A Study of the Kalina Cycle System 11 for the Recovery of Industrial Waste Heat with Heat Pump Augmentation

by

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Abstract

The recovery of industrial waste heat is becoming an area of increased interest due to the ever climbing cost of energy. In the past, the low temperatures that most industrial waste heat is at have prevented the recovery and use of the waste heat stream. Through the application of the Kalina Cycle System 11 (KCS11) with heat pump/refrigeration augmentation, waste heat can be recovered from streams with a lower temperature than would normally be possible.

This thesis investigates the theoretical viability of using a Kalina Cycle System 11 with vapor compression refrigeration cycle augmentation to convert industrial waste heat into useable power, and compares that to a non-augmented KCS11 and an organic Rankine cycle. It was found that with a source temperature of 200 °C, the KCS11 can achieve thermal efficiencies in excess of 30%. By utilizing the correct vapor compression refrigeration cycle to recover the waste heat and supply the waste heat to the KCS11, a portion of the waste heat can be recovered and utilized as a power source.

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List of Abbreviations

COP	Coefficient of Performance
HP	Heat Pump
P	Pressure
T	Temperature
Y	Mass Fraction (Ammonia)
y	Mass Fraction (Double Stage Vapor Compression System)
ω	Mass Fraction (KCS11)
η	Efficiency

Subscripts

th	Thermal
BE	Break Even
NET	Net
cond	Condenser

Chapter 1

Background

1.1 Introduction

Due to the continual rise in the cost and consumption of energy, the utilization of low quality heat sources such as low temperature waste heat has become an area of increased interest [1]. New technologies and power cycles have made the recovery of waste heat more economically attractive.

One of the reasons that low temperature waste heat has become an area of interest is that no process is completely efficient, which is due to irreversibilities in the process [2]. These irreversibilities, such as friction that are present in all mechanical devices, or Joule heating in electrical devices, manifest as an increase in the temperature of the process equipment. In order to prevent premature failure in this equipment, it is necessary to remove the excess thermal energy. The thermal energy that is removed is known as waste heat since historically it has not been economically feasible to recover and use this wasted energy.

As the consumption of energy increases, the amount of waste heat generated will also increase. Low temperature waste heat accounts for approximately 50% of the heat generated in industry [3]. The amount of energy that has historically been lost as waste heat is staggering when the magnitudes of energy consumed is taken into account. In 2006, 21,098 trillion Btu's of energy were consumed just in manufacturing in the United States alone [4].

The Kalina cycle and organic Rankine cycle provide a couple of possible solutions to the problem of recovering the low temperature energy that is usually thrown away in industrial waste heat. Organic Rankine cycles are Rankine cycles that utilize an organic substance such as hydro-carbons or refrigerants as the working fluid. Organic Rankine cycles have been utilized in many ways to recover low temperature energy from various sources such as waste heat, geothermal heat sources, and other renewable heat sources [1, 3, 5-7].

The Kalina cycle is a proprietary power cycle that was developed and patented by Alexander Kalina in the late 1970's and early 1980's [8-10]. The Kalina cycle utilizes an aqueous-ammonia mixture as the working fluid. The use of a binary fluid allows the combination of water and ammonia to be adjusted to optimize the system based on the working parameters ranging from direct fired applications to low temperature waste heat recovery [1, 6, 8-15]. In this study, the Kalina cycle system 11 (KCS11) is analyzed for use as a means to recover the low quality thermal energy flows, waste heat, from industrial applications.

One of the problems encountered in the recovery of waste heat from industrial processes is the management of the waste heat fluid flow. Most industrial processes utilize a closed system for the direct cooling of the equipment. Generally, this closed system would interact with another fluid, called the secondary fluid, through a heat exchanger. The waste heat would be transferred into the secondary fluid and disposed of into the environment. The cooling efficiency of this system is limited to the wet bulb temperature if a water cooling tower is used, the dry bulb temperature if sensible air cooling is used, or the temperature of a local body of water, if one is available, which is used to cool the primary cooling fluid [6]. Likewise, the cooling of the primary cooling fluid is limited to the condenser temperature when any type of power cycle is used to interact with the cooling fluid in an attempt to recover the waste heat from the waste heat stream. The condenser temperature of the power cycle is similarly limited by the environmental conditions. This work will show that through the use of heat pump augmentation, the primary cooling fluid temperature can be controlled to an optimized temperature given the local environmental conditions, which will increase the efficiency of the industrial process, and the maximum temperature of the power cycle can be increased above the temperature of the primary cooling fluid.

1.2 Power Cycles

In order to fully understand the requirements, drawbacks, and benefits of various energy conversion processes, one must have a basic understanding of power cycles and working fluids. In essence, a power cycle is a sequence of thermodynamic processes that a heat engine operates upon, where the system returns to its original state at the conclusion of the cycle. A heat engine is a device that operates between two temperature reservoirs and is intended to do some type of work through the conversion of an energy source [16, 17].

There are various types of power cycles that are defined by a number of factors in how the cycle operates. Some of the various defining factors are; does the working fluid change phase, is the working fluid rejected at the end of the cycle instead of being brought back to the initial state, is the cycle open or closed, and is the heat input of the cycle through an internal or external source [16]? Also, some cycles are designed for a particular application such as vehicle propulsion or electrical power production.

Since this work is concerned with the recovery of waste heat in order to produce usable electrical power, the power cycle that is of most interest is the Rankine cycle. The Rankine cycle is a vapor cycle that is commonly used in the production of electricity. The basic Rankine cycle consists of four processes and is shown in figure 1.1:

1-2 Isentropic compression
2-3 Constant pressure heat addition, usually to a super heated state
3-4 Isentropic expansion in a turbine
4-1 Constant pressure heat rejection to a saturated liquid state

There are several variations on the basic Rankine cycle through the addition of devices and components, but the basic process follows the same path [18].

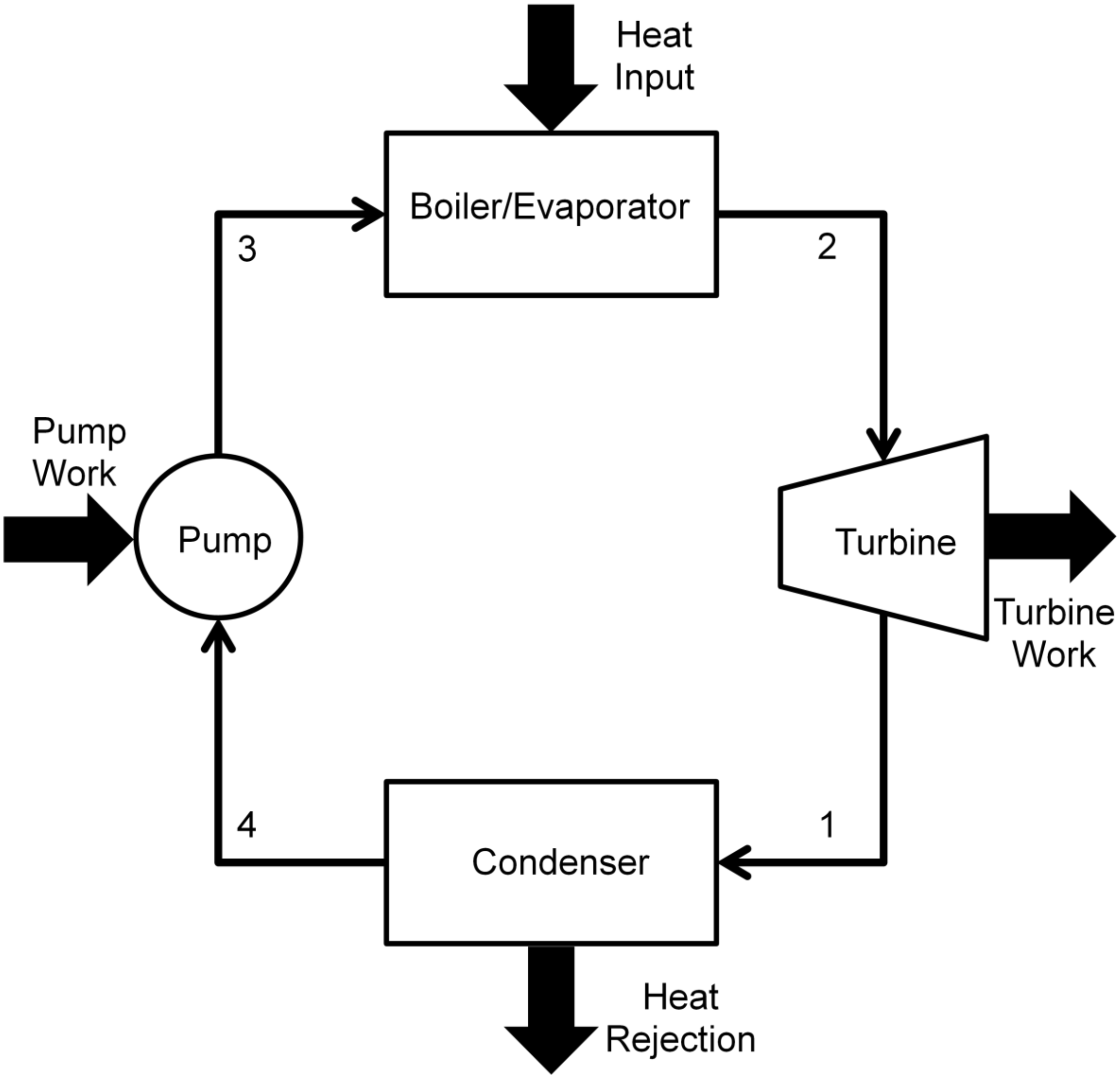

Figure 1.1: Schematic of the Basic Rankine Cycle

1.3 Working Fluids

In thermodynamic power cycles, thermal energy is converted into a useable form of work either for direct drive processes or electrical power production. These cycles require that thermal energy, or heat, be transferred from a high temperature source, into a heat engine, and then rejected to a low temperature sink.

Most devices that operate on a cycle use a fluid, known as the working fluid. The working fluid acts as an energy conduit in the cycle which causes an increase in the heat transfer rate of the cycle and increase the rate that the cycle can operate at. The selection of the working fluid used in any cycle is an important step in ensuring the cycle operates at optimum efficiency. A simple list of desirable characteristics for a pure working fluid has been developed by [19].

1. A critical point that is above the maximum material temperature with a safe maximum pressure. If the maximum pressure is to high at the maximum temperature, material strength problems are encountered.
2. The fluid needs to have a low triple-point temperature to prevent any solidification problems
3. A saturation pressure at the cooling medium temperature that is not too low. If the condenser pressure is too low, it can cause leaking problems.
4. It is desirable to have a large latent heat of vaporization to help minimize the mass flow required.
5. An inverted U shaped saturation dome. The dome shape will help to minimize the formation of droplets in the turbine.

6. The working fluid needs to have a high thermal conductivity.
7. There are also economic and safety characteristics that are as important to the selection of the working fluid as the thermo-physical properties of the fluid. The fluid needs to be inert, cheap, available in large enough quantities, and it needs to be nontoxic.

While the previous list is directed at high temperature working fluid selection, many of the same concepts are important for the selection of a low temperature working fluid, but additional care has to be taken in the selection of the working fluid in respects to the efficiency of a low temperature cycle.

For example, step one in the selection of a high temperature working fluid is concerned with the critical point of the potential working fluid and the saturation pressure near the critical point. The concern is that at these high temperatures and pressures the frequency and severity of material failures would reach a dangerous or uneconomical level. In a low temperature application there is very little concern over material failure since the working fluid does not come anywhere near the maximum temperature of the metals, but the critical point of the fluid is still an important property. It is still desirable to have a working fluid whose critical point is above the maximum temperature of the power cycle.

Because of the new requirements placed on the low temperature cycle working fluid, several organic fluids such as ammonia or various refrigerants have been utilized. When an organic based fluid is used as the working fluid in the Rankine cycle, the cycle is referred to as an organic Rankine cycle. Even though there are a large number of fluids that could be employed as an acceptable working fluid, there are problems with the use of these fluids. The primary problem with using a pure substance as a working fluid in a low temperature is the property of a pure fluid to vaporize at a constant temperature. Because the vaporization temperature is constant when vaporizing a pure substance, there is a loss of useable energy, or exergy. This loss is due to entropy generation, which is increased when heat transfer takes place over a large temperature difference. One of the methods of solving this problem is the use of binary fluids, such an aqueous-ammonia solution. In a binary mixture the condensing and vaporizing temperature varies. This variation in the temperature allows the temperature profile of the working fluid to better match the temperature profile of the temperature source or sink. By matching the temperature profile of the working fluid to the profile of the source, the efficiency of the cycle can be improved by reducing exergy losses through entropy generation [1, 6, 14, 20-21]. A recent cycle developed that utilizes an aqueous-ammonia solution is known as the Kalina Cycle.

1.4 Kalina Cycle

Most simply, the Kalina cycle is a modified Rankine cycle, and was developed in an attempt to reduce the losses incurred by the use of a pure substance working fluid. The goal of the Kalina cycle is that by using a mixture of ammonia and water as the working fluid, the temperature profile of the working fluid will more closely follow the temperature profile of the heat source or sink. There are several variations of the basic Kalina cycle based on the application. For example, the Kalina cycle system five (KCS5) is primarily focused for direct fired applications, the Kalina cycle system six (KCS6) is intended for use as the bottoming cycle in a combined cycle, and the Kalina cycle system eleven (KCS11) is particularly useful as a low temperature geothermal driven power plant cycle [14].

1.4.1 Kalina Cycle System 11 (KCS11)

The Kalina cycle system 11, which for simplicity will now be denoted as KCS11, is a modified Rankine cycle. The KCS11, as with all Kalina cycles, utilizes an aqueous-ammonia mixture as the working fluid. By adjusting the mass fraction of ammonia in the mixture, the KCS11 can be optimized based on the input conditions.

While the KCS11 is a fairly simple power cycle, there are a number of additional steps and parameters that must be understood in order to fully appreciate the cycle. Figure 1.2 shows a basic schematic of the KCS11. The easiest way to understand the cycle process is to step through the cycle, and the easiest place to start from is at state five.

At state 5, the total aqueous-ammonia mixture leaves the evaporator. When considering the Rankine cycle, the working fluid is at least a saturated vapor when it leaves the evaporator or boiler. In the KCS11, the working fluid mixture leaves the evaporator as a saturated mixture. The quality of the mixture is a function of the concentration of ammonia in the working fluid mixture, the temperature of the heat source, and the pressure of the working fluid. Once the working fluid mixture leaves the evaporator, it enters the phase separator. The task of the phase separator is too separate the working fluid into two separate streams. The saturated vapor portion of the working fluid passes through the separator to state 6, and the saturated vapor is an ammonia rich mixture. The saturated vapor continues on to the turbine where it undergoes an isentropic expansion to produce work. The saturated vapor is expanded into a saturated mixture and exits the turbine. The saturated mixture is at state 10. The mass fraction of the working fluid that did not vaporize in the evaporator leaves the separator as a saturated liquid at state 7 and is notated ω. The saturated liquid portion of the working fluid is a weaker ammonia mixture than the saturated vapor portion of the working fluid. The hot saturated liquid is sent to the regenerator. In the regenerator, the saturated liquid gives up some of its thermal energy to the cold working fluid mixture that has left the condenser. The now cooled mixture leaves the regenerator at state 8. Even though the working fluid mixture at state 8 has been acceptably cooled, it is still at the maximum cycle pressure. In order to mix the mass fraction of the working fluid that passed though the regenerator with the mass fraction of the working fluid that was used to drive the turbine, the portion of the working fluid at state 8 has to be brought to a lower pressure. The drop in pressure is accomplished with a throttling valve. The cool, high pressure working fluid expands in the expansion valve and brought to the same pressure as the portion of the working fluid that passed through the turbine at state 9. Now that the two flows of the working fluid are at the same pressure, they enter the absorber. The absorber is the area in the cycle where the two flows are reunited. The recombined mixture leaves the absorber at state 1. Even though the two mass flows are recombined, the mixture is still a saturated mixture. The working fluid then passes through the condenser where heat is rejected and the working fluid is brought back to a saturated liquid. The saturated liquid leaves the condenser at state 2. A pump is then used to isentropically compress the working fluid mixture to the maximum pressure of the cycle to state 3. The cold working fluid then enters the regenerator in order to recover some of the thermal energy used in heating the saturated liquid portion of the working fluid. The cold working fluid mixture is preheated, and leaves the regenerator at state 4. The preheated working fluid mixture then enters the evaporator to start the process over again.

