

## ABSTRACT

HOLALU RAMAKRISHNA, HARSHA. Feasibility Analysis of Waste Heat Recovery from a Steam Jet Vacuum System. (Under the direction of Dr. Stephen Terry.)

Two important and current topics of interest for most industries are emissions and energy efficiency. Current stringent laws regarding pollution control makes reduced emissions and improvements in energy utilization efficiency an important aspect to investigate. This has renewed the interest in waste heat recovery (WHR).

This project proposes to increase the energy efficiency of a Steam Jet Vacuum system, which is designed to convert the pressure energy of a motivating fluid to velocity energy to entrain suction fluid. It then recompresses the mixed fluids by converting velocity energy back into pressure energy, thus creating a vacuum. This is based on the theory that a properly designed nozzle followed by a properly designed throat or Venturi will economically make use of high pressure fluid(steam) to compress from a low pressure region to a higher pressure.

Since the capacity of an ejector is fixed by its dimensions, a single stage has practical limits on the total compression and throughput it can deliver. For greater compression, two or more ejectors are arranged in series. Condensers are used between stages in multi stage units to reduce vapor load, to allow for smaller ejectors and to condense vapors from the final stage. The condensers are the source of waste heat.

Two options for waste heat recovery have been considered in this project, which is to generate power or to provide the cooling needed for the process. An Organic Rankine cycle (ORC) is a thermodynamic process that utilizes low temperature waste heat to boil a refrigerant producing vapor. The refrigerant is forced through a turbine/generator combination in order to produce alternating current. The cooling can be provided by implementing an absorption chiller which utilizes a somewhat similar process to the common vapor compression cycle with few notable differences and Absorption chiller can be powered by low quality waste heat.

This project is a case study of these applications and a brief analysis of the factors affecting performance of the proposed heat recovery cycles. The working fluid for the ORC can be tailored to fit its application, for the following project twelve working fluids were under consideration. The best working fluid for the ORC was suggested at the end of the ORC case study. The thermodynamic models for an ORC and Absorption chiller was developed using a program called Cycle Tempo. It was found that both the applications were a feasible option for waste heat recovery with a payback of less than five years.

Feasibility Analysis of Waste Heat Recovery from a Steam Jet Vacuum System

by  
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## **DEDICATION**

To my family, friends and those who believe in the idea of a world without borders.

## **BIOGRAPHY**

Harsha Holalu Ramakrishna, better known as Harsha H R, was born in Mysore, Karnataka, India to Sheela T R and Ramakrishna H B. He enjoyed the few years before getting into school in Bellur Cross with his grandparents on a farm. This gave him access to grandfather's farm equipment, old motor bike and cars at an early age which influenced him to become a Mechanical Engineer. He completed his pre-primary to degree education in Bangalore. He attended pre-primary and primary education in Anugraha public school. He completed his elementary to senior secondary education in Poorna Prajna education center which was a religiously partial school which made him hate each and every day he spent there. He then went to complete his pre degree education in St. Joseph's Pre-University College with a major in Electronics. He got the admission for BE in Mechanical Engineering with Dr. Ambedkar Institute of Technology. He had scored a low rank in Common Entrance Test for Engineering in Karnataka. He says that getting that rank led him to do some interesting things and Engineering in Ambedkar was the best days of his life. Since the education in India is not very practical, and it favors people with less practicality and more of a bookworm kind, he chose to pursue a Master's of Science at NC State. Education at NC State was not as practical as he expected, until he struck gold by getting to know Dr. Stephen Terry and his team and joined the IAC on May 2012. He worked as a Graduate Energy Engineer / Research Assistant at IAC during which he presented the following thesis. He plans to stay in US for a few more years and pursue a job as Energy Engineer.

## **ACKNOWLEDGMENTS**

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# **Chapter 1**

## **Introduction**

Due to the increasing concern over energy supplies and global warming, the interest for low grade heat recovery has been growing for the last few years.

The NC State Industrial Assessment Center identified a waste heat source in a chemical plant located in North Carolina and the following study has been conducted to check the extent of opportunity for waste heat recovery. Two waste heat recovery options have been considered in this project. The feasibility of either generating power or supplementing the cooling needed for processes in the subject facility will be investigated.

The option to generate power considers units based on an organic Rankine cycle. A waste heat powered chiller will be studied as a cooling alternative. The waste heat source is a significant quantity of steam condensed between stages and also the steam discharged to the atmosphere at the final stage of a Steam Jet Vacuum System. Figure 1 gives a pictorial representation of the project approach.

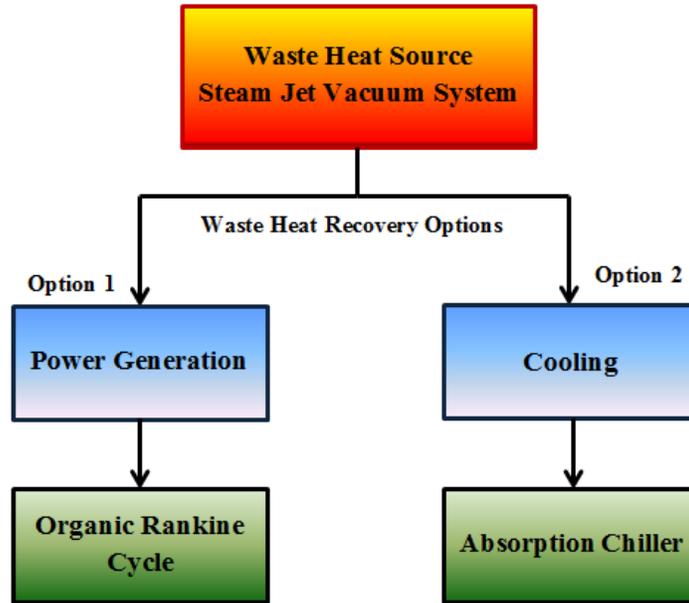


Figure 1 Project approach block diagram

## 1.1 Importance of Waste Heat Recovery

The following research can possibly aid in providing a significant opportunity for improving industrial energy efficiency. The industrial sector uses the largest share of energy in the United States, accounting for more than 28 quads, or 30 percent of all energy consumed domestically. Roughly one third of the energy consumed by industry is discharged as thermal losses directly to the atmosphere or to cooling systems (1).

Waste heat refers to energy generated in industrial processes without being put to practical use. In general, it is at a temperature too low to be used in standard power cycle to generate electricity. Heat recovery units lead to a combination of economic and environmental benefits by making use of the waste heat which has not been put into use.

Industrial waste comes in different states of matter. For instance wood waste, composite waste, waste fiber produced by food industry, exhaust gases, slag, fly ash and sludge are common forms of industrial waste. Some of the waste products can be used as raw materials

by other industries or incorporated on site, can be burned or gasified to produce electricity or useful thermal energy. For example, tri-generation using wood fired boilers has been recommended for furniture industries generating tons of wood waste on a daily basis. Tri-generation refers to producing power, useful thermal energy and process cooling.

The waste recovered from any waste heat source replaces the fossil fuel that would otherwise be used thus reducing the fossil fuel consumption as well as reduced pollutant emissions. In the steam jet vacuum system nearly 95-99% of the energy input to the system is lost as waste heat (2).

## **1.2 Role played by Industrial Assessment Center, NC State**

The projects taken up by NC State, Industrial Assessment Center (IAC) explore novel processes to expand U.S. leadership in advanced manufacturing and the global clean energy economy. The objective of the IAC is to identify, evaluate, and recommend opportunities to conserve energy and reduce energy costs in small to medium-sized plants.

Some of the most widely recommended industrial energy conservation measures by the IAC include but by no means limited to: using compressor waste heat for space heating, condensate return to boilers, boiler tune up, economizers on boiler exhaust, operating motors at maximum efficiency, installing fluorescent lighting fixtures or natural lighting, installing variable speed drives and compressor management.

## **1.3 Factors Influencing the Feasibility of Waste Heat Recovery**

Any waste heat recovery project is influenced by the parameters such as operating schedule of the waste heat source under consideration, heat quantity, quality, composition, minimum exit temperature, implementation cost and the physical limitations for heat recovery.

The total heat content, which is a function of the temperature and flow rate of the heat source is a key factor. It influences the rate of heat transfer between surfaces. The higher the heat content, the better it is for any heat recovery application. The longer operating hours helps to get an early payback.

The chemical composition of the waste stream affects the recovery process and the material selection. The composition and phase of the waste heat stream will determine factors such as thermal conductivity and heat capacitance, which affects the heat exchanger effectiveness. Heat transfer rates are influenced by the deposition of fouling substances on the heat exchanger.

In many cases equipment arrangement presents physical constraints for easy access to the heat source and prevents installation of any additional equipment for recovering heat.

## Chapter 2

### Background

Industrial plants, including the subject plant often vent significant quantities of thermal energy in the form of low pressure steam to the atmosphere, wasting energy, water and water treatment chemicals.

#### 2.1 Steam Jet Vacuum system

In general, high capacity vacuum pumping of upto 0.001 Torr is most economically performed by steam jet vacuum systems. 1 Torr is equivalent to 1 mm Hg, where 1 atm is equal to 760 Torr.

A steam jet vacuum system is a tool that has no moving parts, no lubrication or oil problems, simple in construction, easy installation, insensitive to fouling, low capital cost, easy maintenance and long useful operating lives. But a significant quantity of thermal energy is lost from the system thus leading to a low thermal efficiency

The subject facility uses a multi stage condensing steam jet vacuum system. The unit has four stages with condensing after the second stage. The unit uses a separator for the first and the second stage. The non-condensable gases are removed from condenser by a two stage evacuator. The unit is operated to create pressures in the range of about 0.25 Torr to 10 Torr.

The steam jet vacuum system is made up of ejectors, condensers, separators and interconnecting piping. The separators are basically used to better the quality of the steam being supplied to the steam jet vacuum system. So they are not an integral part of the steam jet vacuum system. Figure 2 shows a 3D-CAD image of the 4 stage condensing steam jet vacuum system used in the subject facility.

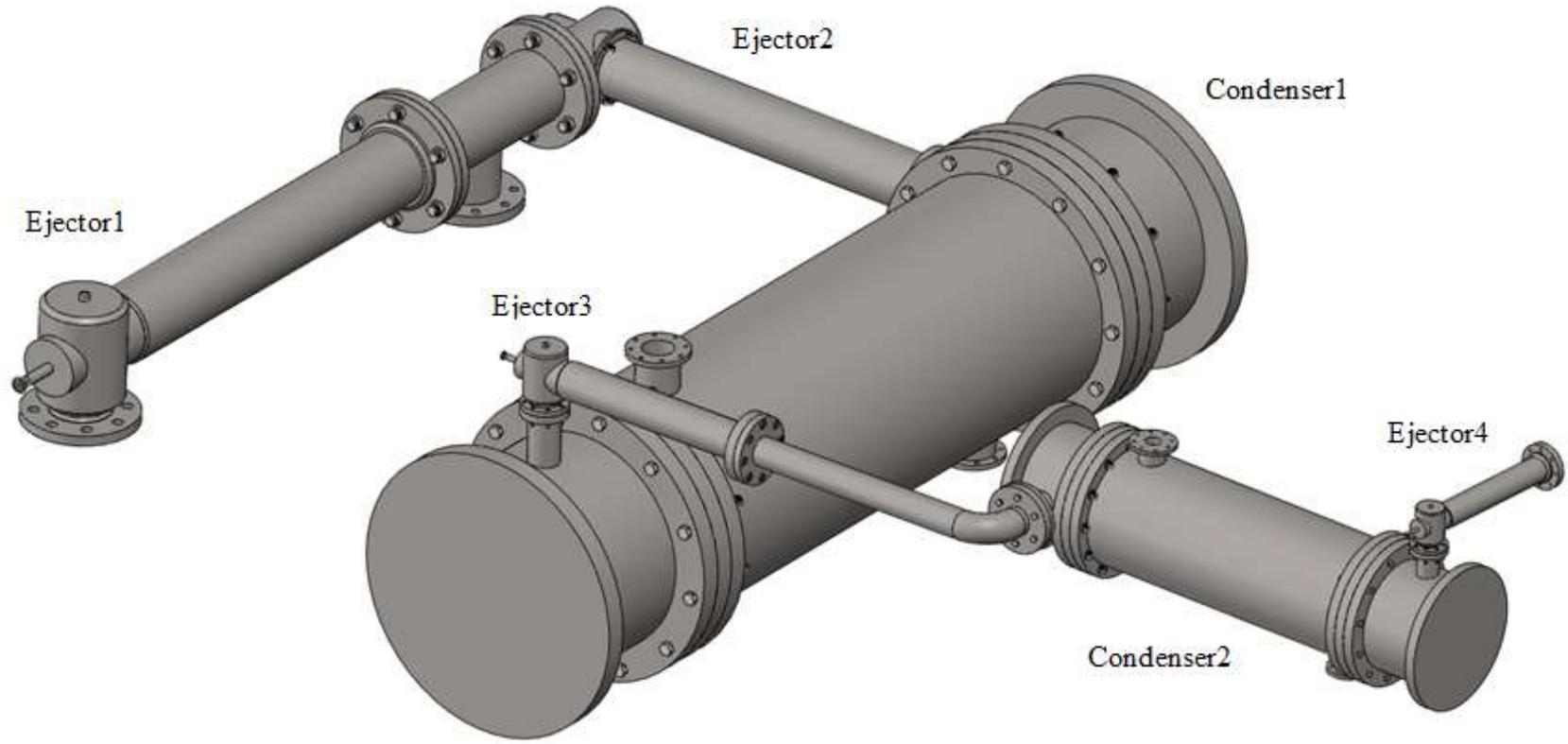


Figure 2 Four Stage Condensing type Steam Jet Vacuum System

Steam Jet Vacuum systems are typically made of cast iron or steel with nozzles of stainless steel. Certain applications require special alloys and plastics. Monel, hastelloy, Ni-resist, Teflon, Titanium and ceramics are some of the materials used (3).

### **2.1.1 Ejectors**

Ejectors are made up of a motive fluid chest, a nozzle that increases the velocity of a fluid at the expense of pressure, a mixing chamber and a diffuser that increases pressure by slowing down the fluid. Figure 3 shows the single stage ejector and its cross section.

Steam ejectors are designed to convert the pressure energy of a motive fluid to velocity energy by an adiabatic expansion in the nozzle. The nozzle exit velocity is usually supersonic in a steam jet vacuum system. The steam jet enters the mixing chamber connecting vessel or a pipe and entrains the suction fluid. The suction fluid can be any fluid that is compatible with the steam and components of the system. The velocity of the mixture drops a little due to entrainment. The mixture enters the diffuser where a portion of the velocity energy is converted to pressure energy in the converging side of the diffuser. The mixture is recompressed in the diverging outlet side of the diffuser. The discharge pressure of the mixture is between the pressures of the motive and suction stream pressure which is 5 to 15 times the suction pressure (3). There is a corresponding rise in mixture temperature as compression occurs. They usually operate under a single set of conditions. Ejector design can be classified either as critical or non-critical. Critical design means that the fluid velocity in the diffuse throat is sonic. In non-critical units the fluid velocity is subsonic.

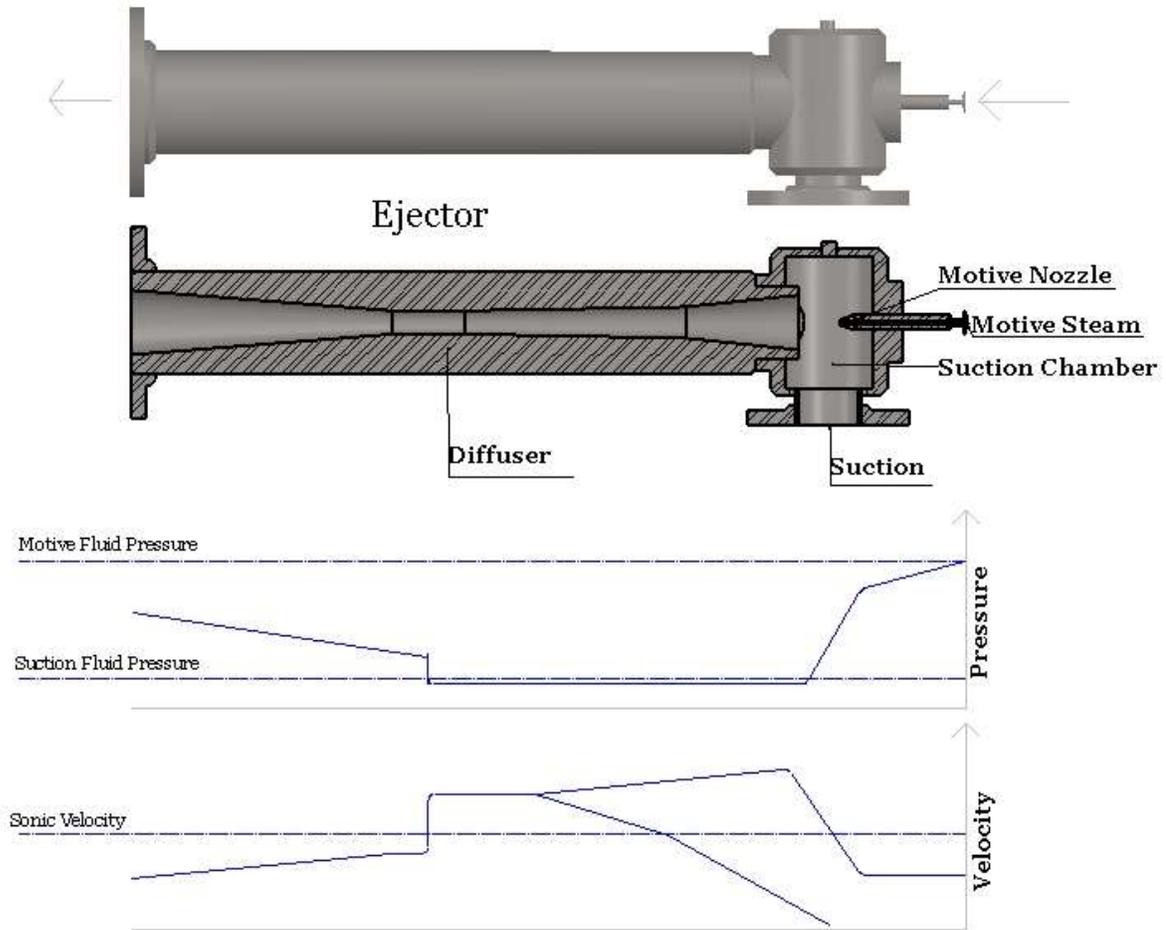


Figure 3 Typical ejector cross-section, pressure and velocity profiles along an ejector operating in critical mode

### 2.1.2 *Condensers*

Condensers are an important part of the steam jet vacuum system. They can be used as a pre-condenser, inter-condenser or an after-condenser depending on the need. The inter-condenser is used to condense the motive steam and any condensable vapor from the preceding stage to reduce the load on the following stage. This allows smaller ejectors to be used and reduced steam consumption. The pre-condenser is used to reduce the load on the first stage for certain specific application. The after condenser is used to ease the discharge of the mixture to the atmosphere. The after condenser does not affect the overall performance of the system.

The specified quantity of cooling fluid must be sent to the condenser and must be below or at the design temperature. If the volume of the cooling fluid drops or temperature is not kept below or at the required level, the temperature and pressure of the vapor in the condenser will rise and the system will cease to operate properly. The subject facility uses a single pass shell and tube condenser. Figure 4 shows the shell and tube heat exchanger and its cross-section.

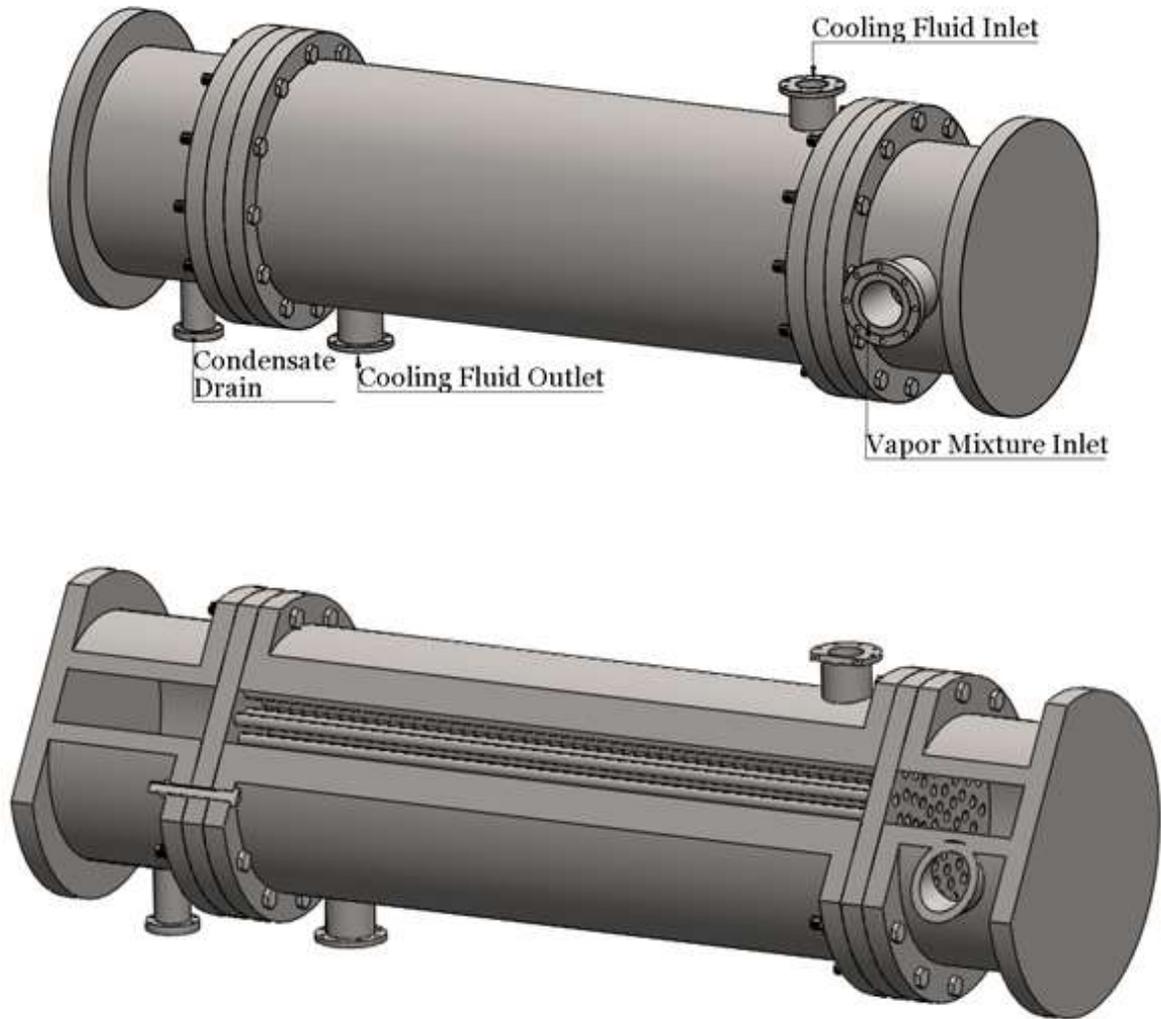


Figure 4 Typical Shell and Tube heat exchanger and its cross-section

Since the operating pressure of the condenser is sub-atmospheric, the condensate collected must be continuously removed which may be accomplished by gravity through a trap or a loop seal tailpipe and with the help of a condensate pump.

## **Condensers Classification**

Condensing systems can be classified into following categories:

- Water-cooled surface condensers
- Direct contact type/Jet condensers
- Air-cooled condensers
- Hybrid condensing systems

### **Water cooled surface condensers**

The most popular condensing systems are the water cooled surface condenser types. Cooling water is pumped directly from the cooling water source into the condenser inlet water box. The tube sheet connects the water box to the tubes, allowing cooling water to pass through the inside of the condenser without contacting the steam condensate circuit. The outside surface of the tubes makes up the condensing surface and it is this surface and the cooling water temperature that dictates the condenser performance. The fluid mixture from the diffuser enters the condenser chamber and condenses upon contact with the outside surface of the cold tubes. The latent heat discharged by the condensed steam is transferred through the tube walls into the cooling water, because the condenser chamber is a fixed a volume and sealed from the ambient air, the state change from saturated steam to saturated liquid occurs at a constant pressure and a constant temperature. The large reduction in the fluid specific volume causes the pressure in the chamber to drop to a vacuum condition, to achieve steady state equilibrium.

Condensate is collected at the bottom of the surface condenser and is then pumped out. The cooling water exits the tubes at an elevated temperature and discharges into the outlet water

box and eventually returns to the cooling source. To maintain a constant pressure inside the condenser a vacuum pump or steam ejector is used to remove non-condensable such as air and other gases.

The most common type of surface condensers is the shell and tube type condensers. Specific forms differ according to the number of shell and tube passes, the simplest one being a single tube and shell pass. Baffles on the tubes can also be used in to increase the effective surface area for heat transfer. The condensation can be on either the shell side or the tube side depending on the condition of the condensing vapors. If the condensing vapors are extremely fouling, tube side condensation is preferred. If fouling is minor, shell side condensing is better due to the higher condensing efficiency.

The surface condensers can be further classified based on the tube design and configuration as spiral tube, square or rectangular tube etc.

### **Direct contact type/Mixing type/Jet condenser**

These kinds of unit are less common these days due to the strict regulations regarding pollution. The cooling water mixes directly with the condensing vapor causing condensation. These units are being used only when the vapors are clean and the initial cost is a factor. Due to direct contact of two fluids the circulating water requirement is much less as compared to other types of condensers. Space requirements and size is also less for jet condenser.

The steam to be condensed and cooling water are directly mixed by breaking up of water in the form of spray and allowing small sized water particles to fall down through the body of steam. It is thus desired to atomize water into small sized particles so that increased surface area is available for heat exchange between hot and cold fluid. Based on the relative movement of two fluids they can be classified as counter flow and parallel flow.

### **Air cooled Condensers**

In areas where water is scarce, sufficient condensing capacity is provided with direct air cooled condenser. The steam enters a central plenum/pipe located above a series of finned tubes, sloped down towards a condensate collection piping system. Because direct water contact is non-existent in air cooled condensers, the exhaust/condensing pressure is dependent on the dry bulb temperature of the ambient air.

### **Hybrid Condenser systems**

Hybrid condensing systems are a suitable combination of an above mentioned condenser systems like direct air cooled condenser and surface condenser. An example of which is an evaporative condenser, it is similar to a cooling tower with few notable differences. It uses a combination of both air and water to provide cooling in the condenser. In this type of condenser the water falls from top through the nozzles over the condenser coil. Water picks up heat from the steam flowing through condenser coil and gets warmed up. This water is recirculated by circulation pump. The flow of air across condenser coil may be natural or forced to enhance the cooling rate. Water evaporated is removed by air leaving unit. Heat required for evaporation is extracted from the steam flowing inside tubes and thus cause phase transformation. For preventing the exit of water vapors with air going out the separator/eliminator is put on top before the final exit by which water vapor are recovered up to certain extent.

