.5}) * L \quad \text{Equation(15)}$$

$$Cost_{Tubing} = \$1950 \quad \text{Equation (16)}$$

$$Cost_{Initialcleaning} = \$350 * D * L \quad \text{Equation(17)}$$

D is the diameter of the shell, L is the length of the shell, and all Cost variables are in units of dollars. Note that the cost of tubing was a fixed cost of \$1,950 as depicted in Janna [26]. These numbers are taken from 2011, so prices may have marginally changed from then, but they are fairly equivalent to pricing quoted from heat exchanger design texts [27]. Just as well, the initial cleaning cost is consistent with the report done by optimizing the cleaning of heat exchangers [28]. The cost calculated using Equation 17 is within \$100 of the estimate given by the report for the optimizing of cleaning heat exchangers.

Using the aforementioned report as a template, the team created a generic fouling model to estimate the optimum time interval to clean the heat exchanger. These models are rough estimates based on the data provided by a 2 pass heat exchanger in a separate experiment as seen in Figure 11, but they are fairly consistent with the heat exchanger data for the Cummins design [29].



**Figure 11:** Optimum desired cleaning time based on fouling experimental data.

The team surmised that the annual cost of maintenance of the different components including cleaning the heat exchanger and maintaining the turbine, compressor, and solar collectors would be in the range of \$250,000. By plotting the initial and annual costs against the money saved from the ORC with solar collectors system, the payoff factor is determined by looking at the intersection of the two lines. After this point, the system begins making money, and this is a simple code that was taken in the intersections.m script file provided as an open source file exchange function via MathWorks [23]. It should be noted that the team took into account the present and future values of the dollar, and the full results can be seen in the Cost Analysis section.

## 5. Final Design Results

For the final design results the GUI operation is explained, then the results of the GUI and other savings idea are shown and explained, next the maintenance of the system is listed and explained.

### 5.1. GUI Operation Instructions

The simulation program created by Team 2 is designed to be a user-friendly program that allows Cummins, Inc. to control different important parameters. The input parameters are the

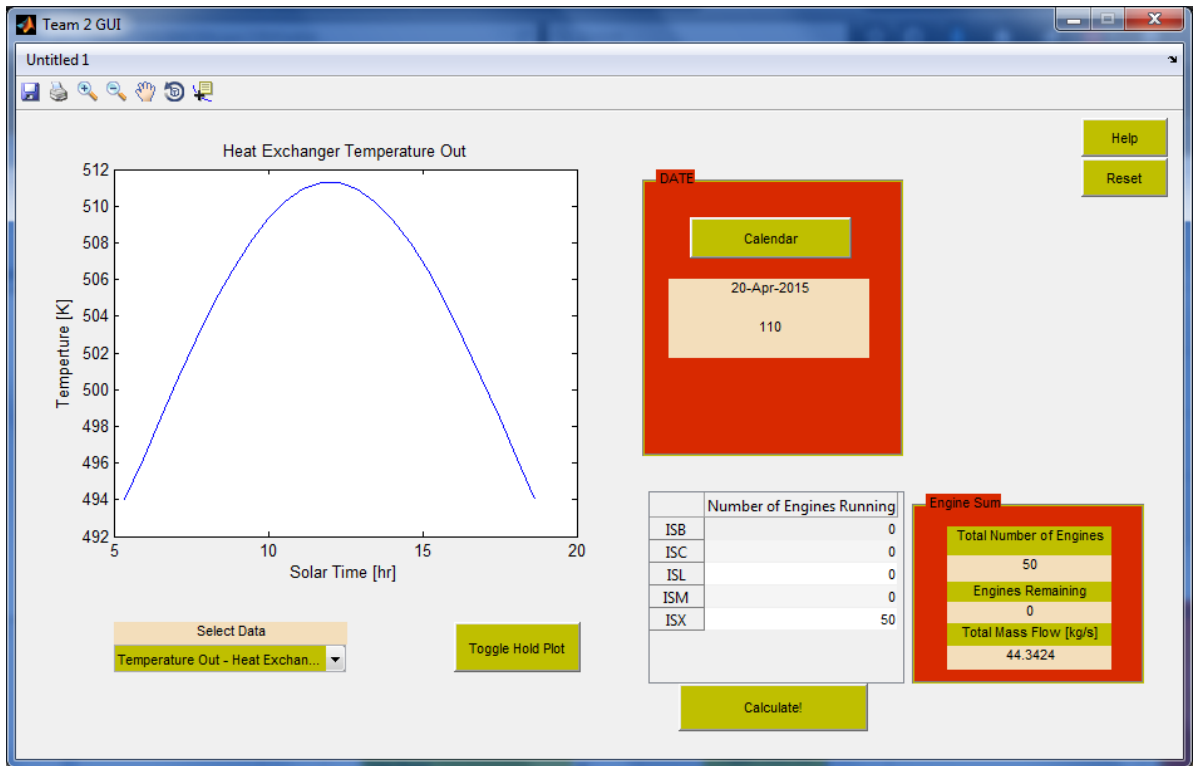
amount of engines running and what day of the year is the system running. The user can then select outputs such as the daily power output, annual power output, and the instantaneous and annual power savings for the system. In order to start the GUI, the user must load the MATLAB program provided by MathWorks. As with any MATLAB program all the files provided by the team must be located within a single working directory. The team has created a zip folder provided for this project that can be saved on the hard drive of the computer that contains all the necessary files. The following files are displayed in Table 24.

Table 24: Required Program Files

Coordinate_Time.m	Cost_Analysis.m	GUI_SIM.fig, GUI_SIM.m
Intersections.m	PipeLoss.m	SIM.asv
SIM.m	Siminstantaneous.m	Solar_Power.xlsx
Solar_Power_2.m	Solar_Thermal_Collectors.m	System_dimension.m
uisedate.m	year.m	

For simplicity and ease to the user, all required files are saved on a zip folder FSU\_Senior\_Design\_Team\_2\_Cummins\_Energy\_Savings.zip. Also, these files can be located in Appendix B of this report or downloaded from Team 2's website.

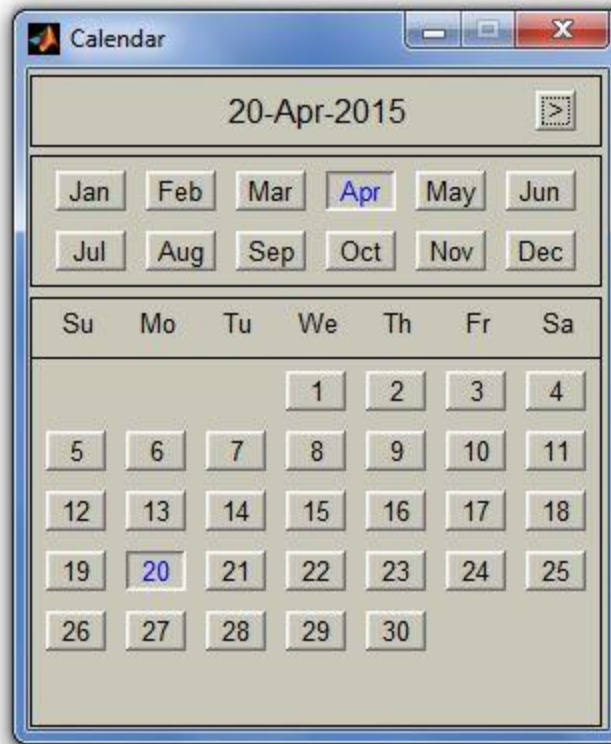
Team 2 built a guided user interface (GUI) using MATLAB. The GUI interface is displayed in Figure 12.



**Figure 12:** Guided User Interface Screen

The main parameters that the user can change are the date and the number of engines running. The user is then able to select various outputs from a dropdown list that shows the plot of the data across a time interval from sunrise to sunset for the selected date.

The program allows the user to specify the amount of engines running for 5 separate engine families: ISB, ISC, ISL, ISM, and ISX. The user clicks on the cell within the table for the entry they would like to edit, and enters a number for the engines running. The program works on the assumption that there will be a maximum of 50 engines running at a given time at the facility, so the sum of the engines running should not exceed this amount. However, the team will give a functionality of the engine cap size to exceed this up to 96 test cells in order to accommodate all test cells located at the CTC.



**Figure 13:** Calendar Input Popup

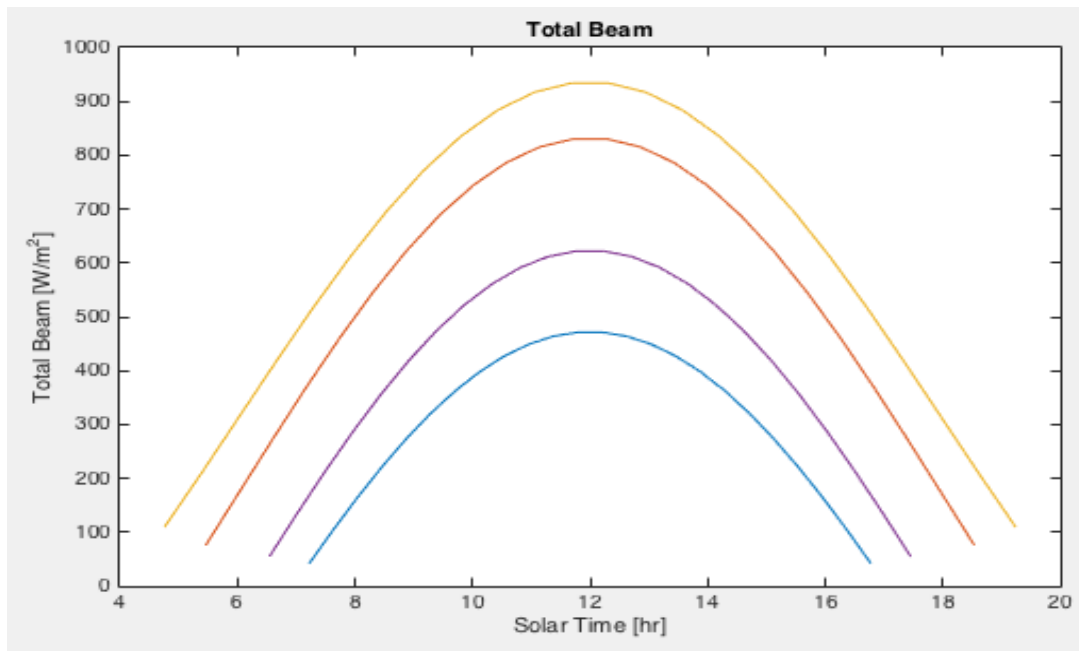
The second input that the user is allowed to vary is the date. In order to change the date, the user must click on the “Calendar” push button that brings up a calendar popup, shown in Figure 4, wherein the user can select any date for the year. The month and day can be selected by clicking on the toggle button for the desired date. In order to edit the year, the user can strike the “+” or “-“ key to either increase or decrease the year, respectively. It should be noted that the calendar will display the current date of the user’s computer upon the first time it is brought up. Once the date is selected, the user is brought back to the GUI, Figure 13, and the selected date is displayed below the “Calendar” push button. The user can repeatedly select different dates, and the date will be overwritten with each successive selection.

Finally, the plots seen in the graph can be selected from the drop down list located beneath the plot. The options available to user are: the daily power output, annual power output, system efficiency, instantaneous and annual power savings for the system. Along with these, the temperatures and other parameters of the system that vary throughout the day may be selected. Additionally, the cost analysis of the system can be selected. Based on the parameter selected,

the plot above the drop down list will change to display the output against the time interval from sunrise to sunset that is determined based on the date selected. The user then has the ability to save, print, zoom in, zoom out, pan, or place a data cursor.

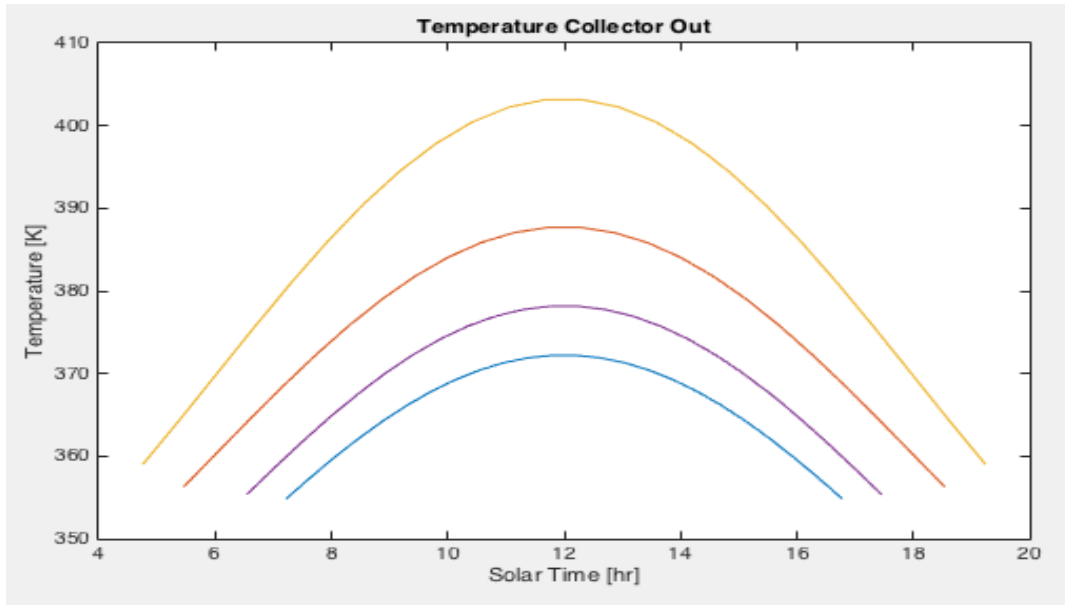
### 5.2.Final Design Results

To show the desired results we overlaid the plots using the toggle button for January 15<sup>th</sup>: blue, April 15<sup>th</sup>: red, July 15<sup>th</sup>: orange, and October 15<sup>th</sup>: purple for each desired output from our code. All equations used are listed in the methodology section. It should be noted that data is only plotted in the daytime for the GUI plots shown. The data should be constant at night, as the solar collectors aren't generating any energy. Also the engine arrangement is assumed to be a constant 50 ISX Engines. The first plotted variable is Total Beam. Here we see that July 15<sup>th</sup> has the highest total beam with a max at 900 W/m<sup>2</sup> as well as the longest time between sunrise and sunset of 14 hours. Next April 15<sup>th</sup> has a max insolation of just over 800 W/m<sup>2</sup> and a 13 hour day. On October 15<sup>th</sup> the max direct beam insolation of the day is 600 W/m<sup>2</sup> with a daytime of 11 hours. As expected, January 15<sup>th</sup> has the lowest max direct beam insolation of 460 W/m<sup>2</sup>, and a daytime of less than 10 hours.



