A SIMULATION STUDY ON PERFORMANCE ENHANCEMENT OF TECHNO-ECONOMIC EFFICIENCY OF OCEAN THERMAL ENERGY CONVERSION CYCLE USING DIFFERENT WORKING FLUIDS

NORAZREEN BINTI SAMSURI

UNIVERSITI TEKNOLOGI MALAYSIA

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NORAZREEN BINTI SAMSURI

A thesis submitted in fulfilment of the
requirements for the award of the degree of
Master of Philosophy

Malaysia-Japan International Institute of Technology
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JULY 2017

I declare that this thesis entitled " *A Simulation Study on Performance Enhancement of Techno-Economic Efficiency of Ocean Thermal Energy Conversion Cycle Using Different Working Fluids*" is the result of my own research except as cited in the references. The thesis has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

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Specially dedicated to beloved mother, father, and husband

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ABSTRACT

Ocean Thermal Energy Conversion (OTEC) is a basis to a promising renewable energy technology in terms of its large and inexhaustible available energy reserves, renewability, stability, and cleaner production. The concept of an OTEC power plant is to harness the energy stored in between the upper layer of warm surface seawater as a heat source, and the cold layer of deep seawater as a heat sink, and the plant operates based on a Rankine cycle to generate electricity at a minimum temperature difference of about 20K, between the source and the sink. This energy is sustainable as long as the sun keeps heating the ocean surface to maintain the temperature difference which may drop slightly at night. However, one limitation is that the plant may perform at a maximum ideal Carnot efficiency of about 6.67% only due to relatively low heat source temperature of about 30^{o}C as well as temperature difference lower than 20K. The other limitation is that ammonia, which is a hazardous working fluid due to its high toxicity, is known to perform well in many existing power plants. Since OTEC is free, a bigger energy input will give bigger energy output, a condition at the expense of capital cost. Hence the influence of the types of working fluid and the corresponding operating conditions can be vital and therefore becomes the focus of this study. In addition to investigate the OTEC power plant's thermodynamic efficiencies, this research also explores the economic efficiencies in term of capital cost per net power output ($/kW) using various working fluids to substitute ammonia. The analyzed working fluids, including that of ammonia, are ammonia water mixture (0.9), propane, and refrigerants (R22, R32, R134a, R143a, and R410a). Accordingly, the results showed that ammonia water mixture gave the best performance in terms of heat transfer characteristics with the best thermodynamic efficiency of 4.04% compared to pure ammonia with 3.21%, propane with 3.09%, followed by refrigerants from 3.03% to 3.13%. In terms of capital cost, propane was economically efficient with 15730 $/kW compared to cost of ammonia water mixture at 16201$/kW, refrigerants from 16990 $/kW to 21400 $/kW, and pure ammonia being the costliest at 21700 $/kW. Despite being lower in its thermodynamic efficiency, propane gave the lowest capital cost and had the lowest toxicity compared to all other working fluids. Therefore, propane has the potential to be used as a clean and safe working fluid that would further enhance the OTEC technology in terms of being economically efficient and environmentally friendly.

ABSTRAK

Penukaran Tenaga Terma Lautan (OTEC) adalah asas kepada teknologi tenaga boleh diperbaharui yang menjanjikan kelebihan tenaga yang besar serta berterusan, stabil, dan pengeluaran yang lebih bersih. Konsep loji kuasa OTEC memanfaatkan tenaga yang tersimpan di antara lapisan atas permukaan air laut panas sebagai sumber haba dan lapisan air laut dalam yang sejuk sebagai takat haba rendah, dan beroperasi berdasarkan kitaran Rankine untuk menjana tenaga elektrik pada perbezaan suhu minimum kira-kira 20K. Tenaga ini berkekalan selagi matahari mampu memanaskan permukaan laut untuk mengekalkan perbezaan suhu air laut walaupun terdapat penurunan suhu pada waktu malam. Walau bagaimanapun, had sistem ini adalah pencapaian kecekapan maksima melalui kitaran Carnot yang unggul hanya 6.67% disebabkan oleh sumber haba yang rendah kira-kira 30^{o}C dan perbezaan suhu yang kurang daripada 20K. Seterusnya penggunaan ammonia sebagai bendalir kerja berprestasi tinggi, namun merbahaya kerana mempunyai ketoksikan yang tinggi. Oleh kerana tenaga input OTEC adalah percuma, perolehan tenaga yang lebih besar akan memberikan pengeluaran tenaga yang besar dan bergantung kepada kos modal. Oleh itu, pengaruh jenis bendalir kerja dalam operasi loji ini amat penting dan menjadi tumpuan kajian ini. Selain menyiasat kecekapan termodinamik untuk loji kuasa OTEC, penyelidikan ini juga meneroka kecekapan ekonomi dari segi kos modal setiap pengeluaran kuasa bersih ($/kW) menggunakan pelbagai cecair bekerja untuk menggantikan ammonia. Bendalir bekerja yang dikaji termasuk ammonia, campuran ammonia dan air (0.9), propana, dan bahan pendingin (R22, R32, R134a, R143a, dan R410a). Keputusan yang diperolehi menunjukkan bahawa campuran ammonia air memberikan pemindahan haba dengan kecekapan termodinamik yang terbaik sebanyak 4.04% berbanding dengan kecekapan ammonia tulen dengan 3.21%, propana dengan 3.09%, disusuli dengan bahan pendingin yang lain dari 3.03% hingga 3.13%. Dari segi kos modal pula, propana memberi kecekapan ekonomi yang terbaik dengan 15730$/kW berbanding kos campuran ammonia dan air pada 16201 $/kW, bahan pendingin yang lain dari 16990 $/kW hingga 21400 $/kW, dan pengunaan ammonia tulen adalah termahal iaitu 21700 $/kW. Walaupun propana lebih rendah kecekapan termodinamiknya, ia mampu memberikan kos modal yang paling rendah serta ketoksikan yang rendah berbanding semua bendalir kerja yang dikaji. Secara kesimpulannya, propana mempunyai potensi untuk digunakan sebagai bendalir kerja yang bersih dan selamat dan dapat meningkatkan lagi teknologi OTEC dari segi kecekapan ekonomi dan mesra alam.

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LIST OF ABBREVIATION

OTEC	-	Ocean Thermal Energy Conversion
WSW	-	Warm Sea Water
CSW	-	Cold Sea Water
ODP	-	Ozon Depletion Potential
GWP	-	Global Warming Potential
OC-OTEC	-	Open Cycle of Ocean Thermal Enery Conversion
CC-OTEC	-	Closed Cycle of Ocean Thermal Energy Conversion
LabVIEW	-	Laboratory Virtual Instrument Engineering Workbench
NIST	-	National Institute of Standards and Technologies
PROPATH	-	Program Package Thermophysical properties of fluids

LIST OF SYMBOLS

Q_E	-	Evaporator heat
Q_C	-	Condenser heat
m	-	Fluid mass flow rate
c_p	-	Specific heat of fluid
$c_{p,ws}$	-	Specific heat of warm seawater
$c_{p,cs}$	-	Specific heat of cold seawater
T_i	-	Inlet temperature
T_o	-	Outlet temperature
η_T	-	Turbine efficiency
η_G	-	Generator efficiency
η_{th}	-	Net thermal efficiency
η_G	-	Net efficiency
m_{ws}	-	Mass warm seawater flow rate
m_{cs}	-	Mass cold seawater flow rate
m_{wf}	-	Mass working fluid flow rate
T_{wsi}	-	Warm seawater inlet temperature
T_{wso}	-	Warm seawater outlet temperature
T_{csi}	-	Cold seawater inlet temperature
T_{cso}	-	Cold seawater outlet temperature
U	-	Overall heat transfer coefficient
h	-	Enthalpy
ΔH	-	Pressure head difference
ΔT_{lm}	-	Log mean temperature difference
L_{ws}	-	Warm seawater length pipe
L_{cs}	-	Cold seawater length pipe
W_N	-	Net power
$W_{T\text{-}G}$	-	Turbine generator power

$W_{P,ws}$	-	Warm seawater pumping power
$W_{P,cs}$	-	Cold seawater pumping power
d	-	Inner diameter pipe
g	-	Gravitational acceleration

