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Analysis of Low Temperature Organic Rankine Cycles

for Solar Applications

by

Yunfei Li

A thesis

Presented to the Graduate and Research Committee

of Lehigh University

in Candidacy for the Degree of Master of Science

in

Mechanical Engineering and Mechanics

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This thesis is accepted and approved in partial fulfillment of the requirements for the Master of Science.

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Date

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Dr. Sudhakar Neti, Thesis Advisor

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Dr. D. Gary Harlow, Chairperson of Department

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ABSTRACT

The present work focuses on Organic Rankine Cycle (ORC) systems and their application to low temperature waste heat recovery, combined heat and power as well as off-grid solar power generation applications. As CO_2 issues come to the fore front and fossil fuels become more expensive, interest in low grade heat recovery has grown dramatically in the past few years. Solar energy, as a clean, renewable, pollution-free and sustainable energy has great potential for the use of ORC systems. Several ORC solutions have been proposed to generate electricity from low temperature sources. The ORC systems discussed here can be applied to fields such as solar thermal, biological waste heat, engine exhaust gases, small-scale cogeneration, domestic boilers, etc.

The current work presents a thermodynamic and economic analysis for the use of ORC systems to convert solar energy or low exergy energy to generate electrical power. The organic working fluids investigated here were selected to investigate the effect of the fluid saturation temperature on the performance of ORCs. The working fluids under investigation are R113, R245fa, R123, with boiling points between 40℃ and 200℃ at pressures from 10 kPa to 10 MPa. Ambient temperature air at 20°C to 30°C is utilized as cooling resource, and allowing for a temperature difference $10\,$℃ for effective heat transfer. Consequently, the working fluids are condensed at 40℃. A combined first- and second-law analysis is performed by varying some system independent parameters at various reference temperatures. The present work shows that ORC systems can be viable and economical for the applications such as waste heat use and off-grid power generation even though they are likely to be more expensive than grid power.

CHAPTER 1

INTRODUCTION

1.1   Motivation

Fossil fuel consumption in the recent years has been increasing and the burning of fossil fuel is said to be a major contributor towards global warming, acid rains, air, water and soil pollution, forest devastation and radioactive substances emissions. Besides the environment, the fossil fuel prices fluctuate considerably, usually going up, with the price of liquid hydrocarbons well over USD 4/gallon in US, and more expensive in most other countries. Most importantly, the quantity of fossil fuels, like petroleum, natural gas and coal can only decrease since they are non-renewable energy resources. As a result, many countries have been investing billions dollars in new energy technologies and demand for sophisticated power supply options is greatly increased.

Currently, recovering low temperature heat, which includes industrial waste heat, geothermal energy, solar heat, biomass, and so on, could be a very critical and sustainable way to solve energy crisis. Comparing to industrial waste heat, geothermal energy and biomass, solar energy has advantages over them due to its usability and low-cost. Utilizing waste heats along with attempts for the use of renewable resources as low grade thermal heat sources have motivated the wider use of ORC.

1.2   Background

One of the earliest uses of Organic Rankine Cycles (ORC) as a power generation system from low heat sources appears to be one presented in 1961 by Israeli solar engineers Harry Zvi Tabor and Lucien Bronicki [1].
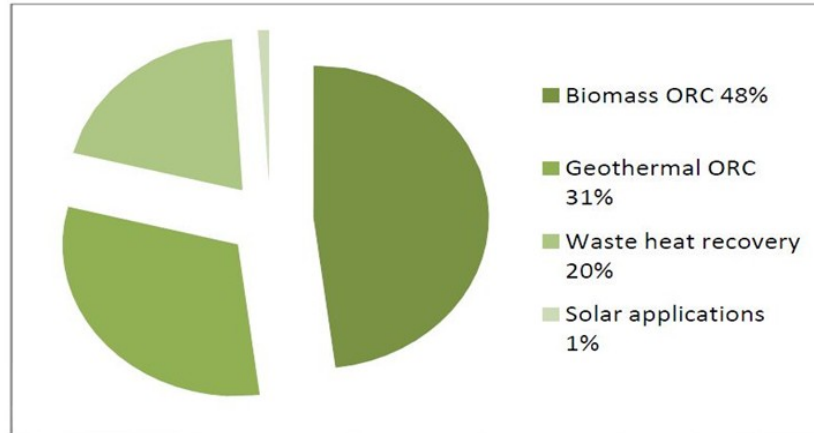
Figure 1.1: ORC market share for different heat sources [2]

Figure 1.1 shows that currently Biomass ORC has a dominant market share and Geothermal which used to be the most significant part is in the second position. Waste heat recovery in different industries which is in third place was result of use in numerous industries and solar ORC still has a huge potential yet to be realized. This technology, due to lack of awareness, currently is only 1% of ORC market. Biomass based application was developed since it was the only proven technology for decentralized applications to generate powers up to $1MW_e$ from solid fuels like biomass [3]. From the 1980s when ORC has been available in the market, more than 200 projects with a total power of about 2,000 MW of electricity have been installed. This is one of the reasons for

the unprecedented growth of ORC usage in recent years. This establishes a promising future for ORC markets in near future.

An Organic Rankine Cycle (ORC) basically resembles the steam cycle according to thermodynamic principles. In an ORC, water is replaced with a high molecular mass organic fluid with a lower saturated boiling temperature in comparison with water. Fluid characteristics make ORC favorable for applications of low temperature conditions (normally less than 400°C) and heat recovery applications at even lower temperatures.

ORCs in general have several advantages over steam Rankine cycle. It is known that working fluids in ORCs have higher molecular weight than water and this will increase the mass flow rate of working fluid for the same sizes of turbine. Larger mass flow rate, can in general, gives better turbine efficiencies with less turbine losses [3].

Turbine efficiencies of about 85% are kept for part load applications and the transient system behavior can be better. Most importantly, boiling point of ORC fluids chosen as working fluids are less than water; hence, they can be used with lower temperatures heat sources [4].

There are some other alternatives for power generation for low temperature heat sources. Kalina cycle and transcritical CO_2 power cycles are two important examples. Due to gliding temperature for Kalina cycle working medium in evaporation and decreasing temperature during condensation, better heat transfer from heat source and to the heat sink is expected which can result in better efficiency for Kalina cycle. However, to achieve this, higher maximum pressure will have to be maintained in the Kalina cycle in comparison with ORC and this could make the Kalina cycle more expensive. More

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components such as an absorber and separator are needed, turbine needs to be a multistage device or have a high rotational speed and since the medium is a mixture and normally is from water and Ammonia, which is deemed to be corrosive. All in all, these will result in bulkier systems, more components thus resulting in higher electricity prices from Kalina cycle in comparison with ORC [4].

Transcritical (CO_2) power cycles can also be used to generate shaft power and electricity from low temperature heat sources same as vehicles exhausts, geothermal and solar energy resources. Phase change, as in a Kalina cycle, occurs in non-isothermal conditions which will enhance the heat transfer but is poor thermodynamically. The working medium is CO_2 with 4-10 time larger slope in vapor pressure diagram (increases in temperature, enthalpy and pressure with entropy or volume) in comparison with many fluids which will give better enthalpy transfer criteria for the cycle.

Transcritical power cycles could produce more power than ORC with better efficiencies. An important obstacle or disadvantage for such improvement is that the medium has properties in between liquid and the gas. This makes the design and development more difficult due to the lack of appropriate expanders which work in multi-phase situations. The more recent achievements in multi-phase turbines make this technology more feasible today. Arrangement of the blades in these multi-phase turbines can be similar to normal impulse axial turbines; however, special shape of the blades will need to be able to withstand the small liquid droplets in multi-phase flow and the associated erosion.
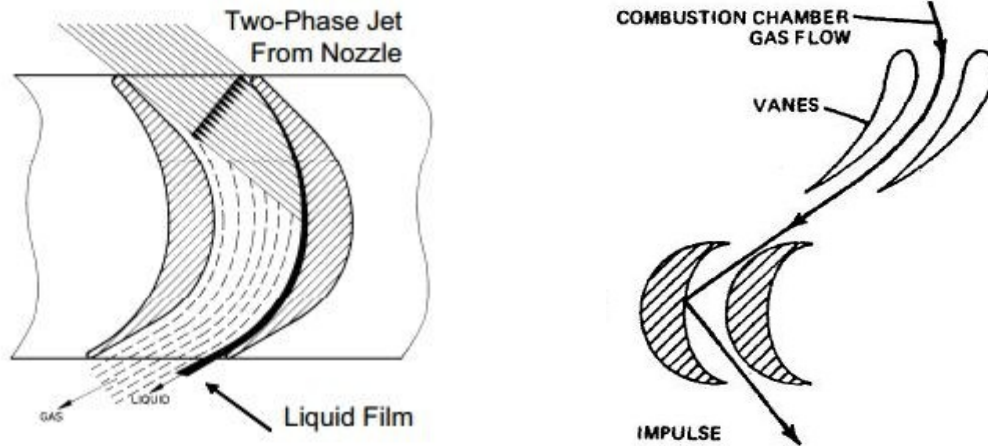
Figure 1.2. Variable phase turbine blades in comparison with normal turbine blades [4]

Overall, for low temperature heat resources, the Organic Rankine Cycle (ORC) systems are the most widely used. ORC systems involve components similar to those in a conventional steam Rankine power plant (a boiler, a work-producing expansion device, a condenser and a pump) but use organic substances as working fluids in order to utilize low-grade heat sources. An ideal working fluid should be non-toxic, eco-friendly, non-corrosive, non-flammable, have a high heat of vaporization and density, isentropic saturation vapor curve, the low-grade boiling point and most importantly be inexpensive.

1.3   Overview of Methodology

An extensive literature survey that included traditional journals, digital libraries and brochures from manufacturers has been conducted. The survey includes system costs, configurations, power and temperature ranges, and special considerations related to ORC technology.

During the evaluation, life cycle cost of ORC, quality and quantity of produced energy and ORC properties as well as its initial cost has been taken into account. The

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components of ORC have been scrutinized and some recommendations have been proposed for cost reduction opportunities. To reach the aforementioned goals, equipment of ORC needs to be selected properly. Issues considered included the working fluid, speed of turbo machineries, degree of superheat, working minimum and maximum temperatures and pressures for some of the applications.

A numerical computational model for thermal and exergy efficiencies, heat transfer factor and turbine size has been developed. The effect of selecting components with different characteristics and ORC features has been studied. The model outputs include information on cycle performance and cost mostly as non-dimensional parameters.

1.4   Objective

Demand for efficient and affordable power generation systems and environmental concerns encourage the investigation of cost effective ORC. ORCs are environmentally friendly option to meet an important need. Rural distributed electricity generation for nearly 2 billion people and better waste heat recovery of different industries in addition to Renewable based power generation can benefit from this.

A comprehensive study for different applications of cycle for several ORC technologies is presented here. Today, Organic Rankine Cycles are commercially available in the MW power range. This paper presents an overview of small scale (kW) ORCs to be applied in decentralized rural space and for scarce of water environments. In this work, an alternative system is proposed, consisting in the coupling between a low-cost solar concentrator and an Organic Rankine Cycle engine. The goal of the project is to provide rural areas, especially in developing countries with a system that can be applied and

manufactured locally and can replace or supplement diesel generators in off grid areas, by generating clean power at a lower level cost.

In the rest of this thesis, each component and process of the ORC in the technology will be studied and suggestions will be made to reduce the overall cost, improve the efficiency and power output of cycle.

CHAPTER 2

SURVEY OF LITERATURE

2.1 Previous Work

During the last twenty years, concentrated solar thermal based electricity generation has significantly increased globally. About one percent of market share of ORC is related to this technology. In the below discussion, benefits and the huge potential for this technology for ORC applications will be introduced.