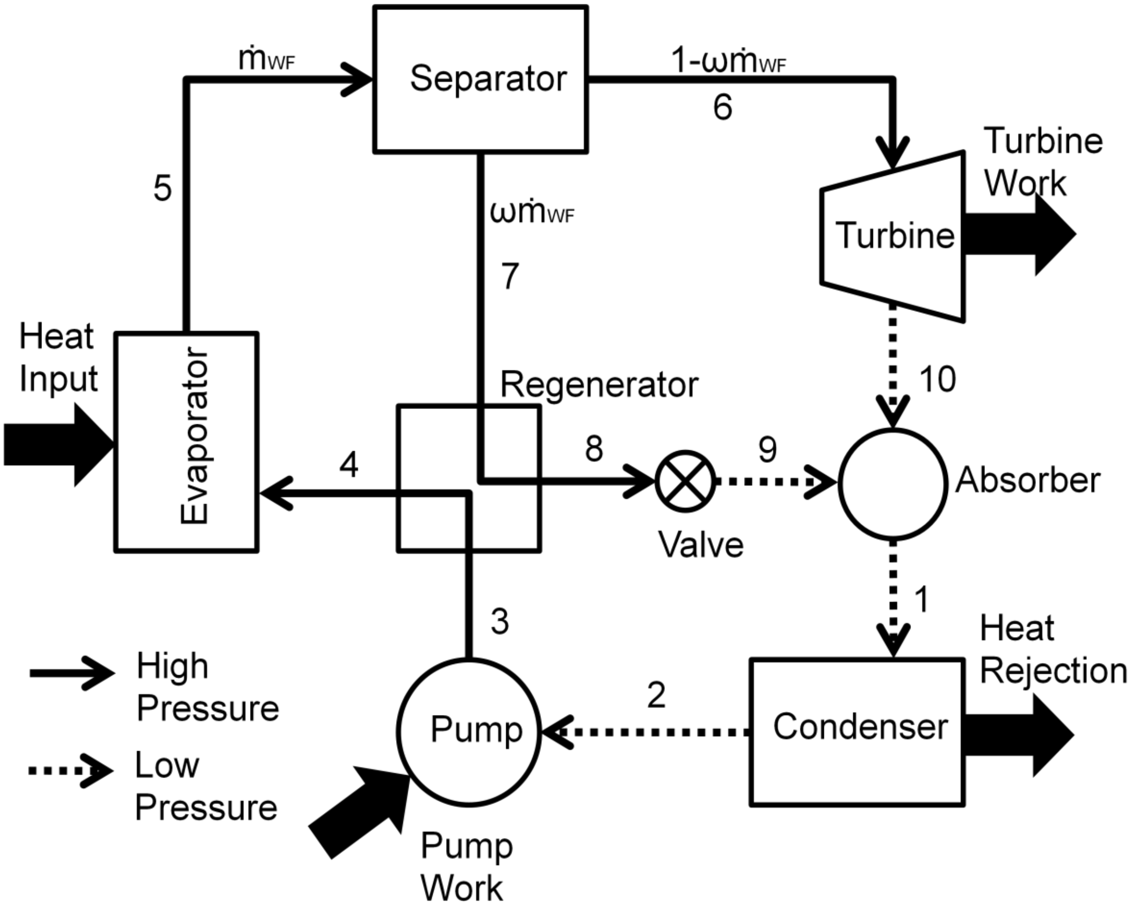

Figure 1.2: Schematic of the Kalina Cycle System 11 (KCS11)

1.4.2 Current Applications of the Kalina Cycle

The Kalina cycle has seen limited deployment as a power cycle for use in geothermal applications. Most notably a Kalina cycle that has been put into operation in Iceland is generating power from a geothermal sourced brine, and the cycle is currently providing 80% of the power required by the local town of Husavik [22].

1.5 Heat Pumps

The KCS11 has a thermal efficiency that is comparable to organic Rankine cycles. The KCS11 has a drawback in respect to the recovery of industrial waste heat sources which is caused by the main component of the cycle that causes such an increase in efficiency, the regenerator. In an organic Rankine cycle, the working fluid enters the evaporator/boiler at the condenser temperature. In the KCS11, due to the regenerator, the working fluid enters the evaporator with a temperature that is close to the maximum cycle temperature. This smaller inlet to outlet temperature difference in the Kalina cycle means that the industrial cooling fluid return temperature is limited by the minimum temperature in the evaporator. This means that the cooling fluid then has to go through an additional cooling process. By using a heat pump in between the industrial process cooling fluid and the KCS11, the temperature of the cooling fluid can be managed and cooled to the desired temperature, regardless of the ambient temperature.

Another problem that has to be faced when attempting to recovery low temperature waste heat flows is a matter of efficiency limitations. The thermal efficiency of a power cycle is limited by the Carnot efficiency. The Carnot efficiency is related to the ratio of temperatures for the cycle, and it shows that the greater the difference in the maximum and minimum working fluid temperatures in a cycle the higher the maximum possible thermal efficiency of that cycle.

$$\eta_{Carnot} = 1 - \frac{T_{min}}{T_{max}} \tag{1.1}$$

The Carnot efficiency reveals a significant limitation with waste heat recovery from industrial sources. Many industrial waste heat flows, even though the flows contain a significant amount of energy, have low temperatures that prevent an efficient method of recovery. The temperature of the waste heat flow needs to be increased in order to make the waste heat accessible.

Through the application of a heat pump, both of the previous limitations can be addressed. The cold side of the heat pump cycle can be set so that the temperature of the cooling fluid can be managed and the industrial process can be controlled to a higher degree. The heat pump cycle can also increase the efficiency of the KCS11 by increasing the maximum temperature of the power cycle. There are several different heat pump configurations that can be utilized depending on the desired cycle boundary conditions.

1.5.1 Standard Heat Pump/Refrigeration Cycle

The standard heat pump cycle is a two phase cycle that is used to force the transfer of energy from a lower temperature environment to a higher temperature environment is known as a heat pump cycle or a refrigeration cycle. The standard heat pump/refrigeration cycle is referred to as a vapor-compression refrigeration cycle. Figure 1.3 shows a schematic of the standard vapor-compression refrigeration cycle.

At state one of the cycle, the refrigerant leaves the evaporator as a saturated vapor. The vapor is then isentropically compressed to a super heated vapor at state two. The refrigerant then enters the condenser at an elevated pressure and temperature. In the condenser the refrigerant cools to a saturated vapor, at the same pressure as state two, and then begins to condense at a constant temperature. Once the refrigerant is condensed to a saturated liquid phase, the liquid exits the condenser at state three. The saturated liquid refrigerant then goes through an adiabatic expansion, from the condenser pressure to the evaporator pressure in the expansion valve, to state four. The refrigerant enters the evaporator as a saturated mixture, and begins vaporizing at a constant temperature and pressure. When the refrigerant has left the evaporator, the vapor-compression refrigeration cycle is complete.

The vapor-compression refrigeration cycle is a very effective cycle that is not only efficient but also reliable. The cycle is increasingly being used to not only cool homes but to warm them as well by using the cycle as a heat pump.

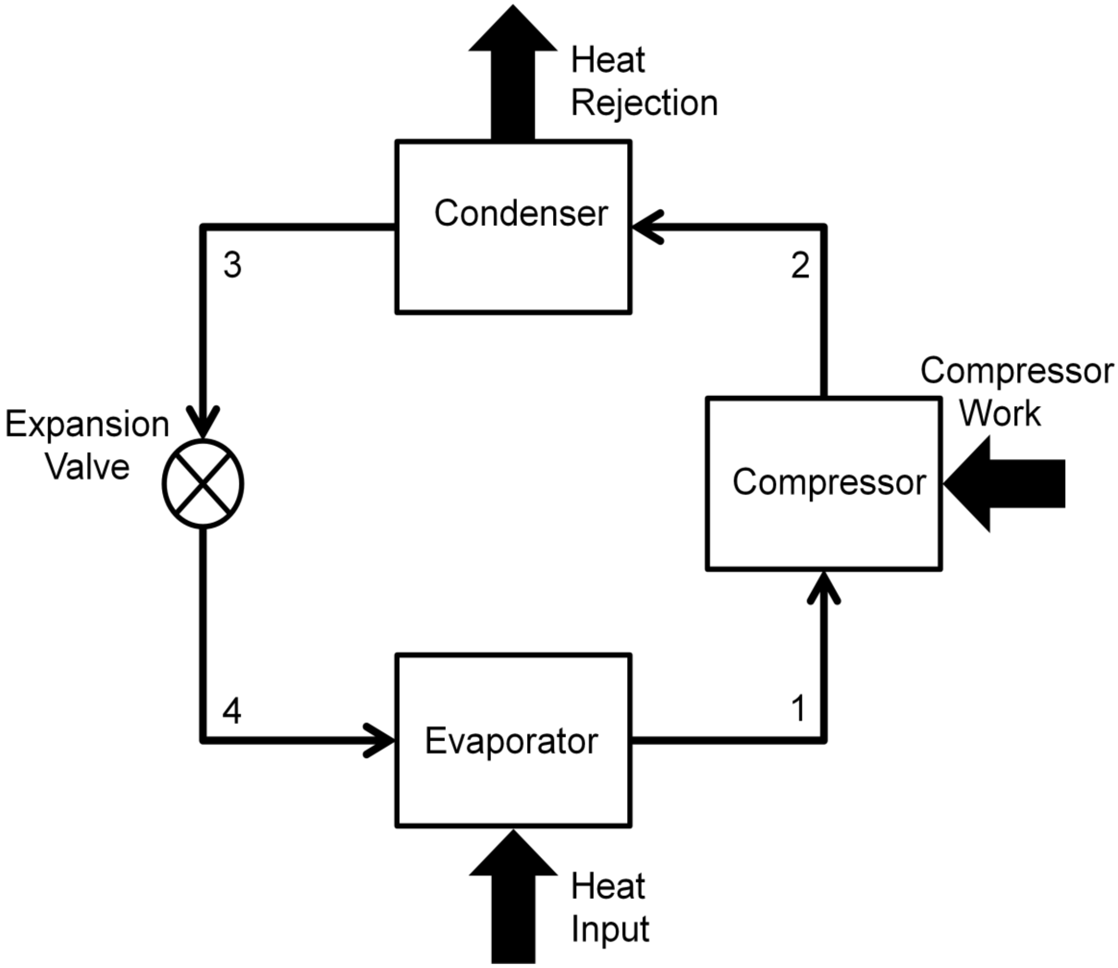

Figure 1.3: Schematic of Basic Vapor-Compression Refrigeration Cycle

1.5.2 Cascade Refrigeration Systems

A cascade refrigeration system is simply a combination of two or more vapor-compression refrigeration cycles. The cascade refrigeration system addresses one of the problems encountered by a single vapor-compression refrigeration cycle; specificall, the coefficient of performance, which is a ratio of the amount of heat added to or received from the heat pump/refrigeration cycle divided by the work required to operate the cycle, drops significantly as the temperature difference across the cycle increases. One of the causes of this loss is the fact that it is difficult for a standard compressor to compress the vapor over the large pressure difference. The cascade refrigeration system allows for a high temperature difference across the cycle by utilizing a low temperature vapor-compression refrigeration cycle that interacts with a high temperature vapor-compression refrigeration cycle through a heat exchanger that acts as the condenser of the low temperature cycle and the evaporator of the high temperature cycle [23]. Figure 1.4 shows a cascade refrigeration system.

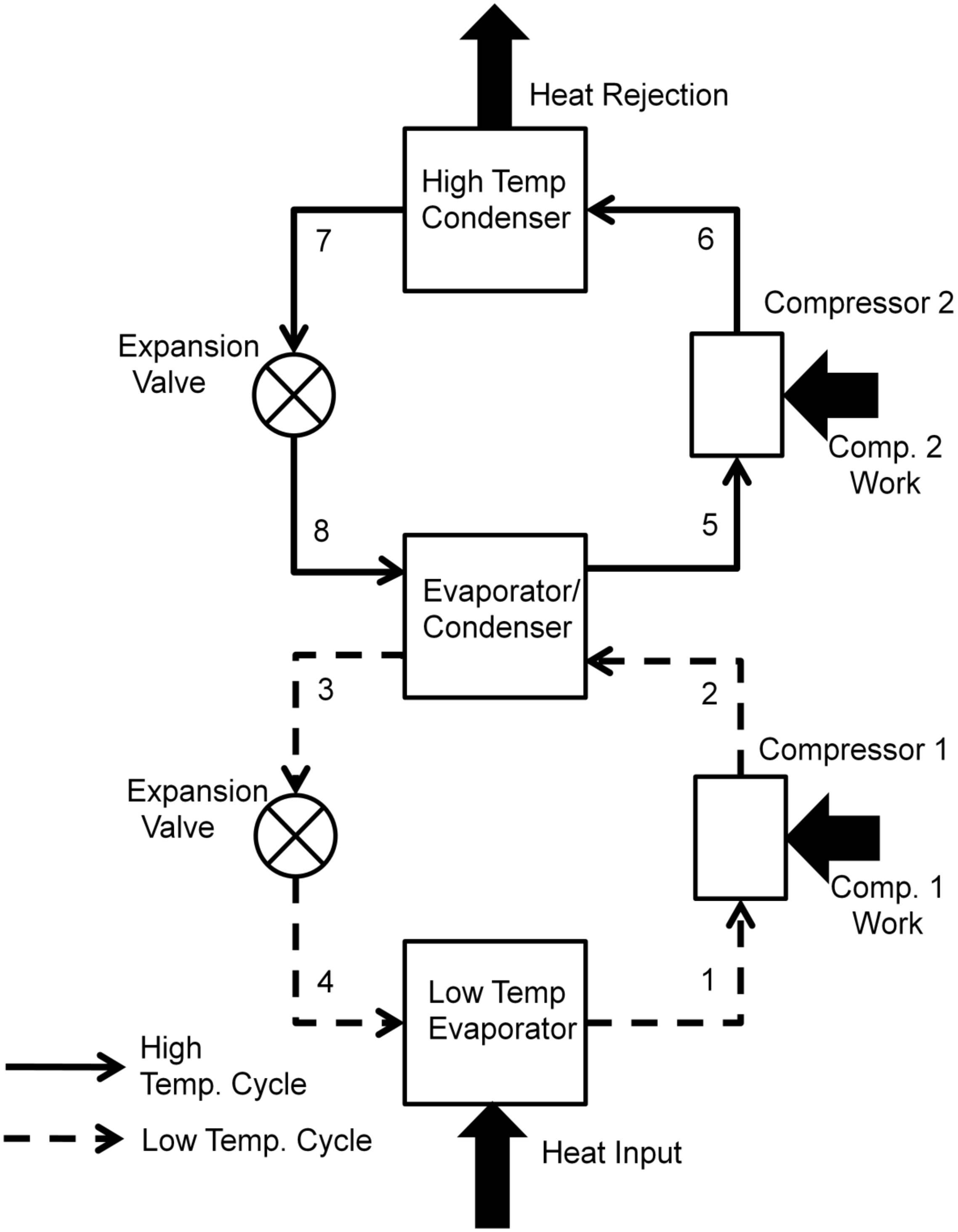

Figure 1.4: Schematic View of a Cascade Vapor Compression Refrigeration System

One of the benefits of the cascade refrigeration cycle is that since the two vapor-compression refrigeration cycles are both closed cycles, one refrigerant that is well suited for low temperature applications can be used in the low temperature cycle and a different refrigerant that is designed for high temperature applications can be used in the high temperature cycle. While being able to use different refrigerants has certain benefits, it is not always necessary or useful. When only a single refrigerant is used, but a large temperature difference is needed, a modified cascade refrigeration system can be utilized.

1.5.3 Multistage Compression Refrigeration Systems

Not all situations require or need the use of two different refrigerants. When the same refrigerant is used in the low temperature cycle and the high temperature cycle a multistage compression refrigeration system can be used in place of the cascade refrigeration system.

The benefit of using a multistage compression cycle when a single refrigerant is used is that the two closed cycles are replaced by a single closed cycle. The heat exchanger that allows the high and low temperature cycles to interact is replaced with a flash chamber. The flash chamber removes the loss that is introduced to the system due to the necessary temperature difference that is present in the interacting heat exchanger of the cascade refrigeration system [23]. Figure 1.5 shows a diagram of a two-stage multistage compression refrigeration system.

