Union College
Union | Digital Works

Honors Theses

Student Work

3-2018

# Multicomponent Working Fluids in Organic Rankine Cycle Evaporators

Jennifer Fromm

# Multicomponent Working Fluids in

# Organic Rankine Cycle Evaporators

By

Jennifer Fromm

*************

Submitted in partial fulfillment

of the requirements for

Honors in the Department of Mechanical Engineering

and for MER 498, Mechanical Engineering Senior Project

UNION COLLEGE

March, 2018

## Abstract

Organic Rankine cycles are a promising technology to convert waste heat energy into usable mechanical or electric power, giving them the potential to reduce fossil fuel emissions generated by traditional energy generation. The heat exchangers of these devices are of particular interest, as maximizing energy extraction from these free heat sources will increase net electrical power output. For this project I created a model to predict the effects of mixture working fluids on the evaporator performance of an organic Rankine cycle generator for a wide range of waste heat source temperatures. This model combines empirically derived heat exchanger performance parameters with the Lemmon and Jacobsen equations of state for mixtures of refrigerants to calculate the overall heat transfer coefficient (the UA value) for the specified entry conditions, allowing for outlet temperatures and net heat transfer to be predicted. Data was collected on a 10”x20” x 40 plate flat plate heat exchanger using cool and warm water at various flow rates. Additional data was provided by Ener-G-Rotors from their refrigerant test bed. Parameters that can be varied within the model are the mass flow rates and inlet temperatures of the heat source and refrigerant, as well as the composition of the refrigerant working fluid. This variability will assist in future system adaptations to new waste heat conditions that could be utilized by organic Rankine cycle technology.

## Acknowledgements

I would first like to acknowledge the help and support I have received from my project advisor Prof. Richard Wilk. Additionally, I would like to thank Mr. Scott Gold and Mr. Mike Newell of Ener-G-Rotors for their support. I would also like to thank Stan Gorski, Paul Tompkins and Robert Harlan for providing invaluable technical support to this project, and Rhonda Becker for her administrative assistance.

Financially this project was made possible by component donations from Ener-G-Rotor, the Union College Department of Undergraduate Research through a Student Research Grant (SRG), and the Union College Presidential Green Grant. Thank you to these supporters.

# Contents

## List of Figures



## List of Tables

# Introduction

Sustainability and the push for green energy are two of the largest driving forces in engineering today, with governmental agencies and corporations being driven to operate with the health of our earth in mind. Environmental engineering comes in a variety of shapes and sizes, as everything from the design of our road way infrastructure to the materials used in food packaging contribute to humankind's impact on our world, but one of the largest contributors to our global footprint is our electrical energy production.

Globally, the majority of our energy is produced by coal and natural gas. The use of these non renewable energy sources is not only concerning economically and politically, as it leads to uncertainty regarding the future supply of these sources, but is also environmentally devastating. The burning of these fossil fuels releases millions of tons of carbon and other elements into the atmosphere, contributing to both global climate change and the pollution of the air and water in the areas where this electricity is produced. Additionally, the extraction of these fuels can be extremely harmful to local human and natural ecosystems. One only needs to watch the daily news see the harm done to people in coal mines, the costs to areas where fracking is taking place, and the animals harmed during oil spills. Clearly, this global trend needs to be reversed, and research into potential sources for alternative energy production and ways to increase the efficiency of our existing energy infrastructure are vital steps in the right direction.

The attractiveness of fossil fuels is clear: oil and coal burn very well and very hot, and have an have an incredible mass to energy storage ratio. The most widely used methods of electrical generation use these high temperate heat sources to vaporize a working fluid that can then generate mechanical work as it is expanded, which can then be converted into electricity, a process known as the Rankine cycle. Working fluids are most commonly water, another technical detail that is easily explained: water is the most abundant fluid in the world, it is non-toxic, non-flammable and relatively non-corrosive, it's thermodynamic properties have been extensively researched and documented, and billions of dollars have already been spent to develop the mechanical equipment to hand it vaporization, expansion, condensation, and compression. And while vaporizes at relatively high temperatures at achievable pressures, the most commonly used heat sources burn at temperatures well above them.

However, low temperature vaporizing fluids, typically organic in nature, are already being used to generate electricity from alternative heat sources in parts of the world where fossil fuels are more costly or green energy is more heavily subsidized. These organic Rankine cycles (ORC) largely expand the list of potential energy sources beyond the traditional fossil fuels, and can also increase the efficiency and net energy production of traditional power plants that are already equipped to burn these fuels. ORCs are currently utilized in geo-thermal plants, solar thermal plants, compost heat recovery, and as bottoming cycles in traditional power plants, where they make use of waste heat that would otherwise be discarded.

However, these cycles are not without their challenges. As stated earlier, the research and development into the mechanical devices that are required to facilitate the processes in a Rankine cycle has already been done for water, but are ongoing for organic working fluids. Additionally, mixtures of working fluids are being investigated because of their potential for customizability to specific heat source temperature, but the behaviors of these mixtures in prefabricated thermodynamic components remains to be studied.

For my senior design project, I partnered with Ener-G-Rotors, a startup company located in Schenectady, NY, that designs and manufactures module ORC devices tailored to their client's excess thermal production, dealing with heat source temperatures in the 65 to 150°C range. The core of their business is their innovative expansion design, a pressure driven positive displacement expander, [1] and they are currently using flat plate heat exchangers as their condensers and evaporators in all of their field implemented modules. Their modules are designed to reduce a building or complex's reliance on externally generated electricity, using waste heat sources to generate electricity that is fed directly to the client.

As Ener-G-Rotors and other field leaders of ORC technology, focus on the utilization of waste heat that would otherwise be discarded, the first law thermodynamic efficiency is not an accurate representation of the cycle's effectiveness. When "fuel" is free, the optimal cycle will extract the maximum energy from the waste heat available, thus the effectiveness of the heat transfer between the heat source and the working fluid is of paramount importance. While extracting more heat may lead to a worsened first law efficiency for the overall cycle, if it increases the overall power generation this is to the benefit of the cycle. My project focused on increasing the effectiveness of the heat transfer that occurs during the vaporization stage of Ener-G-Rotors cycles by utilizing custom made mixtures of working fluids.

The major objective of my project is to create a modeling that will evaluate the effectiveness of mixture working fluids in this stage of an ORC for different heat source temperatures and mass flow rates. This is to be achieved by combining empirical data, and theoretical equations of state to predict the behavior of these fluid mixtures in flat plate heat exchangers.

The two main research areas for this project are the thermal performance of the flat plate heat exchangers and the behaviors of mixtures organic working fluids. I collected empirical data first on a small flat plate heat exchanger previously owned by Union College, and then on a larger heat exchanger provide by Ener-G-Rotors, operating with low pressure water as both the heat sourcse and the heat sinks.  I analyzed this data to calculate the heat transfer coefficients for a variety of inlet conditions. This report will detail the methodology and results the data collection and analysis.

The predictions of working fluid properties are to be made with RefProp, a software developed by NIST specifically to give properties of commonly used thermal fluids and their mixtures. For the refrigerants used this model, which give only a sampling of the potential working fluids that could be used by Ener-G-Rotors and is not a representation of what they use currently or will use in future module implementations, RefProp uses the Lemmon Jacobsen equations of state for refrigerant mixtures.

These findings informed the construction of the computational tool that is the final goal of this project. A description of the methodology and framework of the model is given in the Predictive Model section, as well as a summary of the model's underlying assumptions and limitations.

# Background

## Organic Rankine Cycles

Rankine cycle generators utilize four thermodynamic processes, expansion, condensation, compression, and vaporization to convert thermal energy into mechanical energy.  Organic Rankine cycles are specific Rankine cycles that operate with organic fluids, typically refrigerants or hydrocarbons, and are tend to be smaller in scale and utilize lower temperature heat sources. The ideal working fluid is specific to each heat source and available coolant temperature [2], meaning that while generalizations on the merit of specific hydrocarbons or refrigerants over others exist, the evaluation of the proper fluid must be done on a case by case basis.

Work done on this topic is extensive, and typically involves model predictions of ORC work output for different fluids and temperature ranges using known thermodynamic properties. Jamal Nouman of the Technical University of Stockholm [3] performed a comprehensive study of 105 organically composed working fluid candidatesd for a hypothetical Rankine Cycle, analyzing the effects that additional cycle componentry, such as preheaters or super-heaters, would have upon the output of the system, thus determining not only the optimal working fluid but also the optimal cycle configuration which would optimize the thermal efficiency.

However, optimizing an ORC for maximum thermal efficiency does not always create the contextually optimal ORC configuration.  Numerous engineers have claimed that as the fuel source for typical ORC devices is otherwise wasted or lost, the thermal efficiency, as given in Equation 1, misrepresents the effectiveness of ORC generators because additional heat input does not add to fuel costs.

$$\eta_{TH} = \frac{Work\ Output}{Heat\ Input} \tag{1}$$

Several papers written on the subject detail alternative methods for determining the effectiveness of the ORC devices, including the second law thermal efficiency, given in Equation 2, which gives the ratio of the first law thermal efficiency and the ideal Carnot efficiency [4].

$$\eta_{II} = \frac{W_{Actual}}{W_{Ideal}} = \frac{\eta_{TH}}{1-(T_{max}-T_{min})} = \frac{\eta_{TH}}{\eta_{Ideal}} \tag{2}$$

Other methods for determining the effectiveness of an ORC device include the total heat recovery efficiency [5], which is given by Equation 3.

$$\eta_{T} = \frac{net\ Work}{Potential\ Energy\ of\ Waste\ Heat} = \eta_{TH} * \left[\frac{T_{h\ inlet}-T_{h\ outlet}}{T_{h\ inlet}-T_{WF\ Pre\ Evaporator}}\right] \tag{3}$$

These three methods of determining efficiency offer individual advantages and disadvantages, but sources strongly indicate that the relationship between the heat made available for extraction and the work output is more valuable than the simple value of extraction heat to work output. Thus research into maximizing heat extraction has the potential to increase ORC effectiveness and profitability.

