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# Performance Analysis of ORC Using Effluent from Power Plant

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# Abstract

우리의 생활에 있어서 에너지는 필수적인 요소가 되었지만, 전 세계 인구의 20%, 즉, 아프리카에서 약 9.2%, 인도에서 약 7.36%, 그리고 다른 개발도상국들은 전기 없이 살아가고 있다. 현존하는 종래의 발전 방식은 환경에 다양한 결과를 초래하는 화석연료나 방사능 물질을 연료로 사용하고 있다. 한편, 많은 산업들이 고온의 폐열을 부산물로써 환경에 방출한다. 이러한 폐열은 인근 부지에 설치된 유기랭킨사이클 플랜트의 열교환기로 유입됨으로써 전력을 생산하는데 재이용 될 수 있다. 폐열 회수 유기 랭킨 사이클(WHR)은 친환경적이며, 종래의 증기 랭킨 사이클과 물 대신 유기 유체를 사용한다는 점과 저온과 중온의 온도 범위 사이에 있는60-200°C인 열원을 사용한다는 점을 제외하고는 유사하다. 본 연구의 phase 1에서, 남해 발전소의 폐수와 표층수는 탄화수소계(HCs), 수소화부화탄소계(HFCs) 그리고 HCs, HFCs혼합물인 R245fa를 사용하는 터빈전력생산량 20kW급 사이클의 각각 Heating과 Cooling 열원으로 사용되었다. Phase 1에서 가장 높은 효율을 보인 유체는 Phase 2에서의 폐열회수 유기랭킨 사이클을 모의 실험하고 분석하는데 사용되었다. 또한, 연간의 남해 발전소로부터의 폐수는 heating 매체로 쓰였고, 표층수는 cooling 매체로 1MW 터빈발전량 생산을 위해 사용되었다. PHASE 1 과 2는 터빈전력생산량 20kW급 시스템의 효율, Net Power, 질량유량과 경제성 평가, APRE(터빈출력 대비 증발열량의 비), APRC(터빈출력 대비 응축 열량의 비), 그리고 TTP(펌프일량 대비 터빈전력생산량)과 같은 성능분석 지표들을 통해 분석되었다. Aspen HYSYS는 모델링 및 시뮬레이션 프로그램으로 이용되었다. 결과는, 20kW급 터빈전력생산량의 경우, 고농도 R600a와 R245fa 혼합냉매를 적용한 사이클은 질량 유량비를 R245fa를 작동유체로 사용했을 때 대비, 55%가량 감소시킬 수 있다. 또한, HFCs는 제법 우수하게 작동하였으나, APRE, APRC값이 나타나듯, 높은 펌프

운전비용을 야기하는 큰 질량유량비와 압력을 필요로 했다. 게다가, 1MW 급 터빈 발전량을 얻기 위한 월별 온배수를 볼 때, 겨울철에 가장 높은 시스템 효율인 10.45%를 보였고 이는 가장 효율이 낮았던 가을철보다 28% 높은 수치였다. 또한, 사이클 효율은 TTP와 직접적인 관계가 있고, 질량 유량비와는 반비례했으며, 계산된 TTP의 절반의 수치를 보였다. 비용을 절감하고 폐열회수 유기랭킨 사이클을 최적화 하기 위해서는 적용된 작동유체의 끓는점에 강조를 두며 Net power와 APRE, APRC 그리고 적절한 시스템 효율 등이 균형을 맞추는 최저 허용범위의 압력을 선정하는 것이 이상적이다.

# CHAPTER 1. INTRODUCTION

## 1.1 Background

Even though energy has become a basic necessity in life, it comes as a surprise that about 20% of the world's population is without electricity with about 9.2% in Africa, over 7.36% in India and other developing countries. On the contrary, existing conventional power plants use fossil fuels or radioactive materials as fuel source. Derived consequences include soil erosion, dust, noise, water and toxic pollution, and impacts on local biodiversity. These coupled with increasing demand for energy, more stringent environmental legislation and binding targets for safe and sustainable energy, such as the Montreal Protocol, Kyoto Protocol and European '20-20-20' Climate and Energy package necessitate managing our environment to conserve biodiversity. This can be done by developing and deploying innovative technologies to efficiently offset current fossil fuels dependency while tackling greenhouse gas emissions and assuaging environmental pollution.

A waste heat recovery (WHR) system is one of such inventions and a perfect example of the energy 'trilemma' of balancing the demand for energy security, affordability and low carbon emission. WHR technology is a new capable renewable energy power producing technology from a dedicated heat source using the principle of the conventional steam Rankine cycle (RC). It involves capturing and recycling waste heat from industrial processes to generate electricity, preheat combustion air, absorption cooling process, and space heating. Unfortunately, most of the waste heat from industries are at moderate to low grade temperatures making the steam Rankine cycle economically unviable [1]. Heat is considered to be moderate to low grade when it is less than 370 °C[2].

Waste heat, together with the right working fluid selection can help generate electricity by employing the organic Rankine cycle (ORC). ORC is a promising substitute to steam Rankine cycle for power generation at low to medium heat grades and there have been many studies on it applications[3]. Also, ORC plants are more recognized in the energy generating field and commercially operating with Arizona Public Service Company in the United States building the first new ORC power plant that combines solar technology [4] is shown in Fig. 1.1.

Fig. 1.1 Organic power plant in Saguaro, Arizona [4]

The performance of the ORC plant components, the temperatures of the heat source and sink and the thermophysical characteristics of the selected working fluids are influential to the total efficiency of the Rankine cycle.

More also, using ORC rather than the steam Rankine cycle at low temperature heat source reduces the volume ratio of the working fluid at the turbine inlet and outlet thus avoiding the use of more expensive and complicated turbines thereby making the ORC more cost effective[5].

Employing waste heat recovery (WHR) technology as heat source for an ORC plant can make renewable power generation almost accessible on every continent.

## 1.2 Objectives and methodology

Waste heat from industries are dissipated into the atmosphere increasing fuel consumption due to large amount of flue gas produced which causes an increase in equipment size while upsurging global warming. However, this waste heat can be recovered to generate power to offset electricity consumption on site (especially the cost of high amount of fuel consumed to produce the waste heat) or exported to a local utility if in excess to power a small community.

This study is in two phases. Phase 1 uses the average power plant effluent (waste heat) temperature from a yearly data of effluents for Namhae power plant as heat source and surface seawater as cooling source to generate 20kW gross turbine power for a WHR-ORC employing homologous series of hydrocarbons (HCs); n-pentane, n-hexane, iso-butane, propane (R290) hydrofluorocarbons (HFCs); R245fa, R152a, a mixture of hydrocarbons and another mixture of R245fa and the highest efficiency hydrocarbon. The working fluid with the highest cycle efficiency from phase 1 will be used to re-simulate the WHR-ORC with the yearly effluent data from Namhae power plant and surface seawater as heat and cooling source, respectively, to generate 1MW gross turbine power.

The Aspen HYSYS software package is deployed as the simulation tool for analyzing the cycle performance through performance indicators like cycle efficiency, net power, mass flow rates, and economic analyzers, APRE (ratio of evaporator capacity to gross turbine work), APRC (ratio of condenser capacity to gross turbine work) and TTP (ratio of gross turbine work to pump work).

### 1.3 Waste heat recovery technologies

Waste heat is heat contained in flue gases from industrial actions like fuel combustion and chemical reactions and discarded into the atmosphere at higher temperatures [6]. This heat thrown into the atmosphere can be recycled on the same premises then used on the premises or transported for use elsewhere. In most industrial countries including South Korea, almost 40-50% makes up waste heat with only 15% being recovered and 10% of the total energy consumption going to steel and iron industries. Fig. 1.2 and 1.3 shows the proximity of a typical industrial complex in South Korea to the ocean which makes it a good candidate for WHR-ORC power generation.

Fig. 1.2a View of Namhae Power Plant [7]

In the context of this paper, the heat is transferred to a heat exchanger which is part of an ORC plant to generate power to counterbalance electricity consumption on site. Waste heat can be applied to a variety of medium to low range temperature heat streams.

Fig. 1.2b Geographical location of Namhae Chemical Corporation [7]

Fig. 1.3a View of Yeosu National Industrial Complex [8]

Fig. 1.2b Proximity of Yeosu National Industrial Complex to the Ocean [8]

Recovering waste heat can aid in reducing fossil fuel consumption, associated operating cost and pollutants emission. The amount of heat available is not vital compared to the scheme used in the heat recovery process which is also dependent on the waste heat gas temperature as well as the economics involved. Although, energy lost in waste gases cannot always be fully recovered, much of the heat could be recovered and losses minimized by considering the quantity, quality, composition, minimum allowable temperature and operating schedules, availability and other logistics of the waste heat source and the stream to which the heat will be transferred [1,9]. The aforementioned parameters will help determine the viability of the waste heat recovery and provide understanding into material selection and design configuration and boundaries. The parameters relevant to this paper are briefly discussed below.

### 1.3.1 Heat availability

This describes the nature of the energy source. The higher the quality of energy source, the more available is its energy for use [9]. The heat available can be calculated from;

$$Q = V \times \rho \times Cp \times \Delta T \qquad 1.1$$

Where

$Q$ = heat content (kJ)

$V$ = flow rate of the substance ($m^3$/hr)

$\rho$ = density of the flue gas (kg/$m^{3)}$

$Cp$ = specific heat of the substance (kJ/kg $^oC$)

$\Delta T$ = temperature difference ($^oC$)

### 1.3.2 Heat quantity

The heat content or quantity, measures the amount of energy that can be recovered from a heat stream and how it can be used. Heat quantity is a function of mass flow rate and temperature of the stream [1]. It can be represented as:

$$E = mh(t) \qquad 1.2$$

Where

$E$ =Waste heat loss (W)

$m$ =Waste stream mass flow rate (kg/s)

$h$ = Waste stream specific enthalpy (J/kg)as a function of temperature.

### 1.3.3 Waste heat quality/temperature

The waste heat quality, as determined by its temperature is the most important factor that can define the viability of WHR. Quality is a measure of the usefulness of the waste heat and its capability to cause change while temperature is the measure of a heat source's quality and availability. The higher the heat source temperature, the higher the quality and more cost effective is the heat recovery and as such, the degree of the temperature difference between the heat source and sink can determine the quality of the waste heat. This difference in temperature can help in the material selection and design of heat exchangers [1].

It is essential for cooling source temperature (sink) to be lower than the waste heat temperature (source) to facilitate heat recovery and transfer. Also, the degree of the temperature difference between the heat source and cooling source is a key factor for determining the quality of the heat.

### 1.3.4 Classification and application

Waste heat recovery opportunities are classified in this paper according to Table 1.1. It is worthy to mention that this study will only consider low grade waste heat source since it accounts for more than 50% of the total heat generated in industries, which makes its potential exceed other heat sources even though it has a lower quality. In addition, low to medium grade heat sources are more compatible with heat exchanger materials and practicable with ORC plants for power generation [10].

