Working Fluid Selections in Organic Rankine Cycle Used for Internal Combustion Engine Exhaust Waste Heat Recovery

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ABSTRACT

This project is a feasibility study to investigate the possible substances that can be used in a bottom Rankine cycle in connection with a combustion heat engine, such as a regular gasoline or diesel engine. This Rankine cycle, sometimes called organic Rankine cycle, converts some of the engine exhaust gas energy to useful shaft work. This cycle has a lower high temperature and uses the atmosphere as the low temperature energy reservoir for its condenser. For this reason, the working substance must be selected so the phase change and pressures are suitable for the application. The working substances investigated are mainly within the scope of pure substances. Mixtures are only briefly investigated due to their complexity. The basic approach of this project is a theoretical model of the system built in MATLAB software with the thermal properties provided by NIST (National Institute of Standards and Technology) database. Important characteristics of the cycle, such as thermal efficiency, maximum shaft work produced, high pressure, low pressure and mass flow rate, are calculated with various working substances for different operating conditions. These conditions are important design characteristics, such as exhaust gas temperature, turbine isentropic efficiency and heat exchanger efficiency. The final selection is based on the thermal efficiency of the cycle, net work produced, feasibility of the cycle, and concerns regarding environmental impact and safety. The results of this project are best choices of working substances for different exhaust gas temperatures. Mixtures are briefly studied for the purpose of future research.

Keywords: organic Rankine cycle (ORC), combustion engine, exhaust gas, working substance.

CONTENTS

1. INTRODUCTION

The internal combustion engine is the core power source for many vehicles. Due to the current stringent energy demands and high fuel costs, manufacturers of vehicles have been seeking methods to increase the thermal efficiency of the internal combustion engine to produce more work given the same amount of fuel consumptions [1]. However, researches has indicated that internal combustion engines still lose around 60-70% of the fuel energy through the exhaust gas and the coolant. The exhaust gas tends to have a high enough temperature compared to the coolant temperature [2] so it has a potential to drive a low temperature heat engine cycle. Therefore, Waste Heat Recovery (WHR) from the exhaust gas to achieve higher efficiency of the internal combustion engine is worth attention.

There are various thermodynamic cycles, such as the organic Rankine cycle (ORC), Kalina cycle, Goswami cycle, and trilateral flash cycle, available to convert the low-grade heat in the exhaust gas to usable mechanical work or electricity. Among these cycles, the ORC has shown its high competiveness because of its higher reliability and it is less complex [3].

There are hundreds of substances available in the world to be used in the ORC [4]. The best working fluids should be determined for specific cases. Much research has been done on the working fluid selections for the ORC used as a method of WHR from engine exhaust gas since 1976 [1]. However, most of the research focused on some specific conditions, such as a fixed turbine inlet temperature, a fixed turbine inlet pressure, or a specific type of engine. There is a lack of generic but yet qualitative and simple solutions to the choice of the working fluid in the ORC when used in connection with an internal combustion engine.

The objective of this project is to determine the best choices of working fluids among pure substances for an ORC used in connection with an engine through a simple and generic solution. This project also briefly studies the use of mixtures in the ORC. Due to the time limitation, cycle simulation for mixtures is only to demonstrate the accommodation of the model to mixtures rather than provide the best choices of mixtures. The approach of this project is a literature review and a computer model simulation of a simple ORC cycle.

This report includes a literature review, simulation descriptions, simulation results, sensitivity studies, simulation validations, and a brief study of mixtures.

2. LITERATURE REVIEW

To familiarize the readers with the ORC, a short introduction of the ORC from literature is included in this section. In addition, preliminary working fluid selections were determined through this literature review. The selection process includes a statistical summary of the substances that were studied previously, an introduction of fluid categories, environmental impact and safety concerns.

2.1. Organic Rankine Cycle

The organic Rankine cycle (ORC) is a closed-loop system where a naturally organic working fluid repeatedly circulates through four components, an evaporator (aka. boiler), an expander, a condenser, and a pump, to transform waste heat into mechanical or electrical power [1]. The ORC differs from the traditional steam Rankine cycle because it uses organic fluids with low boiling points instead of water to recover heat from lower temperature heat sources [3]. Figure 1 shows a configuration of an ORC and its corresponding T-s diagram.

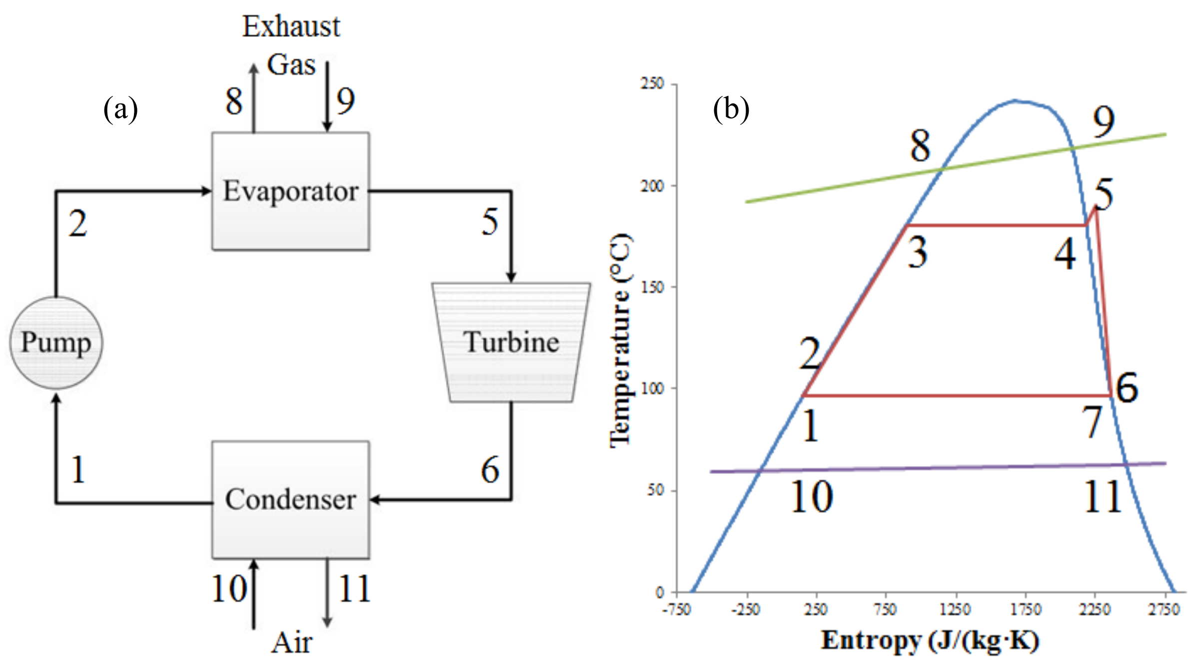

Figure 1. (a) Configuration of an organic Rankine cycle and (b) its corresponding T-s diagram.

As explained by Sprouse, C. and Depcik, C. [1], process 2-5 in Figure 1 is an isobaric heating process through which the working fluid is heated by the engine exhaust gas to capture the waste heat available. This process occurs in an evaporator (boiler). Process 5-6 in Figure 1 is an ideal isentropic process through which the working fluid expands and produces mechanical or electrical power. The expander used in this process is usually a turbine or a displacement expander. The working fluid then goes through the condenser to discharge low-grade heat to the environment, as shown by isobaric process 6-1 in Figure 1. A pump eventually compresses the working fluid back to state 2 through an ideal isentropic process 1-2. This process only requires a small amount of work compared to the rest of the cycle, thus the difference between 1 and 2 cannot be clearly identified on Figure 1 (b). In addition, state 7 indicates the location where saturated vapor occurs in process 6-1. However, since state 6 is already on the saturated vapor line, there is no difference between state 6 and 7 on Figure 1 (b). Further, line 8-9 and line 10-11 only show the simplified temperature profiles of the exhaust gas and the coolant (air), so there are no entropy indications for these two lines.

2.2. Working Fluid Selections

There are hundreds of substances that can be adopted as the working fluid candidates in the ORC, including numerous pure substances and mixtures [4]. To narrow down the number of candidate working fluids and facilitate working fluid selection process, a review of the previous literatures regarding the use of working fluids in the ORC is necessary. Promising working fluid candidates, which will be tested in my model, are summarized at the end of this section. The focus of this section is on pure substances.

2.2.1. Previous research

Previous research on the ORC used in connection with an internal combustion engine is summarized in this section. Due to the limitation of resources available, I used the review done by Sprouse and Depcik [1] as a guideline for this section. Major conclusions from the literature were mostly extracted from [1] instead of the original references because of limited access to the references.

In 1976, Patel and Doyle [5] recovered the exhaust waste heat of a Mack 676 diesel engine in a long haul truck by an ORC using Fluorinol-50 as the working fluid. The operating temperature was between 650 °F (343.3 °C) at turbine inlet and 158 °F (70 °C) at condenser exit. They claimed a 13% increase in maximum power output along with a 15% improvement in fuel economy [1].

In 1985, Badr [6] documented a working fluid selection process for a Rankine cycle engine producing a low power (<10 kW) and operating between 40 °C and 120 °C. 67 prospective working fluids were evaluated, among which three superior candidates were R-11, R-113, and R-114, while R-11 was identified to be unstable at temperature above 120 °C [1].

In 1995, Larjola [7] used an ORC with prototype high-speed oil free turbogenerator-feed pumps to recover heat from a 425 °C source. Among the several working fluids that he tested, toluene showed the best suitability. The toluene-based ORC had 26% efficiency compared to the 11-19% efficiency achieved by a steam Rankine cycle [1].

In 1997, Hung [8] evaluated the performance of six working fluids, benzene, ammonia, R-11, R-12, R-134a, and R-113, by modeling to determine the maximum Rankine cycle efficiency at different turbine inlet temperatures. The result showed that benzene had the highest efficiency from 500-550 K (227 °C – 277 °C) [1].

In another work published four years later, Hung [9] investigated the potentials of benzene, toluene, p-xylene, R-113 and R-123 to recover waste heat from a 10 MW source at 600 K (327 °C) in an ORC. P-xylene showed highest cycle efficiency when a constant 15 °C temperature difference between the turbine inlet and the waste heat source existed. Refrigerants showed better performance as the source temperature decreased [1].

In 2005, El Chammas and Clodic [10] configured an ORC for WHR from the cooling circuit and exhaust of a 1.4 L spark ignition engine in a hybrid vehicle using a 55 °C condensing temperature. They tested water, isopentane, R-123, R-245ca, R-245fa, butane, isobutane, and R-152a as the working fluids. The results indicated that water gave the highest efficiency, followed by R-123, isopentane, and R-245ca [1].

In 2007, Mago et al. [11] investigated the performances of R-134a, R-113, R-245ca, R-245fa, R-123, isobutane, and propane as the working fluids in an ORC operating at low temperatures. R-113 showed highest system thermal efficiency at temperature around 450 K (177 °C) [1].

In the same year, Quoilin [12] described optimization of a small scale ORC through computer simulations. He indicated that R-123 operated efficiently with source temperatures between 100 and 200 °C [1].

In 2009, Ringler et al. [13] found that water was the most appropriate working fluid for an ORC that works with a four-cylinder engine to recover heat from exhaust gas only [1]. The investigation was facilitated by a Dymola modeling tool, thus the engine performance directly linked to the vehicle speeds [1].

In 2010, Espinosa et al. [14] studied the optimal ORC configuration for WHR on commercial trucks. In addition to the optimal configuration discussion, he used computer models to evaluate three working fluids, water, ethanol, and HFC-R-245fa. R-245fa was deemed as the most suitable working fluid [1].

In 2011, Roy et al. [15] parametrically optimized the performance of an ORC using R-12, R-123, R-134a, and R-717 as the working fluids. R-123 demonstrated the highest efficiency for both a constant hear source temperature of 550 K (227 °C) and a variable heat source.

In 2012, Seher, Lengenfelder, Gerhardt, Eisenmenger, Hackner and Krinn [16] compared WHR power produced by an ORC in connection with a diesel engine for a heavy duty commercial vehicle using water, toluene, MM, ethanol, and R-245fa. The results obtained by both simulations and experiments indicated that water or ethanol is the suitable working fluid.

In 2013, Bao and Zhao [4] reviewed working fluid selections for ORC. They summarized selection criteria based on thermodynamic and physical properties, such as latent heat and boiling temperature. They proposed a table of recommended fluids for different applications, working conditions and performance indicators. Over the heat source temperature range of 320-500 K (47-227 °C), 24 working fluids were recommended in their work.

In short, previous researchers have done large amount of work on the ORC used in connection with internal combustion engines. However, the assumed operating conditions of the ORC and the results in their works vary largely. No single working fluid is suitable for all conditions. Therefore, rather than a further detailed study on why the results of these literatures vary largely, a simple statistical summary is more helpful to narrow down the working fluid selections. Based on this consideration, I summarized that the working fluids that were frequently investigated in the literatures are R-11, R12, R-113, R134a, R123, R-245ca, R-245fa, isobutane (R-600a), toluene, benzene, ethanol and water. The working fluids that have demonstrated the highest system efficiencies in the literatures are R-11, R-113, R-114, R-123, R-245fa, toluene, benzene, p-xylene, ethanol and water. Note that R-123 showed the best performance in three studies ([10], [12] and [15]).

2.2.2. Working fluids' category

The classification of the working fluid in an ORC as a dry, isentropic, or wet fluid is one of the most important characteristics of the working fluid [8]. This classification is according to the slope of the vapor saturation curve of the working fluid on a temperature-entropy (T-s) diagram. A dry fluid has a positive slope, a wet fluid has a negative slope, and an isentropic fluid has a nearly vertical (infinitely large) slope [4]. Figure 2 renders this classification by the T-s diagrams of candidate fluids with the saturated vapor curve shown in red.

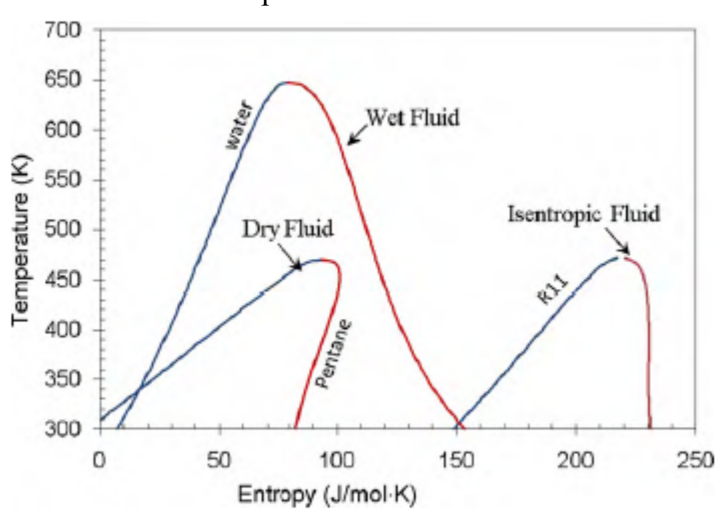

Figure 2. Three types of working fluids: dry, isentropic, and wet [3].

As discussed by Bao and Zhao [4], isentropic or dry fluids can avoid liquid droplet impingent in the turbine blades during the expansion process without the need of being superheated. This can save the cost on the superheated apparatus and reduces the cycle complexity. Therefore, dry or isentropic fluids are more suitable for an ORC system. However, a very dry fluid can cause energy waste and add loads to the condenser, because it leaves the turbine with substantial "superheat" [3]. Thus, a fluid that is too dry should be avoided in the ORC.

Chen, Goswami, and Stefanakos [3] summarized the classifications (dry, wet, and isentropic) of different fluids by two T-ξ charts, where T is temperature and ξ is defined as the inverse of the vapor saturation slope on a T-s diagram. Incorporating the results shown by these two charts and the considerations of latent hear, density and specific heat on the system performance, they proposed benzene, toluene, R-141b, R-123, R-21, R-245ca, R-245fa, R-236ea, R-142b, R-601, R-600 and R-600a as likely candidates for the ORC.

2.2.3. Environmental impact and safety aspects

The major concerns regarding the environmental impact are the ozone depletion potential (ODP), global warming potential (GWP) and the atmospheric lifetime (ALT). The ODP and GWP represent substance's potential contributions to ozone degradation and globe warming. Some working fluids that have already been phased out are R-11, R-12, R-113, R-114, and R-115. Some other working fluids are being phased out in 2020 or 2030, such as R-21, R-22, R-123, R-124, R-141b and R-142b [3].

The fluid's level of danger can be indicated by the ASHRAE refrigerant safety. In general, characteristics such as noncorrosive, non-flammable, and non-toxic are preferred. However, these characteristics are not always practically useful or critically necessary. Many substances that are considered flammable do not ignite if there is no ignition source around. Nevertheless, auto ignition is still a problem for longer alkanes at temperatures above 200 °C. The concentrations and explosion limits for these fluids should be taken into account for safety [3].