## Multicomponent Fluids

Mixing working fluids has widespread promise in the field of ORC as they allow for customization of the cycles vaporizing and condensing stages to optimize heat transfer. Extensive research has been done in the fields of chemistry and chemical engineering to derive equations of state for mixtures of fluids and to make them more robust through empirical research [6]

Very rough estimates of properties can be determined from weighted averages that utilize mole fraction [7]. For example, the critical temperature of a mixture can be predicted by Kay's rule:

$$T_{cm} = \sum_i y_i T_{ci} \tag{4}$$

Where y is the mole fraction of the given fluid, but the higher the accuracy of the prediction that is required, the more complex these equations become. The determination of the thermodynamic properties of mixture fluids is typically done by experimentation and the gathering of empirical evidence on several mixtures [8], which is then extrapolated for different mass concentrations and fluids of similar properties. These models are accurate for single phase fluids which have relatively simple thermodynamic properties, however the modeling of multiphase heat transfer in a multicomponent fluid adds additional complexity to an already complex scenario. [9].

## Flat Plate Heat Exchangers

Heat exchangers are devices that facilitate the transfer of thermal energy from one thermal medium to another and they are used in most facets of thermal engineering and energy generation. They come in multiple forms many dealing with fluid to fluid heat transfer while keeping the two streams separate.

Flat plate heat exchangers are popular heat transfer devices as they are some of the most compact, variable, and effective heat exchangers commercially available. Unlike shell and tube heat exchangers, their uniform flow profile prevents hot or cold spots from forming and ensures consistent heat transfer between the two fluids. They are also adaptable to many different fluids and temperature ranges, however they do have upper pressure limits that prevent their use in large scale designs, such as an industrial power plant producing on the order of thousands of GW a year [10]. Flat plate heat exchangers are very widely used in residential scale heating and cooling systems, making them very easily applicable to modular ORC devices.

Flat plate heat exchangers do present considerable challenges when they are modeled. While shell in tube and other larger scale heat exchangers have been successfully modeled using Computational Fluid Dynamics (CFD) software, flat plate devices have defied such quantification and rely on estimations and empirical data. This modeling is made more difficult by the fact that, as they do in ORC devices, flat plate heat exchangers typically handle multiphase fluids in one of their flow path ways as they act as either a vaporizer or condenser. These multiphase flows are impossible to theoretically predict and computational models rely heavily on empirically derived data that makes estimations realistic. Thus, flat plate heat exchangers are typically modeled on a macro scale with averaged heat transfer coefficients, which prove effective for most uses.

# Experimental Work

As previously stated, the product of this project is a computational tool capable of predicting the effective heat transfer in the vaporizer for a variety of working fluid concentrations and a range of heat source temperatures and mass flow rates. This required physical testing to gather empirical data used to derive constant parameters for the equations of heat transfer. The following section details the set ups used to test the small (3”x8”, 20 plate) heat exchanger provided by Union College and the larger (10”x20”, 40 plates) heat exchanger donated by Ener-G-Rotors, the data resulting from these tests, and the calculations done to determine the heat transfer coefficients for each set of parameters.

## Small Heat Exchanger Set Up

The heat exchanger test set up is pictured in Figure 1.

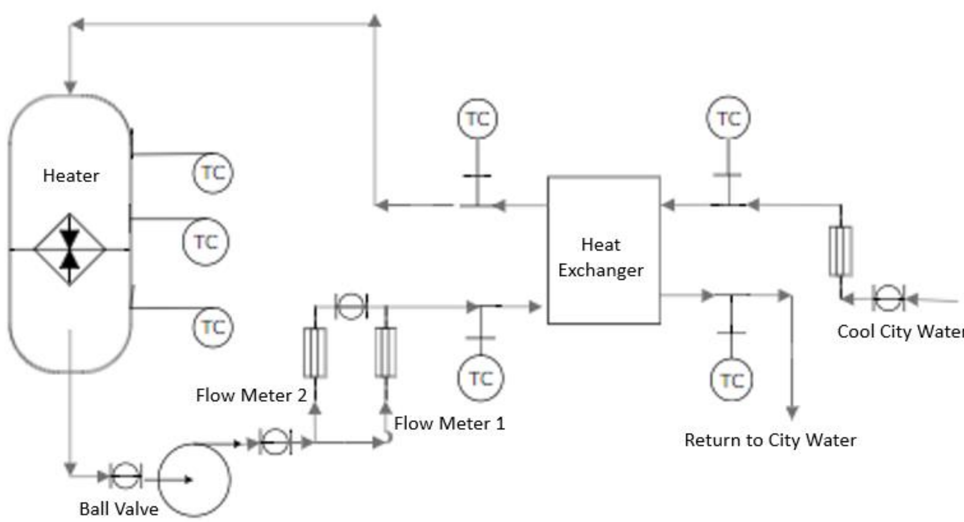



*Figure 1: Heat Exchanger Set Up*

A 15 gallon tank heated by a 1.5 kW immersion heater supplies the hot side while cool municipal water supplied the cool side. To prevent evaporation and heater exposure, hot side temperatures are closely monitored to remain below the boiling point of water, thus the temperature range of the heat source was 30°C to 80°C. The heater is located halfway from the top of the heat source tank, thus the pump must be on and water must circulate for the tank to reach a uniform temperature. The heat sink temperature is not controlled as it was provided directly by municipal water, at temperatures ranging from 18°C to 28°C. Type K $^{1}/_{8}$" diameter professional thermocouples are inserted into to the flow stream via Swagelok pipe fittings, which are inserted into pipe tees inline with the stream. Positive displacement flow meters are mounted vertically in each flow stream. Ball valves control the flow in each side of the heat exchanger and allow for the isolation of the hot side pump. Cool side flow is induced by municipal water pressure and no pump is required. Valves isolate the flow to go through one of two flow meters, which are accurate at two different ranges. All thermocouples are linked to a IOMEGA Data Acquisition Box to continuously monitor temperatures at their respective locations. Full details of the componentry used in the set up are given in Table 1.

*Table 1: Component Specifications for Small Heat Exchanger Set Up*

| Component | Specifications/Description | Supplier | Part Number |
|---|---|---|---|
| Heater | 1.5kW Screw-Plug Mount Immersion Heater, Adjustable, 304 Stainless Steel, 120V AC, Single Phase, 1500W, 9-1/4" Long | McMaster Carr | 3656K159 |
| Thermocouples | Type K 1/8" diameter Thermocouple probes | Omega | KQXL-18U-12 |
| Flow Meter 1 | Full View Flow Meter for water, 1 to 10 GPM, with female pipe fittings | McMaster Carr | 4197K51 |
| Flow Meter 2 | Easy to Install Dual-Scale Flowmeters for water, 0.1- 1 GPM, with female pipe fittings | McMaster Carr | 4400K49 |
| Swagelok Fittings | 1/8" tube to 1/4" pipe fittings, drilled out to accommodate the 1/8" thermocouple probe | McMaster Carr | 5272K291 |

The first heat exchanger tested is an GEA, 20 plate, ¾" threaded connection, 10gpm, 3"x 8" flat plate heat exchanger, with brazed copper plates and a depth of 2 1/4 ". The maximum predicted heat transfer of this model heat exchanger is 1200 BTU/ H / °F - $ft^2$. As previously seen in Figure 1, the heat exchanger was set up with the heat source and sink in counter flow.

## Large Heat Exchanger Set Up

The large heat exchanger was donated by Ener-G-Rotors and was used for preliminary testing of for their 1.5kW production generator module. The heat exchanger schematic is shown in Figure 2.

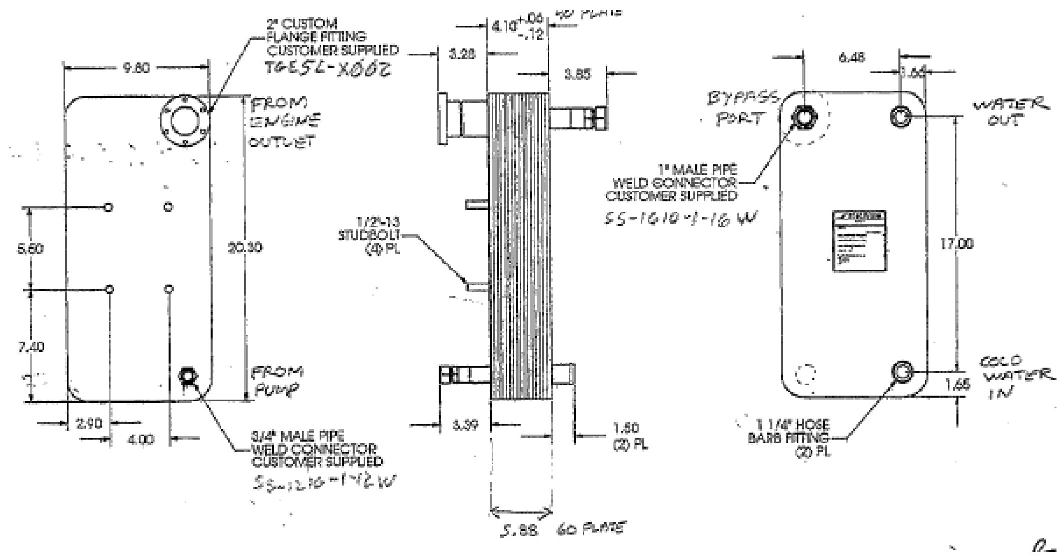


*Figure 2:Ener-G-Rotors test heat exchanger input and output diagram*

For the purposes of this project the heat exchanger was similarly configured with the same valve, flow meter and thermocouple configuration as the small heat exchanger, however as the inlet and outlet pipe diameters were different, there were alternate fittings attaching to connect it with the tap and the hot water tank. These fittings are detailed in Table 2.