It is generally believed that higher temperature and enthalpy for power cycles yield better performance; as a result steam cycle using water is normally used in solar thermal electricity production systems and attempts have been done to achieve higher temperatures by creating steam at higher enthalpies using solar energy and passing it through a turbine [6]. The most common modern technique to produce steam for electricity production or industrial needs has been to utilize a well-established heat transfer and or working fluid. Usually a salt or oil is pumped through a solar field as the heat transfer fluid to heat up the working fluid. The hot oil is then passed through a heat exchanger with water or other working fluid to generate steam. The benefit of a heat transfer fluid is that it remains liquid at high temperatures, which increases heat transfer and ease of pumping. Oils have a liquid working range up to 400℃, salts up to 600℃ and liquid metals used as heat transfer fluids can reach much higher temperatures.

The Solar Total Energy Project in Shenandoah, Georgia used a heat transfer fluid to create electricity and process steam for a textile factory. The heat transfer fluid used was Syltherm 800, a silicon-based fluid produced by Dow-Corning Corporation. The Syltherm carried 2.6 MW equivalent of heat from 114 7-meter-diameter concentrating parabolic dishes which the total aperture area is $4352m^2$ [7]. The fluid was pumped to cavity receivers that were placed at the focal point of each dish. The collector efficiency of the concentrators was 76.9%. Three heat exchangers were then utilized to preheat, boil and superheat water before it was passed through a steam turbine. A portion of the steam generated was used directly by the factory. The remaining hot water was used as a heat sink for a lithium bromide absorption chiller. Although thermal-to-electric conversion is only 11.8%, the total heat work efficiency for all of the processes is 44.6%. The output of the Shenandoah solar energy system is 400kW electricity maximum whereas the input is 4500kW. According to the latest report, $0.15 per kWh cost is expected to be reduced by perfecting the whole system.
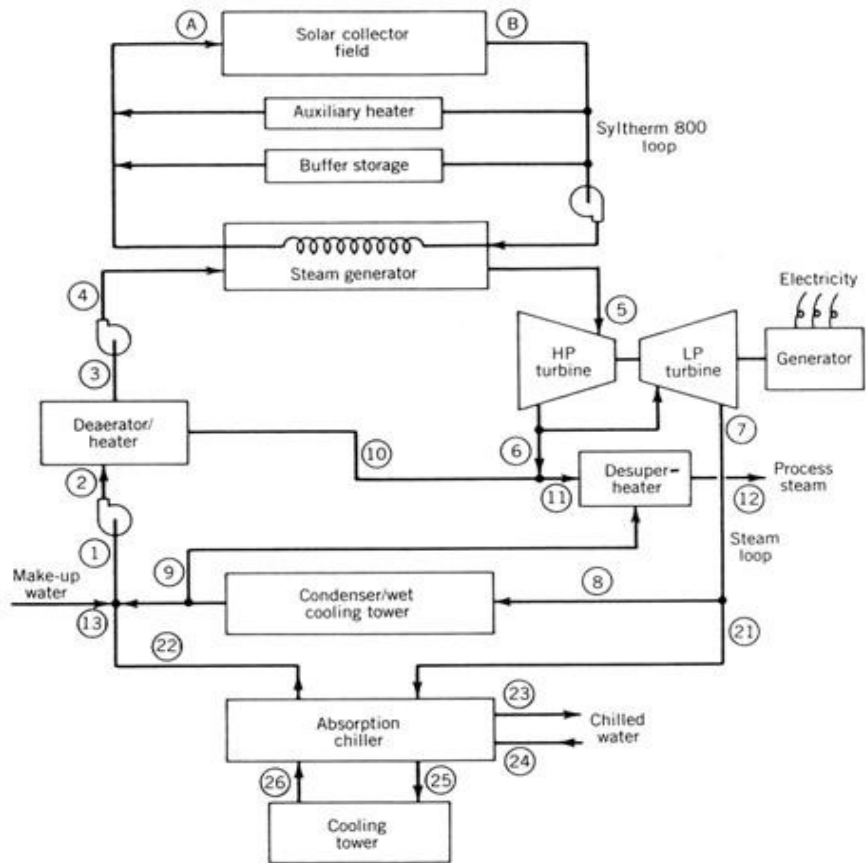
Figure 2.1. Flow diagram for the Shenandoah solar total energy project [7]

However, looking more precisely to the system will show several justifications for using ORC instead of a steam Rankine cycle. First, turbo machineries for steam cycle are not so efficient in related rather low temperatures of solar systems in comparison with conventional systems. In addition, it is necessary to superheat the steam up to 600°C to avoid condensation during expansion in steam turbine. These temperatures for solar thermal systems are so expensive to be achieved. Thermal energy storage is not even commercialized in these temperatures and solar detectors and high concentrator equipment is expensive in these cases. This is precisely where using ORC instead of steam Rankine cycle does not have these drawbacks. No freezing of working fluid during cold weather threatens ORC fluid while this problem exists in steam cycle, which uses

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water. Since in ORC less heat source temperature in comparison with steam cycle is used, less costly material for solar collector and concentrators is not needed and this will reduce the total cost of system in comparison with steam cycle. These systems can be combined with phase change material (PCM) storage system in relatively reasonable costs to increase the period of power generation.

Recovering low- and medium-temperature heat (ranging from 60°C to 200°C), which includes industrial waste heat, geothermal energy, solar heat, biomass and so on, is an important sustainable way to solve the energy crisis. Organic Rankine cycle (ORC) systems are feasible for generating power from these low- and medium-temperature heat sources. The advantages of ORCs are high thermal efficiency, simple turbine, dry expanding process, no emission of exhaust gas, particularly under low temperature and exergy conditions [8]. Theoretical analysis of several working fluids shows that cycle efficiency is very sensitive to evaporating pressure, but insensitive to expander inlet temperature. An ORC testing system was established and operated using R600a as a working fluid and hot water as a heat source [8]. The maximum cycle efficiency of the testing system is 5.2 percent, and the testing result also proves that cycle efficiency is insensitive to heat source temperature, but sensitive to evaporating pressure. The entropy calculations also show that internal and external entropy of the evaporator is the main source of total entropy generation.

Since the boiling point and other physical properties varies for different organic fluids. An investigation of performance for different working fluids based on temperatures and pressures is necessary. A thermodynamic screening of 31 pure component working fluids for ORC is given and also resulted in the use of BACKONE equation of state [9]. In

order to evaluate the thermal efficiency η_{th} of a cycle and the T, ΔH-diagram for some working fluid, its thermodynamic properties must be known and described accurately by a fundamental equation of state (EOS). Reference [9] presents and discusses the BACKONE equation and the parameters for 31 pure components to be considered as working fluids for ORC processes. Most EOS which have sufficient accuracy as, e.g., those included in REFPROP [10] use a large number of substance specific parameters which need to be fitted to an extensive set of precise experimental data. On the other hand, simple general EOS like cubic EOS have only few parameters and need only a small data base but they do not have sufficient accuracy.

The BACKONE equation is a better alternative for EOS. Strictly speaking, BACKONE is a family of physically based EOS [11], which is able to describe thermodynamic properties of nonpolar, dipolar and quadrupolar fluids with good to excellent accuracy. We can assign state number to the inlet and outlet of different devices in the cycle, using BACKONE to calculate the cycle efficiency ideally.

HFC-245fa has some serious considerations as a working fluid for ORCs. HFC-245fa is a suitable working fluid for low and medium temperature cycle (60°C to 200°C) and its properties and potential application in power generation system should be better investigated [16]. The commercially available Refrigeration Grade HFC-245fa meeting Air-Conditioning and Refrigeration Institute Standard 700 has provided a viable option for safe, flexible, and economically efficient conversion of waste heat to electric power using Organic Rankine Cycle technology. Reference [9] explicitly listed useful physical properties of HFC-245fa, like enthalpies, entropies, boiling temperature and latent heat based on given conditions. In addition to this, cycle efficiencies are theoretically

calculated for temperatures ranging from 148°C to 176°C of outlet of evaporator which provides an important reference for our research. It should be noted here that all the assumptions and experiments are based on getting heat from geothermal heat in this paper whereas the current focus is on solar thermal energy. The difference of heat resource would affect energy collector field for ORC. Another article states good properties of some fluids as low 'global-warming' replacement of HFC-245fa and HFC134a in ORC applications.
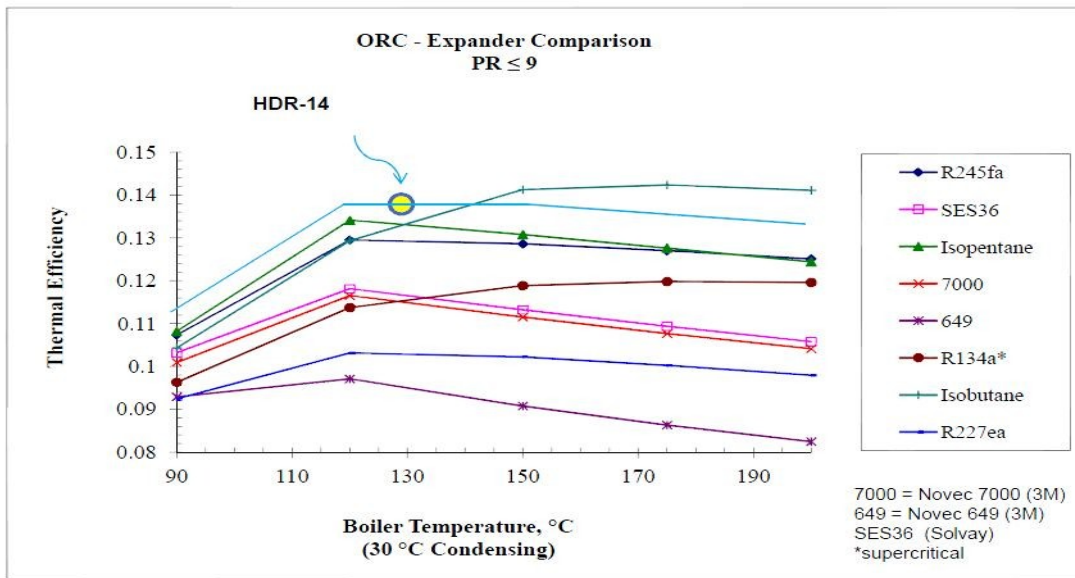
Figure 2.2. Different Organic fluids thermal efficiencies, boiling temperature from 90°C to 200°C at 30°C condensing temperature [11]

In terms of cycle parts, for the solar collector field, researchers at MIT and University of Liège [10] have collaborated with the non-governmental organization Solar Turbine Group (STG) International for the purpose of developing and implementing a small scale solar thermal technology utilizing medium temperature collectors and an ORC. A first unit was installed by STG in 2007 [10], and is shown in Figure 2.3.

Figure 2.3. Solar ORC prototype installed by STG in Lesotho [10]

The goal here is to provide rural areas of developing countries with a system that can be manufactured and assembled locally (unlike PV collectors) and can replace or supplement diesel generators in off-grid areas, by generating clean power at a lower level cost. At the core of this technology is a solar thermal power plant consisting of a field of parabolic solar concentrating collectors and a vapor expansion power block for generating electricity. An electronic control unit is added for autonomous operation, as sub-megawatt scale plants cannot justify the staffing of operating personnel.