## **2.2 Classification of a Steam Jet Vacuum System**

Since the capacity of the single ejector is fixed by its dimensions a single unit has practical limits on the total throughput it can deliver. For greater suction pressure two or more ejectors are connected in series. For greater throughput capacity of gas or vapor two or more ejectors can be arranged in parallel.

The suction capacity can be lowered by a few techniques like increasing the motivating pressure and by controlling suction pressure. The suction pressure can be controlled using a control valve which would create an artificial load. An artificial load can also be created by using the discharge from any one of the ejectors creating a loop. The load can also be taken from the outside if feasible. Few units are designed to turnoff certain stages to control suction.

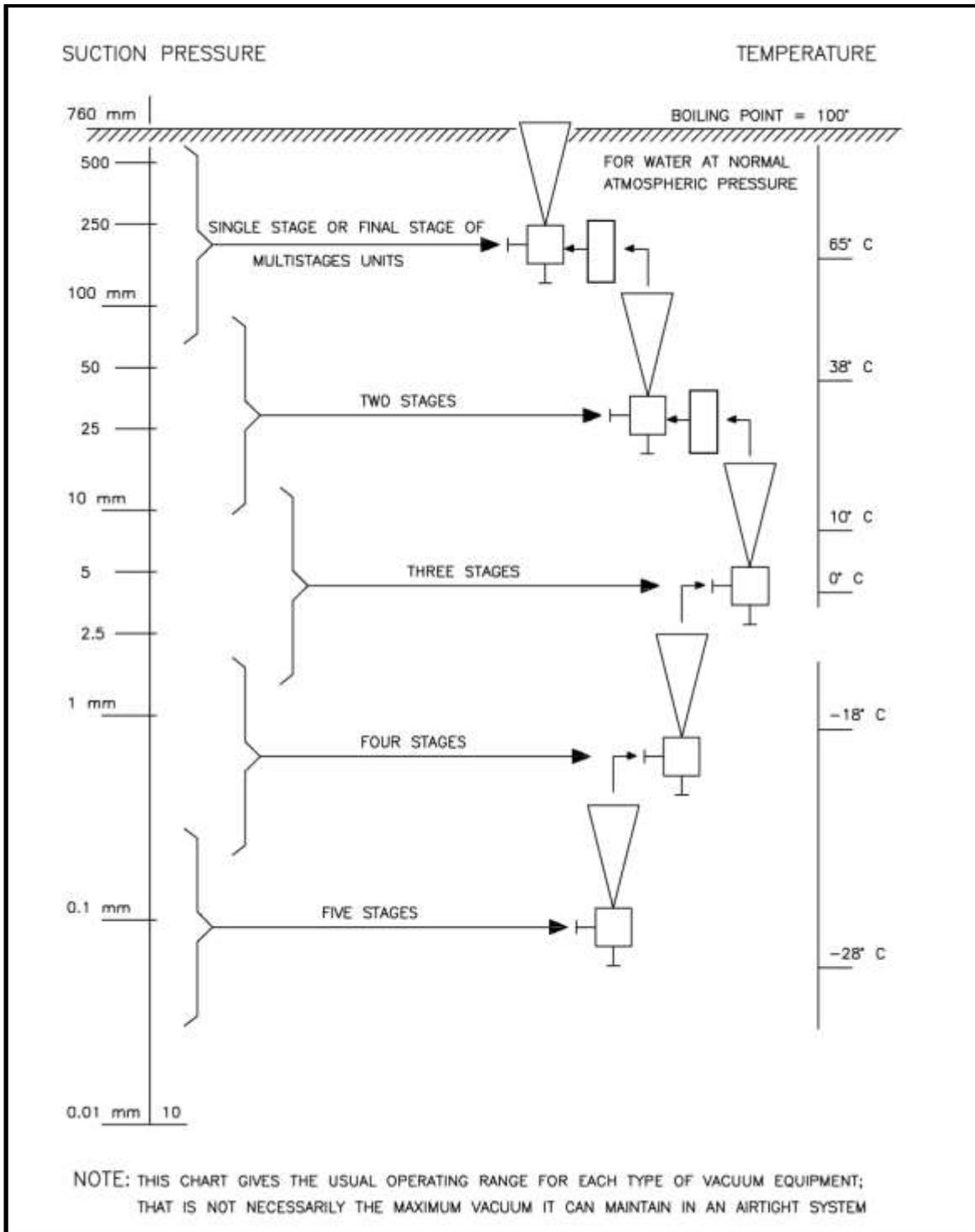


Figure 5 Suction pressures attained by multistage units (4)

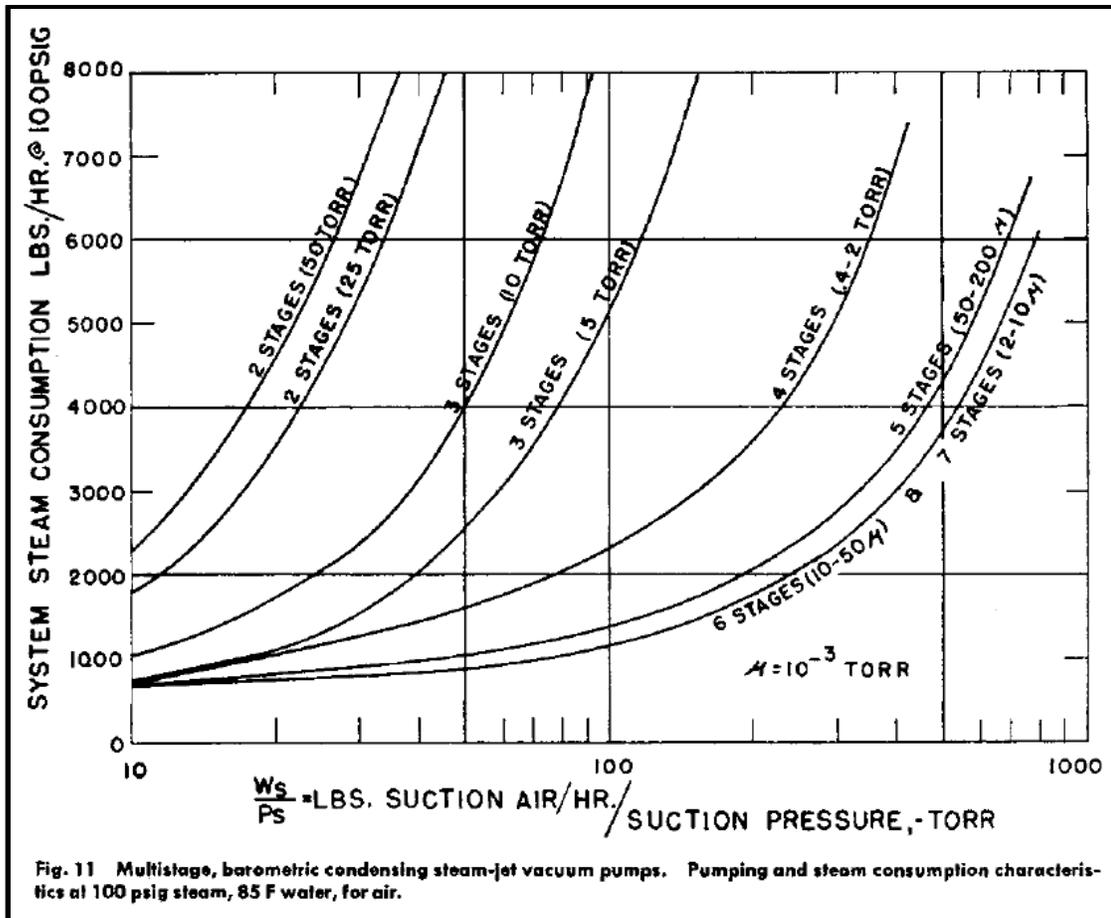


Figure 6 Steam consumption versus Suction quantity (2)

### Single stage

They are simple in design and most common units. They are designed to create low vacuum in the range of 760 Torr to 75 Torr (3). Single stage units typically discharge at or near atmospheric pressure. Single stage ejectors are commonly used as thermo compressors designed to boost low pressure steam to a higher pressure and temperature. Most thermo compressors are used to increase the efficiency of heaters, dryers or evaporators.

### Multi stage non-condensing

They are used where lower suction pressures are required and factors like the first cost, availability of water and space is an issue. These units have higher steam consumption

because each successive stage is required to handle load and motive steam of the stage ahead of it. These units have a higher operating cost.



Figure 7 Two Stage Non-Condensing

### **Multi stage condensing ejectors**

These units use condensers between stages to condense the mixture from the preceding stage to reduce the load on the following stage. Multi stage condensing units can handle pressures ranging from 100 Torr to as low as low as 0.5 Torr (3). They have low operating cost. For extremely low suction pressures a six stage unit could also be utilized.



Figure 8 Two Stage Condensing

### **Multi stage condensing followed by multi stage non-condensing**

These units are used to achieve extremely low vacuums with economy. The non-condensing stage follows the condensing stage. These units are designed to handle extremely low vacuum ranging from 5 Torr to 0.001 Torr (3).

### **Multi stage followed by a liquid ring vacuum pump**

These units are often used to reduce the energy costs. These combination units are designed to handle low absolute pressures in the range of 50 Torr to 1 Torr (3). Higher initial cost and maintenance cost are an issue to look out for, but lower operating costs, reduced space requirements, access to spare parts makes it a little more attractive.

## **2.3 Efficiency**

All steam jet vacuum systems have long been known to have low thermal efficiencies in the range of the 1 to 5 percent (2). For most units energy efficiency is measured as comparison of output to input energy. But in the case of a vacuum system overall efficiency is measured in terms of the entrainment efficiency, nozzle efficiency and the diffuser efficiency. The overall efficiency is given by product of nozzle, diffuser and entrainment efficiency.

### **Entrainment efficiency**

Due to the entrainment of the low velocity suction fluid in the motive fluid there is a loss of kinetic energy owing to impact and turbulence. The fraction of the suction fluid that is successfully removed from the chamber through the exchange of momentum is called the entrainment efficiency.

The entrainment efficiency is a function of molecular weight and temperature of the suction fluid. The higher the molecular weight, the greater the capacity assuming equal motivating fluid quantity. The ejector will handle less fluid with the rise in temperature of the suction fluid.

### **Diffuser efficiency**

It is a measure of the diffuser's ability to increase the pressure of the fluid from inlet to outlet. It is expressed in terms of the ratio of the kinetic energy that can be converted to pressure energy if the fluid is discharged at the actual exit stagnation pressure to the maximum kinetic energy available for the rise in pressure. For most practical diffusers efficiency is a little lower than the isentropic diffuser efficiency.

### **Nozzle efficiency**

The efficiency with which a nozzle converts potential energy into kinetic energy, commonly expressed as the ratio of the actual change in kinetic energy to the ideal change at the given pressure ratio (5).

## **2.4 Advantages and Disadvantages**

An ejector vacuum system is chosen because they are rugged, reliable, leakage problems can be easily rectified, corrosive and slugging fluids can be handled with little modifications, no vibrations leads to easy installation, and finally can handle a high volume of suction fluid.

The quality of the steam is a major factor in the proper functioning of a steam jet vacuum system. If the quality is decreased, a decrease in suction pressure and capacity occurs. Excess steam superheat can also adversely affect the suction capacity of an ejector. It not only decreases the energy efficiency but can also choke the diffuser due to an increase in the specific volume. If moisture is present in the steam a separator and trap should be used to better the quality of steam.

Once a unit is built for a certain set of specifications like motivating pressure, discharge pressure and suction pressure its capacity cannot be increased without changing its internal physical dimensions.

## Chapter 3

# Waste Heat Recovery Applications

Waste heat recovery includes transferring heat between gases and or liquids, so that it can be used in combustion air preheating, preheating boiler feedwater, generating mechanical or electrical power, and using waste heat with a heat pump for heating or cooling facilities. This chapter discusses the power generation and waste heat fired heat pump.

### 3.1 Power generation

Heat sources at different temperatures have varying theoretical efficiency limits for power generation. The Carnot cycle is the most efficient cycle operating between two temperature limits but there are several impracticalities associated with the Carnot Cycle. The cycle that results for power generation once the impracticalities are removed from the Carnot cycle is the ideal Rankine Cycle.

#### 3.1.1 Rankine Cycle

The Rankine cycle is the ideal cycle for vapor power cycles. The ideal Rankine cycle consists of four processes the isentropic compression, constant pressure heat addition, isentropic expansion, and constant pressure heat rejection. Figure 9 shows the schematic and T-s diagram of ideal Rankine cycle and actual vapor power cycle

Saturated water is compressed isentropically by the pump to the operating pressure of the boiler. Water enters the boiler with a slight increase in temperature from the isentropic compression in the pump. The water leaves the boiler as a superheated vapor, which expands isentropically in the turbine producing work by rotating the shaft connected to the electric generator. There is a pressure and temperature drop in the turbine during expansion resulting

in a saturated liquid vapor mixture which is condensed at constant pressure in the condenser. Steam leaves the condenser as saturated liquid and enters the pump completing the cycle.

### ***3.1.2 Actual Vapor Cycle***

The irreversibilities in the components like fluid friction and heat loss to the surroundings causes the actual vapor cycle to differ from the ideal Rankine cycle.

Due to fluid friction in the connecting pipes, condenser, boiler and other components, there is a pressure drop in steam leaving boiler, steam entering turbine and saturated water leaving condenser. Also there is heat loss from the various components, which calls for more heat input through the boiler. The turbine produces a smaller amount of work as a result of irreversibilities. Thus the thermal efficiency given by ideal Rankine cycle is decreased as a result of irreversibilities.

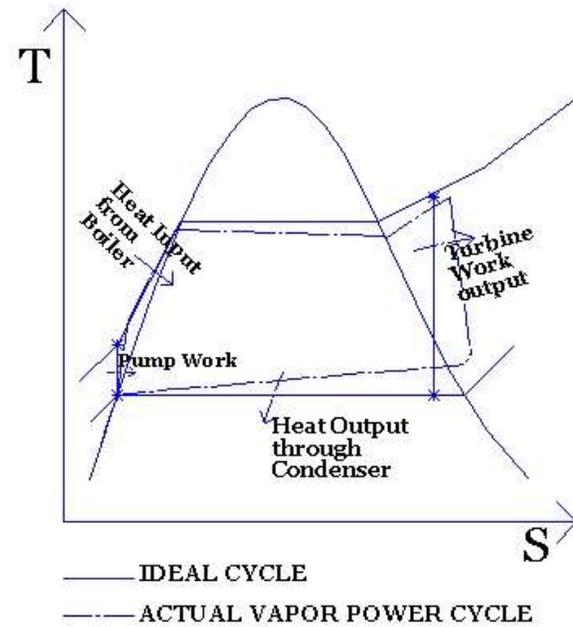


Figure 9 Schematic and T-s diagram of ideal Rankine cycle and actual vapor power cycle

### 3.1.3 Thermal Efficiency

The thermal efficiency is the ratio of energy desired to energy that cost. It can be interpreted as the ratio of the area enclosed by the cycle on a T-s diagram to the area under the heat-addition process. Maximum efficiency at a given temperature is based on the Carnot efficiency. It is given by

$$\eta_{Carnot} = 1 - \frac{T_s}{T_H} \quad (1)$$

where  $T_H$  temperature of the heat source and  $T_s$  is the heat sink temperature. The Carnot efficiency represents the maximum possible efficiency in an engine at a given temperature. The thermal efficiency of the Rankine cycle is determined by (6)

$$\eta_{th-Rankine} = 1 - \frac{q_{out}}{q_{in}} = \frac{W_{net}}{q_{in}} \quad (2)$$

where heat transfer through the boiler is  $q_{in}$ ,  $q_{out}$  is the heat discharged from the condenser,  $W_{net}$  is the total work produced, which is given by either the difference in work produced by the turbine( $W_{turb}$ ) and work consumed by the pump( $W_{pump}$ ) or difference in heat transfer through the boiler( $q_{in}$ ) and heat discharged from the condenser( $q_{out}$ ). The typical efficiency of a Rankine cycle at a power plant is 25-35% (6).

#### Heat Rate

The efficiency of power generation is often expressed in terms of heat rate. Heat rate is an expression of how much heat is supplied to generate 1 kWh of electricity. The smaller the heat rate the greater the efficiency. Disregarding the losses associated with the conversion of shaft power to electric power, the relation between heat rate and the thermal efficiency can be expressed as (6)

$$\eta_{th} = \frac{3412(\text{Btu/kWh})}{\text{Heatrate}(\text{Btu/kWh})} = \frac{3600(\text{kJ/kWh})}{\text{Heatrate}(\text{kJ/kWh})} \quad (3)$$

### 3.1.4 *Methods to Improve the Thermal Efficiency of a Rankine cycle*

Increasing the thermal efficiency can mean large savings for steam power plants around the world. Lowering condenser pressure, superheating steam to higher temperature and increasing the boiler pressure are the possible ways of increasing efficiency.

#### **Reheat**

Due to the metallurgical limits on superheating steam, Reheat is a suitable option for increasing the thermal efficiency of the system.

Figure 10 shows the schematic and T-s diagram of ideal reheat Rankine cycle. In the reheat cycle the steam expands in two stages. In the first stage the steam expands in the high pressure turbine to an intermediate pressure and then sent back to the boiler where it is reheated to inlet temperature of the first turbine. The steam expands in the low pressure turbine to the condenser pressure in the final stage. Reheat is a good option to decrease the moisture content in the turbine. The efficiency could be increased up to 5% by incorporating a single reheat unit (6). The increase in efficiency from the second reheat is about half of that resulting from a single reheat. Due to factors like superheated exhaust, added cost and complexity reheat cycle is limited to single reheat.

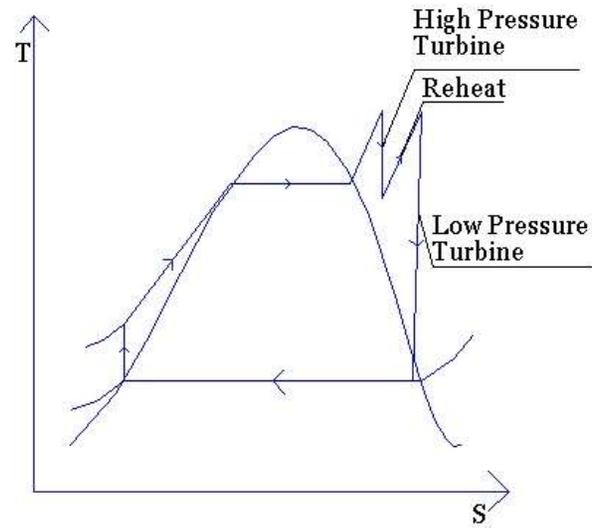
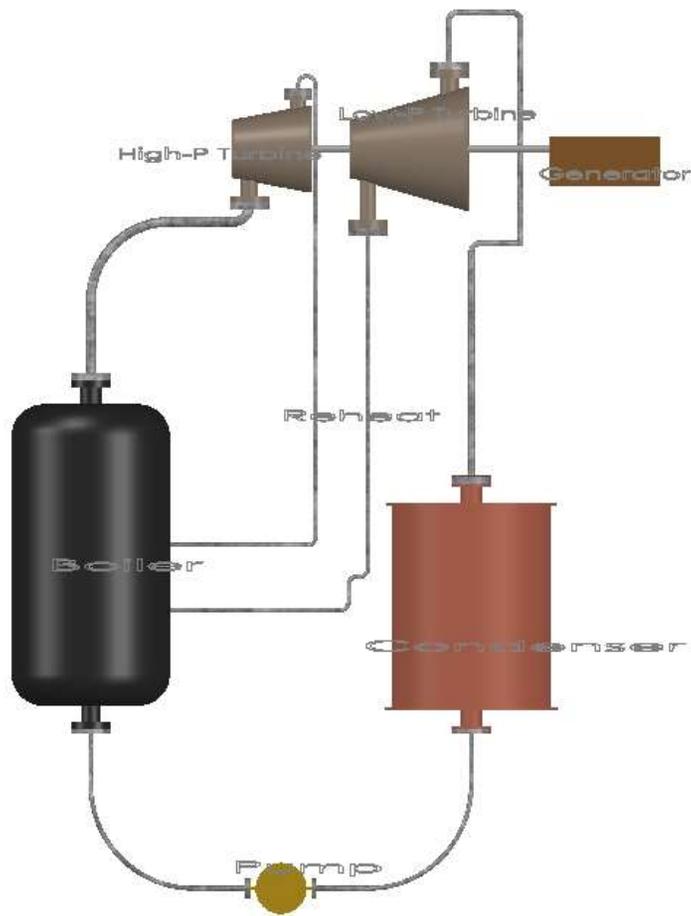


Figure 10 Schematic and T-s diagram of ideal reheat Rankine cycle

## **Regeneration**

Regeneration is the process of heating the feedwater before it enters the boiler to eliminate low temperature heat addition in the boiler which reduces the cycle efficiency. Regeneration also helps in deaerating the feedwater, reducing the load on the final stages of the turbine.

Figure 11 shows the schematic and T-s diagram of ideal regenerative Rankine cycle with open feedwater heater. Regeneration is done by bleeding the steam from the turbine at certain points and used in regenerator or a feedwater heater. The feedwater heater is either the direct contact type or the closed type each having its own advantages and disadvantages. The thermal efficiency is increased as result of regeneration because it raises the average temperature at which heat is transferred to the steam in the boiler.



### ***3.1.5 Organic Rankine cycle***

A standard Rankine cycle was developed to generate power by making use of steam with a high temperature heat source. A Rankine cycle using organic fluids as a working fluid is called Organic Rankine Cycle (ORC). The system involves the same components as in a conventional steam power plant, but operates at much lower temperature because of the properties of the working fluid.

ORC systems can be used to recover waste heat as low as 160° F and hence is beneficial in many aspects like economical utilization of energy, reduced electrical loads and reduced emission. However, efficiencies are much lower, usually 5-10%.

The operation of the ORC greatly depends on thermodynamic properties of the working fluid. ORC's uses organic fluids that have a lower boiling point, higher vapor pressure, higher molecular weight, and higher mass flow compared to water.

### ***3.1.6 Working Fluid for ORC***

To extract the most energy from a heat source using a Rankine cycle, one must determine the working fluid that can provide the best overall efficiency. Water would be the first choice and certainly provides enough reasons for its use in the high temperature range. In the low temperature waste heat application, there are other factors which often ride over those positive factors.

Other working fluids, with better efficiencies at lower heat source temperatures are used in Organic Rankine Cycle.

Selection of a suitable working fluid for the operating temperature range is a major aspect in ORC units because the irreversibilities and system efficiency greatly depend on the thermodynamic properties of the fluid. Fluids can be categorized into three types based on

the slope on their saturated vapor curves i.e. dry fluids have positive slope, and they are, in general of higher molecular weight such as R-113 and Benzene. Wet fluids have negative slopes and they are, in general, of low molecular numbers, such as water; and isentropic fluids have nearly vertical slope saturated vapor curves such as R-11 and R-12. Figure 12 shows the comparison of the types of fluids classified by their slopes on a T-s diagram.

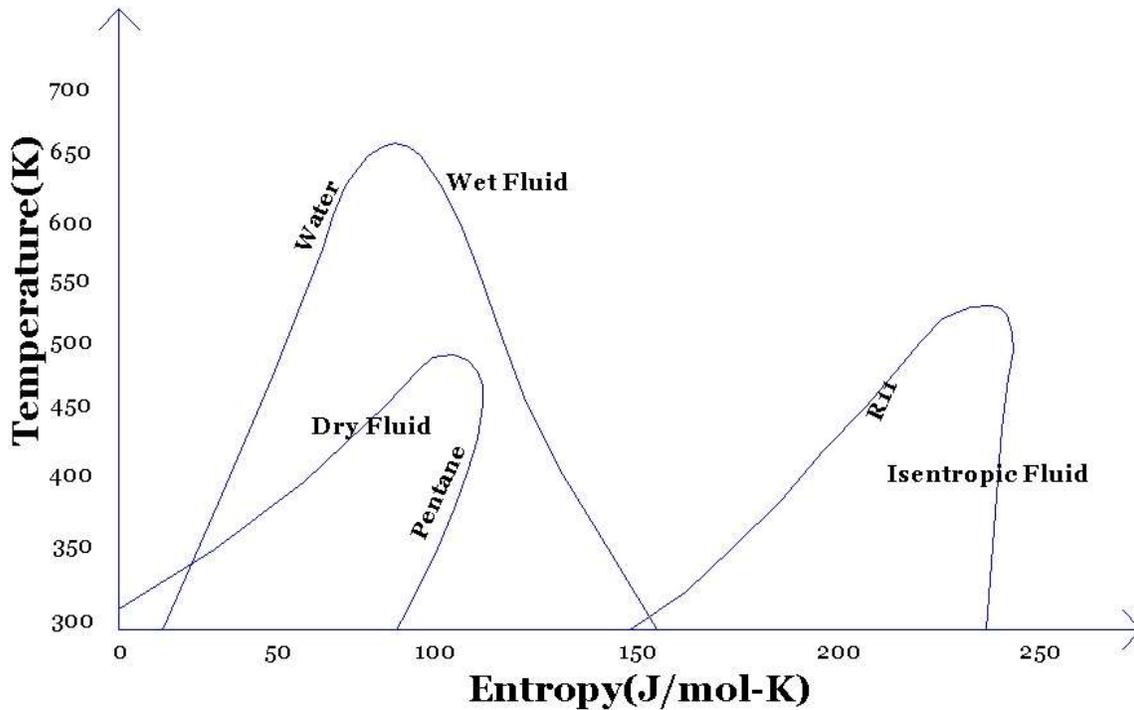


Figure 12 Three types of working fluid: dry, isentropic and wet (7)

Since the project concerns the recovery of waste heat in the range of 250° F to 390° F, the wet fluids are not adequate for ORC systems because they become saturated once they go through a large enthalpy drop after producing work in the turbine, and the condensed liquid can damage the turbine. However dry and isentropic fluids can be used instead. When these organic fluids expand in the turbine it becomes superheated or dryer. Therefore superheating prior to delivery to the turbine is not required.

Good thermodynamic fluid result in high efficiency and low cost systems. A working fluid is more desirable when it exhibits low toxicity, explosion and controllable flammability characteristics, good material compatibility, non-corrosive, and possess adequate chemical stability in the desired temperature range.

Factors that need affect the selection of the working fluid include fluid density, specific heat, latent heat, critical point, thermal conductivity, specific volume at saturation conditions.