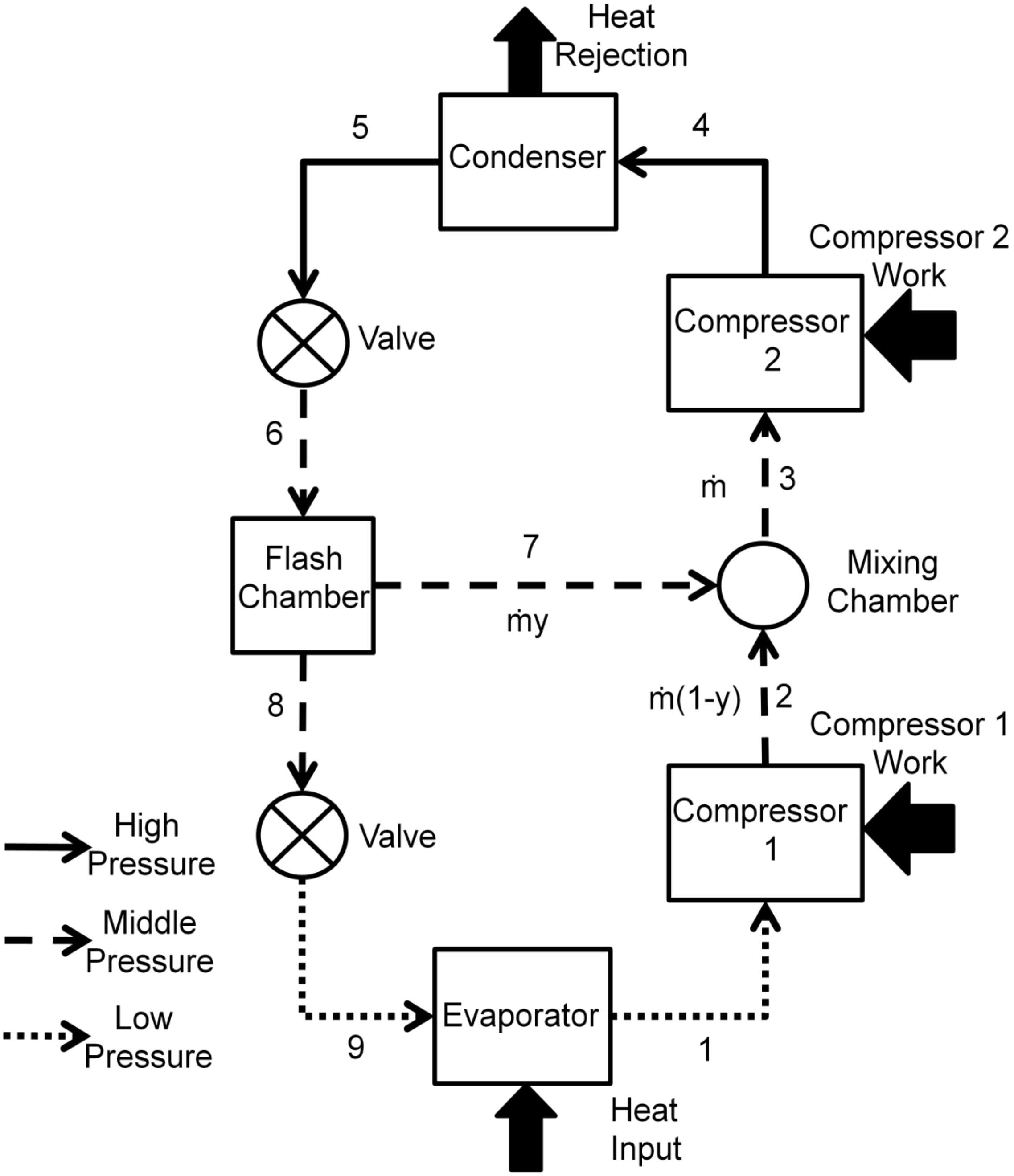

Figure 1.5: Schematic of a Two Stage Multistage Vapor Compression Refrigeration System

The two stage multistage compression refrigeration system is analyzed in the same manner as a vapor-compression refrigeration cycle with a few exceptions, and is easiest to explain starting from the condenser exit. At state five, the total mass flow leaves the condenser as a saturated liquid. The saturated liquid is then adiabatically expanded to a middle pressure, which is set by maximizing the coefficient of performance, to state six in the first of two expansion valves. The saturated mixture then enters the flash chamber. The flash chamber is nothing more than a phase separator. The flash chamber separates the saturated vapor portion of the mass flow and sends it to state seven. The saturated liquid portion of the mass flow is then sent to the second expansion valve at state eight. The saturated liquid is then adiabatically expanded to the evaporator pressure in the second expansion valve to state nine. The saturated mixture is then vaporized in the evaporator to a saturated vapor phase at state one. The saturated vapor is then isentropically compressed to the middle pressure to state two. The super heated vapor of state two enters the mixing chamber where it is mixed with the saturated vapor portion of the mass flow that was separated by the flash chamber. The now recombined mass flow leaves the mixing chamber as a super heated vapor at state three. The super heated vapor is then isentropically compressed to the maximum system pressure at state four. The super heated vapor then enters the condenser where it cools to a saturated vapor, and then condenses at a constant temperature to a saturated liquid state which is state five, and the cycle is completed.

Chapter 2

System Analysis

2.1 Introduction

The recovery of waste heat from an industrial process poses several challenges that must be overcome. Some of these challenges are the low temperature and the high mass flow of the waste heat stream, and the fact that the recovery of the waste heat can not hinder the industrial process. While there are several simple methods of disposing of the waste heat, such as the use of cooling towers or natural water sources, the reliable and efficient recovery of the waste heat requires a more complex approach. One such method is the application of the KCS11 with heat pump augmentation to recover the waste heat and convert the waste heat into a clean reliable source of energy. Another possible option for waste heat recovery is the use of an organic Rankine cycle.

In order to evaluate and appreciate the potential value of a system of thermodynamic cycles, the individual cycles have to be broken down into their simplest components. Once the cycle has been broken down, the individual components can be studied through the application of the first law of thermodynamics. Once the individual components have been evaluated, the individual component results can be recombined to determine the benefit of the cycle.

2.2 Kalina Cycle System 11 (KCS11)

The Kalina cycle is a modified Rankine cycle that uses an aqueous ammonia solution as the working fluid rather than a simple one component substance. Due to the inclusion of ammonia in the working fluid, certain Kalina cycle configurations are very effective in the recovery of low temperature energy sources. This work concentrates on the evaluation of the Kalina Cycle System 11 (KCS11) for the recovery of low temperature waste heat.

One of the primary benefits of using the KCS11 for the recovery and conversion of industrial waste heat in comparison to the use of an organic Rankine cycle (ORC) is that the KCS11 can achieve a higher thermal efficiency than an ORC [22, 24, 25]. The thermal efficiency of the power producing cycle is of upmost importance in order to produce an economically viable system, and can be used as the primary evaluation parameter.

2.2.1 KCS11 First Law Analysis

Through the utilization of the first law of thermodynamics, the KCS11 can be evaluated to determine the optimum operating parameters based on the cycle boundary conditions.

The first law analysis of the KCS11 is carried out by applying an energy balance to the device being evaluated, and in order to simplify the analysis of the KCS11, standard thermodynamic assumptions have been applied to all of the cycle components. It is assumed that the changes in kinetic and potential energies are negligible throughout the cycle. The heat transfer to or from the various heat exchangers are defined as the change in the enthalpies of the working fluid. Also, the work required by the pump and the work produced by the turbine are calculated by the change in enthalpy of the working fluid across the device in question. The pressure reducing valve after the regenerator is assumed to be adiabatic, and so the enthalpy of the fluid is the same on both the inlet and exit side of the valve.

The KCS11 also utilizes a flow separator and a mixing chamber, and both of the devices are assumed to be adiabatic. The phase separator breaks the single saturated mixture flow into an ammonia rich saturated vapor stream that drives the cycle turbine and an ammonia weak saturated liquid stream that transfer its thermal energy to the working fluid entering the evaporator. The mixing chamber is referred to as an absorber, and the purpose of the absorber is to combine the two previously separated flows back into a single flow through an adiabatic mixing process. By recombining the flows together, the ammonia is absorbed back into the water and the mixture can be condensed into a saturated liquid state in the condenser.

The final piece of additional equipment in the KCS11 is the regenerator. The purpose of the regenerator is to preheat the fully combined working fluid before entering the evaporator. The regenerator is assumed to be adiabatic, so we can say that due to the first law the sum of the total energies entering the regenerator has to equal the sum of the total energies leaving the regenerator.

The regenerator has two separate fluid streams. The low temperature stream, from state three to state four, is the fully combined working fluid after leaving the cycle pump. The high temperature flow, from state seven to state eight, is the portion of working fluid that did not vaporize in the evaporator. The high temperature fluid enters the regenerator at the maximum temperature of the cycle. As the high temperature fluid flows through the regenerator, thermal energy is transferred to the low temperature stream. All heat exchangers have a pinch point which is defined as the smallest temperature difference between the two fluid flows in the heat exchanger. The pinch point in the regenerator is between the high temperature flow exit temperature and the low temperature flow inlet temperature. By setting a desired pinch point, the regenerator can be evaluated.

The pinch point value is restricted by the size of the regenerator, and by lowering the pinch point, the thermal efficiency of the KCS11 can be increased. The size of the regenerator, and in turn the effectiveness of the regenerator, is limited by the cost of the equipment in relation to the gain the equipment provides. Since this work is focused on the modeling of the various thermodynamic cycles, the pinch point for the regenerator, and all other heat exchangers evaluated, is set to 4 K. The pinch point value was decided in order to provide a temperature difference that was easily achievable.

The regenerator is evaluated using all standard thermodynamic assumptions. It is assumed that the regenerator is rigid and fully insulated, and that any change in the potential and kinetic energies is negligible. By applying these assumptions, the regenerator can be fully analyzed for the inlet and exit conditions of the fluid flows. The boundary conditions of the regenerator are based on the temperature, pressure, and the composition of the hot and cold streams. We know the composition of the flows since the cold temperature stream is the fully combined working fluid in route to the evaporator, and the hot temperature stream is the mass fraction of the total mixture that did not vaporize in the evaporator. The hot stream inlet temperature is at the maximum temperature of the cycle, and its exit temperature is equal to the cold stream inlet temperature plus the pinch point value of the regenerator. By knowing the inlet and exit temperatures, composition, and pressure of the hot stream, enthalpy of the stream can be calculated at the inlet and exit of the regenerator. The total amount of heat transfer out of the hot stream is equal to the product of the mass flow of the hot stream and the change in its enthalpy. Next, by applying the assumption that the regenerator is adiabatic we know that the heat transfer from the hot stream is equal to the heat transfer to the cold stream. Because we know the inlet temperature, the pressure, the composition, and the heat transfer to the cold stream, the exit temperature for the cold stream can be calculated.

In order to expedite the calculation and evaluation of the KCS11 under various boundary conditions, a software package was used. The software package that was chosen is EES, which is pronounced ease. The software package not only allowed for several thousand equations to be solved simultaneously, but the package has a large database of thermo-physical properties of various fluids. The software allowed for the speedy and accurate calculation of the various fluid properties, which are a function of temperature, pressure, and ammonia concentration. In order to calculate the properties of the aqueous-ammonia working fluid, EES uses a formulation by Ibrahim and Klein [26].

2.3 Organic Rankine Cycle (O.R.C.s)

A Rankine cycle is the primary thermodynamic cycle that is used to convert thermal energy to mechanical work, and in turn, electrical power. Generally, Rankine cycles are utilized when the thermal energy is at a high temperature such as when the thermal source is a boiler or nuclear reactor, but the Rankine cycle can also be used when the thermal source temperature is low.

By taking advantage of the properties of various fluids, the Rankine cycle can be operated at a lower temperature than would be necessary if using water as the working fluid. When water is replaced by a refrigerant or an organic fluid, the Rankine cycle is referred to as an organic Rankine cycle.

2.3.1 Organic Rankine Cycle First Law Analysis

The organic Rankine cycle studied in this work is a direct modification of a standard Rankine cycle. In order to analyze the cycle, standard thermodynamic assumptions are utilized for each of the cycle devices. The turbine and the pump are assumed to be internally reversible, and both of the heat exchangers are assumed to not have any loss. It is also assumed that there is not a change in the kinetic or potential energy of the working fluid throughout the cycle. The organic Rankine cycle can be studied by analyzing each of the components in the cycle.

The first step in the organic Rankine cycle is the exit side of the condenser. As the working fluid passes through the condenser, it is condensed in a constant temperature process to a saturated liquid at the sink temperature. The minimum pressure in the cycle is set based on the sink temperature so the fluid is a saturated liquid at the condenser exit. Once the working fluid leaves the condenser, it goes into the cycle pump. The cycle pump isentropically increases the pressure of the working fluid to the maximum cycle pressure. The working fluid then goes through the evaporator where it is vaporized into a super heated vapor. In a standard Rankine cycle, the evaporator is modeled as a boiler. The working fluid then goes through the turbine where it is isentropically expanded, causing the turbine to spin, to the minimum cycle pressure. The maximum cycle pressure is set based on the fact that the working fluid quality must be high when leaving the turbine. The quality has to be high to avoid damaging the turbine blades from excessive condensate formation. Once the working fluid leaves the turbine it goes into the condenser where the cycle is started again.

An organic Rankine cycle allows low temperature sources to be utilized for the production of electrical power. By using various working fluids and cycle pressures, the organic Rankine cycle can be optimized to maximize its thermal efficiency with standard cycle components.

2.4 Heat Pump/Refrigeration Cycles

The purpose of a heat pump is dependent on which side of the cycle you are evaluating and which side you most interested in. If cooling is desired, a refrigeration cycle is used to remove excess heat from a substance or area, and the evaporator is the most important component in the cycle. On the other hand, if heating is desired, a heat pump is used to efficiently increase the temperature of a substance or area, and the condenser is the most important component of the cycle.

Since a refrigeration cycle is nothing more than a heat pump cycle run in reverse, a heat pump can be utilized to solve several challenges when recovering waste heat from low temperature industrial sources. In fact, a heat pump cycle can recover all of the waste heat from a waste heat stream no matter how low the temperature of the stream, although there are economic and efficiency limits to the utilization of a heat pump for waste heat recovery. At the same time, a heat pump can be used to increase the temperature of the waste heat stream to a more useable level.

The utilization of a heat pump/refrigeration cycle as the cooling method for a waste heat stream provides certain benefits for the management and location of the industrial process that is generating the waste heat stream. The evaporator of a heat pump cycle can be controlled and kept at a constant temperature at all times. By providing a constant sink temperature for the waste heat stream, the industrial process can be optimized to run at the temperature of the cooling fluid stream. Or if the process needs a particular temperature to be the most efficient, the heat pump can be designed so that the return temperature of the waste heat stream will be at the preferred temperature without the fluctuations that are necessary when using the environment as the waste heat sink.

A heat pump also allows a process to be implemented where the average local temperatures are above acceptable limits. When the environment is used as the waste heat sink, the location has to be taken into account. If an industrial process requires that the cooling fluid is returned at a temperature lower than standard cooling methods, such as a water cooling tower, can achieve in a particular area, the industrial process has to be located to a different, cooler, location. Through the use of a heat pump, the industrial process can be implemented in any location since the waste heat cooling process' only interaction with the environment is at the higher temperature condenser.

The removal of the environment as a variable in the design of the process can be accomplished by cooling the heat pump with the KCS11 instead of the environment. Once the refrigeration cycle has recovered the waste heat and cooled the cooling fluid stream, the condenser of the refrigeration cycle can transfer the waste heat to the evaporator of the KCS11 at a higher temperature. Since the KCS11 is supplied the waste heat at a higher temperature than it would if it was used as the recovery/cooling method, the KCS11 can achieve a higher thermal efficiency.

2.4.1 Heat Pump/Refrigeration First Law Analysis

Heat pump/refrigeration cycles are well known and understood thermodynamic cycles. For this work, a standard vapor compression refrigeration cycle and two modified vapor compression refrigeration cycles were modeled using twelve different refrigerants to evaluate the range of performance that could be expected through the application of these cycles. The heat pump coefficient of performance was calculated for each cycle configuration and refrigerant at various boundary condition temperatures.

The standard vapor compression refrigeration cycle is the standard thermodynamic cycle used to convert work into thermal energy, and unlike the conversion of thermal energy to useable work, the conversion is complete with minimal loss. While it is recognized that there is always losses in any process, the isentropic efficiency for the compressor was not considered below 1.0 in this evaluation.

While the standard vapor compression refrigeration cycle is a simple and effective cycle for the cooling of the waste heat stream, it does not perform well over large temperature differences. The drawback to the standard vapor compression refrigeration cycle is that as the compressor increases the pressure of the refrigerant, it becomes more and more super heated. As the refrigerant is heated and compressed, the specific volume of the refrigerant also increases. As the specific volume of the refrigerant increases, it takes more and more work to continue to compress the fluid. The simplest way to reduce the work required to compress the refrigerant is to try and follow the saturated vapor line of the refrigerant’s saturation dome, or to use the vapor compression refrigeration cycle over small temperatures only.

Since it is not always possible, or desirable, to run a heat pump/ refrigeration cycle across a small temperature difference, a different solution has to be found to increase the performance of the cycle. One method for increasing the performance is to use two standard vapor compression refrigeration cycles in series. This configuration is called a cascade vapor compression refrigeration system. The cascade vapor compression refrigeration system has two distinct benefits over a standard vapor compression refrigeration cycle. First, the cascade system can better follow the saturated vapor line of the saturation dome which reduces the amount of work required to achieve the same maximum cycle temperature. Secondly, the cascade vapor compression refrigeration system is not limited to a single refrigerant. By using two different refrigerants, the system can be better designed and optimized for a particular situation.