This heat exchanger is configured as an evaporator, to take in cool working fluid on one side and hot water on the other to vaporize the working fluid for it to enter the Ener-G-Rotors rotary expander.  In our experimental set up, cool water will be run through the 1 ¼" hose barb inlet and outlet ports, and our hot side water will be run through the compression fitting inlet (3/4") and outlet (1"). A custom flange cap was constructed to seal off the flange fitting, as test operations are not anticipated to produce situations in which a bypass would be required. The schematic for this flange is seen in Appendix D: Flange Cap Schematic.

To integrate the thermocouples needed to measure flow temperatures at each inlet and outlet temperatures, inline tees with diameter reduction bushings and Swagelok tube fittings are installed at each connection. The cold side connection hose diameters are also sized down to ¾”. 1 ¼” hose ID and 3/4” pipe OD hose bards connect the outlet hose to the inline tee, which is then connected to a ¾” ID hose via a ¾” hose ID and 3/4” pipe OD hose bard. The compression fittings are likewise converted into ¾” ID pipe through the inline tees. The ¾” tube is connected to the tee via a ¾” tube OD to 3/4” pipe OD compression fitting, and then a ¾” hose ID and 3/4” pipe OD hose barb completes the transition. The 1” compression fitting outlet is connected to a 1” inline tee via 1” tube OD to 1” pipe OD compression fitting, which is then converted by a 1”OD to ¾” OD pipe bushing and the standard on the opposite side of the tee, to a ¾” hose ID and 3/4” pipe OD hose barb. These parts were purchased with funds provided by Union College through their Student Research Grant and are summarized in Table 2.

*Table 2: Component Specifications for Large HX*

| **Component** | **Specifications** | **Supplier** | **Part Number** | **Quantity** |
|---|---|---|---|---|
| Hose Barb | Brass Barbed Hose Fitting, ¾"ID, Swivels until Tightened, ¾" NPTF Male End | McMaster Carr | 5346K43 | 4 |
| Tee Connector | Low-Pressure Threaded Pipe Fitting, Tee Connector, 1 NPT female | McMaster Carr | 4429K255 | 1 |
| Tee Connector | Low-Pressure Threaded Pipe Fitting, Tee Connector, ¾ NPT female | McMaster Carr | 4429K254 | 3 |
| Bushing Adaptor | Low Pressure Brass Threaded Pipe Fitting, Bushing Adaptor with Hex Body, 1 Male x ¾ Female NPT | McMaster Carr | 4429K415 | 1 |
| Yor-Lok Fitting | Yor-Lok Fitting for Copper Tubing, Straight Adaptor for ¾" Tube OD x ¾ NPT Male | McMaster Carr | 5272K294 | 1 |
| Yor-Lok Fitting | Yor-Lok Fitting for Copper Tubing, Straight Adaptor for 1" Tube OD x 1 NPT Male | McMaster Carr | 5272K515 | 1 |
| Hose Barb | Brass Barbed Hose Fitting, 1-1/4"ID, ¾" NPTF Male End | McMaster Carr | 5346K94 | 2 |
| Bushing Adaptor | Low Pressure Brass Threaded Pipe Fitting, Bushing Adaptor with Hex Body, 1 Male x ¼ Female NPT | McMaster Carr | 4429K461 | 1 |
| Bushing Adaptor | Low Pressure Brass Threaded Pipe Fitting, Bushing Adaptor with Hex Body, ¾ Male x ¼ Female NPT | McMaster Carr | 4429K423 | 3 |

## Collected Data

Initially, data was collected to characterize the heat source tank and its heat loss rate when both insulated and uninsulated, to determine the UA values for the two set ups. These data and the accompanying analysis are given in Appendix E: UA Calculations.

Tests were run on the initial small heat exchanger to verify the functionality of the set up before the large heat exchanger could be installed. Data was collected on the inlet and outlet temperatures of the small heat exchanger on with three different combinations of mass flow rates for the inlets and outlets. The inlet temperature on the hot side decreased as the tank itself lost heat to the heat exchanger. Cold side inlet temperature fluctuated with the provided temperature of municipal water. Graphs detailing this data are shown in Appendix F: Small heat exchanger data.

After the installation of the large heat exchanger more data was collected on the inlet and outlet temperatures of the warm and cold water stream for a variety of mass flow rates. Similarly, the hot side temperature decreased slightly as the hot water tank was cooled via the heat exchanger and the cold side temperature fluctuated as the available cooling water changed temperature. Six data points were collected at 6 different mass flow rates and varying input temperatures. The data is summarized in

Table 3. The complete data set is available upon request.

*Table 3: Time averaged temperatures and mass flow rates for experiments on the large heat exchanger*

| *Date/ID* | *Thi (C)* | *Tho (C)* | *Tci (C)* | *Tco (C)* | *mdot h (kg/s)* | *mdot c (kg/s)* |
|---|---|---|---|---|---|---|
| *2/5/18 _1* | 69.83 | 61.00 | 22.85 | 57.21 | 0.225 | 0.058 |
| *2/5/18_2* | 67.00 | 55.42 | 14.48 | 32.19 | 0.067 | 0.067 |
| *2/11/18 _1* | 68.39 | 60.30 | 24.39 | 41.31 | 0.125 | 0.067 |
| *2/11/18 _2* | 67.65 | 58.16 | 17.14 | 41.47 | 0.167 | 0.067 |
| *2/11/18 _3* | 66.00 | 56.63 | 14.43 | 45.20 | 0.208 | 0.067 |
| *2/11/18 _4* | 64.89 | 57.84 | 13.81 | 25.05 | 0.042 | 0.067 |

## Data Analysis

The goal of this data analysis was to determine the overall heat transfer coefficient of the flat plate heat exchanger to provide a basis for the computer model that is the end result of this project. The U was found with the following energy balance equation:

$$\dot{m}Cp(T_{out} - T_{in}) = Q = UA\Delta T_{lm} \quad (5)$$

The change in energy of the cold and hot side would in theory be equivalent, however, as there is heat lost to the environment through the faces of the heat exchanger, the more accurate representation of the quantity of heat transferred is the heat gained by the cold side. Cp is the specific heat of water, 4.2 kJ/kg*K, and the log mean temperature difference is calculated with the following equation:

$$\Delta T_{lm} = \frac{(T_{h\,in} - T_{c\,out}) - (T_{h\,out} - T_{c\,in})}{ln\left(\frac{T_{h\,in} - T_{c\,out}}{T_{h\,out} - T_{c\,in}}\right)} \quad (6)$$

This equation for log mean temperature difference is specifically for separated streams in counter flow.

From these equations the empirical UA value was calculated for each set of parameters tested. These calculations are summarized below in Table 4.

*Table 4: Summary of Empirical UA values, with LMTG and Q values, for each set of parameters*

| *Date/ID* | *UA (kJ/K)* | *LMTD (K)* | *Qc (kJ)* | *Qh (kJ)* |
|---|---|---|---|---|
| *2/5/18 _1* | 0.363 | 23.07 | 8.39 | 8.32 |
| *2/5/18_2* | 0.131 | 37.79 | 4.96 | 3.21 |
| *2/11/18 _1* | 0.151 | 31.29 | 4.72 | 4.24 |

| | | | | |
|---|---|---|---|---|
| *2/11/18 _2* | 0.205 | 33.05 | 6.79 | 6.62 |
| *2/11/18 _3* | 0.284 | 30.26 | 8.59 | 8.17 |
| *2/11/18 _4* | 0.075 | 41.90 | 3.14 | 1.23 |

Note that there is a consistent variance in the heat transferred to the working fluid and from the heat source when a conservation of energy would mandate that they be equal, or at the very least that more heat would be lost from the heat source than is absorbed by the working fluid. This discrepancy can be attributed to the high level of uncertainty imposed upon the Q calculation from the measurement of the volumetric flow meter on the hot side in the experimental set up of plus or minus 1.25 GPM. This corresponded to the uncertainties tabulated in Table 5. By contrast, the cold side flow meter was smaller and contributed an uncertainty of only plus or minus ;0.05 GPM.