Table 1.1 Modified temperature classifications of waste heat sources [1]

| Temp. Range (°C) | Example Heat Sources | Temp. (°C) | Advantages | Disadvantages | Typical Recovery Technologies |
|---|---|---|---|---|---|
| Low [< 230] | Cooling water from; air conditioning and refrigeration | 30-40 | Numerous steam products contain large quantities of low temperature heat. | Impractical low-temperature heat recovery for combustion exhaust due to heat exchanger corrosion.<br><br>Low efficiency power generation. | ORC cycle<br><br>Upgrading via a heat pump to increase temperature for end use. Domestic water heating. |
| | Condensers | 30-50 | | | |
| | furnace doors air compressors | 30-50 | | | |
| | Internal combustion engines | 70-120 | | | |
| Medium [230-650] | Steam boiler exhaust | 230-480 | More compatible with heat exchanger materials.<br><br>Practical for power generation. | | ORC for power generation.<br><br>Transfer to low-temperature process. |
| | Drying and baking ovens | 230-590 | | | |
| | Reciprocating engine exhaust | 320-590 | | | |
| | Gas turbine exhaust | 370-540 | | | |
| High [>650] | Hydrogen plants | 650-980 | High heat transfer rate per unit area.<br><br>High quality energy, available | High temperature creates increased thermal stress on heat exchange materials. | Transfer to medium-low temperature processes.<br><br>Steam generation for process heating |
| | Copper refining furnace | 760-820 | | | |
| | Fume incinerators | 650-1430 | | | |
| | Coke oven | 650-1000 | | | |

### 1.3.5 Waste heat utilization

Waste heat utilization technologies usually involve the direct handling of the waste heat 'just as it is' so there is no need for specialized equipment apart from piping and additional ductwork. Heat exchangers have the widest range of applications in different designs, sizes and capacities limiting their applications to larger plants and complex processes.

Harmonizing the available heat, the accessible quantity and quality, distance between source and demand (a shorter distance may present a more practicable system), the form and condition of waste heat source (especially in instances were heat exchangers will be deployed) as well as the recovery applications can lead to adequate exploitation of the waste heat.

This paper will concentrate on the heat exchanger technology since they have widespread usage and established technologies.

# CHAPTER 2. ORC TECHNOLOGY

## 2.1 Organic Rankine cycle (ORC)

ORC is a continuous, closed-loop in which a working fluid exchanges heat with an heat source in an evaporator to vaporize, expands in a turbine coupled to a generator to produce power, rejects heat to condense in the condenser and pumped back to the evaporator by a circulating or feed pump to repeat the process as depicted in Fig. 2.1.

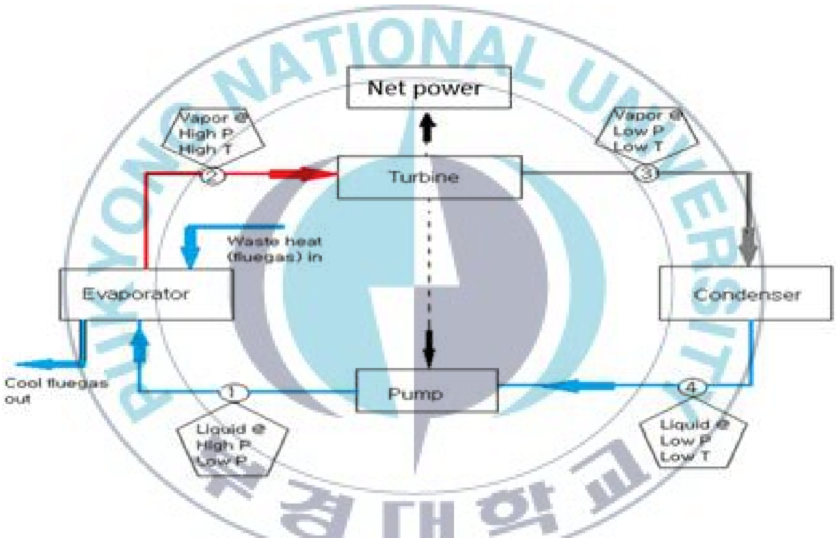


Fig. 2.1 The organic Rankine cycle [11]

It is a promising alternative to steam Rankine cycle for the generation of power utilizing low-grade heat sources within a temperature range of 60-200 °C and seeks to address the problem of steam Rankine cycles by using organic fluids instead of water. Organic fluids are suitable for the ORC because their specific vaporization heat and boiling point temperature are much lower than that of water [12-14]. Other advantages of the ORC over the conventional steam Rankine cycle include, it requires fewer expansion stages with the expander (screw expander) operating at low peripheral speed, usually requires no superheat, has gear free transmission which results in long operating life, less maintenance, fewer parts and are self-running thus constant supervision is not necessary [15,16].

Further, the organic Rankine cycle employs biomass combustion heat, industrial waste heat and renewable energies like geothermal, solar ponds and so on as heat source. Its name, ORC is from the working fluids employed in its operation which are usually "organic" but can employ any working fluid having temperatures within the operating allowable temperature region. Thus its name is exclusively an advertising notion [17].

## 2.2 Literature review on previous ORC studies

There have been many researches on proposed designs aimed at optimizing the ORC. Most of them, however, are mainly reflected in the selection of working fluids, analysis of thermal performance, optimization of system, and transcritical cycles. An Organic Rankine Cycle with Ejector (EORC) was proposed to increase power output capacity and efficiency and compared with Double Organic Rankine Cycle (DORC) which was also introduced in order to analyze and compare the EORC with the ORC [18]. Also, Hettiarachchi et al [19] proposed a cost-effective optimum design criterion for organic Rankine power cycles utilizing low-temperature geothermal heat sources. They optimized the low-temperature geothermal organic Rankine power cycle using the steepest descent method while setting the ratio of the total heat exchanger area to net power output as objective function while varying the geothermal and cooling water velocities as well as the evaporation and condensation temperatures. From their study, they concluded that, the choice of working fluid can greatly affect the power plant cost, with a difference of more than twice in some instances.

Furthermore, a supercritical ORC was further analyzed to improve the efficiency of the system of about 8% [20]. T.C. Hung et al. [21] presented that a suitable selection of the working fluids is a critical factor for achieving an efficient and a safe operation in an organic Rankine cycle (ORC). They emphasized that each working fluid has its own range of applicability according to its thermo-physical properties under the considerations of a high efficiency and a safe operation and listed toxicity, chemical stability, boiling temperature, flash point specific heat, latent heat and thermal conductivity as important factors of the working fluids needed to be considered.

Still, a theoretical thermal efficiency of an ideal ORC was proposed to analyze the effect of refrigerant properties for various heat source temperatures. The parameters for analyses were the thermal efficiency, optimal operation condition and exergy destruction. It was determined that the various refrigerants have little or no impact on the optimal operation of the proposed cycle with the desirable characteristics of the refrigerant being low critical temperature, specific heat and high vaporization latent heat [22].

## 2.3 Working fluid selection

The selection and choice of working fluid is very important in achieving a higher ORC efficiency so most of the research works on ORC are centered on selection of optimal working fluids based on their environmental impact (GWP, ODP), flammability, toxicity and other thermodynamic properties and working conditions while some few others are focused on optimizing the ORC through design and configuration. With low grade heat being the heat source for this study, the choice of working fluid becomes significantly crucial since only few fluids are workable at low temperature and the likely occurrence of irreversibilities in the heat exchangers which can be disadvantageous to the overall system performance of the ORC plant.

### 2.3.1 Factors to consider in working fluid selection

The selection of working fluid depends on many factors some of which include, the accessible heat source temperature, thermodynamic properties, stability and compatibility of fluid with materials in contact and lubricating oil, environmental safety standards, technical feasibility, availability of working fluid and cost. For this study, only a few criteria for working fluid selection relevant to this work are discussed in no special order.

**Thermodynamic properties:** This may be the key property to consider as it has an impact on the environment, plant operation and the system efficiency. The thermodynamic properties consist of a number of co-dependent working fluid properties which include specific heat, heat of vaporization, density, critical point, etc. Nonetheless, when compared, the efficiency and/or desired power output of the selected fluids must be as high as possible for set heat source and sink when the cycle is simulated with a thermodynamic model [23].

**Saturation vapors curves:** This can be clarified using T-s diagrams, as depicted in Fig. 2.2a-c. Wet fluids like water, show a negative slope, isentropic fluids, like R134a, vertical and dry fluids like R600a, positive slope. Unlike wet fluids, isentropic and dry working fluids are most suitable for the ORC as there are no droplets at the end of the expansion to damage the turbine and also show a better performance. However, too positive slope implies an additional load for the condenser since the vapor will leave the turbine with substantial superheat. Thus, a regenerator or recuperator is needed in such instances to preheat the fluid with the substantial superheat exiting the expander before entering the evaporator. On the other hand, isentropic fluids show increased efficiency among the vapor curves without additional investment cost on recuperators [23] and this is the preferred vapor curve since this study focuses on the recovery of low grade heat.

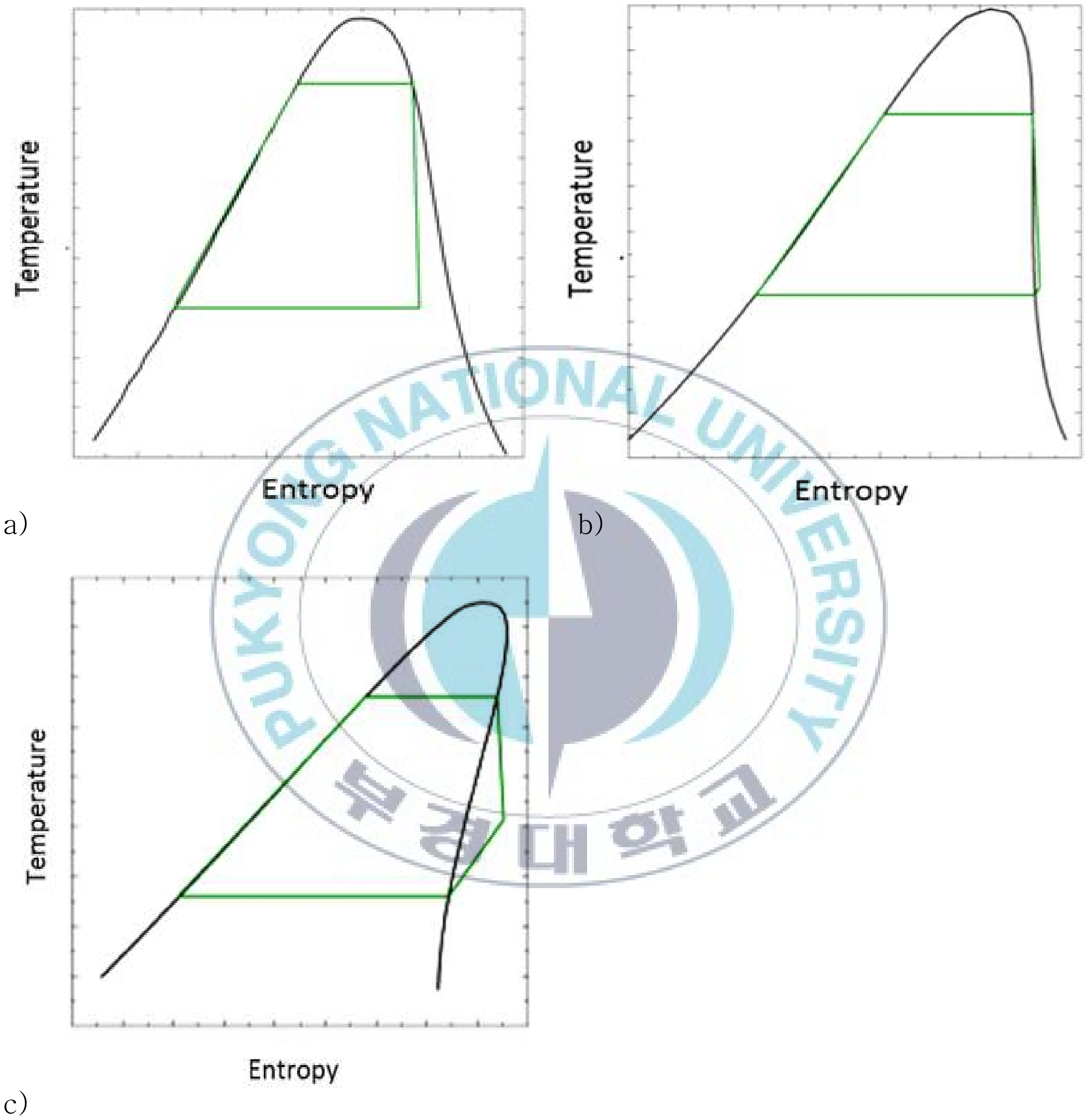


Fig. 2.2 a) Wet, b) Isentropic and c) Dry working fluids[24]



GWP: This represents how much a given mass of a chemical contributes to global warming over a given time period compared to the same mass of carbon dioxide set to the unity. All GWP values represent global warming potential over a 100-year time horizon [25].