2.2.4. Summary

Combining the results from the previous three sections and Bao and Zhao's work [4] (This work was described previously, but the detailed results were not presented for space economy.), I decided to further investigate nine working fluids, which are toluene, benzene, water, ethanol, R-600a (p-xylene), R-245fa, R-245ca, R-123, and R-236ea. Water is a very wet fluid (refer to section 2.2.2), thus it may increase the cycle complexity because of the need of a superheater. However, water is the most commonly used working fluid in the Rankine cycle and can be used for comparisons with other working fluids.

3. SIMULATION DESCRITION

Computer models were built to simulate a simplified ORC in order to test the nine working fluid candidates. This section includes the objectives of the simulation, the simulation assumptions, and an illustration of the model.

3.1. Objectives

The objective of the simulation is to find important cycle characteristics for different working fluids in order to assist working fluid selections. First law thermal efficiency is one of these important characteristics, because it shows us the overall performance of the cycle. However, we are interested in the total performance of the whole system including the engine and the ORC (how much work is increased by adding the ORC) regardless of the cycle performance in many cases. The total performance is better to be shown as the total amount of work produced by the cycle, because the engine is unknown in this project. First law efficiency and amount of work produced are the two major concerns in my model. Besides these two characteristics, we are also interested in the maximum and minimum pressures in the cycle, because these will determine the feasibility of the cycle. We also want to know the mass flow rates of the working fluid in the cycle and the coolant in the condenser, which reflects the size of the cycle and determines whether the cycle is suitable for vehicle use. In addition, the exhaust gas temperature variations should be captured in the model, because the engine works at different conditions.

In short, the objective of this model is to find the first law thermal efficiency of the cycle, the amount of work produced by the cycle, the maximum and minimum pressures (pressure profiles) in the cycle, the mass flow rate of the working fluid in the cycle, the mass flow rate of the coolant in the condenser, while capturing the exhaust gas temperature variations.

3.2. Assumptions and Limits

In order to simplify the model, assumptions and limits were made. These assumptions and limits are subject to a sensitivity study in section 5. Figure 3 shows the assumptions and limits made on the model. Figure 3 is based on Figure 1 (b). Temperature profile of the exhaust gas in the counter-flow evaporator and that of the coolant in the counter-flow condenser are also shown on Figure 3.

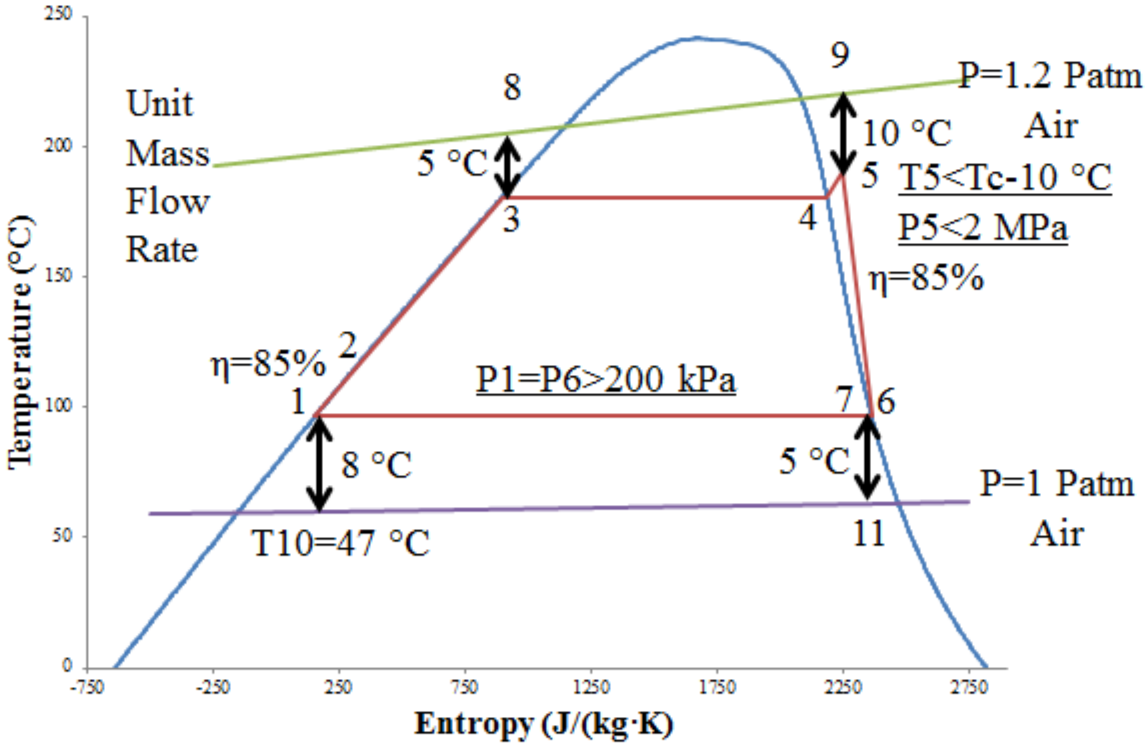

Figure 3. Assumptions and limitations (underlined) on the ORC model.

The exhaust gas was assumed to have the same composition as air. The coolant in the condenser was also assumed to be air, because air is the ultimate coolant on a vehicle. The isentropic efficiencies of both the pump and the expander were assumed to be 85%. The temperature difference between the working fluid and the exhaust gas at the outlet of the evaporator was assumed to be 10 °C ($\Delta T_{9\text{-}5}$ on Figure 3). The temperature difference at the pinch point in the evaporator was assumed to be 5 °C ($\Delta T_{8\text{-}3}$ on Figure 3). The temperature difference between the working fluid and the air at the outlet of the condenser was assumed to be 8 °C ($\Delta T_{1\text{-}10}$ on Figure 3). The temperature difference at the pinch point in the condenser was assumed to be 5 °C ($\Delta T_{7\text{-}11}$ on Figure 3). These temperature differences reflect heat exchanger efficiencies, which should be decided in detail when design specifications are available. The temperature of the air before it enters the condenser was assumed to be 47 °C (T_{10} on Figure 3) according to [10]. The pressure of the exhaust gas in the evaporator was assumed to be 1.2 atmosphere pressure (1.2×101.325 kPa), and the pressure of the air in the condenser was assumed to be one atmosphere pressure. The mass flow rate of the exhaust was set to be unit mass flow rate, because this can provide us with a reference to the calculated mass flow rate of the working fluid and that of the condenser air without knowing the engine displacement and operating conditions. In addition, the heat exchange processes in the evaporator and the condenser were assumed to be externally adiabatic so that there is no heat loss.

Bao and Zhao [4] provided some practical limitations of the ORC. According to their work, the maximum pressure (turbine inlet pressure at 5 on Figure 3) in the cycle cannot exceed 2000 kPa. The minimum pressure (condenser pressure at 1, 6 and 7 on Figure 3) cannot be below 200 kPa. The maximum temperature (turbine inlet temperature at 5 on Figure 3) cannot exceed the critical temperature of the working fluid minus 10 °C, above which working fluids can be instable. When this temperature limit is hit, the model will stop calculating for the current working fluid and jump to the next working fluid. The pressure limits can override the assumptions stated before. For example, the temperature of the saturated water at 200 kPa (the minimum pressure allowed) is about 100 °C [18]. The condensing air temperature (T_{10}) will then be 92 °C ($100\ °\mathrm{C} - \Delta T_{1-10}$) instead of the assumed 47 °C.

3.3. Model Illustration

A general illustration of how the model works is included in this section. The first step of modeling is to find each state numbered on Figure 3 using the assumptions and the given conditions. After each state is known (temperature, pressure, enthalpy, and etc. at each state are known), the model will do calculations to determine the system characteristics, such as efficiency and work produced. In this section, we refer T_x, P_x and h_x to the temperature, pressure and specific enthalpy at the state numbered as x (x is 1, 2, 3 and etc.) on Figure 3. The thermodynamics properties of the fluids for cycle calculations are given by NIST [18] as a MATLAB [19] function. The total model is based on MATLAB.

To better illustrate the state determination process, T_9 (the exhaust gas temperature before it enters the evaporator) and P_5 (turbine inlet pressure) are treated as known variables first. T_1 was calculated by $T_{10} + \Delta T_{1-10}$, whose values were assumed. State 1 can then be determined, because it is saturated liquid. P_6 and P_7 are the same as P_1, because the condensing process is isobaric. State 7 can be determined since it is saturated vapor. T_5 was calculated by $T_9 - \Delta T_{9-5}$, where T_9 is treated as known variables and value of ΔT_{9-5} was assumed. State 5 can be determined due to the known T_5 and P_5. By knowing state 5, P_6, and the assumed turbine isentropic efficiency, state 6 can be determined. P_2 and P_3 are the same as P_5, because process 2-5 in the evaporator is isobaric. State 2 was solved by the known state 1, P_2 and isentropic pump efficiency. State 3 can be determined since P_3 is known, and state 3 is saturated liquid. Further, T_8 is equal to $T_3 + \Delta T_{8-3}$, and T_{11} is equal to $T_7 + \Delta T_{7-11}$, where ΔT_{8-3} and ΔT_{7-11} have their assumed values . As we already assumed pressure of the exhaust gas in the evaporator and the pressure of the air in the condenser, state 8, 9, 10, 11 can be determined. The fluid properties at each important state are now known.

We further calculate the cycle characteristics by the properties at each important state that we have determined. The basic equations to calculate cycle characteristics are given by [18]. The first law efficiency (η) was calculated by Equation 1,

$$\eta = \frac{(h_5 - h_6) - (h_2 - h_1)}{h_5 - h_2} \quad \text{(Eq. 1)}$$

where h_i is the specific enthalpy at different states. The mass flow rate of the working fluid ($\dot{m}_{wf}$) was calculated by Equation 2,

$$\dot{m}_{wf} = \frac{\dot{m}_{ex}(h_9 - h_8)}{h_5 - h_3} \quad \text{(Eq. 2)}$$

where $\dot{m}_{ex}$ is the assumed unit mass flow rate of the exhaust gas. Equation 2 assumes no heat loss in the evaporator. The mass flow rate of the air in the condenser ($\dot{m}_{air}$) was calculated by Equation 3,

$$\dot{m}_{air} = \frac{\dot{m}_{wf}(h_7 - h_1)}{h_{11} - h_{10}} \quad \text{(Eq. 3)}$$

Equation 3 assumes no heat loss in the condenser. The amount of work produced by the cycle ($\dot{W}$) was calculated by Equation 4,

$$\dot{W} = \dot{m}_{wf}(h_5 - h_6) \quad \text{(Eq. 4)}$$

The maximum pressure in the cycle is the turbine inlet pressure P_5, while the minimum pressure in the cycle is the condenser pressure P_1, P_6 or P_7.

So far, the cycle calculations have been based on the statement that T_9 and P_5 are known variables. Given a T_9 (or T_5 since $T_5 = T_9 - \Delta T_{9-5}$), P_5 can vary from the saturated vapor pressure at T_5 to the condensing pressure P_1, P_6 or P_7 (condensing pressure is independent of P_5). Therefore, to optimize the cycle performance, the model will internally calculate the efficiency or work at different P_5 given a T_9, and will only report the cycle characteristics when the cycle performance (either efficiency or work) is optimized at a given T_9. Figure 4 shows this idea.

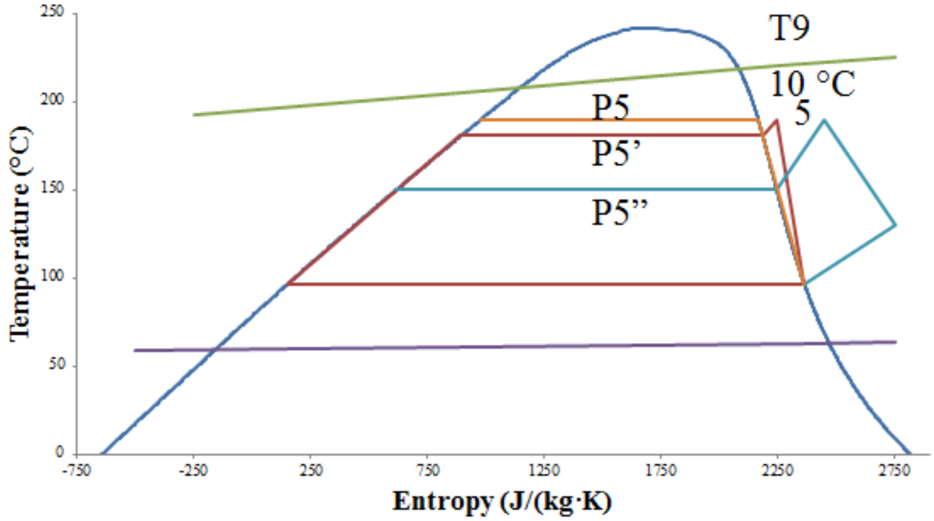

Figure 4. Different pressures (P5, P5', P5", etc.) can be chosen as the turbine inlet pressure. The model picks the pressure that optimizes the cycle performance.

In addition, T_9 (Figure 3 and 4) is the exhaust gas temperature, which can vary according to the engine types and operating conditions. Thus, to capture this variation, the model will plot the cycle characteristics at the optimized state against exhaust gas temperature from 80 °C to 380 °C. Note that all the limitations stated before will be imposed on the model. The MATLAB code of the model is included in Appendix.

4. SIMULATION RESULTS

The model was run for the nine working fluids listed in section 2.2.4. The simulation results and discussions of these results are given in this section. The results include those discussed in section 3.1. In this section, the plotting stops when the temperature reaches the critical temperature minus 10 °C, which is the general practical limit of each working fluid [4].

4.1. Efficiency and Work

Figure 5 shows the optimized first law efficiency at different exhaust gas temperatures. Figure 6 shows the optimized amount of work produced by the cycle assuming a unit mass flow rate of the exhaust gas (1 kg/s).

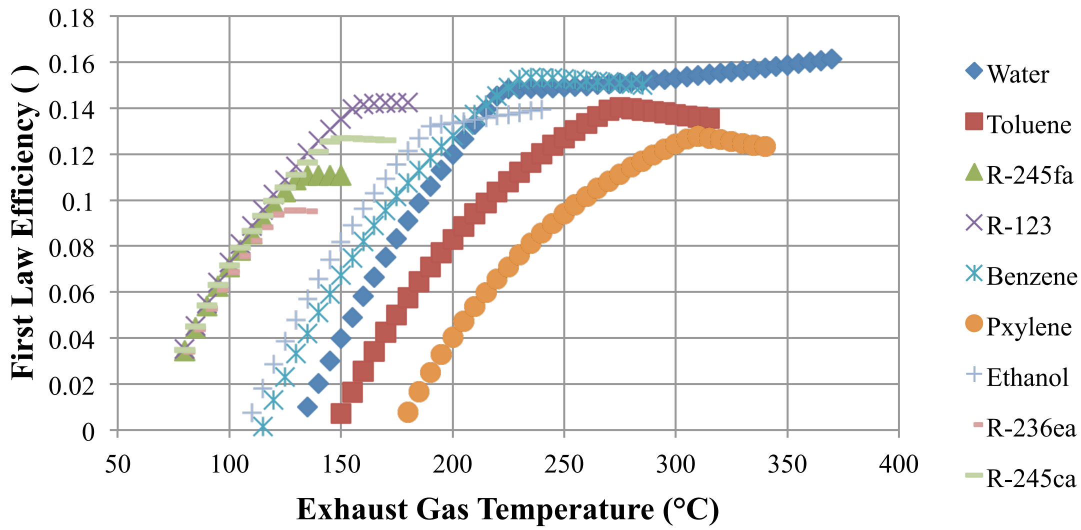

Figure 5. Optimized thermal efficiency against exhaust gas temperature.

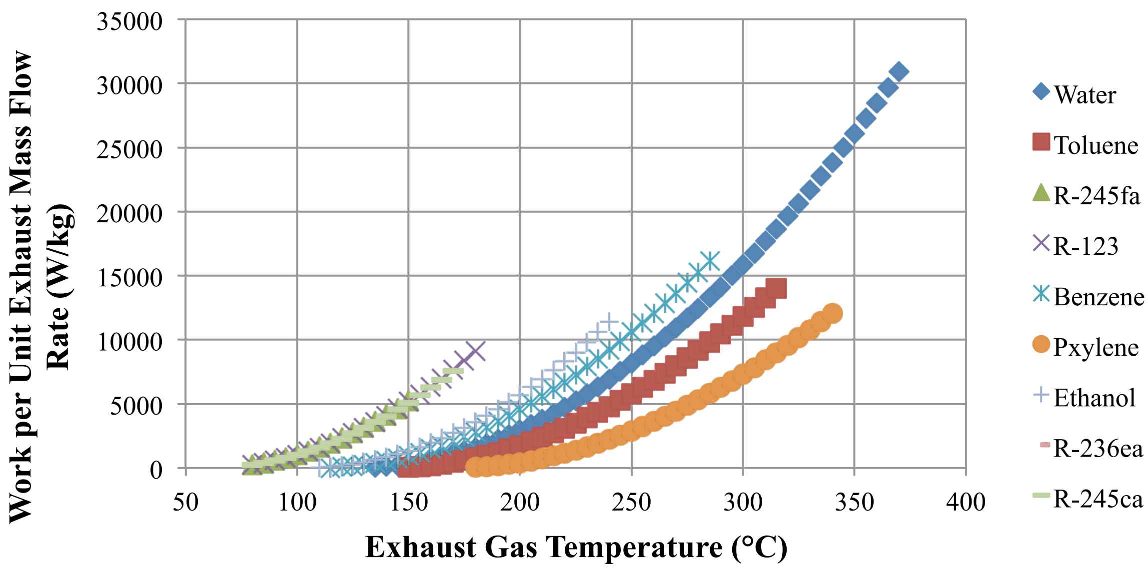

Figure 6. Optimized work per unit exhaust gas mass flow rate against exhaust gas temperature.