*Table 5: Uncertainty of Q calculations*

| *Date/ID* | *Qc (kJ)* | *Qh (kJ)* | *Uncertainty of Qc* | *Uncertainty of Qh* |
|---|---|---|---|---|
| *2/5/18 _1* | 8.39 | 8.32 | 0.264324123 | 0.654995 |
| *2/5/18_2* | 4.96 | 3.21 | 0.156184214 | 0.253179 |
| *2/11/18 _1* | 4.72 | 4.24 | 0.148794881 | 0.333548 |
| *2/11/18 _2* | 6.79 | 6.62 | 0.213862131 | 0.521692 |
| *2/11/18 _3* | 8.59 | 8.17 | 0.2705409 | 0.643155 |
| *2/11/18 _4* | 3.14 | 1.23 | 0.098836038 | 0.096929 |

## Predictive Model

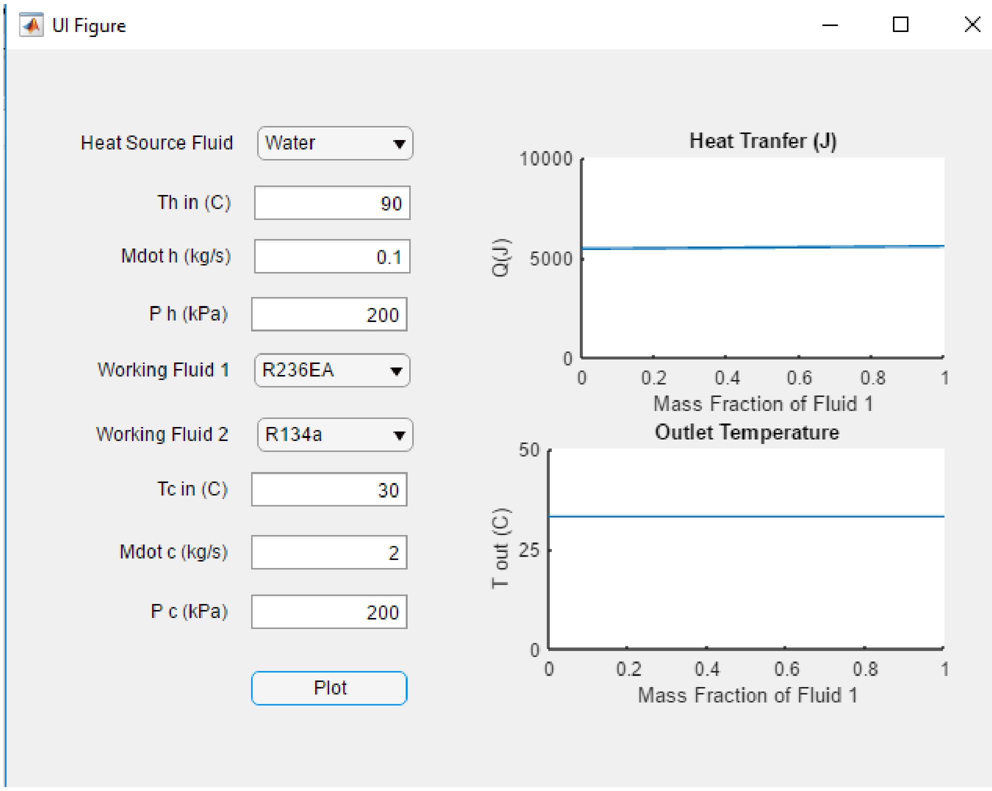



*Figure 3: Image of the Graphic User Interface that front ends the predictive model*

The goal of this project was to produce a working model of the flat plate heat exchanger provided by Ener-G-Rotors with multi-component working fluids. This section will describe the model's underlying structure and the methodology used to generate the predictions and the assumptions made by the modeling program. The MatLab scripts that comprise the code are provided in Appendix G: MatLab Scripts of the final model version.

### Description of the Model

The model is mapped out as shown in Figure 4, which illustrates the inputs and outputs of the model. The subscript 'c' is used to describe characteristics of the cold side or working fluid, while ‘h’ describes characteristics of the hot side or heat source. The subscript ‘I’ denotes properties at inlet conditions, while ‘o’ denotes outlet conditions Note, it is assumed that there is only one inlet fluid on the hot side. SI units are used, Celsius for temperature, Pascals for pressure, and kg/s for mass flow rate.

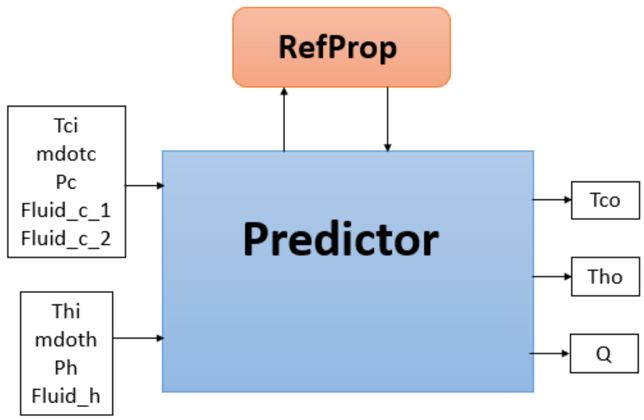



*Figure 4: Predictor model map of inputs and outputs*

Once the model has these inputs, it converts the temperatures to Kelvin and then estimates the average temperature of each fluid to be used for property looks ups. RefProp is then connected to determine the Prandlt number (Pr), the specific heat (Cp), the kinematic viscosity (v) and the thermal conductivity(k) for each fluid or mixture of fluids. Equations described in the following Theoretical Heat Transfer Coefficients sections are used to determine the U of the system, which is dependent on the calculated fluid properties.

Q, the net heat transfer, is then calculated with the following equation:

$$Q = \frac{(e^k - 1)\,(T_{hi} - T_{ci})}{\frac{e^k}{\dot{m}_h C p_h} - \frac{1}{\dot{m}_c C p_c}} \tag{7}$$

Where k is the constant defined in the following equation

$$k = UA\left(\frac{1}{\dot{m}_h Cp_h} - \frac{1}{\dot{m}_c Cp_c}\right) \tag{8}$$

This equation was derived by setting the three known equations for Q equal to one another, as follows:

$$Q = \dot{m}_c Cp_c(T_{co} - T_{ci}) = \dot{m}_h Cp_h(T_{hi} - T_{ho}) = UA\left(\frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{ln\left(\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right)}\right) \tag{9}$$

The value of Q was then used to find Tco and Tho.

It may be noted that the mass fraction for the two fluid mixtures is not a user input, as the output of the Predictor model plots the possible outcome heat flux and outlet temperatures for compositions in a range from 0% to 100% Fluid_2, allowing the user to select the appropriate composition for the specified inlet conditions.

## Theoretical Heat Transfer Coefficients

The calculation of theoretical heat transfer coefficients is integral to the model's functionality. U was calculated with the following equation:

$$\frac{1}{U} = \frac{1}{h_c} + \frac{x}{k} + \frac{1}{h_h} \tag{10}$$

Where $h_c$ is the heat transfer coefficient for the cool side in W/K, $h_h$ is the heat transfer coefficient for the hot side, x is the thickness of the plate between the two streams in m, in this case 0.0005m, and k is the thermal conductivity of the material of the plate in W/m*K, in this case copper, at 200 W/m*K.

According to Bergman's *Fundamentals of Heat and Mass Transfer* [11], for a laminar liquid phase flow through rectangular pipes, the Nusselt number is 8.23 for uniform heat flux and 7.54 for a uniform surface temperature. For this project the Nusselt number was estimated to be 8 as the heat transfer was neither uniform in heat flux or surface temperature. Thus, the heat transfer coefficient for water was found using the following equation:

$$Nu = \frac{hDv}{k} \tag{11}$$

Where D is the effective diameter of the rectangular tube, derived empirically to be 0.029m, v is the kinematic viscosity of the fluid or fluid mixture, and k is the thermal conductivity of the or fluid mixture , and Nu was set equal to 8. For the water to water model, this h was used for both the hot side and the cold side, and for the refrigerant model it was used for the hot side only, as the heat source is assumed to be single phase liquid water.

Two phase flows are intrinsically more complex than single phase flows, and with the nonstandard geometry of the heat exchanger, Nusselt number calculations become more difficult. According to the results of Subbiah [12], the following correlation can be used for fully developed two phase refrigerant flow in a flat plate heat exchanger:

$$Nu = 4.118Re^{0.4}Pr^{1/3} \tag{12}$$

Where Re is the Reynolds number calculated for the specific geometry of the heat exchanger and Pr is the Prandtl number of the fluid, or fluid mixture. This Nu number was then used in Equation 2 to calculate the cold side heat transfer coefficient for refrigerant working fluids.

All fluid specific properties, k, v, Pr, and Cp, were calculated using equations of state internally in RefProp [13], which for refrigerants of the nature handled in this project, utilized the Lemmon and Jacobsen Method for refrigerant mixtures [6].

## Assumptions and Limitations

A primary assumption made in structure of the model is that the temperature dependent fluid properties that changed as the fluids or fluid mixtures changed temperature are adequately approximated by values looked up for a mid-point temperature. This is a rational assumption given that the temperature changes expected within Ener-G-Rotors generators are typically within a range of 80C, limited from above by the boiling point of water at the low operating pressures, and limited from below by the ambient temperature used to vent heat. Thus the single phase fluid properties would not be expected to undergo drastic changes. This is a greater stretch for the two-phase fluids which would clearly change quite dramatically, and an aspect that could be improved upon for future changes made to the model.

A second assumption made is that the pressure drop over the heat exchanger is negligible and can be ignored. This has been verified by data provided by Ener-G-Rotors and should not have a significant effect on the model's accuracy.

The third major assumption is regarding the empirically derived effective diameter that was calculated to calibrate the model to mimic water to water heat transfer as observed in the experimental set up. This effective diameter is assumed to be an adequate value to use when the working fluid is a two-phase refrigerant as well. Given the limitations of Union College's facilities and their inability to handle vaporizing refrigerants, this assumption was unavoidable for the model generated by this project, and undoubtedly has a negative impact on the accuracy of predictions made by the model.

Additionally, the equations of state used to determine properties of mixtures by RefProp are also extrapolated from a limited set of data and are predictions in and off themselves. However, these predictions have been correlated with large quantities of experimental data by NIST and are the most accurate values available without doing independent research on fluid properties, which would be an additional project in and of itself.