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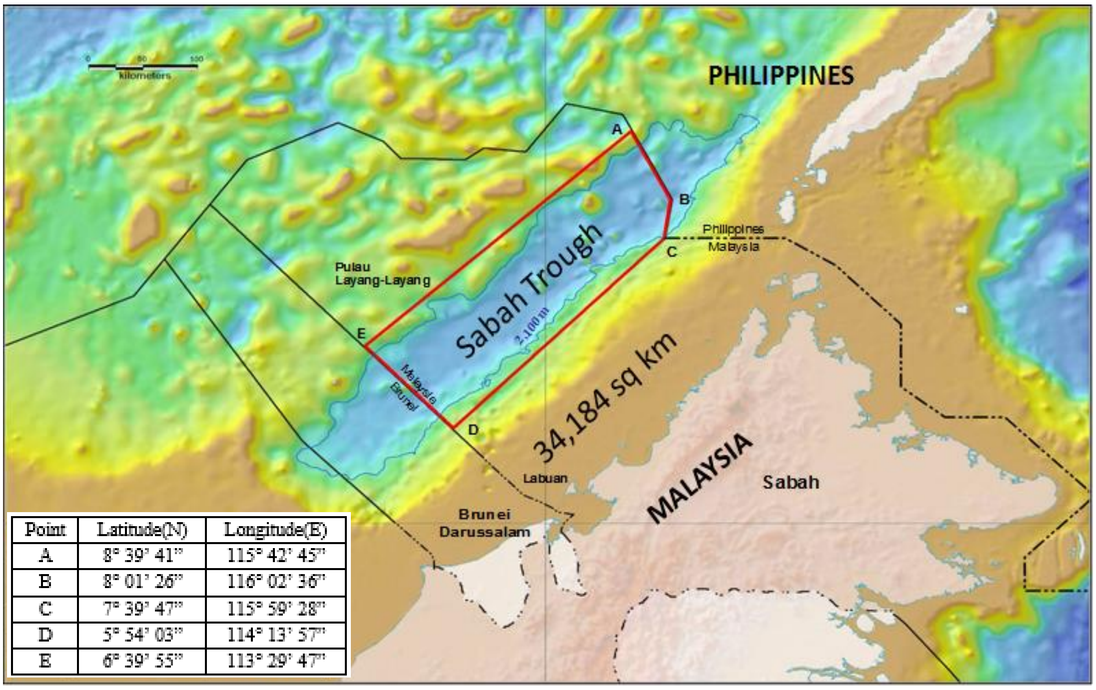
CHAPTER 1

INTRODUCTION

Ocean Thermal Energy Conversion (OTEC) has great potential in an area of deep ocean water where the difference in temperature between the surface water and a certain depth is large enough for an OTEC power plant to operate effectively. The initial concept by Arsonval (1881) stated that the ideal temperature difference required to install an OTEC plant is greater than 20 K. The plant can operate between a heat source (surface seawater at 30 °C) and heat sink (at a seawater depth of 1000 m at 4 °C) (e.g. Uehara, 1979; Vega, 2002; Nihous et al. 2005). The development of the OTEC power plant appears to be based on the open (OC-OTEC) and closed Rankine cycles (CC-OTEC). For optimum performance of the power plant, the operation must be based on the Uehara cycle (Uehara, 1994), with a mixture of ammonia and water being used as the working fluid at 5-6% thermal efficiency and a temperature difference of less than 20 K. However, the use of ammonia-water mixture as the working fluid poses a high health risk (Fuller, 1978).

1.1 Motivation of Study

For a while, Malaysia was not on any global map showing areas with the potential of generating OTEC power. However, thanks to the efforts of the Sapura-Crest Group's Subsidiary, Teknik Lengkap GeoSciences Sdn Bhd in 2008, a marine survey was performed in the South China Sea. This confirmed that the temperature at the bottom of the North-Borneo Trough, known as the Sabah Trough (as shown in Figure 1.1), is about 4 °C when compared to the surface temperature which is at about 29 °C. The above mentioned discovery gave Malaysia the opportunity to successfully operate an OTEC plant. The Sabah Trough is estimated to have a width of 60 km and a length of 100 km, with an average depth of 1000 m. It is essential to perform a fundamental research in order to obtain the best possible performance of OTEC while reducing capital cost.



Point	Latitude(N)	Longitude(E)
A	8° 39' 41"	115° 42' 45"
B	8° 01' 26"	116° 02' 36"
C	7° 39' 47"	115° 59' 28"
D	5° 54' 03"	114° 13' 57"
E	6° 39' 55"	113° 29' 47"

Figure 1.1 The location and area of the Sabah Trough by Sapura-Crest Group's Subsidiary Teknik Lengkap GeoSciences Sdn. Bhd. (2008)

There is still a lack of reliable research data on OTEC technology. This study highlights several new ideas, such as the use of an external heat source to preheat the working fluids and replacing the working fluid with a cheaper and more environment-friendly kind.

1.2 Problem Statement

The first OTEC system was proposed by D'Arsonval (1881) and was based on the closed Rankine cycle, utilizing a low-pressure turbine because of its low operating temperature produced 3% of thermal efficiency (D' Arsonval, 1881). The closed Rankine cycle has been introduced by using an evaporator, a turbine, a condenser and three designated pump for working fluid, SSW and DSW as shown in the Figure 1.2. In 1985, Dr. Kalina has come out with Kalina Cycle which was modified from closed Rankine Cycle with some additional of a separator, an evaporator, an absorber, and a reducing valve. The cycle has improved the OTEC system performance by 4% (Kalina, 1985). Then, Uehara cycle was developed at Saga University in Japan with a 3% higher efficiency than that of a basic closed-cycle OTEC system (Uehara, 1994). Uehara cycle was invented from Kalina Cycle by additional a turbine, a heater, a tank, a pump for working fluid and a diffuser. However, for OTEC carnot efficiency or the ideal cycle for OTEC within temperature 28°C and 5°C is 8.25%. Therefore, the trending has shown that there is a slight increase of the plant efficiency with some expense of plant complexity and plant cost. This thesis will emphasize on the efficiency and also the initial cost by using different working fluids.

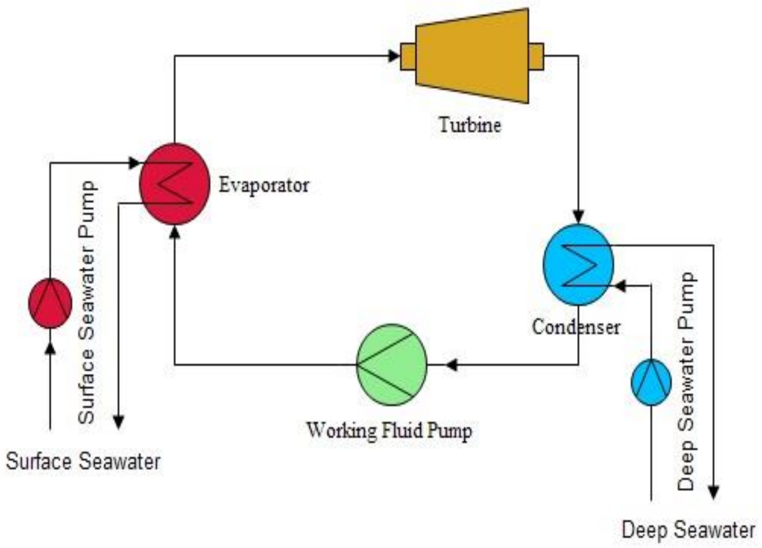


Figure 1.2 Schematic diagram of OTEC Closed Rankine cycle by Arsonval (1881)

Since the energy supplied from the SSW is abundant, this study recommends that preheating be performed by using an outside heat source, as shown in Figure 1.3. A multi-stage turbine, which utilizes an external heat source to perform interstage superheating is proposed to counter the drop in efficiency. However, the multi-stage expansion will increase the cooling requirement of the condenser, thereby increasing the flow rate or power required to pump DSW. Therefore, the proposed closed Rankine cycle OTEC system (as shown in Figure 1.3) uses a closed condenser loop to minimize the possible negative impact on the underwater environment due to the upwelling of DSW. The commercial plant was used titanium for the material of the pipe to overcome biofouling. This modification will also minimize the impact of biofouling on the inside of the tube of the heat exchanger, allowing cheaper materials to be used for manufacturing the condenser. Therefore, this study will encounter the problem on evaporator and condenser as the most expensive components to come out with a closed loop of evaporator and condenser.