As shown by Figure 5, R-123 has larger thermal efficiency over the exhaust temperature range of 80-180 °C. Ethanol has larger thermal efficiency over the exhaust temperature range of 180-210 °C. Benzene shows larger thermal efficiency at 210-280 °C. Finally, water is the best working fluid at exhaust temperature above 280 °C. Figure 6 shows a similar prediction as Figure 5 with the only difference being that ethanol produces greater amount of work from 180 to 240 °C instead of 210 °C.

Refrigerants such as R-123 have lower practical temperature limits [4], thus they are highly recommended for low temperature sources. However, although R-123 has good performance, it will be phased out in 2020 or 2030 [3]. If the ORC will be operated for a larger exhaust temperature range, ethanol or benzene is suggested depending on whether we are more interested in thermal efficiency or work produced. If we want to extract as much work as possible out of a given amount of heat, we should focus on thermal efficiency. Nevertheless, if we only want more work produced regardless of how much heat is taken out of the exhaust gas, we should use Figure 5 as an indicator. Finally, water is still the best solution at higher temperature range.

4.2. Pressure Profiles

Figure 7 shows the maximum pressure profile (turbine inlet pressure) in the cycle when thermal efficiency is optimized at different exhaust gas temperatures. Figure 8 shows the minimum pressure profile (condensing pressure) in the cycle when thermal efficiency is optimized at different exhaust gas temperatures.

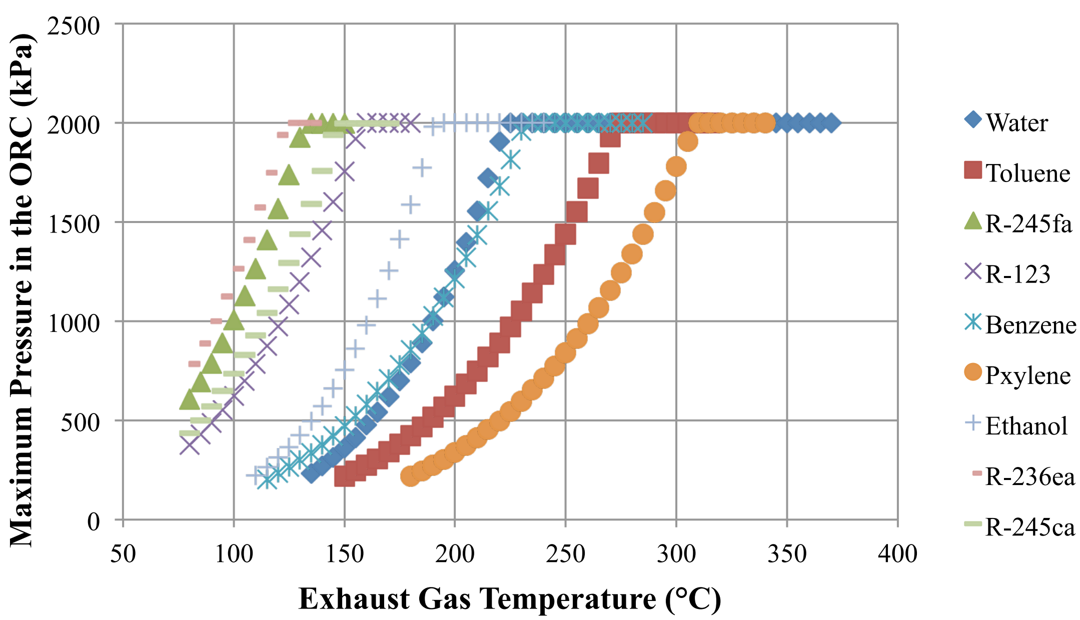

Figure 7. Maximum pressure in the ORC when thermal efficiency is optimized.

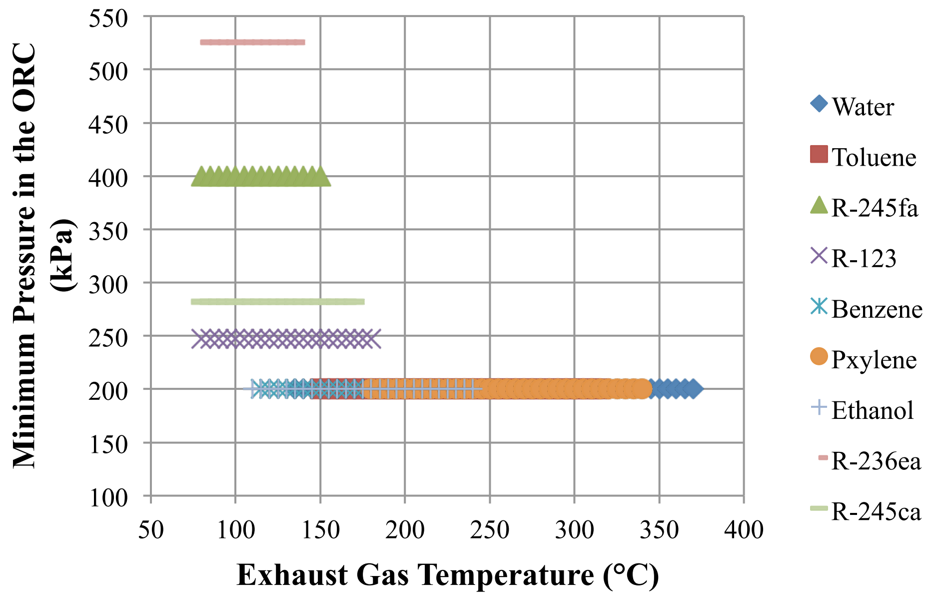

Figure 8. Maximum pressure in the ORC when thermal efficiency is optimized.

As explained in section 3.2, pressure limits have been imposed on the ORC. From Figure 7, we can see that all working fluids reach the upper pressure limit as the exhaust temperature goes up. Comparing to Figure 5, we can see that the upper pressure limit actually prevents the thermal efficiency from further increasing. However, the pressure limit does not "cap" the work in Figure 6. These two observations lead to a conclusion that the increase in the heat taken out of the exhaust gas outruns the increase in the work so that the thermal efficiency drops as the fluid reaches the pressure limit. In another word, the pressure limit essentially limits the temperature at the pinch point in the evaporator (state 3 on Figure 3), and thus limits the exhaust gas temperature at the pinch point (state 8 on Figure 3) since $T_8 = T_3 + \Delta T_{8-3}$. While T_8 (Figure 3) is limited, T_9 (Figure 3) is increasing along the horizontal axis. This gives a steeper exhaust gas temperature profile across the evaporator (temperature difference between states 3 and 5 on Figure 3 also becomes larger), and thus poor thermal efficiency of the cycle. In short, the work is unaffected by the maximum pressure limit, but the heat taken out of the exhaust gas is affected by the limit, which makes the efficiency also affected by the limit.

From Figure 8, we can observe that five working fluids (those non-refrigerants) are right on the lower pressure limit. This implies that these working fluids prefer a lower limit so that they can have better thermal efficiency. This aspect will be further investigated in the sensitivity study.

Pressure limits have already been imposed on the system, thus the pressures in the system are generally feasible. Figure 7 and 8 can be used to design the turbine and the condenser, since the two figures give the pressure profiles when the efficiency is optimized.

4.3. Mass Flow Rates

Figure 9 shows the mass flow rate of the working fluid in the cycle for a unit exhaust mass flow rate when net work produced is optimized. Figure 10 shows the mass flow rate of the coolant, air, in the condenser for a unit exhaust mass flow rate when net work produced is optimized.

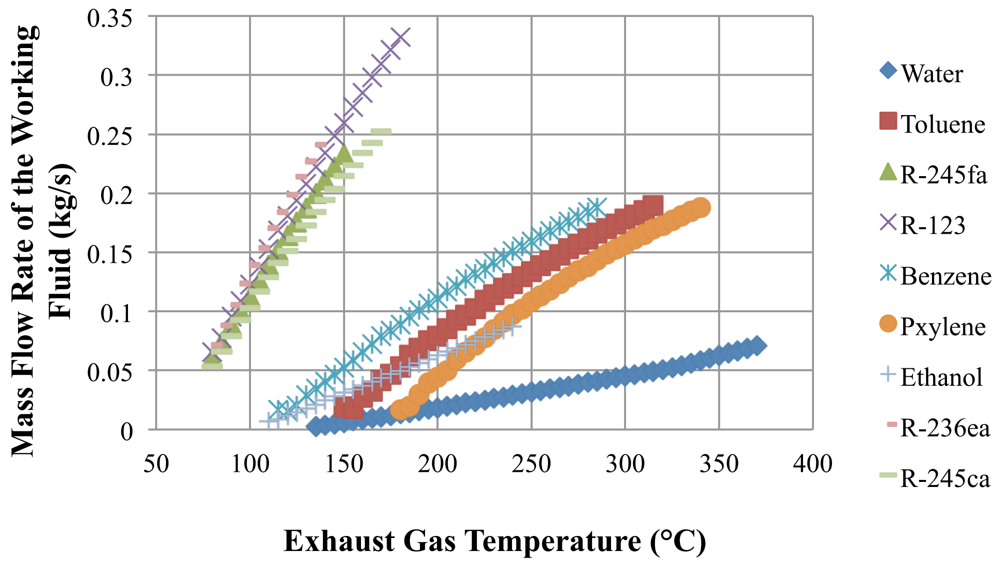

Figure 9. Mass flow rate of the working fluid when work produced is optimized.

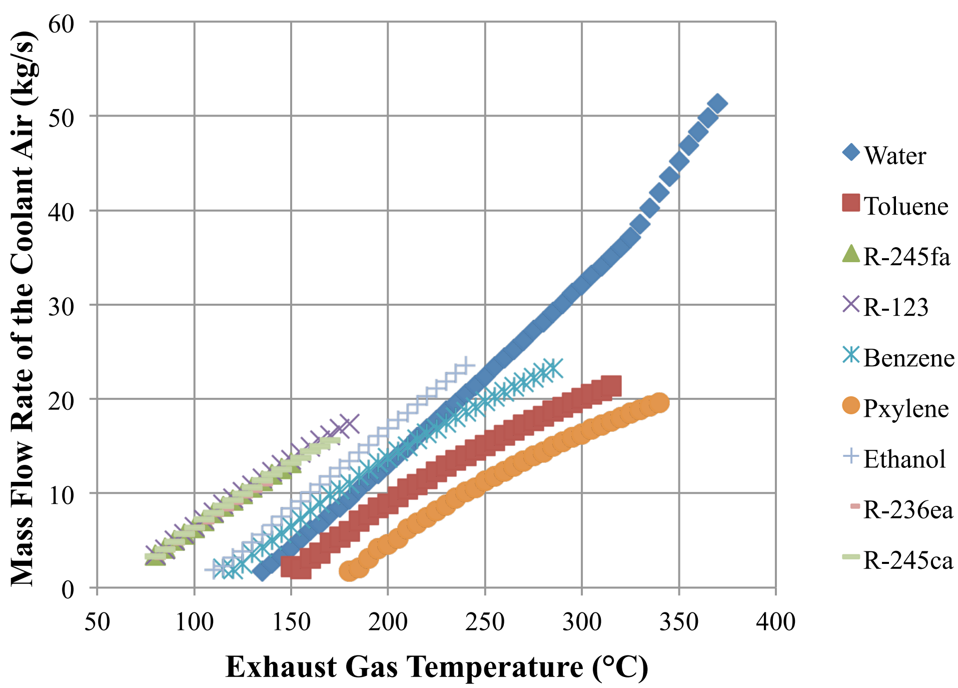

Figure 10. Mass flow rate of the coolant air when work produced is optimized.

We have assumed a unit mass flow rate of the exhaust gas (1 kg/s). From Figure 9, the mass flow rates of all working fluids are very small compared to a unit mass flow rate. This indicates that the size of the ORC is likely to be suitable for a vehicle. However, the mass flow rate of the coolant air in the condenser is large according to Figure 10. Although the condenser can be open to the environment so that the air flow is not constrained by pipes or hoses, the large mass flow rate is still worth attention.

4.4. Summary

I have examined the thermal efficiency, work produced, pressure profiles, and mass flow rates for the nine working fluid candidates in the ORC with changing exhaust gas temperature. I found that refrigerants provide high efficiency at lower temperature range (80-180 °C). R-123 showed the highest thermal efficiency and work produced within this temperature range for the particular assumptions and limitations stated in section 3.2. It is worth noticing that R-123 is being phased out due to environmental concerns discussed in section 2.2.3. Ethanol and benzene show better performance at middle temperature range (180-280 °C). Depending on the specific temperature range and the need of efficiency or work to be optimized, either ethanol or benzene can be chosen as discussed in section 4.1. However, ethanol is a slightly wet fluid, which may require the addition of a superheater in the cycle according to [3]. Water is still the best working fluid at higher temperature range (above 280 °C). This is also confirmed by literature [20], since it states that organic fluids are usually used below 300 °C heat source temperature. The use of water should require substantial superheating as common industrial practice [17].

5. SENSITIVITY STUDY

As discussed in section 3.2, the various assumptions and limitations made are subject to a sensitivity study to further understand the cycle performance. In particular, we are interested in how the changes of maximum pressure limit, minimum pressure limit, isentropic efficiencies, and condensing temperature influence the cycle performance. We have used changing exhaust temperature as the heat source temperature and constant temperature difference between the working fluid temperature and exhaust temperature in the evaporator, thus turbine inlet temperature variations can be directly observed on the plots (It is exhaust temperature minus the assumed temperature difference) and was not studied in this sensitivity study. Further, the evaporator temperature difference assumption, which reflects the heat exchanger efficiencies, was not studied because the variations of it can also be observed on the plots (e.g. 10 °C temperature difference and 200 °C exhaust temperature gives the same data as 8 °C temperature difference and 198 °C exhaust temperature).

Toluene and Pxylene are removed for clarity. The original data for some working fluids are added for comparison.

5.1. Maximum Pressure Limit

As discussed in section 4.2, pressure limits have profound influence on the thermal efficiency and the net work produced. I expect a higher maximum pressure limit and a lower minimum pressure limit can increase both the thermal efficiency and the work produced. The first sensitivity study focuses on the maximum pressure limit. The maximum pressure in the cycle, which is the turbine inlet pressure, is now limited by 2800 kPa. Only Figure 5 and Figure 6 are reproduced accordingly, because thermal efficiency and work produced are the major concerns in this project. The resulting plots are Figure 11 and Figure 12.

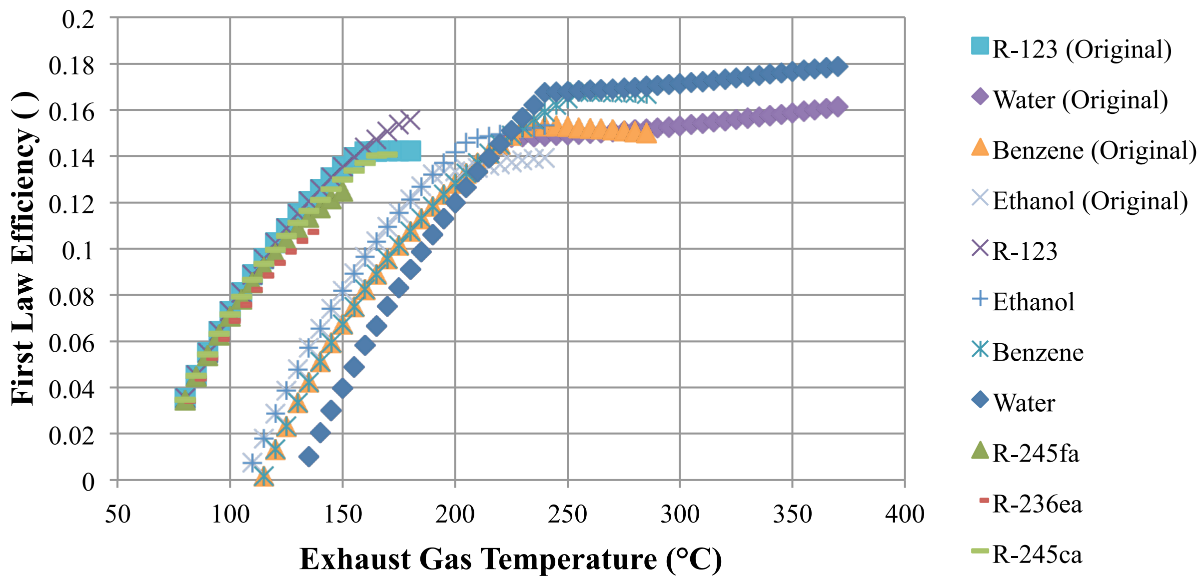

Figure 11. First law efficiency assuming maximum pressure limit is 2800 kPa.