For the turbine, many MW scale turbines which are commercially available can be applied on the ORC, in contrast, few kW scale turbines for ORC could be found for commercial usages right now. Infinity Turbine® provides customized waste heat turbines for solar applications in kW scale [12]. Experimental efficiency of a compact ORC system with a compact rotary-vane-type expander first was being proposed by Japanese scientists [14]. The compact ORC system can be used for power generation from low-

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temperature heat sources such as waste heat from various small-scale heat engines, fuel cells, electric devices, and solar thermal energy. This article [14] proposed an ORC system with a low power output of less than 1kW with a relatively high temperature source ranging from 60°C to 100°C and a cold temperature source ranging from 10°C to 30°C.
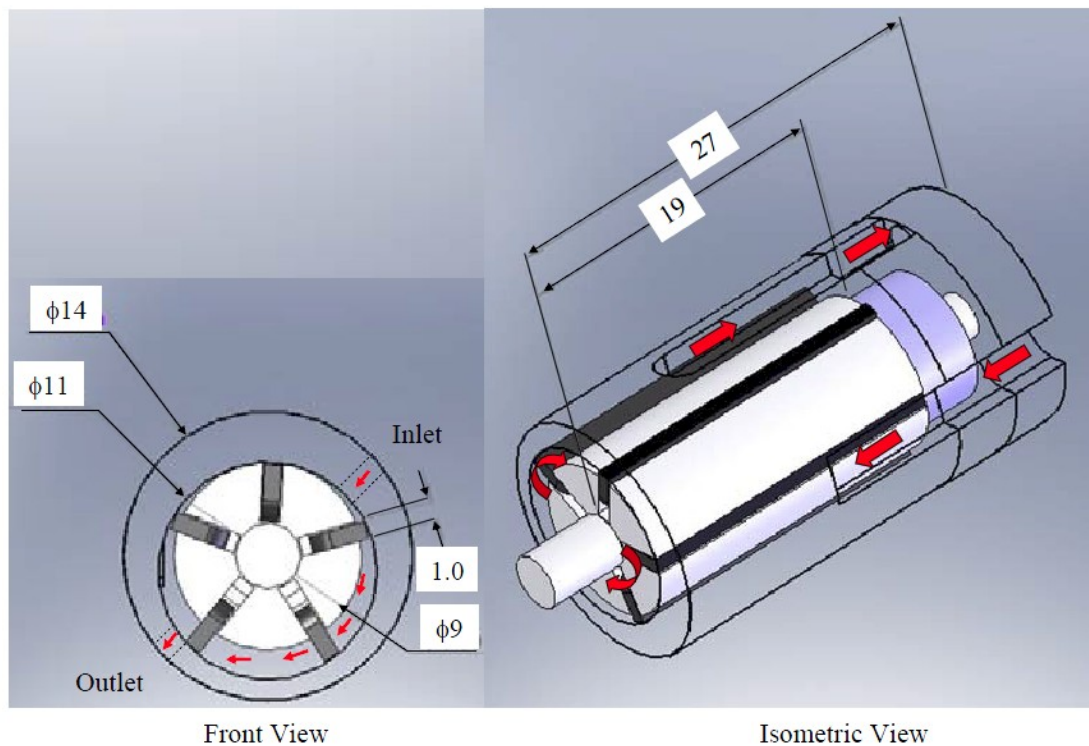
Figure 2.4. Rotary-vane-type expander used in the proposed ORC system [14]

In this system, HFC-245fa was employed as the working fluid. The measured efficiency of the rotary-vane-type expander was 43%, 44%, 48% for a heat source temperature difference (between inlet and outlet) of $\Delta T = 60°C$, 70°C and 80°C, resulting in theoretical thermal efficiencies were 8.85%, 8.72%, and 9.41% respectively. These efficiencies are much lower than normal ORC efficiency usually around 18% due to lower heating temperature and lower pump pressure. Issues such as low expander

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efficiency, high pump power consumption, and lack of insulation must be resolved in order to improve the thermal efficiency of the proposed system.

2.2 Project Content

In this work, a sample ORC based on solar thermal system is proposed. Device selection is the first part of this work mainly based on temperature and pressure. Since the target utilization environment is decentralized rural space, cost is a critical point for being taken into account. In the proposed system, an energy storage tank is added to the cycle as a relative new variant. For short periods solar power outage due to unpredictable weather change (cloudy day, dark cloud), energy storage tank is connected to the inlet of boiler as a back-up heat provider. The size of the tank would be based on the anticipated power outage time(s).

Several organic fluids are used as working fluid for different heating temperature and cycle pressure. Cycle is operated at temperature ranging from 40°C to 200°C while being condensed at 30°C by air. The target net output power considered varies from 5 kW to 1MW. In the later part of this work, theoretical cycle efficiency is given as a judgment of cycle performance showing how the temperature, pressure and fluid selection affect the efficiency. An economic investigation is shown as energy costs for units $/kW and $/kWh format as evaluated in the work.

CHAPTER 3

PROPOSED ORC SYSTEM

3.1   The Proposed Solar ORC System

The ORC system under consideration is expected to operate thermal energies available at cycle temperatures (< 200℃). As with all heat engines, Carnot efficiency will establish upper limits for efficiency and could be an impediment to design trade-offs for maintaining low cost at small scales. For a given level of power output, while lower temperatures enable cost savings in the materials and manufacture of the absorber units, heat exchangers, fluid manifolds and parabolic troughs, lower temperatures also lead to low efficiencies. Device selection has to be designed for different output applications. The target net output power for the current system varies from 5 kW to 1MW.

The main components in the classical Rankine cycle include an evaporator, an expander, a condenser and a recirculation pump to increase the pressure. The heat transfer fluid is heated up in the collector field and driven to the evaporator by the heat transfer fluid pump. Then heat transfer fluid is used to evaporate the organic fluid of the ORC as evaporator which is connected to an expansion unit where useful work is produced. In an ORC with a 'dry' fluid, the recuperator exchanges energy from the superheated exhaust to the pump outlet in order to preheat the liquid before the evaporator, which increases the cycle efficiency. This superheated exhaust is also readily exploitable for cogeneration purpose in order to recover the residual heat and produce hot water, depending on the local demand. Next, working fluid is condensed in the air condenser and pumped back to

the evaporator. A thermal storage is installed in order to attenuate the fluctuations of solar irradiation during the day, like sun is blocked by cloud temporarily, to maintain a stable and continuous energy supply. The thermal storage tank size would be determined based on the maximum outage time of solar irradiation in the following chapters. In the proposed ORC heat exchangers (evaporator, recuperator and condenser) are sized in order to avoid unnecessary water consumption under restrictions.
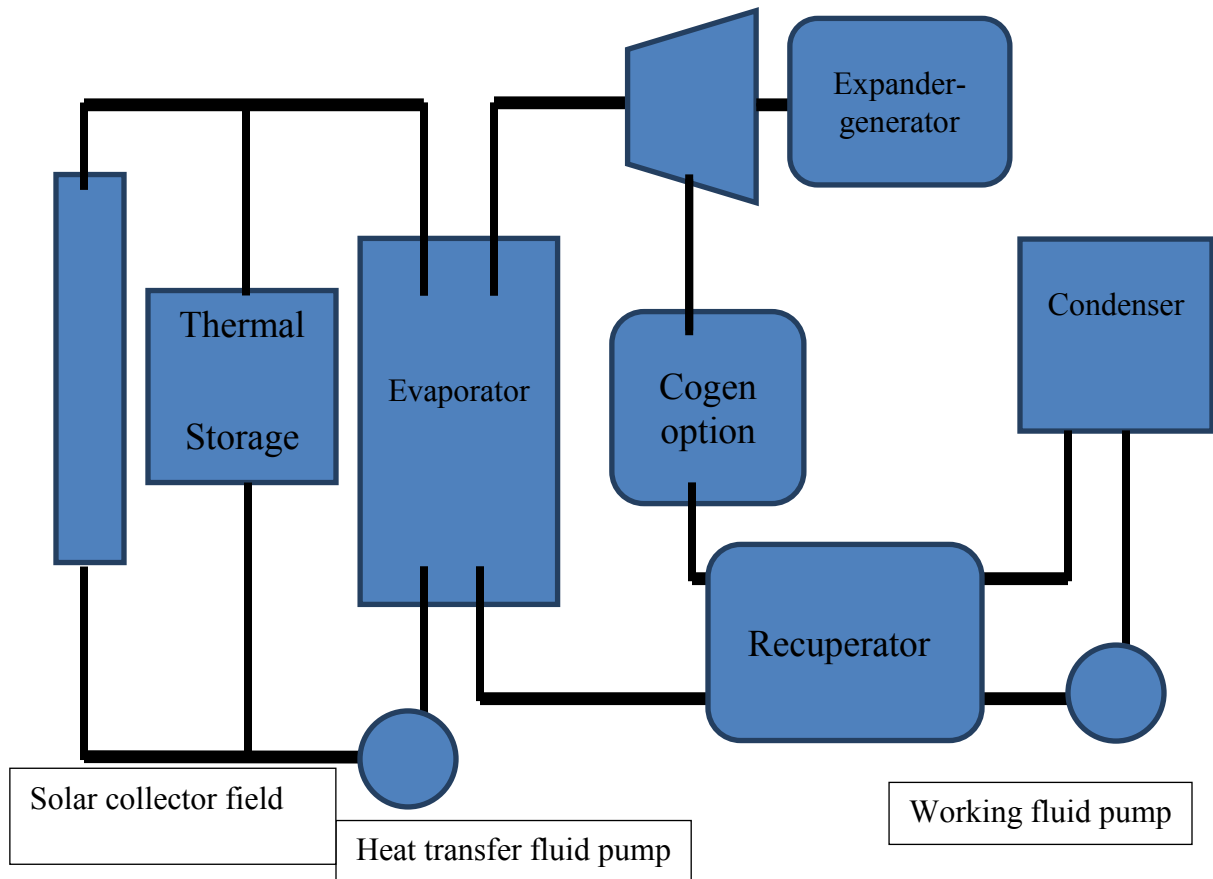

Figure 3.1. Schematic of the proposed solar ORC

The main characteristics of this cycle are the following:

- Objectives of electricity output are 5 kW, 10 kW, 25 kW, 250 kW, 500 kW and 1 MW which determine the devices properties in the cycle.

- Refrigerant fluids to be circulated in this cycle, as working fluids are R-123, R-245fa.

- Heat transfer fluid (HTF): Monoethylene glycol (MEG) with thermal buffering in a thermal storage tank.

- Operation pressure ranges from 10kPa to 10MPa.

As stated above, cost is a crucial parameter for the competitiveness of the technology.

Several strategies and tradeoffs are set up in order to reduce the cost of the installed kWe:

- An electronic control unit is added for autonomous operation since sub-megawatt size plants cannot justify large staffing of operating personnel.

- While the thermodynamic efficiency of heat engines increases with operating temperature, solar absorbers will have greater thermal losses due to convection and radiation. Mitigating these losses under high-temperature operation, e.g. by means of selective high absorptivity and low emissivity coatings and evacuated glass glazing, entails higher costs in materials and manufacturing. These features may be economically justified in megawatt-scale installations, where large collector production volumes lead to lower specific costs and where the cost of land lease is an important consideration. Where the cost of space is not a constraint, however, and where small capacity units of deployment are preferred, optimization for the lowest Levelized cost of Electricity (LCOE) may lead to medium temperature designs involving lower thermal efficiencies and increased footprint, e.g. from 20 m^2/kWe in a typical large scale Concentrating Solar Power (CSP) plantto 35m^2/kWe in a medium temperature CSP plant [15].

- The use of widely available components (mainly from the HVAC) market allows economies of scale compared to components that would be specially designed for ORC units.

3.2 Cycle Efficiency Analysis

As stated above, the working conditions (boiling temperature, turbine inlet/outlet temperature, cycle pressure, condenser temperature) mainly determine the cycle performance. Another important aspect of the operation of ORCs is the selection of the working fluid. The organic working fluid must be carefully selected on the basis of safety, fluid properties, temperature at which thermal energy is available and technical feasibility. There is a wide selection of organic fluids that could be used in ORC such as

chlorofluorocarbons (CFCs), Hydrochlorofluorocarbons (HCFCs), Hydrofluorocarbons (HFCs), and hydrocarbons (HCs). Performances and characteristics of different working fluids for waste heat recovery system can be found in references [15] and [16]. Generally, a good working fluid should exhibit low toxicity, good material compatibility and fluid stability limits, and low flammability, corrosion, and fouling characteristics.