While the cascade system provides several benefits to a standard vapor compression refrigeration cycle, it does have one particular drawback, the cycle temperatures have to overlap. In order for the waste heat to be transferred from the low temperature cycle to the high temperature cycle, the low temperature cycle’s condenser must be at a higher temperature than the high temperature cycle’s evaporator. This means that additional work has to be put into the low temperature cycle that is not utilized in the conversion of the waste heat into useable power.

Even though very little can be done to remove the temperature overlap in a cascade vapor compression refrigeration system, in instances when a single refrigerant is used, the two vapor compression refrigeration cycles can be combined. By combining the two cycles together, the heat exchanger that acts as the low temperature cycle’s condenser and the high temperature cycle’s evaporator can be replaced with a flash chamber. When this is done, the modified refrigeration cycle is called a multi-stage vapor compression refrigeration system.

A multi-stage vapor compression refrigeration system has two additional components that must be added. The first of the components is the flash chamber. The flash chamber is nothing more than a phase separator. The refrigerant leaves the condenser as a saturated liquid and is flashed to a middle pressure by the first expansion valve. Once the refrigerant goes through the expansion valve it enters the flash chamber where the saturated vapor is siphoned off and sent to a mixing chamber, which is the second additional required component for the multi-stage vapor compression refrigeration system. The saturated liquid portion of the refrigerant leaves the flash chamber and enters the second of the expansion valves where it is flashed to the minimum pressure of the system. The refrigerant then goes through the evaporator where it is evaporated to a saturated vapor before it is compressed. After the saturated vapor is compressed in the first of the cycle's two compressors, the refrigerant goes to the mixing chamber where it is recombined with the saturated vapor from the flash chamber. Once the two separate flows are recombined, the refrigerant is compressed in the second compressor and sent to the condenser. When a single refrigerant is used across a large temperature difference, the multi-stage vapor compression refrigeration system generally has a higher coefficient of performance than either a standard vapor compression refrigeration cycle or a cascade vapor compression refrigeration system.

2.5 Current Work Analysis Method

The current work uses the aforementioned vapor compression cycles in conjunction with a KCS11 to investigate the impact on the overall cycle thermal efficiency using waste heat temperature sources that are normally not available for KCS11 use. The heat pump augmented KCS11 thermal efficiencies are then compared to the thermal efficiencies of a non-augmented KCS11 and organic Rankine cycles. Because the thermal efficiency of the power cycles is the primary comparison value, the power cycles are evaluated using the same boundary conditions. By using consistent boundary conditions, variations in the cycles can be reduced and the thermal efficiency values from the different cycles can be compared directly. The same source and sink temperatures are used to evaluate all the power cycles. The vapor compression systems are evaluated using the same refrigerants. The source temperatures that are used for the power cycles are also used for the vapor compression systems.

Chapter 3

Results

3.1 Introduction

In order to appreciate the possibilities of combining a heat pump and a power cycle together, the performance of each cycle has to be studied. Since this work concentrated on studying the possible outcome of combining multiple ideal thermodynamic cycles, thermal efficiency and coefficient of performance plots for an organic Rankine cycle, KCS11, and various vapor compression refrigeration systems are used to determine and compare the output of the various cycles.

3.2 Organic Rankine Cycle

An organic Rankine cycle is a modified Rankine cycle where the only modification is replacing water as the working fluid with an organic fluid such as ammonia, a refrigerant, or a hydro-carbon. For this work, the organic Rankine cycle was analyzed using iso-butane, propane, and ammonia as the working fluid.

In order to calculate the thermal efficiency of the cycle, the maximum cycle pressure was increased while holding the condenser and evaporator temperatures constant. The maximum pressure was limited so that the turbine exit quality was no less than 90%. Figures 3.1 through 3.3 show the thermal efficiency of an organic Rankine cycle using ammonia as the working fluid with a sink temperature of 10°C, 17°C, and 25°C respectively. The *EES* code that was used to formulate the thermal efficiency plots can be found in appendix.

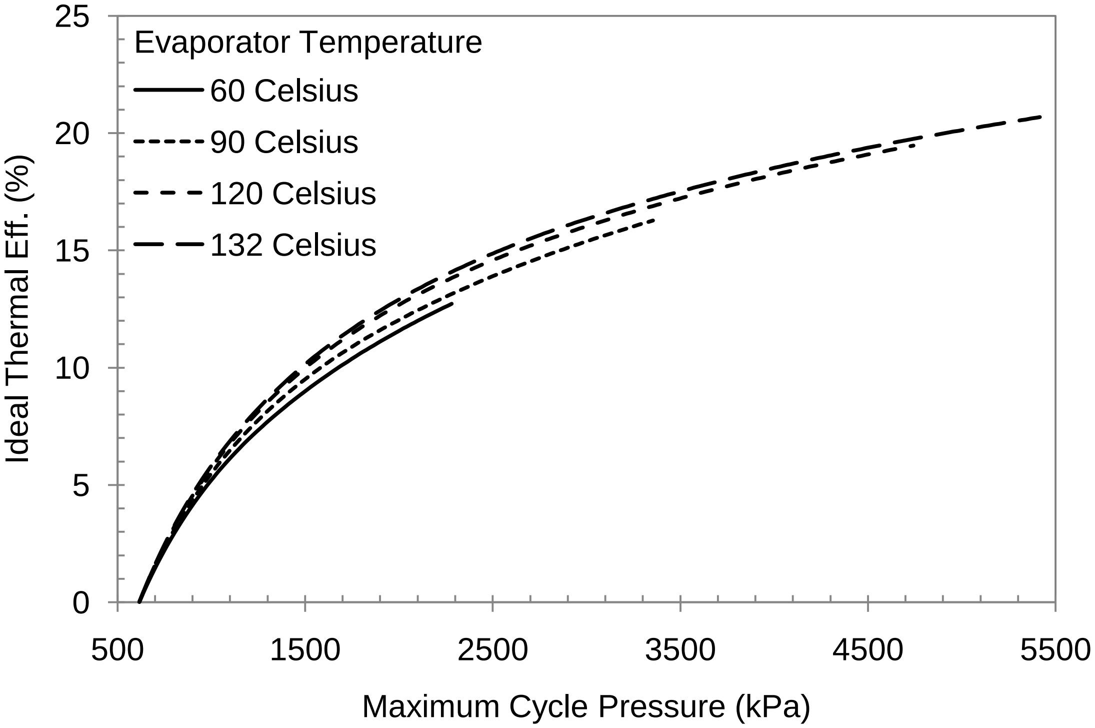

Figure 3.1: Thermal efficiency of NH3 ORC with a condenser temperature of 10°C

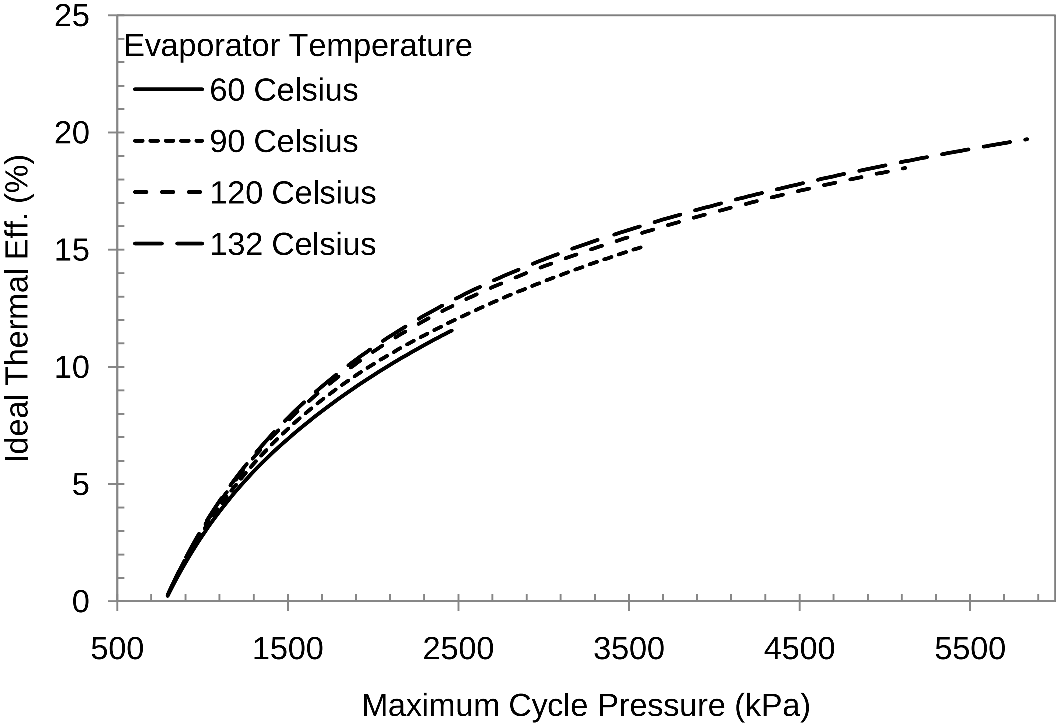

Figure 3.2: Thermal efficiency of NH3 ORC with a condenser temperature of 17°C

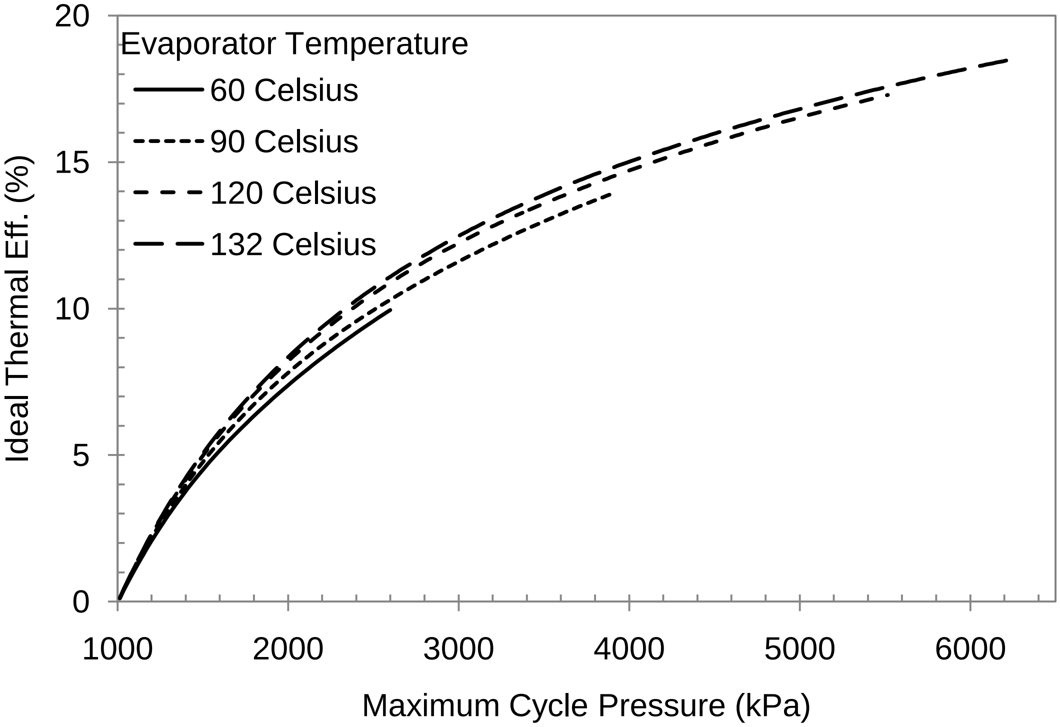

Figure 3.3: Thermal efficiency of NH3 ORC with a condenser temperature of 25°C

3.3 Kalina Cycle System 11 (KCS11)

As stated previously, the KCS11 is a modified Rankine cycle that replaces water as the working fluid with an aqueous ammonia mixture. By replacing the working fluid with a mixture instead of a pure substance, the KCS11 can take advantage of various properties of the mixture. The most important property of a mixture in relation to a pure substance, as far as the KCS11 is concerned, is that the mixture has a variable vaporization temperature. By utilizing the variable vaporization temperature, the concentration of the mixture can be set based on the boundary conditions of the system.

In order to automate the calculation of the thermal efficiency of the KCS11, *EES* was used to step through the mass fraction of ammonia in the working fluid from zero to one, while the maximum cycle pressure, source temperature, and sink temperature were all held constant. The KCS11 thermal efficiency was evaluated at several maximum cycle pressures, evaporator temperatures, and several condenser temperatures. Figures 3.4 through 3.7 show the thermal efficiency curves for the KCS11 as a function of the ammonia mass fraction for several evaporator temperatures. In the plots, the condenser temperature is set at 283K and the maximum pressure is 15 bar, 20 bar, 25 bar, and 30 bar respectively. The *EES* code that was used to calculate the thermal efficiency of the KCS11 can be found in appendix.

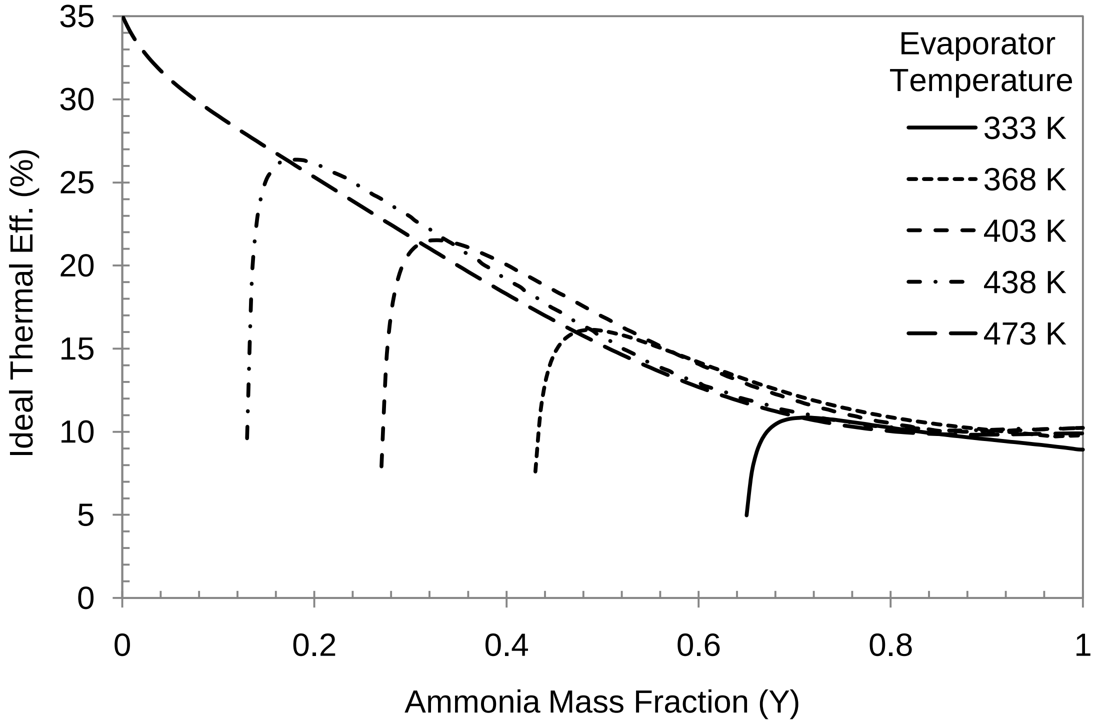

Figure 3.4: Thermal efficiency vs. Y for the KCS11 with a maximum pressure of 15 bar and a condenser temperature of 283K

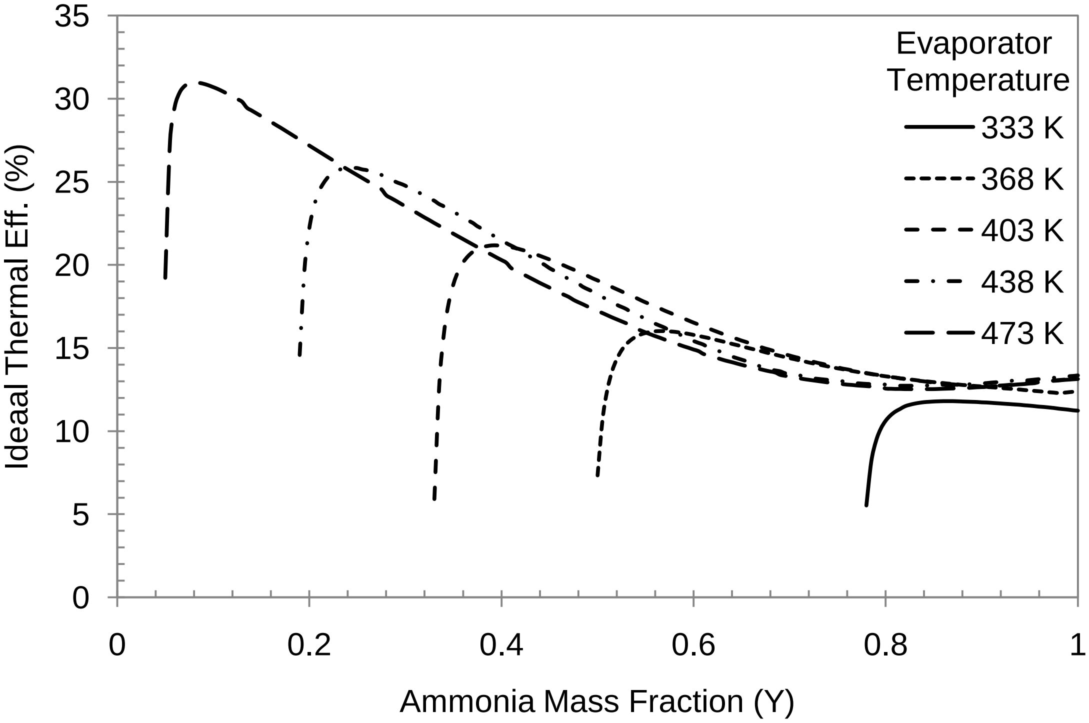

Figure 3.5: Thermal efficiency vs. Y of the KCS11 with a maximum pressure of 20 bar and a sink temperature of 283K