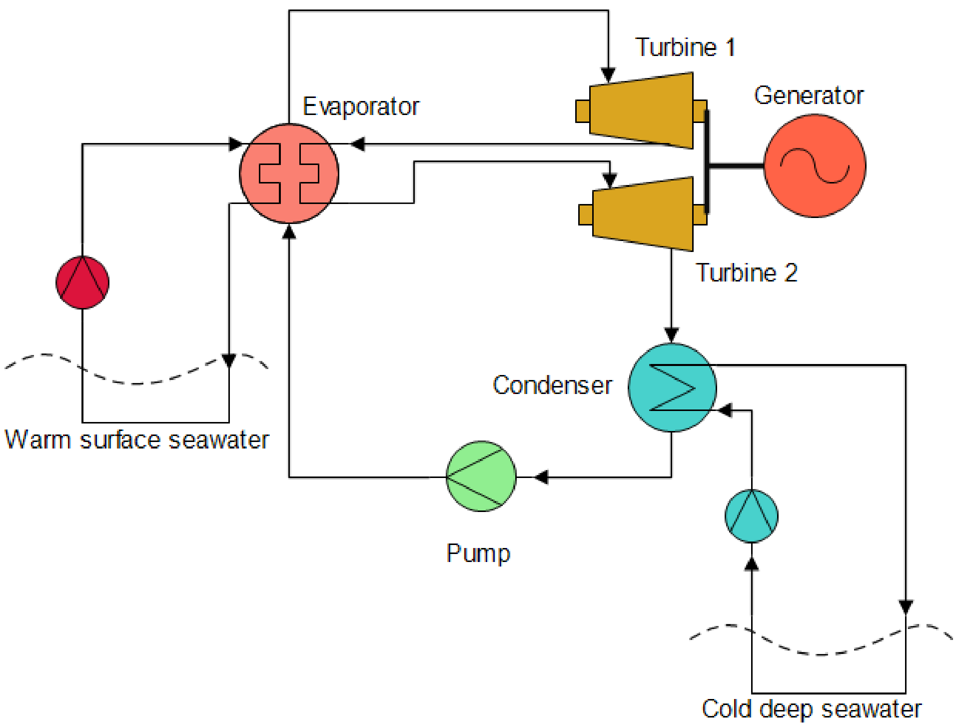


Figure 1.3 Schematic diagram of OTEC Closed Rankine Cycle with modified evaporator heating and condenser cooling systems

Additionally, further research can be carried out to simulate the performance of an OTEC cycle operating on different working fluids. The selection of appropriate working fluids can have a major effect on the overall feasibility and performance of the system. A frequently used working fluid in an OTEC power plant is ammonia (or ammonia-water mixture) because its physical properties are ideal for the OTEC cycle. However, ammonia presents some possibly fatal health risks and hazardous fluid. A vapour leak of up to 1700 ppm dosage could cause respiratory damage in human lungs, and a liquid leak making contact with the eyes or skin could lead to the cold burns (Hoz et al., 1996). More than 50 working fluids have been considered in this study, but some of them were removed because of environmental concerns and their lack of use. Finding other suitable substitutes has been a major obstacle to this research.

1.3 Research Objectives

The main objectives of this research are:

i. To design simulation model the performance of OTEC power cycle;
ii. To improve the performance of a closed rankine cycle by adopting an interstage reheating using warm surface water directly;
iii. To evaluate both thermal and techno-economic efficiencies of OTEC closed rankine cycle using different working fluids

1.4 Research Scope

From the explanation above, the scope of this research lies within the thermodynamic and techno-economic analysis of OTEC cycles because the main focus is on simulating the OTEC cycles with the use of different working fluids. Understanding the characteristics of the new potential working fluids and the effects to the performance of building and operating the Closed Rankine cycle with the proposed new evaporator closed cycle, warm surface sea water heating and condenser closed cycle deep sea water cooling systems developed in the simulation, will assist in getting:

i. An estimate of techno-economic efficiency, which is a more meaningful indicator of the overall performance of the plant;

ii. A significant reduction in the health risk and on the negative environmental effect;

iii. Less expensive and bio-fouling free evaporator and condenser operations.

1.5 Significance of the Study

The importance of this study is hoped to improve the performance as well as the economic efficiency and at the same time the OTEC plant is operated using an environmentally safe working fluid. Furthermore, this study can contribute to the existing literature on OTEC cycles and its working fluid.

1.6 Limitations of the Study

The assumption on simulation of the OTEC cycles is based on steady state conditions by Uehara and Ikegami (1990) and Nihous (2005) in order to derive the appropriate initial values for various working fluids. Furthermore, the selection of working fluids to be tested has been identified by previous studies for OTEC cycle (Ventosa, 2011).

1.7 Research Flow Chart

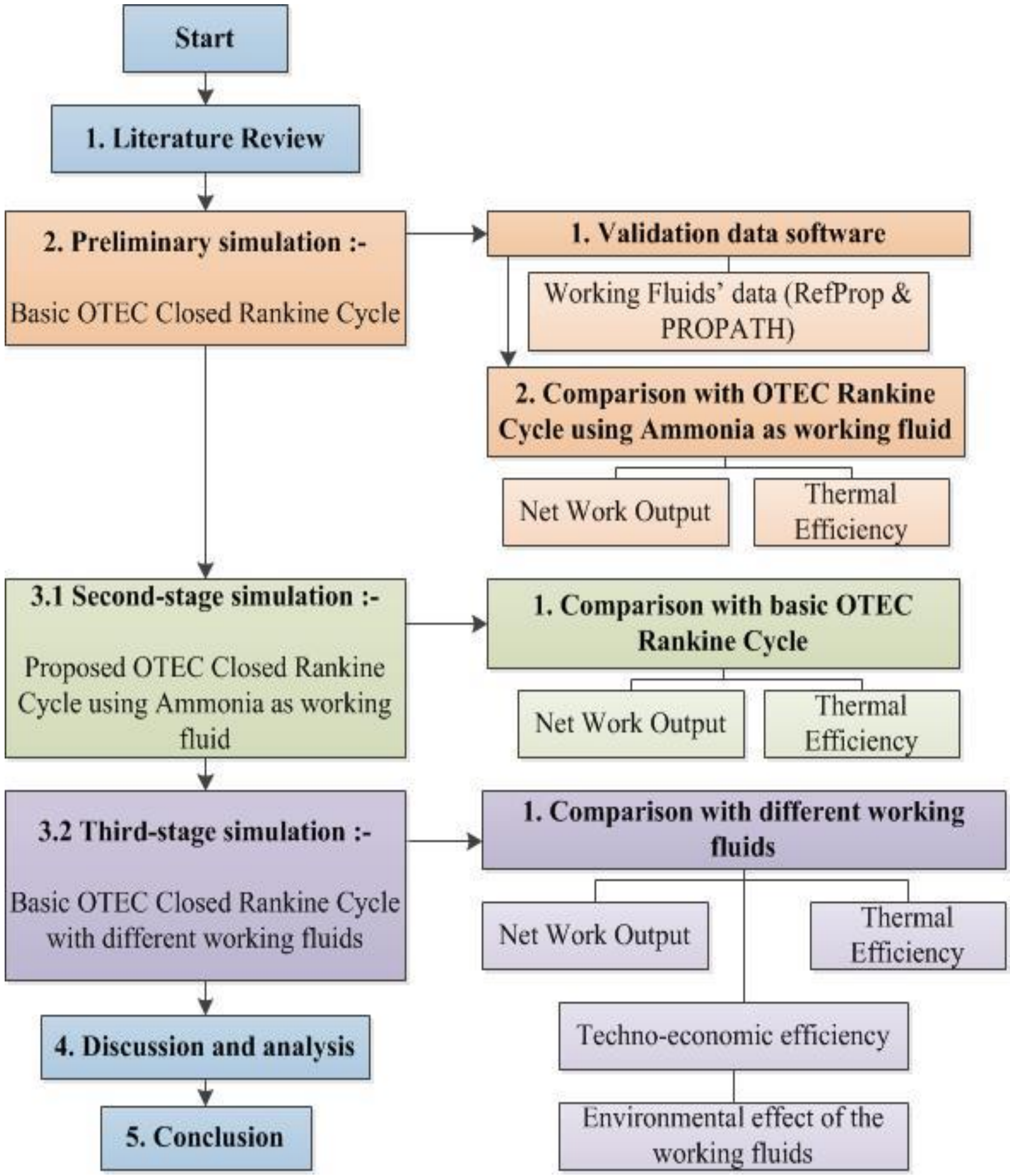


Figure 1.4 Research flow chart

1.8 Structure of Thesis

The rest of the thesis is structured as follows. Literature review on OTEC cycles, working fluids and techno-economic of OTEC plant is summarized and discussed in Chapter 2. In Chapter 3, methodological approaches used in this study are explained. The sub-chapters include validation, analytical techniques of thermodynamic, selection methods of working fluids, etc. are organized for a detailed and complete elaboration. This also includes preliminary results for validation and comparison purposes. In Chapter 4, the results of the second-stage and third-stage simulation are presented. This chapter is presented with comprehensive analysis and discussion which are crucial to the objectives outlined in this study. Lastly, in Chapter 5, the conclusions of the study and recommendation for further works are presented.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

OTEC is a system that converting thermal energy stored in ocean into electricity. The problems with this remarkable renewable energy are low efficiency, higher electricity costs, increased concerns for global warming, and a political commitment to energy security that have made OTEC is attractive to the researchers.