The down fall of organic fluids include that they suffer from chemical deteriorations and decomposition at high temperature. Hence the selected fluid should be stable in the selected operating temperature range.

The range of temperatures seen by an ORC system governs the type of working fluid that can be used. The chemical composition of the fluid begins to break down at higher temperatures and thus they must be chosen so that it remains stable at the system's highest temperature but have a freezing point well below the lowest temperature in the cycle.

### **3.2 Cooling**

While using waste energy for heating purposes is relatively simple, using it to achieve a cooling effect is more complicated. Moving heat against the temperature gradient is accomplished with a heat pump. A refrigerator is a type of heat pump that requires work input to transfer heat from a low temperature to a high temperature to abide with the second law of thermodynamics. Refrigerators are cyclic devices and the working fluids used in the refrigeration cycles are called refrigerants.

Efficiencies of refrigerator are expressed in terms of coefficient of performance. It is given by (6)

$$\text{COP} = \frac{\text{Desired Output}}{\text{Required Input}} = \frac{\text{Cooling Effect}}{\text{Work Input}} \quad (4)$$

A refrigerator is based on the reversed Carnot cycle. The reversed Carnot cycle cannot be approximated in actual devices and is not a realistic model for refrigeration cycles. However, the reversed Carnot cycle can serve as a standard against which actual refrigeration cycles can be compared. Practical refrigerators use a throttling device as compared to turbine in a Carnot refrigerator. The schematic of a Carnot refrigerator and T-s diagram of the reversed Carnot cycle is given below in Figure 13.

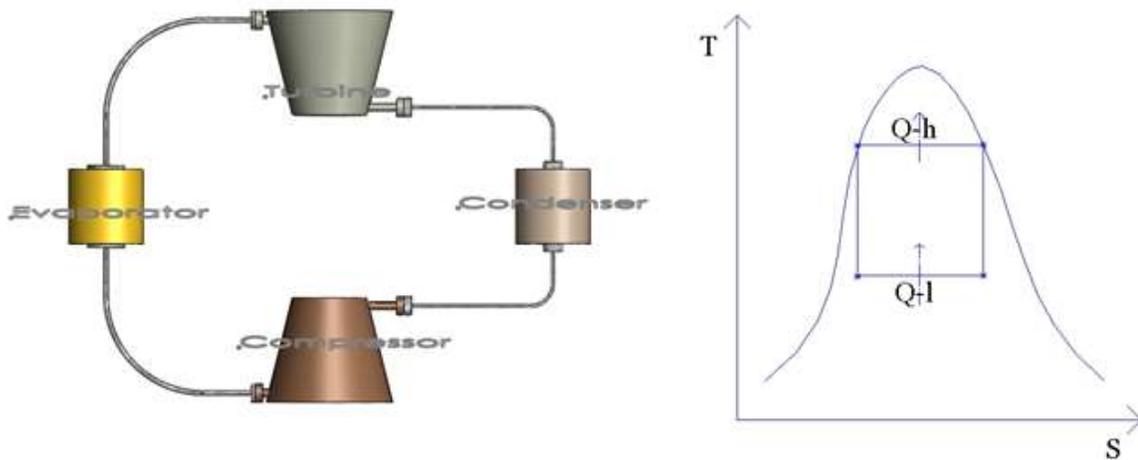


Figure 13 Schematic of a Carnot refrigerator and T-s diagram of the reversed Carnot cycle

### 3.2.1 Absorption Chiller

Absorption chillers can be classified as a heat driven heat pump working on the absorption cycle. Absorption chillers use heat input to provide cooling, i.e. the compressor is replaced by an absorption mechanism which utilizes heat. It allows the use of a variety of heat sources such as: directly using a gas burner, recovering waste heat in the form of hot water or low-pressure steam, or boiler generated hot water or steam. It is designed to operate economically when there is a source of inexpensive thermal energy in the range of 212° F to 392° F (6).

The schematic of an absorption cycle and T-s diagram of the lithium bromide absorption cycle is given below in Figure 14.

An absorption mechanism consisting of an absorber, a pump, a generator, a regenerator, a valve, and a rectifier. The absorption mechanism does the same job as the compressor i.e. to raise the pressure of the refrigerant to the required level. The refrigerant is cooled and condensed in the condenser by rejecting heat to the surrounding. It is throttled to the evaporator pressure and absorbs heat from the refrigerated space as it flows through the evaporator.

The overall coefficient of performance of an absorption chiller under reversible conditions is given by,

$$cop_{absorption} = \frac{Q_L}{Q_{generator} + Q_{pump}} = \left(1 - \frac{T_o}{T_s}\right) \left(\frac{T_L}{T_o - T_L}\right) \quad (5)$$

Where  $Q_L$  is the output,  $Q_{generator}$  is the heat from the source transferred to the generator,  $Q_{pump}$  is the pump work,  $T_o$  is the temperature of the environment or that of the cooling liquid,  $T_s$  is the source temperature and  $T_L$  is the temperature of the refrigerated space.

Absorption chillers have low coefficient of performance. Nonetheless, they can substantially reduce operating costs because of their ability to use low grade waste heat. The pump work involved in the absorption mechanism is relatively small since it is the liquid being pumped. The pump work involved is often neglected in the analysis of the cycle

Absorption chillers are available with cooling capacities ranging from kilowatts to megawatts which match with small residential to large commercial or industrial cooling needs.

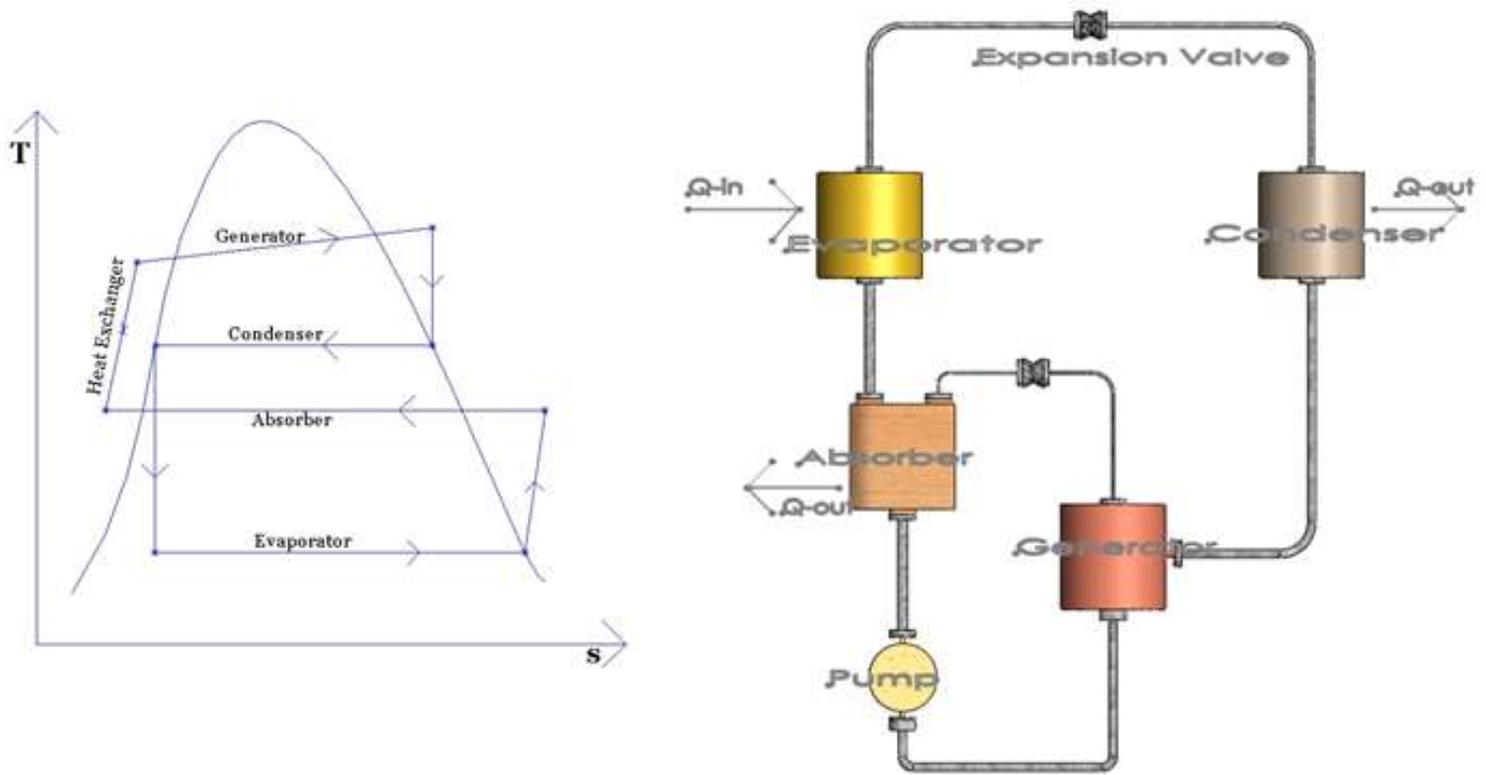


Figure 14 Schematic of an absorption cycle and T-s diagram of the aqueous lithium bromide absorption cycle

### ***3.2.2 Working fluid for Absorption chillers***

The performance of absorption chillers depend on the thermodynamic properties of the working fluid. Ammonia-water is a common working fluid mixture, where ammonia serves as the refrigerant and water as the transport medium. Some systems utilize either water-lithium bromide or water-lithium chloride as the working fluid where water serves as the refrigerant.

The working fluid has to be selected based on the application and heat source. The difference in boiling point between the pure refrigerant and the mixture at the same pressure should be as large as possible. The refrigerant should have heat of vaporization to reduce the flow from between the generator and absorber. The transport properties of the refrigerant and the mixture should be favorable. The mixture should be chemically stable, non-corrosive and non-explosive.

#### **Ammonia water mixture**

The preferred heat source temperature is 203° to 356° F and is used for application requiring temperatures in the range of -40° to 40° F. Chillers using ammonia water mixture operate under much higher pressure compared to lithium bromide water mixture chillers. However, a rectifier is required to separate the ammonia vapor from the water vapor after the generator.

The ammonia-water absorption cycle works as follows:

1. Ammonia vapor leaves the evaporator and enters the absorber, where it dissolves and reacts with water to form ammonia water mixture. This is an exothermic reaction i.e. it gives off heat. The amount of ammonia that can be dissolved in the water is inversely proportional to the temperature in the absorber; hence it is necessary to keep the absorber temperature as low as possible.

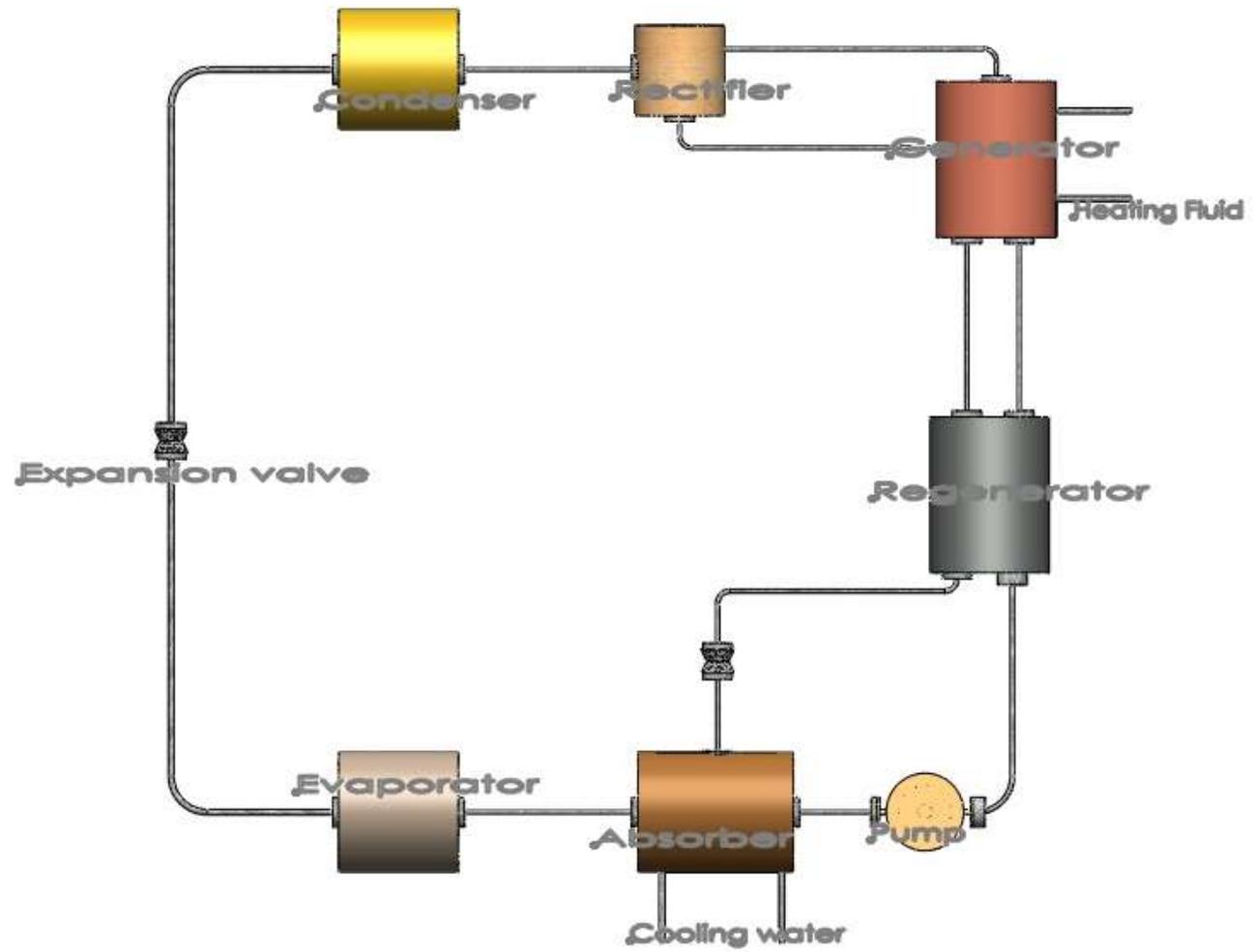


Figure 15 Schematic of an Ammonia absorption refrigeration cycle

2. The ammonia water solution, which is rich in ammonia, is then pumped to the generator. Heat is transferred to the solution from the heat source in the generator. The vapor rich in ammonia passes through the rectifier which separates the water and returns it to the generator.
3. The refrigerant vapor continues its journey through the condenser, throttling device and evaporator completing the cycle.
4. The regenerator is non-mixing type heat exchanger where the weak solution i.e. the solution with less ammonia content gives off the heat it has gained in the generator to the rich solution being pumped from the absorber.

### **Lithium bromide salt solution**

Lithium Bromide-water chillers basically classified into two types, the single effect and the double effect. They both use a concentrated salt solution of water-lithium bromide to produce chilled water. Here water is the refrigerant and lithium bromide is the absorbent. Lithium bromide has a hygroscopic property and a high affinity for water. The lithium bromide creates a high vacuum in the absorber/ evaporator. This causes the refrigerant to boil at around 44° F. In an absorption chiller the evaporator is maintained at low pressures, thanks to the hygroscopic property of lithium bromide. The operation of a lithium bromide-water absorption chiller is heavily influenced by the pressures and concentrations of its working fluid.

### **Single effect Absorption chiller**

The single effect Absorption chillers are simple in construction and hence has a low initial cost compared to double effect Absorption chiller. It is suitable for applications with small cooling capacities and where temperature heat source is low. The single effect chiller operates with a heat source temperature in the range of 176° to 266° F and with a coefficient of performance of 0.65 - 0.75 (8).

## **Double effect Absorption chiller**

The double effect chiller has two stages of generation to separate the refrigerant from the absorbent and hence operates in the range of 266° to 392° F with a coefficient of performance of 0.9 - 1.2 (8). The double effect chillers are more expensive compared to single effect chillers. The major difference between single effect and double effect absorption chiller is that the later incorporates a high temperature generator operating at a higher temperature and pressure in addition to the standard components of a single effect system, resulting in a better utilization of the supplied heat. Figure 16 shows a double effect absorption chiller.

Heat from the high temperature source is utilized in the main high temperature generator to generate the refrigerant. The high temperature refrigerant generated in the high temperature generator exchanges heat gained with the second low temperature generator to further generate the refrigerant. The rest of the cycle is same as a single effect cycle. The heat input to the double effect cycle occurs at a much higher temperature than in the single effect cycle but the heat rejection at the condenser and cooling at the evaporator are at about the same temperatures as single effect cycle, hence resulting in a higher COP.

The flow arrangement of the working fluid into the high and low temperature generator is either serial or parallel. In serial flow units the refrigerant rich solution flows into the high temperature generator first and is generated by the external heat source, this generated solution is then introduced into the low temperature generator and further generated by the refrigerant vapor from the high temperature generator.

In parallel flow arrangement the refrigerant rich solution produced in the absorber is sent to both the high and low temperature generators through the high temperature solution heat exchanger and low temperature solution exchanger, respectively. The parallel arrangement is better against crystallization issues since it operates at a safe distance from the crystallization line and also has a slightly better COP than the serial arrangement.

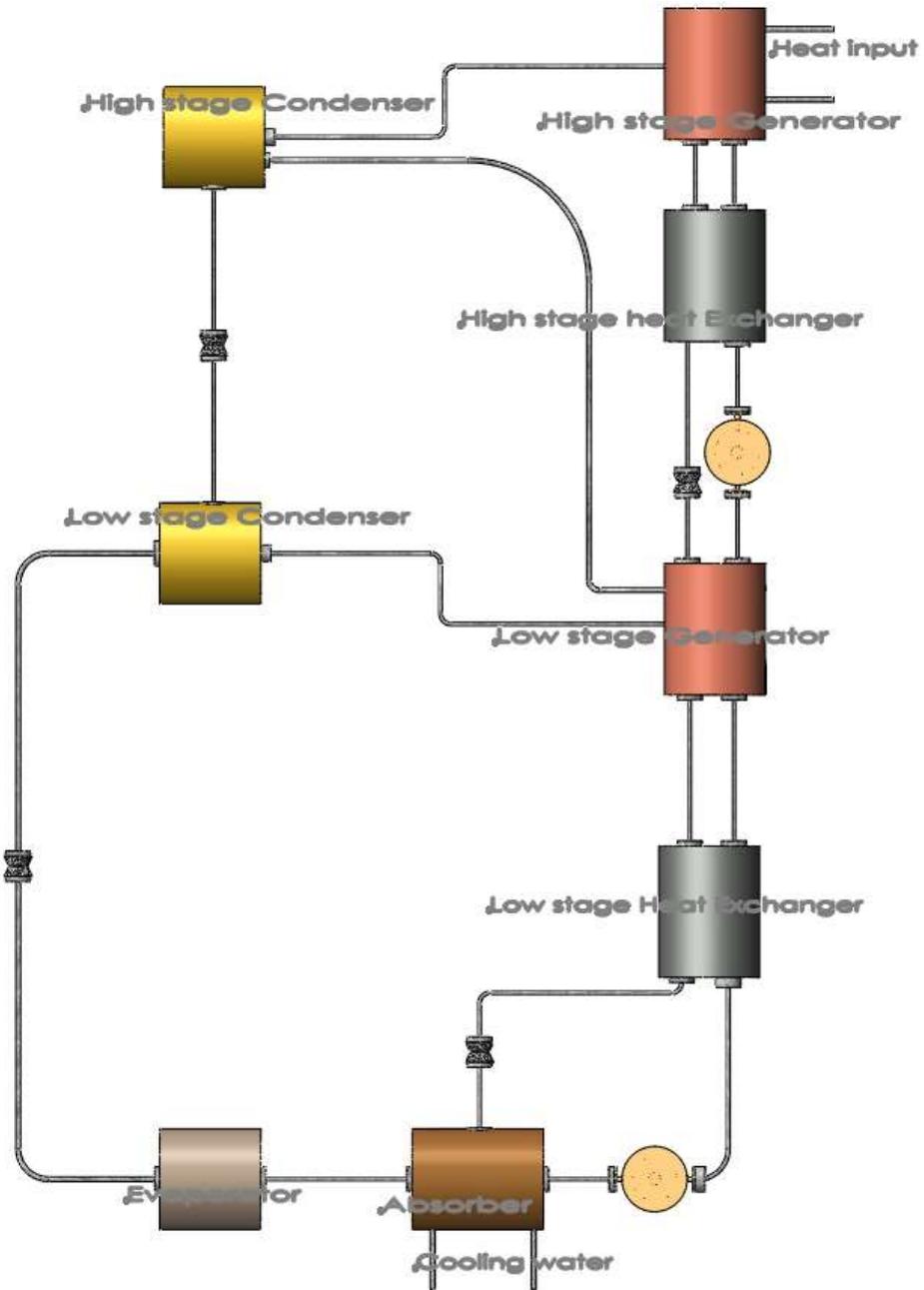


Figure 16 Schematic of a double effect lithium bromide absorption refrigeration cycle

## **Chapter 4**

# **Thermodynamic Modeling and Organic fluids**

### **4.1 Cycle-Tempo**

Cycle tempo is a thermodynamic modeling program developed by Delft University of Technology for optimization of energy conservation systems. The program is very user friendly and is the main software program used in this project.

Cycle Tempo is used to model and optimize systems and cycles for the production of electricity, heat, and refrigeration. These systems are designed with various interconnected components and hence form a complex network of mass and energy flow. This software is mainly used to determine the mass and energy flow of the system. The program contains a large number of apparatuses and working fluids, with which one can model the desired cycle. This gives an opportunity to vary the existing configuration and calculate the efficiency or other parameters that interests the user.

The Cycle Tempo provides a graphical user interface, which makes it easier for the user and also makes the modeling more interactive. The parameters for each component like pressure losses and temperature can be input by double clicking on the respective component. The program processes calculation into a complete collection of data that you need for detailed assessment of the properties of the modeled system. The results can be viewed in user friendly tables and graphs.

## **4.2 Organic Fluids**

From preliminary study, the working fluids that appear to be options for the organic Rankine cycle operating in the evaporating temperature range from 212° to 392° F. The final selection will be made by considering the efficiency of the cycle, cost, and also environmental impact.

Refrigerants are organic fluids that are the most suitable for Organic Rankine Cycle. A refrigerant can be defined as a fluid that has a relatively lower boiling point at atmospheric pressure making it suitable to be used in refrigerating cycle or directly as ice for cooling. Carbon dioxide, ammonia, and water are among some of the natural refrigerants. CFCs, HCFCs, and HFCs are a group of refrigerants that can be classified as synthetic fluids.

### **Water**

Water is used as a refrigerant in lithium bromide absorption chillers. Since water is non-toxic, non-flammable, and abundant it is the most environmental friendly refrigerant. However, since the boiling point of the water is 212°F at atmospheric pressure it has to be maintained at a lower pressure to make it work as a refrigerant.

### **Ammonia**

Ammonia is a natural refrigerant and considered to be one of the most efficient. It is used in both vapor compression cycles and ammonia water absorption chillers as a refrigerant. Ammonia is commonly used in industrial applications and has been in use since it was identified as a refrigerant 100 years ago.

## **Hydrocarbons**

R600a, R601 and R601a are among the common hydrocarbons that are used in vapor compression cycles and have been considered as an option for the Organic Rankine Cycle. An organic compound is made of hydrogen and carbon atoms and is commonly found in crude oil.

## **Chlorofluorocarbons**

Chlorofluorocarbons are the organic compounds that contain carbon, chlorine, hydrogen and fluorine. R12, R113 and R114 are the CFC refrigerants considered for this project. Production of CFCs was phased out by the Montreal Protocol in many countries, but many other countries across the world still produce and use these substances. The CFCs are stable, safe, non-flammable, and efficient. The only problem being the damage it can cause to the ozone.

## **Hydro-chlorofluorocarbons**

HCFCs are a large group of organic compounds, whose structures closely resemble that of CFCs. In addition HCFCs includes one or more hydrogen atoms. They are fairly stable and unreactive. Like the CFCs the use of HCFCs will be phased out in some countries. R142b is the HCFC considered for this project.

## **Hydro-fluorocarbons**

HFCs are relatively new refrigerants and considered to be a replacement for phased out refrigerants. HFCs are supposed to have no ozone depletion potential. HFCs are also used for foam blowing, solvent cleaning, and fire extinguishers. HFCs are a group of man-made

chemicals containing the elements carbon, hydrogen, and fluorine. They are colorless, odorless, and unreactive gases. R134a is the HFC considered for this project.

### 4.3 Refrigerant Properties

Properties such as lower toxicity, no flame propagation, and environmental factors are important components. However, selecting a working fluid for Organic Rankine Cycle goes far beyond just these properties. Power output and Refrigeration capacity are an important and primary concern.

#### Toxicity and Flammability

Toxicity is among the few safety factors to be considered in the selection of the working fluid. Any substance can be lethal in the case of over dosage.

Flammability is another key parameter in evaluating the safety level of the refrigerant. Most substances can burn given the right circumstances. Class 1 in the figure to the right indicates refrigerants that do not show flame propagation when tested in air at 14.7 psia and 70°F (9). Class 2 and Class 3 signify refrigerants with lower and higher flammability respectively. Where A indicates lower toxicity and B indicates higher toxicity. Class A1 would be the best choice of working fluid to be used in various applications as it signifies refrigerants with lower toxicity and do not show flame propagation.

	Lower Toxicity	Higher Toxicity
Higher Flammability	A3	B3
Lower Flammability	A2	B2
No Flame Propagation	A1	B1

Figure 17 Toxicity and Flammability chart

### **Ozone Depletion Potential**

ODP is an index to indicate the ability of the fluid to damage ozone. A value of 1 is given to refrigerants R-11, R12 and R114 which have the highest potential to damage ozone. While the ODP does not play any role in the working of the fluid, it is a major factor to be considered in countries subjected to Montreal Protocol. Any new refrigerant developed should have an ODP of zero in these countries.

### **Global Warming Potential**

GWP is a measure of a substance ability to be a greenhouse gas. The GWP is relative to the warming effect of a similar mass of carbon dioxide for a 100 year time frame (9). GWP is measured in comparison to the GWP of CO<sub>2</sub>, set to unity. Although some refrigerants can reach a GWP value as high as 1000, there is no legislation restricting the use of high GWP fluids.

### **Critical Point**

Critical point is the point on a pressure-enthalpy diagram where the properties of the liquid and vapor refrigerant meet and become indistinguishable. The temperature, density, and composition of the substance are the same for a liquid as they are for a gas at this point. When operating at conditions above the critical point no separate liquid phase will be possible. Cycles requiring condensation will be hard to operate and also reduces the efficiency of the cycle.