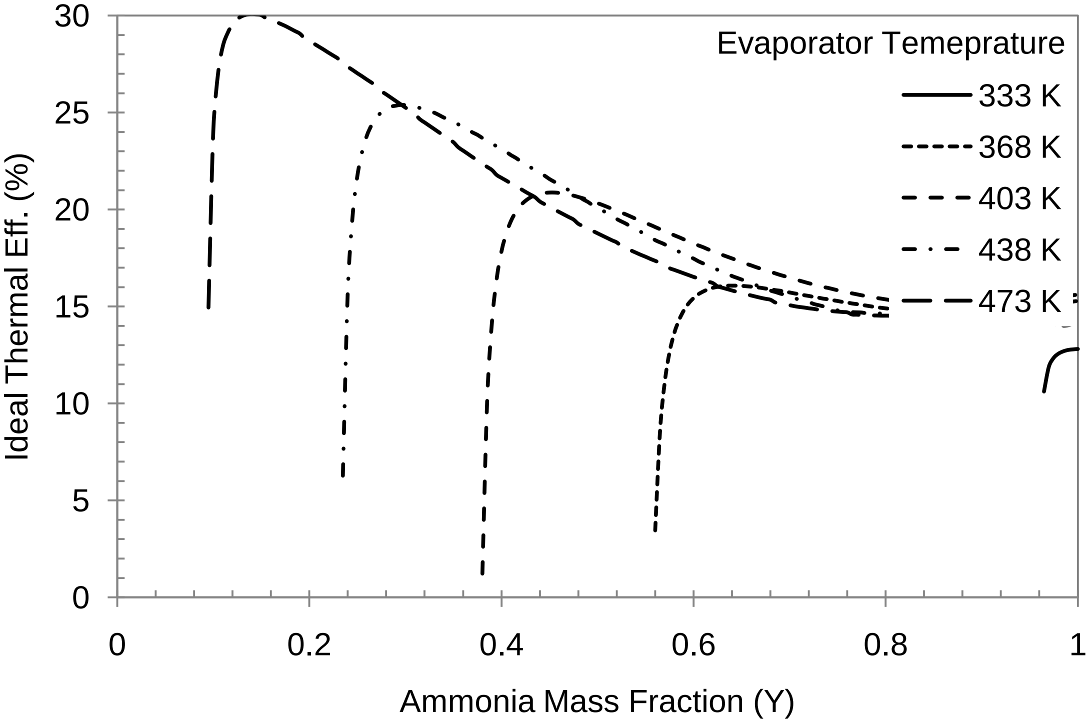

Figure 3.6: Thermal efficiency vs. Y of the KCS11 with a maximum pressure of 25 bar and a sink temperature of 283K

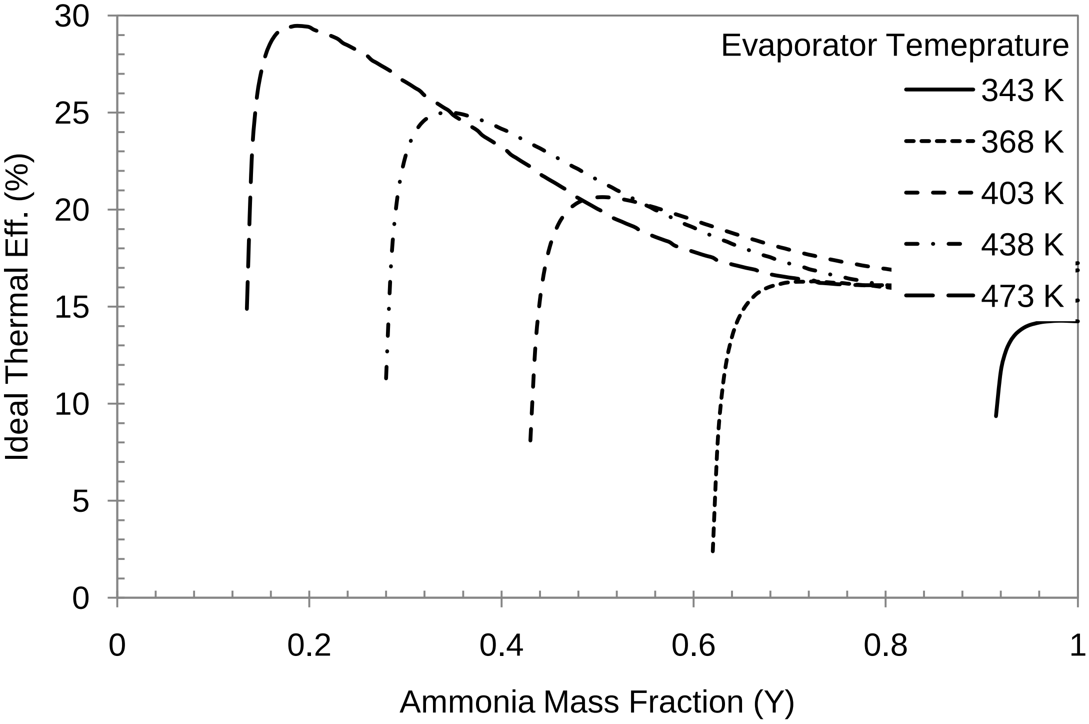

Figure 3.7: Thermal efficiency of the KCS11 with a maximum pressure of 30 bar and a sink temperature of 283K

As the previous plots show, the thermal efficiency of the KCS11 follows a couple trends that need to be considered when optimizing the system. The first trend is fairly obvious and expected, as the source temperature increases the thermal efficiency of the KCS11 increases also. The second, and most important trend, is that the maximum thermal efficiency in relation to the ammonia mass fraction is on an abrupt spike. If the working fluid mixture is too lean in relation to the ammonia mass fraction, the thermal efficiency drops rapidly, but if the working fluid mixture is a little rich in relation to the ammonia mass fraction, the thermal efficiency drops gradually as the ammonia mass fraction is increased. This indicates that for a KCS11 in operation the ammonia mass fraction of the working fluid would need to be rich to avoid a complete loss in the thermal efficiency of the cycle do to a mixing problem or leak.

3.4 Vapor Compression Refrigeration Cycles

The final cycle performances that need to be looked at are for the different vapor compression refrigeration cycles that were studied. Three different vapor compression refrigeration cycles were looked at in this work. The single stage vapor compression refrigeration cycle; the cascade configuration vapor compression refrigeration system, which combines two single stage vapor compression refrigeration cycles; and the double stage vapor compression refrigeration system.

All of the systems were evaluated using *EES* with twelve different refrigerants. The refrigerants that were used were chosen based on their critical point. All of the refrigerants have critical temperatures that are above 120°C except for R134a. R134a was also evaluated due to its widespread availability and common usage.

In order to calculate the coefficient of performance for the single stage vapor compression refrigeration cycle, the evaporator temperature was held constant, and the condenser temperature was increased by stepping through a temperature difference. The condenser temperature was calculated by adding the temperature difference to the evaporator temperature. The total temperature was increased until the condenser temperature was equal to the critical temperature of the refrigerant being evaluated.

For the cascade configuration and the double stage vapor compression refrigeration systems, the temperature difference between the evaporator and the condenser, the low temperature cycle evaporator and the high temperature cycle condenser for the cascade system, is stepped through in the same manner as the single stage vapor compression refrigeration cycle, but for the cascade and double stage vapor compression systems, there is an additional pressure parameter that has to be taken into account in order to calculate the coefficient of performance for the system. By utilizing the maximization function that is built into *EES*, the middle pressure can be used as a maximization variable at every temperature step to find the highest possible coefficient of performance. Figures 3.8 through 3.12 show the coefficient of performance plots for R-718. The evaporator temperature ranges from 0°C, 15°C, 30°C, 45°C and 60°C respectively. The *EES* codes that were used to calculate the coefficient of performance can be found in appendix.

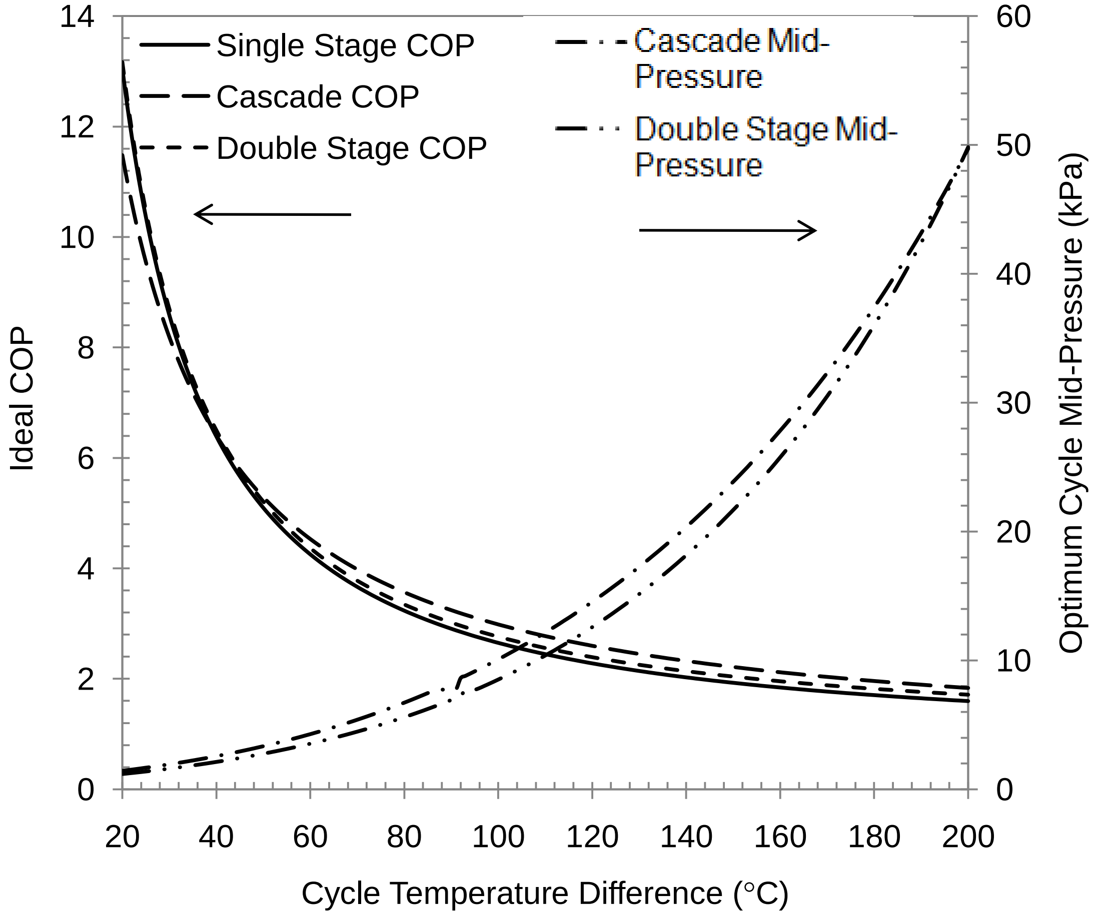

Figure 3.8: COP and the optimum cycle mid-pressure vs. temperature difference with R-718 for an evaporator temperature of 0°C

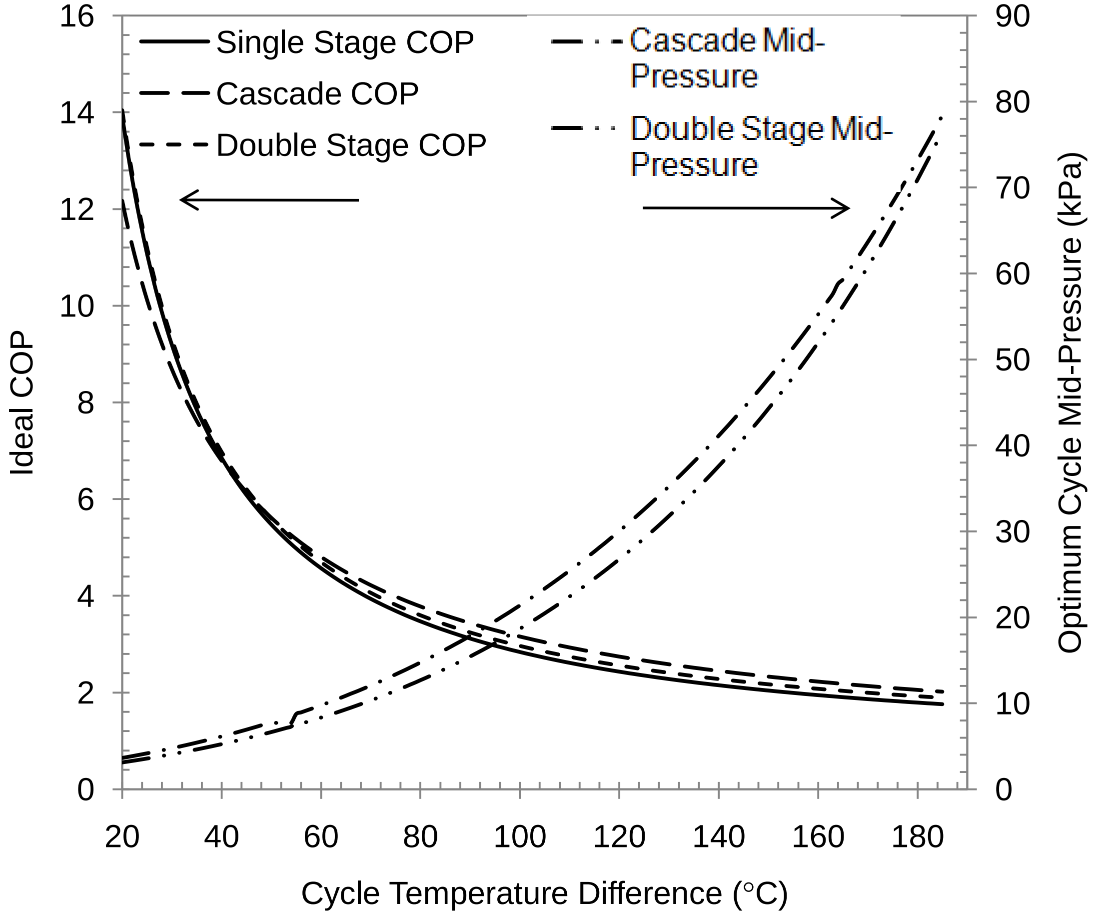

Figure 3.9: COP and optimum cycle mid-pressure vs. temperature difference with R-718 and an evaporator temperature of 15°C