Refrigerants are good candidates for ORC applications because of their low-toxicity characteristics. Another characteristic that must be considered during the selection of organic fluid is its saturation vapor curve. This characteristic affects the fluid applicability, cycle efficiency, and arrangement of associated equipment in a power generation system [17].The slope of the saturation curve in the T–s diagram depends on the type of fluid employed. A dry fluid has a positive slope; a wet fluid has a negative slope; and an isentropic fluid has infinite large slopes. Generally, dry and isentropic fluids are better working fluids for an ORC because they do not condensate after the fluid goes through the turbine, which means the turbine blades have a lower risk of being corroded. The comparison of the temperature–entropy diagram for dry, wet, and isentropic fluids is presented in Figure. 3.2. The working fluids employed in this investigation are classified as follow: R113, R123, R245ca, R245fa, and isobutene are dry fluids, whereas water, R134a, and propane are wet fluids. Isentropic fluids such as R12 and R22 were not included as they are being phased-out and replaced with alternative refrigerants. Hung et al. [17] presented an analysis of ORC using R134a and R113, but they did not include a second-law analysis. Hung [17] performed a second-law analysis of an ORC using R113 and R123. Although some of the fluids selected for this investigation have been already analyzed for different cycles, R113 and R123 are the

only ones that have been specifically evaluated for ORC from the exergy point of view. Therefore, this article presents combined first- and second-law analysis for additional fluids such as R245fa, R113 and R123 for use in ORCs.
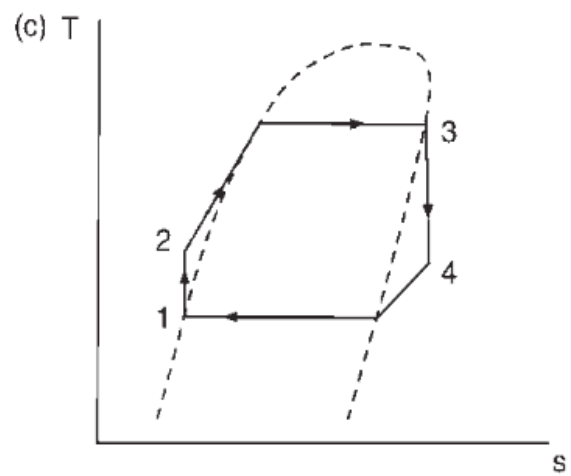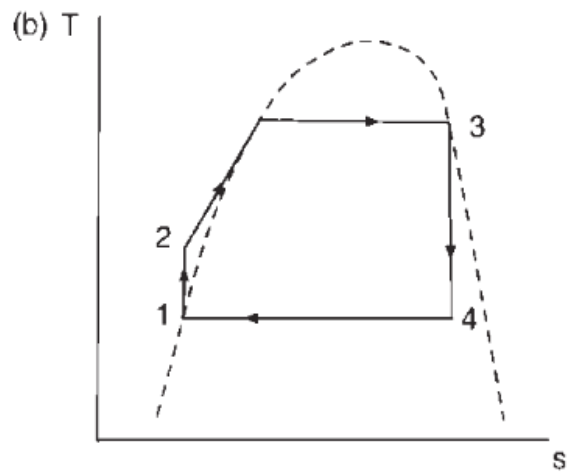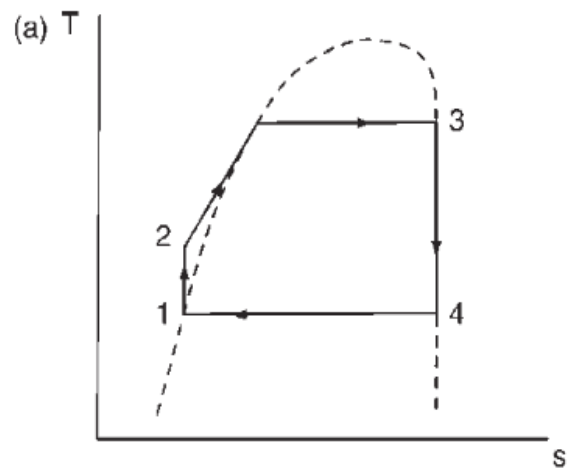
Figure 3.2. Comparison of the working fluids: (a) isentropic, (b) wet, and (c) dry.

The objective of this article is to study the change in thermal efficiency and irreversibilities of an ORC using different working fluids varying system-operating parameters at various reference temperatures. Also, the influence of the fluid boiling point temperature on the performance of ORCs is investigated.

a. Working Fluid

Based on our desired operating conditions (low temperature around $40\,°C$ and high temperature around $200\,°C$, low pressure around 100kPa and high pressure around 10MPa), refrigerant fluid have better efficiency than isobutene fluid.

Comparison of several suitable working fluid properties:

Type	AHARE Number	Molecular Formula	Molecular Mass (kg/mol)	Normal Boiling point(°C)	Critical Temperature (°C)	Critical pressure (absolute kPa)
HFC	R-245fa	C3H3F5	134.0477	15	154.05	3,640
HCFC	R-123	C2HF3Cl2	152.9274	27.6	183.68	3,662
HCFC	R-113	C2Cl3F3	160.921	**47.7**	214	3417

Table.3.1. Properties of working fluids [11]

In order to get an optimal operation temperature and pressure, we have to explore their physical properties carefully.

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The temperature entropy diagrams for some of the ORCs and fluids considered are shown below:

Figure 3.3. Temperature-entropy diagram of R245fa, R123 and R113 [11]

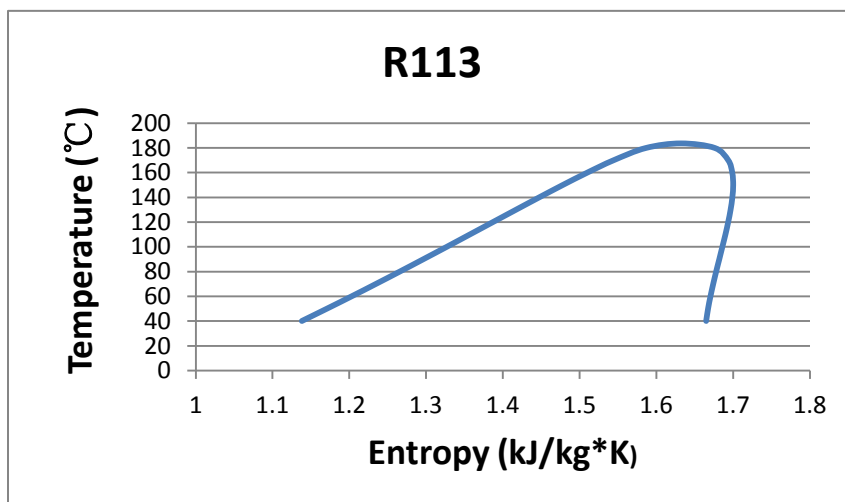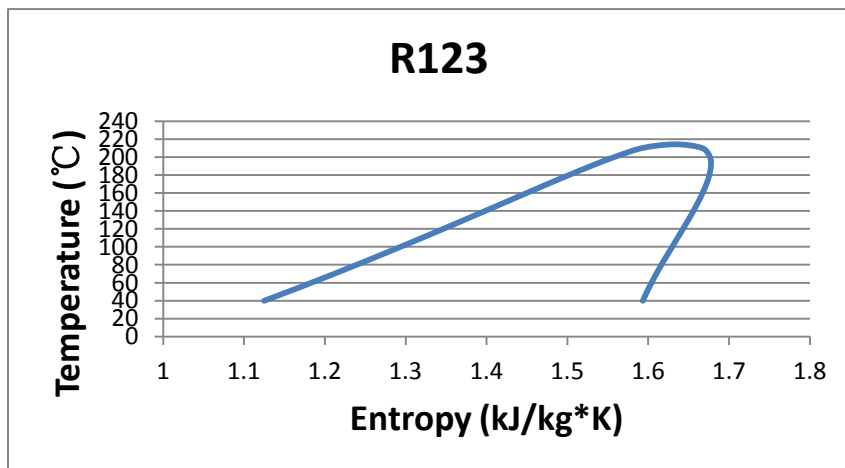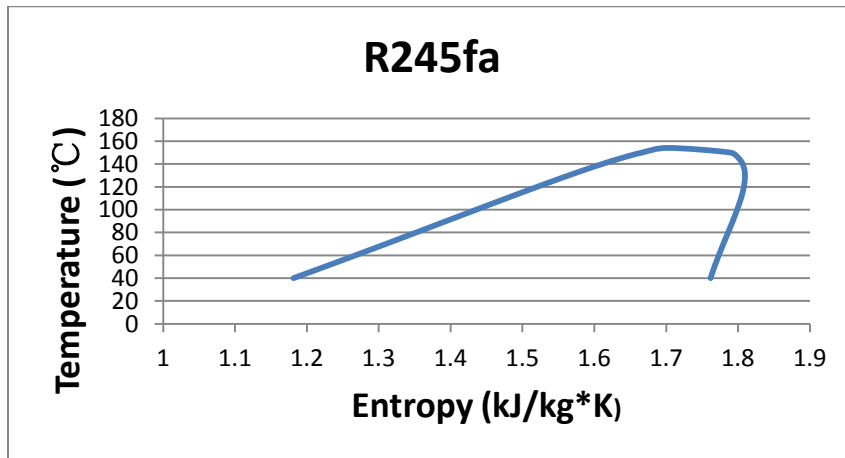From the T-s diagram above we can know that the chosen fluids are all dry fluids which means the fluid at outlet of turbine is at superheated state. A recurperator heat exchanger is supposed to be installed between the outlet of turbine and inlet of evaporator which improves the efficiency of the cycle by taking more of the heat out of the working fluid before it is condensed and transferring the heat to the working fluid before it enters the evaporator. According to a study done by T.C. Hung, et.al. [17], the efficiency of the cycle is closely related to the latent heat of the fluid at low pressure; a greater latent heat at low pressure yields a lower efficiency since a larger portion of the energy carried by the fluid is rejected via the condenser.

In order to calculate the efficiency, some conditions, such as the evaporator outlet temperature Tin, the turbine inlet pressure Pin ($=P_4$) and the turbine outlet pressure Pout should be assigned at each stage. Due to the solar collector field temperature restriction, we assign a target temperature range (40℃ to 200℃) to calculate the cycle efficiency. Pin is expected to be got based on the Tin. Since targeted fluids normal boiling point varies (R113>R123>R245fa), a corresponding sequence is supposed to be listed: Tin(R113)>Tin(R123)>Tin(R245fa).