**Figure 14:** Total Beam Plot

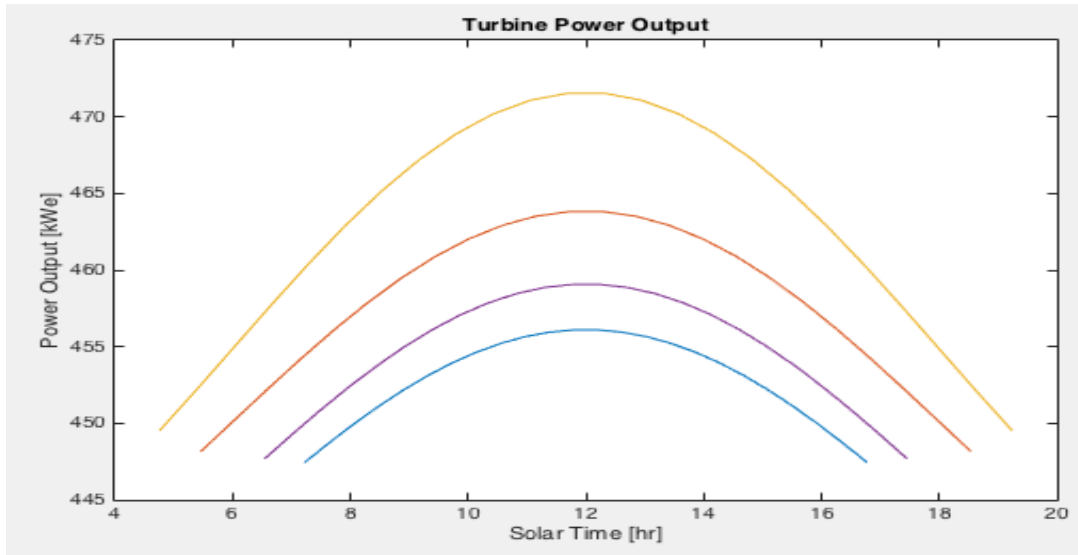




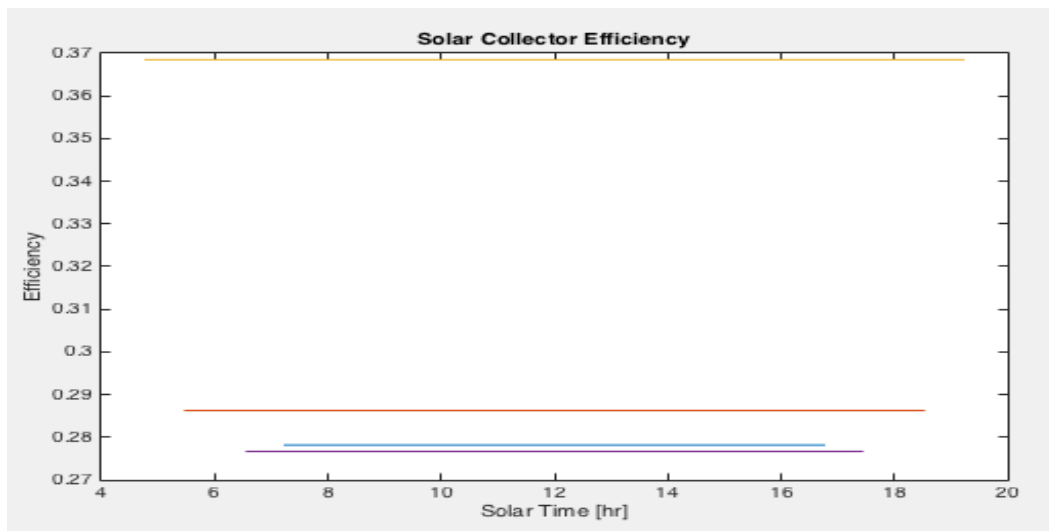
**Figure 15:** Collector Temperature Out

In Figure 15 one can see that the daytime or time between sunset and sunrise is once again 14 hours for July 15<sup>th</sup>, 13 hours for April 15<sup>th</sup>, 11 hours for October 15<sup>th</sup>, and 10 hours for January 15<sup>th</sup>. These daytime lengths will be the same in each graph for this section as the same four days are plotted in each graph. Figure 13 displays the outlet temperature of the Collector arrangement for the four days. Here we see a max  $T_{out}$  of 402 K on July 15<sup>th</sup>, 388 K on April 15<sup>th</sup>, 377 K on October 15<sup>th</sup>, and 371 K on January 15<sup>th</sup>. It can also be seen that for the days selected the maximum temperature rise from the Solar Collectors is 50 K.

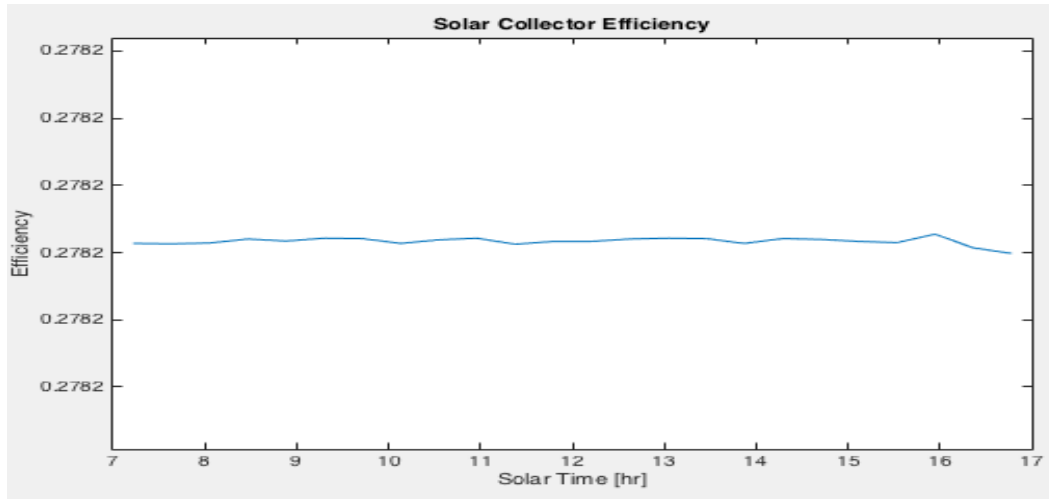
The next plot figure 16 displays the Turbine Power Out for each of the four days. In the figure it can be seen that the max power out of the turbine is found on July 15<sup>th</sup> to be 472 kWe, 463 kWe for April 15<sup>th</sup>, 458 kWe for October 15<sup>th</sup>, and 452 K for January 15<sup>th</sup>.



**Figure 16:** Turbine Power Output



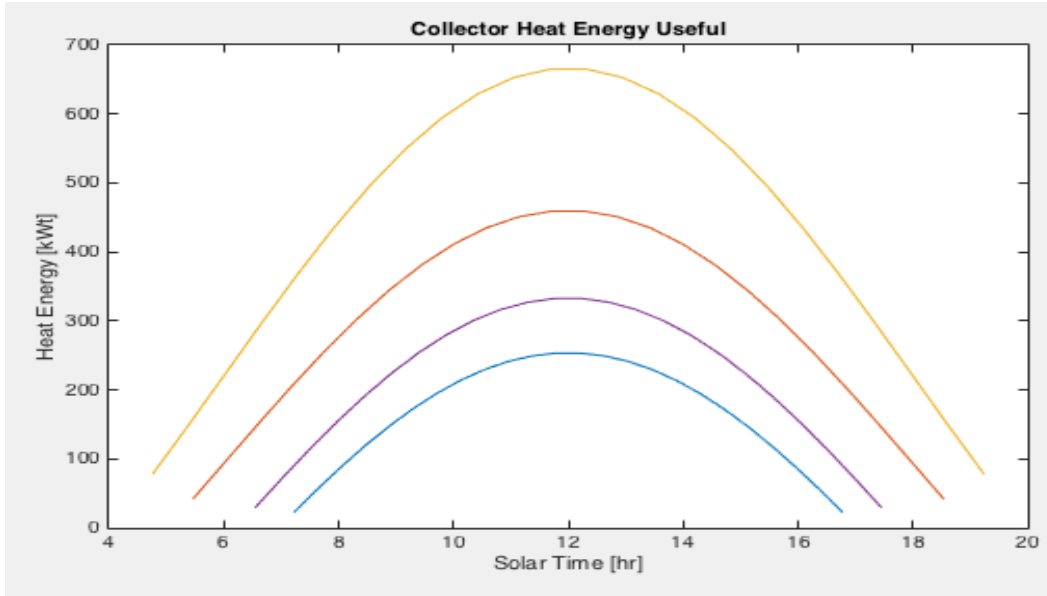
**Figure 17:** Solar Collector Efficiency



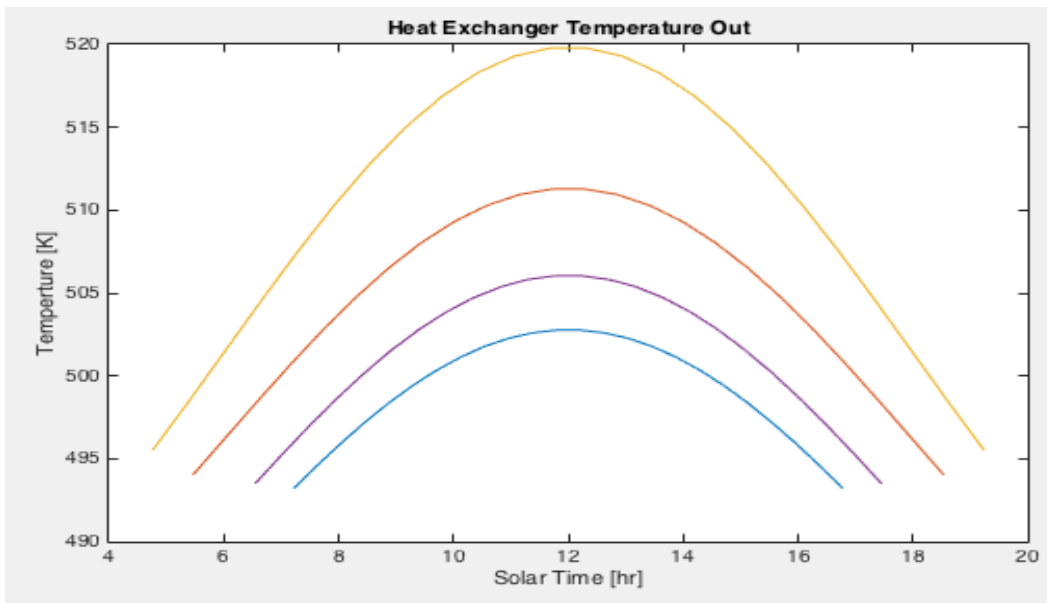
**Figure 18:** Solar Collector Efficiency January 15th

The next data point plotted in figure 15 for the 4 days is Collector efficiency. In the figure we see that on July 15<sup>th</sup> the collector nearly 37% efficient dropping to 28.6% on April 15<sup>th</sup>, 27.9% on January 15<sup>th</sup>, and 27.8% on October 15<sup>th</sup>. From Figure 17 it would seem that each collector efficiency calculated is constant for the day, but as one can see from Figure 18, each collector efficiency is not linear throughout the day but actually varies slightly in the y direction.

The next data set to be plotted is the collector heat energy useful. This data set is a measure of the heat energy transferred to the butane from the collector assembly with losses already accounted for. As expected the Collector useful energy is zero when there is no daylight. One can see a max Energy of 663 kWt on July 15<sup>th</sup>, 452 kWt on April 15<sup>th</sup>, 308 kWt on October 15<sup>th</sup>, and 214 kWt on January 15<sup>th</sup>.