### **Material Compatibility**

The working fluids need to be checked for material compatibility before being used in real world applications. It would affect the cost of the equipment the working fluid comes in contact with. CFCs are the class of refrigerants that are stable and inert. The stability of the CFCs led to their downfall since they have a very long atmospheric lifetime.

Table 1 Refrigerant general and environmental information

Type	Refrigerant Number	Chemical Name	Chemical Formula	Safety Group	Atmospheric Lifetime(yr.)	ODP	GWP
HC	-	Benzene	C <sub>6</sub> H <sub>6</sub>	-	-	-	-
HC	-	Toluene	C <sub>7</sub> H <sub>8</sub>	-	-	-	-
Alcohol	-	Ethanol	CH <sub>3</sub> CH <sub>2</sub> OH	-	-	-	-
HFC	245fa	1,1,1,3,3-Pentafluoropropane	CF <sub>3</sub> CH <sub>2</sub> CHF <sub>2</sub>	B1	7.6	0	1,030
CFC	113	1,1,2-Trichlorotrifluoroethane	CCl <sub>2</sub> CF <sub>2</sub> CF <sub>2</sub>	A1	85	0.8	6,130
HFC	134a	1,1,1,2-Tetrafluoroethane	CH <sub>2</sub> FCF <sub>3</sub>	A1	14	0	1,430
HC	601	n-pentane	CH <sub>3</sub> CH <sub>2</sub> CH <sub>2</sub> CH <sub>2</sub> CH <sub>3</sub>	A3	15	0	20
HC	601a	Iso-pentane	(CH <sub>3</sub> ) <sub>2</sub> CHCH <sub>2</sub> CH <sub>3</sub>	A3	15	0	20
HC	600a	Iso-butane	CH(CH <sub>3</sub> ) <sub>2</sub> CH <sub>3</sub>	A3	15	0	20
HCFC	142b	1-Chloro-1,1-difluoroethane	C <sub>2</sub> H <sub>3</sub> ClF <sub>2</sub>	A2	17.9	0.06	2,400
CFC	12	Dichlorodifluoromethane	CCl <sub>2</sub> F <sub>2</sub>	A1	100	1	10,900
CFC	114	1,2-Dichlorotetrafluoroethane	CClF <sub>2</sub> CClF <sub>2</sub>	A1	300	1	10,000
-	717	Ammonia	NH <sub>3</sub>	B2	-	0	1
-	718	Water	H <sub>2</sub> O	A1	-	0	0

Table 2 Refrigerant operational information

Refrigerant	Molecular Mass	T <sub>critical</sub>		P <sub>critical</sub>		Normal Boiling Point		Freezing point		E <sub>evap</sub> (Btu/lb)
		°F	°C	Torr	Bar	°F	°C	°F	°C	
Benzene	78.11	553.1	289.5	36,200	48.26	176.2	80.1	42	5.5	168.0
Toluene	92.14	318.6	159.2	30,797	41.05	110.7	43.7	-139	-95	151.0
Ethanol	46.07	465.5	240.8	47,254	63.0	173.0	78.3	-173	-113.8	364
R245fa	134.0	309.2	154.0	27,302	36.39	59.5	15.3	-160	-106.6	84.6
R113	187.4	417.4	214.1	25,427	33.89	117.7	47.6	-31	-35	167.2
R134a	102.0	214.0	101.1	30,452	40.59	-15.0	-26.1	-142	-96.6	100.9
R601	72.15	385.9	196.6	25,277	33.69	97.0	36.1	-200	-128.8	153.6
R601a	72.15	369.0	187.2	25,352	33.79	82.0	27.7	-254	-158.8	146
R600a	58.10	274.5	134.7	27,227	36.29	10.9	-11.7	-173.2	-114	183.3
R142b	100.5	278.8	137.1	30,902	41.19	15.4	-9.2	-202.7	-130.4	92.4
R12	120.9	233.5	111.9	31,022	41.36	-21.5	-29.7	-250.6	-157	75.3
R114	170.9	294.2	145.6	24,429	32.57	38.4	3.55	-137.4	-94.1	60.0
Ammonia	17.0	270.0	132.2	85,004	113.3	-27.9	-33.2	-108.0	-77.7	589.5
Water	18.0	705.1	373.9	165,487	220.6	211.9	99.9	32.0	0	971.6

## Chapter 5

# Case Study for Organic Rankine Cycle

### 5.1 Design Specifications

The following are design specifications for the steam jet vacuum system under consideration. The data collected from the facility are as follows:

#### Capacity

- The absolute pressure to be maintained:  
Finishing vessel = 0.29 Torr  
UFPP vessel = 9 Torr
- Composition of the gas to be entrained: steam and air  
The 550 lb/hr mixture consists of 250 lb/hr air and 300 lb/hr steam at 40° F.
- Discharge pressure in the condenser: 400 Torr

#### Supply Steam

- Steam line pressure: 181.7 psig
- Steam temperature: 410° F
- Flow rate: 6,166 lb/hr

#### Mixture temperature and flow rate to the condenser

It is difficult to measure accurately the temperature of the vapor entering the condenser. For a mixture of air and steam the temperature shall be calculated by using the following equation (4).

$$t_m = \frac{W_s C_{ps} t_s + W_a C_{pa} t_a + W_{ms} C_{mps} t_{ms}}{W_s C_{ps} + W_a C_{pa} + W_{ms} C_{mps}} \quad (6)$$

- $t_m$  = Temperature of the mixture °F  
 $W_s$  = Suction steam flow rate, pounds per hour  
 $C_{ps}$  = Specific heat of suction steam at constant pressure corresponding to downstream absolute pressure (0.45 Btu/lbm °F)  
 $t_s$  = Temperature of the suction steam on the downstream side of the nozzle  
 $W_a$  = Air flow rate, pounds per hour  
 $C_{pa}$  = Specific heat of air at constant pressure (0.24 Btu/lbm °F)  
 $t_a$  = Temperature of the air °F  
 $W_{ms}$  = Motive Steam flow rate, pounds per hour  
 $C_{mps}$  = Specific heat of the motive steam at constant pressure corresponding to downstream absolute pressure  
 $t_{ms}$  = Temperature of the motive steam on the downstream side of the nozzle.

The presence of air in the steam causes the temperature on the surface of the heat exchangers to be lower than expected. The heat transfer will be lower and the system efficiency will be reduced. The temperature of the mixture calculated by using the above formula is as given below

$$t_m = \frac{(6166 \times 410 \times 0.45) + (250 \times 0.24 \times 40) + (300 \times 40 \times 0.45)}{(6166 \times 0.45) + (250 \times 0.24) + (300 \times 0.45)} = 385^\circ F$$

Condenser inlet temperature: 385°F

Outlet temperature: 75-77°F

Flow rate considered: 6,466lb/hr

## Cooling water

The overall energy balance can be found the following equations (6),

$$q = \dot{m}_h (I_{h,i} - I_{h,o}) \quad (7)$$

$$q = \dot{m}_c c_p (T_{c,o} - T_{c,i}) \quad (8)$$

Where, I represent fluid enthalpy, T is the temperature and  $c_p$  the specific heat. The subscripts h and c refer to the hot and cold fluids, whereas i and o designate the fluid inlet and outlet conditions.

The required heat transfer can be obtained from the overall energy balance as follows,

$$q_h = 6466 \text{ lb/hr} (1216 \text{ Btu/lb} - 45.3 \text{ Btu/lb}) = 7.569 \text{ MMBTUhr}$$

$$q_c = 7.569 \text{ MMBTUhr} = \dot{m}_c (45.3 \text{ Btu/lb} - 33.4 \text{ Btu/lb})$$

$$\dot{m}_c = \frac{7.569 \text{ MMBTUhr}}{11.9 \text{ Btu/lb}} = 636,050 \text{ lb/hr} = 1,272 \text{ gal/min} = 80 \text{ kg/s}$$

Temperature inlet: 65°F

Flow rate: 636,050 lb/hr

Figure 18 below gives a screen shot of the steam jet vacuum system control program that the facility uses.

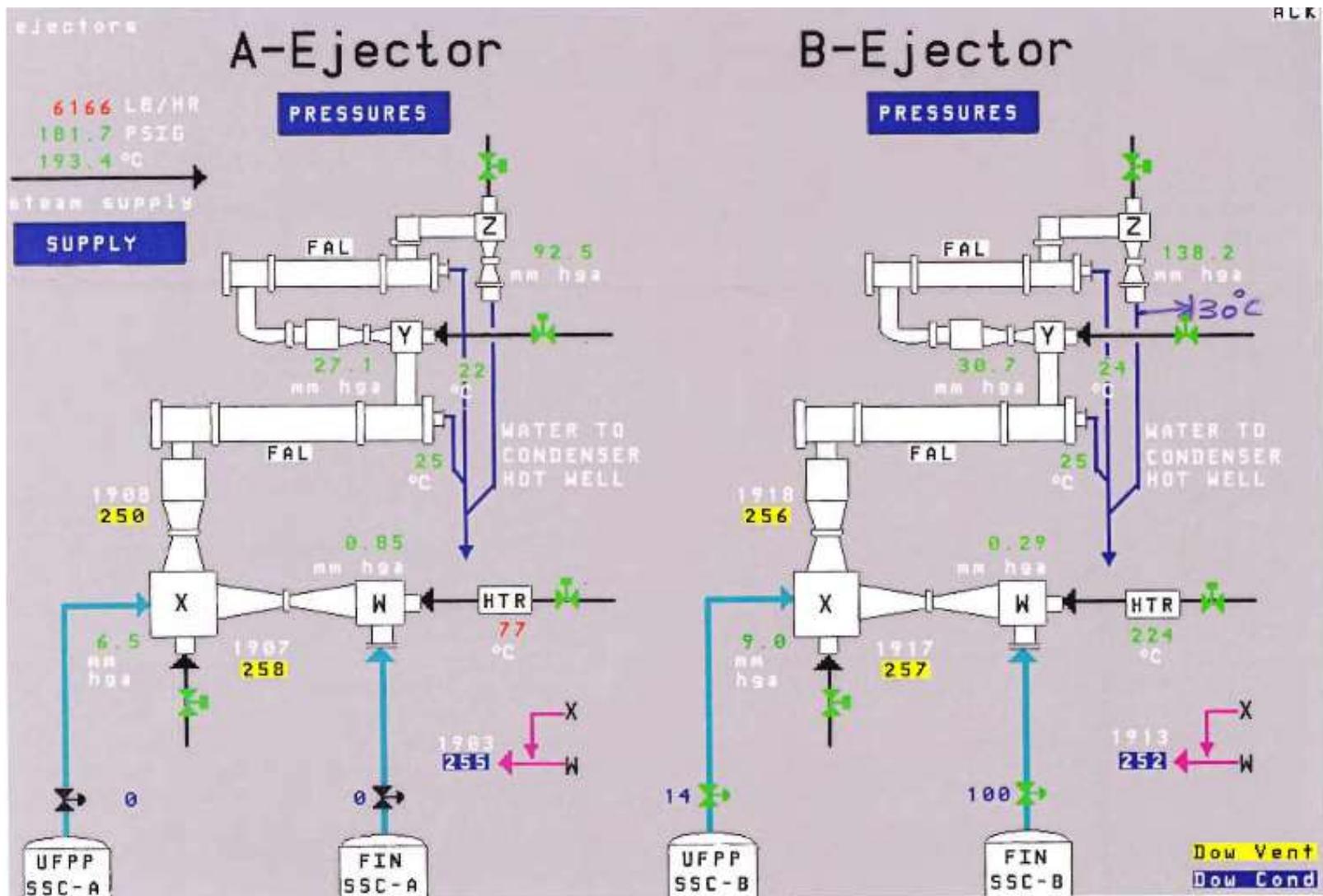


Figure 18 Control Parameters of the Steam Jet Vacuum System

## 5.2 Example Calculation for Organic Rankine cycle

Since the idea here is to replace the cooling water supplied to the condenser of a Steam Jet Vacuum System with an organic fluid and generate electricity using an Organic Rankine Cycle, a number of fluids will be investigated in the following case study. As an example and also to check the credibility of cycle-tempo, the electricity that can be generated by replacing water with R134a will be manually calculated,

### Carnot cycle

The cycle with maximum efficiency is a Carnot cycle. The Carnot efficiency of the cycle operating between 385° F and 86° F, a typical maximum cooling water temperature is

$$\eta_{Carnot} = 1 - \frac{T_s}{T_H} = 1 - \frac{545.67^\circ R}{844.67^\circ R} = 0.353 = 35.3\%$$

The T-s diagram of a simple R134a-Rankine cycle whose boiler is replaced with a condenser of the Steam Jet Vacuum system is given in

Figure 19. R134a enters the heat source at 86°F and 27 bar and is vaporized to about 203°F. After expansion in the turbine, it is condensed in the ORC-condenser at a pressure of 7.9 bar. The calculation for this cycle will be based on steady operating conditions and the irreversibilities are neglected.

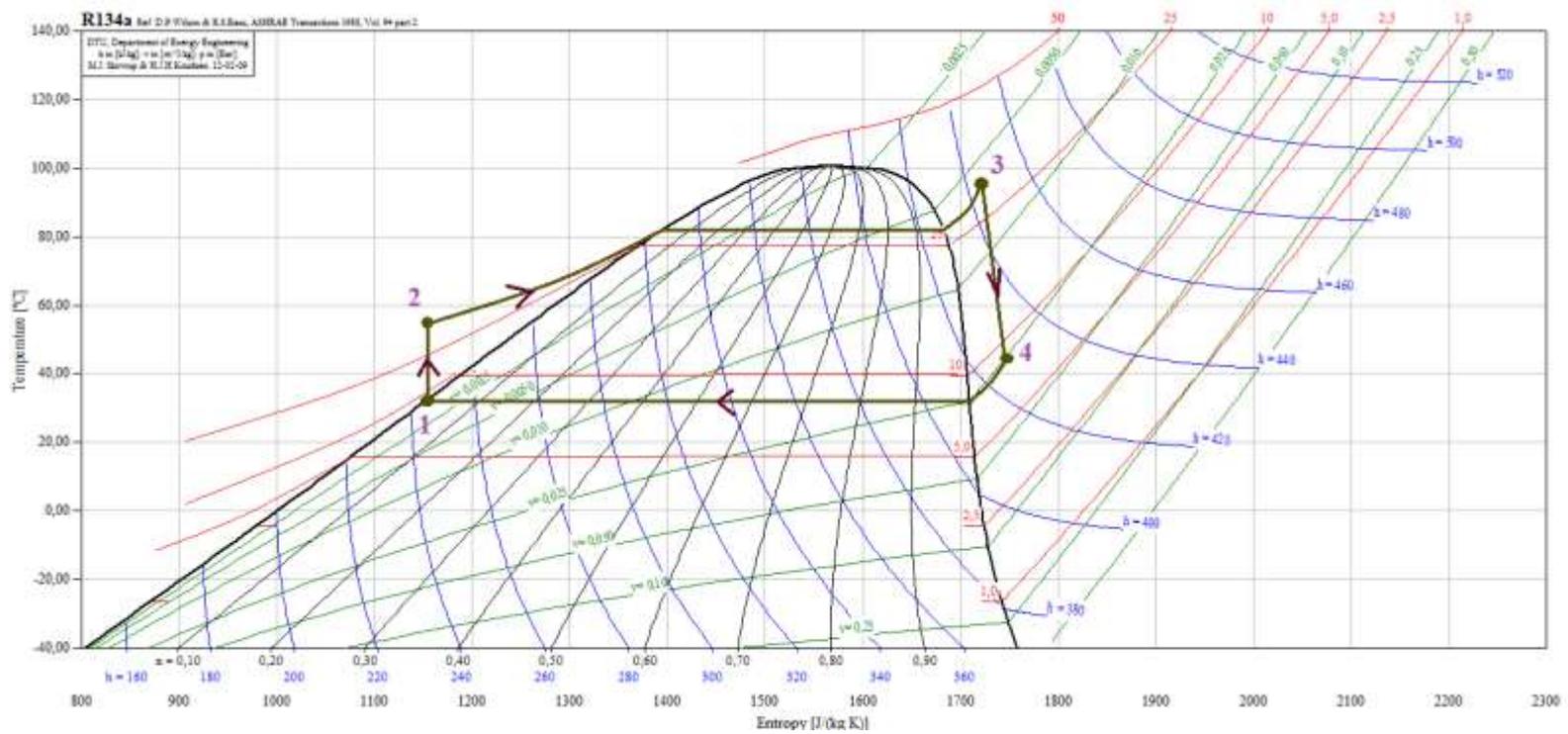


Figure 19 T-s Diagram For R134a

- 1-2: Compression in the pump
- 2-3: Heat addition in the boiler
- 3-4: Expansion in the turbine
- 4-1: Heat rejection in the condenser

First we determine the enthalpies at various points in the cycle, using data from the R134a saturation curve.

$$\begin{aligned} \text{State 1: } P_1 = 7.9 \text{ bar} & & h_1 @ 86^\circ\text{F} = 40 \text{ Btu/lb} \\ & & v_1 = 0.000843 \text{ m}^3/\text{kg} \end{aligned}$$

Where,  $v_1$  is the specific volume.

$$\text{State 2: } P_2 = 27 \text{ bar}$$

$$W_{\text{pump,in}} = \frac{v_1(P_2 - P_1)}{\eta_p} = \frac{0.000843 \times (2700 - 790)}{0.75} = 2.14 \text{ kJ/kg} = 0.92 \text{ Btu/lb}$$

$$h_2 = h_1 + W_{\text{pump,in}} = (40 + 0.92) \text{ Btu/lb} = 40.92 \text{ Btu/lb}$$

$$\begin{aligned} \text{State 3: } P_3 = 27 \text{ bar} & & h_3 = 126 \text{ Btu/lb} \\ T_3 = 203^\circ\text{F} & & s_3 = 0.225 \text{ Btu/lb } ^\circ\text{F} \end{aligned}$$

$$\begin{aligned} \text{State 4: } P_4 = 7.9 \text{ bar} & & \text{saturated vapor} \\ & & h_4 = 117.5 \text{ Btu/lb} \end{aligned}$$

The energy input and output is given by,

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = (126 - 40.92) \text{ Btu/lb} \\ &= 85.08 \text{ Btu/lb} \end{aligned}$$

$$\begin{aligned} q_{\text{out}} &= h_4 - h_1 = (117.5 - 40) \text{ Btu/lb} \\ &= 77.5 \text{ Btu/lb} \end{aligned}$$

The efficiency of the cycle is given by,

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{77.5}{85.08} = 8.9\%$$

The Work produced by the turbine considering the isentropic efficiency,

$$\begin{aligned}W_{\text{turb,out}} &= \eta_T W_{s,\text{turb,out}} \\&= \eta_T (h_3 - h_4) \\&= 0.85 \times (126 - 117.5) \text{ Btu/lb} \\&= 7.225 \text{ Btu/lb}\end{aligned}$$

The net power generated by the cycle is,

$$\begin{aligned}w_{\text{net}} &= W_{\text{turb,out}} - W_{\text{pump,in}} \\&= 7.225 - 0.92 \text{ Btu/lb} \\&= 6.305 \text{ Btu/lb}\end{aligned}$$

Considering the flow rate calculated by the program the power generated is given by,

$$\begin{aligned}W_{\text{net}} &= m (w_{\text{net}}) \\&= 82,937 \text{ lb/hr} \times 6.305 \text{ Btu/lb} \\&= 522,917 \text{ Btu/hr} \\&= 153 \text{ kW}\end{aligned}$$

### **5.3 Cycle Tempo analysis of various working fluids**

The efficiency of the Organic Rankine Cycle is greatly dependent on the operating parameters of the cycle. The temperature and pressure of the vapor entering the turbine are the main parameters that influence the cycle. Since the maximum temperature that can be attained is limited to that of the source temperature by the laws of thermodynamics, there is a limit on the power output that can be retrieved from the project.

Each model has been created to discover the maximum power output that can be extracted from an ORC, i.e. the main comparison point for the cycles is power output, but the final selection will be based on a number of other factors. The models are close to actual operating

conditions, the pinch point for any of the components is no less than 40°F. The best operating parameters were set by varying the input and analyzing output data multiple times.

The cycle has been analyzed for twelve fluids. The fluids were selected based on studies conducted previously regarding ORC units; some of these fluids are being used in commercial ORC units. All the models analyzed are presented in Figures 21 to 36. The cooling source is held constant throughout the analysis and the exit temperature of the energy source is set to a certain best point for that particular model. The ASHRAE Fundamentals handbook and also a program RefProp was used to evaluate the thermodynamic properties of the organic fluids.

Most of the ORC units are custom built and the irreversibilities involved in the units are reduced to a large extent by insulation and recent manufacturing technologies. The isentropic efficiencies considered for the pump, turbine and generator in all the cases is held constant and is given by 75%, 85%, and 85% respectively.

From the above theoretical calculations, we can note that the models do not violate any physical laws. From the above comparison between Carnot efficiency of 35.3% to that of an ORC-R134a of 8.9%, we can see that the ORC-R134a doesn't violate any physical laws.

From all the analysis below the maximum power that could be generated does not exceed 150 kW; hence the system should be sized respectively. It was noted during the analysis that higher boiler pressure leads to a higher thermal efficiency.

The analysis models were developed based on the example models available with the Cycle Tempo program. Each refrigerant was checked for a simple cycle and a regenerative Rankine cycle. The regenerative Rankine cycle involves recovering heat from the exit mixture of the turbine.

### 5.3.1 Procedure for modeling

The components required for the cycle were selected from the pallets and assembled on the graphical interface. The components were connected with pipes containing the desired working fluid. Next, the conditions in each apparatus and pipe were set to values as accurately as possible to the actual operating conditions. The common conditions set were the pinch point, maximum exit temperature, pressure, and efficiencies of the components.

In the Cycle Tempo models above and also in the rest of the ORC models, the content in the pipes is defined in figure 20.

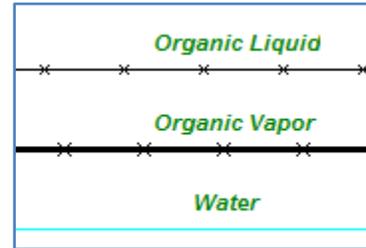


Figure 20 Flow through pipes

The heat source is the condenser of the steam jet vacuum system. Components 7 and 8 above under the heat source are source and sink respectively. The condenser was modeled as a simple heat exchanger. The only flow rate specified is for the heat source in all the cycles. All other flow rates are optimized by the cycle tempo program.

The cooling system operates under its own loop. The pressure for the cooling source was set to 2 bar, about 29 psig at random. This has no effect on the cycle.

After all the operating parameters were set, the program was run and results were calculated as indicated in Figures 21-36 The results can also be presented in the form of tables. Tables 3-6 below show the output obtained from R134a cycle. The output file is a text file that contains the calculated mass flow rates, energy balance, and efficiencies of the components.

Table 3 Data for Pipes

Pipe No.	Medium	Mass flow [kg/s]	Pressure [bar]	Temperature [°C]	Enthalpy [kJ/kg]	Entropy [kJ/kg.K]	Quality [%]
1	R134a	10.475	27	95	28.46	-0.1495	100
			27	94.39	27.66	-0.1517	100
2	R134a	10.475	27.1	31.78	-181.7	-0.7594	0
			27	31.78	-181.7	-0.7594	0
3	R134a	10.475	8	44.37	5.66	-0.1394	100
			7.9	44.21	5.66	-0.1385	100
4	R134a	10.475	7.8	30	-183.93	-0.7613	0
			7.7	29.95	-183.93	-0.7612	0.04
5	Water	79.26	1.5	24.01	100.84	0.3533	0
			1.5	24.01	100.84	0.3533	0
6	Water	79.26	1	18	75.64	0.2678	0
			0.9	18	75.64	0.2679	0
7	Steam	0.815	0.5332	196	2869.79	8.1126	100
			0.5332	196	2869.79	8.1126	100
8	Water	0.815	1	40	167.62	0.5724	0
			1	40	167.62	0.5724	0
9	Water	79.26	2	18.01	75.79	0.268	0
			2	18.01	75.79	0.268	0

Table 4 Heat Balance for Equipment

App. no.	Name	Low end temperature diff. [K]	High end temperature diff. [K]	Transmitted heat flow [kW]
4	Condenser	11.99	6.83	1986.02
2	ORC-Heat Source	8.22	101	2201.46

Table 5 Equipment Efficiencies

App. no.	Name	Isentropic efficiency [%]	Mechanical efficiency [%]
3	Turbine	85	75
1	Pump	75	85.34
6	Pump	75	84.75

Table 6 Net Power Output

Delivered	No.	Apparatus	Energy [kW]	Totals [kW]
Delivered gross power	1	Generator	146.92	146.92
Aux. power consumption	1	Pump	27.43	41.26
	6	Pump	13.83	
Delivered net power				105.66

Table 3 above shows the data for each pipe within the cycle. There are two rows for each pipe. The first row provides details for the inlet of that pipe and the second row deals with the outlet of the pipe. Table 5 shows the mechanical efficiencies of the cycle calculated by the program while the isentropic was set beforehand. Table 6 above we can see the net power produced after deducting the power consumed by the ORC and Cooling system pumps.