As normal ORC, recuperator (2-3) which preheats the working fluid, evaporator (3-4) which changes the phase of working fluid from liquid to vapor by gaining heat from the heat source, turbine (4-5) which expands the working fluid and extract power from it, recuperator will exchange heat from outlet of turbine inlet to inlet of evaporator to preheat fluid (5-3) before it goes into evaporator which can increase the cycle efficiency in a cost effective way. After exchanging heat, condenser (6-1) that removes the heat

from fluid at saturated liquid or mixture of saturated liquid and gas state makes it liquid

and pump (1-2) increases the liquid pressure.
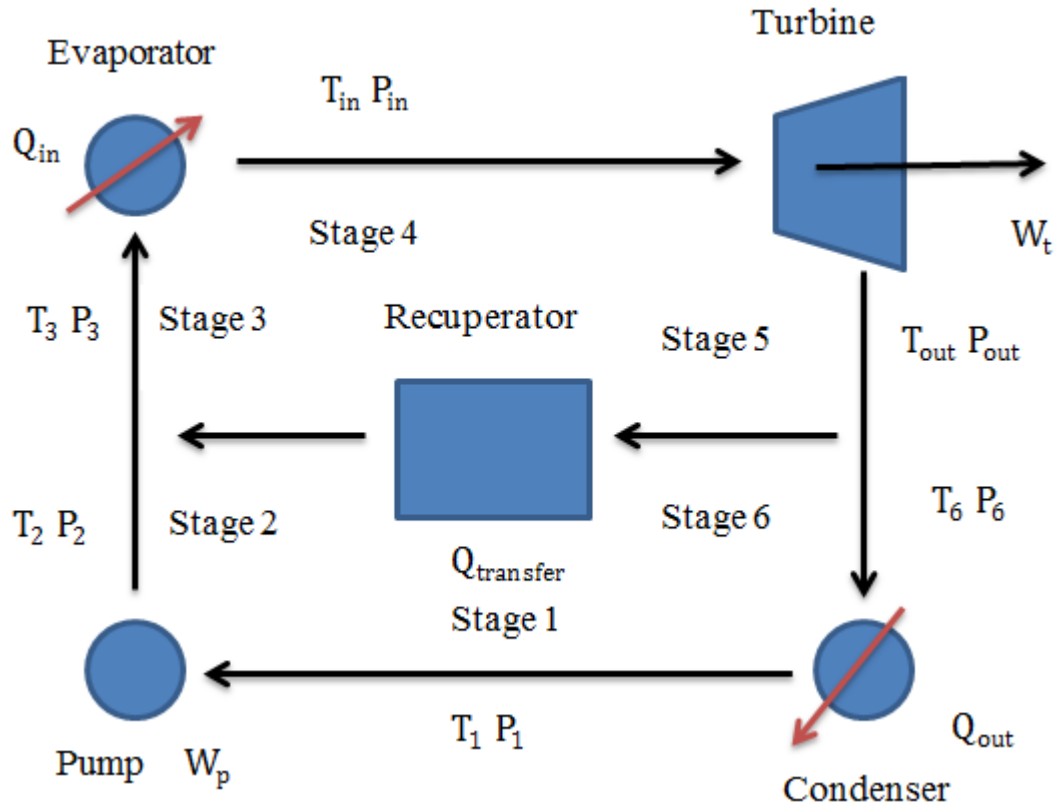
Figure 3.4. Schematic of organic Rankine cycle [15]

The turbine inlet vapor is normally set to be a superheated or a saturated phase at stage 4.

The efficiency is also a weak function of the turbine inlet temperature, i.e. an increase of

superheat in the turbine does not result in a significant increase in efficiency.

Consequently, the ideal situation should be that the fluid at outlet turbine is saturated

vapor. The working fluid passing through the condenser is assumed to be a saturated

liquid (T_6, P_6). The power input is determined by the circulating mass flow rate (kg/s).

The operating conditions of an ORC can greatly affect how efficient the cycle is. The

main factors affecting the efficient operation of an ORC are the maximum temperature,

the maximum pressure, the amount of heat input, and work output from the system. In order to compare how efficient a cycle operates at these different conditions the thermal efficiency can be calculated for each condition and compared. Since many operating points for these cycles exist a theoretical ORC was designed and analyzed multiple times. The cycle was analyzed for three different fluids. For each fluid, the cycle was analyzed at multiple boiler pressures within the system. For each pressure of each fluid multiple iterations were performed to analyze the cycle under different operating conditions. Several different iterations were performed for each pressure of each fluid and only one aspect of the cycle was changed each time.

The tables below show the number of iterations and the parameters altered for each fluid and each pressure. Tables 3.2, 3.3, and 3.4 are for R-245fa, R-123, and R-113 respectively. T_{max} and P_{max} refer to the temperature and the pressure in the boiler of the ORC. W_{out} refers to the work output of the turbine, and $\dot{m}$ is the mass flow rate of the refrigerant within the cycle. The control run, iteration number 1, has a mass flow rate of 1.0kg/s. These are considered to be the default values and are held constant except when the temperature and pressure are varied.

Iteration	Pmax	1200kPa	1000kPa	800kPa	600kPa
1	Tmax	Tsat	Tsat	Tsat	Tsat
2	Tmax	Tsat+5℃	Tsat+5℃	Tsat+5℃	Tsat+5℃
3	Tmax	Tsat+10℃	Tsat+10℃	Tsat+10℃	Tsat+10℃

Table 3.2.Iterations and Altered Parameters for Theoretical
Analysis (R-245fa)

Iteration	Pmax	1800kPa	1500kPa	1200kPa	900kPa
1	Tmax	Tsat	Tsat	Tsat	Tsat
2	Tmax	Tsat+5°C	Tsat+5°C	Tsat+5°C	Tsat+5°C
3	Tmax	Tsat+10°C	Tsat+10°C	Tsat+10°C	Tsat+10°C

Table 3.3.Iterations and Altered Parameters for Theoretical Analysis
(R-123)

Iteration	Pmax	2400 kPa	2000 kPa	1600 kPa	1200 kPa
1	Tmax	Tsat	Tsat	Tsat	Tsat
2	Tmax	Tsat+5°C	Tsat+5°C	Tsat+5°C	Tsat+5°C
3	Tmax	Tsat+10°C	Tsat+10°C	Tsat+10°C	Tsat+10°C

Table 3.4.Iterations and Altered Parameters for Theoretical
Analysis (R-113)

In order to calculate the specified state points within the cycle the first law of thermodynamics was applied to the system. The first law of thermodynamics in equation form is shown below.

$$\Delta E = (\dot{Q}_{in} - \dot{Q}_{out)} - (W_{in} - W_{out)} = 0 \qquad (3.1)$$

Where delta E is the change in energy which is equal to zero for a cycle, Qin is the heat added to the boiler from the waste heat application, Qout is the heat rejected to the environment from the condenser, Wout is the work produced by the turbine generator combination, and Win is the work required by the pump to move the working fluid.

The analysis began with state one, the inlet to the pump. The circulation pump is the driving mechanism of the proposed system. Since the fluid is desired to be condensed by air at 30°C, allowing a 10°C temperature difference for effective heat transferring, the fluid enters the pump at 40°C as saturated. P_1 could be get from checking the fluid thermo table. The working fluid (saturated liquid) leaving the condenser at low pressure P_1 regains high pressure here to P_2. The working fluid is pumped back into the evaporator. The circulation pump W_p is calculated by the following equation:

$$\dot{W}_p = \frac{(P_2 - P_1) \cdot \dot{m}}{\rho} \tag{3.2}$$

Where ρ denote the density of working fluid(saturated condition)

$$h_2 = h_1 + \dot{W}_p \tag{3.3}$$

After the fluid is pumped out as compressed liquid (P_2 T_2), the heat that comes from some part of the latent heat where saturated vapor is condensed preheats the liquid (P_3 T_3):

$$h_3 = h_2 + \dot{Q}_{Transfer} \tag{3.4}$$

Whereas $\quad \dot{Q}_{Transfer} = (h_{5a} - h_6) \times \eta_{recuperator} \ (\eta_{recuperator} = 0.85) \tag{3.5}$

At evaporator part:

$$\dot{Q}_{in} = \dot{m} \times (h_4 - h_3) \tag{3.6}$$

Since it was desired that the fluid be in vapor form at the exit of the evaporator and the quality of state four was specified to be one. The temperature of state four was then assumed to be the temperature at which the quality was one and the pressure was equal to

the first iteration pressure. With two independent properties defined, RefProp could be used to determine the enthalpy, density, specific heat, and entropy of the fluid at state four. Using the ideal assumption that the expansion in the turbine happens isentropically the entropy in state five should be equal to the entropy in state four.

$$S_4 = S_{5s} \tag{3.7}$$

Another independent property is still required to calculate all the variables within state five. To calculate this needed variable, we assume that the pressure of the outlet of the turbine equals the pressure at state 6. With the known entropy and pressure, the enthalpy can be got from RefProp. Since the turbine efficiency is assumed at 85%, the equation for work output from a cycle is modified and is shown in equations below:

$$\frac{(h_4 - h_{5a})}{(h_4 - h_{5s})} = \eta_t = 0.85 \tag{3.8}$$

$$W_t = \dot{m} \cdot (h_4 - h_{5a}) \tag{3.9}$$

The superheated vapor of the working fluid passes through the cellular recuperator which is the most efficient recuperator in the market, transferring some of the waste heat in the exhaust to the compressed fluid, thus preheating it before entering the evaporator stage. Since the liquids have been pre-heated, less power is needed to heat the liquids up to the turbine inlet temperature. By recovering some of the energy usually lost as waste heat, the recuperator can make the ORC significantly more efficient.

$$(h_{5a} - h_6) \cdot \dot{m} \cdot \eta_{recuperator} = \dot{Q}_{Transfer} \tag{3.10}$$

Whereas $\eta_{recuperator}$ is assumed at 85%.

In actual power plant, the fluid leaves the recuperator above the saturated temperature at condenser pressure because a temperature difference of at least a few degrees is required for any effective heat transfer to take place.

The working fluid goes through a constant pressure phase change process in the condenser into a state of saturated liquid, rejecting the latent heat into the environment or the condenser coolant. The pressure of the working fluid within the condenser is equal to the Rankine cycle lower pressure: P_1 and the temperature is equal to the saturation temperature of the pressure P_1. The condenser load, Q_{out}, which is the rate of latent heat rejection from condensing working fluid, can be calculated from the following equation:

$$\dot{Q}_{out} = \dot{m} \cdot (h_6 - h_1) \qquad (3.11)$$

The cycle efficiency is calculated using the following equation.

$$\eta = \frac{W_t - W_p}{\dot{Q}_{in}} \times 100\% \qquad (3.12)$$

As in most Rankine cycles, much of the pressure drop occurs in the turbine and condenser processes, and the pressure rise occurs in the pumping process. However, as explained above, it assumed that the pressure rise occurs in the circulation pump only, and the pressure drop occurs in the turbine only for a simpler thermodynamic analysis. Though ORC systems rarely operate under steady state conditions, this presents analysis assumed steady state operation, no heat loss and pressure changes in the system except the ones as mentioned above. This means that the temperature and pressure at the evaporator outlet equals to the turbine inlet. Though it is well known that the isentropic efficiency of the expansion process depends on the turbine design as well as fluid

properties, in the present work, the adiabatic efficiencies of the turbine and the recuperator are both assumed to be 85 percent.

Once the fluids were selected the ORC performance of each fluid was analyzed at a range of boiler pressures, temperatures, flow rates, and assumed power. These specified operating conditions are shown in Tables 3.3, 3.4, and 3.5. The pressures listed correspond to the maximum pressure in the system which is also the pressure in the boiler and turbine entrance. The analysis was then carried out for each fluid where the boiler pressure and temperature was varied multiple times. Initially, the temperature investigated was the saturation temperature at the given pressure. For the various performance analysis calculations, the temperature was varied from saturation temperature in five-degree increments to obtain the results presented here. Given the amount of enthalpy input to the working fluid in the evaporator (boiler), the temperatures above saturation were achieved in the calculations by decreasing the value of the mass flow rate thus in effect achieving the desired superheat.
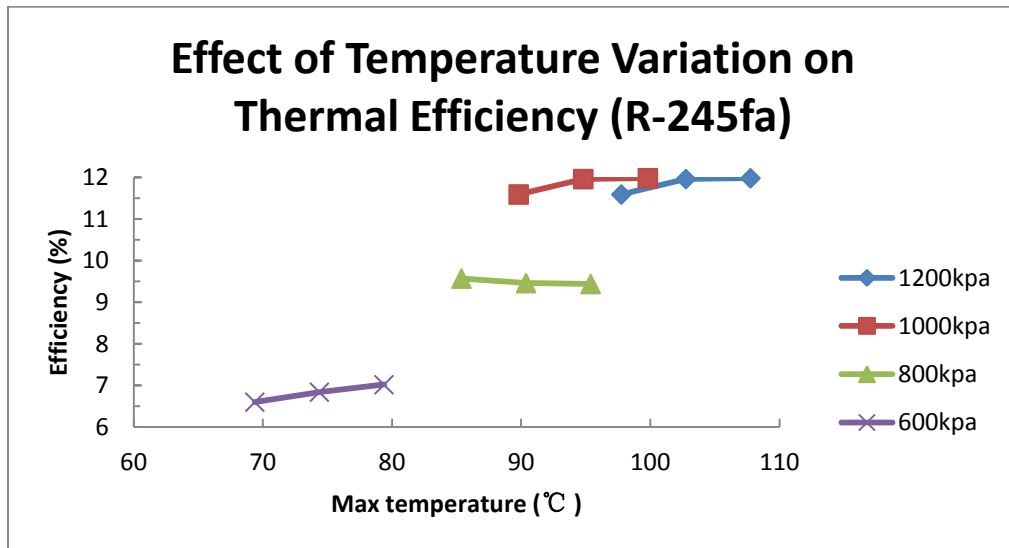

Figure 3.5. Temperature variation on thermal efficiency (R-245-fa)

The graph in Figure 3.5 shows the thermal efficiency variation for a 1 kg/s mass flow rate ORC using R-245fa as the working fluid. The thermal efficiency was calculated at three different temperatures for four different operating pressures. Overall it can be seen that the higher the boiler pressure (with the associated higher saturation temperature) leads to an overall higher thermal efficiency. Since the pressure is proportional to the temperature in the cycle it is expected that the lower pressure will have a lower temperature which would mean that the fluid doesn't absorb as much heat as it would at a higher pressure and would ultimately have a lower thermal efficiency. This claim is supported by Figure 3.5 shown above. It was also expected that increasing the turbine inlet temperature while keeping the boiler pressure constant would cause the thermal efficiency to slightly increase. This is also seen in the graph above. This is mainly due to the heat input being increased while the work output increases larger than the proportional value. It was expected that each pressure grouping would follow the same trends for the increases in maximum temperature but the analysis shows some differences.