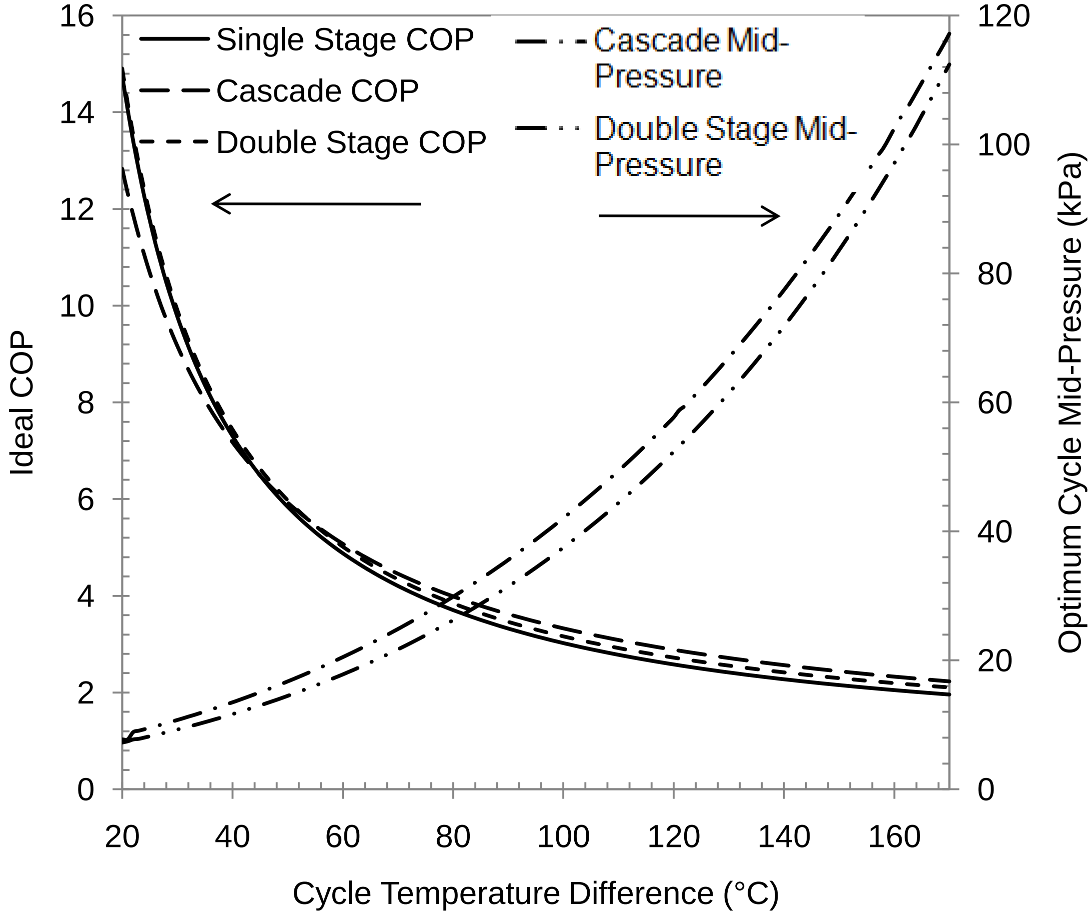

Figure 3.10: COP and optimum cycle mid-pressure vs. temperature difference with R-718 and an evaporator temperature of 30°C

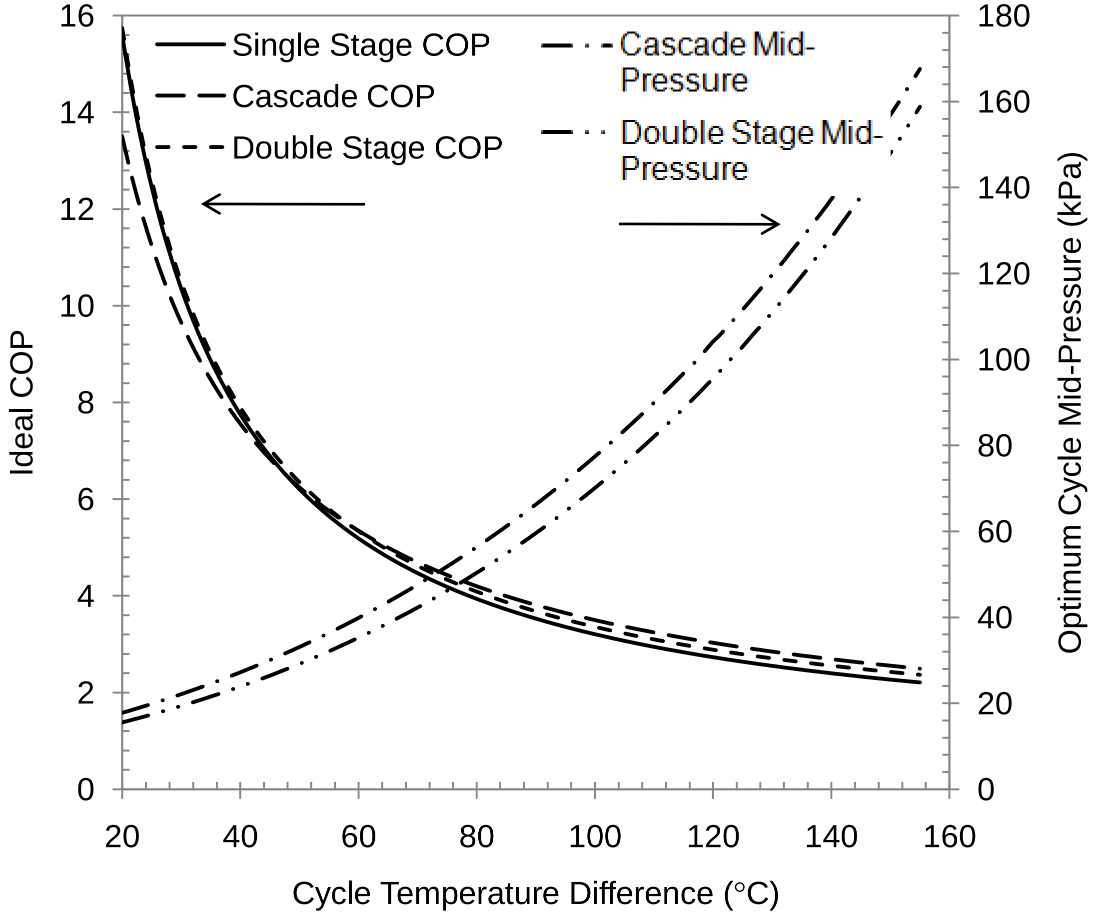

Figure 3.11: COP and optimum cycle mid-pressure vs. temperature difference with R-718 and an evaporator temperature of 45°C

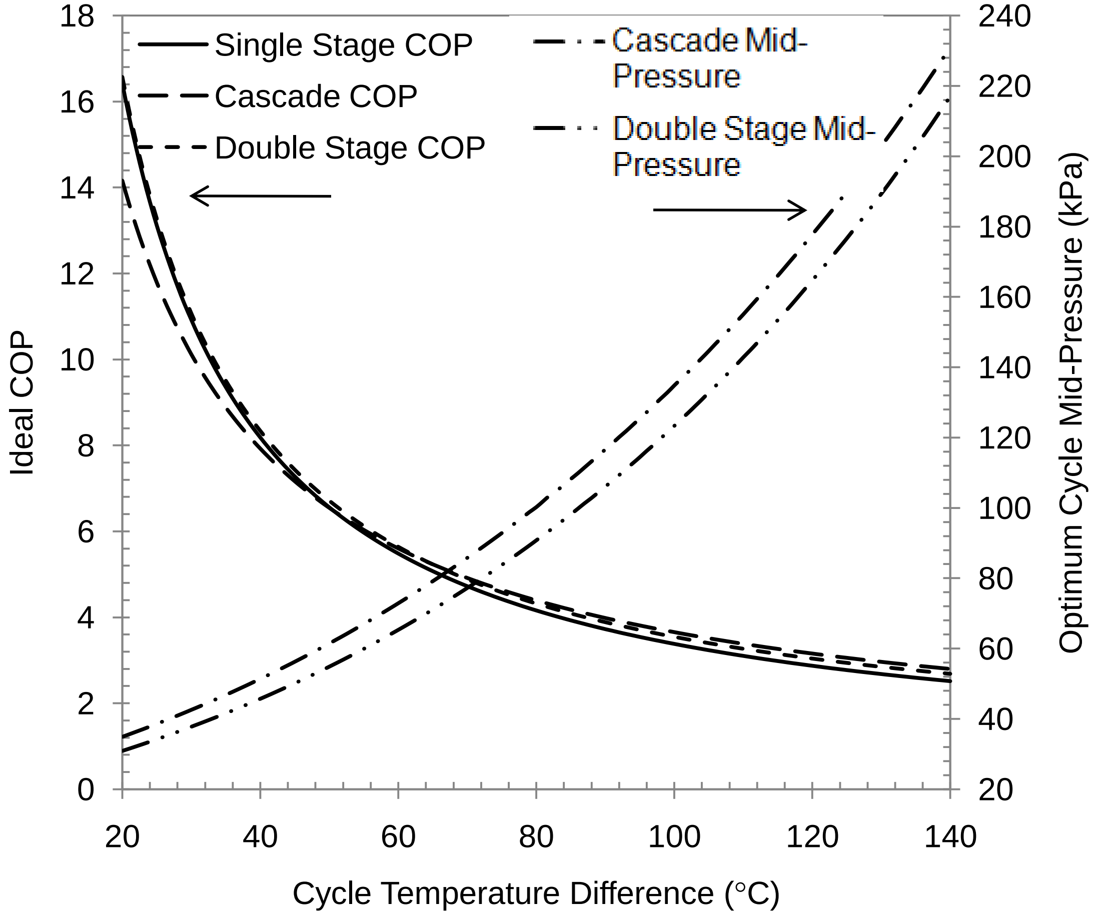

Figure 3.12: COP and optimum cycle mid-pressure vs. temperature difference with R-718 and an evaporator temperature of 60°C

3.5 Combined Systems

Now that the ground work has been laid by studying the individual systems, the various power systems can be compared. In the comparison of the systems, the primary area of interest is the thermal efficiency of the systems. By using the thermal efficiency as the measure of interest, the value of the systems can be directly related to one another since the goal of any power system is to convert thermal energy to mechanical work.

In the comparison, a couple values have to be set so that the power cycle outputs can be compared. First, there is an assumed pinch point of 4°C for all heat exchangers. This includes the evaporators, condensers, and the regenerator. Secondly, the power cycles were evaluated at the same source and sink temperatures. The only exception to the comparison is that the organic Rankine cycles are limited to lower source temperatures by their critical temperatures, where as the KCS11 is capable of operation at much higher temperature ranges than the organic Rankine cycles. Also, the heat pump augmented KCS11 is limited to source temperatures of 120°C or less. This is due to the temperature limitations of the aqueous ammonia equation of state that was used.

The pinch point value was necessary to avoid the assumption of ideal heat exchangers, and as well as in the calculation of the thermal efficiencies by setting the evaporator exit temperature, condenser exit temperatures, and the temperature difference in the regenerator of the KCS11. Due to the direct and significant affect the pinch point has on the thermal efficiency of the power cycle being evaluated, and since this work concentrated solely on the thermodynamic analysis, the same pinch point was used at all locations. The actual magnitude was chosen simply as a ‘realistic’ pinch point value for a conservatively sized fluid to fluid heat exchanger. A pinch point of 2K was used by [1].

When augmenting the KCS11 with a heat pump, there are several values of interest in addition to the thermal efficiency of the KCS11. First is the coefficient of performance of the heat pump cycle. For the purposes of this study, the coefficient of performance values that were considered is three, four, and five. Secondly, the source temperature, or the temperature of the waste heat stream, is extremely important in the augmentation of the KCS11.

The value of the source temperature and the coefficient of performance are closely linked when evaluating the vapor compression cycle. In most instances, as the source temperature increases, the temperature difference increases with the same coefficient of performance. This means that with an increase in the source temperature, the use of the vapor compression cycle will in turn create an even greater increase to the temperature of the KCS11 evaporator which will increase the thermal efficiency of the KCS11.

The following tables are used to compare the thermal efficiencies generated by this work of a KCS11, a heat pump augmented KCS11, and an organic Rankine cycle. All three power cycles were evaluated at various source and sink temperatures. The source temperatures represent the temperature of a waste heat stream that could be used to feed the power cycle. Tables 3.1 and 3.2 display the thermal efficiencies for the non-augmented KCS11, and the organic Rankine cycles. Tables 3.3 and 3.4 show the thermal efficiencies for the heat pump augmented KCS11.

Table 3.1 displays the thermal efficiencies based on the source temperature, sink temperature, the maximum cycle pressure, and the ammonia mass fraction. Table 3.2 shows the thermal efficiencies of the organic Rankine cycles based on the working fluid, source temperature, and the sink temperature. Table 3.3 and table 3.4 show the thermal efficiencies of a heat pump augmented KCS11. The heat pump augmented KCS11’s thermal efficiency is listed based on the source temperature, the heat pump condenser temperature, the sink temperature, the maximum cycle pressure, ammonia mass fraction, and the coefficient of performance of the heat pump.

The refrigerant used for the heat pump augmented KCS11 was R718 because it provided the largest temperature difference. The temperature difference across the augmenting heat pump is listed in table 3.5. The temperature difference is listed based on the source temperature, the coefficient of performance, the cycle configuration, and the cycle mid-pressure.

Table 3.1: Thermal efficiencies and ammonia mass fraction for KCS11 at various source and sink temperatures.

KCS11								
Source Temp. (°C)	Sink Temperature = 10°C							
	15 bar		20 bar		25 bar		30 bar	
	η_{th}	Y	η_{th}	Y	η_{th}	Y	η_{th}	Y
60	9.35	0.746	10.38	0.92	No Vapor			
80	12.25	0.589	12.56	0.687	13.11	0.799	13.78	0.911
100	15.46	0.481	15.34	0.558	15.37	0.63	15.56	0.71
120	18.64	0.385	18.34	0.457	18.13	0.517	18.01	0.576
140	21.58	0.303	21.19	0.366	20.88	0.423	20.63	0.475
160	24.39	0.218	23.86	0.285	23.16	0.315	23.15	0.384
180	27.29	0.129	26.62	0.195	26.05	0.251	25.65	0.296
200	32.01	0.021	29.62	0.101	28.79	0.159	28.26	0.207
Source Temp. (°C)	Sink Temperature = 17°C							
	15 bar		20 bar		25 bar		30 bar	
	η_{th}	Y	η_{th}	Y	η_{th}	Y	η_{th}	Y
60	0.761	88	8.56	0.913	No Vapor			
80	10.71	0.585	10.98	0.683	11.47	0.789	12.12	0.903
100	13.98	0.478	13.87	0.554	13.9	0.632	14.07	0.702
120	17.19	0.386	16.91	0.454	16.72	0.515	16.6	0.572
140	20.18	0.297	19.79	0.366	19.47	0.415	19.25	0.473
160	23.04	0.218	22.53	0.283	22.11	0.337	21.8	0.383
180	26.06	0.128	25.35	0.195	24.81	0.249	24.36	0.296
200	30.87	0.021	28.43	0.101	27.61	0.158	27.09	0.205
Source Temp. (°C)	Sink Temperature = 25°C							
	15 bar		20 bar		25 bar		30 bar	
	η_{th}	Y	η_{th}	Y	η_{th}	Y	η_{th}	Y
60	5.645	0.73	6.478	0.904	No Vapor			
80	8.928	0.582	9.167	0.677	9.606	0.78	10.2	0.894
100	12.27	0.474	12.18	0.55	12.21	0.624	12.36	0.695
120	15.53	0.383	15.27	0.451	15.09	0.511	14.98	0.572
140	18.57	0.3	18.19	0.365	17.9	0.419	17.67	0.469
160	21.51	0.216	21.02	0.278	20.6	0.334	20.27	0.38
180	24.66	0.127	23.9	0.194	23.35	0.248	22.9	0.295
200	29.55	0.021	27.08	0.1	26.28	0.157	25.7	0.204

Table 3.2: Thermal efficiencies of an organic Rankine cycle at various source and sink temperatures

ORC			
Source Temp. (°C)	Sink Temperature = 10°C		
	iso-butane	propane	ammonia
60	11.39	11.64	11.62
80	15.07	15.32	14.07
100	17.98	17.81	16.36
120	20.27		18.51
140	Above Critical Temperature		
160			
180			
200			
Source Temp. (°C)	Sink Temperature = 10°C		
	iso-butane	propane	ammonia
60	9.667	9.924	10.12
80	13.56	13.86	12.92
100	16.64	16.55	15.27
120	19.04		17.47
140	Above Critical Temperature		
160			
180			
200			
Source Temp. (°C)	Sink Temperature = 25°C		
	iso-butane	propane	ammonia
60	7.635	7.889	7.93
80	11.82	12.16	11.56
100	15.09	15.1	13.99
120	17.65		16.27
140	Above Critical Temperature		
160			
180			
200			

Table 3.3: Thermal efficiencies of a heat pump augmented KCS11 for various source and sink temperatures. (COP equal to 3 and 4)