**Figure 19:** Collector Heat Energy Useful

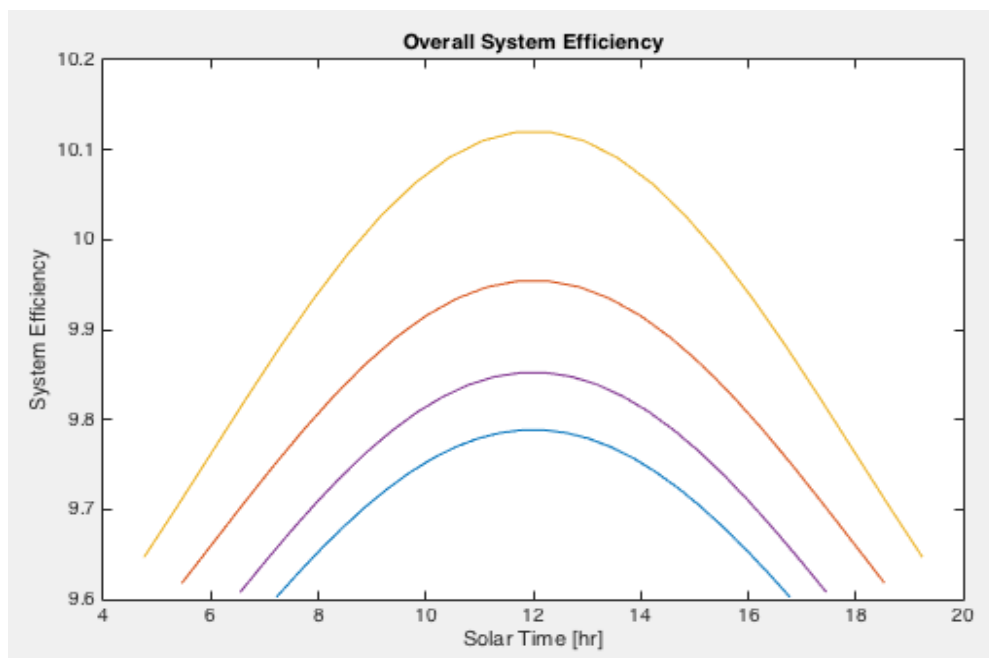


**Figure 20:** Heat Exchanger Temperature Out

The next plotted data set is the heat exchanger outlet temperature Figure 20. This temperature change is on top of the temperature change due to the solar collectors Figure 13 and represents the temperature change due to the exhaust gasses. In the figure we see a max temperature for July 15<sup>th</sup> of 520 K, 511 K for April 15<sup>th</sup>, 506 K on October 15<sup>th</sup>, and 503 K on January 15<sup>th</sup>. Here one can see that the temperature changes due to the solar collectors are not all

transferred in the heat exchanger because the daily temperature change in the heat exchanger is less than the daily temperature change in the solar collector but by only a small fraction.

The next 3 figures are used to display the overall system results. In Figure 21 below the overall system efficiency for the 3 days is plotted. At noon on July 31<sup>st</sup> we see the system efficiency rise up to over 10.1%, 9.94% on April 31<sup>st</sup>, 9.83% on October 15<sup>th</sup>, and 9.79% on January 15<sup>th</sup>. The reason we see a higher system efficiency in august is not only due to the high direct beam insolation (Figure 14) during summer but also during summer the ambient temperature is a great deal higher than winter, requiring less heat to be lost to the surroundings.

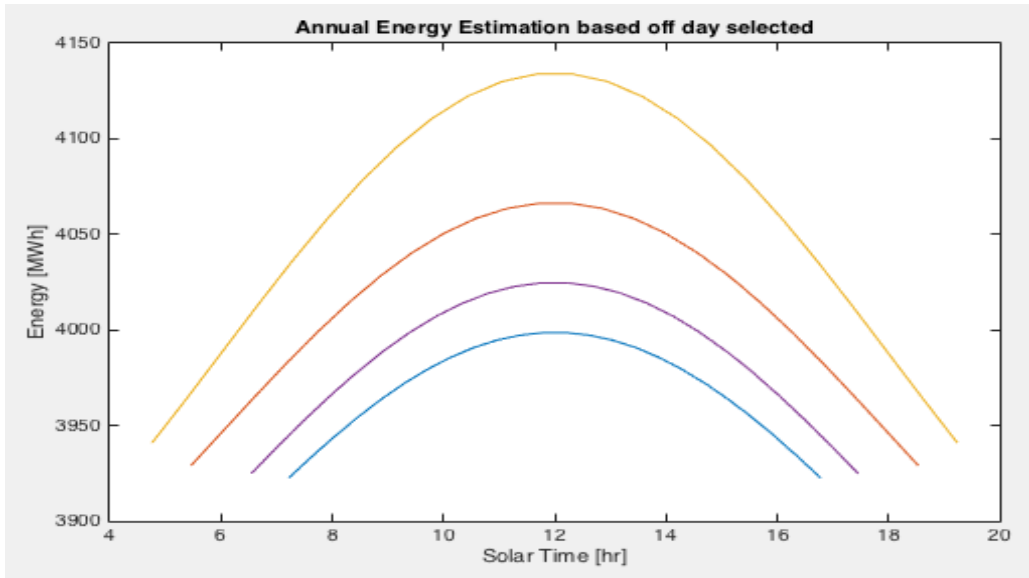


**Figure 21:** Overall ORC System Efficiency

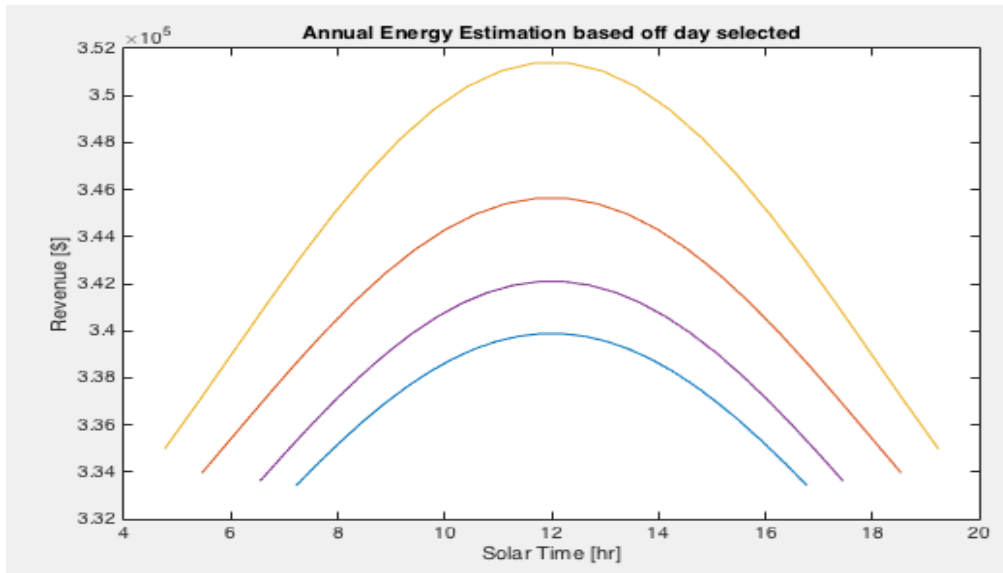
The next two plots are instantaneous estimations of the annual energy generated based off the given day and time. The first one Figure 22 shows the annual energy in MWh for the 4 selected days. In this plot we see a max energy of 4135 MWh if the entire year was noon on July 15<sup>th</sup>, 4063 MWh if the entire year was noon on April 15<sup>th</sup>, 4018 MWh if the entire year was noon on October 15<sup>th</sup>, 3992 MWh if the entire year was noon on January 15<sup>th</sup>.

Figure X10 is similarly to Figure 20 an instantaneous estimation. Figure 21 displays the Annual Revenue Generated at 8.5 cents per kWh given to us from Duke Energy in Columbus Ohio. Here we see a max annual revenue on July 15<sup>th</sup> at noon of approximately \$351,500.00,

\$345,750.00 at noon on April 15<sup>th</sup>, \$342,000.00 at noon on October 15<sup>th</sup>, and \$339,800.00 at noon on January 15<sup>th</sup>.



**Figure 22:** Annual Energy Generated instantaneous (at given time and day for whole year)



**Figure 23:** Annual Revenue Generated Instantaneous (at given day and time for whole year)

Table 25: Total Energy Table

Annual	GJ	kWhele	\$generated
ORC	14222	3,950,587	\$335,799.91
Solar Panels	Implemented in the ORC now		
Chillers			\$223,095.00
Wind Turbine	9389	2,608,076	\$221,686.50
total	23,611	6,558,664	\$780,581.40
savingsrate	4.35%		

Table 25 above displays the total results for each system we plan on implementing. The solar panels are neglected as we switched to solar collector to be implemented in the cogeneration ORC. This change will be further described in design improvements section later. The total savings comes out to 23,611 GJ, 6,558,664 kWh, and \$780,581.40. This comes out to 4.35% of the CTC's annual consumption.

### 5.3. Projected Maintenance

The regular maintenance section is intended for the maintenance of the exhaust system power generation system and each individual component that makes up the system. This is more useful for the user instead of maintaining the simulation program which comes down to maintaining a computer. The most important maintenance is performed on the heat exchanger, the cost of maintenance increases exponentially with weeks between cleanings. It is suggested that the heat exchanger is cleaned every 10-12 weeks to maximize efficiency and minimize costs. In this range the cost of each cleaning is in the \$50,000 to \$60,000 per cleaning totaling to an annual cost of about \$275,000.

The Solar Collectors also require maintenance and cleaning. As earlier stated, we have selected the AE series Collectors specifically the AE 3000 system. These collectors operate best when the glass is clean and unobstructed. If the solar collectors become dirty, wash them with mild soapy water and rinse. Remove any branches or leaves that do not naturally fall off or are blown away by the wind. This debris should be minimal due to the fact that the solar collectors are located on the roof. Cost of this cleaning is minimal, but must be monitored throughout the year. These duties can be added to any worker already on staff, which does not require hiring new staff members.

Team 2 has selected the Corken 6 inch single stage horizontal reciprocating Compressor. This compressor uses 7 quarts (6.6 Liters) of oil as lubricant, which must be monitored and changed every 2200 hours of operation along with filter 4225. Piston rings and piston ring expander also must be replaced every 4400 hours of operation (2200 hours if non-lubricated). Packing cups, spacers, O-rings, springs bushing, bearings, and retainer rings would also need multiple replacements within lifespan of the machinery [30]. A more detailed chart is located in Appendix A. Another maintenance requirement for the compressor and turbines is the monitoring of their blades. The compressor and turbine system should be inspected every 8 weeks for any fatigue failures, corrosion effects and regular maintenance. Even though n-butane is a non-corrosive material, routine maintenance checks are required. Many other components can become corroded or cracked and must be monitored. A bi-monthly inspection will consist of a full system cleaning and component analysis. This service requires a team of field engineers to ensure a proper inspection was conducted. The compressor and the turbine can be inspected at the same time to minimize the amount of down time experienced the estimated maintenance cost for the turbine system equates to be \$35,555.28 [31].

## **6. Environment, Reliability and Economic Analysis**

### *6.1.Environmental Analysis*

There are associated health concerns when harnessing and rerouting exhaust gasses. Some molecules that need to be regulated include carbon dioxide, particulate matter, and methane hydrocarbons. Carbon dioxide and methane hydrocarbons are hazardous because they destroy the Ozone layer, which protects us from harmful UV rays. Particulate matter on the other hand is harmful because it can kill you if entered into the body through normal respiration. Because of these concerns Team 2 will calculate the emissions of these harmful by-products. Team 2 calculated the highway speed the Cummins ISX15 600 consumes fuel at a rate of 11.67 G/h. According to emissions data from a study conducted by Cummins and the American Transportation Research Institute during this hour running all 96 ISX engines would produce 1,174 kg of carbon dioxide, 1.167 g Particulate Matter, and 130 g of Methane Hydrocarbons. These pollutants are well under the EPA regulations and would be found, analyzed, and disposed before recirculating into the exhaust gas heat converter.