Table 2.1: Overview of OTEC studies since 1979 until 2016

Authors (year)	Target	Cycle	Working Fluids	Achievement
Ganic and Wu (1979)	Working fluids for OTEC	Closed Rankine cycle	Ammonia, Freon-114 and Propane	Ammonia the best fluid
Uehara *et al.* (1998)	Performance analysis of Uehara cycle	Uehara cycle	Ammonia-water mixtures	5.0% of thermal efficiency
Halimic *et al.* (2003)	Alternative refrigerant for Refrigeration cycle	Refrigeration cycle	R12, R401a, R290 and R134a	R134a similar to R12
Yeh *et al.* (2005)	Maximum output of OTEC power plant	Closed Rankine cycle	Ammonia	Optimization of pipe length, pipe diameter and flowrate of seawater

Table 2.1: Overview of OTEC studies since 1979 until 2016 (continued)

Authors (year)	**Target**	**Cycle**	**Working Fluids**	**Achievement**
Ikegami *et al.* (2006)	Experimental performance using Uehara cycle	Uehara cycle	Ammonia-water mixtures	1.5% to 2% thermal efficiency for 5hours of operation
Tong *et al.* (2008)	Performance for OTEC Closed cycle	Closed Rankine cycle	Ammonia, R11, R12, R22, R502	R22 is suitable working fluid
Zhang *et al.* (2012)	Research on the Kalina cycle	Kalina cycle and Closed Rankine cycle	Ammonia-water mixtures	4% of thermal efficiency
Semmari *et al.* (2012)	Novel Carnot-based cycle for OTEC	CAPILI cycle	R21, R114, R123, R152, R152a, R600, R600a, R134a, R134, RC318, R138	1.9% thermal efficiency and R134a is the best fluid
Liu *et al.* (2012)	Performance of GuoHai cycle	GuoHai cycle	Ammonia-water mixtures	5.1% of thermal efficiency
Aydin *et al.* (2015)	Performance analysis of Closed Rankine cycle with solar preheating	Closed Rankine cycle	Ammonia, R22, R32, R134a	The superheating case increases the thermal efficiency 1.9% to 3%
Kulkarni and Joshi (2016)	Performance of working fluids in OTEC technology and different applications	Closed Rankine cycle	Ammonia, R22, R600a, R227ea, R290, R404a, R410a, R744, R1270, R236fa, R245fa, R134a	Ammonia qield the highest efficiency but R744 offered few turbine problem and lowest vapor expension

Table 2.1 shows the overview of OTEC studies from 1979 until the most recent in 2016. It shows that the achievement of the researcher to improve the gap in OTEC technology. Recently, OTEC technology has come out with combination with solar technology (Aydin, 2015). It was proven that the combination is successful on thermal efficiency was increased. However, this study will be focused on improving the OTEC technology itself.

The trend of the research has shown in Figure 2.1 that refers to the percentage of previous research focus on the OTEC topic from 1979 to 2016. From the pie chart, the highest research focus is 19% on environmental concern of the OTEC power plant is mainly regarding the ammonia usage as the working medium. To address the environmental issue, about 16% of the researchers is concern about the selection of working fluids. The second highest of the research focus on the equipment size was 17% and heat exchanger technology was 15%, respectively. If the optimization has been achieved for equipment size and the suitable heat exchanger technology, it can reduce the capital cost to build the OTEC power plant. Apart from that, small amount of research focus on OTEC topics from the pie chart was power generation by OTEC plant, pumping power consumption, manufacturing cost, and the plant improvement. Consequently, this study will emphasize on the OTEC cycle improvement and selection of suitable economic, efficient working fluids towards the OTEC cycle.

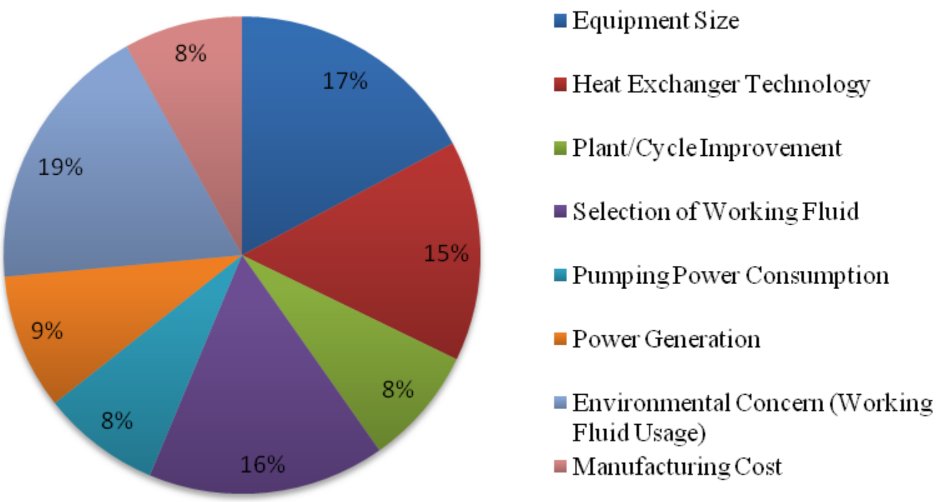


Figure 2.1 The percentage of previous research focus on OTEC topics from 1979 to 2016

In terms of OTEC cycle with different working fluids which is the primary focus of this research, Ganic and Wu (1979) conducted a simple computer model on the effect of three different working fluids (ammonia, propane, and freon-114) on the size of OTEC heat exchangers and system performance. The model using seven different combinations of shell-and-tube heat exchangers are considered to compare with the three fluids. The study had shown that ammonia as the best fluid, although in some cases only by a small margin. Kalina (1982) identified that ammonia-water mixture plays a key role in the Kalina Cycle and a great competitor against the conventional Closed Rankine Cycle. With regard to the new ammonia-water mixture with Kalina Cycle, Uehara (1994) argued with an invented Uehara Cycle by additional absorption and extraction process that allows this system to give 1-2% higher than Kalina Cycle. This new ammonia water mixture greatly improved the efficiency of the power cycle. Based on the idea, Iqbal and Starling (2000) pointed out that mixtures of working fluid gives advantages in OTEC cycles compared to pure-fluid. The efficiency of the Closed Rankine Cycle had been increased from 2.6% to 3.0% and improved performance characteristic under conditions of progressive fouling of heat exchanger. However, ammonia and chlorine can damage the eyes, skin, and can inhibit respiration. The risks are higher if accidents occur in the system involving these chemicals. So, the main question to be answered here is whether there are other suitable working fluids that are safe and economically efficient to replace ammonia using the Closed Rankine cycle, like in this present study.

2.2 Type of OTEC Cycle

OTEC is able to generate power on a continuous of baseload basis as long as the sun keeps heating the SSW. There are three basic modalities of OTEC systems which is Closed Cycle, Open Cycle and Hybrid Cycle. In the closed-cycle, the temperature difference is used to vaporize and condense a working fluid to drive a turbine-generator to produce electricity. In the open-cycle, warm surface water is introduced into a vacuum chamber where it is flash-vaporized. This water vapor drives a turbine-generator to generate electricity. Remaining water vapor (essentially distilled water) is condensed using cold sea water. This condensed water can either return back to the ocean or be collected for the production of potable water. The hybrid-cycle combines characteristics of the closed and the open cycles, and has great potential for applications requiring higher efficiencies for the co-production of energy and potable water. Figure 2.2 shows that the reviews from 1979 to 2016 that investigate specifically about the varieties of OTEC cycle, such as Carnot cycle, CAPILI cycle, Closed-Rankine cycle, Open cycle, Regenerative cycle, Kalina cycle, Uehara cycle, and Guohai cycle.

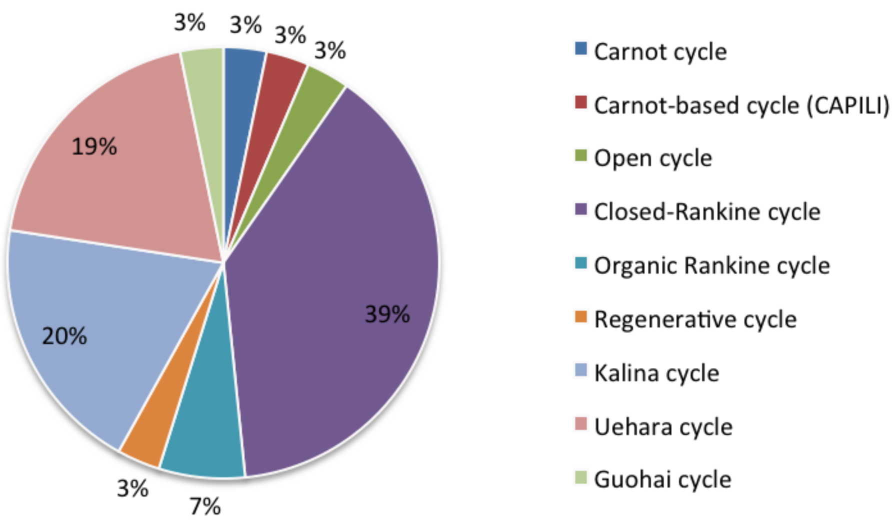


Figure 2.2 The percentage of previous research focus on OTEC cycle from 1979 to 2016

2.2.1 Carnot Cycle

OTEC is a system of converting heat energy stored in parts of ocean into electricity by using the temperature difference between surface water and cold water of about 20K over a depth of 500m to 1000m in the tropical zone. With these operating conditions, the limiting theoretical Carnot energy conversion efficiency of an OTEC plant is 8.6%. The low maximal energy conversion efficiency means that about 92% of the thermal energy extracted from the ocean is wasted and only transits from the warm to the cold heat exchanger of the OTEC plant. Therefore, it is necessary to have a large heat exchanger area and seawater flow rates to produce a relatively small amount of power.