Heat Pump Augmented KCS11										
Sink Temperature = 10°C										
Source Temp. (°C)	HP T_{cond} (°C)	COP=3 / η_{BE}=33.3%				HP T_{cond} (°C)	COP=4 / η_{BE}=25%			
		P_{MAX} (bar)	η_{TH}	η_{NET}	Y		P_{MAX} (bar)	η_{TH}	η_{NET}	Y
60	183	15	27.8	-5.5	0.11	145	10	22.8	-2.2	0.2
80	211	25	30.3	-3	0.11	171	15	25.9	0.92	0.17
100	239	45	32.2	-1.1	0.13	197	20	29.1	4.12	0.11
120	266	65	34.3	0.96	0.11	222	30	31.5	6.48	0.1
Sink Temperature = 17°C										
Source Temp. (°C)	HP T_{cond} (°C)	COP=3 / η_{BE}=33.3%				HP T_{cond} (°C)	COP=4 / η_{BE}=25%			
		P_{MAX} (bar)	η_{TH}	η_{NET}	Y		P_{MAX} (bar)	η_{TH}	η_{NET}	Y
60	183	15	26.5	-6.8	0.11	145	10	21.5	-3.5	0.2
80	211	25	29.3	-4.1	0.11	171	15	24.4	-0.6	0.15
100	239	45	31.2	-2.2	0.13	197	20	27.9	2.93	0.12
120	266	65	33.3	-0	0.11	222	30	30.4	5.43	0.1
Sink Temperature = 25°C										
Source Temp. (°C)	HP T_{cond} (°C)	COP=3 / η_{BE}=33.3%				HP T_{cond} (°C)	COP=4 / η_{BE}=25%			
		P_{MAX} (bar)	η_{TH}	η_{NET}	Y		P_{MAX} (bar)	η_{TH}	η_{NET}	Y
60	183	15	25.2	-8.2	0.11	145	10	19.9	-5.1	0.2
80	211	25	27.9	-5.4	0.11	171	15	23.2	-1.8	0.17
100	239	45	29.9	-3.4	0.13	197	20	26.6	1.57	0.11
120	266	65	32.1	-1.2	0.11	222	30	29.1	4.14	0.1

Note: P_{MAX} is limited to no less than 10 bar and Y (ammonia mass fraction) is limited to no less than 0.1. η_{BE} is the break even efficiency for the prescribed coefficient of performance. $\eta_{NET} = \eta_{TH} - \eta_{BE}$

Table 3.4: Continued thermal efficiencies of a heat pump augmented KCS11 (COP equal to 5)

Heat Pump Augmented KCS11 continued					
Sink Temperature = 10°C					
Source Temp. (°C)	HP T_{cond} (°C)	$COP=5\backslash\eta_{BE}=20\%$			
		P_{MAX} (bar)	η_{TH}	η_{NET}	Y
60	124	10	19.7	-0.3	0.29
80	149	10	22.1	2.14	0.15
100	173	15	26.3	6.26	0.16
120	197	20	29.1	9.12	0.11
Sink Temperature = 17°C					
Source Temp. (°C)	HP T_{cond} (°C)	$COP=5\backslash\eta_{BE}=20\%$			
		P_{MAX} (bar)	η_{TH}	η_{NET}	Y
60	124	10	18.2	-1.8	0.28
80	149	10	22.1	2.05	0.18
100	173	15	25	4.96	0.16
120	197	20	27.9	7.93	0.12
Sink Temperature = 25°C					
Source Temp. (°C)	HP T_{cond} (°C)	$COP=5\backslash\eta_{BE}=20\%$			
		P_{MAX} (bar)	η_{TH}	η_{NET}	Y
60	124	10	16.4	-3.6	0.3
80	149	10	20.5	0.52	0.18
100	173	15	23.6	3.55	0.16
120	197	20	26.6	6.57	0.11

Note: P_{MAX} is limited to no less than 10 bar and Y(ammonia mass fraction) is limited to no less than 0.1. η_{BE} is the break even efficiency for the prescribed coefficient of performance. $\eta_{NET} = \eta_{TH} - \eta_{BE}$

Table 3.5: Temperature difference, mid-pressure, and configuration of augmentation heat pump using R-718 in cascade operation

Source Temp (°C)	COP	Temperature Difference (°C)	Mid-Pressure (kPa)
60	3	127	170
	4	89	99
	5	68	71
80	3	135	360
	4	95	216
	5	73	158
100	3	143	699
	4	101	426
	5	77	315
120	3	150	1253
	4	106	773
	5	81	579

3.6 Case Study

While the thermal efficiency results of the organic Rankine cycle, the Kalina Cycle System 11, and the heat pump augmented Kalina Cycle System 11 are discussed in the previous sections, a specific case study can help to determine the validity of a combined cycle in comparison to power cycles that are individually implemented. For this case study, the source and sink temperatures were set at 60°C and 10°C respectively. With source and sink temperatures set, we can look at the possible net thermal efficiency with a combined cycle based on the coefficient of performance, and compare that to the individual cycle efficiency for the specified temperatures.

With the boundary condition temperatures set, the remainder of the specifications for the cycles can be decided on. R718 is used as the refrigerant in a cascade vapor compression cycle to maximize the temperature increase across the cycle. The maximum cycle pressure in the KCS11 is set to 15 bars, and a pinch point of 4°C is applied to all of the heat exchangers in evaluating the cycles.

The coefficient of performance for the cascade vapor compression system is stepped down from 20 to 2. With the coefficient of performance known, the condenser exit temperature for the high temperature cycle can be calculated. The evaporator temperature for the KCS11 is set to be the exit temperature of the high temperature condenser minus the pinch point. Since multiple cycles are interacting, a waste heat value of 1 MW is used to calculate the thermal efficiency and the net thermal efficiency of the combined cycles.

Table 3.6 shows the values used to compare the combined cycle based on the coefficient of performance. The fist column lists the coefficient of performance of the cascade vapor compression system. The heat released by the vapor compression system is given in the second column, which is equal to the waste heat value plus the work of the vapor compression system. The third column is the temperature of the condenser followed by the temperature of the KCS11 evaporator. The following columns show the optimum ammonia mass fraction for the KCS11, the thermal efficiency of the KCS11, the break even thermal efficiency, the amount of excess work produced by the combined cycle, and then the net thermal efficiency of the combined cycle. The thermal efficiency of the KCS11 is the ratio between the work produced by the turbine minus the work required for the pump divided by the amount of heat added to the KCS11. The break even thermal efficiency is found by determining what thermal efficiency is required so that the work produced by the KCS11 is the same amount of work is required by the cascade vapor compression system. The work out column is found by subtracting the work required by the pump and the cascade vapor compression system from the work produced by the turbine. Then the net thermal efficiency of the combined cycle is found by dividing the excess work of the combined cycle by the magnitude of the waste heat added to the combined cycle.

Since the thermal efficiency is the variable used to compare the cycles, we can take the case from table 3.6 with the highest net thermal efficiency and compare that to the Carnot efficiency and the thermal efficiency of an individually implemented KCS11 and ORC. With a coefficient of performance of 20, the combined cycle has a net thermal efficiency of 5.82%, while a KCS11 can achieve a thermal efficiency of 10.38%. An ORC can achieve a thermal efficiency of 11.64% when propane is used as the working fluid. This shows that the combined cycle cannot achieve a net thermal efficiency that is possible for the individually implemented power cycles.

Table 3.6: Case study for a source temperature of 60°C and a sink temperature of 10°C

COP	Q_{out} (kJ)	T_{cond} (°C)	T_{evap} (°C)	Y Optimum	η_{th} (%)	η_{BE} (%)	W_{out} (kJ)	η_{NET} (%)
20	1053	68.7	64.7	0.67	10.56	5.03	58.20	5.82
19	1055	69.4	65.4	0.66	10.67	5.21	57.57	5.76
18	1059	70.6	66.6	0.65	10.84	5.57	55.80	5.58
17	1063	71.7	67.7	0.64	11	5.93	53.93	5.39
16	1067	72.9	68.9	0.64	11.18	6.28	52.29	5.23
15	1072	74.4	70.4	0.63	11.41	6.72	50.32	5.03
14	1077	75.9	71.9	0.62	11.63	7.15	48.26	4.83
13	1083	77.8	73.8	0.61	11.92	7.66	46.09	4.61
12	1091	80.1	76.1	0.59	12.27	8.34	42.87	4.29
11	1100	82.8	78.8	0.57	12.7	9.09	39.70	3.97
10	1110	85.8	81.8	0.56	13.17	9.91	36.19	3.62
9	1125	90	86	0.52	13.64	11.11	28.45	2.85
8	1143	94.9	90.9	0.51	14.63	12.51	24.22	2.42
7	1166	101.4	97.4	0.47	15.69	14.24	16.95	1.69
6	1201	111	107	0.43	17.24	16.74	6.05	0.61
5	1251	124	120	0.37	19.25	20.06	-10.18	-1.02
4	1334	144.4	140.4	0.29	22.19	25.04	-37.99	-3.80
3	1501	182.5	178.5	0.12	27.66	33.38	-85.82	-8.58

Chapter 4

Discussion

4.1 Overview

Economics, due to the second law of thermodynamics, has always been one of the limiting factors for the recovery of low temperature waste heat streams. The possible amount of thermal energy that can be recovered from any given waste heat stream is limited by the Carnot efficiency, and that is for a completely reversible heat recovery process. Due to the limitations placed on the recovery of waste heat by the temperature of the waste heat stream and the thermal sink, methods of waste heat recovery are needed that can operate as efficiently as possible at low temperatures, and can possibly increase the temperature difference between the waste heat stream and the thermal sink used by the waste heat recovery process.

This work concentrated on studying the viability of utilizing a Kalina cycle system 11 with heat pump augmentation to produce useable power from low temperature waste heat streams. The Kalina cycle, because of its use of an aqueous ammonia solution as the working fluid, is able to operate within a greater range of temperatures than other low temperature recovery methods such as an organic Rankine cycle. By implementing a heat pump as an intermediate cycle between the waste heat stream and the Kalina cycle, the waste heat can be recovered at temperatures that are lower than standard cooling methods, such as a water cooling tower, can achieve. The temperature of the waste heat can also be increased before it is transferred into the Kalina cycle.

The model results for the heat pump augmented Kalina cycle were compared to model results for a non-augmented Kalina cycle and an organic Rankine cycle. All cycles were evaluated to a maximum temperature value. The organic Rankine cycle's maximum temperature was limited to the critical temperature of the working fluid being evaluated. The non-augmented KCS11 was limited to a maximum temperature of 200°C. The heat pump augmented KCS11 was limited to waste heat streams of 120°C or less.

In comparing the non-augmented KCS11 to an organic Rankine cycle, the tabulated thermal efficiencies show that at the lowest temperatures evaluated, the organic Rankine cycle operates at higher thermal efficiencies than the KCS11. For example, for a waste heat stream at 80°C, an organic Rankine cycle using propane as the working fluid has a thermal efficiency of 15.3%, while a non-augmented KCS11 can have a thermal efficiency of 13.8%. Based on these values, the organic Rankine cycle has an advantage at the lower temperature values evaluated by this work. While the organic Rankine cycles showed an advantage at the lowest temperature values considered, the KCS11 was able to close the performance gap in the middle temperature values. At 120°C, the KCS11 could have a thermal efficiency value of 18.6%, while an organic Rankine cycle using ammonia as the working fluid has a thermal efficiency value of 18.5%. Based purely on the thermal efficiency comparison between a non-augmented KCS11 and an organic Rankine cycle, for extremely low temperature waste heat streams in the range of 60°C to 120°C, the organic Rankine cycle is the preferred method of thermal energy conversion. In temperature ranges from 120°C to 200°C, the KCS11 provides exceptional thermal efficiencies for the conversion of waste heat to useable power.

While the organic Rankine cycle has an advantage to the non-augmented KCS11 at the lower portion of the temperature range used in this work, by augmenting the KCS11 with a vapor compression refrigeration cycle, a heat pump, the temperature of the waste heat can be increased so that the thermal efficiency gains of the KCS11 can be realized. For example, if we take a waste heat stream of 80°C and use it to compare a heat pump augmented KCS11 to an organic Rankine cycle, the organic Rankine cycle using propane as the working fluid has a thermal efficiency of 15.3%, but a heat pump augmented KCS11 has a thermal efficiency of 30.3% using a heat pump with a coefficient of performance of 3.

At first glance, the heat pump augmented KCS11 is the obvious choice in low temperature waste heat recovery, but the thermal efficiency of the power cycle does not give the whole picture. The problem with using a vapor compression refrigeration cycle to increase the temperature of the waste heat stream is the work required to operate the cycle. For a heat pump that has a coefficient of performance of 3 to break even, the power cycle it is supplying has to have a thermal efficiency of 33%. That means that if the power cycle does have a thermal efficiency of 33%, all of the power output of the power cycle is being used to drive the vapor compression cycle. The vapor compression cycle work is free, but none of the waste heat stream is actually being converted into useable energy. The waste heat is being dumped into the heat sink used by the power cycle. So to find the combined thermal efficiency for the heat pump augmented KCS11we have to subtract the breakeven thermal efficiency from the thermal efficiency of the KCS11.

What this indicates is that in order to implement heat pump augmentation based purely on the thermal efficiency; the power cycle being augmented needs to have a very high thermal efficiency, the coefficient of performance of the vapor compression cycle needs to be high, or a combination of a high thermal efficiency with a high coefficient of performance are required. So now if we compare the waste heat recovery from the same 80°C waste heat stream we can get a better idea of what method might be preferred. For the non-augmented KCS11, we can obtain a thermal efficiency of 12.6%, the organic Rankine cycle using propane as the working fluid can have a thermal efficiency of 15.3%, and the heat pump augmented KCS11 can achieve a thermal efficiency of 22.1% using a vapor compression cycle that has a coefficient of performance of five. With the heat pump augmented KCS11 though; we need to subtract the breakeven thermal efficiency, which for a coefficient of five is 20%. Once we subtract the breakeven thermal efficiency, we find that only 2.1% of the waste heat is being converted into useable power.

Based on the thermal efficiency trends of the models tested in this work, the ideal cycle for low temperature waste heat recovery is the organic Rankine cycle with the KCS11 being a suitable replacement at temperatures above the critical temperature of the fluids used in an organic Rankine cycle. The amount of work required by the vapor compression refrigeration cycle to increase the waste heat temperature to an acceptable level is simply too high.

4.2 Future Work Recommendations

Based on previous work that has been done [1, 6, 9, 11, 14], the Kalina cycle can operate at higher efficiencies than an organic Rankine cycle is able to achieve. While this work was based on ideal thermodynamic cycles, it is theorized that the organic Rankine cycle would suffer a greater loss in thermal efficiency due to losses in the cycle turbine and pump than a KCS11 would suffer. This is because only a small portion, about 15%, of the mass flow in the KCS11 passes through the cycle turbine. In order to obtain a more accurate comparison of the various power cycles, small scale testing of the various power cycles is needed.

The thermal efficiency comparison of the power cycles only shows a portion of the solution to any given problem. In order to fully rule out the use of heat pump augmentation of a power cycle, a more detailed study of the economics and the individual applications; such as the process location environment, the cycle boundary conditions, and current waste heat disposal methods are needed.

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Appendix

The appendix is used to provide the *EES* codes that were used to calculate the results investigated in this work. For each code that is provided, a description of how the code was used is also provided in addition to the notes written in the code.

A.1 Organic Rankine Cycle Code

The organic Rankine cycle was the easiest to code since it follows a simple four stage Rankine cycle. In order to calculate the thermal efficiency of the cycle; a working fluid, source temperature, sink temperature, and pinch point temperature need to be set. Once the boundary conditions are set, a parametric table is created with the desired maximum pressure range in the first column of the table. All other desired parameters are set to additional columns in the table. Once the solve table command is used, *EES* runs the code for each pressure value from the parametric table. In other words, the software runs the code for each row of the table. The remaining columns that are set up display the corresponding information of that run.

```
                    {Organic Rankine Cycle}
{The function Therm_Eff is used to limit the output of the code to the range
desired based on the critical pressure and the turbine outlet quality.}
        Function Therm_Eff(P_test,P2,w_turbine,qh,x4)
        If (P_test < P2) OR (x4 < 0.9) Then Therm_Eff := 0
        Else Therm_Eff := (w_turbine/qh)*100
        End

P$ = 'Ammonia'        {P$ is the variable used to call the correct working fluid.}

T_source = 140 [C]  {Temperature of the heat pump condensor or cooling fluid
                     flow.}
T_sink     = 25 [C]  {Temperature of the heat sink used for the power cycle
                     condensor.}
T_pinch   = 4 [C]    {T_pinch is the pinch point for all heat exchnagers.}
{P_max     = 2500}   {P_max is an independent variable to find maximum
                     efficiency.}

Pcrit = P_crit(P$)   {Pcrit is the critical pressure for the fluid. Pcrit is used to limit
                     P_max.}

ETA_Pump     = 1     {Isentropic efficiency for the pump.}
ETA_Turbine = 1      {Isentropic efficiency for the turbine.}
```

```
T1 = T_sink + T_pinch       {The temperature leaving the condenser is the sink
                            temperature plus the pinch point.}
x1  = 0                     {The fluid is a saturated liquid as it leaves the
                            condenser.}
P1 = Pressure(P$,T=T1,x=x1)
h1  = Enthalpy(P$,T=T1,x=x1)
s1  = Entropy(P$,T=T1,x=x1)
v1  = Volume(P$,T=T1,x=x1)

w_pump = (v1*(P2-P1))/ETA_Pump      {Pump work is found assuming the
                                    pump is isentropic.}

P2 = P_max                  {P_max is stepped through in the loop.}
h2  = h1 + w_pump
T2 = Temperature(P$,P=P2,h=h2)

P3 = P2
T3 = T_source - T_pinch     {Maximum cycle temperature is set by the source
                            temperature minus the pinch point.}
P_test = P_sat(P$,T=T3)     {P_test is used to insure that P_max does not exceed
                            the saturation pressure for the maximum cycle
```

```
                              temperature.}
h3  = Enthalpy(P$,T=T3,P=P3)
s3  = Entropy(P$,T=T3,P=P3)

s4    = s3                    {Assuming isentropic expansion in the turbine.}
P4   = P1
h4s  = Enthalpy(P$,P=P4,s=s4)
h4   = h3 - (h3 - h4s)*ETA_Turbine     {Actual enthalpy after the turbine based
                                        on the isentropic efficiency of the
                                        turbine.}
T4  = Temperature(P$,P=P4,h=h4)
x4   = Quality(P$,P=P4,h=h4)

w_turbine = h3 - h4
qh          = h3 - h2

ETA_th = Therm_Eff(P_test,P2,w_turbine,qh,x4)      {Thermal efficiency is
                                                    output as a percentage.}
```