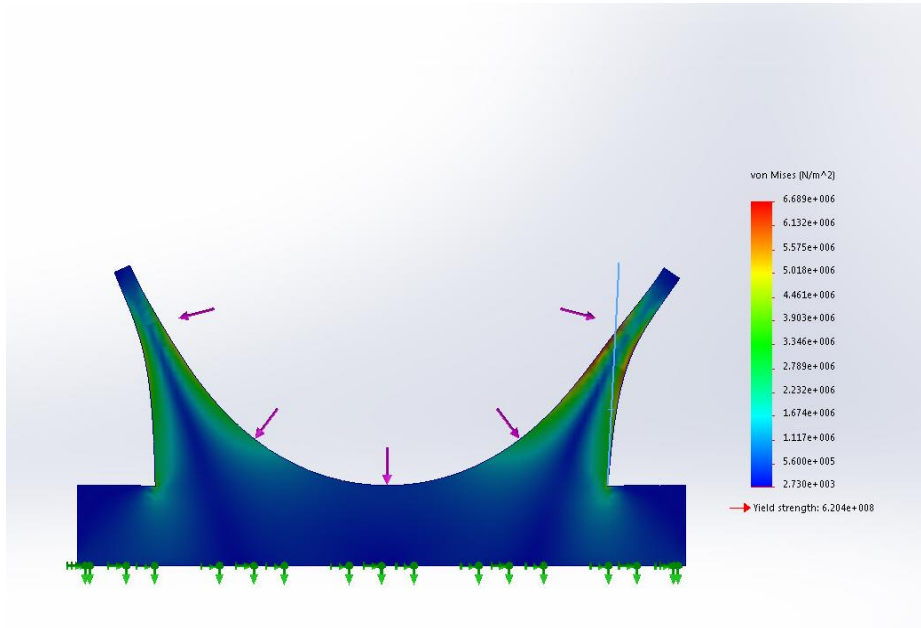
## *6.2. Reliability Analysis*

In Team 2's system, there are several subsystems that are included: heat exchanger, power generation, solar collectors, compressor, and storage tank. The heat exchanger is the system that will transfer the energy from the exhaust gases of the testing engines to the working fluid. The shell of the heat exchanger has been designed to be able to handle a maximum pressure of 1MPa as well as being correctly insulated so there is no heat loss to the exterior through the shell. The shell is also made out of steel and with proper maintenance, will be able to withstand the corrosion due to the impurities of the hot exhaust gases coming out of the engine. The pipes of the heat exchanger are made out of brass because it is cost effective, a good thermal conductor and will have no problem withstanding the pressure and temperatures that the heat exchanger is capable of achieving. The baffles are made out of steel and their main purpose in the heat exchanger is to redirect the hot exhaust gases in the heat exchanger in a path that will maximize the efficiency of the heat exchanger. The turbine that was chosen for the ORC is the Siemens SST-060; this turbine is able to handle the temperatures and pressure of our system without any problem. The operating pressures and temperatures of Team 2's system do not reach the maximum allowable pressures and temperature of the turbine, 13.1MPa and 510C respectively. The solar collectors will help to improve the efficiency of the system by increasing the inlet temperature to the heat exchanger to achieve a higher output temperature and therefore a higher efficiency, because of the large amount of tubing through the solar collector the head loss through the piping has been considered in order to have no issues when the system is running.

One of the possible failures of the system would be that the working fluid is highly flammable making a potential for explosion. The system is well insulated from any outside interaction and greatly reduces this risk. Also, the operating conditions throughout the system (1.2MPa and 145C) maintain n-butane as a superheated vapor throughout the system. The turbine is made to work with ORC working fluids. The working parameters through the system are well within the stable conditions for n-butane.

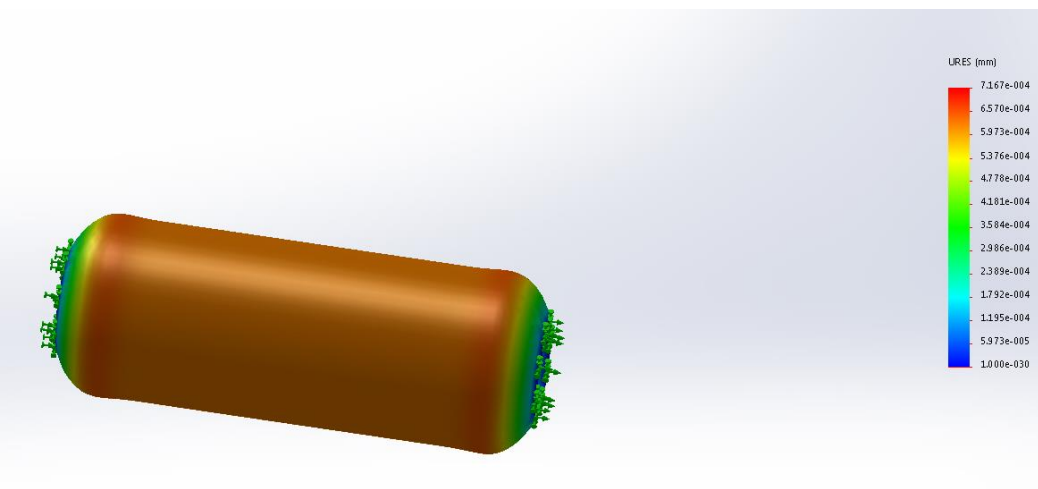
FEA analysis was performed in the weakest parts of the system which were the support of the heat exchanger due to the high loads it must withstand, and the tubes inside the heat exchanger because they are made out of brass, which is relatively weak compared to steel. The results of the FEA analysis were satisfactory and each part had a factor of safety higher than 10. Figure 24 shows the stress analysis performed to the supports of the heat exchanger. The

supports are made out of one inch steel, the stress analysis conducted with the entire weight of the heat exchanger (45Mg) concentrated on the support system. In reality, the weight will be distributed throughout the support system.



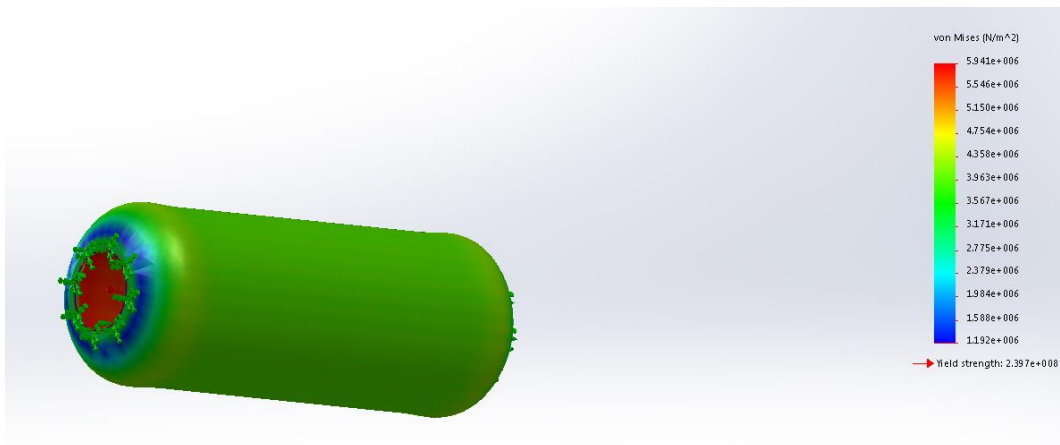
**Figure 24:** Stress Analysis for Support Beam

Figure 23 shows the material displacement in a brass pipe with the smallest diameter in the system experienced to the maximum pressure experienced by the system of 1.0 MPa. The magnification of the picture is more than two thousands.



**Figure 25:** Pressure Analysis for Heat Exchanger Pipe

Figure 26 shows that the maximum material deflection experienced by the piping is  $7.17 \times 10^{-4}$  mm deflection. The piping does not deform from the pressure in the piping which proves that the system is reliable to withstand the pressures. The final analysis that was conducted was a stress analysis on the piping material to determine if any plastic deformation or fracture occurred caused by the pressures.



**Figure 26:** Stress Analysis on Piping

As Figure 26 displayed, the stresses experience by the piping material does not exceed the yield stress of the brass and can be considered a reliable system.

### 6.3. Economic Analysis

The initial cost of most components was determined by contacting the manufacturer to determine the price. The heat exchanger cost was determined using an initial cost equation 18 [31].

$$C = 1800D^{2.5}L + 350DL + 1950 \quad \text{Equation (18)}$$

Where L is the length and D is the diameter of the shell side of the heat exchanger. This equation incorporates the initial cost of the shell, tubing, and an initial cleaning fee. The initial prices for each component are listed in Table 26.

Table 26 – Initial Cost of Individual ORC Components

<b>Component</b>	<b>Price</b>
Heat Exchanger	\$ 591,950.00
Turbine	\$ 500,000.00
Compressor	\$ 450,000.00
Storage Tank	\$ 1,965.00
1600 Gal of n-Butane	\$ 1,040.00
Solar Collector	\$ 400,000.00
Piping	\$ 2,000.00

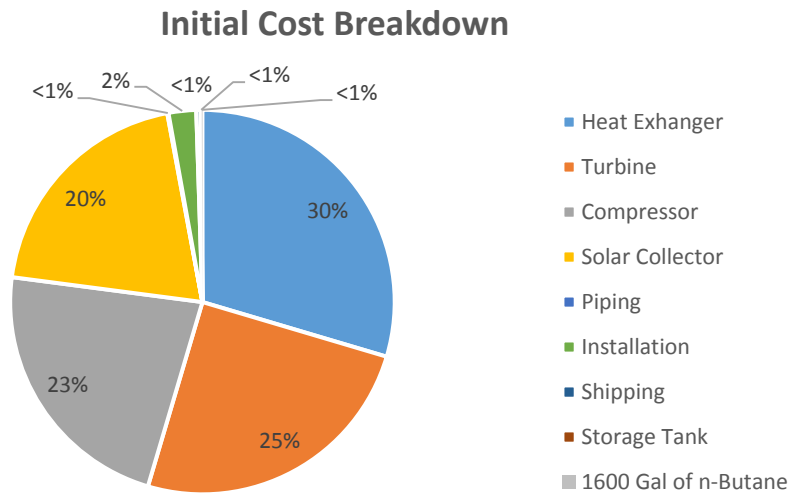
After calculating initial costs on all the components an analysis of shipping and installation costs was completed. The shipping costs were done using the uship website where the distance and weight of the shipped component were used to provide an average shipping cost [32]. The initial location for each component was taken from the company websites for the turbine, compressor, and storage tank. Since the heat exchanger is custom and needs to be manufactured our sponsor provided information of a nearby shipyard where it would be possible to manufacture [33]. The piping location was found by using a local pipe distributor in Indiana. The solar collector location was determined by using a solar collector manufacturer, Nusun Solar who are located in Indiana. The shipping costs are shown in Table 27.

Table 27 – Shipping Cost of Individual Component

<b>Component</b>	<b>Shipping Cost</b>
Heat Exchanger	\$ 5,700.00
Turbine	\$ 230.00
Compressor	\$ 512.00
Solar Collector	\$ 121.00
Piping	\$ 173.00
Storage Tank	\$ 445.00
1600 Gal of n-Butane	\$ 143.00

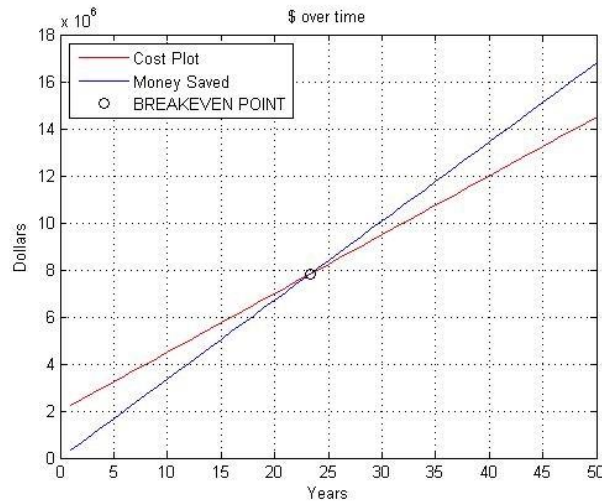
Installation costs were approximated by assuming the time and man power needed to complete the installation. The time for installation was assumed to be one month since the

system is so large and it was determined that one engineer and a team of ten skilled workers would be needed to install the system. Assuming that they would be working a 40 hour work week for 4 weeks at an average price per hour the total installation cost was determined to be \$46,400 [34]. Figure 27 shows a pie chart of all the initial costs. These costs total to \$2,000,623.00.



**Figure 27:** Initial Cost Breakdown of all costs

Finally the maintenance cost was determined by using data from average maintenance costs for the turbine and compressor. The heat exchanger was maintenance was calculated by using the equations from “Optimizing the cleaning of Heat Exchangers. The annual maintenance cost was determined to be \$250,000 a year. The initial and annual costs were plotted on the same plot as the money gained from the generated energy of the system in Figure 27. Figure 28 shows a breakeven point of 23 years. After this time the system will be generating a profit for Cummins. This shows that economically the overall system is a feasible option for Cummins.



**Figure 28:** Time (years) to Payoff Initial and Annual Costs

## 7. Improvements

Many Improvements have been made throughout the year to improve our design. This section contains the improvements Team 2 did to their initial system design and also provides some further design improvements that could be implemented if needed.

### 7.1. Design Improvements

The first being the upgrade from solar panels to cogeneration with solar collectors and exhaust gasses together in the ORC. If paired with our proposed absorption chiller trigeneration will be implemented. The chiller change from their current chiller to a modern absorption chiller lowers annual operation and maintenance costs an estimated \$223,095.00.

Another component that has undergone various changes is the heat exchanger. The heat exchanger dropped in size to reduce initial cost and mass. The original design was made to save Cummins 10% of their energy footprint. The original weighed 20 tons and was the size of a building. The new design is made to reduce initial costs, weighs just under 10 tons, and is only one quarter of the size. Another design improvement, the cleaning time of the heat exchanger is optimized to improve heat exchanger efficiency and lower annual operating costs. We find according to a fouling model created by Conco Consulting Corp. that 10-12 weeks is the proper time between each heat exchanger cleaning. During cleaning three pipe cleaners are implemented: plastic molds (pigs), brushes, or metal cleaners (2-6 blades). These cleaners are propelled by water at 300 psig. The Heat exchanger can also receive a more intense clean using a

hydrodrill or other compressed air driven cleaner. It is recommended that the heat exchanger is hydrodrilled once annually (approximately every 5 cleans).