## Conclusions

This project was a success in that it delivered a working model to aide Ener-G-Rotors if they choose to integrate mixture working fluids into future installations of 1.5kW generators. However, due to simplifying assumptions made for the framework of the model, the initial hypothesis that mixture working fluids would increase heat transfer with zeotropic expansion was neither confirmed nor denied. Recommendations for continuations of this project would be to obtain access to the facilities at Ener-G-Rotors and  collect additional data on the behaviors of mixture fluids in flat plate heat exchangers. This will allow for a more careful analysis of the potential for zeotropic expansion.

## References


[1] Ener-G-Rotors, "Ener G Rotors Technology," 2016. [Online]. Available: www.ener-g-rotors.com. [Accessed 29 Septermber 2017].

[2] Y. Haoshui, F. Xiao and W. Yufei, "Working Fluid Selection for Organic Rankine Cycle (ORC) Considering the Characteristics of Waste Heat Sources," *Industrial and Engineering Chemistry Research,* vol. 55, pp. 1309-1321, 2016.

[3] J. Nouman, "Comparative Studies and Analsyses of Working Fluids for Organic Rankine Cycles (ORC)," Technical University of Stockholm, Energy Department, Stockholm Sweden, 2012.

[4] G. Angelino and P. Colonna Di Paliano, "Multicomponent Working Fluids for Organic Rankine Cycles (ORC)," *Energy,* vol. 23, no. 6, pp. 449-463, 1998.

[5] B.-T. Liu, K.-H. Chien and C.-C. Wang, "Effect of working fluid on organic Rankine cycle for waste heat recovery," *Energy,* vol. 29, pp. 1207-1217, 2004.

[6] E. W. Lemmon and R. T. Jacobsen, "Equations of State for Mixtures R-32, R-125, R-134a, R143a, and R-152a," *Jounal of Physical and Chemical Reference Data,* vol. 33, pp. 593-620, 2004.

[7] R. C. Reid, J. M. Prausnitz and T. K. Sherwood, The Properties of Gases and Liquids, New York: McGraw-Hill Book Company, 1977.

[8] T. J. Bruno, L. S. Ott, T. M. Lovestead and M. L. Huber, "Relating Complex Fluid Composition and Thermophysical Properties with the Advanced Distillation Curve Approach," *Chemical Engineering Technology,* vol. 33, no. 2, pp. 363-=376, 2010.

[9] N. Padoin, A. T. Dal'Toe, L. P. Rangel, K. Ropelato and C. Soares, "Heat and mass transfer modeling for multicomponent multiphase flow with CFD".

[10] S. Kalac, A. E. Bergles and F. Mayinger, Heat Exchangers: Thermal-Hydraulic Fundamentals and Design, Washington: Hemisphere Publishing Corporation, 1981.

[11] T. L. Bergman, A. S. Lavine, F. P. Incropera and D. P. Dewitt, The Fundamentals of Heat and Mass Transfer, 7th, Ed., Jefferson City: John Wiley and Sons, 2011.

[12] M. Subbiah, "The Characteristics of Brazed Plate Heat Exchangers with Different Chevron Angles," in *Heat Exchangers- Basic Design Applications*, Rijeka, Croatia, Intech, 2012, pp. 397-424.

[13] E. Lemmon, M. Huber and M. McLinden, *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1,* Gaithersburg: National Institute of Standards and Technology, Standard Reference Data Program, 2013.

## Appendix A: Student Research Grant Proposal

Jennifer Fromm
Student Research Grant Proposal
Investigation of Mixture Working Fluids in Low Temperature Generators
Advisor: Prof. Richard Wilk
Fall 2017/Winter 2018

### *Statement of Problem*

Energy extraction from low temperature or waste heat is a promising process for energy efficient technology. Rankine cycles have long been used to convert thermal energy into work, and the performance of popular, high critical temperature working fluids (such as water) has been well documented and researched. However, the behaviors of low temperature vaporizing working fluids, for example refrigerants R-134a, R236fa, and 245fa, and particularly the behaviors of mixtures of these fluids[1], has yet to be thoroughly documented. Additionally, new international policies are continuously restricting the usage of CFCs and HCFCs, spurring further research into more environmentally friendly fluids. Thus, further research into these behaviors is valuable to the development of these energy extractors and has the potential to considerably increase the effectiveness of such technologies.

### *Project Description*

In my Senior Research project, I will focus on the performance of these 'cool', low temperature vaporizing, working fluids operating in a Rankine Cycle generator that utilizes a pressure driven expander created by Ener-G-Rotors. This expander is unique as it requires relatively low working fluid flow rates to operate, but a high-pressure gradient during the expansion process, which presents unique design challenges and solutions in the design of the accompanying cycle componentry, particularly the evaporator and condenser components.

I will gather data on the flat plate heat exchanger utilized in the Ener-G-Rotors 5kW generator (provided by Ener-G-Rotors) with working fluid and heat source flow rates at approximately 5gpm, with heat source temperatures ranging from 180°F to 220°F, and cooling water supplied at 50°F to 70°F. These flow rates and temperatures can be supplied by the heat exchanger test bed in Prof. Richard Wilk's Energy Lab, however connection componentry will need to be purchased.

This data will be used to characterize the heat exchanger performance and heat transfer rates, particularly the overall heat transfer coefficients and the effectiveness, thus allowing me to extrapolate and model the heat exchanger's performance with multiphase working fluids other than water as well as mixtures. Specifically, fluids including R-134a, R-236fa, R-245fa, butane isobutane, and ammonia will be theoretically investigated. Due to safety and budgetary restraints, these fluids will not be physically tested on Union Property, however it is possible that Ener-G-Rotors will verify my computational predictions in their onsite laboratory.

Using fundamental heat transfer equations, the fundamental of fluid mechanics, and the overall heat transfer coefficients calculated during physical testing, I will create a model for the 5kW generator heat exchanger. Additionally, data collected on the performance of a larger heat

[1] DiPippo, Ronald. "Second Law assessment of binary plants generating power from low-temperature geothermal fluids". *Geothermics* 33 (October 2004): 575.

exchanger used in the Ener-G-Rotors will be used to extrapolate the code to analyze larger heat exchangers, an important feature for the future applications of the model.

This code will also require the calculation of theoretical properties of binary mixtures of working fluids[2], an important part of the investigation which could increase efficiency due to the non-isothermal (zeotropic) vaporization phase during evaporation and condensing, and the potential for tailor-made fluids that could optimize performance across the entire range of acceptable heat source and cooling fluid temperatures. These predictions will be done using equations of, which are derived from various mixing principles.

### *Project Deliverables*

The results of this project will include a detailed analysis of the above proposed working fluids in the 5kW generator heat exchanger, as well as analysis of binary mixtures of the three refrigerants R-134a, R-236fa and R-245fa, a working coded model for the 5kW generator heat exchanger and potentially a generalized model adaptable for other heat exchanger configurations, particularly the 35kW generator heat exchanger. This will be presented to the Mechanical Engineering Faculty of Union College to be graded as my Senior Design Project, and to the Union College Community during the Steinmetz presentation day in May 2018. Additionally, all findings and the computerized model will be made available to Ener-G-Rotors for their future use.

---

[2] Christophe Coquelet, C`eline Houriez, Jamal El Abbadi. "Prediction of thermodynamic properties of refrigerant fluids with a new three-parameter cubic equation of state." *International Journal of Refrigeration* 69 (2016): 418-436.

*Budget Proposal*

Numerous materials are needed to establish the working test bench for the analysis of the heat exchanger used in the 5Kw Ener-G-Rotors generator. The heat exchanger (valued at ~$600) itself will be loaned by Ener-G-Rotors with the potential for long term extensions of component. Union college is already in possession of several thermal reservoirs, which can supply the high and low temperatures needed for the exchanger analysis. Data acquisition can be done by the data acquisition systems supplied by the Mechanical Engineering Department and will not be consumed. Remaining to be purchased are plumbing fixtures that can accommodate the large diameter heat exchanger input and output ports, which are 1" diameter, larger than the current ¾" set up currently in place in Prof. Wilk's Energy Lab. Additionally, thermocouples to supplement those already present in the lab will be needed to continuously monitor fluid temperatures. Shipping costs are rough estimations, but are included to prevent the Mechanical Engineering Department from needing to supplement the cost as is typical for Senior Research Project.

Table 1: Itemized Budgetary Summary

| Item | Description | Supplier | Quantity | Unit Price | Total | Grand Total |
|---|---|---|---|---|---|---|
| Copper Tubing | 3' of 1" diameter Copper Tubing for plumbing of the 5kW Plate Heat Exchanger | McMaster Carr | 1 | $57.83 | $57.83 | |
| Ball Valve | Bronze, 1" Diameter, Ball valve to seal off external plumbing during diagnostics and troubleshooting of the heat exchanger | McMaster Carr | 4 | $21.44 | $85.76 | |
| Elbow Connectors | 1"D 90degree brass threaded elbow connector | McMaster Carr | 4 | $33.14 | $132.56 | |
| Brass Inline Tee connectors | 1" to 1/4" Tee connectors to allow for thermocouple insertion into the fluid flow stream | McMaster Carr | 4 | $30.91 | $123.64 | |
| | | | | | | $399.79 |

## Appendix B: Presidential Green Grant Proposal

Jennifer Fromm
Green Grant Proposal
Investigation of Mixture Working Fluids in Low Temperature Generators
Advisor: Prof. Richard Wilk
Fall 2017/Winter 2018

### *Statement of Problem*

Energy extraction from low temperature or waste heat is a promising process for green, energy efficient technologies. There is a wealth of untapped energy potential that is discarded as low temperature fluid, either in the cooling steps of large industrial plants or in facilities that utilize warm or hot water. Rankine cycles have long been used to convert thermal energy into work, and in Europe and some parts of the USA, new Rankine cycle generators are being developed and utilized to convert these waste heat sources into useable electricity, decreasing fuel consumption and reducing emissions from traditional electricity sources.