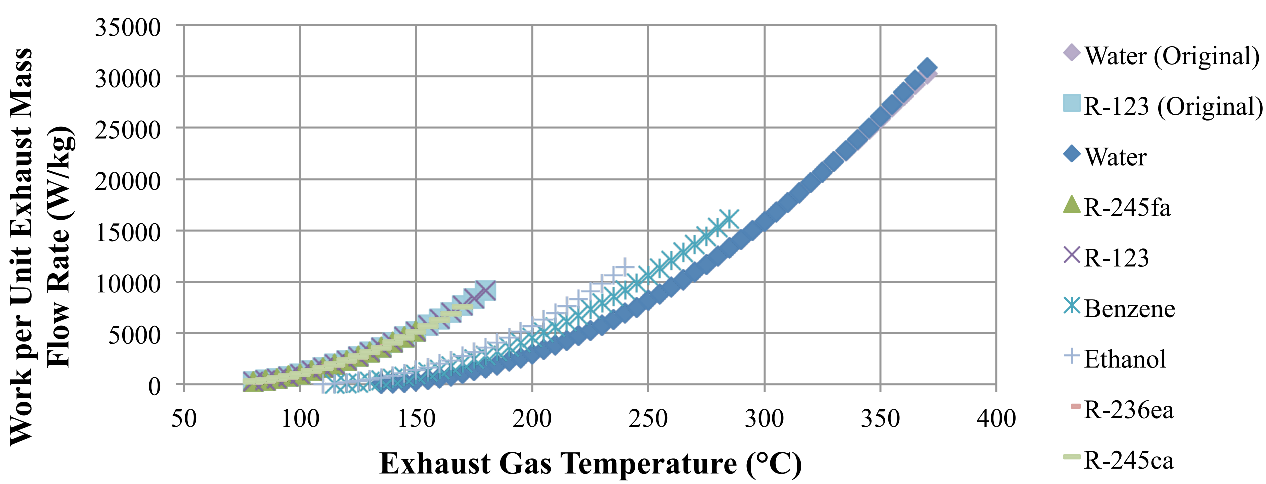

Figure 12. Work produced assuming the maximum pressure limit is 2800 kPa.

The general shapes of the curves on Figure 11 are very similar to those on Figure 5, while the maximum thermal efficiency that can be reached for each working fluid increases. The “flat” regions on the curves of R-123, R-245fa, R-245ca and R-236ea on Figure 5 disappear on Figure 11, because the efficiency is “free” to increase given the larger pressure limit. An important difference between Figure 11 and Figure 5 is that water gives a higher efficiency starting from 225 °C on Figure 11. Figure 12 is very similar to Figure 6, indicating that work is insensitive to the maximum pressure limit. The reason why only efficiency is sensitive to the maximum pressure limit was explained in section 4.2.

5.2. Minimum pressure limit

Minimum pressure (condensing pressure) limit is lowered to 170 kPa with the other assumptions unchanged. Figure 5 and Figure 6 were reproduced as Figure 13 and Figure 14. P-xylene have been removed due to some obviously erroneous data points.

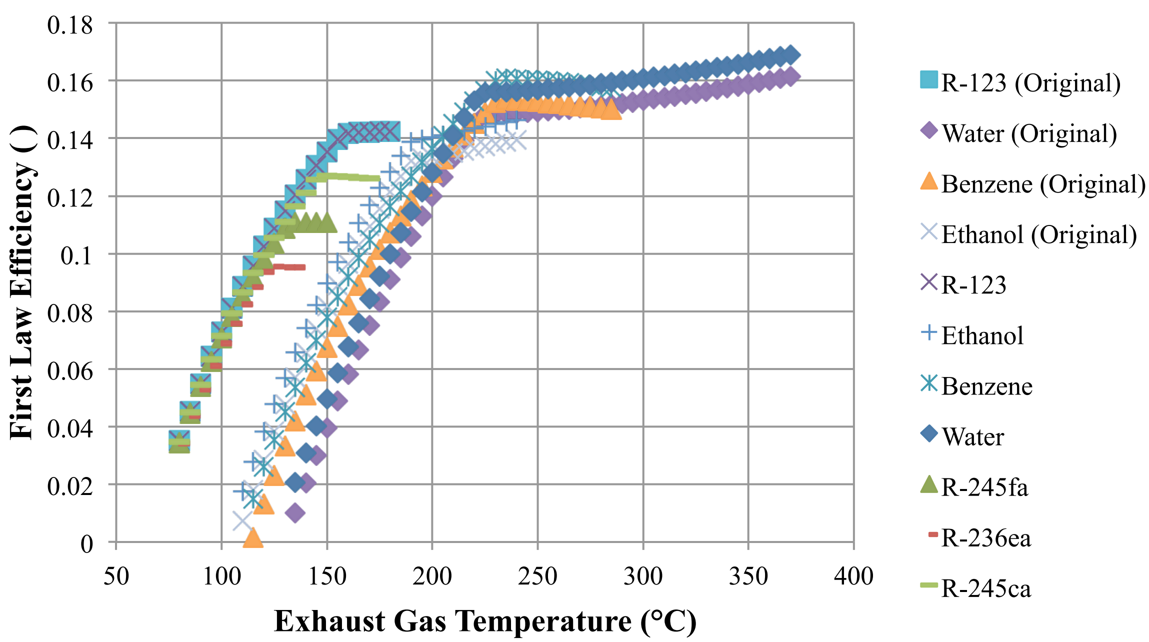

Figure 13. First law efficiency assuming minimum pressure limit is 170 kPa.

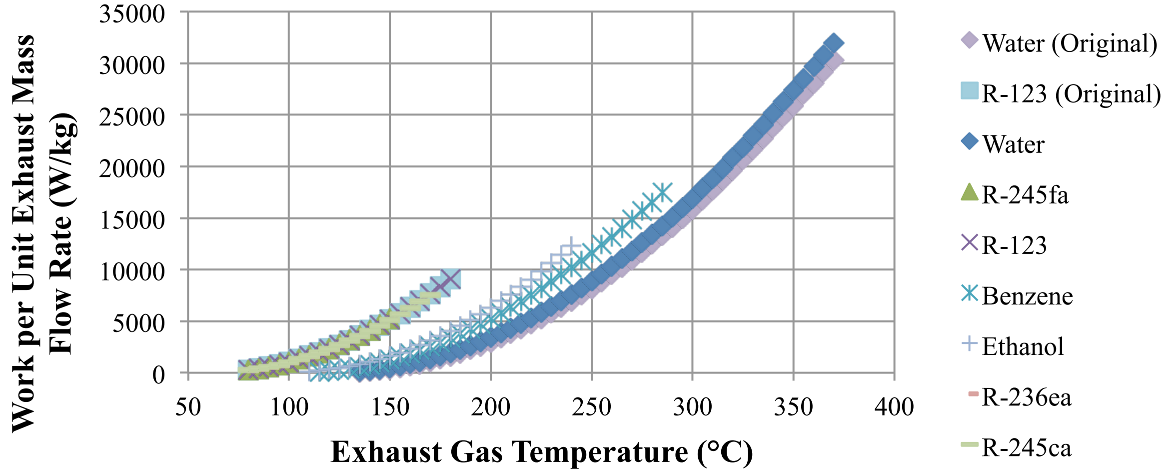

Figure 14. Work produced assuming minimum pressure limit is 170 kPa.

Figure 13 and Figure 14 are very similar to Figure 5 and Figure 6. However, by allowing a lower minimum pressure (condensing pressure) the performances of non-refrigerants increase. From Figure 8, refrigerants do not require very low minimum pressure (condensing pressure), thus their performances were unaffected. In this case, the thermal efficiency of non-refrigerants increased by about 5% (for water at 250 °C) of the original thermal efficiency. The work also increased by about 9% (for water at 250 °C) of the original work. These percentages were calculated from the internal data.

5.3. Isentropic Efficiencies

Another important assumption is the isentropic efficiency. I lowered the isentropic efficiencies of both the expander and the pump to 65% to see its influence. Figure 15 shows the new efficiency plot, and Figure 16 shows the new work plot.

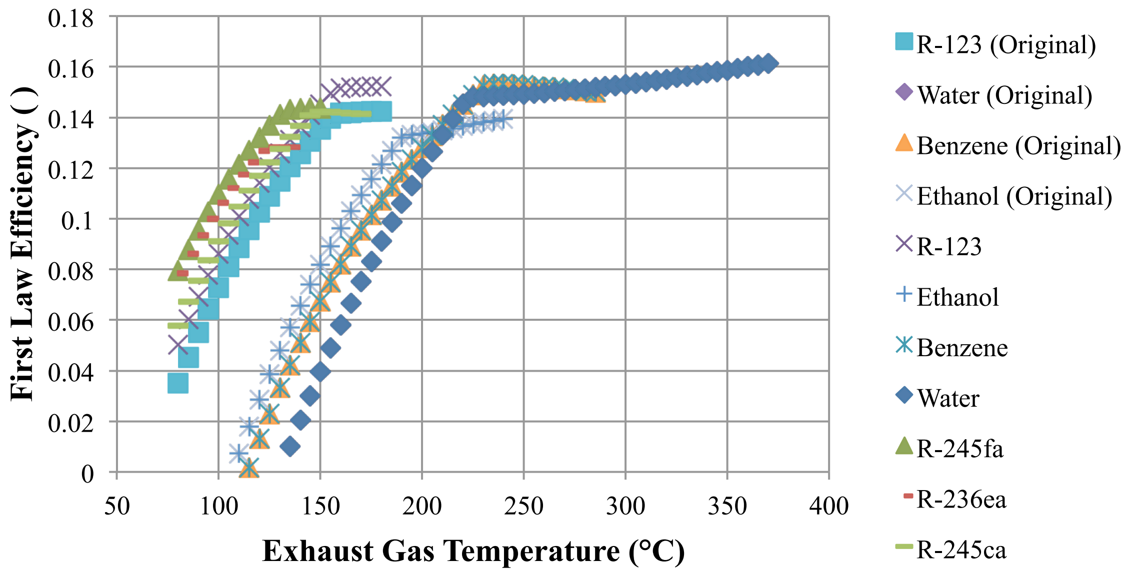

Figure 15. First law efficiency assuming the isentropic efficiencies of the turbine and the pump are 65%.

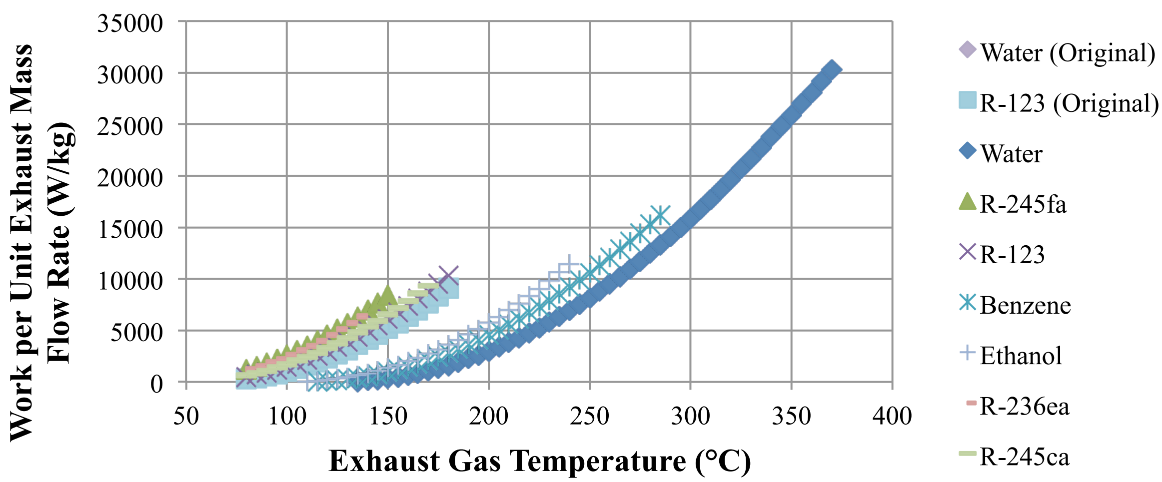

Figure 16. Work produced assuming the isentropic efficiencies of the turbine and the pump are 65%.

By comparing Figure 15 to Figure 5, I found that the first law efficiencies for refrigerants decrease by roughly 26% (at 150 °C for R-123) of the original efficiency, and for non-refrigerants by about 24% (at 250 °C for water). From Figure 16 and Figure 6, the work for refrigerants drops by about 25% (at 150 °C for R-123) of the original work, and for non-refrigerants by approximately 24% (at 250 °C for water). The drops are expected because we have introduced more irreversibilities into the cycle. The efficiencies of the refrigerants are sensitive to the isentropic efficiencies, but their work is a little less sensitive to the isentropic efficiencies. Conversely, other working fluids are equally sensitive to isentropic efficiencies in terms of work and efficiency.

5.4. Condensing Temperature

Condensing temperature is another important aspect in the cycle. I lowered the coolant air temperature to 25 °C with other assumptions unchanged to generate Figure 17 and 18.

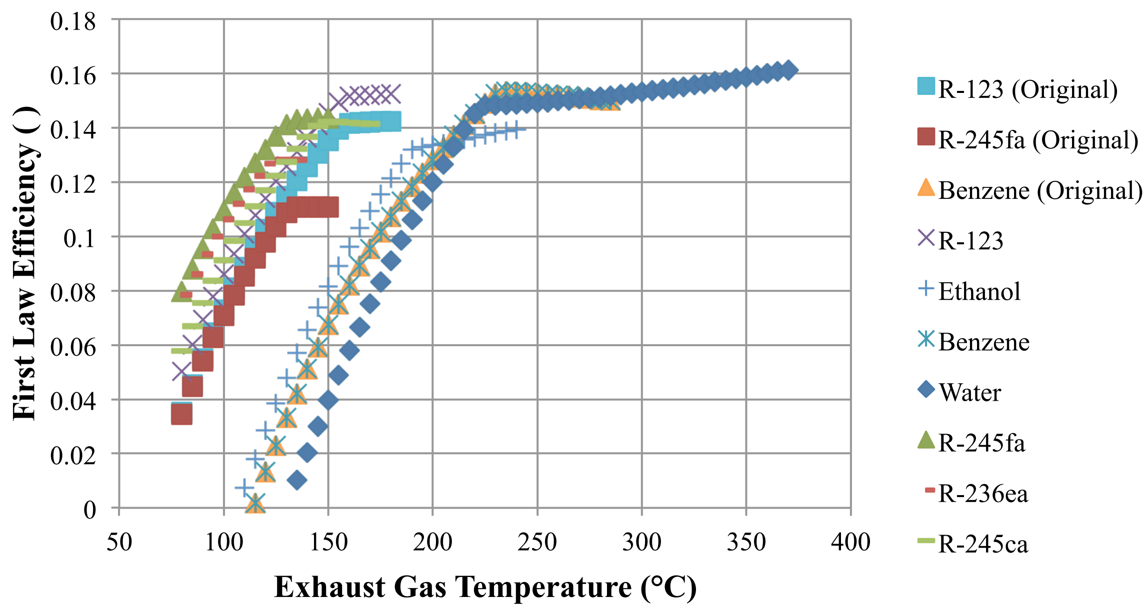

Figure 17. First law efficiency assuming the coolant air temperature is 25 °C.