Another design improvement was made regarding wind turbines. The wind turbines were changed to direct drive to avoid mechanical energy losses found in other 3 proposed generation methods. It is found that the cost of the structure required to support the direct drive generator is far less than the losses from transferring the energy to the ground.

As stated in the background, many working fluids were considered for the Cogeneration cycle proposed. At first we selected superheated water vapor but found that it requires a relatively high operating temp to heat water enough to superheat it in the system. Due to this relatively high boiling temperature (100C) we chose to switch our working fluid to n-butane (-0.4C). The Auto-ignition temperature of n-butane is 430 C [35], which the ORC we plan to implement will never reach due to efficiency limitations and engine exhaust temp of 400F.

## 7.2. *Further Improvements*

During the year we have made a vast plethora of design improvements some of which are mentioned above. Now we will look at some design improvements that weren't made but should be considered in the future. The first possible improvement is the solar collection system could be expanded if taken off the roof. With this change the roof length and width limitations will no longer limit collector assembly size and much larger sized assembly could be implemented. We chose to keep the collectors on the roof to limit heat and pressure losses in the pipes. As of now the ORC implements flat plate Collectors. A solar concentration system could be implemented to improve Insolation seen by the collectors and in turn heat transferred to the n-butane. We chose not to pursue this method due to the high initial costs of solar concentrators.

Another design improvement is the implementation of energy storage. Energy storage could also be utilized to store some of the energy generated by the sun to be added to the system at night. This process uses a separate working fluid, usually a salt, to store the heat energy. Due to the requirement of another heat exchanger, storage tank, and working fluid we chose not to pursue this method.

Another improvement is increasing the ratio of Heat Exchanger Efficiency vs. cost, as technology improves peak component efficiency is constantly raised as price usually drops. In the future better, cheaper, more efficient heat exchangers may be designed.

One easy implementation as stated earlier running the chiller through the ORC system could be employed to use tri-generation to generate energy. The absorption chiller is presently being used separately from the ORC.

The last recommended future idea is to improve the expansion process (turbine). One option could be to design a scroll expander than can replace the turbine. The scroll expander, or other new design could be theoretically more efficient and could improve efficiency and/or lower costs.

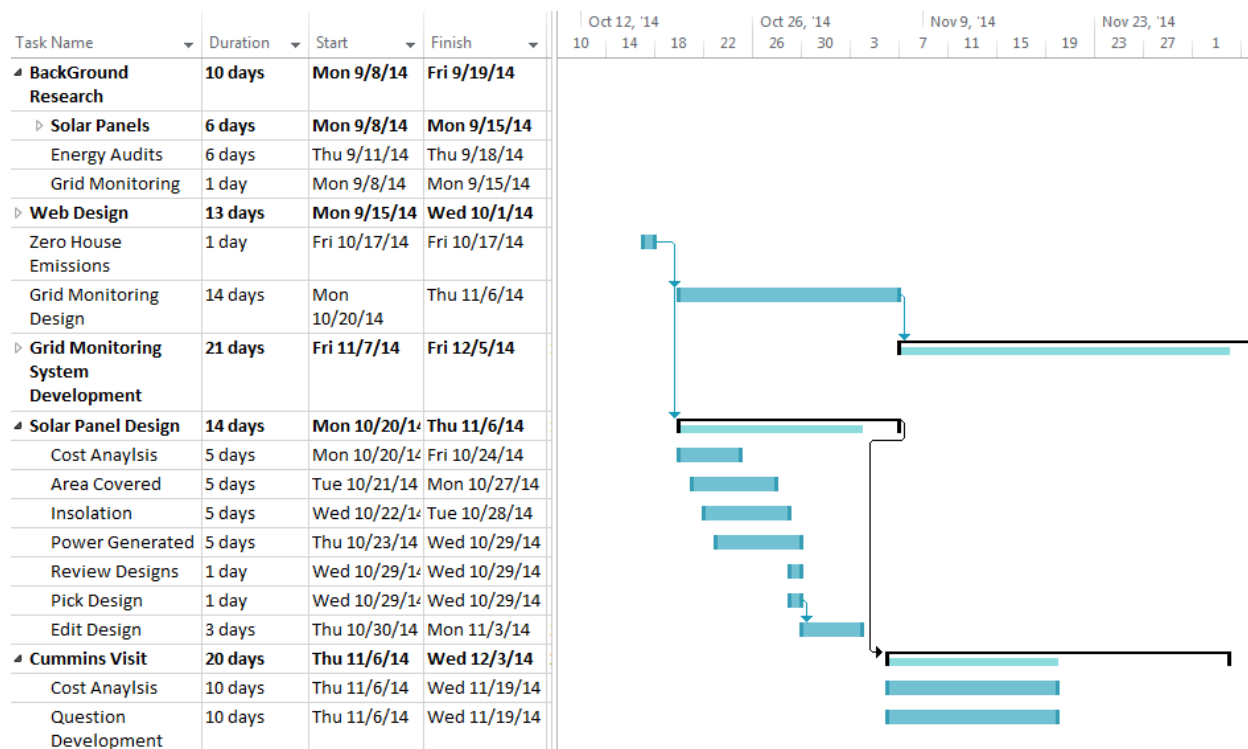


## 8. Project Management

The following section contains a description of Team’s 2 evolution of scheduling throughout the project, resources used, procurement of product and communication between sponsor, advisor and course facilitator.

### 8.1. Scheduling Adaptations

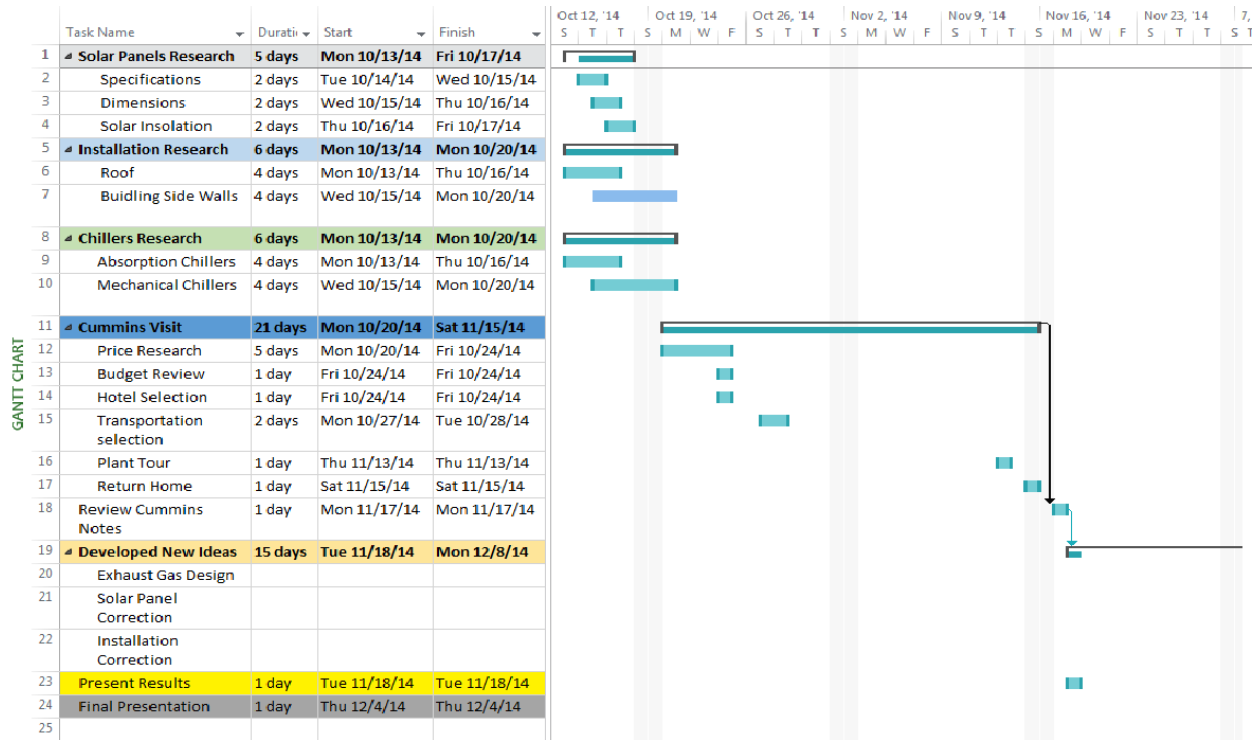
As the project progressed throughout the year, the critical path varied. This was caused by the varying final product deliverables and taking into account obstacles that were not initially thought of. Figures 27-30 show how each the Gantt chart changed throughout the completion of the project.



**Figure 29:** Fall 14 Initial Gantt Chart

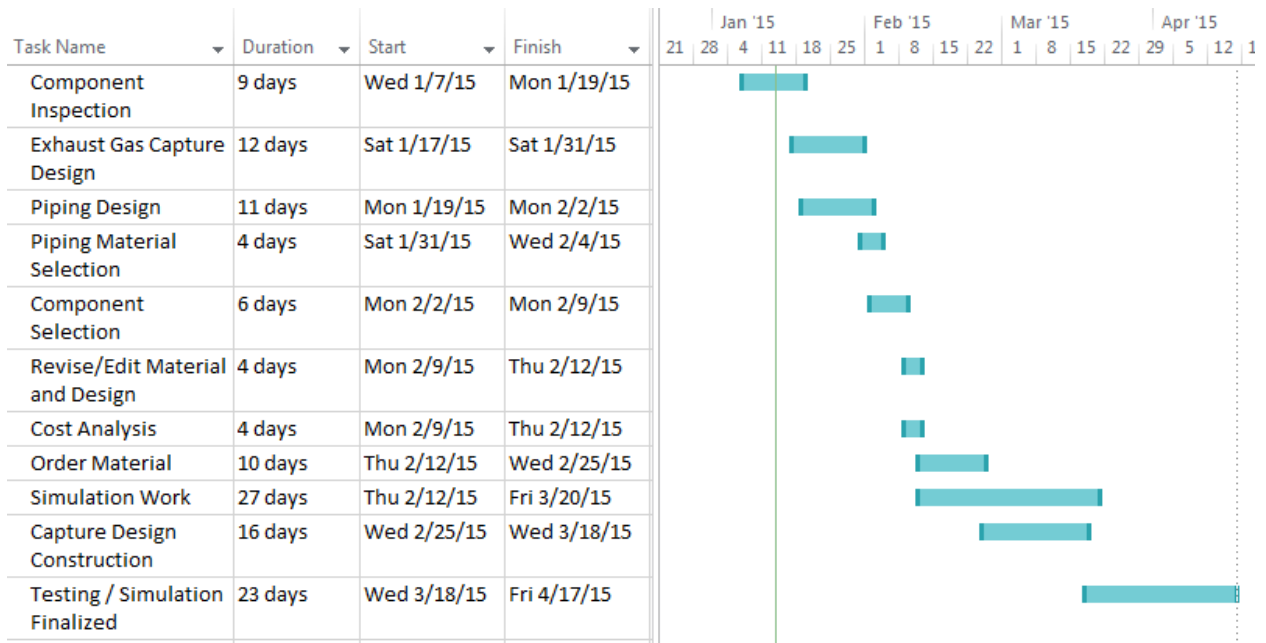
Figure 29 was the initial schedule Team 2 set in the beginning of the semester which divided up different aspects of the project from the beginning to the end of the semester. This was an ideal schedule when the project was first established and no real research was conducted. The main goal was to visit the CTC and gather as much information as possible before the visit. The following figure shows a schedule after a significant amount of research was conducted and some of the tasks were completed from the first Gantt chart. Team 2 had developed 5 ideas that were worth doing further research in as mentioned in the section 3.1 of the report. As Figure 29

shows, team 2 did research on the 5 ideas in order to present their energy saving ideas to Dr. Hays on November 14<sup>th</sup> 2014. After the CTC trip, Team 2 were able to pin point which idea was going to save Cummins the most energy (exhaust power generation) and changed their focus on trying to develop a system to save the CTC energy.

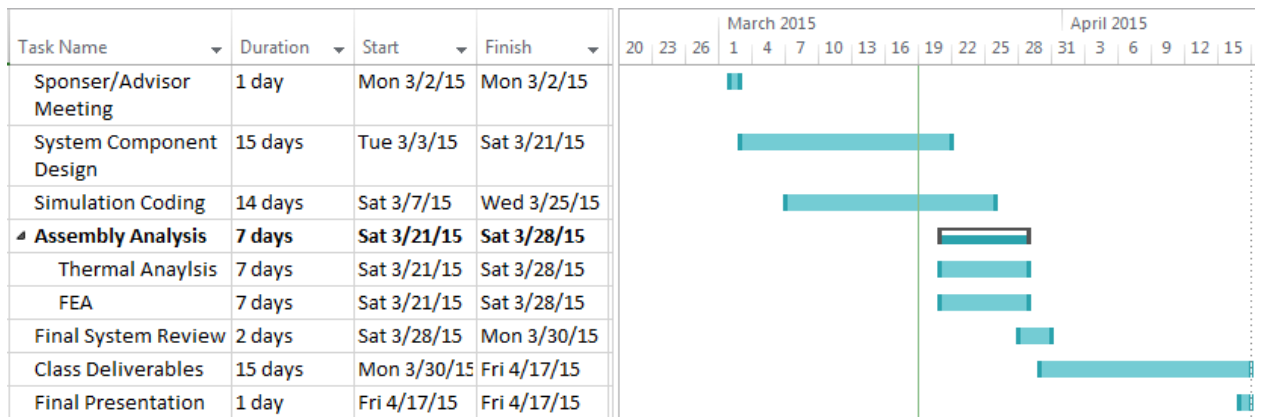


**Figure 30:** Fall 14 Mid-Semester Gantt Chart

Figure 30 shows the work conducted and the future work that was planned on being conducted for the prototype system that was going to be implemented. Eventually everyone involved on the project agreed that a simulation package for an exhaust capturing system was more beneficial for Cummins and all focus was diverted towards the development of an exhaust capturing system as can be seen in Figure 31.