**ODP**: This indicates the potential of a substance to destroy the ozone layer relative to chlorofluorocarbon-11 (CFC-11) fixed at one (ODE=1.0). Thus, a substance with ODP of 2 is twice as harmful as CFC-11 which has been phased out. Most refrigerants have ODP of 0 or close to 1 since those with ODP over 1 are being phased out by the Montreal Protocol.

**Availability and cost**: The selected refrigerant must be commercially available and at a reasonable cost. Most hydrocarbons and alkenes (used with proper precaution) used in chemical and refrigeration industries are easily obtainable and quite cost effective.

**Chemical stability**: With the exception of water, organic fluids tend to chemically decompose and deteriorate at high temperatures and pressures. The chemical stability of the working fluid can determine the allowable maximum heat source temperature.

**Vapor density**: This property is very important particularly for fluids with very low condensing pressure. A high vapor density is desired as low density involves high fluid velocities leading to higher volume flow rate, higher pressure drops in heat exchangers and increased expander size which directly increases cost.

**Viscosity**: A low liquid and vapor viscosity is necessary to minimize frictional losses in heat exchangers and maximize convective heat exchanger coefficient while reducing power consumption.

**Thermal conductivity**: A high thermal conductivity is required to achieve high heat transfer coefficients in the heat exchangers used.

**Pressures**: The maximum acceptable pressure is desirable for optimal efficiency in the ORC. High evaporating pressures lead to higher investment costs and increased complexity and the condensing pressure should be higher than the atmospheric pressure to prevent system air infiltration.

**Heat capacity**: A low liquid heat capacity can lead to low energy recovery from the heat source and reduced total cycle efficiency when working with finite heat sources.

**Vapor specific volume**: This gives an indication of condenser size as fluids with high saturation vapor volumes require larger condensing equipment making the system bulky and costly.

**Enthalpy difference**: In order to reduce flow rate and increase efficiency the enthalpy reduction of the working fluid in the expander needs to be large.

**Melting point**: This should be lower than the lowest operating ambient temperature to prevent the working fluid from freezing despite seasonal changes.

**Fluid compatibility**: The fluid should have mutual solubility with lubricating oils and not react when in contact with materials (heat exchangers, pipes, seals etc.). Generally, most lubricating oils are compatible with organic fluids with refrigerant-oil reactions accelerating with increased heat source temperature.

Therefore, a rough methodology of fluid selection can therefore be summarized in order of priority according to [24] as;

a) Defining the working temperature range for the heat source and heat sink;

b) Focusing on fluid environmental and safety concerns (Montreal Protocol);

c) Evaluating thermodynamic properties and determining cycle efficiencies;

d) Verifying availability of expansion machines in terms of rational operating range.

More also, Chen et al. [26] screened 35 potential working fluids for ORC and presented that although ORC cycles do not have a good thermal match with their heat sources, using fluids with high density and latent heat provide high turbine work output and superheating is only needed for wet fluids because dry and isentropic fluids perform better and Saleh et al. gave a thermodynamic screening of 31 pure component working fluids for ORC using BACKONE equation of state [27].

Besides, a study [28-29] comparing selected working fluids showed that for a proposed DOTEC cycle, wet fluids with high critical temperatures and heat addition rates possess superior turbine power and least pump work and a proposed performance criterion showed R717 as the most promising working fluid among nine others simulated and analyzed using Aspen HYSYS simulation software.

Likewise, the effect of using mixtures as working fluids in ORC was examined with a simulation model for heat sources of 150°C and 250°C and it showed a potential increase of 16% and 6% respectively, in cycle efficiency per heat source temperature and the boiling point temperature of the fluid has a strong effect on the system cycle efficiency with R245fa showing the best efficiency for temperatures between 107°C and 157°C and isobutane, for temperatures lower than 107°C[30-31].

Additionally, the performance analysis of a supercritical ORC system driven by exhaust heat was evaluated for 18 organic fluids based on their working parameters (exergy efficiency, expander size, etc.) and results showed that increasing expander inlet temperature improves exergy efficiency and net power output with R152a and R143a standing out as the best fluids [32]. Supercritical ORC using geothermal energy as heat source can obtain thermal efficiency as high as 21% at 200°C source temperature and 10°C sink temperature and thermal efficiencies higher than 12% for medium source temperatures with optimum pressure ration dependent on heat source temperature (125–150°C)[33].

More over, a regenerative ORC using dry organic fluids, to convert waste energy to power from low-grade heat source shows higher efficiency with minimum irreversibility compared to the basic ORC, the high heat capacity, lower latent heat-to-heat capacity ratio and higher gas density of HFC-245fa makes it suitable for ORC applications as it improves the heat exchanger performance and overall efficiency and employing ORC for biomass applications has an added advantage of efficiency improvement at higher source temperatures [34-35].

Finally, different papers were compared in terms of their target application, condensing temperature and evaporating temperature range and it follows that despite the numerous working fluid studies, there is no single optimal fluid for the ORC since most authors used different hypothesis and/or working conditions and the objective meaning of optimization could vary depending on the target application as solar applications have maximized cycle efficiency whereas WHR applications require maximized output power [36].

On improving ORC efficiency through working fluids, a 1MW solar plant ORC module using n-pentane as working fluid provided an efficiency of 20% [4]. Based on this and other past studies on working fluid selection [5,37-38], this study employs hydrocarbons (HCs); n-pentane, n-hexane, iso-butane, propane (R290), hydrofluorocarbons (HFCs); R245fa, R152a and a mixture of pure hydrocarbons. Aside hydrocarbons being environmental friendly, they can save about 36% on energy cost and are also efficient conductors of heat with generally low operating pressures reducing the pressure on piping and the system as a whole [39].

Hydrofluorocarbon working fluids especially R245fa are also efficient for moderate temperature systems due to their relatively high critical temperatures. Thus, a mixture of R245fa and the hydrocarbon which records the highest efficiency is also employed to access the performance since a mixture can allow power generation at a wide temperature range and reduce irreversibilities by allowing countercurrent flow condensing process in the condenser during heat rejection compared to single working fluids. The mass fractions of the mixtures are varied in a ratio of 9:1. The thermo-physical properties of the fluids for this study is presented in Table 2.1.

Table 2.1 Thermo-physical properties of selected fluids.

| Working fluid | Molecular weight | Boiling point temperature | GWP | ODP | Safety |
|---|---|---|---|---|---|
| Pentane | 72.1 | 36.1 | 4 | 0 | A3 |
| Hexane | 86.18 | 69 | 3 | 0 | A3 |
| Isobutane | 58.1 | -11.7 | 3 | 0 | A3 |
| R290 | 44.1 | -42 | 3.3 | 0 | A3 |
| R152a | 66 | -25 | 0 | 0 | A2 |
| R245fa | 134 | 15.3 | 0 | 0 | B1 |
| A2:Lower flammability; A3:Higher flammability; B2: No flame propagation | | | | | |

Even though the considered Hydrocarbons are flammable, existing ORC processes at higher temperatures employ R601 as working fluid making it not much of a worry in ORC fluid selection [5].

## 2.4 ORC component selection

### 2.4.1 Heat exchangers

Conservation of energy is very imperative in process design and the calculation of the minimum cooling and heating requirement for a heat exchanger network shows substantial energy savings which is an essential tool and stage in determining the cost of preliminary design. The first law of heat integration states that the amount of heat that has to be supplied or removed is the difference between the heat available in the hot streams and the heat required for the cold streams and the type of power plant determines the type and number of heat exchangers to use. Heat exchangers are used for transferring energy or heat between two physically separated fluids (from one stream to another). They make up to 30% of the total ORC plant cost. The heat exchangers mainly empl Most of the studies carried on ORC is based onoyed in the ORC involves the evaporator and condenser.

Heat extraction from low heat temperature requires high heat area which means a large sized evaporator, condenser and if necessary recuperator and these incurs additional cost. Efficient and optimum heat exchangers can outweigh the cost of the system so heat exchangers with good heat transfer characteristics like titanium flat plate can be helpful [40]. Their use and technology is well established in industries and they are available in various designs and configurations to suit the diverse needs, materials and operating conditions. After the flow rate, size of heat exchanger and other analysis of the flow characteristics have been established, Table A1 can be used as a guide for the selection of heat exchangers for WHR applications. In WHR, high heat efficiencies are neither always essential nor preferred and recovery equipment with a lower efficiency may suit the job requirement if all of the heat in the waste heat stream cannot be utilized. This will provide the device advantages like easier maintenance, lower pressure drop etc. From Table A1 (Appendix A), the preferred heat exchanger is the shell and tube.

Although, Plate heat exchangers are also very efficient and compact in size, they are usually for pressures lower than those encountered in power plants. This makes shell and tube heat exchangers the most viable in ORC applications. Also, shell and tube heat exchangers are available in a wide range of standard sizes with many combinations of materials for the tube and shells. The number of passes (that is, the number of times the fluid in the tubes passes through the fluid in the shell) and the type of flow, being it concurrent or countercurrent can help in the selection of the types of shell and tube heat exchanger but a one pass design is preferred from a thermodynamic point of view since it gives a pure countercurrent flow for higher efficiencies [41]. A more expensive but fouling resistant material can be chosen for heat exchangers in order to avoid corrosions and increase the life span of the power plant. The working fluid from the shell can be made to pass over baffled tubes as shown in Fig. 2.3 to increase the effectiveness of the heat exchanger.