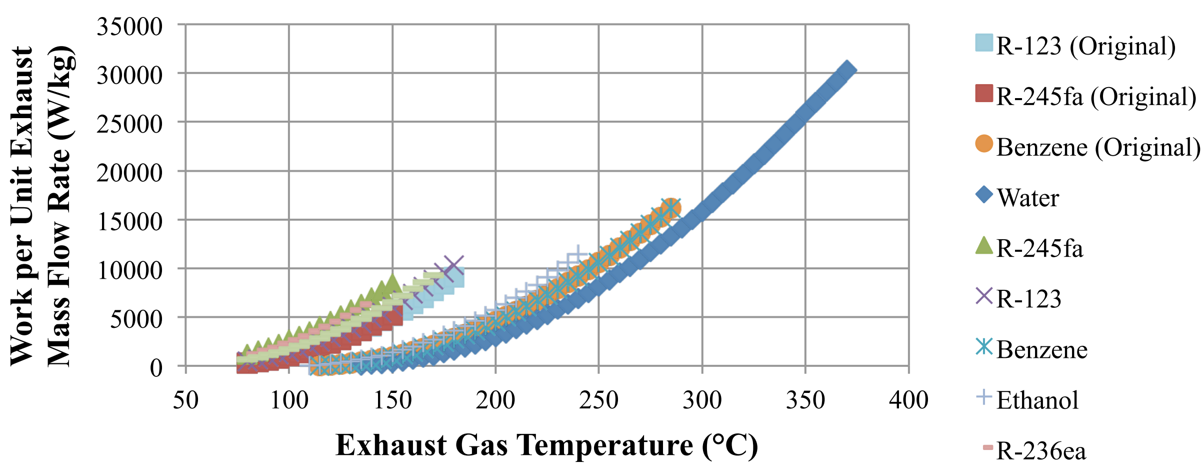

Figure 18. Work produced assuming the coolant air temperature is 25 °C.

By comparisons between Figure 17 and Figure 5, the efficiencies of refrigerants increased. Most importantly, the order of the refrigerants changed. R-245fa instead of R-123 showed the best performance over other refrigerants at exhaust temperature below 150 °C. From Figure 18 and Figure 6, refrigerants showed slightly better performance under the current assumption. R-245fa provided the largest amount of work. The condensing temperature for non-refrigerants are usually limited by the minimum pressure limit (condensing pressure), thus their performances were unchanged in this sensitivity study.

5.5. Summary

Sensitivity tests have been conducted on maximum and minimum pressure limits, isentropic efficiencies and condensing temperature. I found that the performances (i.e. efficiency and work) of the cycle are sensitive to all these conditions. For changes in maximum pressure limit, minimum pressure limit and isentropic efficiencies, the changes in performances follow our normal expectations, and the best working fluid choices are not different from those in section 4.4. However, changing condensing temperature caused the best choice of the working fluid at exhaust temperature below 150 °C to be R-245fa.

6. MODEL VALIDATION

To check the validity of the model, the T-s diagrams of R-123, benzene, ethanol and water, which are the best working fluid choices summarized in section 4.4, are plotted at key exhaust temperatures using the original assumptions. Theses T-s diagrams are then compared with the general T-s diagram of an ORC suggested by the literature. The data of the vapor dome are provided by NIST [18]. The states of each working fluid in the ORC are extracted from the internal data given by the model when efficiency is optimized. The T-s diagrams are shown in Figure 19-22.

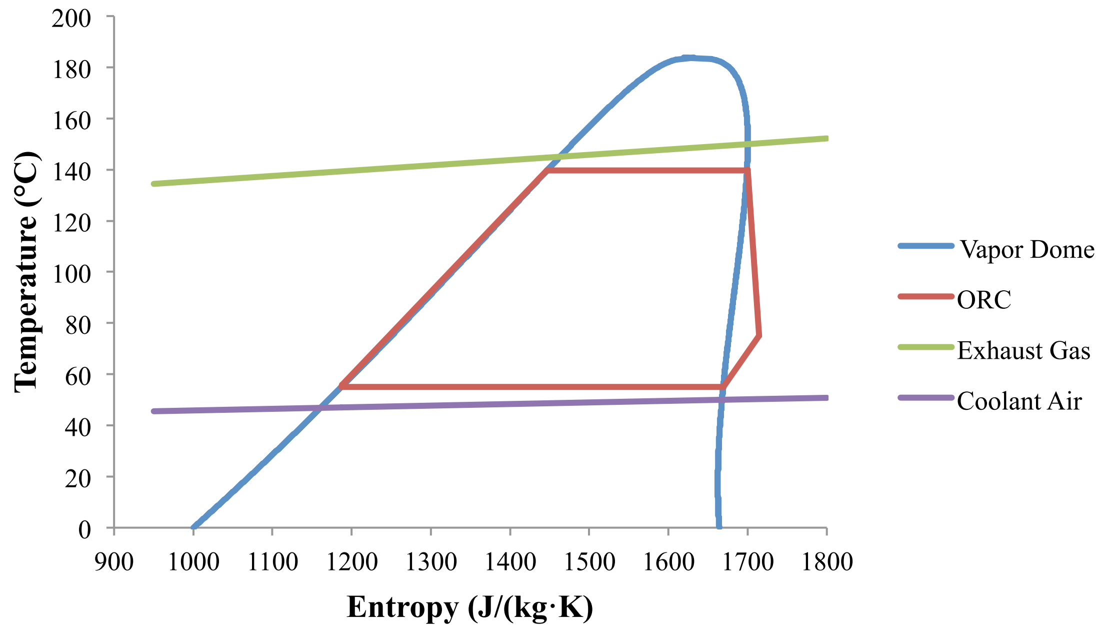

Figure 19. T-s diagram of the ORC for R-123 at an exhaust temperature of 150 °C.

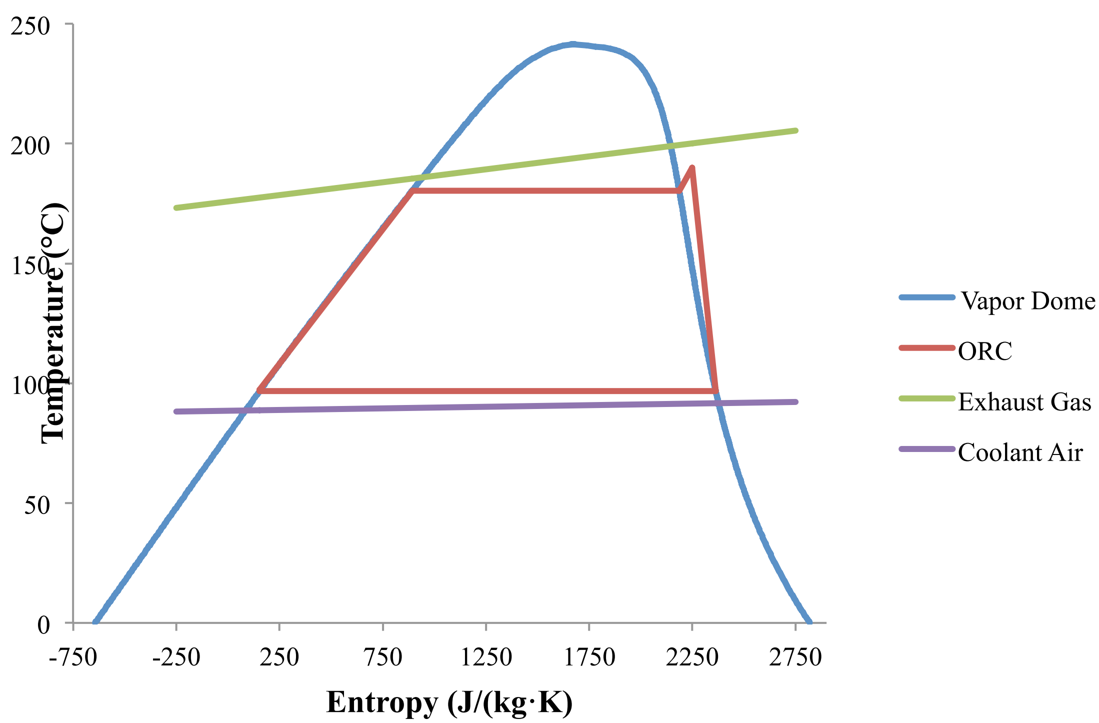

Figure 20. T-s diagram of the ORC for ethanol at an exhaust temperature of 200 °C.

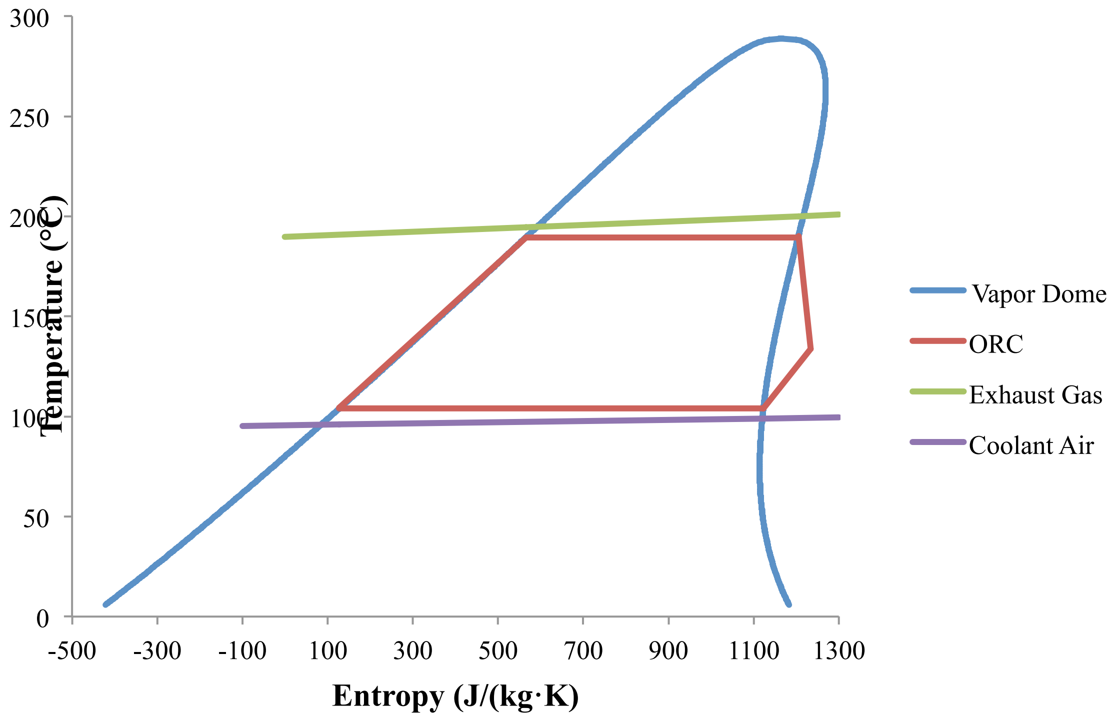

Figure 21. T-s diagram of the ORC for benzene at an exhaust temperature of 200 °C.

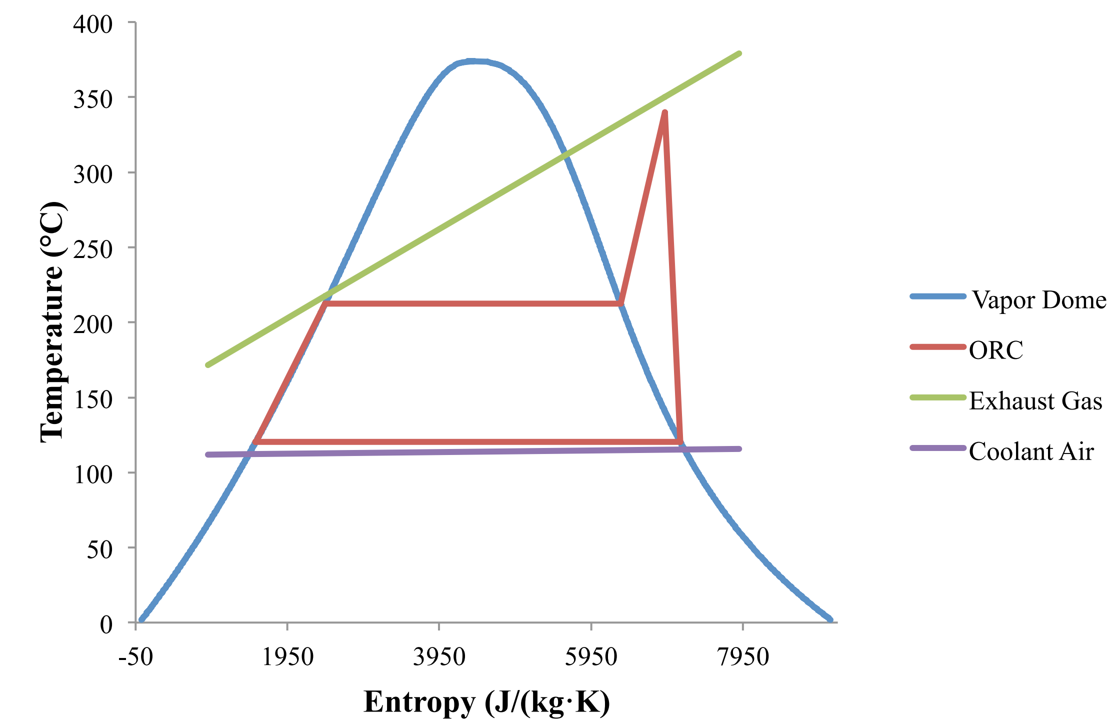

Figure 22. T-s diagram of the Rankine cycle for water at an exhaust temperature of 350 °C.

Compared to [3], Figure 19-22 have demonstrated the validity of the model because they match the literature [3]. The two dry fluids (section 2.2.2) do not require superheating in the model, which is confirmed by [3]. Ethanol is a slightly wet fluid, and water is a very wet fluid. Both of them require superheating to optimize the cycle performance, as expected in [3].

7. MIXTURE

Mixtures were briefly studied in this project. The main objective is to show the accommodation of the model to mixtures instead of selecting the best mixtures.

7.1. Introduction

Pure substances boil and condense isothermally, making the temperature profile of the working fluid giving a mismatch between the temperature profile of the heat source and that of the sink [4], as shown in Figure 19-22. This creates large irreversibilities in the cycle, reducing the cycle performance and maximum possible mass flow rate [20]. Zeotropic mixtures have the properties of boiling and condensing at changing temperature at constant pressure. These fluids thus are more suitable to be used in the ORC because they can closely match the temperature profiles of the heat source and sink to increase the cycle performance [20]. However, using mixtures can introduce more variables that need to be optimized into the cycle. Specifically, constitutions (i.e. combinations of different substances) and compositions (i.e. fractions of substances in a mixture) are the two most significant variables [4]. Researches about the applications of zeotropic mixtures in the ORC are few and usually focus on specific conditions [20].

7.2. Simulation

Due to the time limitation of this project, only one mixture was investigated through my model and compared with the performance of pure substances. The objective of this simulation is to demonstrate the accommodation of my model to mixtures rather than search for the best candidate to be used in the cycle. The mixture studied is R245fa-pentane with a mole fraction of 32% R245fa and 68% pentane (mass fraction is 47% R245fa and 53% pentane), which was selected from [20]. The constitution and composition of the mixture was fixed in this preliminary study. The results are shown in Figure 23 and 24.

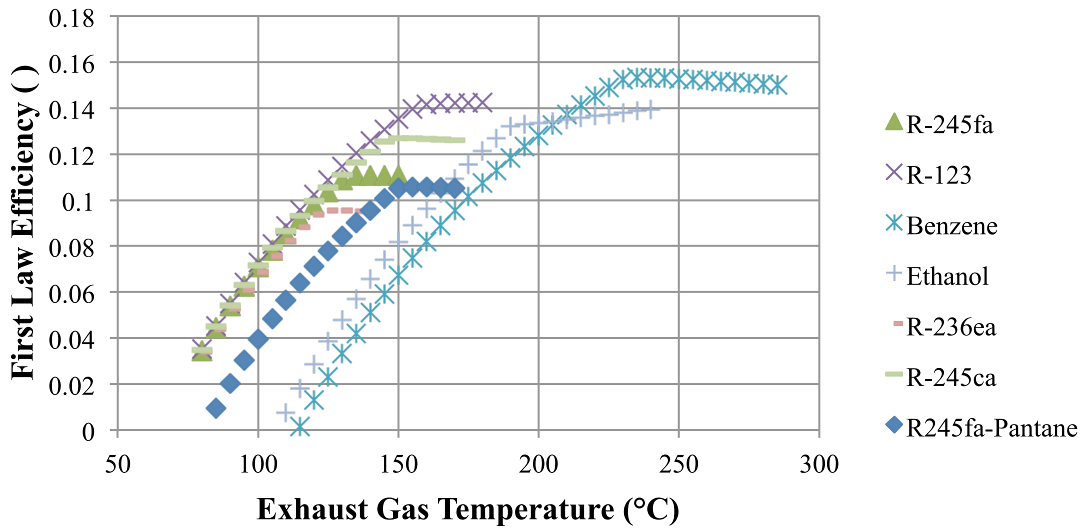

Figure 23. Optimized thermal efficiency against exhaust gas temperature with one mixture (R245fa-pentane) of fixed constitution and composition.