**Figure 31:** Spring 15 Initial Gantt Chart



**Figure 32:** Spring 15 Final Gantt Chart

The final Gantt chart displays the work team 2 did in the last 10 weeks of the semester. From March 2<sup>nd</sup> to April 17<sup>th</sup> 2015, Team 2 focused on developing a simulation package that would take into account all the research conducted and assumptions generated in order to develop a theoretical power generation system.

The initial schedule developed by Team 2 as can be seen in Figure 29, was completely different from the final schedule team 2 constructed as shown in Figure 32. The project evolved a couple of times which Team 2 were able to adapt and create a schedule to allocate the necessary time to

complete the project. Team 2 were able to complete the project by generating a simulation package.

*8.2.Resources*

The resources that Team 2 had to their disposable were the S.T.R.I.D.E lab, College of Engineering (COE) machine shop, COE computer labs and MATLAB tutorials. These resources were maximally utilized throughout the entire year of the project. Since the final deliverable of the project was to code a fully functional Simulation package as the one presented previously, the COE computer labs and MATLAB tutorials were used. Other resources that team 2 utilized the most was the Florida State University (FSU) library online database. Since the project was based dominantly on theoretical ideas and ideas to generate energy saving problems, the FSU library database was used extensively to find the latest research conducted in this area. The FSU database supplemented with textbooks available in the COE library was where most of the information Team 2 were able to find. Also, google scholar was another resource utilized which assisted in finding the latest research in renewable energy, energy audits, power generation and the most up to date components for the power generation system.

*8.3.Procurement*

Team 2 did not have a final product that needed fabrication because the final deliverable was a simulation package completely coded in MATLAB. Since Team 2 had access to MATLAB through the COE, there was no money spent on purchasing the program. The only amount of money that was spent on this project was the traveling cost to Columbus, Indiana. Team 2 had a budget of \$2,000 at the beginning of the semester and \$1,380 were used for all of Team 2’s members to travel to the CTC. Table 28 is a summary of the amount of money spent for the trip to the CTC.

Table 28 - CTC Trip Cost

<b>Item</b>	<b>Total Cost</b>
Food (\$27 a day for 5 people)	\$405
Car Rental	\$550
Gasoline	\$125
Hotel	\$300

Food was budgeted at \$27 a day for 5 people which equated to \$405. A minivan was rented from Enterprise Rent-A-Car for 3 days and unlimited miles. Team 2 stayed at the Days Inn for \$75 dollars a night for two rooms. A total of \$1,380 was spent on the 3 day trip to Columbus, Indiana.

#### 8.4. *Communication*

Communication throughout Team 2 was handled excellently. There was constant communication between the members of Team 2 throughout the entire development of the project. Team 2 members were constantly updated on each other's progress and completion of different parts. Team 2 was in constant communication through Facebook messages, emails, text messages, phone calls, meeting reminders, daily meetings and the occasional weekend retreat. Team 2 proved to be a fully functional group with impeccable communication. Team 2 members were also very understanding of the other members work load and schedules. Unfortunately, the communication between the sponsor, advisor and course facilitator was not met with the same enthusiasm.

Team 2 did not have the same communication success with the sponsor, the advisor and the course facilitator. In the entire fall semester, Team 2 were in communication with the sponsor a total of 5 times to schedule a trip to visit the CTC and provide any updates Team 2 accomplished. Since the sponsor was located in Columbus Indiana, telephone and emails were the dominate form of communication. In the spring semester, communication increased significantly. Team 2 scheduled weekly meetings Dr. Hays was always available to meet and provided valuable input on completing the project.

Team 2 were in constant contact with the course facilitator, Dr. Gupta, in the spring semester. Dr. Gupta was available throughout the entire year during his office hours, emails or by appointment. Team 2's communication with Dr. Gupta was minimal in the beginning of the project. Team 2 were unsure of the direction of the project and should have consulted the course facilitator, but, were reluctant to do so because Team 2 were trying to figure out the problem. Team 2 realized that help was required and started initiating communication with Dr. Gupta on a weekly basis by emails, visiting Dr. Gupta's office hours and scheduling appointments. Also, any emails that were shared with Dr. Hays and the advisor, Dr. Ordonez, Dr. Gupta was cc'd in order to keep him up-to-date with the progress of the project. It should also be noted that Dr.

Gupta's TA's were available by appointment and assisted Team 2 whenever they had time to do so.

Team 2 were unfortunately not in constant communication with the advisor Dr. Ordonez. Team 2 understands the busy schedule Dr. Ordonez had throughout the semester. Dr. Ordonez was available to meet after class for a few minutes so Team 2 had members meet with him after classes. Team 2 also tried to initiate weekly meetings with Dr. Ordonez, which were beneficial at the beginning of each semester. Once school was in mid semester, meetings were cancelled. Team 2 tried to email and call Dr. Ordonez phone office but did not receive a reply. Communication between Team 2 and the advisor could have been better. Team 2 tried to initiate communication with Dr. Ordonez but were not met with the same enthusiasm.

## **9. Lessons Learned**

This senior design project was challenging, eye opening and overall a good learning experience for Team 2. The first law that should never be forgotten is Murphy's Law. A major take away that this project taught Team 2 was how vital communication is to the success of any project. With so many people involved with a project, everyone has their own perspective on completing the same task. In order to avoid any confusion and misunderstanding between team members and supervisors, communication must be there. Another important lesson learned by Team 2 is to clearly define and establish goals. Goals should be specific and not ambiguous. By having precise goals, a team schedule can be accurately generated and easy to follow. A well-developed schedule will keep the progress of a project or any task easier to accomplish because deadlines can be established and follow. Overall, the experience was beneficial to Team 2's professional career. If the purpose of Senior Design was to simulate real world experience, then the message was received. By working on a senior design project, more specifically the Cummins Energy Savings, Team 2 has gained valuable experience that will be forever remembered.

## 10. Conclusion

The comprehensive assessment detailed in this report covered many aspects of the design process for this project. As mentioned previously, the work done in the first semester for determining energy saving ideas laid the ground work for the progress that was made in the final semester. The work done this semester focused primarily on designing a method to harness the energy of the exhaust gases from the diesel engine testing at the CTC. Additionally, Team 2 was tasked with simulating the results of the research, analysis, and design of a heat recovery system in a Guided User Interface (GUI). By recovering the waste heat from the engine testing, Cummins has the potential to save upwards of \$350,000 annually if they choose to implement the Organic Rankine Cycle with Solar Collector pre-heaters that Team 2 has designed and simulated in the MATLAB GUI. The team found that for the design parameters chosen (the mass flow of 50 ISX engines running for an entire year), the payoff factor would be slightly less than 20 years. This is not an optimal payoff period of time, and this long payoff period can be attributed to the fact that the system is generally 8-10 % efficient overall, which is only slightly lower than the projected 15% efficiency of smaller size ORC systems provided by companies like Siemens. However, the fact remains that the ORC system that Team 2 has designed will help reduce the overall energy consumption by a calculated 1-2%. While this falls short of the initial goal of saving Cummins 10% of their energy consumption, combining the savings from the system designed by Team 2 with the ideas from the first semester helps to drive the energy reduction percentage to a total of 4.35% of the overall energy consumption of the facility. In order to reach this massive amount of savings, a steep initial investment is required. For the system designed by Team 2, this equates to just over \$2 million in initial costs, and an annual cost of \$250,000. Basic summation reveals that installing this system will save the CTC about \$100,000 a year, but the benefits will only begin to offset the high cost in about 20 years. Cummins, Inc. has already implemented many different energy savings ideas at the Cummins Technical Center, and had extremely successful results in reducing their energy consumption drastically over the past 5 years. Ultimately, after an exhaustive analysis of the many different aspects of ORC systems, and the resultant calculations from Team 2's Simulation Package, pursuing this course of action is not an optimal solution for the present time. Since this method of heat recovery is a relatively new field being pursued for alternative energy and energy saving methods, it is highly likely that there will be further enhancements made to the ORC system that

will make it viable in the near future. There is the possibility that with further enhancements to the design, the ORC system could have a payoff factor within 12-15 years. Therefore, it is strongly suggested that the facility continues to monitor the development of ORC technology over the next 3-5 years, and that further analysis is done using the simulation materials provided by Team 2 to assist in determining the optimum time to install the ORC system with Solar Collector pre-heaters.



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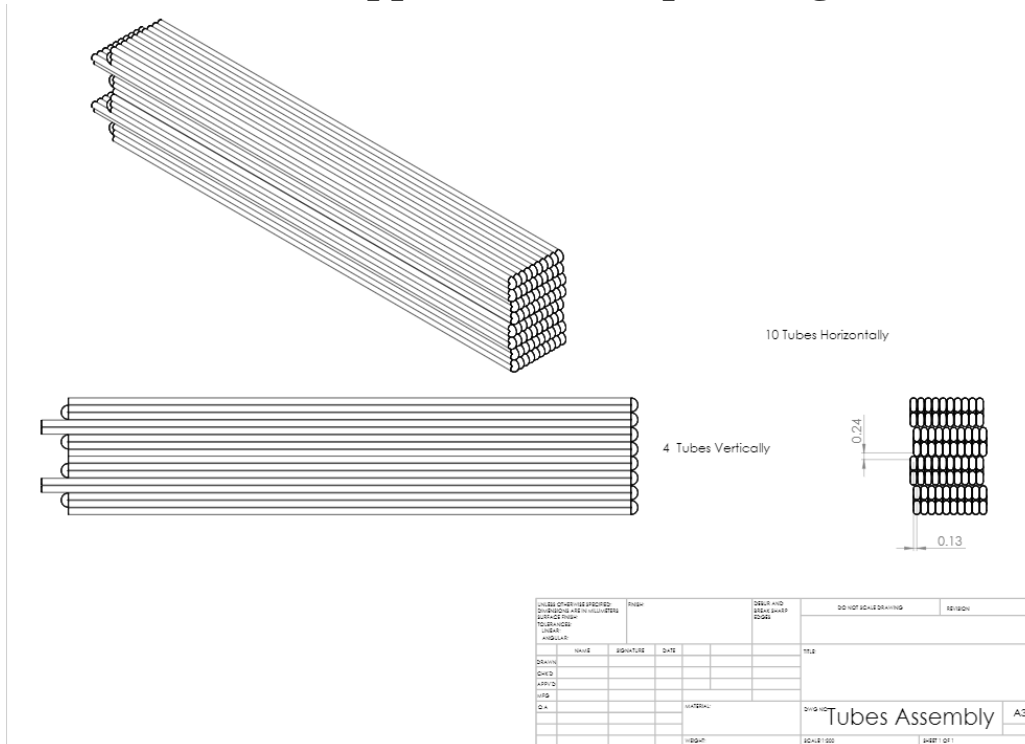
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## Appendix A – Helpful Diagrams



**Figure 33: Heat Exchanger Tubes**

### Chapter 3—Routine Maintenance Chart

Item to Check	Daily	Weekly	Monthly	Six Months	Yearly
Crankcase oil pressure	●				
Compressor discharge pressure	●				
Overall visual check	●				
Crankcase oil level			● <sup>2</sup>	● <sup>2</sup>	
Drain liquid from accumulation points		● <sup>3</sup>			
Drain adapters and distance pieces		●			
Clean cooling surfaces on compressor and intercooler (if any)		●			
Lubricator supply tank level (if any)		●			
Check belts for correct tension			●		
Inspect valve assemblies				●	
Lubricate motor bearings in accordance with manufacturers' recommendations				●	
Inspect motor starter contact points					●
Inspect piston rings <sup>1</sup>				● <sup>1</sup>	● <sup>1</sup>

<sup>1</sup>Piston ring life varies greatly, depending on application, gas and operating pressures. Consult factory for additional recommendations for your specific application.

<sup>2</sup>Change oil every 2,200 hours of operation or every 6 months, whichever occurs first. If the oil is unusually dirty, change it as often as needed to maintain a clean oil condition. Change replacement filter 4225 with every oil change.

<sup>3</sup>Liquid traps should be drained prior to startup.