Organic Rankine cycle (ORC) generators, named for their organic low temperature vaporizing working fluids, can be used in a variety of settings where traditional water Rankine cycles give insufficient temperature ranges. For example, due to high energy costs, ORC technology is widely used in Europe with biomass, geothermal, and solar energy converters[1]. In the US, where the eco-political climate is less friendly to green energy sources, ORC technology has large potential on the industrial level or local, to produce electricity from the waste heat of a laundry facility[2]. ORC have even been applied in compost heat recovery systems, a previously untapped field[3]

These generators operate using low temperature vaporizing working fluids to maximize the energy extraction from the low-quality heat sources that they use. The behaviors of such low temperature vaporizing working and particularly the behaviors of mixtures of these fluids[4], has yet to be thoroughly documented. Further research into these behaviors is valuable to the development of these energy extractors and has the potential to considerably increase the effectiveness of such technologies, and thus amplify the sustainability benefits they offer.

### *Project Description*

As my mechanical engineering senior research project, I am focusing on the performance of these 'cool', low temperature vaporizing, working fluids in a Rankine Cycle generator created by Ener-G-Rotors. This expander is unique as it requires relatively low working fluid flow rates to operate, but a high-pressure gradient during the expansion process, which presents new design challenges and solutions in the design of the accompanying cycle componentry, particularly the evaporator and condenser components.

I am gathering data on the flat plate heat exchanger utilized in the Ener-G-Rotors 5kW generator (provided by Ener-G-Rotors) with working fluid and heat source flow rates at approximately 5gpm, with

[1] R. Rowshanzadeh, "Performance and cost evaluations of Organic Rankine Cycle at different technologies," Stockholm, 2010.
[2] Ener-G-Rotors, "Ener-G-Rotors' Waste Heat to Power Solution Increases Efficiency of Combined Heat and Power System at Bates Troy, Inc.," 19 April 2016. [Online]. Available: http://www.ener-g-rotors.com/ener-g-rotors-waste-heat-to-power-solution-increases-efficiency-of-combined-heat-and-power-system-at-bates-troy-inc/. [Accessed 10 September 2017].
[3] M. M. Smith, J. D. Abner and R. Rynk, "Heat Recoveryt from Composting: A Comprehensive Review of System Design, Recovery Rate, and Utilization," *Compost Science and Utilization*, 2016.
[4] R. DiPippo. "Second Law assessment of binary plants generating power from low-temperature geothermal fluids". *Geothermics* 33 (October 2004): 575.

heat source temperatures ranging from 180°F to 200°F, and cooling water supplied at 60°F to 70°F. These flow rates and temperatures can be supplied by the heat exchanger test bed in Prof. Richard Wilk's Energy Lab. Some connection componentry to specific to this new heat exchanger is being purchased through a Student Research Grant, however additional funding will allow for more flexibility in the operating set up and a more comprehensive study mechanism.

This data will be used to characterize the heat exchanger performance and heat transfer rates, particularly the overall heat transfer coefficients and the effectiveness, thus allowing me to extrapolate and model the heat exchanger's performance with multiphase working fluids other than water as well as mixtures. Specifically, fluids including R-134a, R-236fa, R-245fa, butane isobutane, and ammonia will be theoretically investigated. Due to safety and budgetary restraints, these fluids will not be physically tested on Union Property. While the Union College Mechanical Engineering Department has access to limited thermodynamic data on these fluids through EES: Engineering Equation Solver software, all documentation provided by Ener-G-Rotors on their current device uses the NIST database RefProp (NIST Refernce Fluid Thermodynamic and Transport Properties Database). This database is not open source and charges an accessing fee, but would greatly increase the computational functionality of the

Using this RefProf software, fundamental heat transfer equations, the fundamentals of fluid mechanics, and the overall heat transfer coefficients calculated during physical testing, I will create a model for the 5kW generator heat exchanger. Additionally, data collected on the performance of a larger heat exchanger used in the Ener-G-Rotors will be used to extrapolate the code to analyze larger heat exchangers, an important feature for the future applications of the model.

This code will also require the calculation of theoretical properties of binary mixtures of working fluids[5], an important part of the investigation which could increase efficiency due to the non-isothermal (zeotropic) vaporization phase during evaporation and condensing, and the potential for tailor-made fluids that could optimize performance across the entire range of acceptable heat sources and cooling fluid temperatures. These predictions will be done using equations of state, which are derived from various mixing principles, and the data accessed using RefProp.

### *Project Deliverables*

The results of this project will include a detailed analysis of the above proposed working fluids in the 5kW generator heat exchanger, as well as analysis of binary mixtures of the three refrigerants R-134a, R-236fa and R-245fa, a working coded model for the 5kW generator heat exchanger and potentially a generalized model adaptable for other heat exchanger configurations, particularly the 35kW generator heat exchanger. This will be presented to the Mechanical Engineering Faculty of Union College to be graded as my Senior Design Project, and to the Union College Community during the Steinmetz presentation day in May 2018. Additionally, all findings and the computerized model will be made available to Ener-G-Rotors for their future use to the benefit of future designs of their Rankine cycle generator.

To adhere to the Presidential Green Grant reporting requirements, in addition to the final research report submitted to the Union ME Faculty, summarized progress reports will be made available to U-Sustain at the end of each term of the project's duration (Fall 2017 and Winter 2018).

---

[5] Christophe Coquelet, C'eline Houriez, Jamal El Abbadi. "Prediction of thermodynamic properties of refrigerant fluids with a new three-parameter cubic equation of state." *International Journal of Refrigeration* 69 (2016): 418-436.

*Budget Proposal*

As stated above, I have applied for a student research grant to cover the costs of the initial materials needed to establish the working test bench for the analysis of the heat exchanger used in the 5Kw Ener-G-Rotors generator. The heat exchanger (valued at ~$600) itself will be loaned by Ener-G-Rotors with the potential for long term extensions of component. Union college is already in possession of several thermal reservoirs, which can supply the high and low temperatures needed for the exchanger analysis. Data acquisition can be done by the data acquisition systems supplied by the Mechanical Engineering Department and will not be consumed. However, to allow for greater flexibility in data acquisition, which will increase the variability of my computation model, I am requesting funds for additional flow meters and thermocouple temperature sensors, and an additional immersion heater. Additionally, I am requesting funding for the purchase of the NIST RefProp thermodynamic database software, which is used by Ener-G-Rotors in their cycle analysis. Free educational versions of RefProp are publicly available, however they offer only a limited selection of fluid data and do not cover the working fluids to be analyzed during this project.

Table 1: Itemized Budgetary Summary

| Item | Description | Supplier | Quantity | Unit Price | Total | Grand Total |
|---|---|---|---|---|---|---|
| RefProf | Downloadable Thermodynamic Database | NIST | 1 | $325.00 | $325.00 | |
| Type K Thermocouple | Temperature sensor to monitor fluid properties | OMEGA Instruments | 4 | $38.00 | $152.00 | |
| Immersion Heater | 1500 kW immersion tank heater | McMaster Carr | 1 | $250.00 | $250.00 | |
| Flow Meter | Inline water flow meter, for flows from 1-10 gpm | McMaster Carr | 2 | $137.20 | $274.40 | |
| Brass Inline Tee connectors | 1" to 1/4" Tee connectors to allow for thermocouple insertion into the fluid flow stream | McMaster Carr | 4 | $30.91 | $123.64 | |
| | | | | | | $1125.04 |

## Appendix C: Midterm Presentation Milestone Slides

### ANALYSIS OF MULTICOMPONENT WORKING FLUIDS IN FLAT PLATE HEAT EXCHANGERS

Jennifer Fromm
MER 497 Fall Milestone
Advisor: Prof. Richard Wilk

### Organic Rankine Cycle Background

[illegible]

### Ener-G-Rotors Mechanism

[illegible]

### Project Motivation

- ORC effectiveness limited by heat source and coolant temperatures
- Binary fluids offer potential to increase heat exchange effectiveness
  - 'custom' fluids
  - Vaporization temperature 'glide'

### Project Deliverables

- Report on heat exchanger effectiveness
- Analysis of Multi-component fluids in heat exchangers
  - Effect of temperature 'glide'
- Mathematical model for heat exchange in vaporizer and condenser
  - Account for multi-phase fluids

### Progress Update

- Test bed construction
- Characterization of UA of small heat exchanger (20 plates, 5x3")
- Research on binary mixture equations of state

## Test Bed Description

- Hot water reservoir (15 gallons)
  - 1.5kW immersion heater
- Small heat exchanger
  - Counter flow w/ cool municipal water
- Variable flow on hot and cold side
- Instrumentation
  - Thermocouples in tank, on inlet and outlet
  - Flow meters

## U Analysis

- Multiple data collection cycles
- U of small exchanger is 1030 W/K*m^2
- Given U: 8000 W/K*m^2