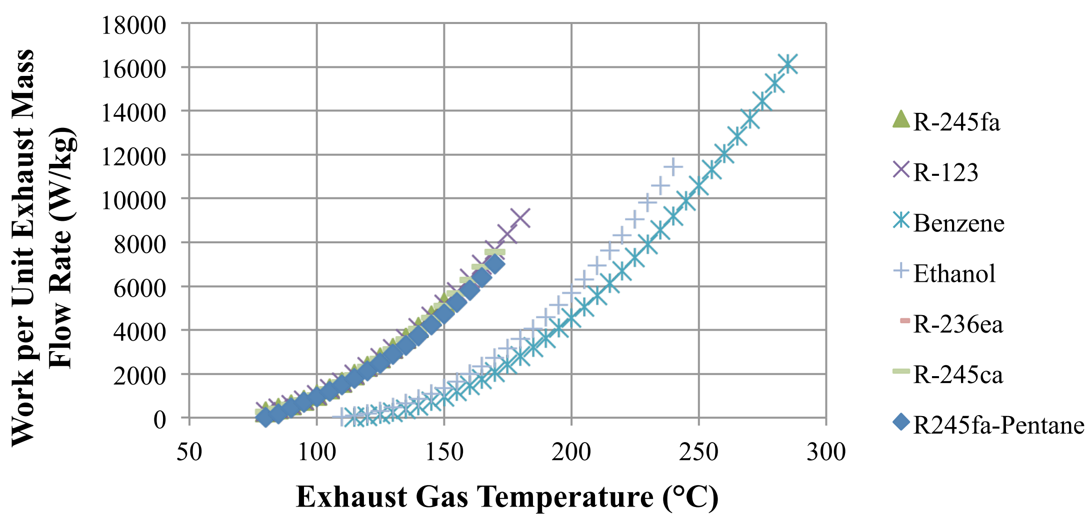

Figure 24. Optimized work per unit exhaust gas mass flow rate against exhaust gas temperature with one mixture (R245fa-pentane) of fixed constitution and composition.

From Figure 23 and 24, I found that the working range of R-245fa changed (85-170 °C instead of 80-150 °C) as pentane is added. However, the cycle performance decreased as pentane is added, which is most likely caused by three reasons. First, the assumed temperature difference between the working fluid and the exhaust gas in the evaporator may not be a reasonable assumption, since the temperature profiles should be more closely matched [20]. Second, the composition (fractions of the components in the mixture) has not been optimized for simplicity, which can reduce the cycle performance. Third, there is a possibility that adding pentane to R-

245fa is essentially a detrimental effect i.e. the constitution (behavior of the substance) is not optimized.

7.3. Validation

As performed previously for pure substances, a validation of the model through plotting T-s diagram is conducted. The exhaust temperature used in this validation is 150 °C. The diagram is shown in Figure 25.

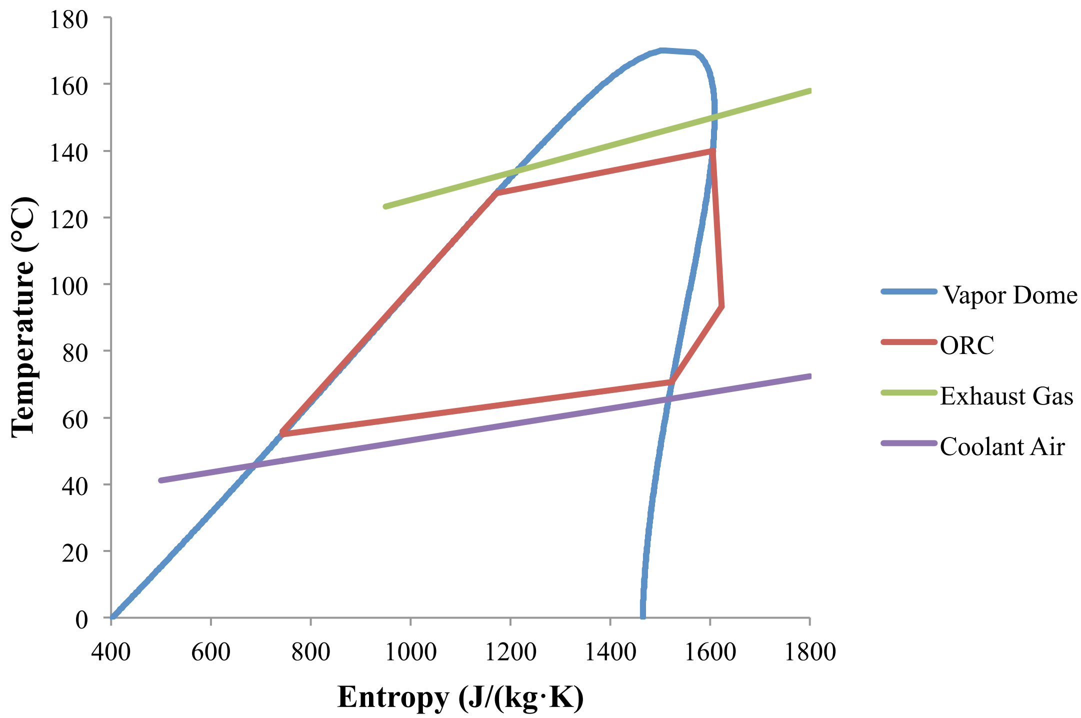

Figure 25. T-s diagram of the ORC for R245fa-pentane at an exhaust temperature of 150°C.

From Figure 25, clear temperature glides can be observed for both of the heat exchange processes as the substance changes phase, which is expected. The diagram is similar to one presented in the literature [3]. This shows that my model can potentially accommodate mixtures for future extension of this research.

7.4. Summary

The model has demonstrated its accommodation to mixtures and the potential for better performance. However, the constitutions and compositions of the mixtures need to be optimized in the model in further research.

8. CONCLUSIONS

I have conducted a literature study and developed a model for simulations for the selections of working fluids used in an ORC connected to an engine. of the substances covered in this work R-123 showed the highest thermal efficiency and work produced at the lower exhaust temperature range (80-180 °C). However, for lower condensing temperature, R-245fa can also be the best choice at this exhaust temperature range. It is worth noticing that R-123 is being phased out due to environmental concerns [3]. Ethanol and benzene show better performance at the middle temperature range (180-280 °C). Depending on the specific temperature range, either ethanol or benzene can be chosen. However, ethanol is a slightly wet fluid, which may require the addition of a superheater in the cycle according to [3]. Water is still the best working fluid at higher temperature range (above 280 °C). The use of water should require substantial superheating as common industrial practice [17].

Mixtures were briefly studied in this project. The model has demonstrated its accommodation to mixtures. However, the constitutions and compositions of the mixtures need to be optimized in the model as work left for future research.

ACKNOWLEDGEMENTS

I wish to thank Dr. Claus Borgnakke and Mr. Saili Li for their assistances on this project.

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APPDENDIX

MATLAB codes of the model described in section 3 are followings.

Code 1:

```
clear
clc
WorkingFluidArray=['water   ';'toluene ';'R245fa  ';'R123    ';'benzene
';'pxylene ';'ethanol ';'R236ea  ';'R245ca  '];
DefaultSetting=input('Default Settings (y or n): ','s');
if DefaultSetting=='n'
IsenEffTurb=input('Input the isentropic efficiency of the turbine: ');
IsenEffPump=input('Input the isentropic efficiency of the pump: ');
TempDiffEvap=input('Input the inlet temperature difference (C) in the
evaporator: ');
TempDiffCond=input('Input the outlet temperature difference (C) in the
condenser: ');
TempAirInCond=input('Input the temperature (C) of the air used for cooling in
the condenser: ');
MaxTempExGasInEvap=input('Input the maximum inlet temperature (C) of the
exhaust gas in the evaporator: ');
MinTempExGasInEvap=input('Input the minimum inlet temperature (C) of the
exhaust gas in the evaporator: ');%%%%%%%%%%%
TempInTurbIncre=input('Input the increment of the inlet temperature of the
turbine (graph resolution): ');
PresInTurbIncre=input('Input the increment of the inlet pressure of the
turbine (optimization resolution): ');%%%%%%%%%%
MaxPresAllow=input('Input the maximum pressure (kPa) allowed in the system:
');%%%%%%%
MinPresAllow=input('Input the minimum pressure (kPa) allowed in the system:
');%%%%%%%
else%!!!!!!!!!!!!!To be added in Model 4
IsenEffTurb=0.85;
IsenEffPump=0.85;
TempDiffEvap=10;
TempDiffCond=8;
TempAirInCond=47;
TempInTurbIncre=5;%Default 1
MaxTempExGasInEvap=380;
MinTempExGasInEvap=80;%%%%%%
PresInTurbIncre=2;%%%%%%%%
MaxPresAllow=2000;%%%%%
MinPresAllow=200;%%%%%%
end
MaxTempExGasInEvap=MaxTempExGasInEvap+273.15;
MinTempExGasInEvap=MinTempExGasInEvap+273.15;%%%%%%%%%%
TempAirInCond=TempAirInCond+273.15;
FirstLawEffArraySize=0;
for LoopArraySize=(MinTempExGasInEvap-
TempDiffEvap):TempInTurbIncre:(MaxTempExGasInEvap-TempDiffEvap)%%%%%%%%%%%
FirstLawEffArraySize=FirstLawEffArraySize+1;
end
WorkingFluidSize=size(WorkingFluidArray);
PlotFirstLawEff=zeros(WorkingFluidSize(1),FirstLawEffArraySize);
```

```
PlotMaxPresArray=zeros(WorkingFluidSize(1),FirstLawEffArraySize);
PlotMinPresArray=zeros(WorkingFluidSize(1),FirstLawEffArraySize);
for i=1:1:WorkingFluidSize(1)
WorkingFluid=WorkingFluidArray(i,:);
FirstLawEffArray=zeros(1,FirstLawEffArraySize);
MaxPresArray=zeros(1,FirstLawEffArraySize);
MinPresArray=zeros(1,FirstLawEffArraySize);
LoopCounterTInTurb=0;
CriticalT=refpropm('T','C',0,'',0,WorkingFluid);
for TempInTurb=(MinTempExGasInEvap-
TempDiffEvap):TempInTurbIncre:(MaxTempExGasInEvap-TempDiffEvap)%%%%%%%%%%%%%
LoopCounterTInTurb=LoopCounterTInTurb+1;
TempOutCond=TempAirInCond+TempDiffCond;%%%%%%%%%%
PresOutCond=refpropm('P','T',TempOutCond,'Q',0,WorkingFluid);%%%%%%%%%%%%%
%if (TempInTurb<CriticalT)
%MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,WorkingFluid);%%%%%%%%%%%%
%else
%MaxPresInTurb=refpropm('P','C',0,'',0,WorkingFluid);%%%%%%%%%%%
%end
%if (MaxPresInTurb>MaxPresAllow)%%%%%%
%MaxPresInTurb=MaxPresAllow;%%%%%%%%
%end
if (PresOutCond<MinPresAllow)%%%%%%%%
PresOutCond=MinPresAllow;%%%%%%%%%
TempOutCond=refpropm('T','P',PresOutCond,'Q',0,WorkingFluid);
end
MaxPres=0;%%%%%%%
MinPres=0;%%%%%%
MaxFirstLawEff=0;%%%%%%%%
if (TempInTurb<CriticalT-10)%%%%%%%%%
MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,WorkingFluid);
if (MaxPresInTurb>MaxPresAllow)%%%%%%
MaxPresInTurb=MaxPresAllow;%%%%%%%%
end
for PresInTurb=PresOutCond:PresInTurbIncre:MaxPresInTurb%%%%%%%%%%%
[HInTurb,SInTurb]=refpropm('HS','T',TempInTurb,'P',PresInTurb,WorkingFluid);%
%%%%%%%%
IsenSInCond=SInTurb;
PresInCond=PresOutCond;
IsenHInCond=refpropm('H','P',PresInCond,'S',IsenSInCond,WorkingFluid);
HInCond=HInTurb-IsenEffTurb*(HInTurb-IsenHInCond);
TempInCond=refpropm('T','P',PresInCond,'H',HInCond,WorkingFluid);
TempInPump=TempOutCond;
PresInPump=PresOutCond;
[HInPump,SInPump]=refpropm('HS','T',TempInPump,'P',PresInPump,WorkingFluid);
IsenSOutPump=SInPump;
IsenHOutPump=refpropm('H','P',PresInTurb,'S',IsenSOutPump,WorkingFluid);
HOutPump=(IsenHOutPump-HInPump)/IsenEffPump+HInPump;
TempOutPump=refpropm('T','P',PresInTurb,'H',HOutPump,WorkingFluid);
FirstLawEff=((HInTurb-HInCond)-(HOutPump-HInPump))/(HInTurb-HOutPump);
if MaxFirstLawEff<FirstLawEff%%%%%%%%%%%
MaxFirstLawEff=FirstLawEff;
MaxPres=PresInTurb;
MinPres=PresInPump;
end
end
end
```

```
FirstLawEffArray(LoopCounterTInTurb)=MaxFirstLawEff;
MaxPresArray(LoopCounterTInTurb)=MaxPres;
MinPresArray(LoopCounterTInTurb)=MinPres;
end
PlotFirstLawEff(i,:)=FirstLawEffArray;
PlotMaxPresArray(i,:)=MaxPresArray;
PlotMinPresArray(i,:)=MinPresArray;
end
PlotExGasTemp=MinTempExGasInEvap-273.15:TempInTurbIncre:MaxTempExGasInEvap-
273.15;%%%%%%%%%%
figure
for j=1:1:WorkingFluidSize(1)
plot(PlotExGasTemp,PlotFirstLawEff(j,:),'.')
hold on
end
legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
xlabel('Exhaust Gas Temperature (C)')
ylabel('First Law Efficiency')
title('ORC Efficiency for Different Working Fluids')
figure
for j=1:1:WorkingFluidSize(1)
plot(PlotExGasTemp,PlotMaxPresArray(j,:),'.')
hold on
end
legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
xlabel('Exhaust Gas Temperature (C)')
ylabel('Max Pressure in System (kPa)')
figure
for j=1:1:WorkingFluidSize(1)
plot(PlotExGasTemp,PlotMinPresArray(j,:),'.')
hold on
end
legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
xlabel('Exhaust Gas Temperature (C)')
ylabel('Min Pressure in System (kPa)')
```