For these reasons, this study will be focused on the improvement of the net efficiency if using other suitable working fluids and reduction of electricity generation costs.

2.2.2 Closed Rankine Cycle

A heat engine of an OTEC plant can take the form of a Closed Rankine cycle, which employs a working fluid that is commonly used in refrigeration application. The working fluid is evaporated at high pressure by warm surface seawater and expanded through a gas turbine that drives a generator. The expanded vapour is then condensed in a condenser by the cold deep seawater as coolant. Finally, a liquid pump brings the condensed ammonia from the low pressure condenser to the high pressure evaporator and the process is repeated. Closed Rankine cycle by Semmari *et al.* (2012) came out with a thermal efficiency of 3.8%.

2.2.3 Open Cycle

An Open Cycle operates similarly to the Closed Rankine cycle except that the warm seawater acts as the working fluid as in Figure 2.3. Warm seawater is flash-evaporated at the evaporator and the resulting vapor drives a low-pressure steam turbine and is condensed by the cold seawater in the condenser. In this variant, the condensate, which is a desalinated water, is a byproduct that can be used as a freshwater source. While direct heat exchangers can be employed in Open Cycle OTEC, which represents the major cost savings compared to Closed Rankine cycle OTEC, this Open Cycle needs to operate under partial vacuum conditions and without noncondensable gas. This specific operation increases parasitic power consumption and results in a lower efficiency. Furthermore, the low density of steam requires very large turbines to produce a significant power level. For these reasons, most OTEC plants operate on the Closed Rankine cycle or its variants.

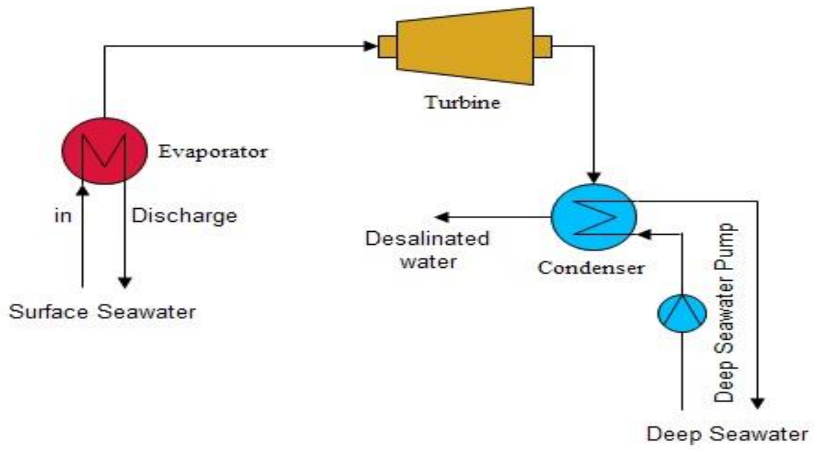


Figure 2.3 Schematic diagram Open Cycle. Adapted from Vega (2002)

2.2.4 CAPILI Cycle

The CAPILI cycle by Semmari *et al.*, (2012) is especially designed so that the compression and expansion can take place without sub-cooling the liquid or superheating the vapor. This is made possible by using an inert liquid so-called "liquid of transfer (LT)", which acts as a liquid piston moving alternately between two insulated cylinders. The two cylinders are connected alternately to the evaporator and the condenser that operate at different pressure levels. The liquid of transfer is characterized by a very low saturation pressure and substantial immiscibility with the working fluid.

2.2.5 Regenerative Cycle

The operation of this cycle by Mago et al. (2008) is similar to that of the Closed-Rankine cycle. However, the only difference is the additional part of an open feed water heater. In this cycle, a part of the steam is extracted from the turbine and goes through an open feed water heater to preheat the liquid refrigerant entering into the evaporator as shown in Figure 2.4.

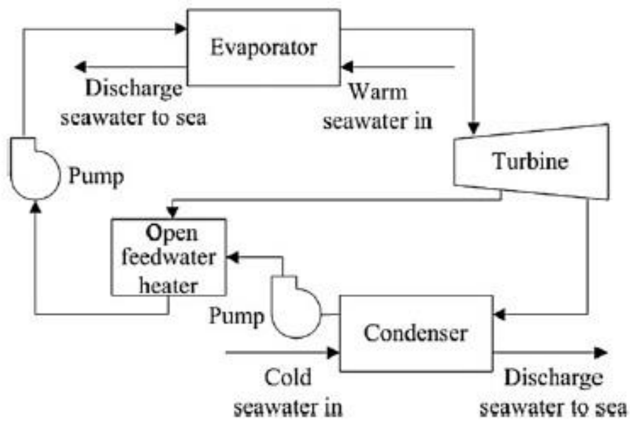


Figure 2.4 Schematic diagram of regenerative cycle by Mago *et al.*, (2008)

2.2.6 Kalina Cycle

The Kalina cycle system is shown in Figure 2.5. The thermal efficiency by using Kalina cycle is 4%, investigated by Zhang *et al.* (2012). Kalina cycle has absorption and distillation equipment added to the cycle. Kalina cycle uses ammonia-water mixture as a working fluid. Compared with Rankine cycle, this cycle system has one more extraction subsystem. This system utilizes the working fluid, which is a mixture of ammonia and water. The mixture passes through an evaporator and some of it is turned into vapor. Next, in the separator, ammonia and water vapor separate from the ammonia solution. The ammonia and water vapor operate a turbine and then flow out from the turbine. Meanwhile, regenerator removes some of the heat from the separated ammonia solution coming out of the separator. The ammonia solution is then mixed again in a mixer with ammonia and water vapor coming out of the turbine and then returns to the condenser. After being condensed by cold seawater, it is pumped through the regenerator and sent to the evaporator again.

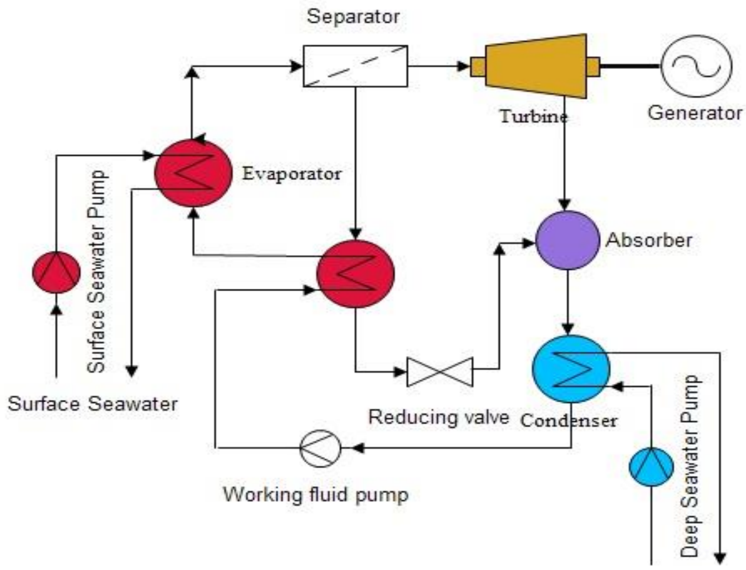


Figure 2.5 Schematic diagram of Kalina cycle. Adapted from Kalina (1984).

2.2.7 Uehara Cycle

The highest of OTEC plant efficiency is produced by Uehara cycle which was developed and invented by Uehara (1994). The improved cycle obtained a thermal efficiency of 6% by using ammonia-water mixtures and 15K temperature difference. The Uehara cycle uses ammonia-water mixture as the working fluid. The cycle extracts heat from the turbine to preheat the condensate that was contributing to plant complexity as in Figure 2.6. This cycle consists of evaporator, pumps, two turbine-generator units or two stage turbine generators, condensers, separators, a regenerator, two heaters, a reducing valve, and an absorber. Operation of this cycle includes producing power and absorption or extraction processes and its thermal efficiency is 1-2% higher than that of the Kalina cycle. In the process of evaporation, the working fluid saturated vapour reduces the working fluid irreversibility in the process of absorption. In the process of condensation, the low content of ammonia and the small temperature difference reduce the irreversibility of condensation, and condense the working fluid completely at low pressure at the same time. The Uehara cycle uses the poor ammonia solution recuperative cycle and the middle recuperative cycle, and reduces the heat loss during the cycle, compared to Rankine cycle of which the theoretical thermal efficiency increases.