A.2 KCS11 Code

The code for the KCS11 is considerably more complicated than the organic Rankine cycle code. This is partially because the KCS11 has an additional independent variable, the ammonia mass fraction, but also because the KCS11 has nearly three times the number of state points that must be evaluated. In order to use the supplied KCS11 code, a number of parameters have to be set. The maximum cycle pressure, source temperature, sink temperature, and the pinch point for the heat exchangers have to be set. A parametric table is then generated with the mass fraction of ammonia in the working fluid, Y, is stepped through from zero to one in the first column. Additional columns can be setup to display the desired information found when the code is run.

```
                    {Kalina Cycle System 11 (KCS11)}
{This EES code is to calculate the efficiency of the Kalina Cycle System 11. The
heat exchangers and the regenerator are considered adiabatic and have the
prescribed pinch point applied.}

{The function THEFF is used to limit the output of the code to prevent displaying
negative thermal efficiencies.}
FUNCTION THEFF(w_net,qh)
IF (w_net<=0) OR (qh<=w_net) THEN
THEFF := 0;
```

```
ELSE
THEFF := (w_net/qh)*100
ENDIF
END

{The function Qu_check is needed when calculating the thermal efficiency of the
KCS11 at the extremes of its pressure range based on the source temperature.}
FUNCTION Qu_check(Qu)
IF (Qu<=0) THEN
Qu_check := 0
step1 := 0
 ELSE
 step1 := Qu
ENDIF
IF (Qu>=1) THEN
Qu_check := 1
step2 := 0
ELSE
step2 := Qu
ENDIF
IF (step1=step2) THEN Qu_check := Qu ELSE a := 0
END
```

```
T_source = 470 [K]           {Temperature of the waste heat flow or source.}

T_sink      = 298 [K]        {Temperature of the heat sink of the cycle.}

T_pinch    = 4 [K]           {Pinch point of all of the heat exchangers.}

P_max      = 20 [bar]        {The maximum pressure in the cycle.}


ETA_pump = 1        {Isentropic efficiency of the pump. ETA_pump <=1}

ETA_turb    = 1     {Isentropic efficiency of the turbine. ETA_turb <=1}


{Y = .5203}   {Y is the mass fraction of the ammonia in the total mixture. The

              value is input in a table.}


P1 = P2; x1 = Y;             {State 1 is before the condenser.}

h1 = w*h9 + (1-w)*h10;

Call NH3H2O(234, P1, x1, h1: T1, P_1, x_1, h_1, s1, u1, v1, Qu1)


Qu2 = 0; T2 = T_sink + T_pinch;

 x2 = x1                     {State 2 is leaving the sondenser.}

Call NH3H2O(138, T2, x2, Qu2: T_2, P2, x_2, h2, s2, u2, v2, Qu_2)
```

```
w_pump = (v2*(P_max - P2)*100)/ETA_pump   {Multiplying by 100 converts the
                                           pressure from bars to kPa.}

P3 = P_max; x3 = x2; h3 = h2 + w_pump;
Call NH3H2O(234, P3, x3, h3: T3, P_3, x_3, h_3, s3, u3, v3, Qu3)


h4 = q_regen + h3   {h4 is found by assuming the change in the enthalply in the
                     cold fluid in the regenerator is equal to the change in the
                     enthalpy of the hot fluid stream.}
P4 = P3; x4 = x3;
Call NH3H2O(234, P4, x4, h4: T4, P_4, x_4, h_4, s4, u4, v4, Qu4)

T5 = T_source - T_pinch;
x5 = x4; P5 = P4;
Call NH3H2O(123, T5, P5, x5: T_5, P_5, x_5, h5, s5, u5, v5, Qu5)

w = 1 - Qu_check(Qu5)      {This is the fraction of the total mass that does NOT
                            vaporize and passes through the regenerator.}

T6 = T5; P6 = P5; Qu6 = 1;
Call NH3H2O(128, T6, P6, Qu6: T_6, P_6, x6, h6, s6, u6, v6, Qu_6)
```

```
T7 = T5; P7 = P5; Qu7 = 0;
Call NH3H2O(128, T7, P7, Qu7: T_7, P_7, x7, h7, s7, u7, v7, Qu_7)


T8 = T3 + 4 [K];        {The exit temperature for the regenerator is set to 4 higher
                        than the condenser temp due to a pinch point.}
P8 = P7; x8 = x7;
Call NH3H2O(123, T8, P8, x8: T_8, P_8, x_8, h8, s8, u8, v8, Qu8)

q_regen = w*(h7 - h8)

h9 = h8; P9 = P1; x9 = x8;
Call NH3H2O(234, P9, x9, h9: T9, P_9, x_9, h_9, s9, u9, v9, Qu9)

P10 = P1; x10 = x6;s10s = s6;
Call NH3H2O(235, P10, x10, s10s: T10, P_10, x_10, h10s, s_10s, u10s, v10s,
Qu10s)

h10 = h6 - ETA_turb*(h6 - h10s)

w_turb = (1 - w)*(h6 - h10)

w_net = w_turb - w_pump
```

```
qh = h5 - h4

ql  = h1 - h2


ETA_th = THEFF(w_net,qh)
```

A.3 Vapor Compression Refrigeration Cycle Codes

The *EES* codes that were used to calculate the coefficient of performance for the different vapor compression refrigeration cycles is included in this section. All of the vapor compression cycles were testes with several refrigerants; R123, R124, R134a, R141b, R142b, R152a, R236fa, R245fa, R600, R600a, R717, and R718. The refrigerants were chosen based on their inclusion in the *EES* fluids database, and their favorable critical temperatures.

A.3.1 Single Stage Vapor Compression Cycle Code

The single stage vapor compression refrigeration cycle is the easiest to code and understand since it uses the standard vapor compression refrigeration model. It is a four state cycle that is well known. In order to run the provided code, there are several parameters that need to be set. The working fluid, source temperature, sink temperature, and the pinch point for the heat exchangers need to be set in the code. Then a parametric table is generated with the temperature difference across the cycle in the first column. The desired values that are calculated by the code can be set in additional columns.

```
                 {Single Stage Vapor Compression Refrigeration Cycle}


F$ = 'R134a'            {Working fluid}
ETA_comp = 1            {The isentropic efficiency of the compressor.}

T_source = 100 [C]  {T_source is the temperature of the waste heat stream.}
```

```
T_pinch    = 4 [C]                    {T_pinch is the pinch point applied to the
                                      evaporator.}
T_low       = T_source - T_pinch  {T_low is the condenser temperature, which is
                                      the minimum temperature in the cycle.}



T1 = T_low   {T1 is set based on the source temperature and the pinch point of
             the heat exchanger.}
x1  = 1      {The working fluid leaving the evaporator is a saturated vapor.}
P1 = Pressure(F$,T=T1,x=x1)
h1 = Enthalpy(F$,T=T1,x=x1)
s1 = Entropy(F$,T=T1,x=x1)

P2 = P3
s2 = s1
h2s = Enthalpy(F$,P=P2,s=s2)
h2    = h1 + (h2s - h1)/ETA_comp{Actual enthalpy at 2 is found using the
                                      isentropic efficiency of the compressor.}



{DELTAT = 45}         {Delta T is the temperature difference between the
                      evaporater exit temp and the condenser exit temp.}
```

```
T3 = T1 + DELTAT  {The temperature at state 3 is set in the table in order to
                  evaluate the COP of the cycle at different points.}
x3  = 0           {The working fluid leaving the condenser is a saturated
                  liquid.}
P3 = Pressure(F$,T=T3,x=x3)
h3 = Enthalpy(F$,T=T3,x=x3)

h4 = h3
P4 = P1

ql  = h1 - h4
qh = h2 - h3
w   = h2 - h1

COP_HP    = qh/w
COP_REF  = ql/w
```

A.3.2 Cascade Vapor Compression System Code

The cascade vapor compression configuration is slightly more complicated than the single stage configuration. The cascade configuration puts two single stage vapor compression refrigeration cycles together at one of the heat exchangers. In the cascade configuration, the low temperature condenser interacts with the high temperature evaporator. Since both of the cycles are closed, and the working fluids do not mix, multiple refrigerants can be used based on the application. As in the previous vapor compression cycle; the source temperature, sink temperature, pinch point, and refrigerant needs to be set. For this configuration, there are two refrigerants that need to be set, a low temperature and a high temperature refrigerant. There is also an additional parameter that is used as an optimization variable. The compressor exit pressure for the low temperature refrigerant. This mid-pressure is used as the optimization parameter by using the imbedded maximization function in *EES*.

To operate the provided code, a parametric table needs to be set up where the temperature difference between the high temperature cycle condenser and the low temperature cycle evaporator is in the first column. Then the Min/Max Table function is selected. When the function is selected, a window opens allowing the user to choose to minimize or maximize the function for a particular variable. Maximizing the COP is chosen then the optimization variable and boundaries have to be set. The variable P2 is chosen, and then the pressure boundaries are set. In this work, since the refrigerant was the same in both cycles, the pressure boundaries was set so that P2 could not be below the high temperature cycles evaporator pressure or above the high temperature cycles condenser pressure. The smaller the possible range of the variable, the more accurate the results are.

```
                {Cascade Vapor Compression Refrigeration Configuration}

WH = 1000 [kW]     {The WH power is required to calculate the COP since there
                   are two seperate mass flows, but the actual value is not
                   important.}

A$ = 'R718'   {Working fluid used in the low temperature cycle, cycle A.}
B$ = A$       {Working fluid used in the high temperature cycle, cycle B. In this
              study we have limited all systems to use the same fluid.}

ETA_comp1 = 1      {The isentropic efficiency of the low temperature heat pump
                   compressor.}
ETA_comp2 = 1      {The isentropic efficiency of the high temperature heat pump
                   compressor.}

T_source = 120 [C] {T_source is the temperature of the waste heat stream.}
T_pinch    = 4 [C] {T_pinch is the minimum temperature difference between the
                   low pressure condenser and high pressure evaporator.}
```

```
T_low       = T_source - T_pinch          {T_low is the saturation temperature of
                                          the low temperature cycle evaporater.}

                  {Low Temperature Vapor Compression Cycle}
T1 = T_low
x1  = 1         {The working fluid is a saturated vapor leaving the evaporator.}
P1 = Pressure(A$,T=T1,x=x1)
h1 = Enthalpy(A$,T=T1,x=x1)
s1 = Entropy(A$,T=T1,x=x1)

{P2 = ?}        {P2 is set by maximizing the heat pump COP based on the
                maximum and minimum temperatures.}
s2 = s1
h2s = Enthalpy(A$,P=P2,s=s2)
h2 = h1 + (h2s - h1)/ETA_comp1

x3 = 0;         {The working fluid is a saturated liquid as it leaves the condensor.}
P3 = P2
T3 = Temperature(A$,P=P3,x=x3)
h3 = Enthalpy(A$,T=T3,x=x3)

h4 = h3
P4 = P1
```

```
ma = WH/(h1 - h4)   {This is the mass flow of the refrigerant in the low
                    temperature cycle.}
QHA = ma*(h2 - h3)

                    {High Temperature Vapor Compression Cycle}
T5 = T3 - T_pinch
x5 = 1 {The working fluid is a saturated vapor as it leaves the evaporator.}
P5 = Pressure(B$,T=T5,x=x5)
h5 = Enthalpy(B$,T=T5,x=x5)
s5 = Entropy(B$,T=T5,x=x5)

s6 = s5
P6 = P7
h6s = Enthalpy(B$,P=P6,s=s6)
h6 = h5 + (h6s - h5)/ETA_comp2

{DELTAT = ?}        {DELTAT is the temp difference between the exit temp of the
                    low temp evaporator and the exit temp of the high temp
                    condenser.}
T7 = T1 + DELTAT
x7 = 0 {The working fluid is a saturated liquid as it leaves the condenser.}
P7 = Pressure(B$,T=T7,x=x7)
```

```
h7 = Enthalpy(B$,T=T7,x=x7)


h8 = h7

P8 = P5


mb = QHA/(h5 - h8)            {Mass flow of the refrigerant in the high temperature

                              cycle, cycle B.}


Q_L   = ma*(h1 - h4)

Q_H  = mb*(h6 - h7)

W_A = ma*(h2 - h1)

W_B = mb*(h6 - h5)

W = W_A + W_B


COP_HP = Q_H/W
```

A.3.3 Multi-Stage Vapor Compression Configuration Code

The multi-stage vapor compression system is an augmentation of the cascade vapor compression configuration. The multi-stage configuration, which in this work was limited to only two stages, removes the heat exchanger that connects the low temperature and high temperature vapor compression cycles in the cascade configuration when the same refrigerant is used in both cycles. The heat exchanger is replaced with a flash chamber that is nothing more than a phase separator. The double stage configuration is slightly more complicated than the cascade configuration because the mass flow is separated into two different flows at the flash chamber. The portion of the mass that is flashed to a vapor by the first expansion valve is denoted as ‘y’ and is sent to a mixing chamber after the low pressure compressor. The portion of the mass flow that is flashed is a function of the mid-pressure, which is the optimization variable used to maximize the coefficient of performance for the system. The code is set up and ran the same way as the cascade configuration code.

```
                    {Double Stage Vapor Compression System}


F$ = 'R134a'        {Working Fluid}


ETA_comp1 = 1       {Isentropic efficiency of the low pressure compressor.}
ETA_comp2 = 1       {Isentropic efficiency of the high pressure compressor.}
```

```
T_source = 120 [C]  {T_source is the temperature of the waste ehat stream.}
T_pinch    = 4 [C]      {T_pinch is the pinch point applied to the cycle evaporator in
                        respect to the source temperature.}
T_low = T_source - T_pinch          {T_low is the temperature in the low pressure
                                    evaporater. It is set in the table.}

T1 = T_low
x1 = 1                  {Working fluid leaves the evaporater as a saturated vapor.}
P1 = Pressure(F$,T=T1,x=x1)
h1 = Enthalpy(F$,T=T1,x=x1)
s1 = Entropy(F$,T=T1,x=x1)

s2 = s1
{P2 = ?}                {P2 is used as the optimization variable}
h2s = Enthalpy(F$,P=P2,s=s2)
h2 = h1 + (h2s - h1)/ETA_comp1 {Actual enthalpy is found using the compressor
                                    efficiency.}

h3 = (1 - y)*h2 + y*h7          {y is the mass fracion of the working fluid that is a
                                saturated vapor after the first expansion valve.}
P3 = P2
s3 = Entropy(F$,P=P3,h=h3)
```

```
s4 = s3

P4 = P5

h4s = Enthalpy(F$,P=P4,s=s4)

h4 = h3 + (h4s - h3)/ETA_comp2 {Actual enthalpy is found using the compressor
                                efficiency.}


{DELTAT =?}        {DELTAT is the temperature difference in the exit
                   temperatures of the evaporater and condenser.}

T5 = T1 + DELTAT         {T5 is the condenser exit temperature, and it is set in
                         the table.}

x5 = 0             {The working fluid leaves the condenser as a saturated
                   liquid.}

P5 = Pressure(F$,T=T5,x=x5)

h5 = Enthalpy(F$,T=T5,x=x5)


h6 = h5

P6 = P2

x6 = Quality(F$,P=P6,h=h6)


y = x6             {y is the mass fraction that leaves the phase seperator and
                   goes to the mixing chamber in a saturated vapor.}
```

```
x7 = 1

P7 = P6

h7 = Enthalpy(F$,P=P7,x=x7)


x8 = 0          {The portion of the mass that leaves the phase seperator and goes

                to the second expansion valve is a saturated liquid.}

P8 = P6

h8 = Enthalpy(F$,P=P8,x=x8)


h9 = h8

P9 = P1

x9 = Quality(F$,P=P9,h=h9)


ql   = (1-y)*(h1 - h9)        {The heat transfer into the system has to be multiplied

                              by the working fluid mass fraction across that stage

                              so that the correct COP can be found.}

qh  = h4 - h5

w1  = (1 - y)*(h2 - h1)

w2   = h4 - h3


wnet = w1 + w2


COP_HP = qh/wnet
```