Code 2:

```
clear
clc
WorkingFluidArray=['water  ';'toluene';'R245fa ';'R123
';'benzene';'pxylene';'ethanol';'R236ea ';'R245ca '];
DefaultSetting=input('Default Settings (y or n): ','s');
if DefaultSetting=='n'
IsenEffTurb=input('Input the isentropic efficiency of the turbine: ');
IsenEffPump=input('Input the isentropic efficiency of the pump: ');
TempDiffEvap=input('Input the inlet temperature difference (C) in the
evaporator: ');
TempDiffEvapPinch=input('Input the temperature difference (C) at the pinch
point in the evaporator: ');
TempDiffCondAirIn=input('Input the outlet temperature difference (C) in the
condenser: ');
```

```
TempAirInCond=input('Input the temperature (C) of the air used for cooling in
the condenser: ');
TempExGasInEvap=input('Input the inlet temperature (C) of the exhaust gas in
the evaporator: ');
PresExGasEvap=input('Input the pressure (kPa) of the exhaust gas in the
evaporator: ');
MFlowExGas=input('Input the mass flow rate (kg/s) of the exhaust gas: ');
TempInTurbIncre=input('Input the increment of the inlet temperature of the
turbine: ');
MaxTempExGasInEvap=input('Input the maximum temperature (C) of exhaust gas:
');
MinTempExGasInEvap=input('Input the minimum temperature (C) of exhaust gas:
');
TempExGasInEvapIncre=input('Input the temperature increment of exhaust gas:
');
PresInTurbIncre=input('Input the increment of the inlet pressure of the
turbine (optimization resolution): ');%%%%%%%%%%
MaxPresAllow=input('Input the maximum pressure (kPa) allowed in the system:
');%%%%%%%
MinPresAllow=input('Input the minimum pressure (kPa) allowed in the system:
');%%%%%%%
else
IsenEffTurb=0.85;
IsenEffPump=0.85;
TempDiffEvap=10;
TempDiffEvapPinch=5;
TempDiffCondAirIn=8;
TempDiffCondAirOut=5;
TempAirInCond=47;
PresAirCond=101.325;
TempInTurbIncre=5;%Default 1
PresExGasEvap=1.2*101.325;%Assume 1.2Patm
MFlowExGas=1;%Assume unit mass flow rate
MaxTempExGasInEvap=380;
MinTempExGasInEvap=80;
PresInTurbIncre=10;%%%%%%%%
MaxPresAllow=2000;%%%%%
MinPresAllow=200;%%%%%%
end
MinTempExGasInEvap=MinTempExGasInEvap+273.15;
MaxTempExGasInEvap=MaxTempExGasInEvap+273.15;
TempAirInCond=TempAirInCond+273.15;
MaxWorkArraySize=0;
for LoopArraySize=MinTempExGasInEvap:TempInTurbIncre:MaxTempExGasInEvap
    MaxWorkArraySize=MaxWorkArraySize+1;
end
WorkingFluidSize=size(WorkingFluidArray);
PlotMaxWork=zeros(WorkingFluidSize(1),MaxWorkArraySize);
PlotMFlowatMaxW=zeros(WorkingFluidSize(1),MaxWorkArraySize);
PlotVFlowatMaxW=zeros(WorkingFluidSize(1),MaxWorkArraySize);
PlotTInTurbatMaxW=zeros(WorkingFluidSize(1),MaxWorkArraySize);
PlotCondAirMFlowatMaxW=zeros(WorkingFluidSize(1),MaxWorkArraySize);
for i=1:1:WorkingFluidSize(1)
WorkingFluid=WorkingFluidArray(i,:);
MaxWorkArray=zeros(1,MaxWorkArraySize);
MFlowatMaxW=zeros(1,MaxWorkArraySize);
VFlowatMaxW=zeros(1,MaxWorkArraySize);
```

```
AirCondMFlowatMaxW=zeros(1,MaxWorkArraySize);
LoopCounter=0;
CriticalT=refpropm('T','C',0,'',0,WorkingFluid);
for TempInTurb=(MinTempExGasInEvap-
TempDiffEvap):TempInTurbIncre:(MaxTempExGasInEvap-TempDiffEvap)%%%%%%%%%%%%%%
TempOutCond=TempAirInCond+TempDiffCondAirIn;%%%%%%%%%%
PresOutCond=refpropm('P','T',TempOutCond,'Q',0,WorkingFluid);%%%%%%%%%%%%%
%if (TempInTurb<CriticalT)
%MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,WorkingFluid);%%%%%%%%%%%%
%else
%MaxPresInTurb=refpropm('P','C',0,'',0,WorkingFluid);%%%%%%%%%%%
%end
%if (MaxPresInTurb>MaxPresAllow)%%%%%%
%MaxPresInTurb=MaxPresAllow;%%%%%%%%
%end
if (PresOutCond<MinPresAllow)%%%%%%%%
PresOutCond=MinPresAllow;%%%%%%%%%
TempOutCond=refpropm('T','P',PresOutCond,'Q',0,WorkingFluid);
end
MFlowRateAtMaxW=0;%%%%%%%
VFlowRateAtMaxW=0;%%%%%%
AirCondMFlowRateAtMaxW=0;
MaxWork=0;%%%%%%%%
if (TempInTurb<CriticalT-10)%%%%%%%%%
MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,WorkingFluid);
if (MaxPresInTurb>MaxPresAllow)%%%%%%
MaxPresInTurb=MaxPresAllow;%%%%%%%%
end
for PresInTurb=PresOutCond:PresInTurbIncre:MaxPresInTurb%%%%%%%%%%%
[DInTurb,HInTurb,SInTurb]=refpropm('DHS','T',TempInTurb,'P',PresInTurb,Workin
gFluid);%%%%%%%%%
IsenSInCond=SInTurb;
PresInCond=PresOutCond;
IsenHInCond=refpropm('H','P',PresInCond,'S',IsenSInCond,WorkingFluid);
HInCond=HInTurb-IsenEffTurb*(HInTurb-IsenHInCond);
[DInCond,TempInCond]=refpropm('DT','P',PresInCond,'H',HInCond,WorkingFluid);
TempInPump=TempOutCond;
PresInPump=PresOutCond;
[HInPump,SInPump]=refpropm('HS','T',TempInPump,'P',PresInPump,WorkingFluid);
IsenSOutPump=SInPump;
IsenHOutPump=refpropm('H','P',PresInTurb,'S',IsenSOutPump,WorkingFluid);
HOutPump=(IsenHOutPump-HInPump)/IsenEffPump+HInPump;
TempOutPump=refpropm('T','P',PresInTurb,'H',HOutPump,WorkingFluid);
[TempEvapPinch,HOutPumpPinch]=refpropm('TH','P',PresInTurb,'Q',0,WorkingFluid
);
TempExGasOutEvapPinch=TempEvapPinch+TempDiffEvapPinch;
HExGasInEvap=refpropm('H','T',TempInTurb+TempDiffEvap,'P',PresExGasEvap,'AIR.
ppf');%Assume exhaust gas is air
HExGasOutEvapPinch=refpropm('H','T',TempExGasOutEvapPinch,'P',PresExGasEvap,'
AIR.ppf');%Assume exhaust gas is air
%if (DInTurb<DInCond)
%MinD=DInTurb;
%
%MinD=DInCond;
%end
MFlowRate=(HExGasInEvap-HExGasOutEvapPinch)*MFlowExGas/(HInTurb-
HOutPumpPinch);
```

```
Work=MFlowRate*((HInTurb-HInCond)-(HOutPump-HInPump));

TempAirOutCond=TempOutCond-TempDiffCondAirOut;
HAirInCond=refpropm('H','T',TempInPump-
TempDiffCondAirIn,'P',PresAirCond,'AIR.ppf');
HAirPinchCond=refpropm('H','T',TempAirOutCond,'P', PresAirCond,'AIR.ppf');
if (TempOutCond<TempInCond)%superheated at the condenser inlet
HCondInPinch=refpropm('H','T',TempOutCond,'Q',1,WorkingFluid);
else
HCondInPinch=HInCond;
end
CondAirMFlow=(HCondInPinch-HInPump)*MFlowRate/(HAirPinchCond-HAirInCond);

if (Work>MaxWork)
MaxWork=Work;
MFlowRateAtMaxW=MFlowRate;
%VFlowRateAtMaxW=MFlowRate/MinD;
AirCondMFlowRateAtMaxW=CondAirMFlow;
end
end
end
LoopCounter=LoopCounter+1;
MaxWorkArray(LoopCounter)=MaxWork;
MFlowatMaxW(LoopCounter)=MFlowRateAtMaxW;
%VFlowatMaxW(LoopCounter)=VFlowRateAtMaxW;
AirCondMFlowatMaxW(LoopCounter)=AirCondMFlowRateAtMaxW;
end
PlotMaxWork(i,:)=MaxWorkArray;
PlotMFlowatMaxW(i,:)=MFlowatMaxW;
%PlotVFlowatMaxW(i,:)=VFlowatMaxW;
PlotCondAirMFlowatMaxW(i,:)=AirCondMFlowatMaxW;
end
PlotTempExGasInEvap=MinTempExGasInEvap-
273.15:TempInTurbIncre:MaxTempExGasInEvap-273.15;
figure
for j=1:1:WorkingFluidSize(1)
plot(PlotTempExGasInEvap,PlotMaxWork(j,:),'.')
hold on
end
legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
xlabel('Exhaust Gas Temperature (C)')
ylabel('Maximum Work (J)')
title('Maximum Work versus Exhaust Gas Temperature for Different Working
Fluids')
figure
for j=1:1:WorkingFluidSize(1)
plot(PlotTempExGasInEvap,PlotMFlowatMaxW(j,:),'.')
hold on
end
legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
xlabel('Exhaust Gas Temperature (C)')
ylabel('Mass Flow Rate at Maximum Work (kg/s)')
title('Mass Flow Rate at Maximum Work versus Exhaust Gas Temperature for
Different Working Fluids')
```

```
%figure
%for j=1:1:WorkingFluidSize(1)
%plot(PlotTempExGasInEvap,PlotVFlowatMaxW(j,:),'.')
%hold on
%end
%legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
%xlabel('Exhaust Gas Temperature (C)')
%ylabel('Maximum Volumetric Flow Rate at Maximum Work (m^3/s)')
%title('Maximum Volumetric Flow Rate at Maximum Work versus Exhaust Gas
Temperature for Different Working Fluids')
figure
for j=1:1:WorkingFluidSize(1)
plot(PlotTempExGasInEvap,PlotCondAirMFlowatMaxW(j,:),'.')
hold on
end
legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
xlabel('Exhaust Gas Temperature (C)')
ylabel('Condenser Air Mass Flow Rate at Maximum Work (kg/s)')
title('Condenser Air Mass Flow Rate at Maximum Work versus Exhaust Gas
Temperature for Different Working Fluids')
```

MATLAB codes of the model described in section 7.2 are followings.

Code 1:

```
clear
clc
WorkingFluidArray=['water   ';'toluene ';'R245fa  ';'R123    ';'benzene 
';'pxylene ';'ethanol ';'R236ea  ';'R245ca  '];
DefaultSetting=input('Default Settings (y or n): ','s');
if DefaultSetting=='n'
IsenEffTurb=input('Input the isentropic efficiency of the turbine: ');
IsenEffPump=input('Input the isentropic efficiency of the pump: ');
TempDiffEvap=input('Input the inlet temperature difference (C) in the 
evaporator: ');
TempDiffCond=input('Input the outlet temperature difference (C) in the 
condenser: ');
TempAirInCond=input('Input the temperature (C) of the air used for cooling in 
the condenser: ');
MaxTempExGasInEvap=input('Input the maximum inlet temperature (C) of the 
exhaust gas in the evaporator: ');
MinTempExGasInEvap=input('Input the minimum inlet temperature (C) of the 
exhaust gas in the evaporator: ');%%%%%%%%%%%
TempInTurbIncre=input('Input the increment of the inlet temperature of the 
turbine (graph resolution): ');
PresInTurbIncre=input('Input the increment of the inlet pressure of the 
turbine (optimization resolution): ');%%%%%%%%%%
MaxPresAllow=input('Input the maximum pressure (kPa) allowed in the system: 
');%%%%%%%
MinPresAllow=input('Input the minimum pressure (kPa) allowed in the system: 
');%%%%%%%
else%!!!!!!!!!!!!!To be added in Model 4
IsenEffTurb=0.85;
IsenEffPump=0.85;
TempDiffEvap=10;
TempDiffCond=8;
TempAirInCond=47;
TempInTurbIncre=5;%Default 1
MaxTempExGasInEvap=200;
MinTempExGasInEvap=80;%%%%%%
PresInTurbIncre=2;%%%%%%%%
MaxPresAllow=2000;%%%%%
MinPresAllow=200;%%%%%%
end
MaxTempExGasInEvap=MaxTempExGasInEvap+273.15;
MinTempExGasInEvap=MinTempExGasInEvap+273.15;%%%%%%%%%%
TempAirInCond=TempAirInCond+273.15;
FirstLawEffArraySize=0;
for LoopArraySize=(MinTempExGasInEvap-
TempDiffEvap):TempInTurbIncre:(MaxTempExGasInEvap-TempDiffEvap)%%%%%%%%%%%
FirstLawEffArraySize=FirstLawEffArraySize+1;
end
FirstLawEffArray=zeros(1,FirstLawEffArraySize);
MaxPresArray=zeros(1,FirstLawEffArraySize);
MinPresArray=zeros(1,FirstLawEffArraySize);
LoopCounterTInTurb=0;
CriticalT=refpropm('T','C',0,'',0,'R245fa','pentane',[0.47 0.53]);
```

```
for TempInTurb=(MinTempExGasInEvap-
TempDiffEvap):TempInTurbIncre:(MaxTempExGasInEvap-TempDiffEvap)%%%%%%%%%%%%%%
LoopCounterTInTurb=LoopCounterTInTurb+1;
TempOutCond=TempAirInCond+TempDiffCond;%%%%%%%%%%
PresOutCond=refpropm('P','T',TempOutCond,'Q',0,'R245fa','pentane',[0.47
0.53]);%%%%%%%%%%%%%
%if (TempInTurb<CriticalT)
%MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,WorkingFluid);%%%%%%%%%%%%
%else
%MaxPresInTurb=refpropm('P','C',0,'',0,WorkingFluid);%%%%%%%%%%%
%end
%if (MaxPresInTurb>MaxPresAllow)%%%%%%
%MaxPresInTurb=MaxPresAllow;%%%%%%%%
%end
if (PresOutCond<MinPresAllow)%%%%%%%%
PresOutCond=MinPresAllow;%%%%%%%%%
TempOutCond=refpropm('T','P',PresOutCond,'Q',0,'R245fa','pentane',[0.47
0.53]);
end
MaxPres=0;%%%%%%%
MinPres=0;%%%%%%
MaxFirstLawEff=0;%%%%%%%%
if (TempInTurb<CriticalT-10)%%%%%%%%%
MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,'R245fa','pentane',[0.47
0.53]);
if (MaxPresInTurb>MaxPresAllow)%%%%%%
MaxPresInTurb=MaxPresAllow;%%%%%%%%
end
for PresInTurb=PresOutCond:PresInTurbIncre:MaxPresInTurb%%%%%%%%%%%
[HInTurb,SInTurb]=refpropm('HS','T',TempInTurb,'P',PresInTurb,'R245fa','penta
ne',[0.47 0.53]);%%%%%%%%%
IsenSInCond=SInTurb;
PresInCond=PresOutCond;
IsenHInCond=refpropm('H','P',PresInCond,'S',IsenSInCond,'R245fa','pentane',[0
.47 0.53]);
HInCond=HInTurb-IsenEffTurb*(HInTurb-IsenHInCond);
TempInCond=refpropm('T','P',PresInCond,'H',HInCond,'R245fa','pentane',[0.47
0.53]);
TempInPump=TempOutCond;
PresInPump=PresOutCond;
[HInPump,SInPump]=refpropm('HS','T',TempInPump,'P',PresInPump,'R245fa','penta
ne',[0.47 0.53]);
IsenSOutPump=SInPump;
IsenHOutPump=refpropm('H','P',PresInTurb,'S',IsenSOutPump,'R245fa','pentane',
[0.47 0.53]);
HOutPump=(IsenHOutPump-HInPump)/IsenEffPump+HInPump;
TempOutPump=refpropm('T','P',PresInTurb,'H',HOutPump,'R245fa','pentane',[0.47
0.53]);
FirstLawEff=((HInTurb-HInCond)-(HOutPump-HInPump))/(HInTurb-HOutPump);
if MaxFirstLawEff<FirstLawEff%%%%%%%%%%%
MaxFirstLawEff=FirstLawEff;
MaxPres=PresInTurb;
MinPres=PresInPump;
end
end
end
FirstLawEffArray(LoopCounterTInTurb)=MaxFirstLawEff;
```

```
MaxPresArray(LoopCounterTInTurb)=MaxPres;
MinPresArray(LoopCounterTInTurb)=MinPres;
end
PlotExGasTemp=MinTempExGasInEvap-273.15:TempInTurbIncre:MaxTempExGasInEvap-
273.15;%%%%%%%%%%
figure
plot(PlotExGasTemp,FirstLawEffArray,'.')
hold on
xlabel('Exhaust Gas Temperature (C)')
ylabel('First Law Efficiency')
title('ORC Efficiency for Different Working Fluids')
figure
plot(PlotExGasTemp,MaxPresArray,'.')
hold on
xlabel('Exhaust Gas Temperature (C)')
ylabel('Max Pressure in System (kPa)')
figure
plot(PlotExGasTemp,MinPresArray,'.')
hold on
xlabel('Exhaust Gas Temperature (C)')
ylabel('Min Pressure in System (kPa)')
```