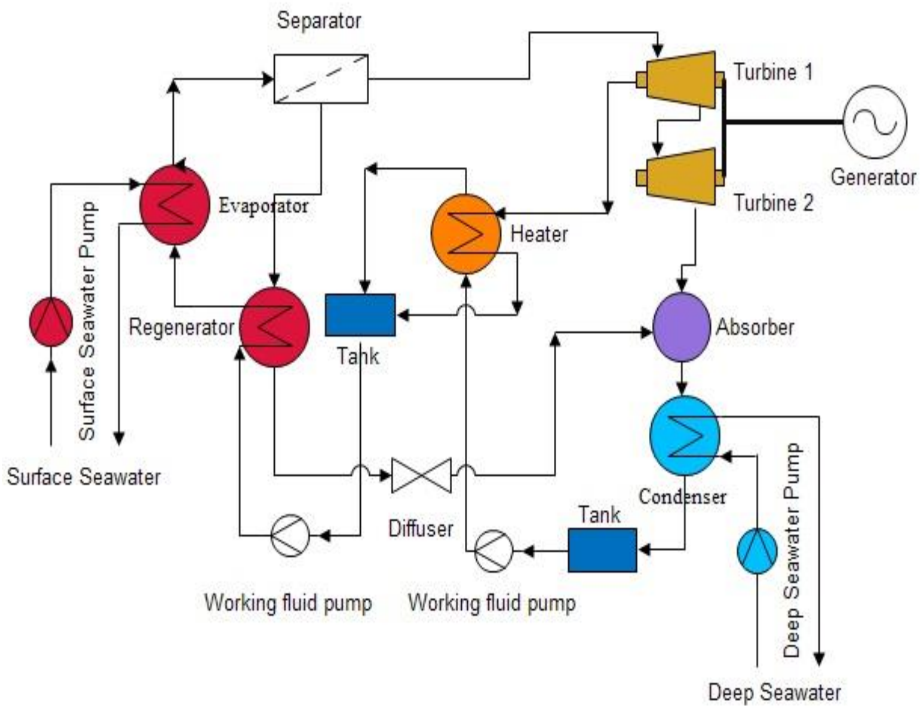


Figure 2.6 Schematic diagram of Uehara cycle by Uehara (1994)

2.1.8 Guohai Cycle

The latest cycle was developed in 2012 by Wei *et al.* (2012) called GuoHai cycle. The GuoHai cycle system is based on Closed-Rankine cycle and Uehara cycle. This new cycle system uses ammonia-water mixture as the working fluid. The GuoHai cycle has a slightly higher thermal efficiency than that of Uehara Cycle but has still not been commercialized yet. GuoHai cycle, as shown in Figure 2.7, is proposed by the First Institute of Oceanography in China. The difference between Guohai cycle and all other OTEC cycles is that when the working fluid through the regenerator heats the basic solution to saturate, the weak ammonia solution from the separator heats the basic solution from the condenser in the regenerator.

This cycle comes out with the improvement on the two regenerators (Regenerator 1 and Regenerator 2), so the basic solution can absorb more heat from the ammonia solution, therefore more ammonia can be saved to do work to improve the thermal efficiency.

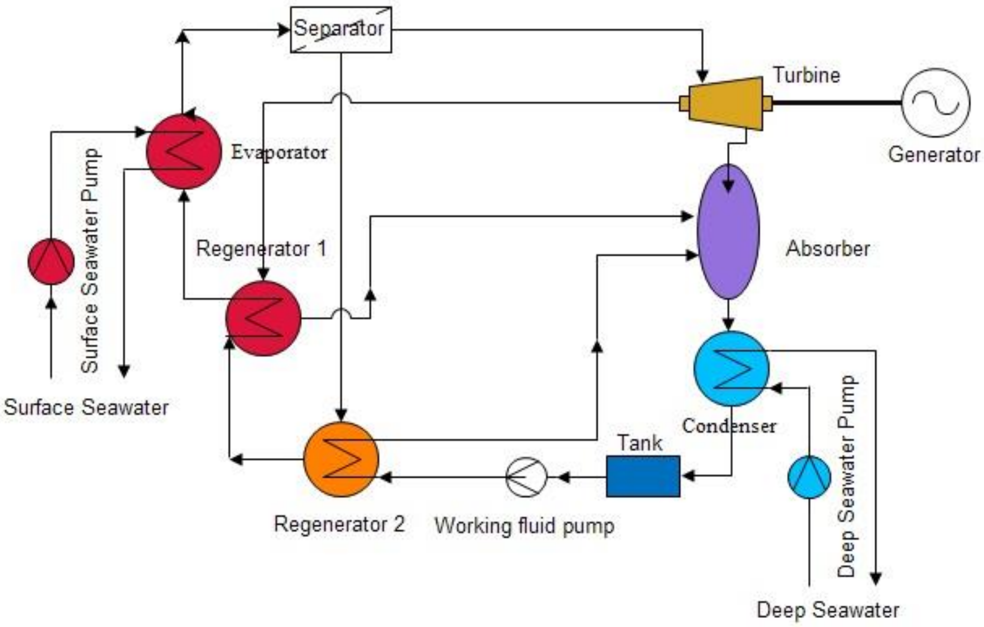


Figure 2.7 Schematic diagram of Guohai cycle by Wei Min Liu *et al.* (2012)

2.3 Working Fluids

The second section of this project is on the selection of working fluids. Ammonia has been considered the best working fluid because it has a suitable boiling temperature (28°C – 32°C) for the OTEC purpose by Ganic and Wu (1979). However, it is toxic and therefore can be hazardous to the environment. Recent development of working fluids shows that Ammonia can be replaced of other working fluids with zero Ozone Depletion Potential (ODP) and zero Global Warming Potential (GWP). Finding other suitable substitutes is a big challenge to the project. A research done by Victoria

(2011) and by Rozi and Henry (2012) have shown better results with the use of other working fluids such as Hydrocarbon (HC) and hydroflourocarbons (HFCs). The other findings are R123 as the best performance but the fluid contributes to the ODP and GWP. Therefore, it is suggested for isopentane as the second best performance and environmental friendly working fluid for the system. However, the applications of their research are for waste heat systems, but can still be operated at low temperature.

The paper that reviews about 35 working fluids and analyzes the influence of fluid properties on the cycle performance is written by Chen et al.(2010). They have categorized the working fluids under three characteristics which are dry, isotropic, or wet fluid depending on the T-s diagram. The selection of working fluid becomes easier when we look into the characteristic of the working fluids. Purvis et al. (1997) has interpreted the data of working fluids into a table with Ozone Depleting Potential (ODP) and Global Warming Potential (GWP) of selected refrigerant. The results obtained for HFC-134a are zero ODP and 1300 GWP with low toxicity and zero flammability. Therefore, the importance of this study is to study other working fluids that have been suggested in the previous research work to be implemented for the OTEC cycle purpose. It is essential to conduct a fundamental research in order to get the best performance of OTEC in a way to reduce capital cost.

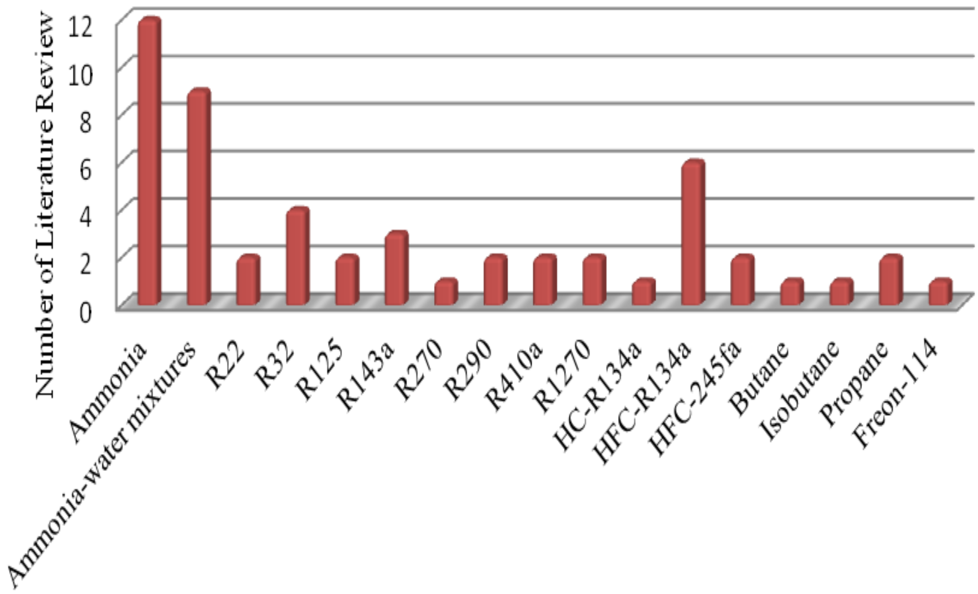


Figure 2.8 The numbers of OTEC previous research focusing on different working fluids from 1979 to 2016