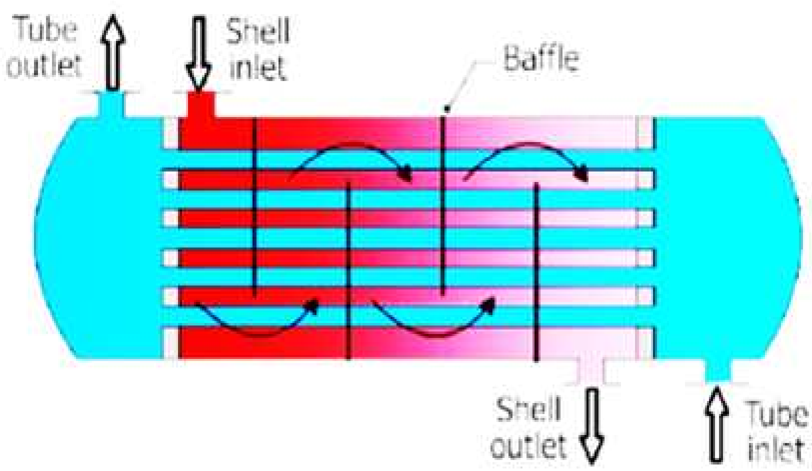


Fig. 2.3 Schematic of the shell and tube heat exchanger [41]

### 2.4.2 Turbine

Turbine is the most important part in an ORC system. The purpose of the turbine is to change the potential energy of pressurized gases into rotational kinetic energy through the expansion of the high pressure vapour organic fluid. The stream of high-pressure vapor organic fluid expands in the turbine, causing its internal part to rotate. The rotor is connected by a shaft to the generator which changes rotational kinetic energy into electricity. The selection of the turbine for a given operating conditions and working fluid depends on its compactness, reliability, performance and technical limitations such as speed, temperature and supply. In most fluid energy power extraction processes, turbines can be substituted for expanders and vice versa as they almost perform the same functions. However, expanders are not ideal for kinetic energy applications since the fluid does not go through significant expansion process [42].

### 2.4.3 Pumps and pumping system

Pumping systems consume the most electrical energy in industries with percentile consumption ranging from about 25 to 50. Thus, there is the need to curb energy consumption through good operating practices and smart designing in order to reduce cost [43]. Typical pumping systems consist of prime movers, pumps, valves, piping and an ultimate tail end equipment (which in this case is a heat exchanger). Pumping systems use pressure to transfer liquids from a source to a required destination or circulate liquids around a system. They overcome static and friction head losses due to the pressure required to make the liquid flow.

Since most proposed prototype ORC plants operate as a closed loop circulating system, the only loss the pump must overcome is friction loss (dynamic head loss) which is proportional to the square of the flow rate. Pumps are generally classified as dynamic or positive displacement according to their mode of operation or how they add energy to a fluid. Positive displacement pumps are widely used for pumping highly viscous fluids whereas, centrifugal pumps are employed where less viscous fluids (liquids especially water) are used and are the most used in industries (about 75%) due to its simplicity, long life span and minimal maintenance. A pump's performance can be graphically expressed as head against flow rate as shown in Fig. 2.4 and 2.5. The head increases with increasing system resistance leading to a decrease in flow rate for dynamic pumps and possibly, a zero flow rate which can burn out the pump. Positive displacement pumps display an almost constant head but they usually pump a fixed quantity after each revolution. This implies, the pump can be damaged should the delivery pipe be congested due to pressure rise [44].

To identify the effect of an installed pump, the system curves and the pump can be overlaid and the point of intersection of the two curves chosen as the operating point.

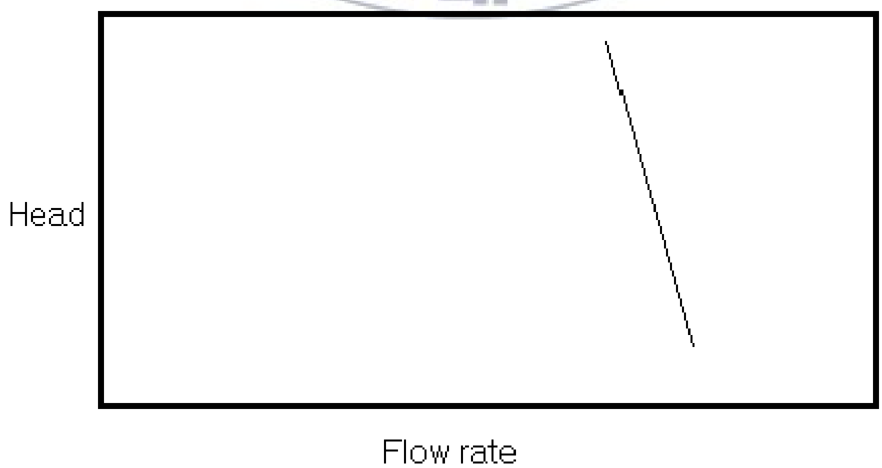


Fig. 2.4 Performance curve for a dynamic pump [44]

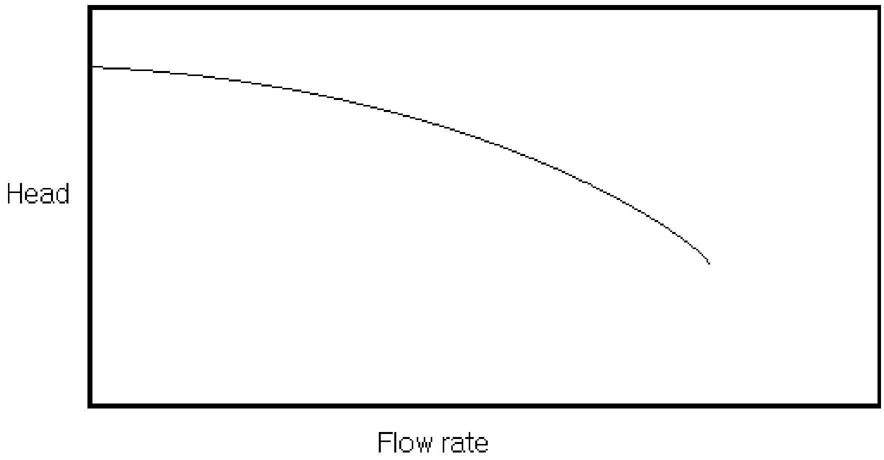


Fig. 2.5 Performance curve for a positive displacement pump [44]

This will help to predict the pump behavior because when the flow and head is calculated, the actual system curve will operate at a flow and head similar to that calculated in reality.

The choice of pump for a given application depends largely on how the pump head-flow characteristics match the requirement of the system downstream of the pump and the fluid properties such as density, viscosity, particulate content and vapor pressure.

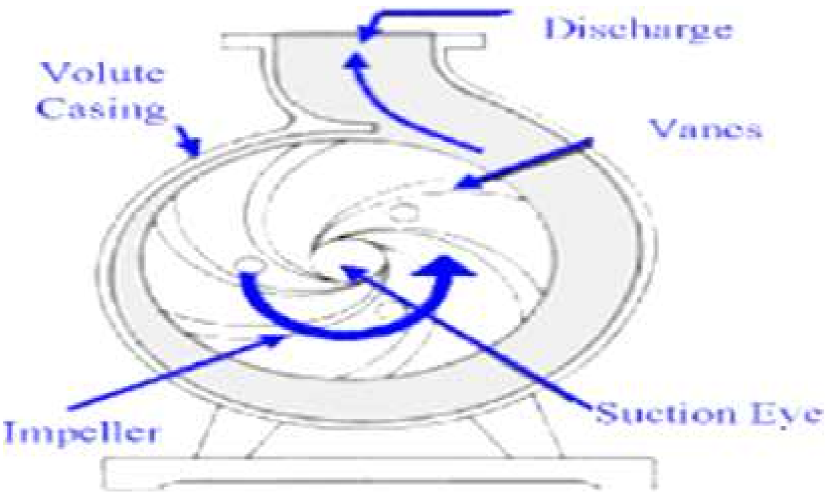


Fig. 2.6 Liquid flow path of the centrifugal pump [45]

From the aforesaid characteristics, the recommended pump for this ORC study is the centrifugal pump. Fig. 2.6 and 2.7 shows the liquid flow path and main components of a centrifugal pump.

## 2.5 Design boundary conditions

Setting the design boundaries is a prerequisite for the best performance of the ORC plant. The heat source temperature, mass flow rate, heat sink temperature and type of fluid used to transfer heat to the evaporator have immense effect on the ORC performance.

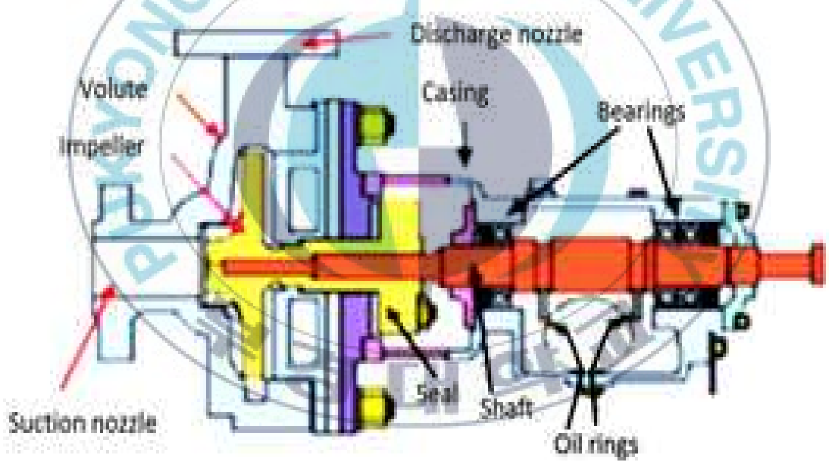


Fig. 2.7 Main components of the centrifugal pump [45]

With all these boundary conditions fixed, the size of the component especially the heat exchangers can also optimize power capacity. However, the size of the individual components, the cost, piping works involved, availability of space, and other financial analysis such as payback time (which should be low), internal rate of return (the more the internal rate of return, the faster the investment return benefit) must be considered before configuring an ORC plant [45]. Also, for economic feasibility, it is required to have an ORC system that effectively recovers waste heat over a wide temperature range from multiple low grade heat sources, is operational and requires only minimal maintenance.

# CHAPTER 3. SYSTEM MODELING

## 3.1 System description of the WHR-ORC

As shown in Fig. 3.1, the basic WHR-ORC consists of an evaporator, turbine, condenser and a pump. Waste heat from the power plant effluent enters the evaporator and exchanges heat with the working fluid to vaporize it. The vaporized working fluid from state 2 then expands in a turbine coupled to a generator to produce power. It exits at state 1 and enters the condenser to exchange heat with surface sea water. The liquefied working fluid then enters the feed pump (state 4) where its pressure is increased and sent to the evaporator (state 3) to repeat the cycle.