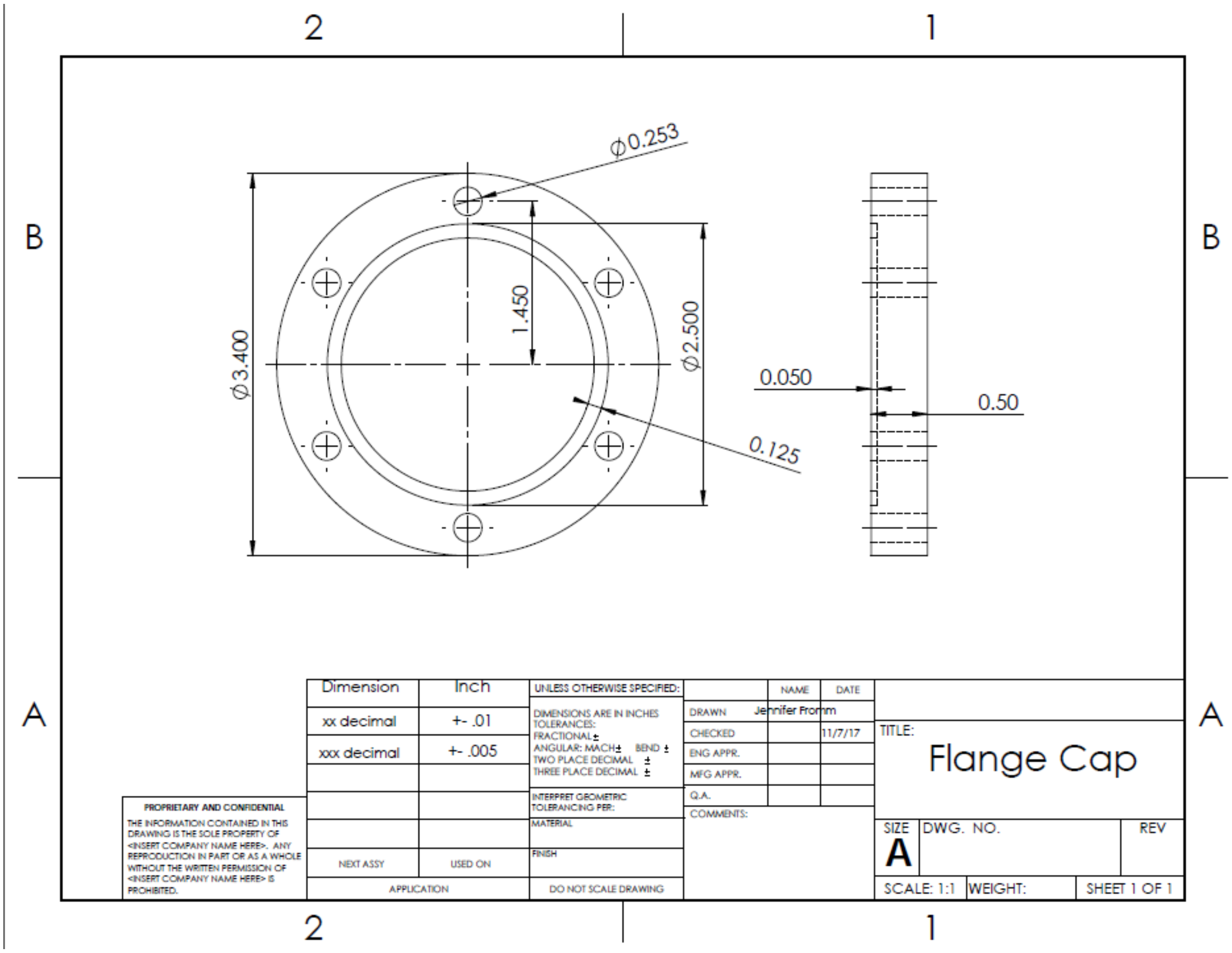
## Equations of State for Binary Fluids

- Use theoretical properties and empirical data (from outside sources)
- Calculate fluid properties as function of composition
  - T, P, s, h, Cp

## Future Steps

- Install donated HX from Ener-G-Rotors
- Compile research on calculation of multi-phase heat transfer
- Construct mathematical algorithm

## Appendix D: Flange Cap Schematic



## Appendix E: UA Calculations

The tank was heated using the immersion heater, the pump was run and water circulated through the heat exchanger until it reached a uniform temperature, and then the heater was turned off, and the temperature monitored until it reached near room temperature. The three thermocouples inserted into the stream recorded the internal fluid temperature, and these three measurements were averaged to get the mean tank temperature. The change in mean temperature over time is plotted below in Figure 5 and Figure 6, which give the data for the tank without and with insulation (respectively).

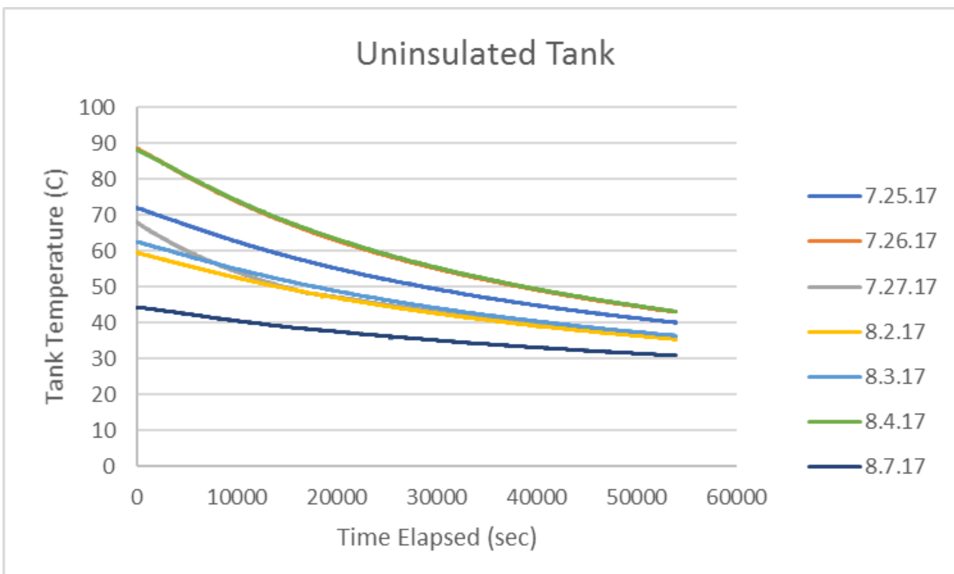



*Figure 5: Graph of the average temperature of the uninsulated heat source tank*

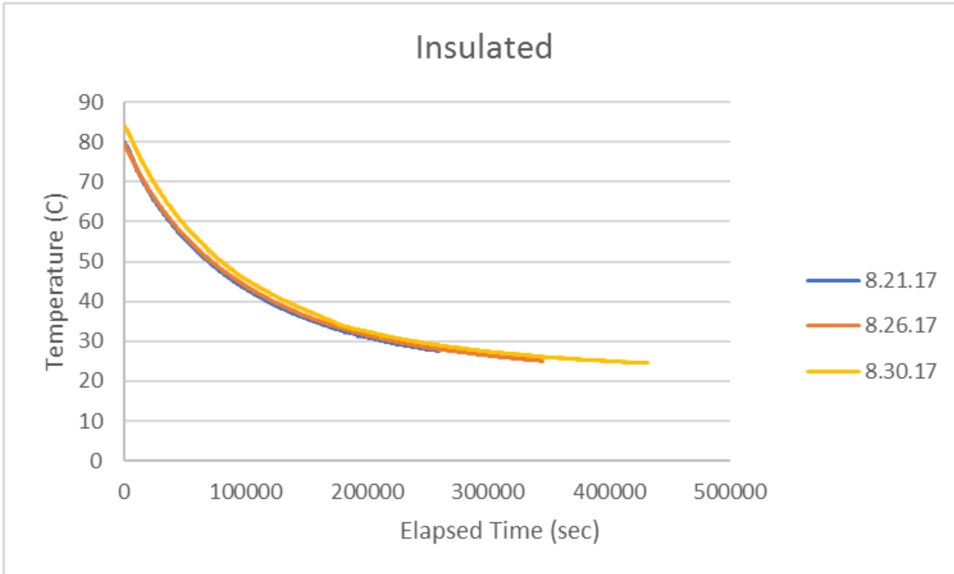



*Figure 6: Graph of the insulated heat source tank*

To determine the UA of each tank, the temperature gradients ($\theta$) were found with the following equations:

$$\theta = T_{Ave} - T_{\infty} \qquad (1)$$

$$\theta_i = T_i - T_{\infty} \qquad (2)$$

Where $T_{\infty}$ is the ambient temperature, $T_{Ave}$ is the mean temperature of the tank, and $T_i$ is the initial tank temperature at time = 0. The quotient of these temperature gradients was then plotted as a function of time, and exponential curves were fit to these graphs to determine the coefficient on time, with is equal to $\frac{1}{\tau}$, as shown in the following equation:

$$\frac{\theta}{\theta_1} = exp\left(-\frac{t}{\tau}\right) = \exp\left(\frac{U}{\rho VC} A_s t\right) \qquad (3)$$

The average of these time constants was found for both the insulated and uninsulated tank, from which the UA value for each set up was determined with the following equation:

$$UA = \frac{1}{\tau}\rho V \mathrm{C} \qquad (4)$$

Where the density was assumed to be the density of water at 1000 kg/m$^3$, the Volume was the volume of the tank at 15 gallons, and the specific heat is of water and 1008 J/kg*K.

Note that this calculation assumes the lumped capacitance of the tanks, a reasonable assumption given the small temperature differentials relative to the mass and heat capacitance of the tank. The method also assumes that the heat stored in the metal tank enclosure is negligible and does not account for them in the calculations.

## Appendix F: Small heat exchanger data

The first data trial is detailed in Figure 7. The volumetric flow rate of both the cool and hot side water were 1 gpm.