Table 7 below gives a detail of the power output and mass flow rates calculated from the models for each refrigerant considered. Figures 21-36 show the model results for the refrigerants considered, that are summarized in Table 7 . The net power output listed is obtained after deducting the power consumed to operate the pumps

Table 7 Results Compiled from the Analysis of Organic Fluids

Refrigerant	Cycle Type	Organic Fluid-Flow Rate(kg/s)	Cooling water-Flow Rate(kg/s)	Auxiliary power consumption (kW)		Delivered Gross Power (Generator-kW)	Net Power Delivered (kW)
				ORC-pump	Cooling Pump		
Benzene	Simple	0.003	0.285	0.24	1.26	7.6	6.1
	Regen	0.232	4.596	0.25	1.02	8.0	6.8
Toluene	Simple	0.146	3.241	0.12	0.79	3.7	2.8
	2stage	0.15	3.321	0.13	0.81	3.8	2.9
	Regen	0.15	3.089	0.15	0.69	4.1	3.3
Ethanol	Simple	0.152	6.323	0.12	1.55	2.9	1.2
	2stage	0.147	6.128	0.12	1.5	2.8	1.2
	Regen	0.152	6.114	0.11	1.36	3.1	1.6
R245fa	Simple	8.542	72.354	7.46	12.63	120.3	100.2
	Regen	9.08	72.033	7.9	11.52	125.4	105.9
R113	Simple	0.781	5.714	0.38	1.4	8.7	7.0
	2stage	0.964	7.059	0.47	1.71	10.8	8.6
	Regen	0.733	5.377	0.92	1.2	14.9	12.7
R134a	Simple	10.475	79.26	27.43	13.83	146.9	105.7
	Regen	10.604	78.487	28.71	12.47	148.7	107.6
R601	Simple	4.6	77.736	3.97	13.56	117.8	100.2
	Regen	4.992	77.737	4.77	12.35	128.7	111.6
R601a	Simple	5.129	78.650	5.17	13.73	125.1	106.2
	2stage	5.622	86.214	5.77	15.03	137.1	116.3
	Regen	5.461	76.976	6.0	12.23	137.1	118.8
R600a	Simple	5.357	79.194	15.36	13.81	127.6	98.4
	Regen	5.536	78.267	16.57	12.43	144.0	115.0
R142b	Simple	8.586	78.84	12.36	13.75	127.9	101.8
	Regen	9.057	78.228	13.9	12.43	143.3	117.0
R12	Simple	14.398	79.581	24.08	13.89	137.9	100.0
	Regen	14.173	78.76	17.6	19.5	140.3	103.2
R114	Simple	12.946	80.121	2.83	13.98	94.9	78.1
	Regen	14.671	79.752	10.13	12.67	107.7	84.9

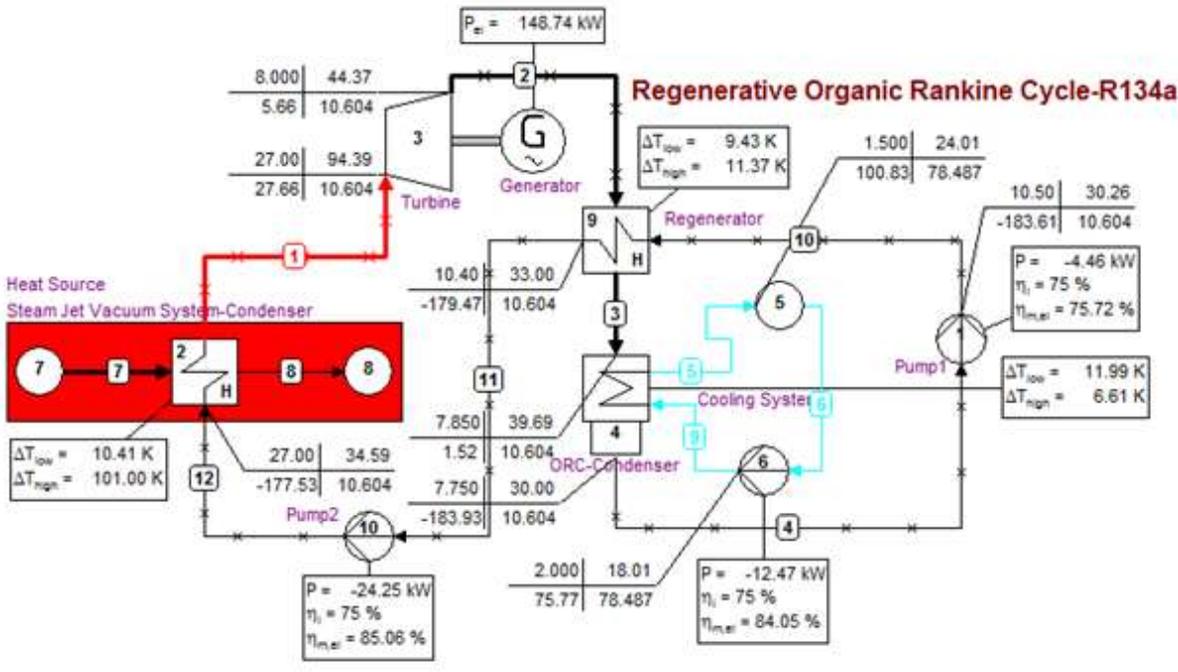
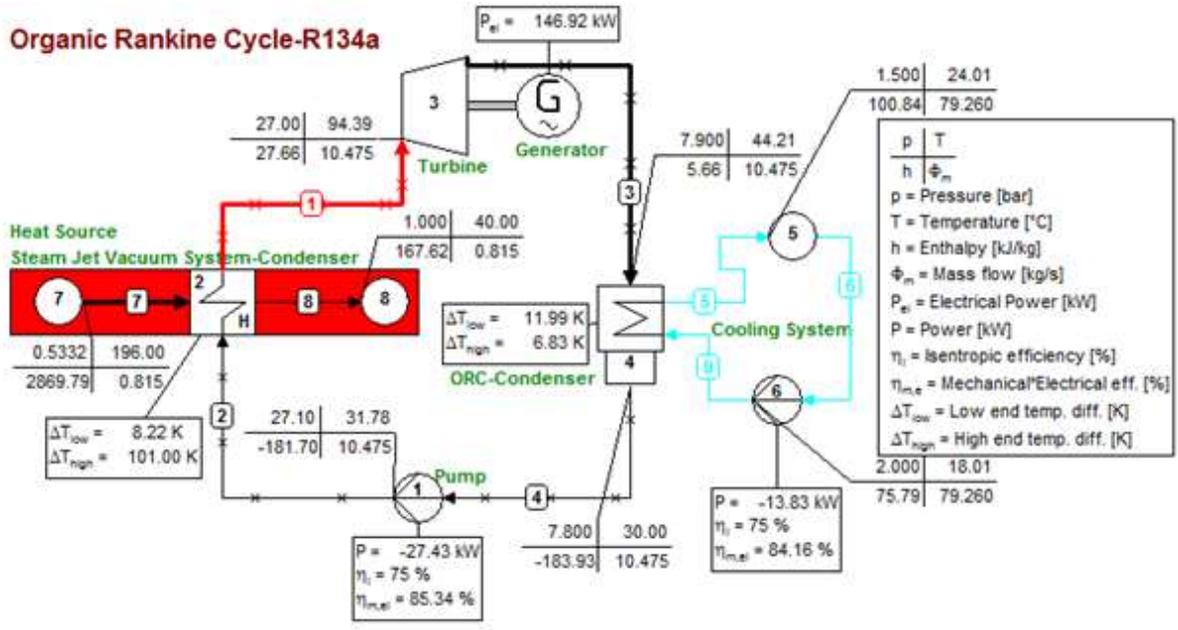


Figure 21 Cycle Tempo Analysis models for R134a

## Organic Rankine Cycle-Benzene

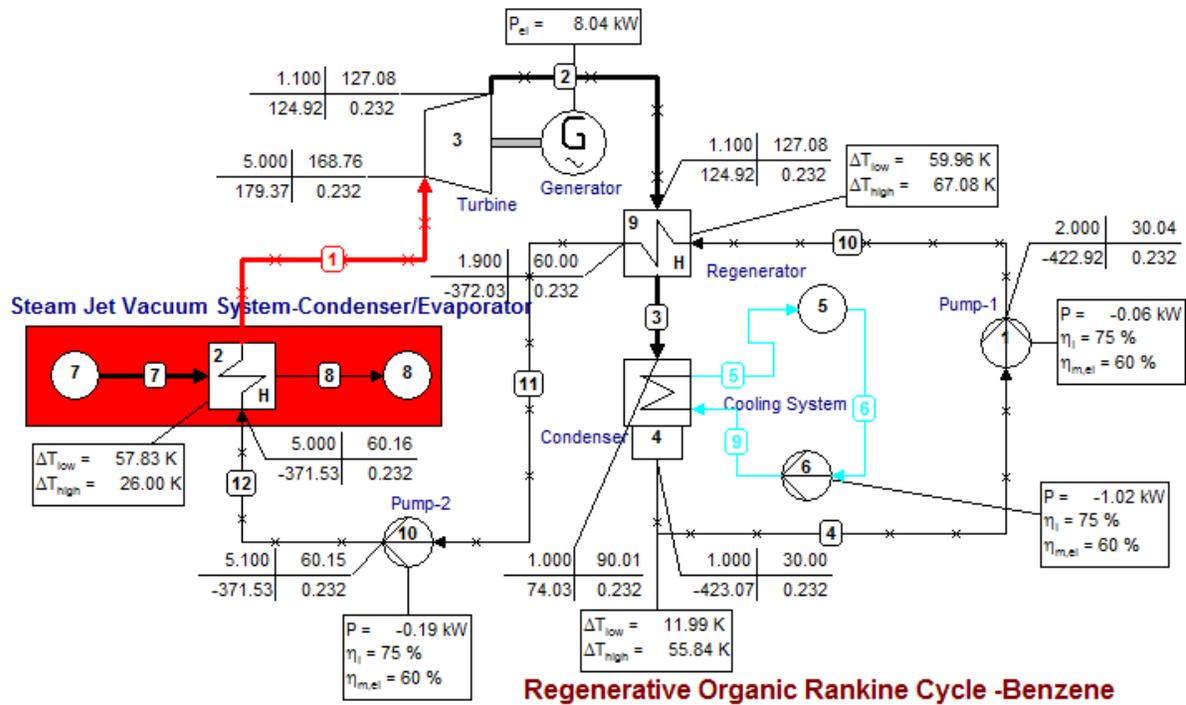
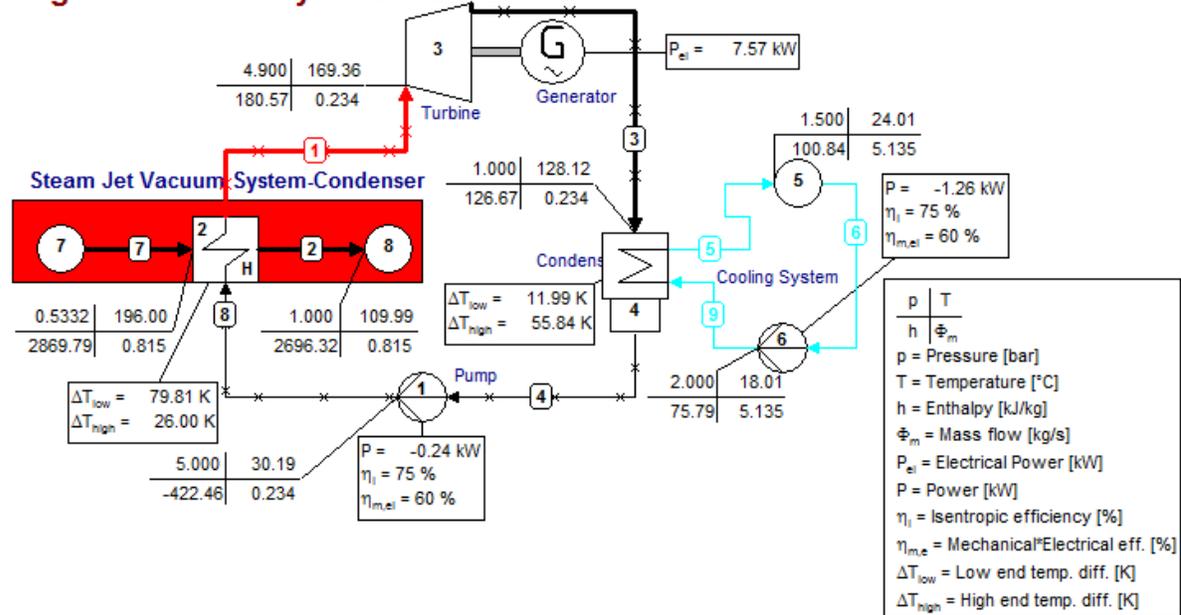
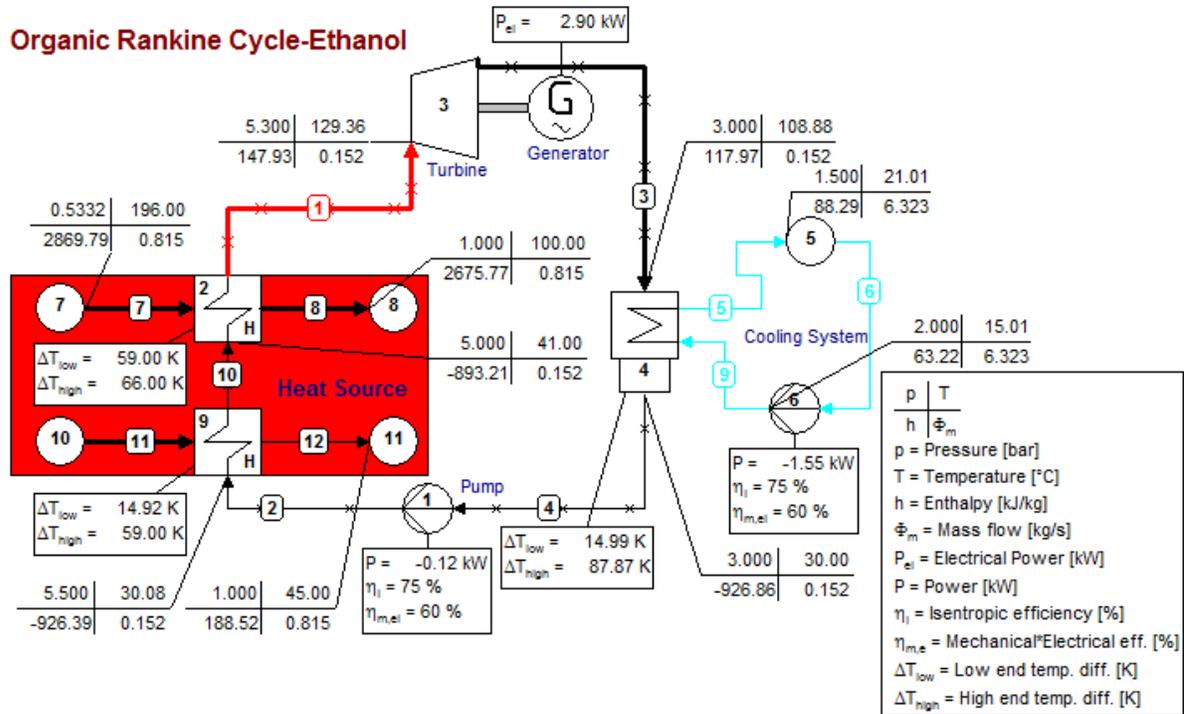


Figure 22 Cycle Tempo Analysis models for Benzene

### Organic Rankine Cycle-Ethanol



### Organic Rankine Cycle-Ethanol

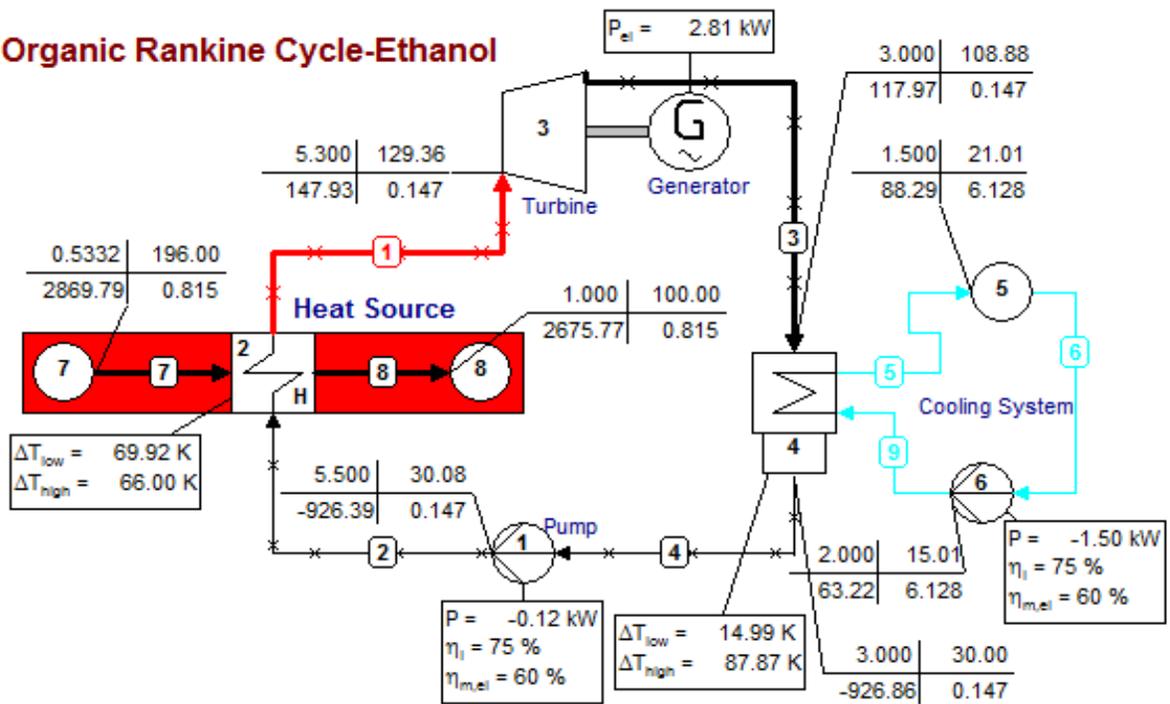


Figure 23 Cycle Tempo Analysis models for Ethanol

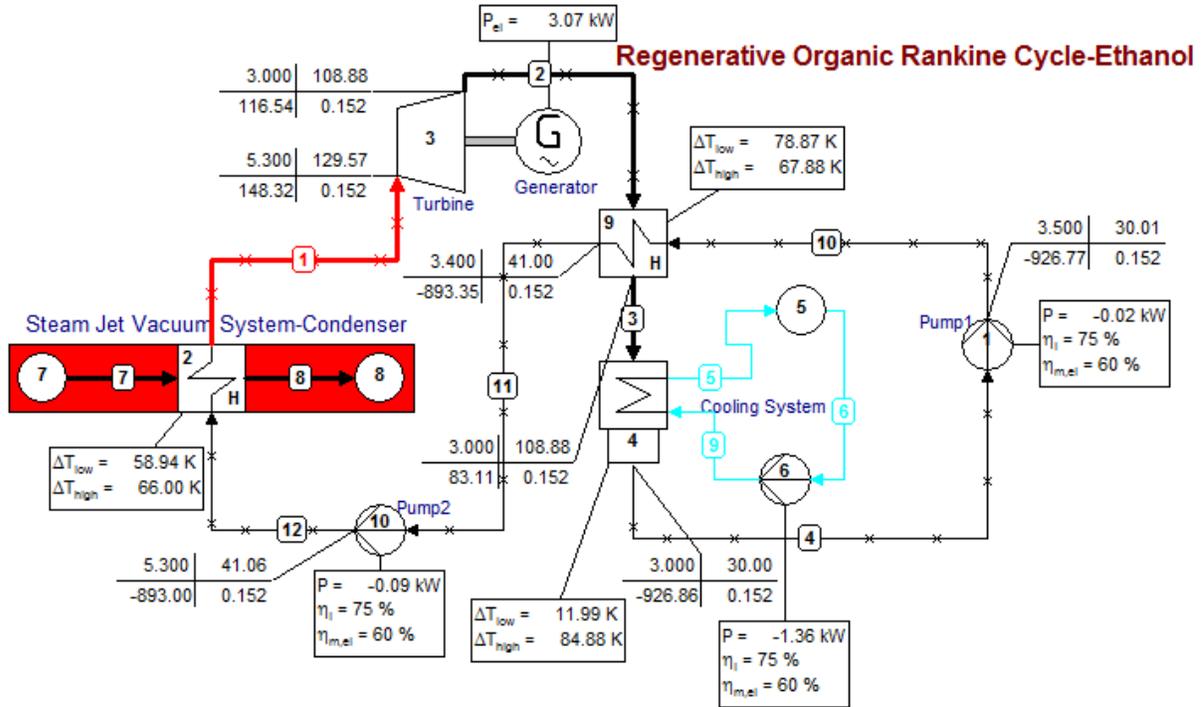


Figure 24 Cycle Tempo Analysis model for Ethanol

The staging of condensers helps to have a lower pinch point i.e. the steam condensed into water will in turn not be heated by the cooling liquid, which is heated up while condensing steam. Staging boilers/condensers is very similar to a regenerative cycle; the working fluid is being preheated before going into the final stage of the boiler.

The stage 2 of the boiler of ORC/condenser of Steam Jet Vacuum System is at higher operating temperatures compared to stage 1. Stage 2 is operating between 196°C and 100°C, while stage 1 is operating between 100°C and 45°C. Here, boiler of ORC/condenser of Steam Jet Vacuum System is the heat source.

This result in an increase in the flow rate of the working fluid and also an increase in power generated.

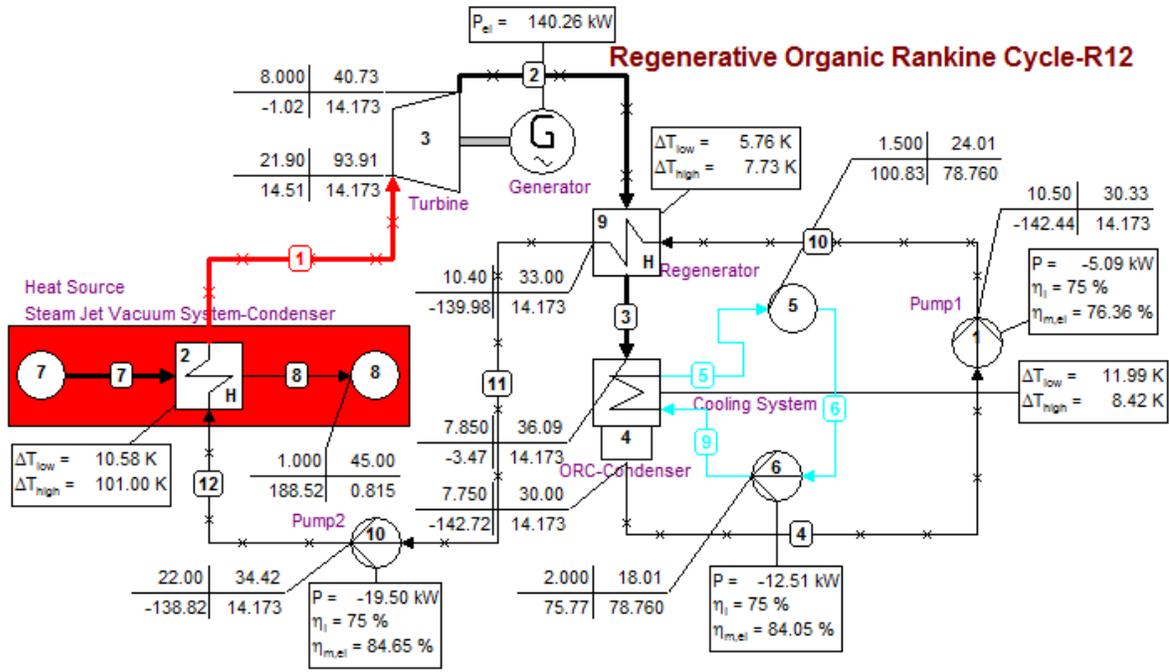
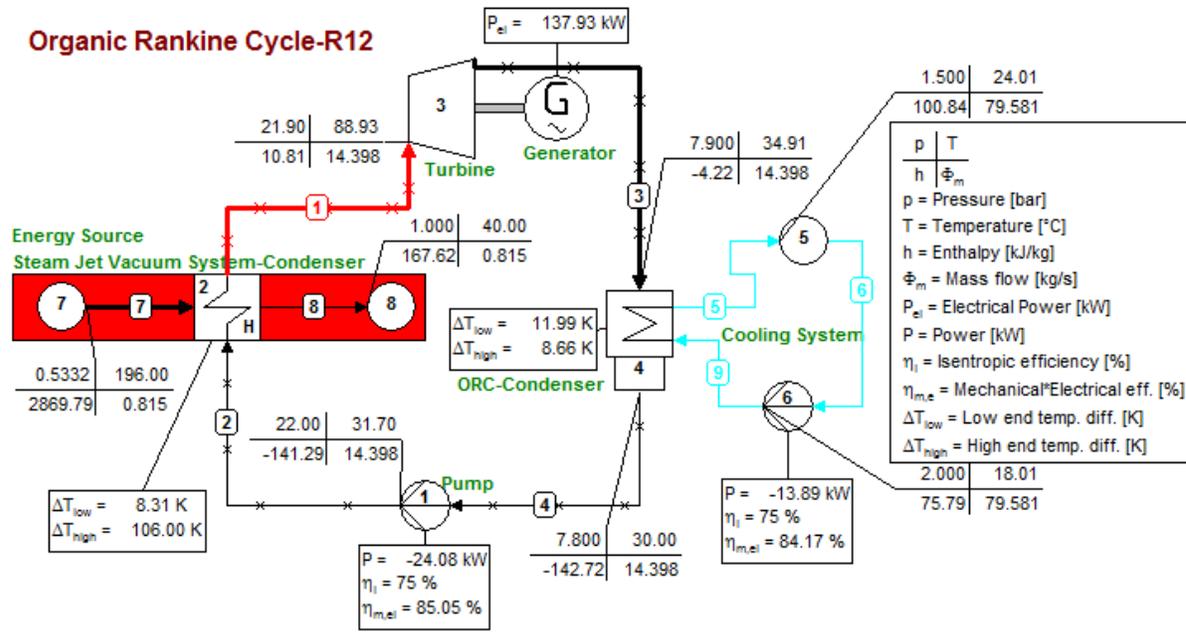
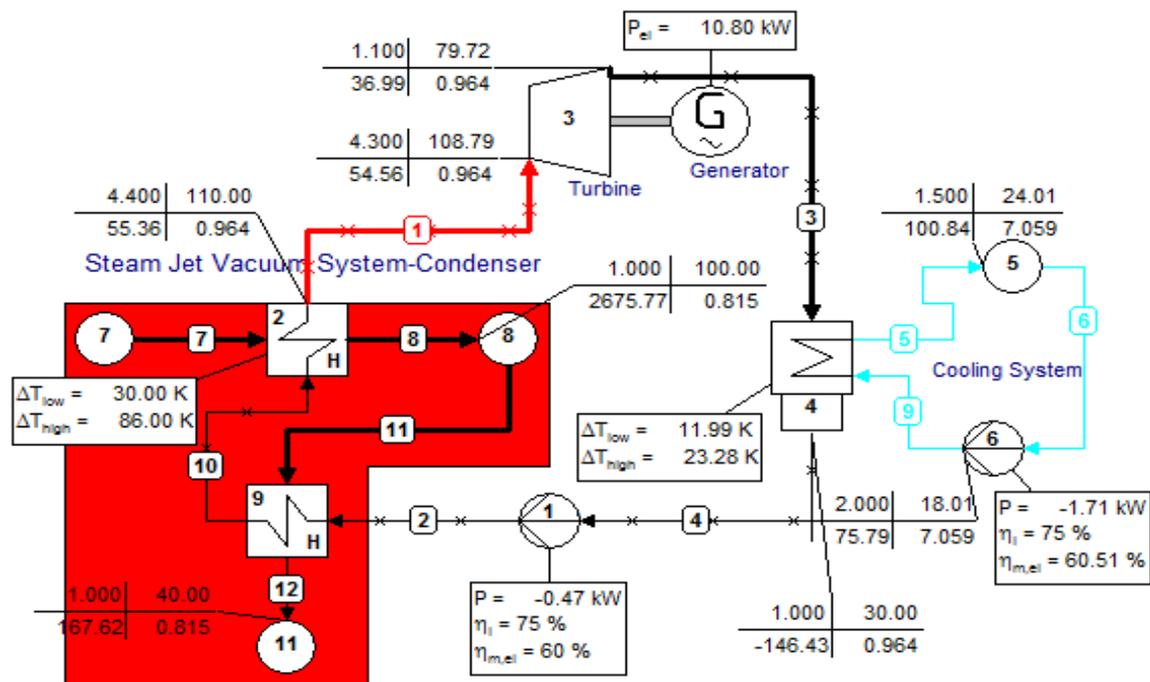
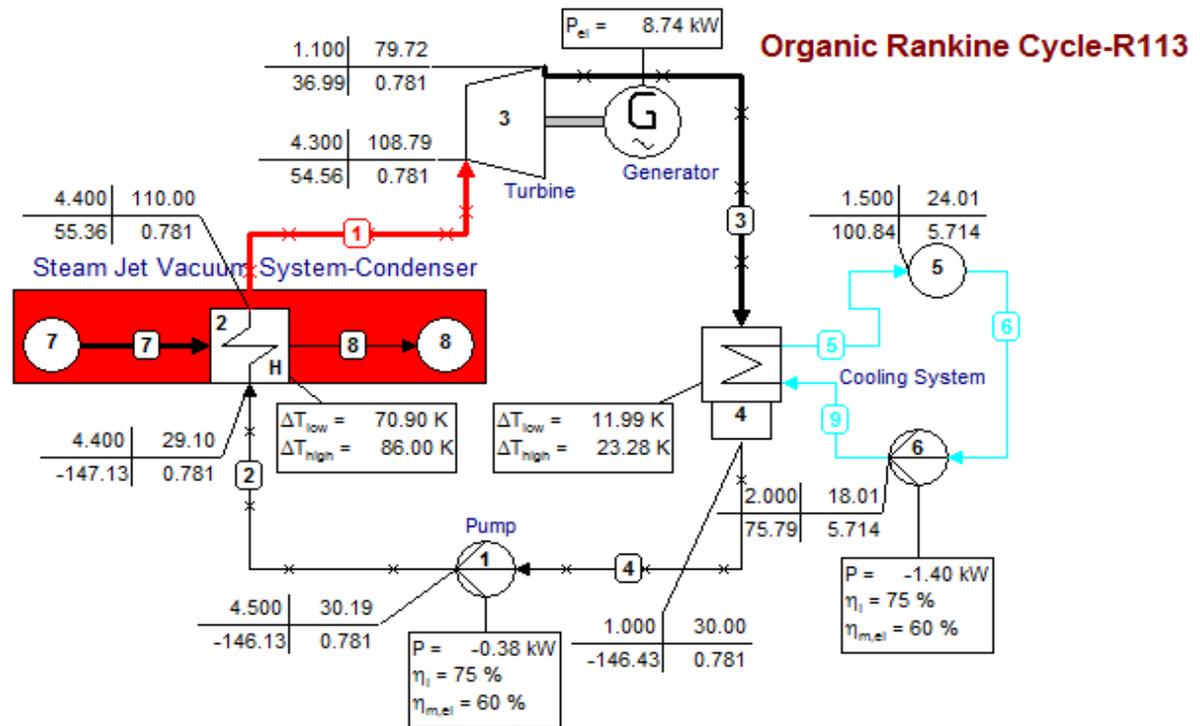
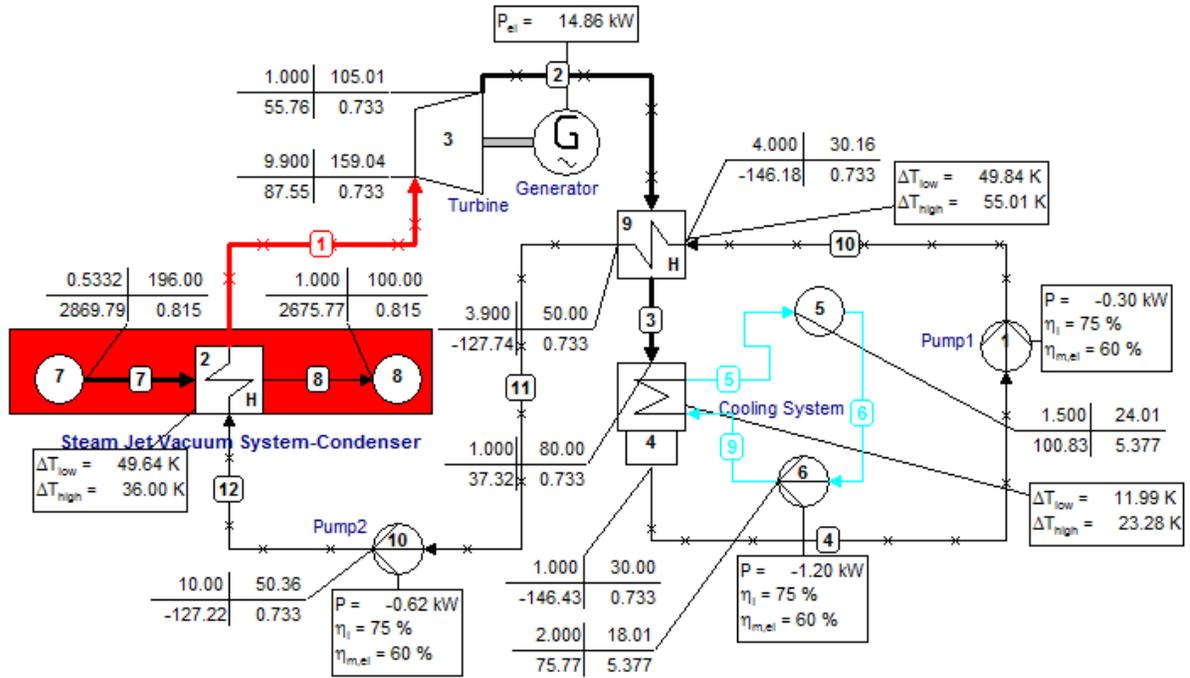