Code 2:

```
clear
clc
WorkingFluidArray=['water  ';'toluene';'R245fa ';'R123
';'benzene';'pxylene';'ethanol';'R236ea ';'R245ca '];
DefaultSetting=input('Default Settings (y or n): ','s');
if DefaultSetting=='n'
IsenEffTurb=input('Input the isentropic efficiency of the turbine: ');
IsenEffPump=input('Input the isentropic efficiency of the pump: ');
TempDiffEvap=input('Input the inlet temperature difference (C) in the
evaporator: ');
TempDiffEvapPinch=input('Input the temperature difference (C) at the pinch
point in the evaporator: ');
TempDiffCondAirIn=input('Input the outlet temperature difference (C) in the
condenser: ');
TempAirInCond=input('Input the temperature (C) of the air used for cooling in
the condenser: ');
TempExGasInEvap=input('Input the inlet temperature (C) of the exhaust gas in
the evaporator: ');
PresExGasEvap=input('Input the pressure (kPa) of the exhaust gas in the
evaporator: ');
MFlowExGas=input('Input the mass flow rate (kg/s) of the exhaust gas: ');
TempInTurbIncre=input('Input the increment of the inlet temperature of the
turbine: ');
MaxTempExGasInEvap=input('Input the maximum temperature (C) of exhaust gas:
');
MinTempExGasInEvap=input('Input the minimum temperature (C) of exhaust gas:
');
TempExGasInEvapIncre=input('Input the temperature increment of exhaust gas:
');
PresInTurbIncre=input('Input the increment of the inlet pressure of the
turbine (optimization resolution): ');%%%%%%%%%%
```

```
MaxPresAllow=input('Input the maximum pressure (kPa) allowed in the system:
');%%%%%%%
MinPresAllow=input('Input the minimum pressure (kPa) allowed in the system:
');%%%%%%%
else
IsenEffTurb=0.85;
IsenEffPump=0.85;
TempDiffEvap=10;
TempDiffEvapPinch=5;
TempDiffCondAirIn=8;
TempDiffCondAirOut=5;
TempAirInCond=47;
PresAirCond=101.325;
TempInTurbIncre=5;%Default 1
PresExGasEvap=1.2*101.325;%Assume 1.2Patm
MFlowExGas=1;%Assume unit mass flow rate
MaxTempExGasInEvap=200;
MinTempExGasInEvap=80;
PresInTurbIncre=10;%%%%%%%%
MaxPresAllow=2000;%%%%%
MinPresAllow=200;%%%%%%
end
MinTempExGasInEvap=MinTempExGasInEvap+273.15;
MaxTempExGasInEvap=MaxTempExGasInEvap+273.15;
TempAirInCond=TempAirInCond+273.15;
MaxWorkArraySize=0;
for LoopArraySize=MinTempExGasInEvap:TempInTurbIncre:MaxTempExGasInEvap
    MaxWorkArraySize=MaxWorkArraySize+1;
end



MaxWorkArray=zeros(1,MaxWorkArraySize);
MFlowatMaxW=zeros(1,MaxWorkArraySize);
VFlowatMaxW=zeros(1,MaxWorkArraySize);
AirCondMFlowatMaxW=zeros(1,MaxWorkArraySize);
LoopCounter=0;
CriticalT=refpropm('T','C',0,'',0,'R245fa','pentane',[0.47 0.53]);
for TempInTurb=(MinTempExGasInEvap-
TempDiffEvap):TempInTurbIncre:(MaxTempExGasInEvap-TempDiffEvap)%%%%%%%%%%%%%
TempOutCond=TempAirInCond+TempDiffCondAirIn;%%%%%%%%%%
PresOutCond=refpropm('P','T',TempOutCond,'Q',0,'R245fa','pentane',[0.47
0.53]);%%%%%%%%%%%%%
%if (TempInTurb<CriticalT)
%MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,WorkingFluid);%%%%%%%%%%%
%else
%MaxPresInTurb=refpropm('P','C',0,'',0,WorkingFluid);%%%%%%%%%%%
%end
%if (MaxPresInTurb>MaxPresAllow)%%%%%%
%MaxPresInTurb=MaxPresAllow;%%%%%%%%
%end
if (PresOutCond<MinPresAllow)%%%%%%%%
PresOutCond=MinPresAllow;%%%%%%%%%
TempOutCond=refpropm('T','P',PresOutCond,'Q',0,'R245fa','pentane',[0.47
0.53]);
end
```

```
MFlowRateAtMaxW=0;%%%%%%%
VFlowRateAtMaxW=0;%%%%%%
AirCondMFlowRateAtMaxW=0;
MaxWork=0;%%%%%%%%
if (TempInTurb<CriticalT-10)%%%%%%%%%
MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,'R245fa','pentane',[0.47
0.53]);
if (MaxPresInTurb>MaxPresAllow)%%%%%%
MaxPresInTurb=MaxPresAllow;%%%%%%%%
end
for PresInTurb=PresOutCond:PresInTurbIncre:MaxPresInTurb%%%%%%%%%%%
[DInTurb,HInTurb,SInTurb]=refpropm('DHS','T',TempInTurb,'P',PresInTurb,'R245f
a','pentane',[0.47 0.53]);%%%%%%%%%
IsenSInCond=SInTurb;
PresInCond=PresOutCond;
IsenHInCond=refpropm('H','P',PresInCond,'S',IsenSInCond,'R245fa','pentane',[0
.47 0.53]);
HInCond=HInTurb-IsenEffTurb*(HInTurb-IsenHInCond);
[DInCond,TempInCond]=refpropm('DT','P',PresInCond,'H',HInCond,'R245fa','penta
ne',[0.47 0.53]);
TempInPump=TempOutCond;
PresInPump=PresOutCond;
[HInPump,SInPump]=refpropm('HS','T',TempInPump,'P',PresInPump,'R245fa','penta
ne',[0.47 0.53]);
IsenSOutPump=SInPump;
IsenHOutPump=refpropm('H','P',PresInTurb,'S',IsenSOutPump,'R245fa','pentane',
[0.47 0.53]);
HOutPump=(IsenHOutPump-HInPump)/IsenEffPump+HInPump;
TempOutPump=refpropm('T','P',PresInTurb,'H',HOutPump,'R245fa','pentane',[0.47
0.53]);
[TempEvapPinch,HOutPumpPinch]=refpropm('TH','P',PresInTurb,'Q',0,'R245fa','pe
ntane',[0.47 0.53]);
TempExGasOutEvapPinch=TempEvapPinch+TempDiffEvapPinch;
HExGasInEvap=refpropm('H','T',TempInTurb+TempDiffEvap,'P',PresExGasEvap,'AIR.
ppf');%Assume exhaust gas is air
HExGasOutEvapPinch=refpropm('H','T',TempExGasOutEvapPinch,'P',PresExGasEvap,'
AIR.ppf');%Assume exhaust gas is air
%if (DInTurb<DInCond)
%MinD=DInTurb;
%
%MinD=DInCond;
%end
MFlowRate=(HExGasInEvap-HExGasOutEvapPinch)*MFlowExGas/(HInTurb-
HOutPumpPinch);
Work=MFlowRate*((HInTurb-HInCond)-(HOutPump-HInPump));

TempAirOutCond=TempOutCond-TempDiffCondAirOut;
HAirInCond=refpropm('H','T',TempInPump-
TempDiffCondAirIn,'P',PresAirCond,'AIR.ppf');
HAirPinchCond=refpropm('H','T',TempAirOutCond,'P', PresAirCond,'AIR.ppf');
if (TempOutCond<TempInCond)%superheated at the condenser inlet
HCondInPinch=refpropm('H','T',TempOutCond,'Q',1,'R245fa','pentane',[0.47
0.53]);
else
HCondInPinch=HInCond;
end
CondAirMFlow=(HCondInPinch-HInPump)*MFlowRate/(HAirPinchCond-HAirInCond);
```

```
if (Work>MaxWork)
MaxWork=Work;
MFlowRateAtMaxW=MFlowRate;
%VFlowRateAtMaxW=MFlowRate/MinD;
AirCondMFlowRateAtMaxW=CondAirMFlow;
end
end
end
LoopCounter=LoopCounter+1;
MaxWorkArray(LoopCounter)=MaxWork;
MFlowatMaxW(LoopCounter)=MFlowRateAtMaxW;
%VFlowatMaxW(LoopCounter)=VFlowRateAtMaxW;
AirCondMFlowatMaxW(LoopCounter)=AirCondMFlowRateAtMaxW;
end

PlotTempExGasInEvap=MinTempExGasInEvap-
273.15:TempInTurbIncre:MaxTempExGasInEvap-273.15;
figure

plot(PlotTempExGasInEvap,MaxWorkArray,'.')
hold on

xlabel('Exhaust Gas Temperature (C)')
ylabel('Maximum Work (J)')
title('Maximum Work versus Exhaust Gas Temperature for Different Working
Fluids')
figure

plot(PlotTempExGasInEvap,MFlowatMaxW,'.')
hold on


xlabel('Exhaust Gas Temperature (C)')
ylabel('Mass Flow Rate at Maximum Work (kg/s)')
title('Mass Flow Rate at Maximum Work versus Exhaust Gas Temperature for
Different Working Fluids')
%figure
%for j=1:1:WorkingFluidSize(1)
%plot(PlotTempExGasInEvap,PlotVFlowatMaxW(j,:),'.')
%hold on
%end
%legend('water  ','toluene','R245fa ','R123
','benzene','pxylene','ethanol','R236ea ','R245ca ')
%xlabel('Exhaust Gas Temperature (C)')
%ylabel('Maximum Volumetric Flow Rate at Maximum Work (m^3/s)')
%title('Maximum Volumetric Flow Rate at Maximum Work versus Exhaust Gas
Temperature for Different Working Fluids')
figure

plot(PlotTempExGasInEvap,AirCondMFlowatMaxW,'.')
hold on

xlabel('Exhaust Gas Temperature (C)')
ylabel('Condenser Air Mass Flow Rate at Maximum Work (kg/s)')
```

```
title('Condenser Air Mass Flow Rate at Maximum Work versus Exhaust Gas
Temperature for Different Working Fluids')
```

MATLAB codes of the model described in section 6 and section 7.3 are followings. Data should be retrieved manually and plotted using Excel (Microsoft, Inc.).

```
clear
clc
WorkingFluidArray=['water   ';'toluene ';'R245fa  ';'R123    ';'benzene
';'pxylene ';'ethanol ';'R236ea  ';'R245ca  '];
DefaultSetting=input('Default Settings (y or n): ','s');
if DefaultSetting=='n'
IsenEffTurb=input('Input the isentropic efficiency of the turbine: ');
IsenEffPump=input('Input the isentropic efficiency of the pump: ');
TempDiffEvap=input('Input the inlet temperature difference (C) in the
evaporator: ');
TempDiffCond=input('Input the outlet temperature difference (C) in the
condenser: ');
TempAirInCond=input('Input the temperature (C) of the air used for cooling in
the condenser: ');
MaxTempExGasInEvap=input('Input the maximum inlet temperature (C) of the
exhaust gas in the evaporator: ');
MinTempExGasInEvap=input('Input the minimum inlet temperature (C) of the
exhaust gas in the evaporator: ');%%%%%%%%%%%
TempInTurbIncre=input('Input the increment of the inlet temperature of the
turbine (graph resolution): ');
PresInTurbIncre=input('Input the increment of the inlet pressure of the
turbine (optimization resolution): ');%%%%%%%%%%
MaxPresAllow=input('Input the maximum pressure (kPa) allowed in the system:
');%%%%%%%
MinPresAllow=input('Input the minimum pressure (kPa) allowed in the system:
');%%%%%%%
else%!!!!!!!!!!!!!To be added in Model 4
IsenEffTurb=0.85;
IsenEffPump=0.85;
TempDiffEvap=10;
TempDiffEvapPinch=5;
TempDiffCondAirIn=8;
TempDiffCondAirOut=5;
TempAirInCond=47;
PresAirCond=101.325;
TempInTurbIncre=5;%Default 1
PresExGasEvap=1.2*101.325;%Assume 1.2Patm
MFlowExGas=1;%Assume unit mass flow rate
MaxTempExGasInEvap=380;
MinTempExGasInEvap=80;
PresInTurbIncre=10;%%%%%%%%
MaxPresAllow=2000;%%%%%
MinPresAllow=200;%%%%%%
end
MaxTempExGasInEvap=MaxTempExGasInEvap+273.15;
MinTempExGasInEvap=MinTempExGasInEvap+273.15;%%%%%%%%%%
TempAirInCond=TempAirInCond+273.15;

T9=150+273.15;
T10=TempAirInCond;
WorkingFluid='R123';
CriticalT=refpropm('T','C',0,'',0,'R245fa','pentane',[0.47 0.53]);
TempInTurb=T9-TempDiffEvap;
```

```
TempOutCond=TempAirInCond+TempDiffCondAirIn;%%%%%%%%%%
PresOutCond=refpropm('P','T',TempOutCond,'Q',0,'R245fa','pentane',[0.47
0.53]);%%%%%%%%%%%%%

if (PresOutCond<MinPresAllow)%%%%%%%%
PresOutCond=MinPresAllow;%%%%%%%%%
TempOutCond=refpropm('T','P',PresOutCond,'Q',0,'R245fa','pentane',[0.47
0.53]);
end
MaxFirstLawEff=0;%%%%%%%%
T5=0;P5=0;T6=0;P6=0;T7=0;P7=0;T1=0;P1=0;T2=0;P2=0;T3=0;P3=0;T4=0;P4=0;

MaxPresInTurb=refpropm('P','T',TempInTurb,'Q',1,'R245fa','pentane',[0.47
0.53]);
if (MaxPresInTurb>MaxPresAllow)%%%%%%
MaxPresInTurb=MaxPresAllow;%%%%%%%%
end
for PresInTurb=PresOutCond:PresInTurbIncre:MaxPresInTurb%%%%%%%%%%%
[HInTurb,SInTurb]=refpropm('HS','T',TempInTurb,'P',PresInTurb,'R245fa','penta
ne',[0.47 0.53]);%%%%%%%%%
IsenSInCond=SInTurb;
PresInCond=PresOutCond;
IsenHInCond=refpropm('H','P',PresInCond,'S',IsenSInCond,'R245fa','pentane',[0
.47 0.53]);
HInCond=HInTurb-IsenEffTurb*(HInTurb-IsenHInCond);
TempInCond=refpropm('T','P',PresInCond,'H',HInCond,'R245fa','pentane',[0.47
0.53]);
TempInPump=TempOutCond;
PresInPump=PresOutCond;
[HInPump,SInPump]=refpropm('HS','T',TempInPump,'P',PresInPump,'R245fa','penta
ne',[0.47 0.53]);
IsenSOutPump=SInPump;
IsenHOutPump=refpropm('H','P',PresInTurb,'S',IsenSOutPump,'R245fa','pentane',
[0.47 0.53]);
HOutPump=(IsenHOutPump-HInPump)/IsenEffPump+HInPump;
TempOutPump=refpropm('T','P',PresInTurb,'H',HOutPump,'R245fa','pentane',[0.47
0.53]);
FirstLawEff=((HInTurb-HInCond)-(HOutPump-HInPump))/(HInTurb-HOutPump);
if MaxFirstLawEff<FirstLawEff%%%%%%%%%%%
T5=TempInTurb;P5=PresInTurb;T6=TempInCond;T1=TempInPump;T7=T1;P1=PresInPump;P
7=PresInPump;P6=P7;
T2=TempOutPump;P2=P5;
end
end
[T3,S3]=refpropm('TS','P',P2,'Q',0,'R245fa','pentane',[0.47 0.53]);
[T4,S4]=refpropm('TS','P',P2,'Q',1,'R245fa','pentane',[0.47 0.53]);
[T7,S7]=refpropm('TS','P',P7,'Q',1,'R245fa','pentane',[0.47 0.53]);
[T1,S1]=refpropm('TS','P',P1,'Q',0,'R245fa','pentane',[0.47 0.53]);
S2=refpropm('S','T',T2,'P',P2,'R245fa','pentane',[0.47 0.53]);
S5=refpropm('S','T',T5,'P',P5,'R245fa','pentane',[0.47 0.53]);
S6=refpropm('S','T',T6,'P',P6,'R245fa','pentane',[0.47 0.53]);
T8=T3+TempDiffEvapPinch;
T11=T7-TempDiffCondAirOut;
S10=S1;S11=S7;S8=S3;S9=S5;
```

```
T1=T1-273.15;T2=T2-273.15;T3=T3-273.15;T4=T4-273.15;T5=T5-273.15;T6=T6-
273.15;T7=T7-273.15;T8=T8-273.15;T9=T9-273.15;
T10=T10-273.15;T11=T11-273.15;
TArray=[T1 T2 T3 T4 T5 T6 T7 T8 T9 T10 T11];
SArray=[S1 S2 S3 S4 S5 S6 S7 S8 S9 S10 S11];
%Vapor Dome
CriticalT=refpropm('T','C',0,'',0,'R245fa','pentane',[0.47 0.53]);
Counter=0;
incre=0.5;
for T=273.15:incre:CriticalT;
    Counter=Counter+1;
end
VDomeT1=273.15:incre:CriticalT;
VDomeS1=zeros(1,Counter);
VDomeT2=zeros(1,Counter);
VDomeS2=zeros(1,Counter);
DomeT1=zeros(Counter,1);
DomeT2=zeros(Counter,1);
DomeS1=zeros(Counter,1);
DomeS2=zeros(Counter,1);
Counter2=1;
for T=273.15:incre:CriticalT
VDomeS1(Counter2)=refpropm('S','T',T,'Q',0,'R245fa','pentane',[0.47 0.53]);
Counter2=Counter2+1;
end
VDomeT1(Counter2-1)=CriticalT;
VDomeS1(Counter2-1)=refpropm('S','C',0,'',0,'R245fa','pentane',[0.47 0.53]);
Counter3=1;
for T=273.15:incre:CriticalT
    VDomeT2(Counter3)=CriticalT-incre*(Counter3-1);
    Counter3=Counter3+1;
end
Counter4=1;
VDomeT2(1)=CriticalT-0.01;
for T=273.15:incre:CriticalT
VDomeS2(Counter4)=refpropm('S','T',VDomeT2(Counter4),'Q',1,'R245fa','pentane'
,[0.47 0.53]);
Counter4=Counter4+1;
end
VDomeT2(1)=CriticalT;
VDomeS2(1)=refpropm('S','T',T,'Q',0,'R245fa','pentane',[0.47 0.53]);
VDomeT1=VDomeT1-273.15;
VDomeT2=VDomeT2-273.15;


DomeT1(:,1)=VDomeT1(1,:);
DomeT2(:,1)=VDomeT2(1,:);
DomeS1(:,1)=VDomeS1(1,:);
DomeS2(:,1)=VDomeS2(1,:);

plot(VDomeS1,VDomeT1,'k-')
hold on
plot(VDomeS2,VDomeT2,'k-')
hold on
```