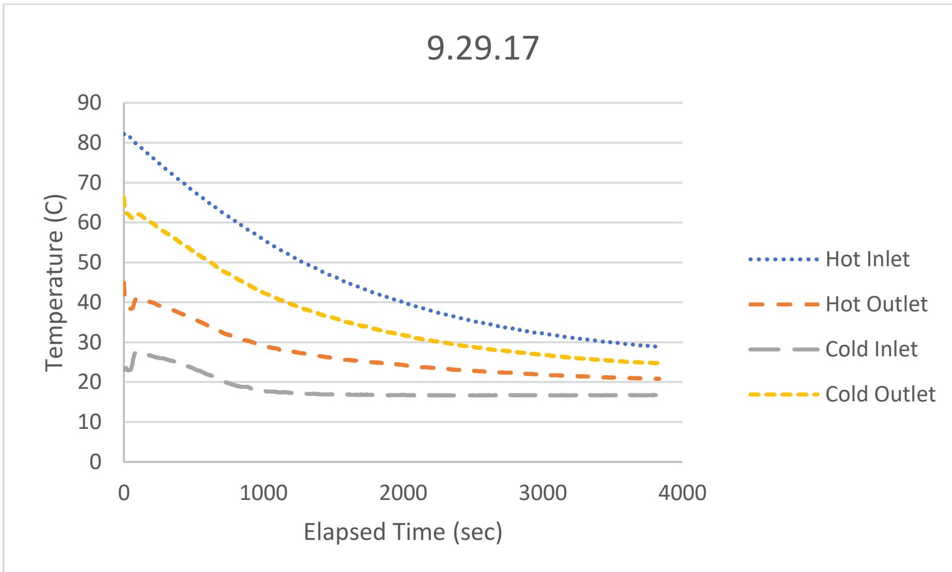



*Figure 7: Temperature plot for data trial on 9.29.17.*

For the second trial, the volumetric flow rate of the cold side was reduced to maintain the high temperature of the heat source tank, to 0.6 gpm, which the hot side flow volumetric flow rate was maintained at 1 gpm.

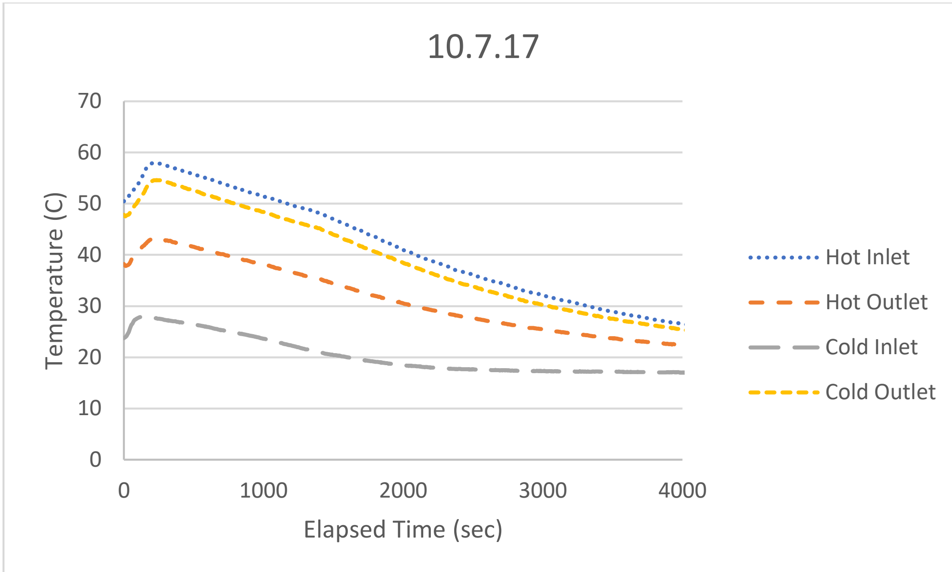



*Figure 8: Temperature Plot for data trial on 10.7.17*

For the third trial, the flow rate of both the hot and cool side was reduced to 0.6gpm.

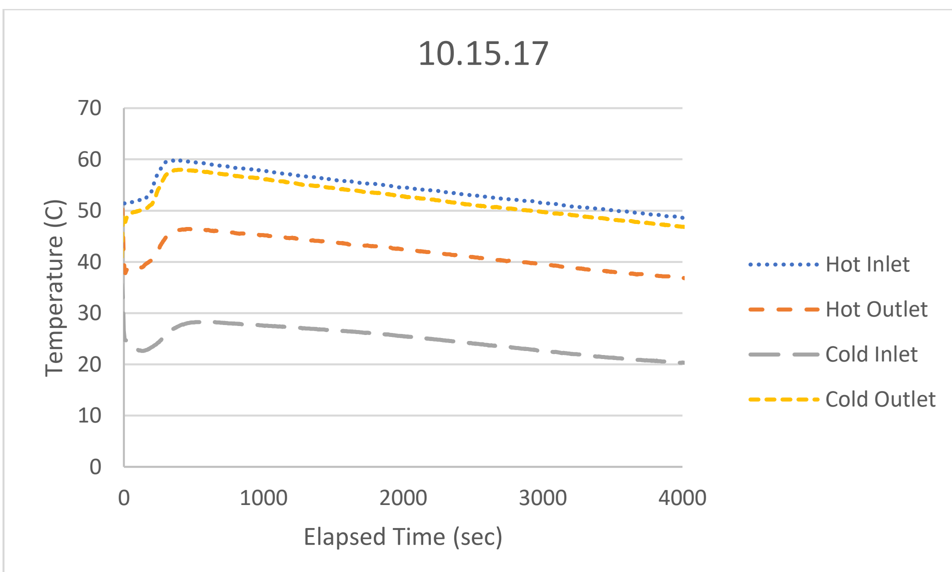



*Figure 9: Temperature plot for data trail 10.17.17*

Data files for all trials are available upon request.

## Appendix G: MatLab Scripts of the final model version

```
function [Q, Tco_Celcius, x] = predictor_app(app)

            %% This function takes in values for the inlet temperatures
            %% of the hot and cold side fluids, the names of the fluids,
            %% and their mass flow rates.
            %% It calculates the outlet temperatures and heat transfer
            %% over a range of fluid concentrations and outputs them as
            %% vectors which can be plotted.

            %specify inputs
            ThiC = app.Thi_C.Value; %C
            TciC = app.Tci_C.Value; %C
            Thi = ThiC+ 273; %K
            Tci = TciC + 273; %K
            mdot_h = app.m_h.Value; %mass flow rate of cold side kg/s
            mdot_c = app.m_c.Value; %mass flow rate of hot side kg/s
            F_hot = app.Fluid_hot.Value; %name of heating fluid
            F_cold_1 = app.Fluid_cold_1.Value; %name of Refrigerant
component 1
            F_cold_2 = app.Fluid_cold_2.Value; %name of Refrigerant
component 2
            Pc = app.Pressure_c.Value; %estimated Pressure in KPa
            Ph = app.Pressure_h.Value; %estimated Pressure in KPa

            %Givens
            Tave = (Thi + Tci)/2; %Average temp in K
            T_LU_h = Tave+ 0.5*(Thi - Tave);
            T_LU_c = Tave + 0.5*(Tci - Tave);
            %Heat exchanger inputs
            Dh = 0.029;%m, effective Diameter of the 'tube', empiracally
derived to fit data
            A = 2.4; %m^2, area of heat transfer in the heat exchanger
            Nu_l = 7.5; %Nusselt number for laminar flow in infinite
rectangular 'tube'
            x_copper = 0.0005; %m
            k_copper = 200; %W/m*K
            %Initialize Output Vectors

            Z = zeros(1,11);
            UA = Z;
            Q = Z;
            Tco = Z;
            Tho = Z;
```

```
            Tco_Celcius = Z;
            Tho_Celcius = Z;

            %loop to vary composition
            x = 0:0.1:1;
            for n = 1:length(x)
                X = [x(n), (1-x(n))] ; %composition of refrigerant by mass

                %look up
                k_c = refpropm('L', 'T', T_LU_c, 'P', Pc, F_cold_1,
F_cold_2, X);
                k_h = refpropm('L', 'T', T_LU_h, 'P', Ph, F_hot);
                Cp_c = refpropm('C', 'T', T_LU_c, 'P', Pc,
F_cold_1,F_cold_2, X);
                Cp_h = refpropm('C', 'T', T_LU_h, 'P', Ph, F_hot);
                Pr_c = refpropm('^', 'T', T_LU_c, 'P', Pc,
F_cold_1,F_cold_2, X);
                Pr_h = refpropm('^', 'T', T_LU_h, 'P', Ph, F_hot);
                V_c = refpropm('V', 'T', T_LU_c, 'P', Pc, F_cold_1,
F_cold_2, X);
                V_h = refpropm('V', 'T', T_LU_h, 'P', Ph, F_hot);
                %Calculate 2-phase nusselt number
                Re_c = mdot_c * (Dh / V_c);
                Nu_2p = 4.188 * Re_c .^0.4 .* Pr_c .^(1/3);
                %Calculate UA
                U = 1 / ((Dh/(k_c*Nu_2p)) + (x_copper/k_copper) +
(Dh/(k_h*Nu_l))); %water-refrigerant
                UA(n) = U*A;

                %Calculate the exit temperatures for 2 one phase
                k = UA(n) * ( (1/(mdot_h*Cp_h)) - (1/(mdot_c*Cp_c)));
                Q(n) = ((exp(k) - 1)*(Thi - Tci)) / ((exp(k)
/(mdot_h*Cp_h)) - (1/(mdot_c* Cp_c)));
                Tco(n) = (Q(n)/(mdot_c*Cp_c)) + Tci;
                Tho(n) = -(Q(n)/(mdot_h*Cp_h)) + Thi;
                Tco_Celcius(n)= Tco(n) - 273;
                Tho_Celcius(n) = Tho(n) - 273;
            end
        end
```