Figure 25 Cycle Tempo Analysis models for R12



**Organic Rankine Cycle-R113--Two Stage Condenser**

Figure 26 Cycle Tempo Analysis models for R113



**Regenerative Organic Rankine Cycle-R113**

Figure 27 Cycle Tempo Analysis model for R113

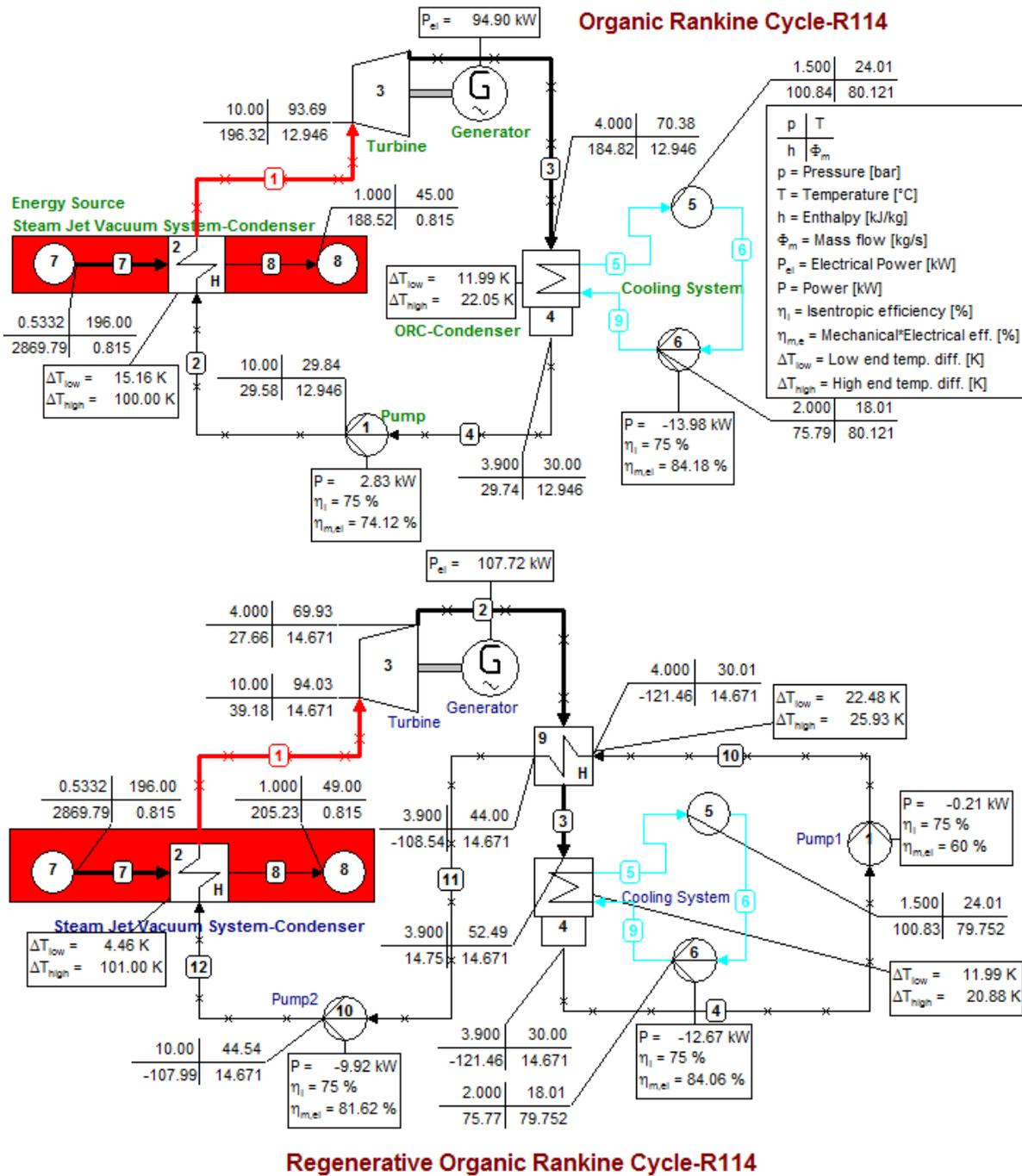
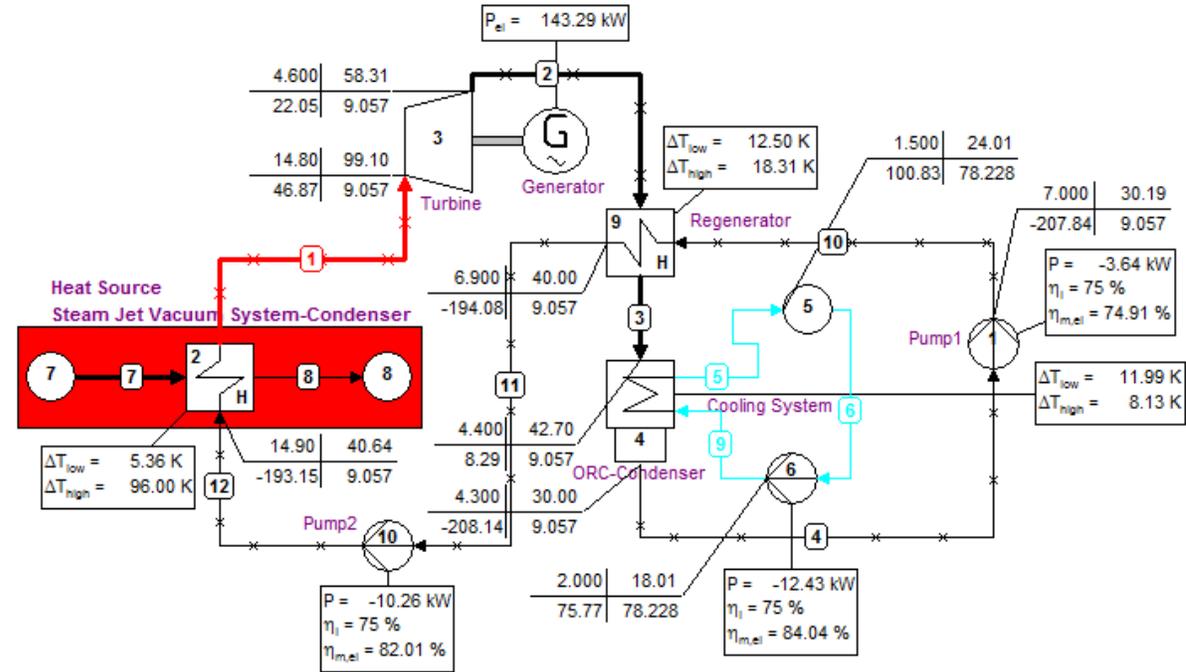
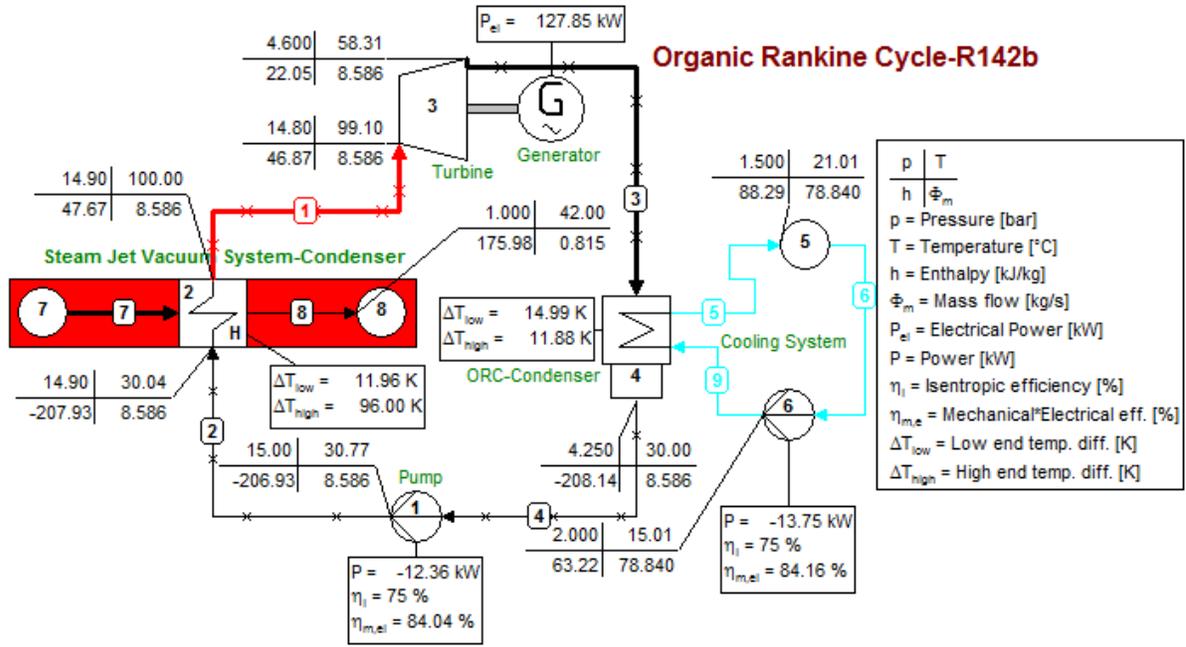


Figure 28 Cycle Tempo Analysis models for R114



**Regenerative Organic Rankine Cycle-R142b**

Figure 29 Cycle Tempo Analysis models for R142b

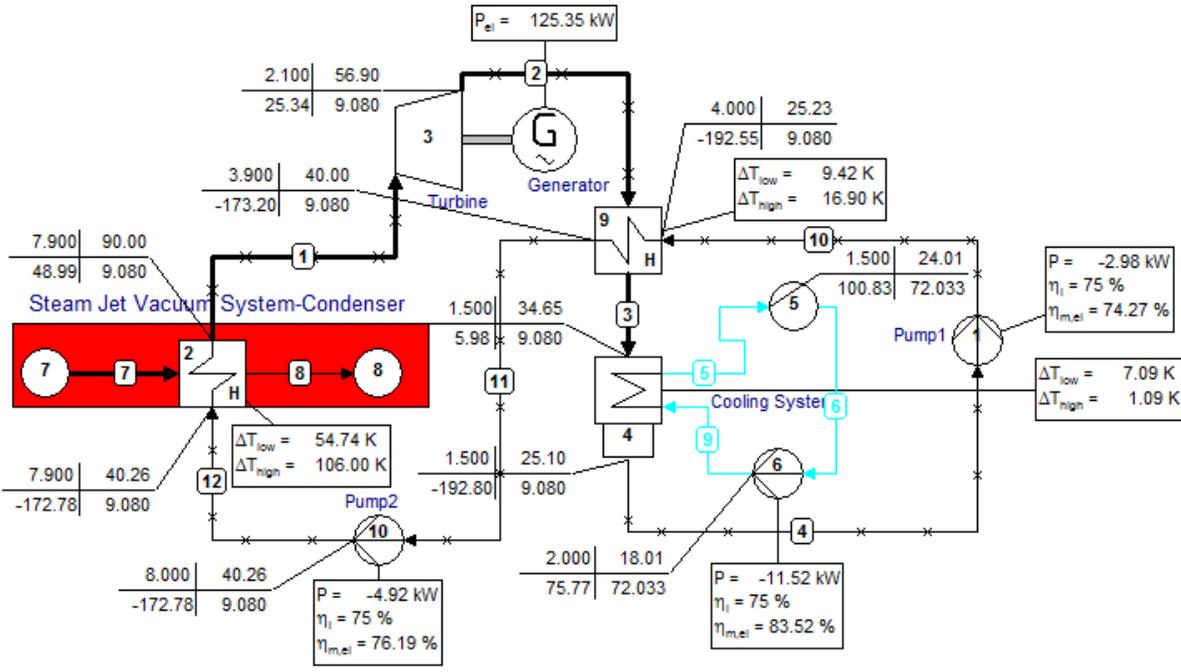
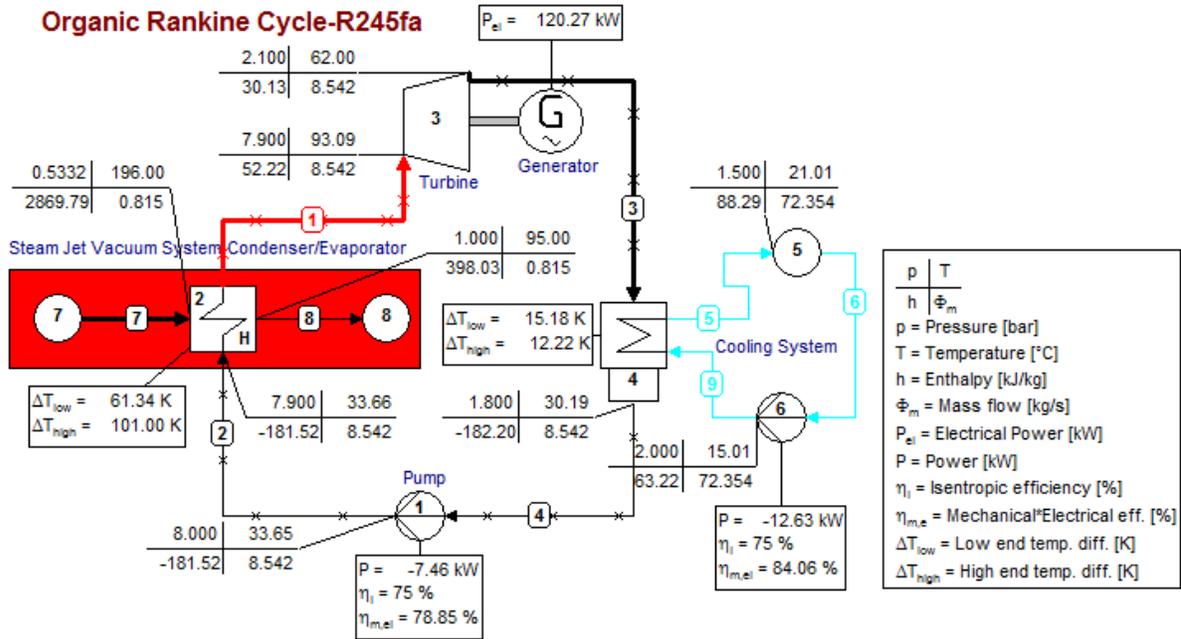
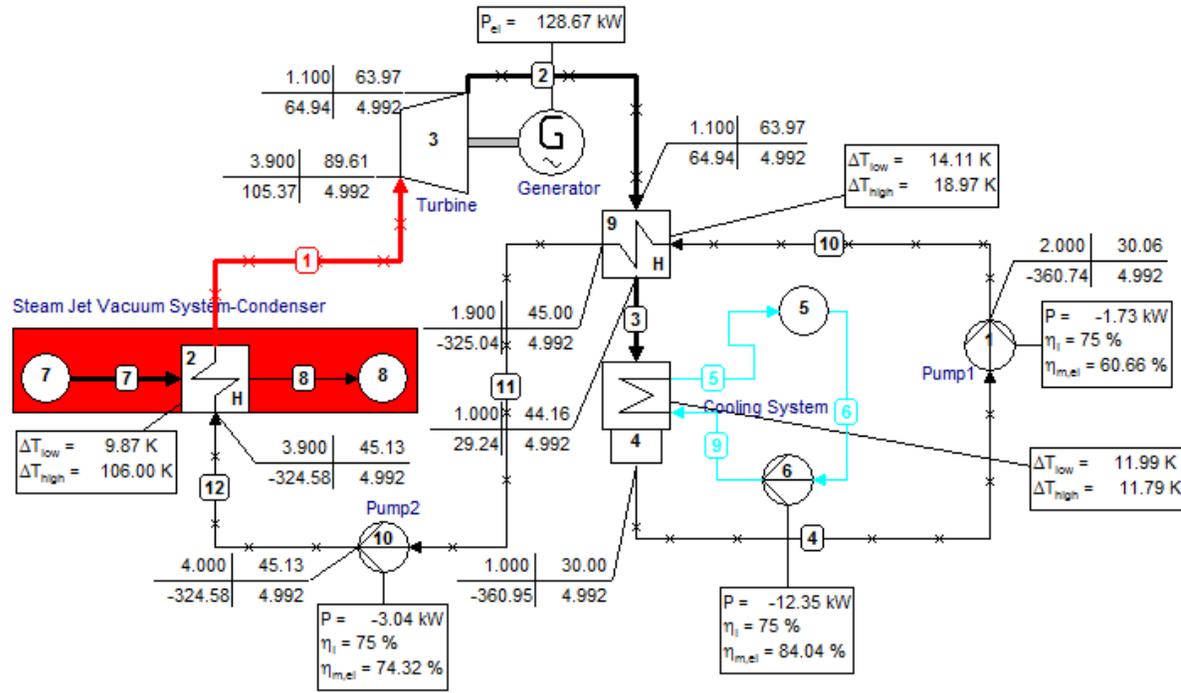
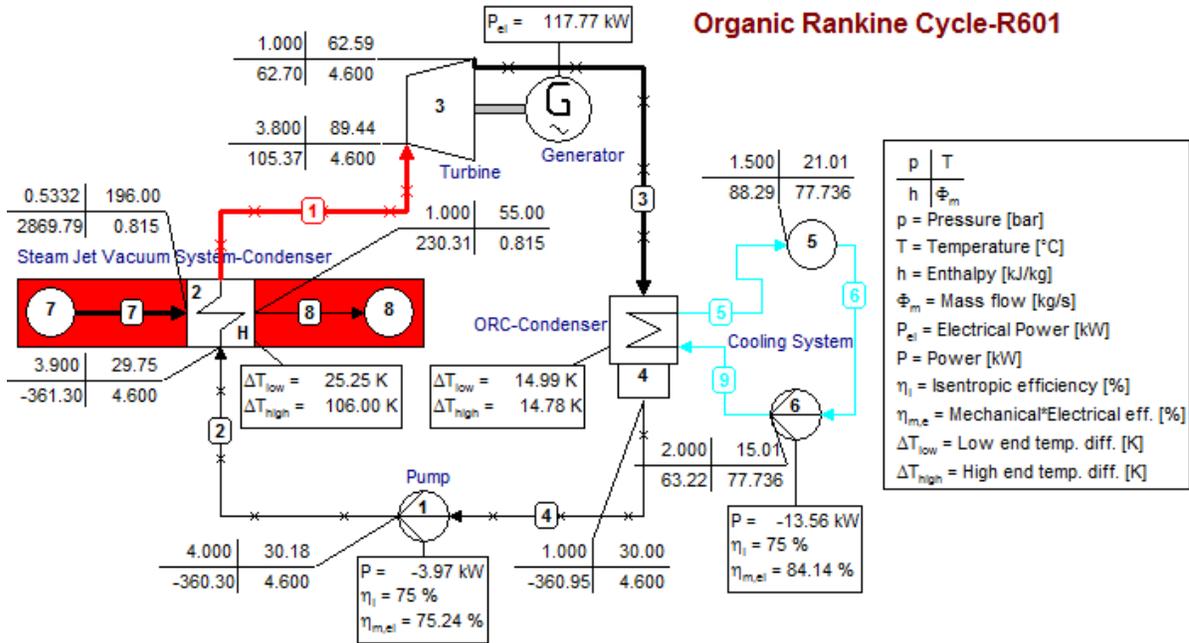