**Figure 34: Compressor Maintenance Chart [26]**

## Appendix B – MATLAB CODE

```

% Cost Analysis Heat Exchanger
% Senior Design
% Last Updated: 3/22/15

D = 4; % Units [m]
D_old = 5.5;
L = 10; % Units [m]
L_old = 21;

Cs = 1800*(D^2.5)*L; % Cost of the shell % Units [$]
Cf = 350*D*L; % Initial cleaning cost % Units [$]
Ct = 1950; % First cost of the tubing % Units [$]
C = Cs + Cf + Ct

Cs = 1800*(D_old^2.5)*L_old; % Cost of the shell % Units [$]
Cf = 350*D_old*L_old; % Initial cleaning cost % Units [$]
Ct = 1950; % First cost of the tubing % Units [$]
C = Cs + Cf + Ct
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Senior Design
%PipeLosses/Heat losses

clc
clear all

numberofpoints = 20;
e = 0.0046;%Commerical Steel Roughness
PD = 0.5; % [m]
thickness = 0.0307; % [m]
g = 9.81;
nB_density = 2.48; % [kg/m^3] Vapor stage
Leq = 25; % [m]
T_ambient = 20; %Degrees C

%%%Piping Material Properties
k_bearsteel = 43; %W/m*K
cp_nbutane = 2282; %J/kg*K
k_nbutane = 0.0342;

%%%Butane_mass_flow = 40; % [kg/s]
AreaPipe = (PD^2)*(pi/4);
dynamic_viscosity = (0.0000124);%Pa*s
kinematic_viscosity = dynamic_viscosity / nB_density;

%%%%%%%% Heat Exchanger to Turbine %%%%%%%%%
F = zeros(1,numberofpoints); %rows x columns
P_turbinein = zeros(1,numberofpoints);
%%%VBT = Butane_mass_flow / (AreaPipe * nB_density)
P_boilerout(1:numberofpoints) = 1000000; % PressureofVaporout Pa
Vbt = linspace(1,25,numberofpoints); %[m/s] Velocity From boiler to inlet of
Turbine %%%%%%%%%
Re = (Vbt.*PD ./ kinematic_viscosity);

```

```

i = 1;
j = 1;
n = 1;

%Finding the Frictional Factor for different Re #
while i <= numberofpoints

if Re(j) > 2100
    F(n) = (-2.*log((e./3.7065.*PD) - (5.0452./Re(j)).* ...
        log(((1/2.8257).*(e./PD)^1.1098))+ ...
        (5.8506./(Re(j).^(.8981))))).^(-2);
else
    F(n) = 64./Re(j);
end

P_turbinein(n) = -
(nB_density*g).*((F(n).*(Leq/PD)).*((Vbt(n).^2)./(2*g)))...
- (P_boilerout(n)./(nB_density*g));

i = i+1;
j = j+1;
n = n+1;
end

Reynolds_Number1 = transpose(Re);
Friction_Factor1 = transpose(F);
Pressure_Turbine_in = transpose(P_turbinein);

Pr = kinematic_viscosity / k_nbutane;
Nu = 0.023.*(Reynolds_Number1.^0.8).*(Pr^0.3);
h1a = (Nu.*k_nbutane)./PD;
critical = k_nbutane./h1a
R_conv1 = 1./(pi.*PD.*Leq.*h1a);
R_conduction1 = log((PD+(2*thickness))/PD) ./ (2*pi*Leq*k_bearsteel);

R_conv2 = 1 ./ (pi*PD*Leq)

%%%%%% Turbine To Compressor %%%%%%
P_lossturbine = 500000;
P_turbineout = P_turbinein - P_lossturbine;
transpose(P_turbineout);
F2 = zeros(1,numberofpoints); %rows x columns
P_compressin = zeros(1,numberofpoints);
Leq2 = 25; % [m]
Vtc = linspace(1,6,numberofpoints) ; %[m/s] %Velocity From boiler to inlet
of Turbine
Re2 = (Vtc.*PD ./ kinematic_viscosity);

i = 1;
j = 1;
n = 1;

%Finding the Frictional Factor for different Re #
while i <= numberofpoints

if Re2(j) > 2100
    F2(n) = (-2.*log((e./3.7065.*PD) - (5.0452./Re2(j)).* ...

```



```

        log(((1/2.8257).*(e./PD)^1.1098))+ ...
        (5.8506./(Re2(j).^(.8981))))).^(-2);
else
    F2(n) = 64./Re2(j);
end

P_compressin(n) = -
(nB_density*g).*(F2(n).*(Leq2/PD)).*(Vtc(n).^2)./(2*g))...
- (P_turbineout(n)./(nB_density*g));

i = i+1;
j = j+1;
n = n+1;
end
Pressure_Compressor_in = transpose(P_compressin)

%%%%%%%% Compressor To Boiler %%%%%%%%%

P_compressout(1:numberofpoints) = 1100000; % this would be a set value
F3 = zeros(1,numberofpoints); %rows x columns
P_boilerin = zeros(1,numberofpoints);
Leq3 = 25; % [m]
Vcf = linspace(1,6,numberofpoints) ; %[m/s] %Velocity From boiler to inlet
of Turbine
Re3 = (Vcf.*PD ./ kinematic_viscosity);
i = 1;
j = 1;
n = 1;

while i <= numberofpoints

if Re3(j) > 2100
    F3(n) = (-2.*log((e./3.7065.*PD) - (5.0452./Re3(j)).* ...
        log(((1/2.8257).*(e./PD)^1.1098))+ ...
        (5.8506./(Re3(j).^(.8981))))).^(-2);
else
    F3(n) = 64./Re3(j);
end

P_boilerin(n) = -(nB_density*g).*(F3(n).*(Leq3/PD)).*(Vcf(n).^2)./(2*g))...
- (P_compressout(n)./(nB_density*g));

i = i+1;
j = j+1;
n = n+1;
end
Pressure_Boiler_in = transpose(P_boilerin);
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Team 2 Cummins Energy Savings
% Simulation Package

clear all,clc,close all

%% Exhaust Properties
rho_air = 0.7461; % kg/m^3 [1]
Cp_air = 1.0061; % kJ/kg*K

```

```

K_air = 1.4019; % Specific Heat Ratio cp/cv
Cv_air = Cp_air / K_air; % kJ/kg*K

prop_air = [ rho_air Cp_air K_air Cv_air];

%% ENGINE SERIES DETAILS

% Engine Power rating values. All values are in Horsepower.
ISB_HP = [ 185 190 205 210 225 240 245 260 275 ];
ISC_HP = [ 225 240 260 285 300 315 330 350 ];
ISL_HP = [ 310 330 ];
ISM_HP = [ 280 310 330 350 370 400 425 450 500 ];
ISX_HP = [ 400 450 475 500 600 ];

% Exhaust temperature values.
% All values are in Farenheit.
ISB_OUT_TEMP = [ 698 801 831 857 892 812 812 886 956];
ISC_OUT_TEMP = [ 706 746 765 833 860 919 927 966 ];
ISL_OUT_TEMP = [ 891 933 ];
ISM_OUT_TEMP = [ 670 721 742 720 737 737 969 789 965 ];
ISX_OUT_TEMP = [ 655 696 842 905 975 ];

% Mass flow rate values.
% All values are in CFM.
ISB_CFM_OUT = [ 1257 1250 1246 1313 1311 1456 1456 1592 1673 ];
ISC_CFM_OUT = [ 1417 1485 1578 1531 1578 1686 1758 1841 ];
ISL_CFM_OUT = [ 1681 1740 ];
ISM_CFM_OUT = [ 1523 1528 1610 1778 1853 1853 2171 2030 2341 ];
ISX_CFM_OUT = [ 2036 2218 2504 2633 3202 ];
% CONVERSION TO VALUES IN kg/s.

ISB_kgs_OUT = ISB_CFM_OUT*(.0283168)*(1/60) * rho_air;
ISC_kgs_OUT = ISC_CFM_OUT*(.0283168)*(1/60) * rho_air;
ISL_kgs_OUT = ISL_CFM_OUT*(.0283168)*(1/60) * rho_air;
ISM_kgs_OUT = ISM_CFM_OUT*(.0283168)*(1/60) * rho_air;
ISX_kgs_OUT = ISX_CFM_OUT*(.0283168)*(1/60) * rho_air;

ISB_avg_mdot = mean(ISB_kgs_OUT);
ISC_avg_mdot = mean(ISC_kgs_OUT);
ISL_avg_mdot = mean(ISL_kgs_OUT);
ISM_avg_mdot = mean(ISM_kgs_OUT);
ISX_avg_mdot = mean(ISX_kgs_OUT);
EngineAverages = [ISB_avg_mdot, ISC_avg_mdot, ISL_avg_mdot,ISM_avg_mdot,
ISX_avg_mdot];
%% Types of engines and mass flow
massFlowOut = [ ];
i = 1;
j = 1;
k = 1;
NumOfEngines = 0;
EngineSeries = 0;
NumEng = zeros(1,5);

while EngineSeries ~= 6
    disp('Please choose 1 = ISB,2 = ISC,3 = ISL,4 = ISM,5 = ISX,6 = DONE')
    EngineSeries = input('Which engine series is this?');
    % Entries allowed: ISB,ISC,ISL,ISM,ISX.

```

```

    if EngineSeries == 1
        massflow = ISB_kgs_OUT;
        i = 1;
    elseif EngineSeries == 2
        massflow = ISC_kgs_OUT;
        i = 2;
    elseif EngineSeries == 3
        massflow = ISL_kgs_OUT;
        i = 3;
    elseif EngineSeries == 4
        massflow = ISM_kgs_OUT;
        i = 4;
    elseif EngineSeries == 5
        massflow = ISX_kgs_OUT;
        i = 5;
    elseif EngineSeries == 6
        break;
    else
        error('Please select a value between 1-6.')
    end

    NumOfEngines = input('How many of these engines are running currently?');
    NumEng(i) = NumOfEngines;

    CombinedMassFlow(i) = EngineAverages(i)*NumEng(i); % Store the number of
    engine series into combined mass flow array. The sum of this array is the
    total mass flow.
end % END WHILE

disp(' The mass flow rate [kg/s] for different types of engines in the series
are:')
TotalMassFlow = sum(CombinedMassFlow)

%% Temperature Exhaust

T_in = 400; % Degree (F) {typically 400F}
T_C = (T_in - 32)* (5/9); % Output Degree Celsius
T_K = T_C + 273.15; % Output convert to Kelvin.
Tout_exhaust = 212;% [Degree F] (corresponds to 100C, picked from as design
parameter that determines size of heat exchanger.
Tout_exhaust = ((Tout_exhaust-32)*(5/9)) + 273.15; % [K]

%% Working Fluid Properties - Butane
% DENSITY
rho_but = 2.48; % [kg/m^3]
% ABSOLUTE VISCOSITY
visc_but = 7*10^(-6); % [Pa*s]
% SPECIFIC HEAT
Cp_but=1.675; %[kJ/kg*K]
Cp_but_turb=2.2; %[kJ/kg*K]
gamma = 1.096; % Specific heat ratios
% THERMAL CONDUCTIVITY
k_but = 1.70*10^(-2); % [W/m*K]
% PRANDTL NUMBER
Pr_but = 0.69; % [UNITLESS]
% MASS FLOW RATE

```

```

mdot_but = 7.00; % [kg/s]
% RUN Solar_Thermal_Collectors function. (Daily)
Day = input('What day of the year is it? (1-365)');
Time = input('What time is it?');
T_Collector_out = Solar_Thermal_Collectors(Day,Time);

% TEMPERATURE INLET
Tin_but = T_Collector_out; % [K]
% Run code from solar collectors.
% TEMPERATURE OUTLET
Tout_but = 145 + 273; % [K]

%% Heat Generated From Exhaust Gases
Q_exhaust = TotalMassFlow*prop_air(2)*(T_K - Tout_exhaust) % Units [kW]
%% Heat Available After Exchange
Q_available = Q_exhaust*.19
T_out_heatexchange = (Q_available/(mdot_but*Cp_but))+Tin_but
T_out_C = T_out_heatexchange - 273.15
%% Turbine Code
Poweroutput = 0.5*Q_available;
T_turbine_in = T_out_heatexchange;
P_turbine_in = 1000; %call from pipe loss code kPa
P_turbine_out = 500; %kPa
T_turbine_out = T_turbine_in .* ((P_turbine_out./P_turbine_in) .^(1-
(1/gamma)));
dh = Cp_but_turb*(T_turbine_in-T_turbine_out) % Units: [kJ/kg*K]

True_powerout = mdot_but*dh
Poweraccuracy = True_powerout/Poweroutput *100
%% Compressor Code
T_compress_in = T_turbine_out;
P_compress_out = 1100; %kPa
P_compress_in = P_turbine_out; %%% WE can call from pipe loss code.
T_compress_out = T_compress_in.* ((P_compress_out ./ P_compress_in) .^(1-
(1/gamma)));

%% Annual Energy
SystemEfficiencyGuess = (Poweroutput/Q_exhaust)*100 % Percent
SystemEfficiencyTrue = (True_powerout/Q_exhaust)*100 % Percent
Annual_Energy = True_powerout*3600*runningtime*365; % Units [kJ/day]
Annual_Consumption = 7.6*10^(11); % Units [kJ/year]
Annual_Energy_Savings = (Annual_Energy/Annual_Consumption)*100; % Percent
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Solar Thermal Collectors
% Senior Design Project
function [T_Collector_out] = Solar_Thermal_Collectors(day,time)

numberofpoints = 24;
SRT = xlsread('Solar_Power','Sheet1','G11:G375'); % Sunrise Time
SST = xlsread('Solar_Power','Sheet1','H11:H375'); % Sunset Time

%day = input('What day of the year is it = ');
time_rise_solar = SRT(day);
time_set_solar = SST(day);