For a pressure of 800 kPa there is an unusual nonlinear decrease in thermal efficiency with an increase in maximum temperature (possibly a consequence of property variations). With an increase of five degrees Celcius each time the thermal efficiency first decreases by 0.11% then decreases by about 0.02%. This shows that there is an optimum temperature and pressure that can be found for the operating condition.

The other three pressures, 1200 kPa, 1000 kPa and 600 kpa displayed expected trends of monotonic increase in thermal efficiency with a linear increase in maximum temperature. This calculation analysis was repeated for other working fluids; the results are shown and discussed below.
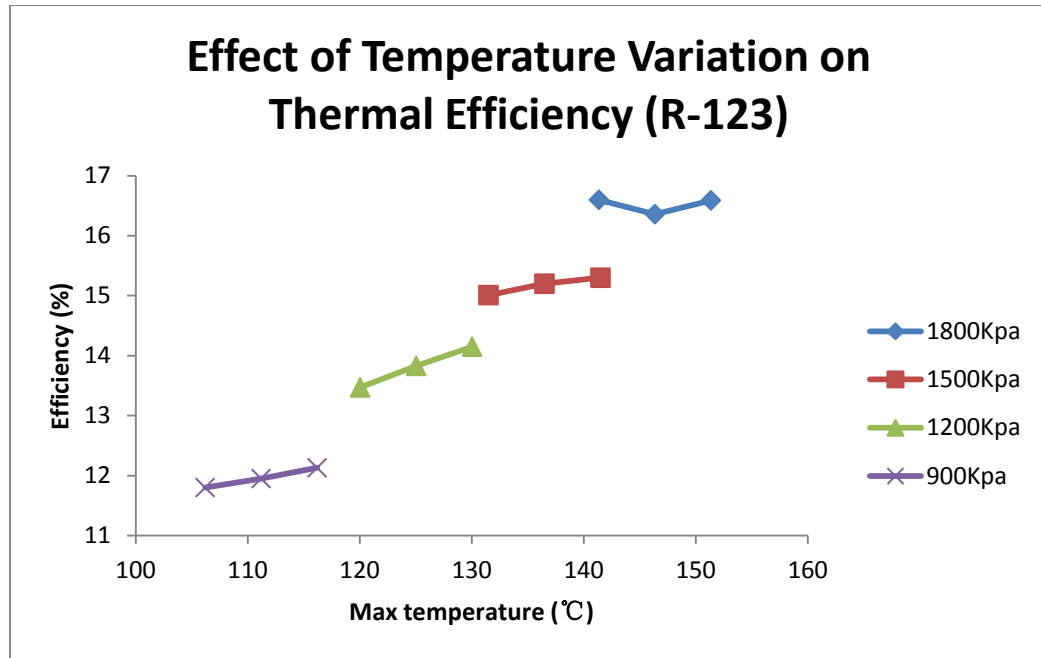
Figure 3.6. Temperature variation on thermal efficiency (R-123)

The data here shows that the fluid would be more efficient in higher temperature applications, such as a solar thermal ORC rather than just with use of low temperature waste heat. The trends shown in the above plot are consistent with those expected, and show a slightly increase in thermal efficiency with an increase in maximum temperature for each given pressure. The trend shown in Figure 3.6 above is the same as the trend shown with R-245fa at low working pressures. The maximum efficiency shown in Figure 3.6 is about 14% at pressure of 1200 kPa which is higher than the maximum efficiency seen when using R-245fa as a working fluid at the same pressure. Since the ORC in the previous study, using R-245fa as the working fluid had a maximum efficiency of about 12% it is assumed again that R-245fa could be utilized in a more efficient method if higher evaporator pressure is pumped.

The energy analysis was performed for another refrigerant, R-123. The results for each iteration and analysis of an ORC are shown below using R-123 as the working fluid within the cycle.
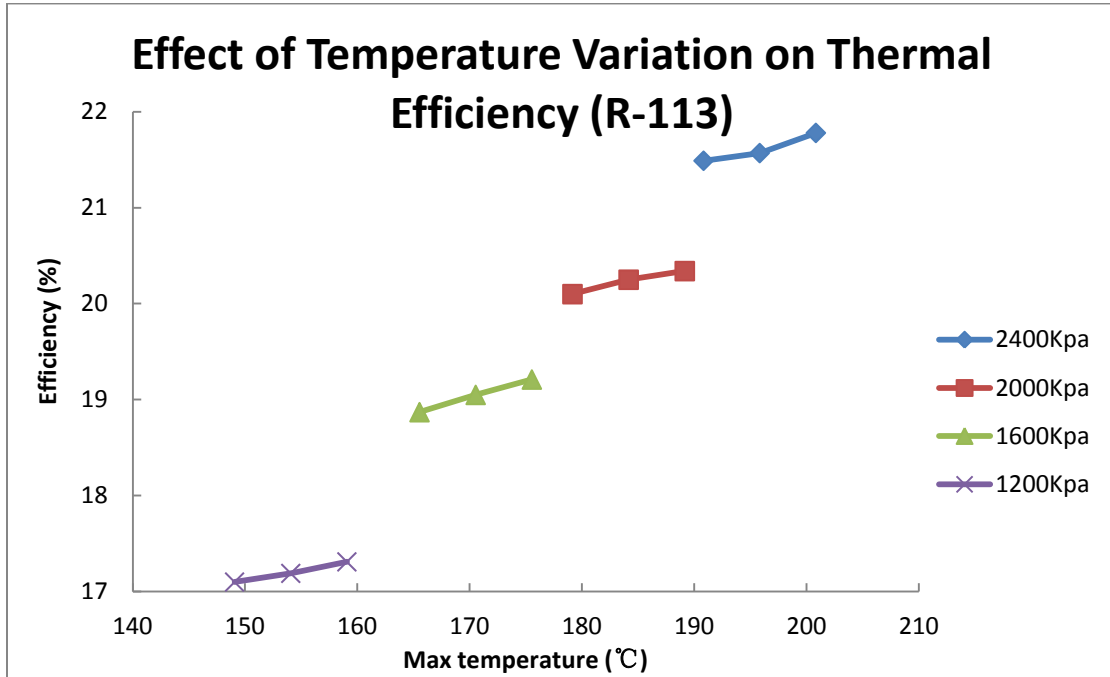
Figure 3.7. Temperature variation on thermal efficiency (R-113)

The trend displayed in Figure 3.7 shows that as the maximum temperature in the cycle is increased the thermal efficiency increases as well. Even though the graph appears to show the efficiency increasing the values of thermal efficiency only increase by about 0.1% with an increase of five degrees Celsius in the maximum temperature. Since the thermal efficiency remains relatively constant for each temperature setting this means that for the working fluid of R-113 the thermal efficiency of the cycle is a weak function of the maximum temperature in the cycle. In order to find the optimized fluid based on the efficiency of the operating conditions of the ORC, an efficiency-temperature chart while keeping the pressure constant is desirable for this analysis with a target temperature range

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expected to be assumed to calculate the ORC efficiency. Based the solar collector field conditions and heat transfer fluid properties, 100℃ to 200℃ is the range we may normally use. Since R-113 has the highest boiling point which means at same pressure level it will have the highest saturation temperature, the pressure is expected to be determined at the value corresponding to the 100℃ saturated temperature which is around 440 kPa.
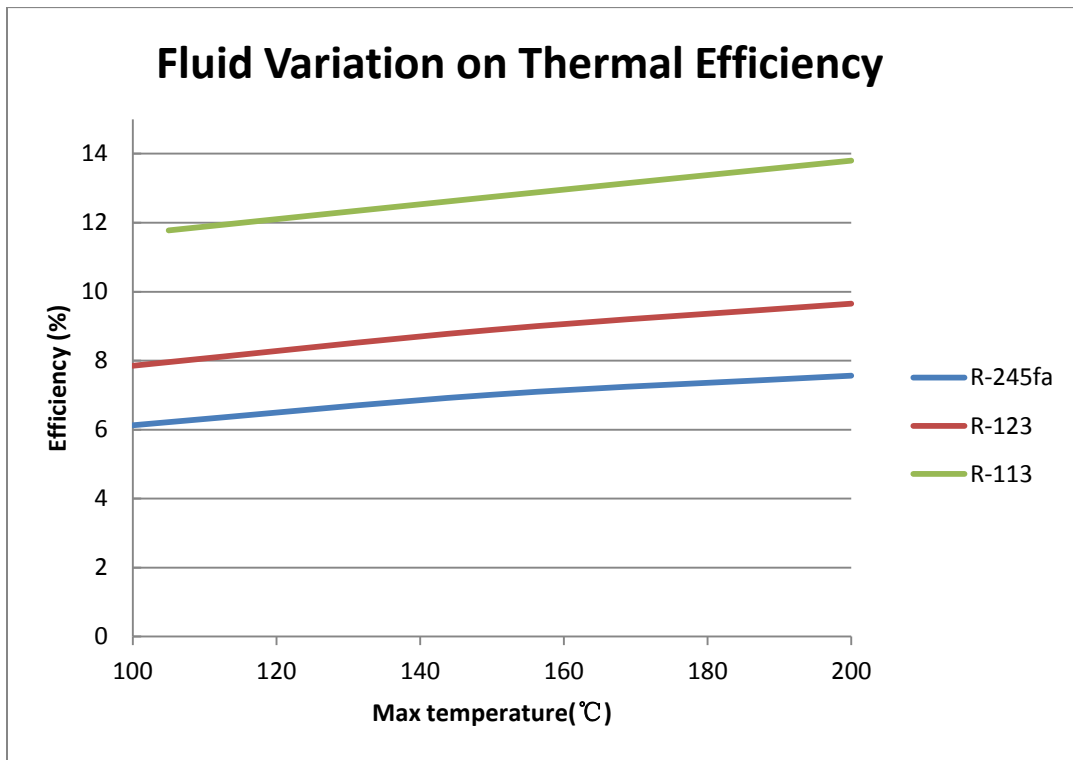
Figure 3.8. Variation of thermal efficiency for various working fluids at 440 kPa pumping pressure

Overall it can be seen that the operating conditions of the ORC's are mainly defined by the temperature and the pressure within the boiler. The thermal efficiency for the cycle appears to be very dependent on the working fluid rather than the operating conditions of the ORC. It was also found that the pressure variation exerts tremendous influence on

ORC efficiency whereas temperature variation has little impact on it. In addition to this, from Figure 3.8 it is clear that the R-113 has a more inclined slope for thermal efficiency than other two fluids as temperature goes up. Because at a given pressure, R-113's saturated temperature most approaches the maximum temperature which means less temperature difference between the saturated point and inlet of evaporator temperature. Consequently, a conclusion could be made that the optimum fluid should be chosen based on the maximum pressure of the ORC. At a certain temperature, lower saturated pressure fluid is expected to have better performance when the ORC is operated at lower maximum pressure, and vice versa. For example, Using R-113 as the working fluid is most efficient when the pressure or temperature is lowest and using R-245fa is the most efficient when the pressure or temperatures are very high.