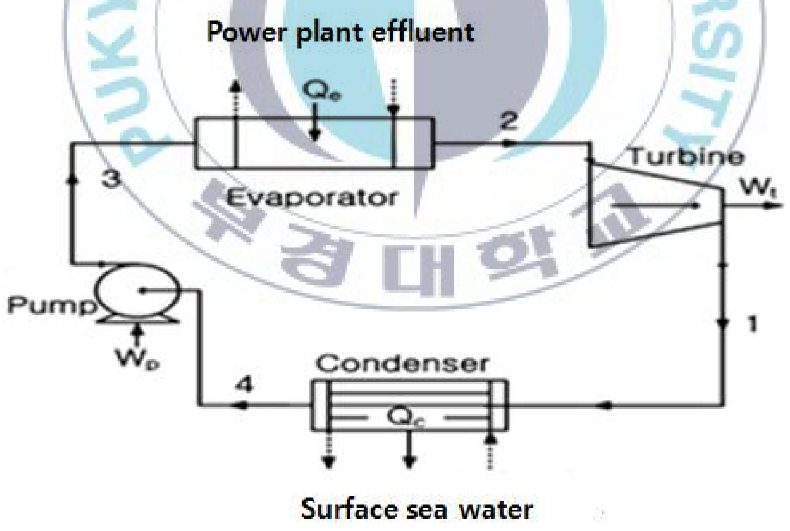


Fig. 3.1 Schematic diagram of the WHR-ORC [46]

Depending on the location of the ocean, the surface seawater temperature can be between 25-30°C. Fig. 3.2 and 3.3 presents the described OTEC process in the T-s and P-h diagrams.

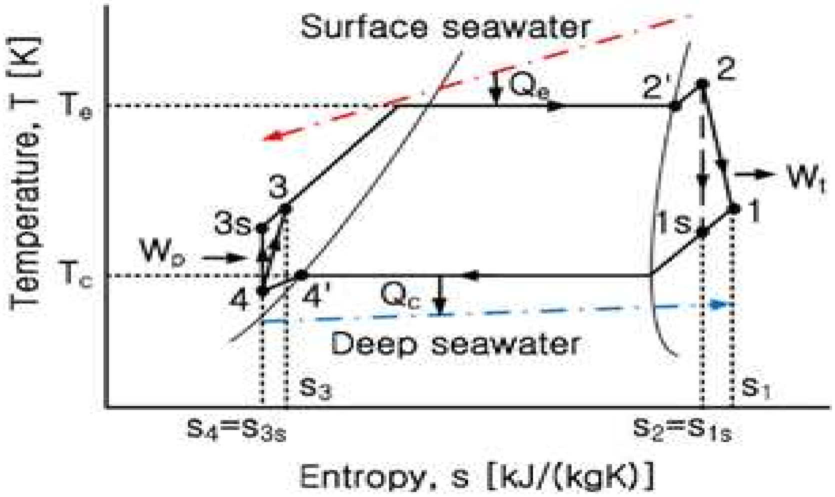


Fig. 3.2 T-s diagram of the WHR-ORC system [46]

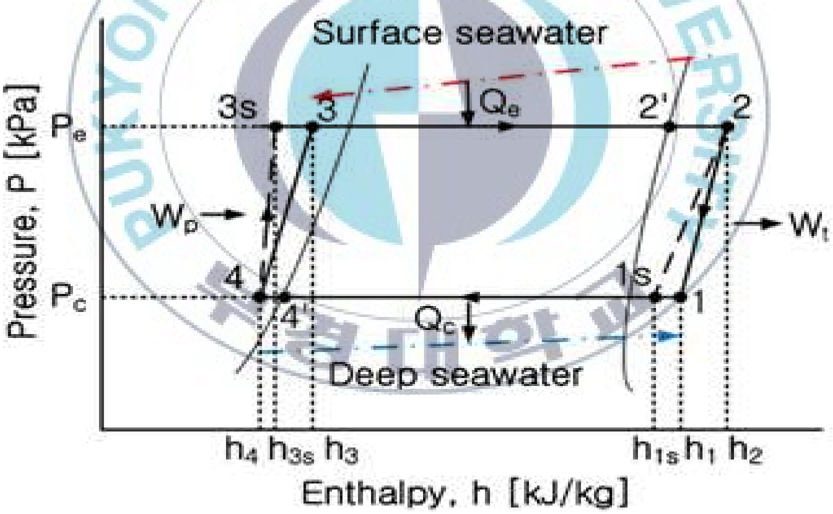


Fig. 3.3 P-h diagram of the WHR-ORC system [46]

## 3.2 Mathematical analysis

The mathematical model is analyzed as follows:

### 3.2.1 Heat addition process

In this process heat is absorbed from the heat-laden thermal oil to the working fluid in the evaporator. This process is considered isobaric but friction in evaporator pipes can cause slight pressure drops. The working fluid enters the evaporator at state 3 and leaves at state 2. The heat added to the working fluid can be determined as

$$Q_{Evaporator} = \dot{m}_{re}\left(h_{Evaporator\ exit} - h_{Evaporator\ inlet}\right) \quad 3.1$$

Where

$Q_{Evaporator}$ = heat added to the working fluid

$\dot{m}_{re}$ = mass flow rate of refrigerant in kg/s

$h_{Evaporator\ exit}$ = enthalpy at evaporator outlet

$h_{Evaporator\ inlet}$ = enthalpy at evaporator inlet

### 3.2.2 Expansion process

The working fluid expands in the turbine to generate energy. This process is considered isentropic although the turbine efficiency is always less than 100%. The useful turbine output work can be estimated as

$$W_{turbine} = \dot{m}_{re}\left(h_{turbine\ inlet} - h_{turbine\ outlet}\right).\eta_t \quad 3.2$$

Where

$W_{turbine}$ = useful work produced by turbine

$h_{turbine\ inlet}$ = vapor enthalpy at turbine inlet

$h_{turbine\ outlet}$ = vapor enthalpy at turbine outlet

$\eta_t$ = efficiency of turbine

### 3.2.3 Heat rejection process

The condenser rejects heat from the working fluid. This process is also considered isobaric neglecting pressure drop in condenser pipe. The working fluid leaves the condenser as saturated liquid. The amount of heat rejected can be deduced as

$$Q_{Condenser} = \dot{m}_{re}\left(h_{Condenser\ inlet} - h_{Condenser\ outlet}\right) \qquad 3.3$$

Where

$Q_{Condenser}$ = amount of heat rejected in the condenser

$h_{Condenser\ inlet}$ = enthalpy at condenser inlet

$h_{Condenser\ outlet}$ = enthalpy at condenser outlet

### 3.2.4 Pump work

The working fluid leaves the condenser as saturated liquid from state 4 and pumped to the evaporator at constant entropy. Thus, the pump work can be expressed as

$$W_{Feed\ Pump} = \dot{m}_{re}\left(h_{Feed\ inlet} - h_{Feed\ outlet}\right)/\eta_p \qquad 3.4$$

Where

$W_{Feed\ Pump}$ = feed pump work

$h_{Feed\ inlet}$ = enthalpy at feed pump inlet

$h_{Feed\ outlet}$ = enthalpy at feed pump outlet

$\eta_p$ = efficiency of pump

And

$$Net\ power = W_{turbine} - W_{Feed\ Pump} \qquad 3.5$$

Finally, the system efficiency of the WHR-ORC can be estimated as

$$\eta = \frac{W_{Turbine} - W_{Pump}}{Q_{Evaporator}} \qquad 3.6$$

Economic analyzers adapted from are used to analyze the WHR-ORC in order to help select the most optimized fluid. The economic analyzers are presented in Table 3.1.

Table 3.1 Economic analyzers for the WHR-ORC

| Economic Analyzer | Deductions |
|---|---|
| Evaporator Sizing | APRE = Evaporator Capacity/Gross turbine work (3.7) |
| Condenser Sizing | APRC = Condenser Capacity/Gross turbine work (3.8) |
| Turbine to pump ratio | TTP = Gross turbine work/Pump work (3.9) |

## 3.3 Simulation conditions

As shown in Table 3.2, the efficiencies of the turbine and pump are fixed within their typical range of applicable efficiencies however, they can be manipulated for specific accuracies as per the designer. The optimization variables for this study are the working fluid evaporator inlet pressure, turbine outlet pressure and the heat exchanger LMTD's. After these factors are selected and fixed the working fluid evaporator inlet mass flow rate is varied until the desired gross turbine power is achieved. Thus, the fixed variables can be manipulated for optimized system performance as per desired output. Table 3.3 presents the monthly data for one year from Namhae power plant effluent and surface seawater while Fig. 3.3 shows a picture of the waste heat source heat from the heater drain from Namhae power plant.

Table 3.2 Simulation assumptions and inputs for 10MW/20kW gross turbine power

| Parameters | Inputs |
|---|---|
| **Efficiencies** | |
| Feed Pump | 65% |
| Turbine | 85%/80% |
| **Pressure drops** | |
| Heat source in evaporator | 50kPa |
| Surface seawater in condenser | 50kPa |
| Between working fluid and heat exchangers | 10kPa |
| **Log mean temperature difference (LMTD)** | |
| Evaporator | 3.5°C |
| Condenser | 3.5°C |
| **Temperature variations** | |
| Change in heat source evaporator inlet and outlet temperature | 10°C |
| Change in cooling source condenser inlet and outlet temperature | 7°C |

Table 3.3 Evaporator and condenser input data for the WHR-ORC

| Phase 1 | | |
|---|---|---|
| Month | Heat source temperature (°C) | Cooling source (surface sea water) (°C) |
| Average monthly temperature | 75 | 17 |
| Phase 2 | | |
| 1. January | 74 | 8 |
| 2. February | 74 | 8 |
| 3. March | 75 | 11 |
| 4. April | 75 | 14 |
| 5. May | 76 | 18 |
| 6. June | 76 | 21 |
| 7. July | 76 | 23 |
| 8. August | 76 | 24 |
| 9. September | 77 | 25 |
| 10. October | 76 | 21 |
| 11. November | 75 | 16 |
| 12. December | 75 | 11 |

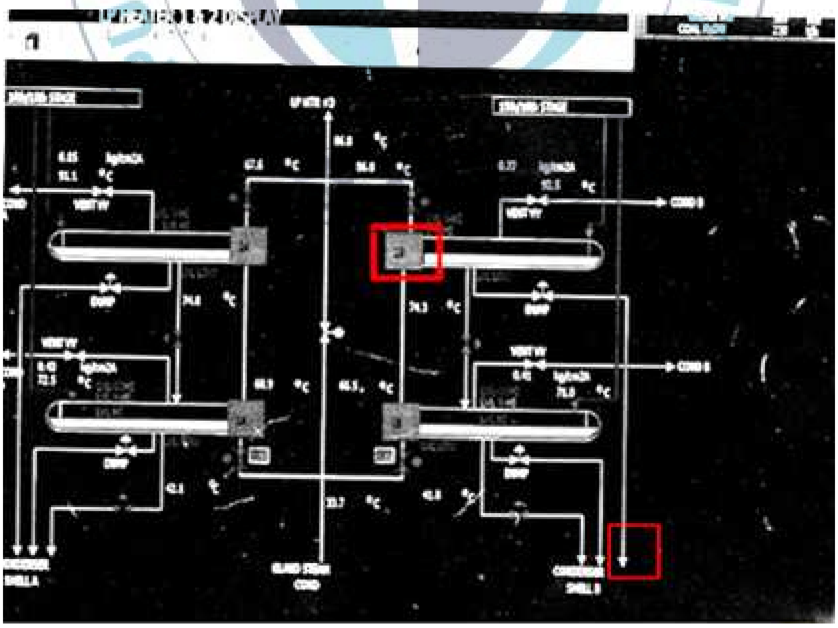

Fig. 3.4 Source of effluent from the heater drain of Namhae power plant

# CHAPTER 4. SIMULATION RESULTS AND DISCUSSIONS

The results from the simulation are presented in figures and tables and discussed as follows.