2.4 Techno-economic Efficiency of OTEC

Plant operation for OTEC technology do not use any fuel. Therefore, the major cost component could be construction cost of the infrastructure. According to the findings by Lennard (1987) a 10 MW capacity of floating OTEC power plant would cost 10,000USD/kW. The closed rankine cycle and ammonia as the working fluid was used. A 10MW floating plant had been designed using plate-fin heat exchanger which estimated to consume the highest percentage of cost component of the plant. This research had been stopped because of the exorbitant estimated cost until further development of the design of the demonstration plant until it would be suitable for production. Similarly, Rogers *et. al* (1990) stated that a land based of 1MW OTEC plant would cost 18,000USD/kW. The study conducted to enhance the economic prospect for both open cycle and closed rankine cycle. They highlighted the key importance was the demonstration of technical feasibility of using the OTEC flash evaporation to produce seawater. Thus, from the findings of such ‘by-product’ of fresh water, notably improves the cost effectiveness in producing electrical energy. Subsequently, from the research of ‘Economics of Ocean Thermal Energy Conversion’ by Vega (1992) it was found that for the 10 MW open cycle plant with second stage water production estimated by 14,700USD/kW for the capital cost of the plant. The research continues by Vega (2010) for 100MW floating plant for 7900USD/kW. The capital cost of 53.5 MW is just about 8430 USD/kW for this study. The differential value of the plant cost was calculated by converting to the present day cost using the USA 20-year average for equipment price-index inflation. The estimated cost value also based on the implementation of similar technologies, that later generation design will reach cost reductions of as much as 30% because in the past, the OTEC work did not yield a single order because there were no real customers for the technology. According to the previous literature, the capital cost was calculated based on ammonia as working fluid. Evidently, this study will come out with the comparison for economic efficiency by using different working fluids.

CHAPTER 3

RESEARCH METHODOLOGY

In this chapter, methodological approaches was organized in the following sections which is basics of Laboratory Virtual Instrument Engineering Workbench (LabVIEW), RefProp 9 and PROPATH are introduced in Section 3.1. Then, calibration and validation of fluids' data are explained in Section 3.2 and the details of analytical techniques of thermodynamic are discussed in Section 3.3. The steps on selection of working fluids are provided in Section 3.4. In Section 3.5, preliminary simulation that was conducted to determine the power potential of closed rankine cycle using ammonia as working fluid is explained with comparison with the previous literature study. Last but not least, a summary of this chapter is included in Section 3.6.

3.1 Introduction

LabVIEW is a system designing platform developed by National Instruments which is used for visual programming languages. Its use in various engineering fields (such as electrical, mechanical, aeronautical etc.) has contributed to the development of the biggest and complex applications in the world in order to meet future requirements. LabVIEW provides flexibility to users via intuitive graphical programming that helps in reducing the test development time. A LabVIEW simulation is performed in five steps: requirements gathering, application architecture, development, testing and validation, and utilization, as shown in Figure 3.1 below.

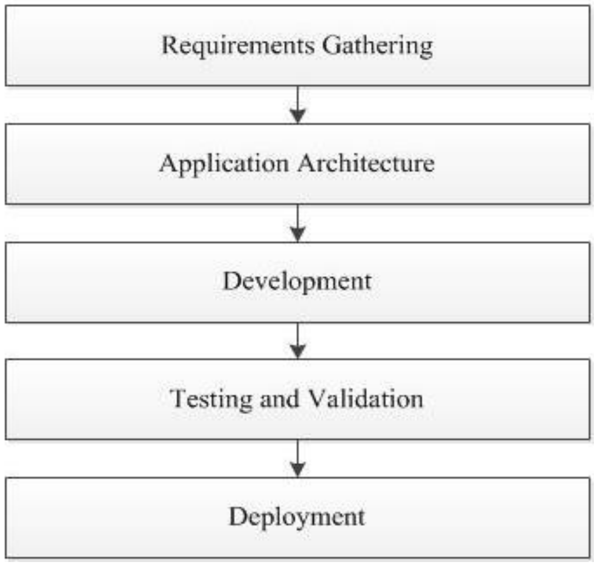


Figure 3.1 The advanced application development with LabVIEW by National Instrument (2014)

The thermodynamic model has been created in LabVIEW and linked to the working fluid data base in NIST RefProp 9 and PROPATH. The thermodynamic model of OTEC cycle is created in LabVIEW to run numerical calculation, simulations and compare the working fluids from a thermophysical perspective.

The source of the fluids data is as follows:

i. For properties of Ammonia, the table of Ammonia by Rogers and Mayhew (1992);
ii. For refrigerants and organic fluids, the National Institute of Standards and Technologies (NIST) RefProp 9 (2013);
iii. For ammonia-water mixtures, the equations of state for Ammonia-Water Mixtures by Ibrahim and Klein (1993) from Program Package for Thermophysical properties of fluids (PROPATH).

3.2 Calibration and Validation of The Working Fluids' Data

An important cornerstone of any methodology is validation and benchmarking. For basis in calculation, the knowledge of thermodynamic properties of the fluids which are enthalpy, enthropy, viscosity, and thermal conductivity is required. In this simulation model, three alternatives for the calculation of the properties of all the fluids has been integrated. Figure 3.2 presents in a ν, T-diagram the comparison of saturations point calculated between the three alternative sets of data. According to the Figure 3.2, the difference between the data amounts to 0.34%, which is less than 3%. The trending line between all the data gives almost the same value. Therefore, the calibration by using either one of the source of data is acceptable. Since the calculation using the table produced by Rogers and Mayhew (1992) is time consuming, hence, the calculation is done using RefProp 9 and PROPATH.

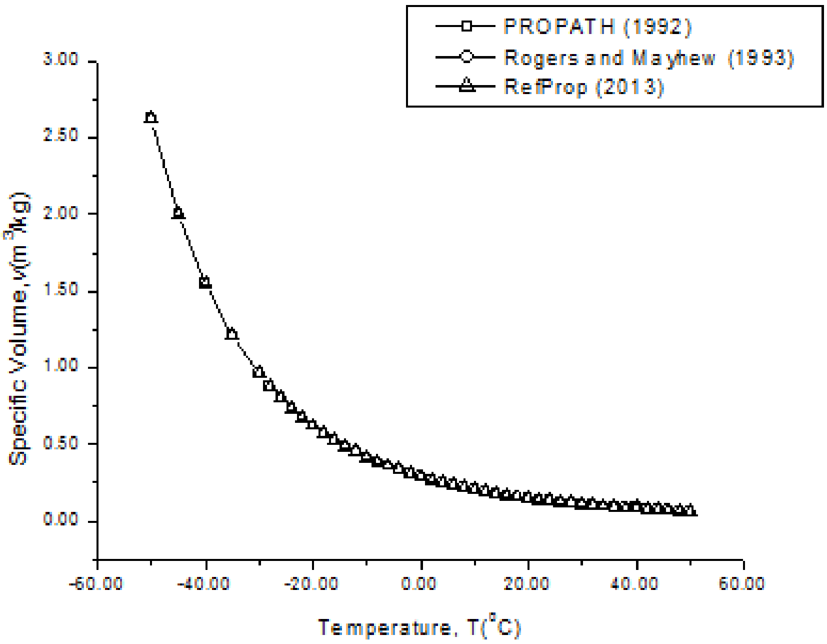


Figure 3.2 Comparison data ν, T-diagram from table by Rogers and Mayhew (1993), PROPATH (1992), and RefProp (2013)

The trending line between all the data is same, the calibration between the source of data is accepted as for the Figure 3.3. Since the calculation using the table produce by Rogers and Mayhew (1992) formulation is time consuming, the following calculation is done using RefProp 9 and PROPATH.

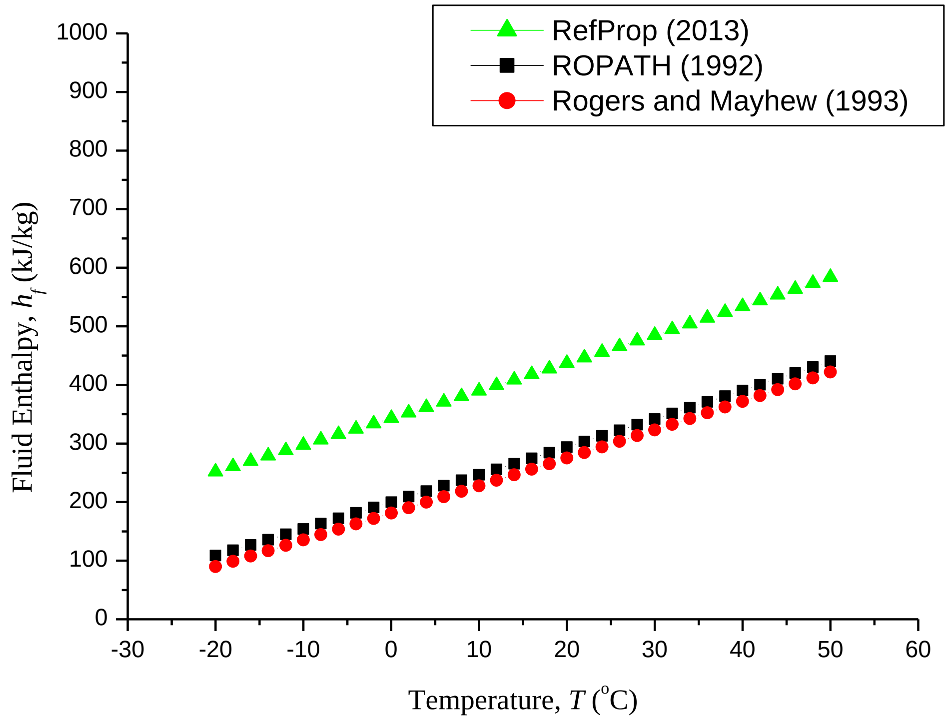


Figure 3.3 Comparison data h_f, T-diagram from table by Rogers and Mayhew (1993), PROPATH (1992), and RefProp (2013)