Another factor that should be taken into account is the influence of the recuperator on the cycle isentropic efficiency especially for those which have higher evaporated temperature. When the fluid is heated to higher temperatures, the fluid at outlet of the turbine is also expected to be higher temperature correspondingly. Since the outlet temperature of recuperator is constant which there is a 5℃ difference between the condenser and recuperator for effective heat transfer. Larger temperature difference means that larger enthalpy is used for pre-heating the fluid before it goes into evaporator. The higher energy contributed to decreasing $\dot{Q}_{in}$ is the main reason that the cycle efficiency slightly increases as the maximum cycle temperature increases.

Another conclusion that can be easily made is thatthe cycle efficiency is expected to go down when the fluid is heated at higher temperature if the recuperator is not used at the outlet of turbine. This is mainly due to the heat input being increased while the work output

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remains constant which leads to a decrease in thermal efficiency. To minimize initial capital cost, recuperator is expected not be applied in the ORC when the maximum cycle temperature is relatively low (usually 20℃ and lower than the condensing temperature in this case). This particular issue needs further analysis from thermodynamic as well as economic pay back considerations.

CHAPTER 4

ECOMONIC ANALYSIS AND DISCUSSION

4.1 Device Selection

An ORC can use thermal energy from a wide range of sources as its heat input. Most commercially available ORC systems use either hot water or heated thermal oil as the transport fluid into the evaporator/boiler to transfer energy to the ORC working fluid (e.g. R-245fa). A liquid medium is preferred as the energy transport fluid into the evaporator since it will have larger energy densities and lower volumetric compared to gases. This means that the waste heat from the source may have to be run through a heat exchanger (with the associated less than unity effectiveness) containing either thermal oil or water before it enters the ORC. While there are many possible heat transfer fluids, one potential energy transport fluid is Duratherm 450 and makes good the heat transfer fluid due to the properties below:

- Optimal Temp. Range: 30°F up to 450°F (-1°C to 232°C)

- Duratherm 450 is specifically engineered for applications requiring process heating and cooling efficiently.

- Economical and thermally stable, Duratherm 450 heat transfer fluid offers an excellent alternative to costly synthetics and aromatic fluids while delivering precise and efficient operation down to 30°F.

Its temperature range best fits the cycle needs with the current targeted temperature range, from 40°C to 200°C. Duratherm 450 is designed to be used in heating and cooling process in engineering applications and the price of this fluid at $3.5/Liter is competitive with other candidates.

Storage of energy for the operation of ORC systems during non-sunny hours requires thermal energy storage and details pertaining to a thermal storage tank also need careful considerations.

Molten salt technology has been used thermal energy storage; and thermal energy collected by solar means, can be used to generate electricity in bad weather or at night. It was demonstrated in the Solar Two project from 1995-1999. Such storage systems have an annual return efficiency of 99%; i.e., the energy lost by the storage system is as little as one percent. The molten salt mixtures used in such systems vary with the most widely used systems containing mixtures of sodium nitrate, potassium nitrate and calcium nitrate. These salts are non-flammable and nontoxic, and have been used in the chemical and metals industries as a heat-transport fluid. Experience with such systems in non-solar applications is valuable. In one storage scheme, the Nitrate salt-mixture that melts at 131°C (268 F) is used. It is kept in the liquid state at 288 °C (550°F) in an insulated "cold" storage tank. The liquid salt is pumped to gain energy from solar collectors to 566°C (1,051 F) and saved in the hot storage tank. The storage tanks are well insulated and thermal energy can be usefully stored for up to a week. When electricity is needed, the hot salt is pumped to a conventional steam-generator to produce superheated steam or in the present application into the ORC evaporator for a turbine/generator as used in most conventional Rankine cycle power plants. A 100 MW_e turbine would need a tank of about 30 feet (9.1 m) tall and 80 feet (24 m) in diameter to drive it for four hours by this design. Several solar power plants in Spain use this thermal energy storage concept [20].

Unfortunately, this burgeoning technology is not ready for applications with ORC, since the salts used in the molten salt technology are too high in the temperature range (450°C

to 600°C). Instead, a well-insulated tank that stores sensible heat in the form of hot transport fluid (Dowrtherm to Silicon oil) can easily meet the current needs. The size of the tank is expected to be determined by the estimated time of operation, typically about three hours.

An example system is considered here. Assuming the mass flow rate of around 1 kg/s and with R-245fa as working fluid being heated to 200°C:

$$\dot{Q}_{in} = \dot{m} \times (h_4 - h_3) \tag{4.1}$$

Working fluid properties [21] are used here: density of Duratherm 450 at 220°C is 725 kg/m^3, heat capacity is 2.68 kJ/kg-K, and a 10°C temperature decrease for effective heat exchange. Based on the above data, a tank with volume of $27.3 m^3$ is expected to be sufficient to supply energy for up to 3 hours when there is no solar power input.

Turbine or expander in ORC is normally the most costly device. As a result, the selection of an application specific turbine is an important task.

In this section, issues relevant to selection of different types of turbines or expanders are discussed. The ORC turbine is based on the preferred power range, ORC turbine speed, enthalpy, and degree of superheat or quality of inlet fluid into turbine, lubrication and sealing type. In Figure 4.1 it is shown how the turbine can be chosen based on power range and rotational speed. In this monogram, lines indicate that expanders could have higher operational speeds for larger powers. Expanders with 10 kW output and with less than 50 revolutions per minute rotational speed are possible. In contrast for turbines, less rotational speed also means less power for a given torque generated based on fluid conditions. In low power ranges, screw expanders with low rotational speeds potentially

using wet fluid can be used. This can also be the case for compressors that could be lubricated by their own fluids.
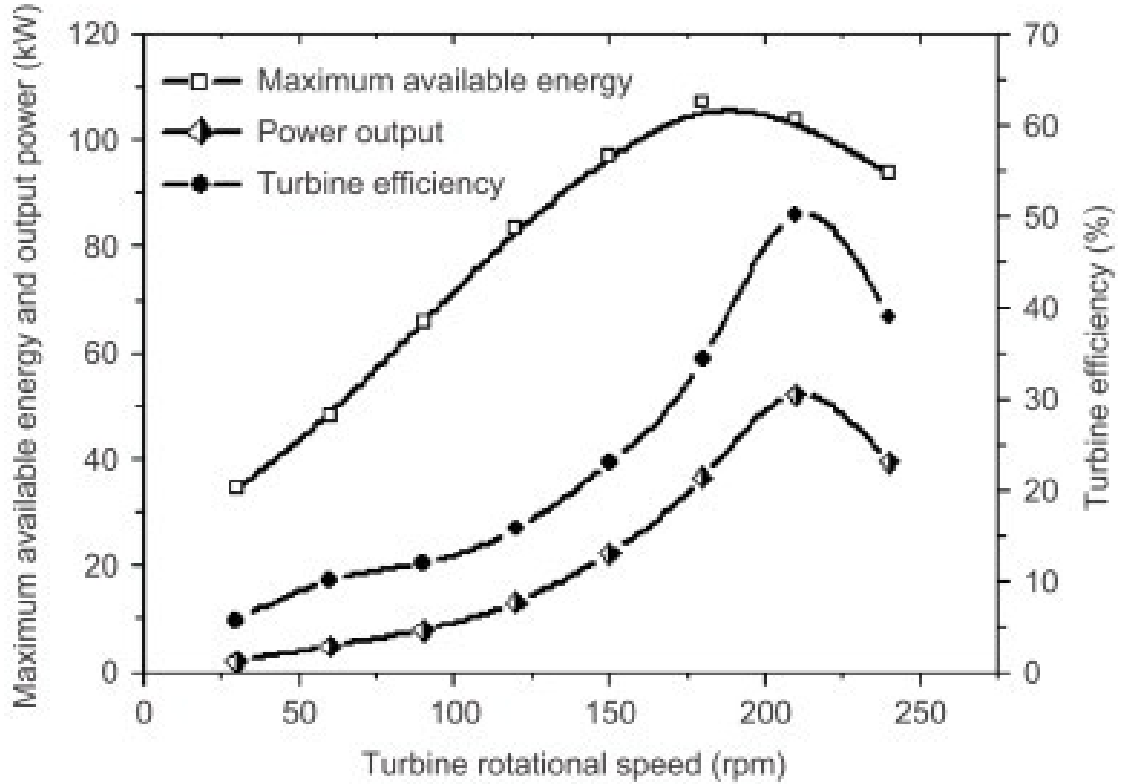
Figure 4.1. Effect on Turbine efficiency of variation of Turbine rotational speed [22]

From Figure 4.1, it is clear that the turbine efficiency reaches the peak when rotational speed is around 200 rpm with maximum available energy and output power at 110 kW. After evaluating commercially available turbines in the market, Infinity Turbine® ORC Power LLC-20 is chosen for further analysis based on its nominal rotational speed 150~250rpm and available power output 50~300kw.

Turbo-expanders extract energy from a fluid and they do this by converting the enthalpy / energy of the high pressure, high temperature fluid into kinetic energy of the impeller and shaft by reducing the cross sectional flow area (nozzle section). The nozzles need to be

arranged around the entire perimeter of the turbine such that each turbine vane is constantly being impacted upon by fluid. The turbine vane depth should equal the nozzle section depth and should be maximized so that any gaps (fluid leakage paths) between the housing and the turbine are a small percentage of the overall depth. In some cases, the nozzles can be reduced in size if there is restricted space, such as when the turbine is small.

Figure 4.2. Inflow Radial Turbine manufactured by Infinity Turbine®

The evaporator / heat exchanger is used as boiler in the ORC in which the energy transfer from the transport fluid to the working fluid occurs.

The working fluid (e.g. R-245fa) at the outlet of the recuperator is expected to be air-cooled since the present ORC systems are expected to be operated in locations with water shortage. Air cooled condensers are of two types: natural convection and forced convection. In the natural convection type, the air flows over it in natural a way depending upon the temperature of the condenser coil. In the forced air type, a fan operated by a motor blows air over the condenser coil. Although natural convection is a

more economical choice, force convection is preferred since they are more effective particularly at the smaller sizes of interest here. Anticipated costs for various ORC components are presented in Table 4.1.

Component	Description	Industry Supplier	Approximate Cost data	Quantity
1. Collectors	14 m^2 reflection concentrating collectors	Alanod	US $ 1000	1
2. Working fluids	HFC-245fa	Dupont	US $ 33 /kg	10
3.WF Pumps	Plunger	Hypro, Various	US $ 300 /unit	1
4.Heat Transfer Fluids	Duratherm 450	Various	US $ 3.5 /Liter	20
5.HTF pumps	V-20 power steering	Vickers	US $ 290 /unit	1
6.fluid motors	24VDC	Leeson, Various	US $ 300 /unit	1
7.Turbines	Inflow Radial Turbine	Infinity Turbine®	US $ 6000/unit	1
8.Heat exchangers	Brazed plate BP-410-30	ITT BRAZEPAK, various	US $ 1500 /unit	1
9.Air condenser	Forced convection finned copper tube	Carrier, Various	US $ 4.5 /kWh rejection	10
10. Sensor based control system	Microcontroller and feedback loops	MIT/STG	US $ 800	1
Total cost				US $ 11,000

Table 4.1. Costs of Various ORC components

4.2 Discussion

California is being considered as the general location where the solar ORC is applied to compete with the higher electrical utility rates there. As it is well known, Southern California is well suited for solar applications due to the longer sunshine hours available in the year.