## 4.1 Phase 1

Table 4.1 Evaporator inlet pressures for selected working fluids

| Working fluid | Evaporator inlet pressure(kPa) |
|---|---|
| Pentane | 262.6 |
| Hexane | 98.95 |
| Isobutane | 1008 |
| R152a | 1836 |
| R245fa | 1354 |
| R290 | 2470 |
| Pentane/Hexane (90/10) | 248.1 |
| Pentane/Hexane (10/90) | 117.4 |
| Pentane/Isobutane (90/10) | 327.8 |
| Pentane/Isobutane (10/90) | 864.8 |
| Isobutane/R245fa (10/90) | 1271 |
| Isobutane/R245fa (90/10) | 1021 |

As seen in Fig. 4.1.1, the R152a and R245fa both being hydrofluorocarbons, showed the highest mass flow rates for the pure or single fluids whereas the mixture of R245fa and isobutane showed the highest for the mixtures. Although R245fa has a higher boiling point temperature compared to R152a from Table 2, it is a dry fluid so left the turbine with substantial heat thereby increasing its mass flow rate. It recorded an increment of 46.9% in mass flow rate compared to R152a which is a wet hydrofluorocarbon. For the hydrocarbons, hexane, showed the highest mass flow rate with pentane being the lowest.

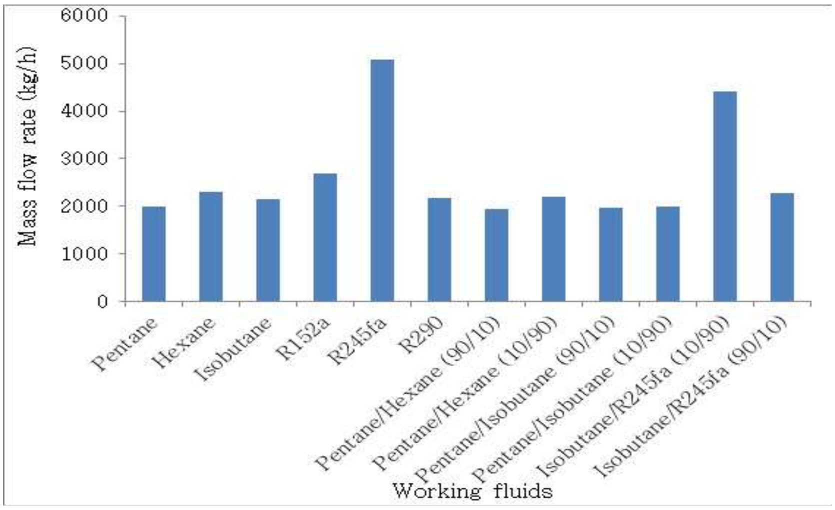


Fig. 4.1.1 Comparison of mass flow rate for selected working fluids

However, a mixture of 9:1 pentane and hexane presented the lowest mass flow rate for the hydrocarbon mixtures while 9:1 isobutane/R245fa reduced the flow rate of pure R245fa by 55%.

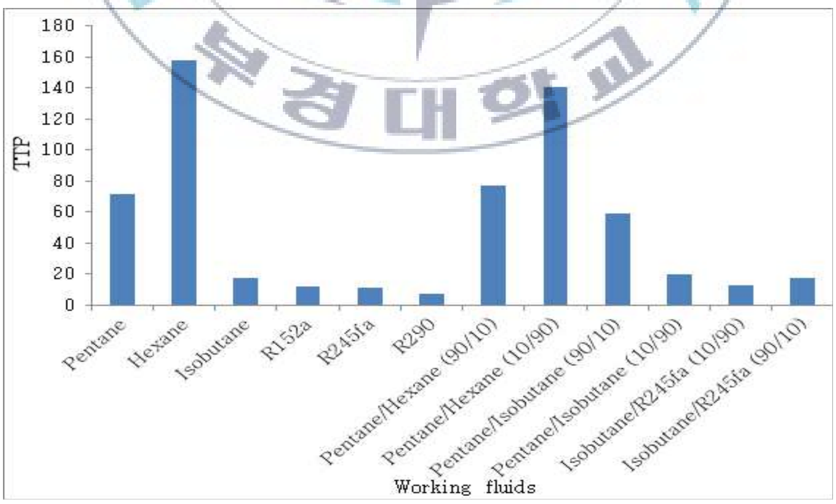


Fig. 4.1.2 Comparison of TTP for selected working fluids

From Fig. 4.1.2, the hydrofluorocarbons performed poorly in TTP with pentane and hexane showing high TTPs while a 1:9 mixture of pentane/hexane showed the highest TTP for the mixtures. From Table 2, boiling point temperature shows a direct correlation with TTP for hydrocarbons as the higher the boiling point temperature and molecular weight, the higher the TTP and an indirect correlation for the hydrofluorocarbons. TTP shows the performance of the turbine to pump work, that is, the power produced to that consumed. Therefore, a high TTP is desirable and the hydrocarbons (pentane, hexane and isobutane) are the most favourable working fluids in that sense.

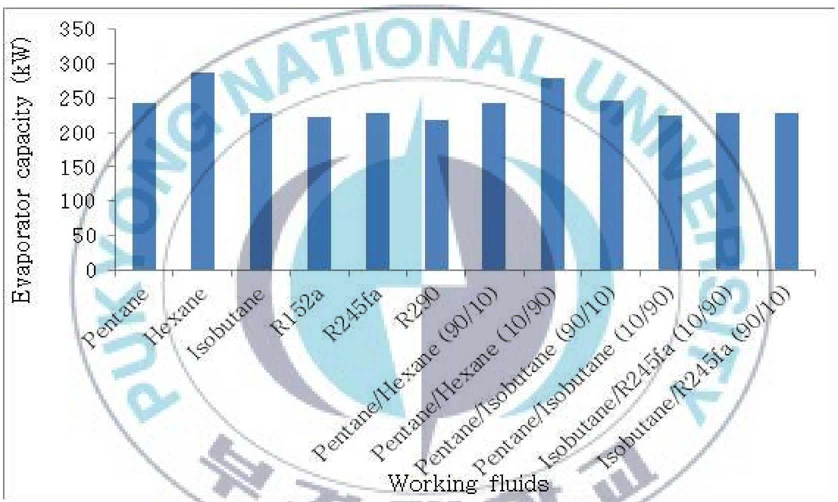


Fig. 4.1.3 Comparison of evaporator capacity for selected working fluids

As depicted in Fig. 4.1.3 and 4.1.4, evaporator capacity is proportional to APRE with all the pure fluids showing fairly high values and the mixtures slight decrements. A higher evaporator capacity has a corresponding high APRE which is a sizing factor for the evaporator. From Table 2, a low molecular weight coupled with low boiling point temperature leads to a lower APRE because the evaporator inlet vapour density is reduced leading to a lower capacity. Based on the economic analyzer deduction from Table 3.3, a lower APRE implies a small sized evaporator which is desired to reduce cost. Thus, R290 and 1:9 mixture of pentane/isobutane presents the lowest APRE.

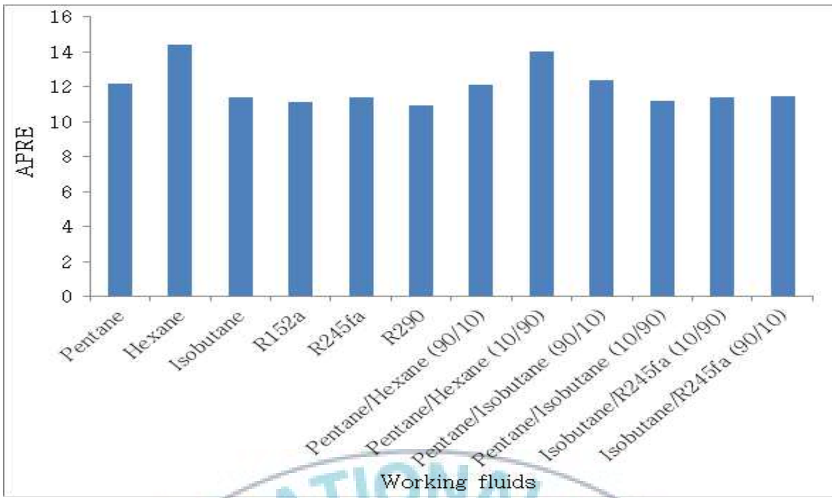


Fig. 4.1.4 Comparison of APRE for selected working fluids

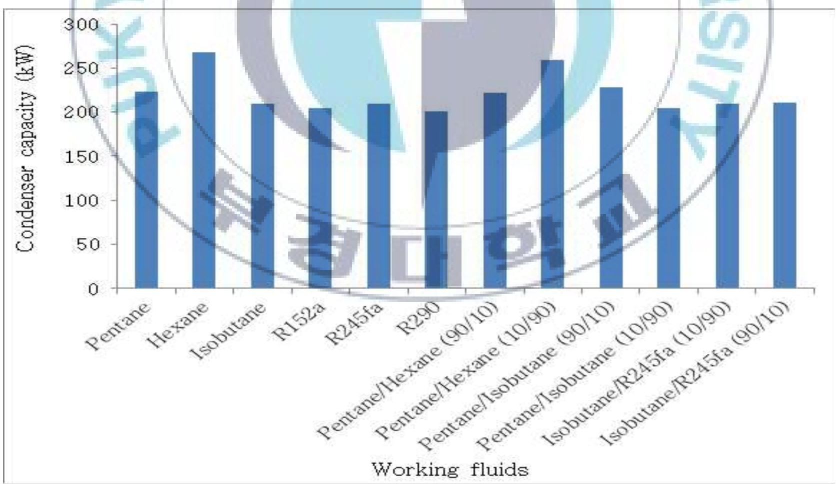


Fig. 4.1.5 Comparison of condenser capacity for selected working fluids

Figs. 4.1.5 and 4.1.6 displayed similar characteristics as Fig. 4.1.3 and 4.1.4. This shows that the properties exhibited by the fluid at the evaporator inlet has a direct implication on the condenser inlet characteristics of the system as per the aforementioned simulation conditions. So, R290 and 1:9 mixture of pentane/isobutane presents the lowest APRC as well.