Figure 30 Cycle Tempo Analysis models for R245fa





**Regenerative Organic Rankine Cycle-R601**

Figure 32 Cycle Tempo Analysis models for R601

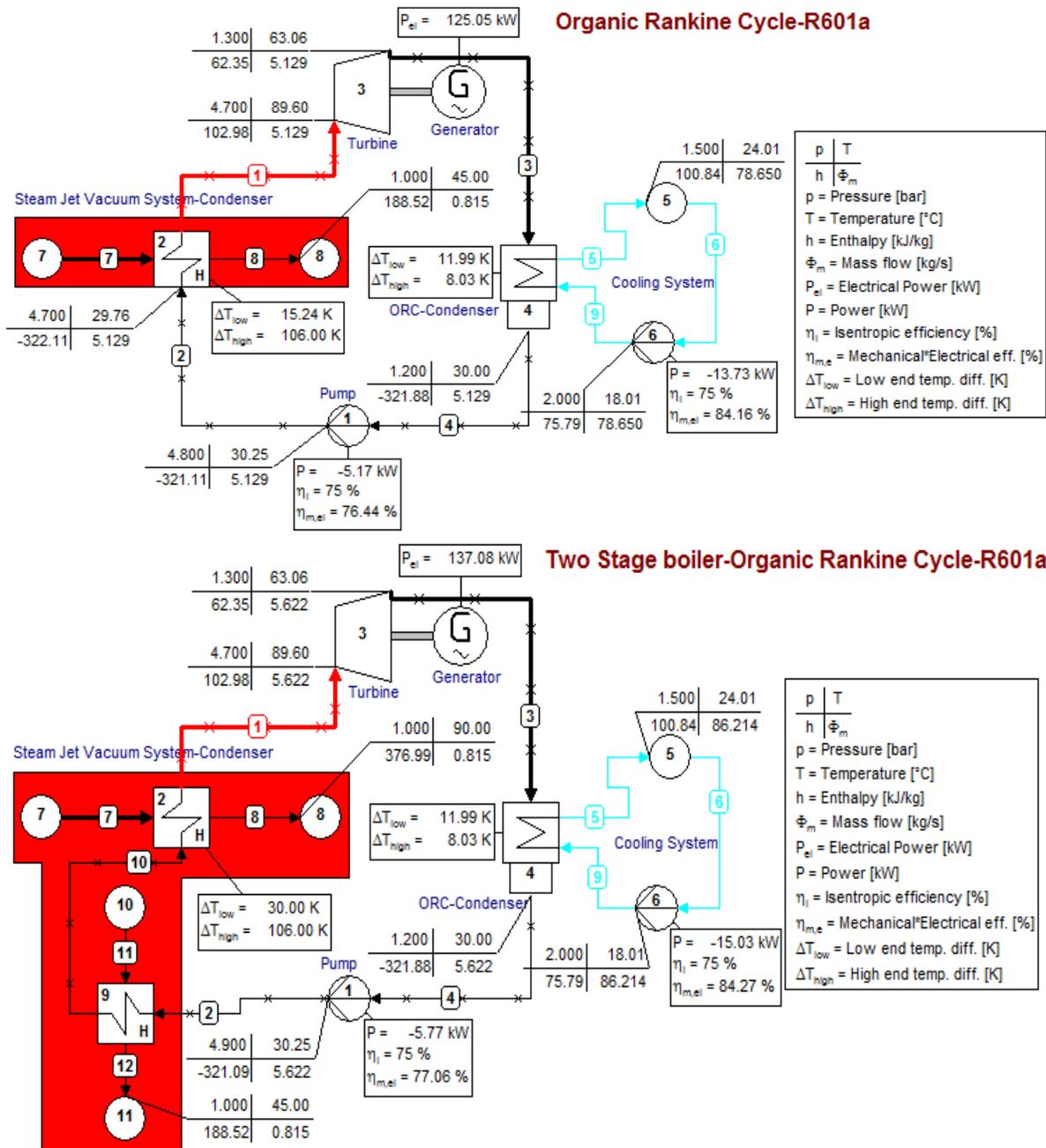


Figure 33 Cycle Tempo Analysis models for R601a

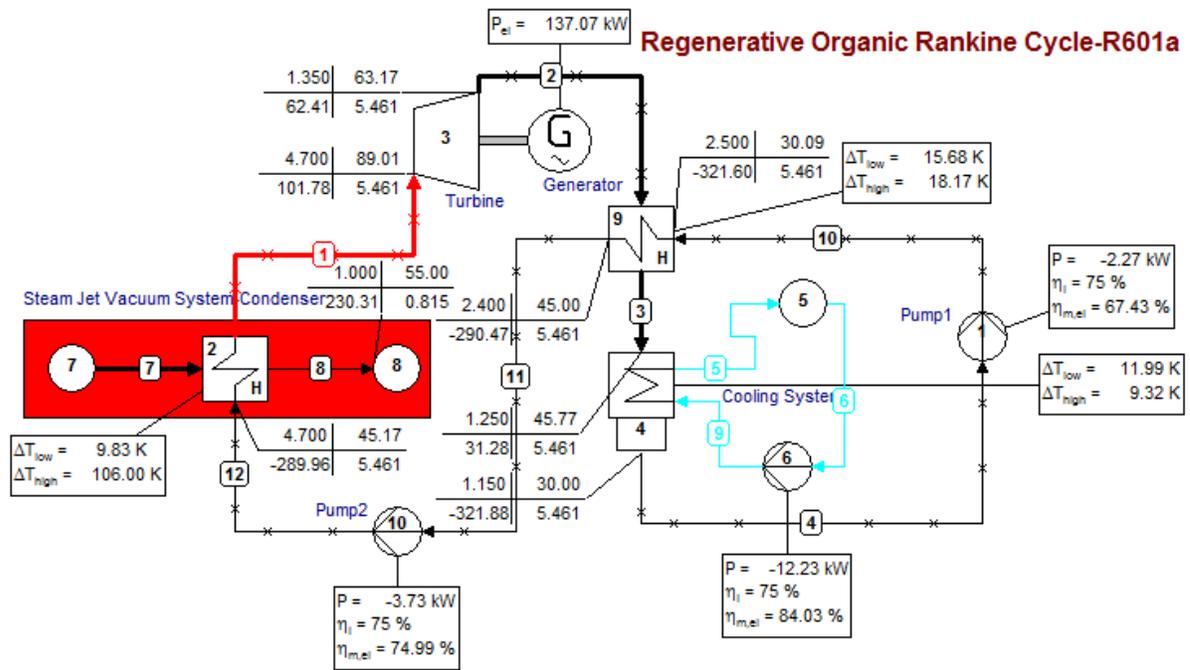


Figure 34 Cycle Tempo Analysis model for R601a

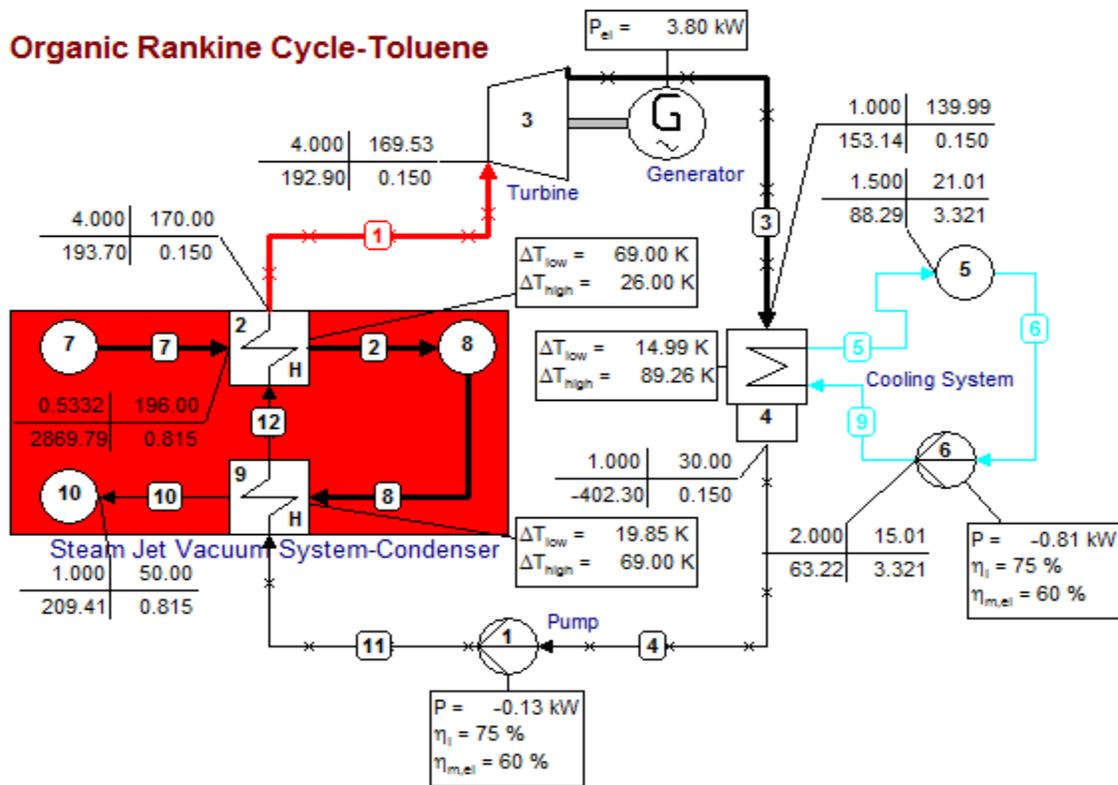
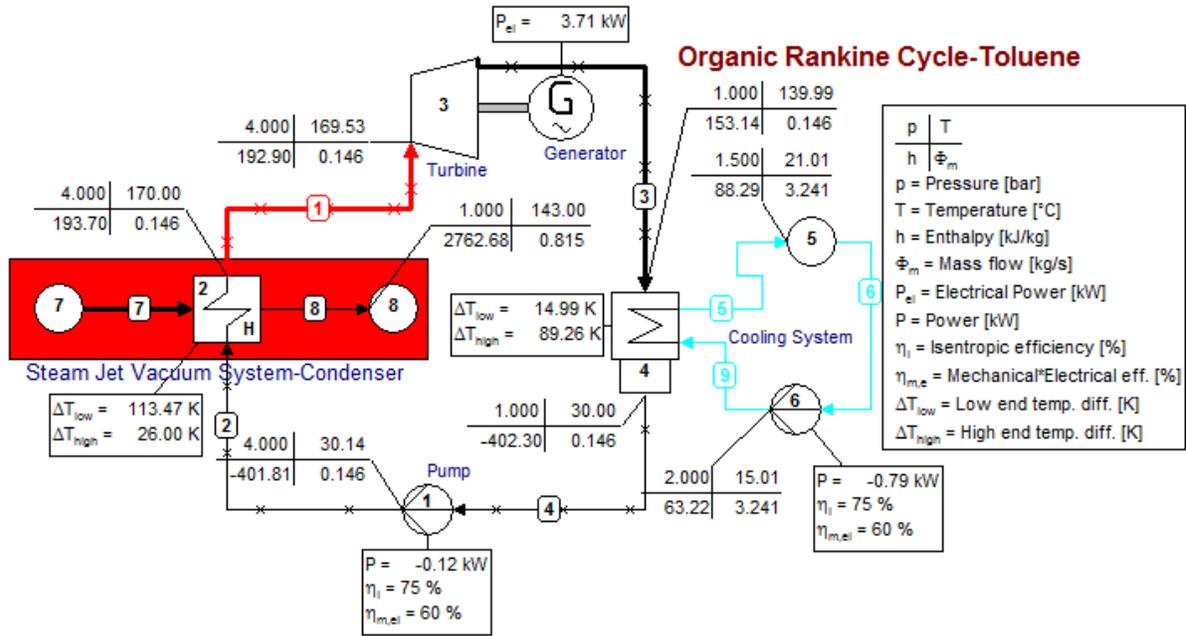


Figure 35 Cycle Tempo Analysis models for Toluene

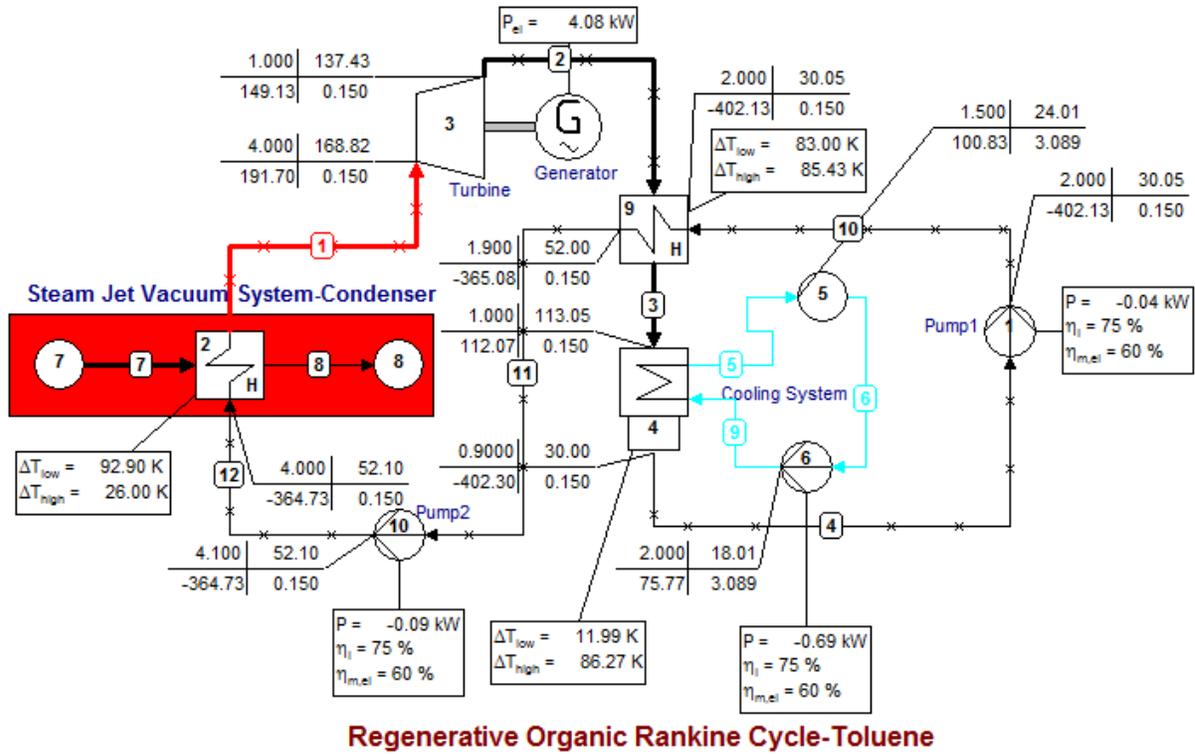


Figure 36 Cycle Tempo Analysis models for Toluene

The graph in Figure 37 was plotted to select the working fluid. Negative factors such as the safety factor, the ODP, the GWP, atmospheric lifetime and flow rates were deducted from the power output to generate the graph.

Table 8 below explains the level of importance and the values assigned to generate the graph. The values were assigned to reach uniformity in the selection process. This was all done in order to make sure that the maximum value of the preceding term was higher than the maximum value of the succeeding term, i.e. to indicate that the power generated held a higher importance than the ODP and so on.

Table 8 Selection process for refrigerants

Level of Importance		Assigned Values	Maximum Value Attained after Manipulation
Power		$\times 100$	11,884
ODP		$\times 200$	200
Safety Factor	A1	0	125
	B1	25	
	A2	50	
	B2	75	
	A3	100	
	B3	125	
GWP		$\div 100$	109
Cooling water flow rate		-	86
Atmospheric lifetime		$\div 4$	75
Organic fluid flow rate		$\times 4$	56

From the graph below we can see that R601a is the best fluid to work with in this case, since the environmental factors of R601a are good. i.e. R601a has an ODP of zero, and a low GWP of 20 compared to other organic fluids. The power output calculated for a regenerative R601a Rankine Cycle is 118.84 kW and that for a simple R601a Rankine cycle is 116.28 kW.

R601 and R142b are second in line for the selection of a working fluid. Using R142b as a working fluid, it was possible to generate 116.96 kW with a regenerative R142b Rankine Cycle, but with an ODP of 0.06 and GWP 2400 it would not be a suitable option in countries subjected to Montreal Protocol. R601 is a better option in these countries, with a capacity to generate 111.56 kW from a regenerative Rankine cycle and an ODP of 0 and GWP of 20. R601 is a better option over R142b.

R134a is also up there as an option. With a capacity to generate 107.56 kW, an ODP of 0, a GWP of 1430 and a safety factor of A1. So the final selection should be made between R601a, R601, R142b and R134a by considering the cost of the system and the cost of these refrigerants. R245fa is used by some commercial ORC manufactures. However, it should be considered after the four refrigerants mentioned above.

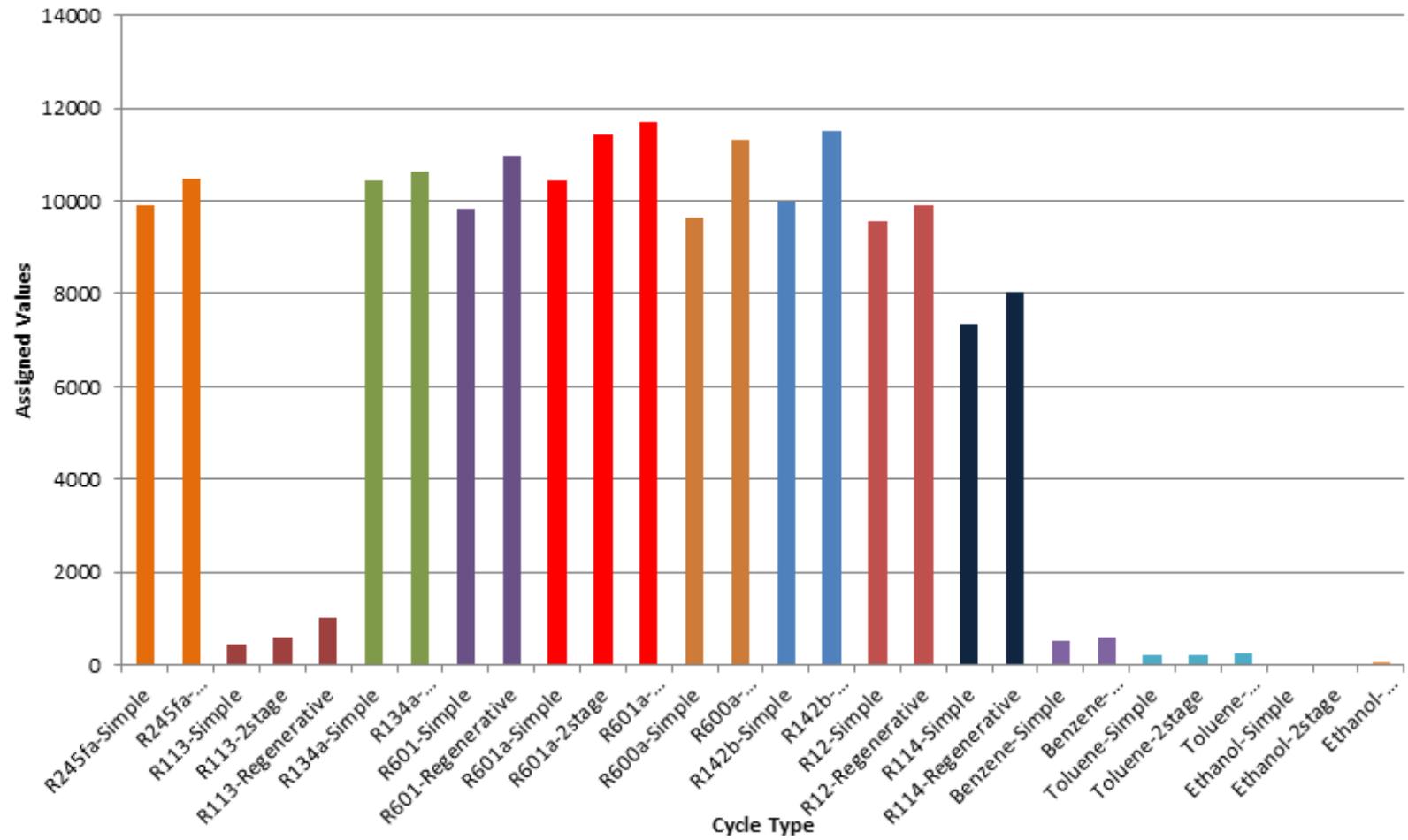


Figure 37 Selection of the Working Fluid

## Chapter 6

### Case Study for Ammonia Absorption Chiller

The Absorption Chiller is a second option for this project, since the absorption chillers are cumbersome.

The lithium bromide chillers are designed to house the absorber, the condenser and the generator inside a single module to keep the whole system under vacuum. Hence, it is difficult to model with the Cycle Tempo software. There is a limit on the maximum refrigeration that can be attained since the water is the refrigerant in the case of lithium bromide chiller. As a result, it was decided to model the ammonia absorption chiller. It is easier to customize an AAR chiller to a particular heat source and need.

The following model was developed to realize the refrigeration effect possible from replacing the generator of AAR with the condenser of the Steam Jet vacuum System. The model is loosely based on the AAR example model provided with the Cycle Tempo program.

It was noticed during the analysis that the COP of an AAR decreases rapidly as generator temperature decreases. The operating parameters were assigned after several trials and with the help of PTX (Pressure, Temperature and Mass Fraction) Diagram for the mixture of Ammonia and Water. Figure 40 is a PTX diagram with a simple Ammonia Chiller plotted in it.

The cycle maximum pressure is set in order to have a reasonable pinch point across the cycle components. The cycle minimum pressure is determined by the mixture saturation condition at the evaporator set point temperature.

Figure 38 below shows a schematic of the AAR chiller. The generator is modeled as a two stage condenser and the distillation column as a drum. The refrigerant vapor is condensed in

condenser number 7. The refrigerant is then divided by a valve in order to supply a fraction back into the drum in order to keep the cycle running. The liquid refrigerant is passed through the expansion valve and then the low pressure and temperature refrigerant is passed through an evaporator to produce the refrigerating effect. The refrigerant vapor is absorbed by the absorber and then then pumped back into the drum to be generated, thus completing the cycle.

The refrigerating effect of the cycle is 28.64 Tons with a pump work of 1 kW. The system requires a cooling water flow of 8.5 kg/s at 59°F. Figure 39 below shows the results of the analysis of the Cycle Tempo model of AAR. With a coefficient of performance of 0.04 the cycle is very inefficient use of waste heat.

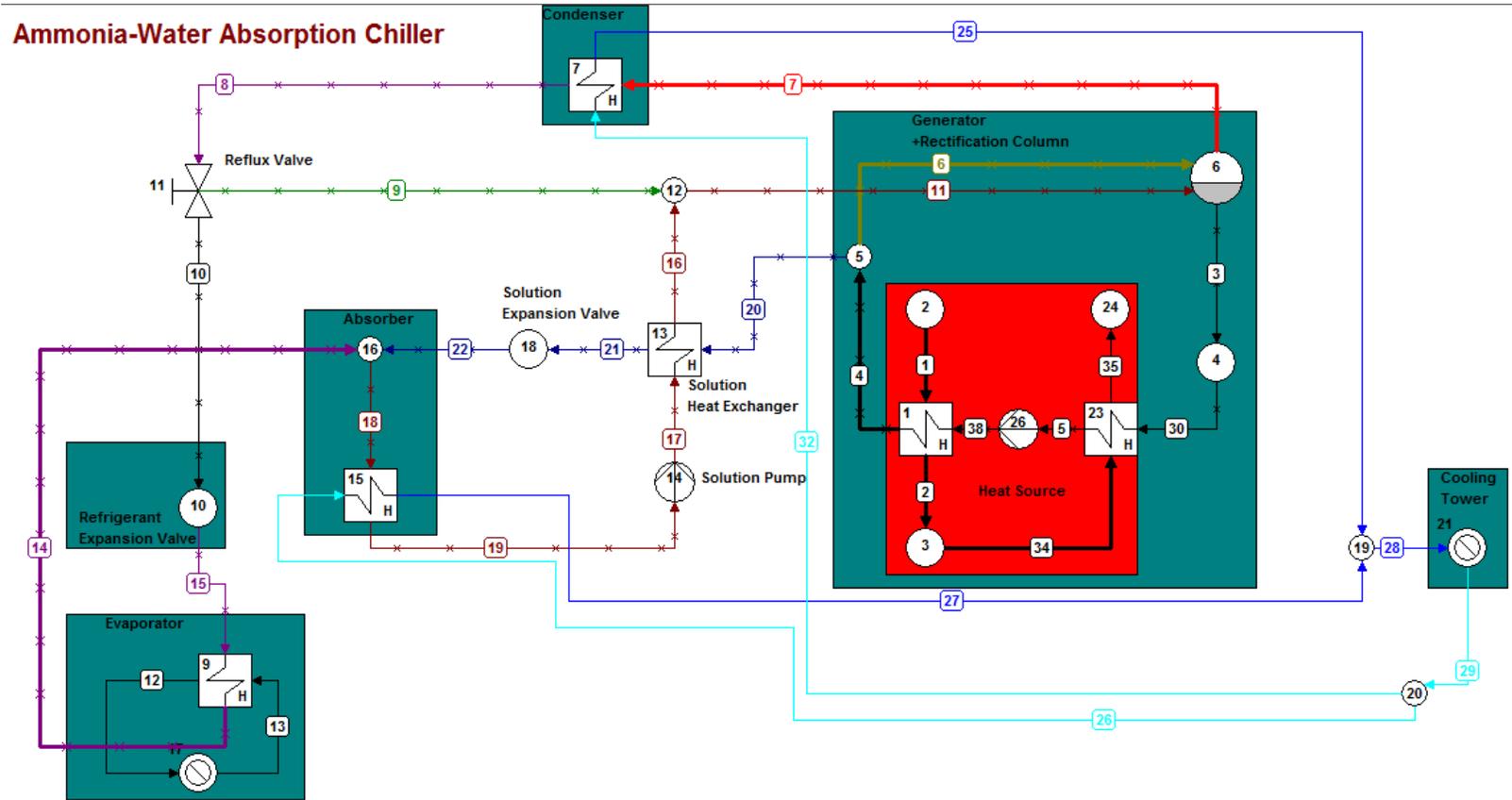


Figure 38 Schematic of Ammonia Absorption Chiller in the Cycle Tempo Program



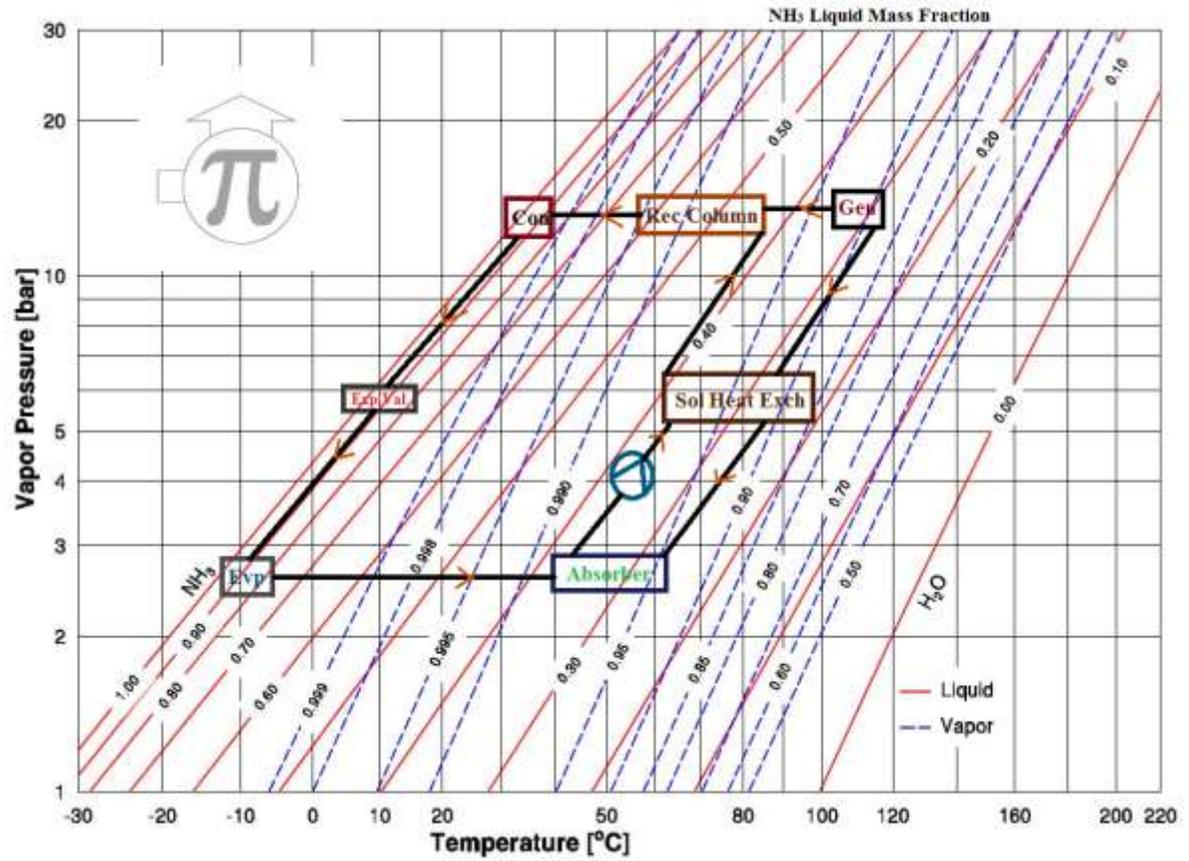


Figure 40 Representation of a simple Ammonia Absorption chiller on the PTX diagram (10)

## Chapter 7

### Economic Analysis

The feasibility of any engineering project is greatly dependent on the economic factor. The economic analysis helps estimate cost savings and the capital investment required to implement the ORC or an AAR. Important factors that need to be considered during the implementation of a particular project is cost savings from electricity, system installed costs, and maintenance costs.