```

```

solar_time = linspace(time_rise_solar,time_set_solar,numberofpoints);
solar_angle = zeros(1,numberofpoints);
zenith = zeros(1,numberofpoints);
alpha = zeros(1,numberofpoints);
as = zeros(1,numberofpoints);
cosi = zeros(1,numberofpoints);

%%% Coordinates for Indiana %%%
I_o = 1367; %W/m^2
a = 0.14;
h = .192; % km above sea level
AM = 1.5; % Air Mass Index [100 units]
aw = 0;
rho = 0.4;

lamda = Coordinate_Time(39,12,4.49298); % lattitude
W = Coordinate_Time(85,54,13.0428); % Lonigtude
beta = 0;

%%% Average Insolation Equation %%%
I_local = I_o.*((1-a.*h).*(0.7.^((AM).^0.678))) + (a.*h);
declination = 23.44.*sind(360*((day-80)/365.25));

i = 1;
j = 1;

while i <= numberofpoints
    solar_angle(i) = (360/24).*(solar_time(i) - 12);
    zenith(i) = acosd(sind(declination).*sind(lamda) + ...
        cosd(declination).*cosd(lamda).*cosd(solar_angle(i)));

    alpha(i) = (90 - zenith(i));
    as(i) = sind(cosd(declination).*sind(solar_angle(i))./cosd(alpha(i)));

    if lamda > declination
        as(j) = 180 - abs(as(i));
    else
        as(j) = as(i);
    end

    cosi(i) = cosd(alpha(i)).*cosd(as(i)).*sind(beta) +
sind(alpha(i)).*cosd(beta);

    j = j+1;
    i = i+1;
end

Direct_Beam = I_local.*cosi; % Regardless of what we use this is correct
(Horizontal equations and elevated equations)
z = 1;
%{
while z <= numberofpoints
    if Direct_Beam(z) < 0

```

```

        Direct_Beam(z) = 0;
    else
        Direct_Beam(z) = Direct_Beam(z);
    end
    z = z+1;
end
%}
if (day < 32)
    C = 0.048;
    Temperature_ambient = 2 + 273;
elseif (day < 59) && (day >= 32)
    C = 0.050;
    Temperature_ambient(1:numberofpoints) = 4.556 + 273;
elseif (day < 90) && (day >=59)
    C = 0.061;
    Temperature_ambient(1:numberofpoints) = 10.944 + 273;
elseif (day < 120) && (day >=90)
    C = 0.087;
    Temperature_ambient(1:numberofpoints) = 17.444 + 273;
elseif (day < 151) && ( day >= 120)
    C = .111;
    Temperature_ambient(1:numberofpoints) = 22.667 + 273;
elseif (day < 181) && (day >= 151)
    C = .124;
    Temperature_ambient(1:numberofpoints) = 27.722 + 273;
elseif (day < 212) && (day >=181)
    C = .126;
    Temperature_ambient(1:numberofpoints) = 29.444 + 273;
elseif (day < 243) && (day >= 212)
    C = .112;
    Temperature_ambient(1:numberofpoints) = 28.889 + 273;
elseif (day < 273) && (day >= 243)
    C = .082;
    Temperature_ambient(1:numberofpoints) = 25.333 + 273;
elseif (day < 304) && (day >= 273)
    C = 0.063;
    Temperature_ambient(1:numberofpoints) = 18.333 + 273;
elseif (day < 334) && (day >= 304)
    C = 0.053;
    Temperature_ambient(1:numberofpoints) = 11.111 + 273;
else
    C = 0.057;
    Temperature_ambient(1:numberofpoints) = 3.333 + 273;
end

Diffuse_Beam = C.*I_local;
Total_Beam = Direct_Beam + Diffuse_Beam;

%% Collector Energy Balance

massflownbutane(1:numberofpoints) = 7; %kg/s
cp_nbutane = 1900; %J/kg*K
Collector_area = 2100; %m^2 CTC Roof
Solar_transmittance = 0.92; %AE Collector
Solar_Absorptance = 0.96; %AE Collector
Collector_storage = 0; %J

```

```

Global_Conductance = 8; %W/m^2*K
Insolation = Total_Beam * Solar_transmittance;      %correct %W/m^2
G = massflownbutane / Collector_area;
T_f_in = 80 + 273.15;
T_f_out = T_f_in + (50 .* C ./ .126) .* (Insolation ./ max(Insolation));
F_r = (G * cp_nbutane .* (T_f_out - T_f_in)) ./ ((Solar_Absorptance .*
Insolation) - Global_Conductance * (T_f_in - Temperature_ambient));
heat_Useful = F_r .* Insolation .* Solar_Absorptance .* Collector_area -
(Global_Conductance .* (T_f_in - Temperature_ambient) .* F_r .*
Collector_area);
%Heat_Useful = abs(heat_Useful)
%Heat_lost = (Global_Conductance .* Collector_area .* (T_f_out + 273.15 -
Temperature_ambient))
Heat_lost = (Insolation .* Collector_area) - heat_Useful;
Temperatedifference = heat_Useful ./ (massflownbutane.*cp_nbutane);

T_Collector_out = Temperatedifference + T_f_in; %K
T_Collector_out_C = T_Collector_out - 273.15 %C

TotalTempchange = sum(Temperatedifference); % Not used.

Heat_total = heat_Useful + Heat_lost;

Collector_efficiency = heat_Useful ./ Heat_total; % Should be 30-40%
%{
while tcounter<=48
    if tcounter/2 < time_rise_solar || tcounter/2 > time_set_solar
        t_Collector_out(tcounter) = T_f_in;
    else
        t_Collector_out(tcounter) = t_Collector_out(tcounter);
    end
    tcounter = tcounter + 1;
end
%}
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Solar Thermal Collectors year
% Senior Design Project

clear all
clc

numberofpoints = 20;
SRT = xlsread('Solar_Power','Sheet1','G11:G375'); % Sunrise Time
SST = xlsread('Solar_Power','Sheet1','H11:H375'); % Sunset Time

solar_angle = zeros(365,numberofpoints);
zenith = zeros(365,numberofpoints);
alpha = zeros(365,numberofpoints);
as = zeros(365,numberofpoints);
cosi = zeros(365,numberofpoints);
solar_time_matrix = zeros(365,numberofpoints); % Size 365x20
Direct_Beam = zeros(365,numberofpoints);

lamda = Coordinate_Time(39,12,4.49298); % lattitude

```

```

W = Coordinate_Time(85,54,13.0428); % Longitude
beta = 0;
I_o = 1367; %W/m^2
a = 0.14;
h = .192; % km above sea level
AM = 1.5; % Air Mass Index [100 units]
aw = 0;
rho = 0.4;
I_local = I_o.*((1-a.*h).*(0.7.^(((AM).^0.678))) + (a.*h));
i = 1;
j = 1;
for day = 1:365

    time_rise_solar = SRT(day); % Reads the sunrise time, assigns here.
    time_set_solar = SST(day); % Reads the sunset time, assigns here.
    solar_time = linspace(time_rise_solar,time_set_solar,numberofpoints)';
    declination = 23.44.*sind(360*((day-80)/365.25));

while i <= numberofpoints % While i runs through 20 evenly spaced columns
    solar_time_matrix(j,i) = solar_time(i); % This should assign the solar
times for each day.
    solar_angle(j,i) = (360/24).*(solar_time_matrix(j,i) - 12);

    zenith(j,i) = acosd(sind(declination).*sind(lamda) + ...
        cosd(declination).*cosd(lamda).*cosd(solar_angle(j,i)));
    alpha(j,i) = (90 - zenith(j,i));
    as(j,i) =
sind(cosd(declination).*sind(solar_angle(j,i))./cosd(alpha(j,i)));

    if lamda > declination
        as(j,i) = 180 - abs(as(j,i));
    end % END IF STATEMENT

    cosi(j,i) = cosd(alpha(j,i)).*cosd(as(j,i)).*sind(beta) +
sind(alpha(j,i)).*cosd(beta);
    Direct_Beam(j,i) = I_local.*cosi(j,i);

%% CLOUDINESS FACTOR
if (day < 32)
    C = 0.048;
    Temperature_ambient = 2 + 273;
elseif (day < 59) && (day >= 32)
    C = 0.050;
    Temperature_ambient(1:numberofpoints) = 4.556 + 273;
elseif (day < 90) && (day >=59)
    C = 0.061;
    Temperature_ambient(1:numberofpoints) = 10.944 + 273;
elseif (day < 120) && (day >=90)
    C = 0.087;
    Temperature_ambient(1:numberofpoints) = 17.444 + 273;
elseif (day < 151) && (day >= 120)
    C = .111;
    Temperature_ambient(1:numberofpoints) = 22.667 + 273;
elseif (day < 181) && (day >= 151)
    C = .124;
    Temperature_ambient(1:numberofpoints) = 27.722 + 273;
elseif (day < 212) && (day >=181)

```



```

        C = .126;
        Temperature_ambient(1:numberofpoints) = 29.444 + 273;
    elseif (day < 243) && (day >= 212)
        C = .112;
        Temperature_ambient(1:numberofpoints) = 28.889 + 273;
    elseif (day < 273) && (day >= 243)
        C = .082;
        Temperature_ambient(1:numberofpoints) = 25.333 + 273;
    elseif (day < 304) && (day >= 273)
        C = 0.063;
        Temperature_ambient(1:numberofpoints) = 18.333 + 273;
    elseif (day < 334) && (day >= 304)
        C = 0.053;
        Temperature_ambient(1:numberofpoints) = 11.111 + 273;
    else
        C = 0.057;
        Temperature_ambient(1:numberofpoints) = 3.333 + 273;
    end

    Diffuse_Beam(j,i) = C.*I_local.*(1 + cosd(beta)) ./ 2;
    Reflected_Beam(j,i) = rho.*I_local.*sind(alpha(j,i) + C).*(sind(beta/2)^2);
    Total_Beam(j,i) = Direct_Beam(j,i) + Diffuse_Beam(j,i) + Reflected_Beam(j,i);

    i = i+1;

end % END WHILE LOOP
    i = 1;
    j = j+1;

end
    solar_time_matrix;
    solar_angle;
    zenith;
    alpha;
    as;
    cosi;
    Direct_Beam;
    Diffuse_Beam
    Total_Beam

```

## Appendix C - Solar Collector Code further Explanation

Solar declination

$$\delta = 23.44 \sin \left[ 360 \left( \frac{d - 80}{365.25} \right) \right] \text{ degrees,}$$

**Figure 35:** Solar declination Equation

```
%%% Average Insolation Equation %%%
I_local = I_o.*((1-a.*h).*(0.7.^(((AM).^0.678))) + (a.*h));
```

**Figure 36:** Average Insolation Equation

$$I_{b,c} = I_{b,N} \cos i,$$

$$\cos i = \cos \alpha \cos (a_s - a_w) \sin \beta + \sin \alpha \cos \beta$$

**Figure 37:** Direct Beam Radiation Equation

```
Diffuse_Beam = C.*I_local;
```

**Figure 38:** Diffuse Beam Radiation Equation

```
Total_Beam = Direct_Beam + Diffuse_Beam;
```

**Figure 39:** Total Beam Equation

```
Insolation = Total_Beam * Solar_transmittance;
```

```
G = massflownbutane / Collector_area;
```

**Figure 40:** Insolation and Mass Flow Ratio

$$F_R = \frac{Gc_p(T_{f,out} - T_{f,in})}{\alpha_s I_s - U_c(T_{f,in} - T_a)}$$

**Figure 41:** Heat Removal Factor

$$q_u = A_c F_R [\alpha_s I_s - U_c (T_{f,in} - T_a)]$$

**Figure 42:** Heat Transferred with absorptivity

or

$$q_u = A_c F_R [\tau_s \alpha_s I_c - U_c (T_{f,in} - T_a)]$$

**Figure 43:** Heat Transferred with Transmittance

$$\eta_c = \frac{q_u}{I_c A_c}$$

**Figure 44:** Solar Thermal Collector Efficiency

## Biography



**Danny Baker:** Dan Baker is a senior in Mechanical Engineering with an aerodynamics focus. Dan worked as a laboratory assistant in the cryogenics laboratory at the National High Magnetic Field Laboratory during his sophomore year. After graduation, he will be moving to Ann Arbor, Michigan to start his career while obtaining an MBA.



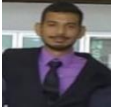
**Warren Bell:** Warren Bell is a third year intern at Myco Trailers LLC. He is proficient in Excel, MATLAB, MathCAD, Kubotek Keycreator, and Creo 2.0 (Pro-e) and will be graduating from the Florida State University with a BS in Mechanical Engineering in May 2015.



**Daniel Carnrike:** Daniel Carnrike will graduate from Florida State University on May 2<sup>nd</sup>, 2015. He has developed a passion for thermal fluid science and will be specializing in Fluid Mechanics and Heat Transfer upon graduation. During his educational career, Daniel has worked as a Mechanics of Materials and Dynamics teaching assistant. Daniel will pursue his masters in Mechanical Engineering at the Florida State University graduate program.



**Kyle Fields:** Kyle Fields is a senior Mechanical Engineering student at the Florida State University with a focus in Fluid Mechanics and Heat Transfer. Kyle has previously interned for Cummins as an Application Engineering Intern in the Filtration division located in Cookeville, TN. He is the team leader for the project.



**Marvin Fonseca:** Marvin Fonseca is a senior Mechanical Engineering student at Florida State University with experience in manufacturing. With this knowledge, he will be able to come up with different ideas to reduce energy consumption. He is in charge of the web design and development and any CAD design.