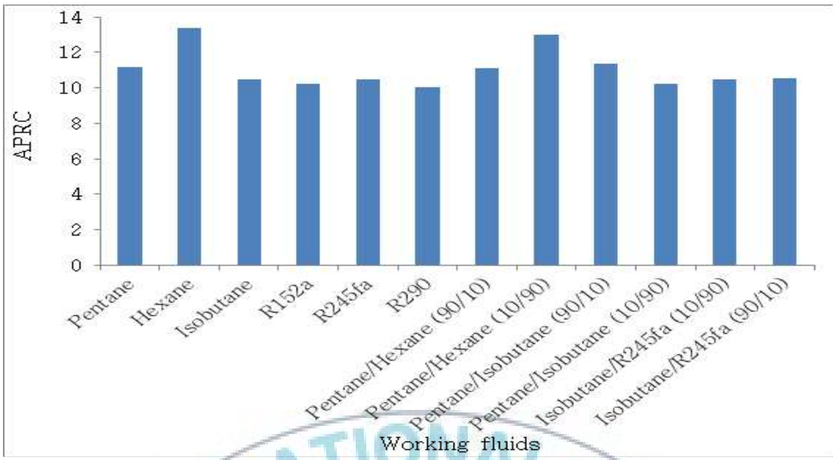


Fig. 4.1.6 Comparison of APRC for selected working fluids

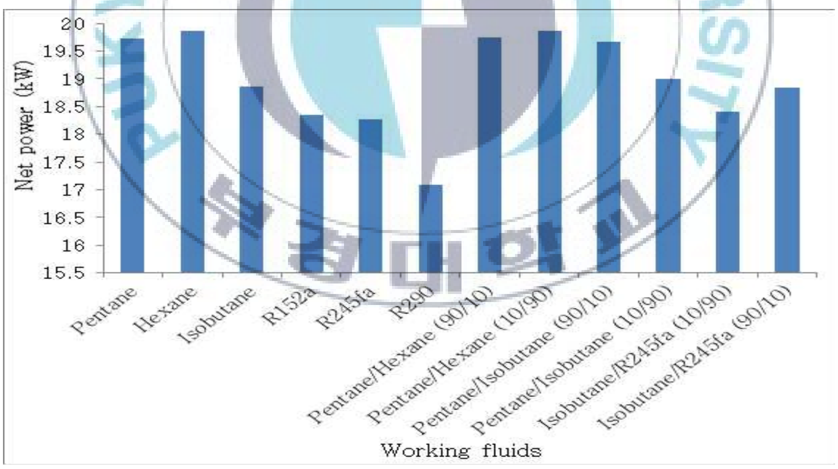


Fig. 4.1.7 Comparison of net power for selected working fluids

From Fig. 4.1.7, the hydrocarbons generally showed high net powers with the exception of R290. Hexane had the highest net power only 0.1% higher than that recorded by the highest mixture. This result is conforms to the evaporator inlet pressures from Table 4.1 of the working fluid. Turbines convert differential pressure to energy and a larger differential pressure leads to a high energy output. As such, a low pressure correlated to a high net power for the hydrocarbon and inverse for the hydrofluorocarbons.

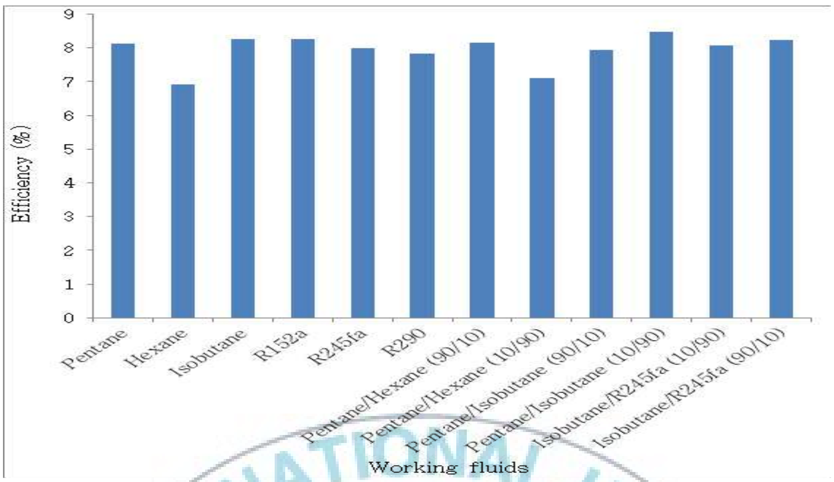


Fig. 4.1.8 Comparison of efficiency for selected working fluids

As shown in Fig. 4.1.8, all the fluids showed high efficiencies emphasizing their positive effect on the WHR-ORC cycle performance. Hydrofluorocarbons R245fa and R152a both had high efficiencies with R245fa which had a lower efficiency than R152a being about 3.2% lower than the highest hydrocarbon and 13.6% higher than the least hydrocarbon efficiency. Nevertheless, the hydrocarbon mixture of 1:9 pentane/isobutane recorded the highest efficiency among the selected working fluids.

## 4.2 Phase 2

To generate 1MW gross power using the monthly power plant effluent as heat source and surface sea water as cooling source, the results obtained from the simulation is presented as follows.

Table 4.2 Monthly evaporator inlet temperature and pressure as generated by HYSYS

| Month | Evaporator inlet temperature (°C) | Evaporator inlet pressure (kPa) |
|---|---|---|
| 1. January | 15.54 | 1020 |
| 2. February | 15.54 | 1020 |
| 3. March | 18.62 | 1041 |
| 4. April | 21.67 | 1037 |
| 5. May | 25.75 | 1057 |
| 6. June | 28.80 | 1053 |
| 7. July | 30.83 | 1051 |
| 8. August | 31.83 | 1049 |
| 9. September | 32.86 | 1073 |
| 10. October | 28.80 | 1053 |
| 11. November | 23.71 | 1035 |
| 12. December | 18.62 | 1041 |

From Fig. 4.2.1, working fluid evaporator inlet mass flow rate is highest in September about 0.57% higher than August and lowest in January and February. In Table 4.2, there is a positive correlation between working fluid evaporator inlet temperature, pressure and mass flow rate. The higher the evaporator inlet temperature and pressure, the higher the mass flow rate.

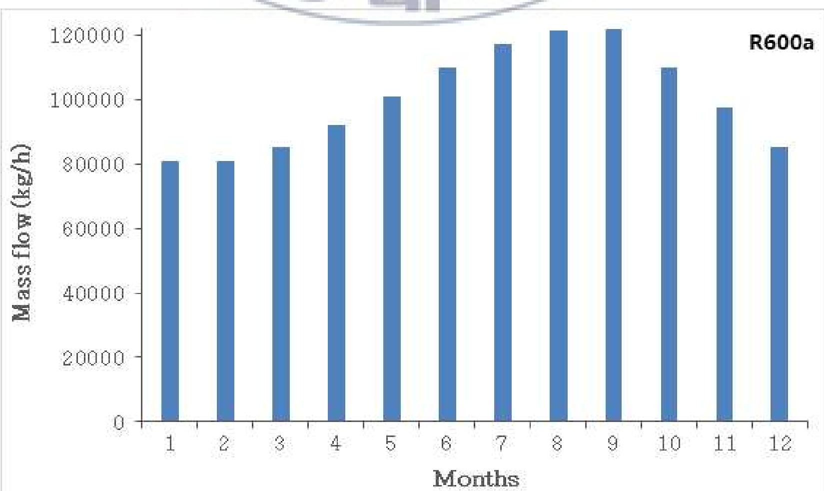


Fig. 4.2.1 Comparison of mass flow rate at various months

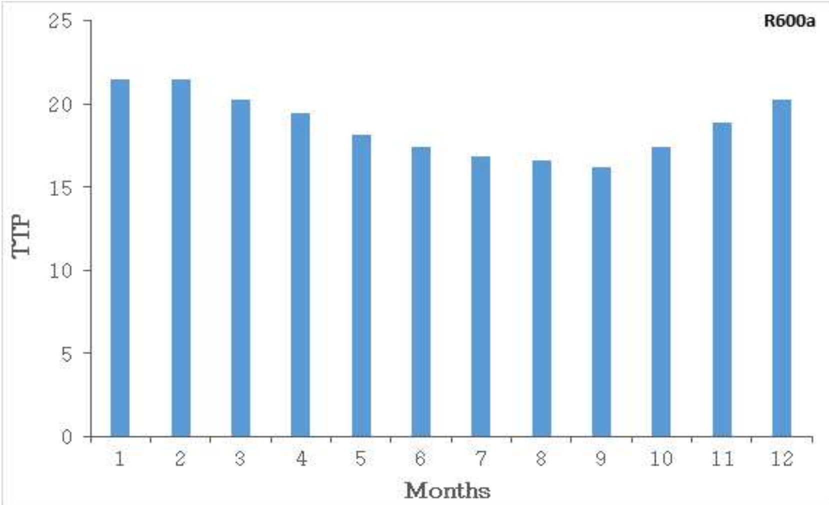


Fig. 4.2.2 Comparison of TTP at various months

From Fig. 4.2.2, TTP increased with an increase in cooling source temperature. The heat dissipation in the condenser was fairly poor due to the high condenser inlet temperature of the cooling source presented in Table 3.3 which increased the temperature of the working fluid exiting the condenser into the pump thereby increasing pump work while reducing TTP. The lowest TTP is seen in September and TTP is inversely proportional to mass flow rate.

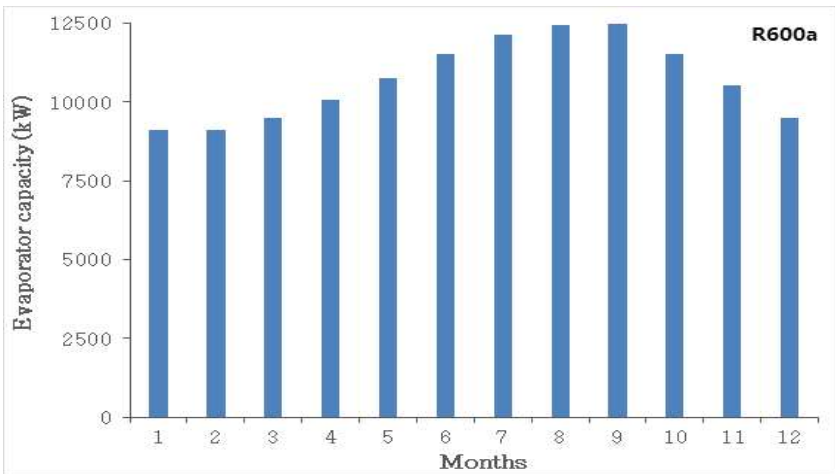


Fig. 4.2.3 Comparison of evaporator capacity at various months

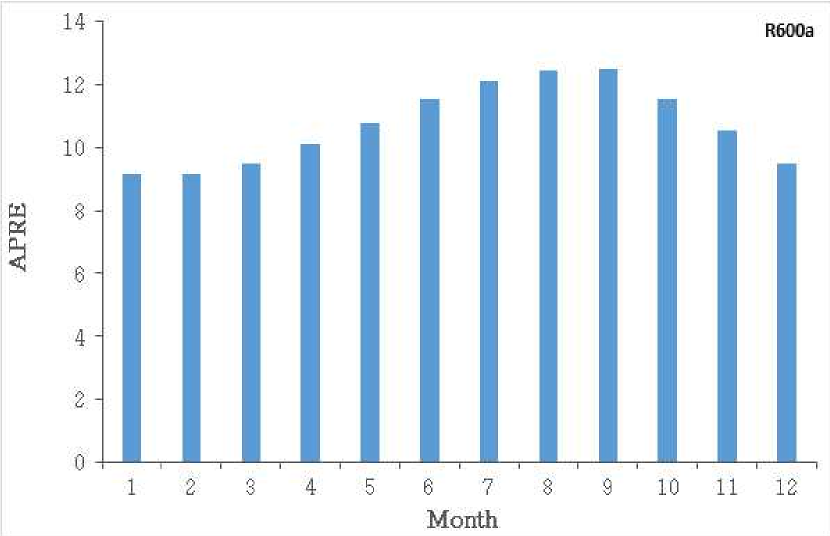


Fig. 4.2.4 Comparison of APRE at various months

There is a direct relationship between Figs. 4.2.3, 4.2.4, 4.2.5 and 4.2.6. The evaporator capacity, APRE, condenser capacity, and APRC have a proportional relationship with mass flow rate and change in evaporator inlet heat source and condenser inlet cooling source temperature with the highest being recorded in September.