It was noted that regenerator usage is applicable for superheated working fluid at turbine entrance and it will help maximize the performance. However, even such cases, better efficiency of the ORC may not compensate for the cost of regenerator.

Higher evaporator pressure gives better efficiencies and there could be a small range of temperature versus heat exchangers performance to optimize the ORC cost.

City	Sunny Days	Partly Sunny Days	Total Days with Sun
Los Angeles	186	106	292
Sacramento	188	77	265
San Diego	146	117	263

Table 4.2 Annual days of sunshine in CA cities [23]

City	Solar Radiation
Los Angeles	4.8
Sacramento	4.3
San Diego	5.3

Table 4.3. Direct Beam Solar Radiation for Concentrating
collectors ($kWh/m^2/day$), Uncertainty $\pm$ 8% [24]

Some relevant NREL data is shown in Tables 4.2 and 4.3. The data indicates that an estimated 1,400 kWh/m²/year energy is received from the sun in Los Angeles. For a 110 kW system, at a nominal 1,000 W/m², 110 m² solar collection areas will be needed. For 110 m² collection area, in Los Angeles, approximately 154,000 kWh energy equivalents will be collected. Assuming the heat exchanger efficiency is 85% at the solar collector field, the system total efficiency would be:

$$\eta_{total} = \eta_{heat\,transfer} \times \eta_{ORC} \qquad (4.1)$$

For such an ORC system in CA, 15,700 kWh electrical energy is expected to be generated based on 12% ORC efficiency.

The estimated cost per ORC unit as shown in Table 4.1 is around $110,000, and it could have $2,000 additional operation and maintenance costs. The $110,000 investment depreciated over 20 years OR per year cost is $5,500 and a total of $7,500 including operation and maintenance. Thus the cost per 1 kWh = $ 0.47 per kWh. That is high but realistic and can be better if the efficiency is better than 12%.

For smaller sizes power units (10~ 50 kW), a scroll-type expander would be used as turbine for electric power generation. Smaller size turbines are much less expensive, however, the ORC system efficiency is comparatively low due to the low power input to heat the working fluid. A thermal efficiency of 3.1% was achieved, and a clear path forward has been presented to achieve 9.7% efficiency. This experiment validates the use of Hydrofluoroether- 7000 (HFE- 7000) as a working fluid in a Rankine cycle – both in a custom application or as a drop-in replacement for less-desirable fluids. Such a system

has potential uses that include bottoming cycles for industrial processes and inexpensive non-concentrating solar-thermal power plants [25].
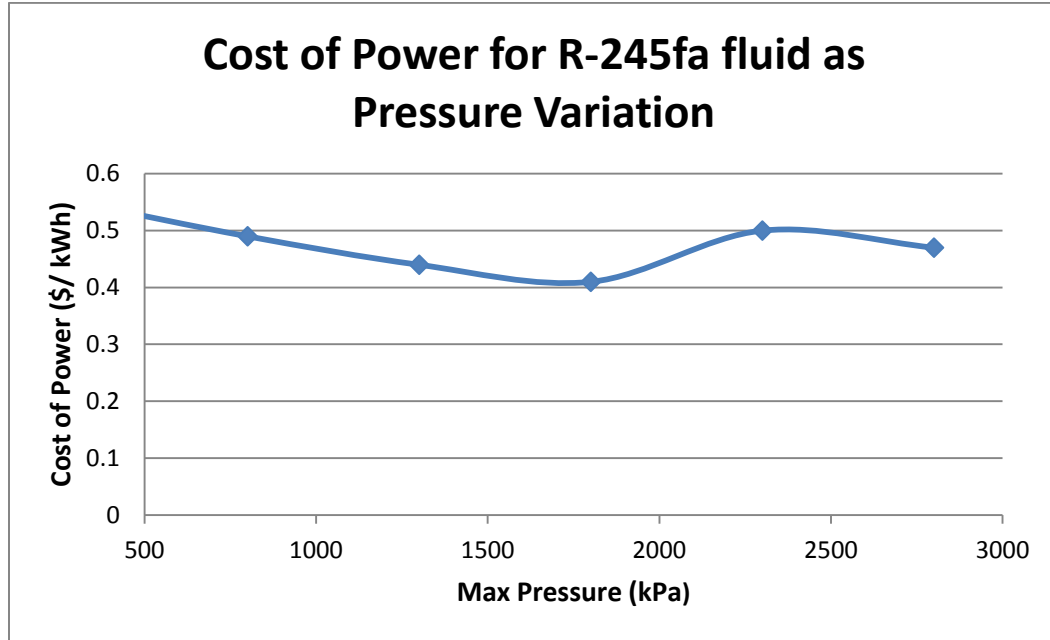

Figure 4.3. Cost of Power for ORC using R-245fa fluid as Pressure Variation

The ORC becomes most cost effective when the maximum cycle pressure is around 1,800 kPa (as in Figure 4.3). For operations at higher pressures, the turbine and pump need to be changed as the fluid needs to be pumped to a pressure which is higher than the maximum applicable pressure of turbine and pump. Higher-pressure systems will involve higher costs with the system efficiency increasing as the pressure increases. The ORC system is expected to have more cost effectiveness when it is operated at optimized higher pressures.

Another factor that affects the efficiency and cost effectiveness is the use or lack of a recuperator. As the data listed in Table 4.3 indicates, the recuperator (heat exchanger) for preheating the fluid before it enters evaporator is around $26,000 per unit which is

approximately one quarter of the total system price. As the analysis in chapter 3 shows, a larger temperature difference between condensing temperature and evaporating temperature would contribute to larger energy input during the preheating of the fluid. Thus a recuperator is not needed when the working fluid is evaporated to a relatively low temperature.
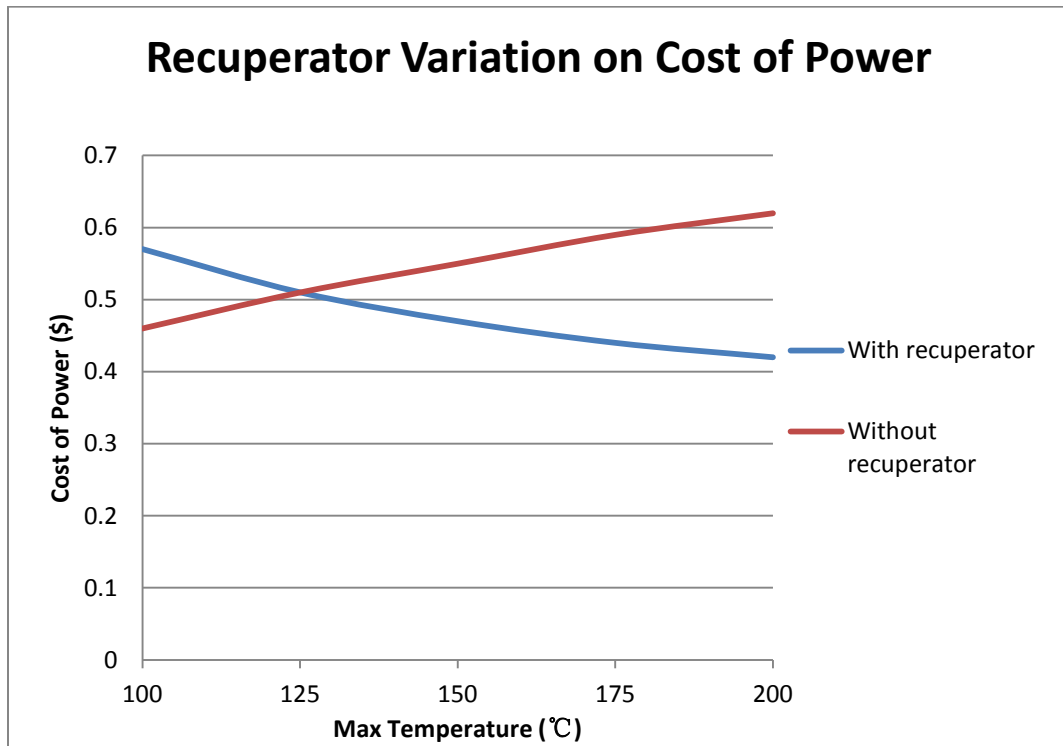

Figure 4.4. Component Variation on Cost of Power when using R-245fa fluid

From Figure 4.4, an optimum for the maximum cycle temperature can be seen as around 126 °C. Consequently, when the evaporating temperature is below 126 °C, a recuperator is required to be connected between outlet of turbine and inlet of evaporator to maximize the cost effectiveness.

Based on the performed investigation and market evaluation, ORC systems could be used for solar applications cost effectively though not with grid-parity. ORC systems can also be used as bottoming cycles of other low temperature waste heat recovery units cost effectively.

Solar based ORC could be implemented effectively for off-grid applications quite often more cost effectively than distributed PV power generation systems with no subsidies. Solar ORC systems can incorporate thermal energy storage enabling power generation when there is no sun. ORC system use as in combined heat and power (CHP) or Tri-generation units are options that can further improve total efficiency and reduce the cost. For each ORC application, power and temperature range, best working fluid need to be chosen carefully.

The cost of electrical power generation using ORC is higher than the typical normal grid power cost which is around 3~6 cents per kWh. Solar application of ORC can be optimized based on the distributed off-grid applications using the procedures used in this work and the result can have a great impact on performance improvement and cost reduction. Despite the higher cost of this distributed power generation using ORC, there could be many off-grid and waste heat recovery applications that could make ORC systems attractive.

CHAPTER 5

CONCLUSION

The present work focuses on power generation applications based on small scale Organic Rankine Cycle (ORC) that can be very useful for distributed power generation for solar, off-grid and waste heat recovery. The present work considered applicable cycle, fluid selection, efficiency analysis based for independent parameters, device design and economic analysis. The conclusions of this study can be summarized as follows:

1. The choice of working fluids is essentially based on the temperature in the condenser since energy rejection is a vital part of the ORC systems. In the current work condensing temperature condensing temperature (40°C) and applicable boiling temperature (up to 200°C) are assumed.

2. The thermal efficiency for the ORC appears to be dependent on the working fluid properties in relation to the maximum temperature and pressure of the ORC and to a smaller extant on the operating conditions of the ORC.

3. The ORC maximum pressure or the saturation pressure corresponding to the chosen maximum temperature influences the cycle efficiency.

4. It was found that R-245fa is a more efficient working fluid at lower temperatures than R123 thus may be better suited for waste heat recovery and tri-generation. It was also found that R-113 was more efficient at higher power systems because it had efficiency over 10% when the cycle was pushed to operate at higher pressure

conditions. Though operation with different fluids is different, most of the fluids are more efficient at higher pressures since they are able to produce more power at a higher pressure.

5. The ORC cycle was also analyzed including some less than ideal isentropic efficiency for the pump and turbine. This showed that the overall ORC thermal efficiency was linearly proportional to the isentropic efficiency of the turbine.

6. At a certain temperature, lower saturated pressure fluid is expected to have better performance when the ORC is operated at lower maximum pressure and vice versa.

7. For small size ORC and low cost ORC cycles, use of a recuperator is expected not to be cost effective in the ORC when the maximum cycle temperature is relatively low.

8. The work presented is a rudimentary analysis of ORC to evaluate the differences of three working fluids chosen, namely R-245fa, R-113 and R-123. Additional work particularly related to long term viability and operation and maintenance of ORC systems is need to take better advantage of ORC systems.

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Vita

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