The proposed ORC system would cost in the range of \$1,000-\$2,000/kw to implement according to a manufacturer of Organic Rankine power generation systems (11). Considering the maximum cost of \$2,000/kW a 150kW ORC system would cost a \$300,000 to implement. Since the ORC system will require some electrical work an additional cost of \$20,000 is considered for this case

A previous project regarding absorption chiller suggested that an absorption chiller costs around \$1,000/ton (12). Hence, a 30 ton Absorption chiller would cost \$30,000. An additional cost of \$10,000 is considered for pipes and fitting. The system will require some training to be operated, which can add another \$1,000 to the total implementation cost i.e. \$41,000.

Federal and State governments offer numerous tax incentives for the energy efficiency and renewable energy projects. The economic incentive for installing an ORC system should be checked with the authorities. The incentives are usually expressed as a percent of the electric savings or a percent of the implementation cost, so if incentives considered it will change the payback considerably.

## 7.1 Utility Considerations

Rate schedules determine the payment that the customers have to make for their energy use. The rate schedules vary based on the type of user such as residential, commercial and industrial. Energy price is much lower for an industrial user compared to a residential user, since they use much less energy.

In the flowing analysis, in order to maintain confidentiality, it has been decided to use Large General Service – Time of Use-23 rate schedule. The LGS-TOU has been chosen, since it is among the common type of rate schedule the industries are subjected to by power companies.

The savings that can be realized from implementing the 116 kW ORC system is given in Table 9 below. The savings are based on 24 hours, 7 days a week basis. An average of on peak and off peak is considered for kWh savings.

Table 9 Savings Realized from Implementing 116 kW ORC System

Month	Average Electricity charge (\$/kWh)	Demand Charge	kWh/Month	kW Savings (\$/Month)	kWh Savings (\$/Month)	Overall Savings (\$/Month)
January	0.0457	14.25	85,560	1,653	3,910	5,563
February	0.0457	14.25	77,280	1,653	3,532	5,185
March	0.0457	14.25	85,560	1,653	3,910	5,563
April	0.0457	14.25	82,800	1,653	3,784	5,437
May	0.0457	14.25	82,800	1,653	3,784	5,437
June	0.0457	19.56	85,560	2,269	3,910	6,179
July	0.0457	19.56	85,560	2,269	3,910	6,179
August	0.0457	19.56	85,560	2,269	3,910	6,179
September	0.0457	19.56	82,800	2,269	3,784	6,053
October	0.0457	14.25	85,560	1,653	3,910	5,563
November	0.0457	14.25	82,800	1,653	3,784	5,437
December	0.0457	14.25	85,560	1,653	3,910	5,563
<b>Total</b>				<b>22,300</b>	<b>46,038</b>	<b>\$68,338/yr.</b>

The savings that can be realized from installing the 28 ton Absorption refrigeration system is given by the following calculation below. The facility already operates a 1,700 ton water cooled vapor compression chiller. Electrically powered vapor compression chillers have an efficiency of about 0.6 kW/ton. Thus, getting the system to reduce its load by 28 tons on a maximum will save

$$\begin{aligned}\text{Chiller power savings} &= 28 \text{ ton} \times 0.6 \text{ kW/ton} \\ &= 16.8 \text{ kW}\end{aligned}$$

With the 1kW power consumed by the Absorption chiller pump, the savings is further reduced to 15.8 kW.

$$\begin{aligned}\text{Cost Savings} &= (15.8 \text{ kW} \times 8,400 \text{ hrs/yr} \times \$0.0457/\text{kWh}) \\ &\quad + (15.8 \text{ kW} \times 12 \text{ months} \times \$14.25/\text{kW}) \\ &= \$8,767/\text{yr}.\end{aligned}$$

However, the actual savings realized is slightly lower since the power consumption will not reduce proportionally to the load reduced. Investing in an absorption system when a cooling system is already in place does not make a lot of sense.

The payback period for both an ORC system with a \$320,000 implementation cost and an Absorption chiller with a \$41,000 implementation cost is 57 months. Absorption chillers are generally known to be difficult to operate. In comparison the ORC systems are easier to operate.

The price of the electricity is a major deciding factor to implement this project. Historical trends state that the electricity price has been increasing over the years. In areas where energy prices are expensive, the following projects will have a much better payback.

## Chapter 8

### Conclusion

From the earlier study it can be concluded that this project has a huge potential to be implemented. The power generation can be increased with an increase in flow rate of the waste heat. It can be seen in the economics section that the feasibility of this project are greatly dependent on the on the electric cost. Also, there are other non-quantifiable advantages such as environmental factors associated with waste heat recovery. The following waste heat recovery technologies offset the fossil fuels that would otherwise be used to generate the power.

#### Next Steps:

- Further investigate which of the two proposed systems should be implemented.
- Investigate incentives. The incentives will further make the project more economical.
- Contact professional consulting services to further verify the credibility of the project.
- Staging the present condenser will help to increase the power generation by 10%.

#### Future Work:

- The condenser will require some modifications in order to operate with a refrigerant, though minimal it should be studied for its negative aspects.
- A refrigerant with better critical values compared to operating range will certainly help the Organic Rankine Cycle. So selection is a constant process with the development of new refrigerants.
- A working model of the ORC if built in the IAC will certainly help the students understand the system better.
- The condenser of the system is modeled to operate at atmospheric pressure. If the operating pressure of the system is reduced below atmospheric the power generation

is further increased. The lower the pressure that the condenser can be operated in, the higher the power generation. If the condenser of the ORC is cooled with another refrigerant the power generation can be increased further as indicated in Figure 41 below.

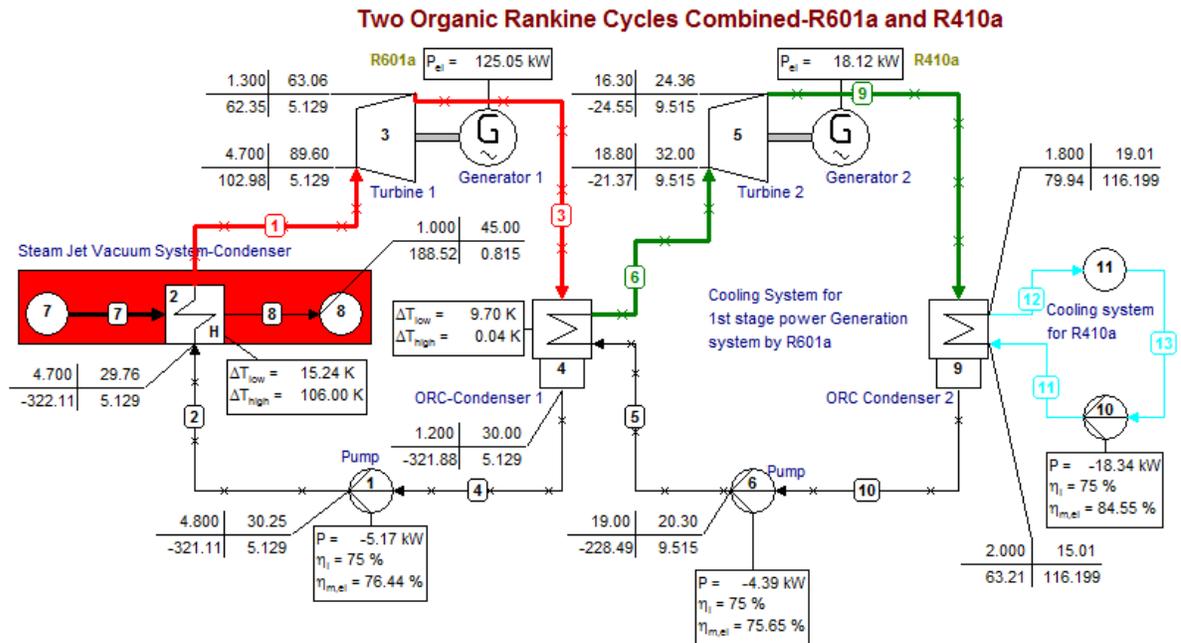


Figure 41 An example of Two Organic Rankine cycles Combined/ Condenser of first stage cooled by another refrigerant generating power

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## APPENDICES

## Appendix A

### *Unit and Conversion*

A unit of measurement is a definite magnitude of a physical quantity, defined and adopted by convention or by law, that is used as a standard for measurement of the same physical quantity (13). The following table provides the necessary units for measurement and conversion that was used in the following project.

Table 10 Conversion Factors

<b>Dimension</b>	<b>Metric</b>	<b>English</b>
Pressure	1 Pa = 1 N/m <sup>2</sup>	1 Torr = 0.99 mm Hg
	1 kPa = 10 <sup>3</sup> Pa = 10 <sup>-3</sup> MPa 1 atm = 101.325 kPa = 760 mm Hg	1 barr = 750 Torr
Perfect Vacuum = 0 Torr = 0 Pa		
Temperature	T(K) = T(°C) + 273.15 ΔT(K) = ΔT(°C)	T(R) = T(°F) + 459.67 T(°F) = 1.8 T(°C) + 32
Energy, heat, work, Internal Energy, Enthalpy	1 kJ = 1000 J = 1000 Nm 1 kJ/kg = 1000 m <sup>2</sup> /s <sup>2</sup> 1 kWh = 3600 kJ	1 kJ = 0.94782 Btu 1 Btu = 1.055 kJ 1 Btu/lbm = 2.326 kJ/kg
Flow Rate	1 kg/s = 3.543 ton/hr	1 lb/s = 0.4536 kg/s
Power	1 W = 1J/s 1 kW = 1000 W = 1.341 hp 1 hp = 745.7 W	1 kW = 3412.14 Btu/h 1 hp = 0.7068 Btu/s 1 BHP = 33.475 KBtu/h

## **Some Definitions**

### **Quality of steam:**

Quality of the steam is the amount of vapor in the mixture i.e. saturated vapor has a quality of 100% and saturated liquid has a quality of 0 %. Quality can be calculated by dividing the mass of the vapor by the mass of the total mixture:

$$\text{Quality} = \frac{m_{\text{vapor}}}{m_{\text{total}}}$$

Where m indicates mass.

### **Superheated steam:**

Superheated steam is steam at a temperature that is higher than its boiling point at the absolute pressure. The steam can lose internal energy i.e. lose some temperature without resulting in a change of state i.e. condensing into liquid. Superheated steam is steam at a temperature that is higher than its vaporization point at the absolute pressure where the temperature measurement is taken; and therefore, the steam can cool by some amount, resulting in a lowering of its temperature without changing state from a gas, to a mixture of saturated vapor and liquid.

## Appendix B

### *Saturation Curves for Refrigerants*

The following are examples of the saturation curves got from the NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP).

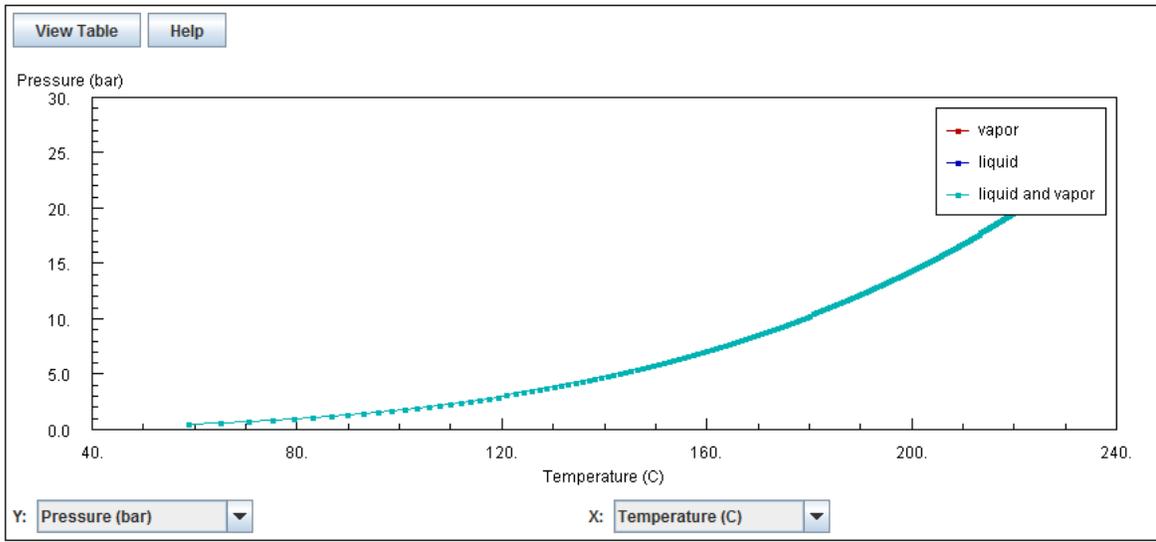


Figure 42 Benzene Saturation Curve

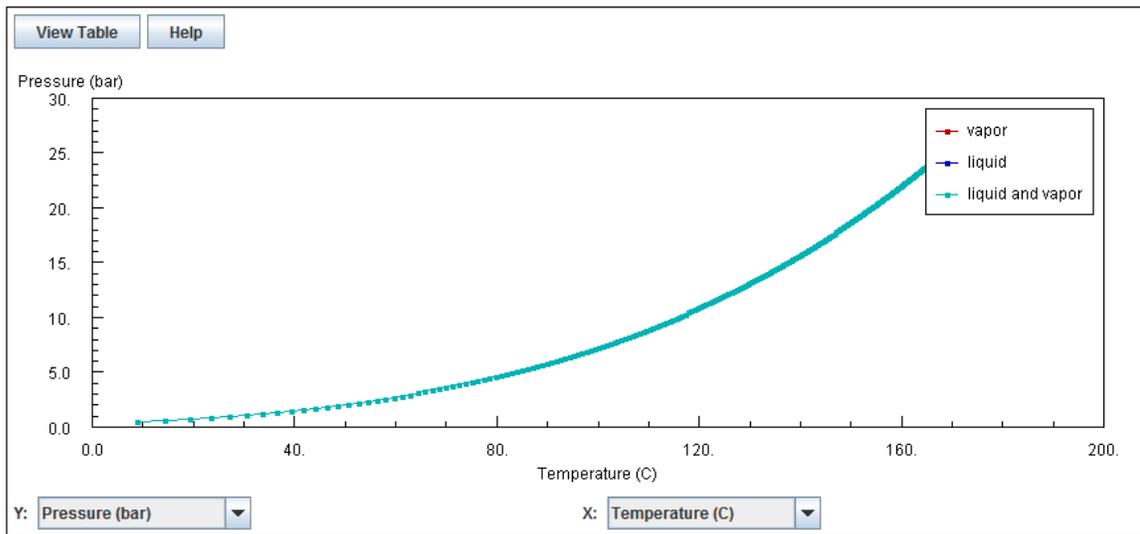


Figure 43 R601a Saturation Curve

## Appendix C

*Additional images used for thesis presentation*

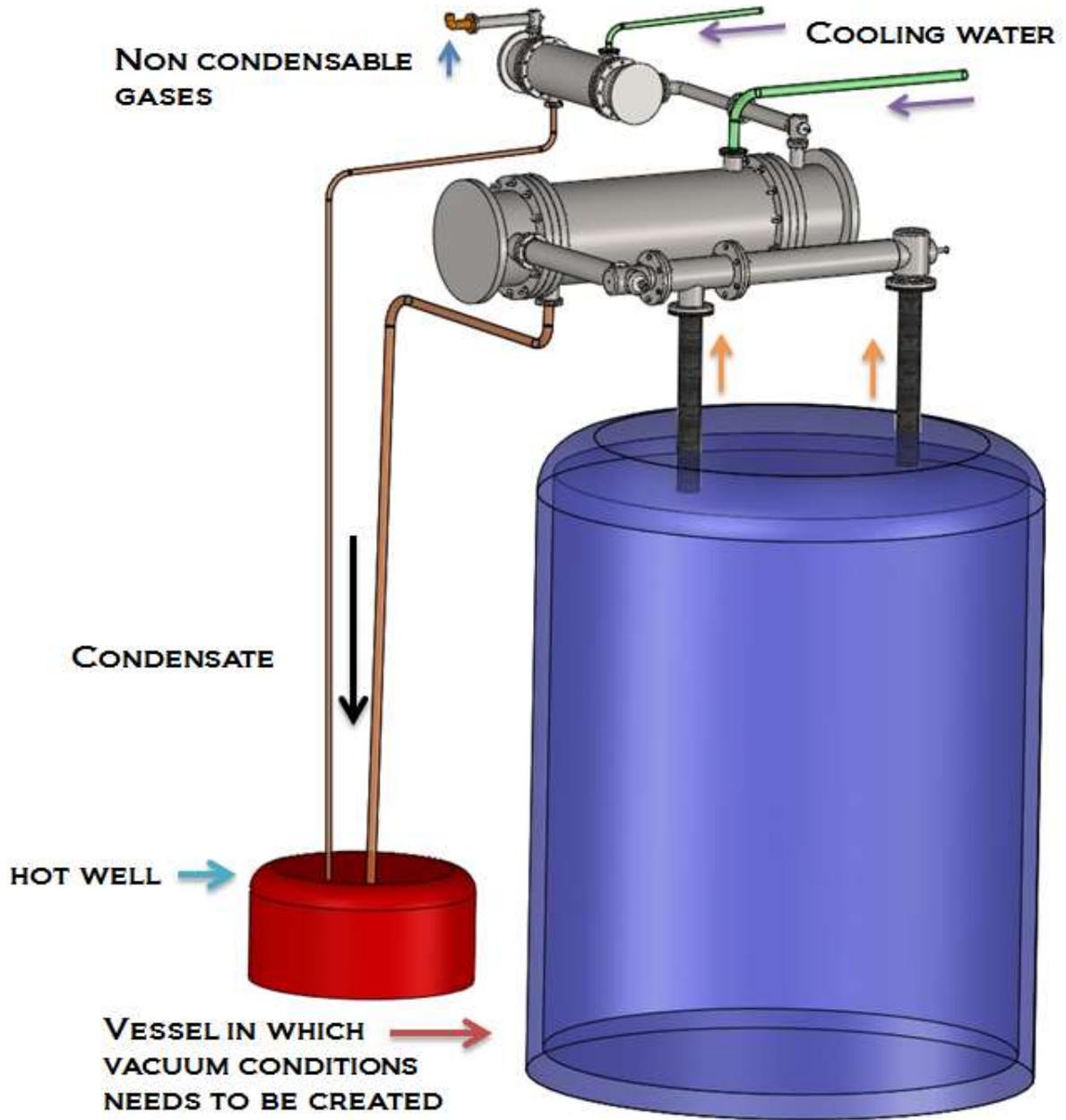


Figure 44 Steam Jet Vacuum System connected to other components

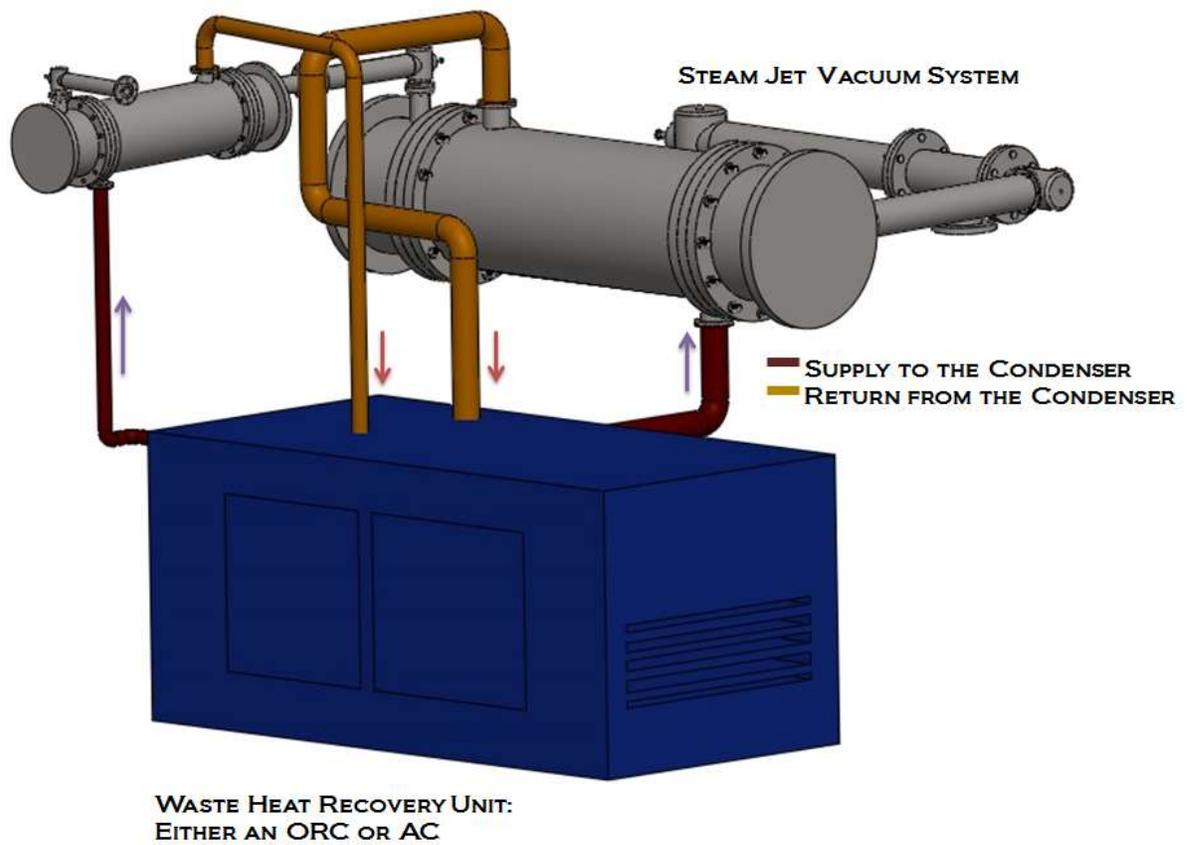


Figure 45 Example of Combined Steam Jet Vacuum System and Waste heat recovery unit

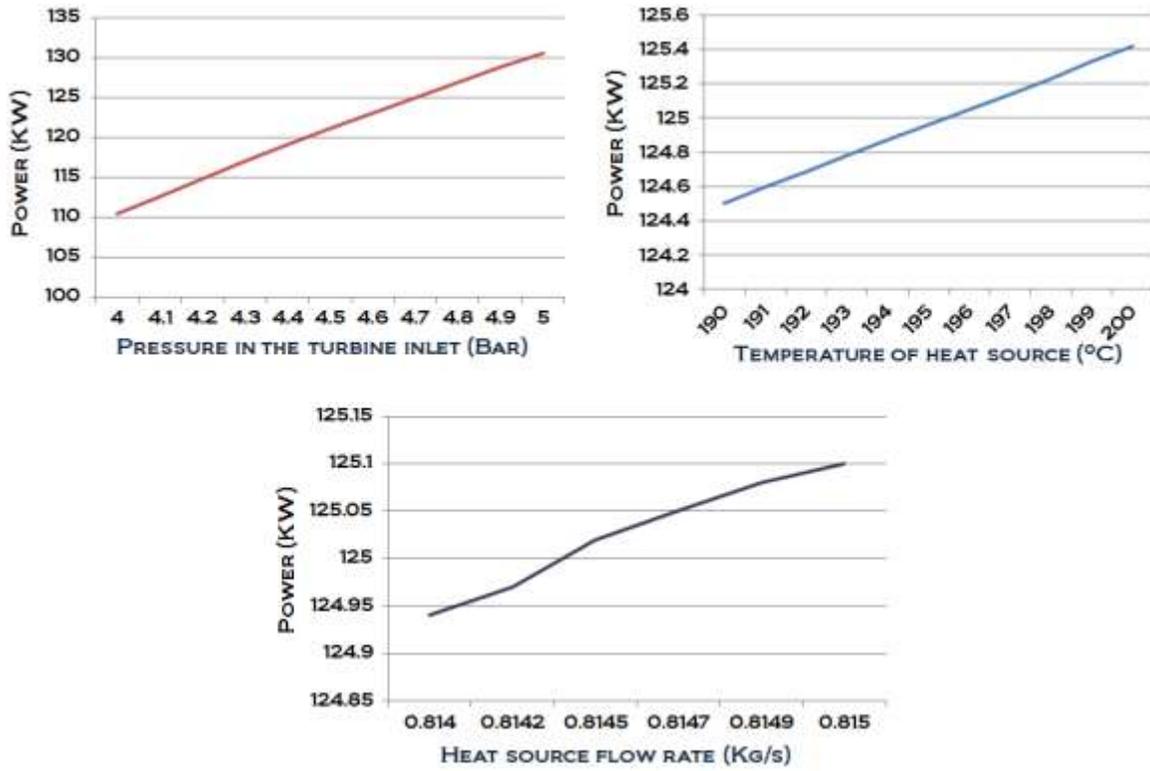


Figure 46 Effect of heat source temperature, flow rate and turbine inlet pressure on power generated