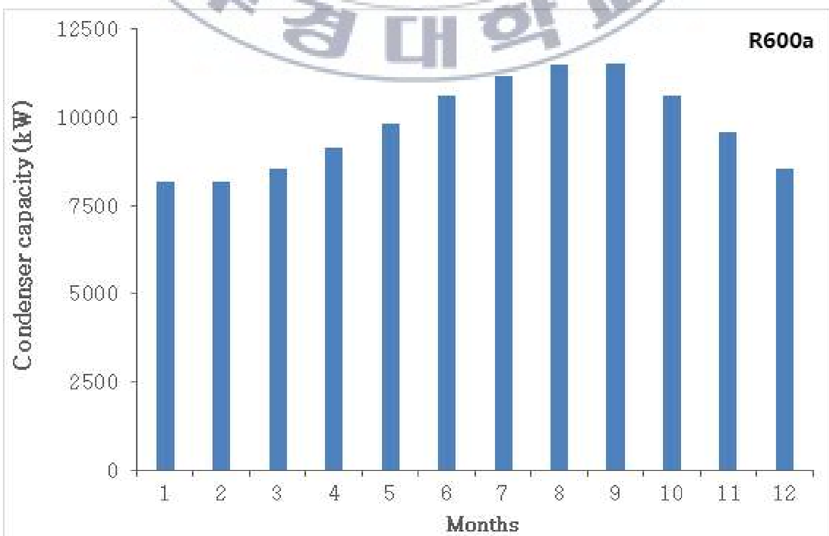


Fig. 4.2.5 Comparison of condenser capacity at various months

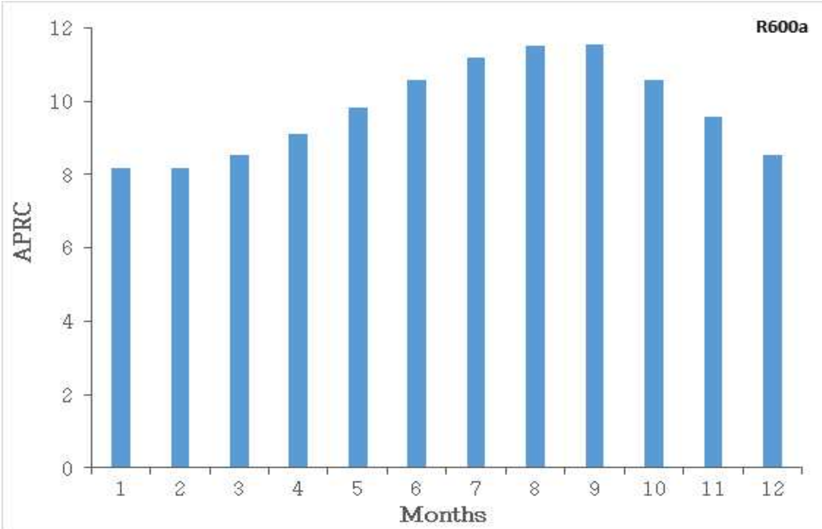


Fig. 4.2.6 Comparison of APRC at various months

From Fig. 4.2.7, the results are no different from that displayed in the foreseen figures as the month with the highest mass flow rate and pressure shows the lowest net power and vice versa. That is, for a particular choice of fluid with a fixed heat and cooling source temperature, net power is relatively dependent on the evaporator inlet mass flow rate.


Fig. 4.2.7 Comparison of net power at various months

However, compared to the increment of 33.8% in Fig. 4.2.1 by September which recorded the highest mass flow rate over January, February having the lowest mass flow rates, September showed 1.56% decrement in net power in contrast to January, February. This slight decrement is due to the closeness of the monthly evaporator inlet pressures from Table 4.2 as the difference between the highest and lowest pressure is 5.2%. Thus, the evaporator inlet pressure is a strong determinant of the net power rather than the mass flow rate as the larger the difference between evaporator inlet pressure for different heat source and cooling source temperatures, the larger the difference in net power and vice versa.

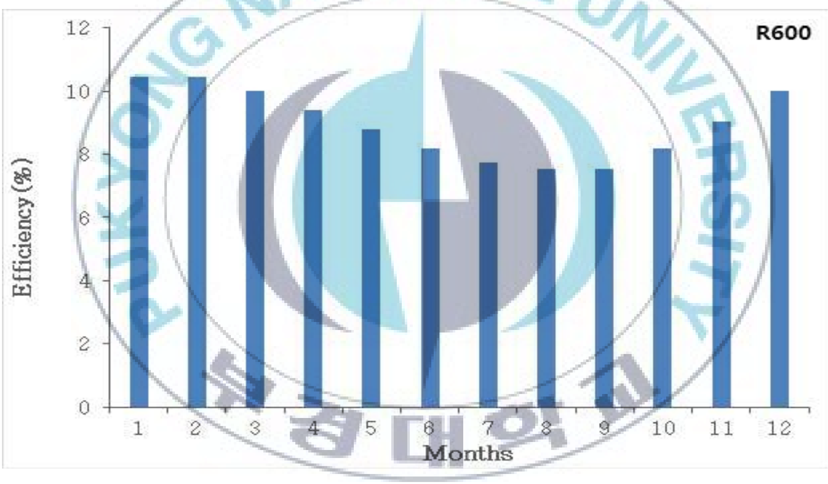


Fig. 4.2.8 Comparison of efficiency at various months

From Fig. 4.2.8, WHR-ORC cycle efficiency has a direct relationship with TTP and an inverse relationship with mass flow rate. Also, January, February with the highest efficiency of 10.45% has a corresponding TTP of 21.45 from Fig. 4.2.2 and September with the lowest efficiency of 7.52%, a TTP of 16.19. Therefore, a ratio of 1:2 can be used to predict (give a rough assumption) the cycle efficiency based on the calculated TTP. Thus, the turbine to pump ratio (TTP) is a determining factor of the WHR-ORC cycle efficiency outcome.

# CHAPTER 5. CONCLUSION

WHR-ORC (Waste heat recovery) power generation is a perfect example of the energy 'trilemma' of balancing the demand for energy security, cost-effectiveness and low carbon emission from a dedicated heat source where high temperature heat source is accessible. It is applicable especially at medium to low range temperature heat streams which make up more than 50% of the total heat generated in industries and more compatible with heat exchanger materials and a power generation alternative for industrial countries like South Korea to counterbalance on site electricity consumption or sent to a local utility to produce power for small community if in excess. Hydrocarbons and hydrofluorocarbons are environmental benign and good fluid choices for optimizing a WHR-ORC using effluent from a power plant and surface seawater as heat and cooling source, respectively.

A high concentration of R600a to R245fa mixture can reduce the cycle's mass flow rate by 55% as opposed to when using only R245fa as working fluid. On the cycle analyzers, the properties exhibited by the fluid at the evaporator inlet has a direct implication on the condenser inlet characteristics of the system at the assumed simulation conditions, Thus, a high evaporator capacity relates to a high condenser capacity and high APRE and APRC, respectively, with the hydrocarbons presenting the smallest APRE, and APRC and the biggest TTP which is more desirable when sizing down of the cycle is concerned. A low evaporator inlet pressure correlates to a high net power for the hydrocarbons and inverse for the hydrofluorocarbons with the pure and mixed hydrocarbons having the highest net power. Generally, for the selected fluids, high evaporator inlet pressures is linked to high cycle efficiency with the hydrofluorocarbons performing fairly well but required higher mass flow rates. These (high pressures and mass flow rate) can increase the cost of pumping systems and components as the lowest APRE and APRC for the hydrofluorocarbons is 1.84% and 1.37% bigger than the smallest of the hydrocarbons, correspondingly.

The higher the evaporator inlet temperature and pressure, the higher the mass flow rate. High condenser inlet cooling source increases pump work thereby decreasing TTP in autumn and TTP is inversely proportional to evaporator inlet mass flow rate. WHR-ORC cycle efficiency has a direct relationship with TTP and an inverse relationship with mass flow rate with a ratio of 1:2 as a determining factor for predicting the performance of the cycle's efficiency at different heat and cooling source temperatures. The highest system efficiency of 10.45% is recorded in winter about 28% higher than the least recorded in autumn.

For any WHR-ORC application, the application target is the amount of power which can be recovered but a high system efficiency indicates that the heat source is being utilized. Based on these, it is idle to select the minimum allowable pressure which offers a balance between net power, components costs as well as tolerable system efficiency. By using the economic analyzers, the selection of working fluid candidates for an optimized waste heat recovery organic Rankine cycle can be narrowed with much emphasis and considerations given to the boiling point temperature of the selected fluids as boiling point temperature is crucial to the fluid's performance.

Simulating and studying the range of operating conditions for a wide range of working fluids can help develop a data base for the selection of the most optimized working fluid for an optimized ORC system. Therefore, further studies on more hydrocarbon working fluids is recommended as they are safe and show good efficiencies.

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## NOMENCLATURE

| | | |
|---|---|---|
| Comm. | : Commercial | |
| Cont. | : Contamination | |
| Diff. | : Differential | |
| Equip. | : Equipment | |
| GWP | : Global warming potential | |
| h | : Enthalpy | [kJ/kg] |
| H.E | : Heat exchanger | |
| Inter. | : Intermediate | |
| m | : Mass flow rate | [kg/hr] |
| ODP | : Ozone depletion potential | |
| OTEC | : Ocean thermal energy conversion | |
| ORC | : Organic Rankine cycle | |
| Q | : Heat | [kW] |
| Temp. | : Temperature | |
| W | : Work done | [kW] |
| WHR | : Waste heat recovery | |

Table A1 operation and application characteristics of heat exchangers

| Comm. heat transfer equip. | Specifications for waste heat recovery unit | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Low temp. 0-120℃ | Inter. temp. 120-650 ℃ | Moisture recovery | Permits large diff. temp. | Packaged units available | Can be retrofitted | No cross contamination | Compact size |
| Shell and tube H.E | • | • | | • | • | • | • | • |
| Finned-tube H.E | • | • | | • | • | • | • | • |
| Waste Heat Boiler | • | • | | | • | • | • | |
| Spiral H.E | • | • | | | • | • | • | • |
| Plate H.E | • | • | | • | • | • | • | |
| Run-Around System | • | • | | • | • | • | • | |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Heat Wheel | • | • | • | | • | • | b | |
| Heat Pipe | • | • | | c | • | • | • | • |

a. Can be constructed from corrosive materials but possible extensive damage to equipment from le
b. Cross-contamination can be limited to less than 1% by mass when a purge section is added.
c. Phase equilibrium properties of internal fluid can limit allowable temperature and temperature dif

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