In this study, PROPATH and RefProp will be both used to find the thermophysical properties of working fluids. Similarity with Morisaki & Ikegami (2012) also used both PROPATH and RefProp for their research on evaluating ammonia, ammonia-water mixture and HFC245fa. Kim *et al.* (2009) conducted a simulation study about the system of OTEC by using condenser effluent from a nuclear power plant using PROPATH for ammonia-water mixture thermophysical properties data. Similarity, Goto *et al.* (2011) conducted their investigation by using PROPATH for binary mixture and came out with the accuracy of a simulation model that was evaluated from the experimental results. According to Karla *et al.* (2012), the RefProp has a database of thermodynamic and thermophysical properties of fluids from 17,000 pure components. Therefore, a benchmark from the previous study will be compared with the outcome later.

3.3 Analytical Techniques of Thermodynamic

The simulation was modelled over the thermodynamic analysis of the performance of OTEC Rankine cycle. The Rankine cycle consists of four main components: evaporator, turbine, condenser and coolant pump. Certain assumptions were made to simplify the analysis and the evaluation of the simulation (Yuan et al., 2014). They are given as follows:

i. Each of the components is in steady state.

ii. Pressure drop and heat losses are neglected.

iii. The system is fully insulated.

iv. Isentropic efficiency is given for all pumps and turbines.

The energy balance equations due to the steady state condition, total energy entering a system equal to total energy leaving the system which is in Equation 3.1

$$E_{in} = E_{out} \tag{3.1}$$

or it can be elaborated as in Equation 3.2

$$W_{in} + Q_{in} + \sum \dot{m}_{in} = W_{out} + Q_{out} + \sum \dot{m}_{out} \tag{3.2}$$

where $\dot{Q}$ is the rate of heat transfer; $\dot{m}_{in}$ and $\dot{m}_{out}$ is mass flow rate, inlet and outlet of the system; W_{in} and W_{out} is work inlet and outlet of the system. By assuming the system is fully insulated and no heat loss; Q_{in} = 0, Q_{out} = 0, and

W_{out}=0; the energy balance in the pump is written as Equation 3.3:

$$W_{in} + \sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{3.3}$$

From Equation (3.3) the work supplied can be expressed as Equation 3.4

$$W_{in} = \sum \dot{m}_{out} - \sum \dot{m}_{in} \tag{3.4}$$

Heat rate supplied to the cycle (evaporator), $\dot{Q}_e$ is given as Equation 3.5

$$\dot{Q}_e = \dot{m}_{wf} \, \Delta h_e \tag{3.5}$$

Heat rate rejected from the cycle (condenser), $\dot{Q}_c$ is calculated as Equation 3.6

$$\dot{Q}_c = \dot{m}_{wf} \, \Delta h_c \tag{3.6}$$

Where h_{out} and h_{in} refers to entalphy at evaporator and condenser. Heat rate absorbed from the warm sea water, $\dot{Q}_{e,ws}$ is shown as in Equation 3.7

$$\dot{Q}_{e,ws} = \dot{m}_{ws} \, c_p \Delta T_{ws} \tag{3.7}$$

Heat rate rejected into the cold seawater, $\dot{Q}_{c,cw}$ is Equation 3.8

$$\dot{Q}_{c,cw} = \dot{m}_{cs} \, c_p \Delta T_{cs} \tag{3.8}$$

where, $\dot{m}_{ws}$ and $\dot{m}_{cs}$ are the mass flow rate of warm and cold sea water, respectively. c_p is the seawater specific heat capacity at constant pressure. The working fluid pump, $W_{P_{wf}}$ and the turbine work, W_T is written as the Equation 3.9 and Equation 3.10

$$W_{P_{wf}} = \dot{m}_{wf} \, v(P_2 - P_1) \tag{3.9}$$

$$W_T = \dot{m}_{wf} \, \Delta h \tag{3.10}$$

where, Δh is the enthalpy difference in the turbine system. By Uehara and Ikegami (1990), the working fluid pumping power, P_{Pwf} in the Equation 3.11; The warm sea water pumping power, P_{ws} ; and the cold sea water pumping power, P_{cs} given as the Equation 3.12 and Equation 3.13

$$P_{Pwf} = \frac{\dot{m}_{wf}\ \Delta H_{wf}\, g}{\eta_{wf,p}} \qquad (3.11)$$

$$P_{ws} = \frac{\dot{m}_{ws}\ \Delta H_{ws}\, g}{\eta_{ws,p}} \qquad (3.12)$$

$$P_{cs} = \frac{\dot{m}_{cs}\ \Delta H_{cs}\, g}{\eta_{cs,p}} \qquad (3.13)$$

where ΔH is the pressure difference. The net power output, P_n is generated as Equation 3.14

$$P_n = P_G - P_{ws} - P_{cs} - P_{p_{wf}} \qquad (3.14)$$

3.4 Selection of Working Fluid

The working fluid plays a key role in the cycle. An optimum working fluid must have the necessary thermophysical properties equivalent to its application and remain chemically stable in the preferred temperature range. The selection of working fluid affects the efficiency, operating conditions, environmental impact and economic viability of the system. The criteria for determining a potential working fluid for the OTEC closed Rankine cycle system at various conditions is explained in this section. Hung (2010) and Sami (2012) have identified the important factors that influence the thermodynamic and thermophysical requirements of this system namely;

boiling point, latent heat, specific heat, thermal conductivity, flash point, boiling temperature, chemical stability, and toxicity, as shown in Figure 3.4.

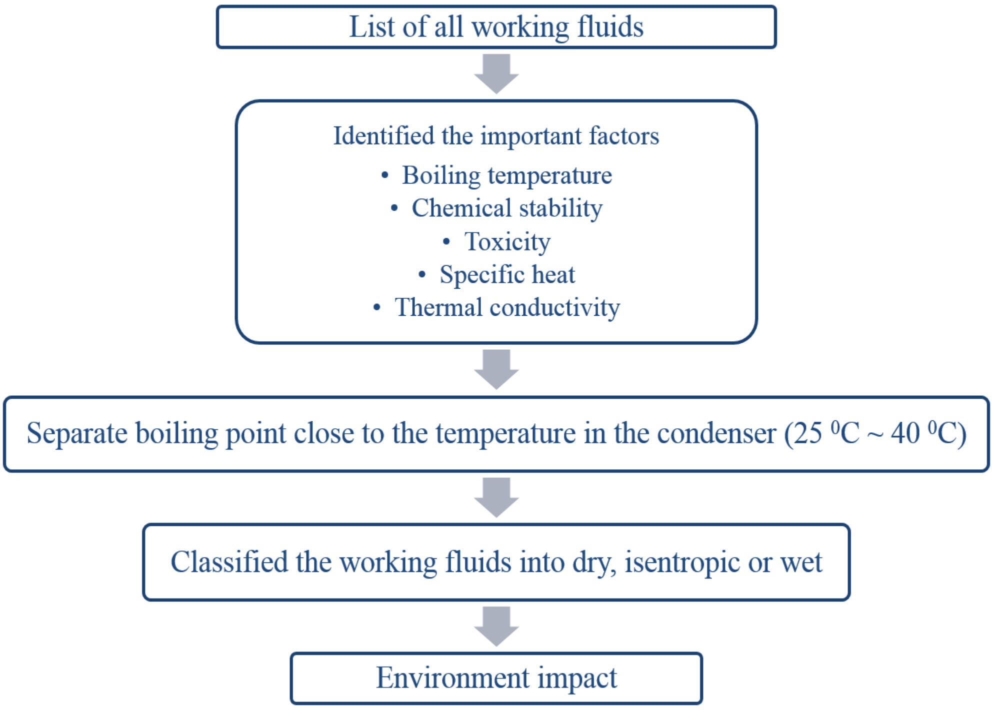


Figure 3.4 Selection of working fluids by Hung et al. (2010) and Sami (2012)

In this case study, the OTEC Closed Rankine cycle used the working fluids boiling point close to the evaporator operating temperature, which approximately 25^{0}C to 40^{0}C (Ventosa, 2010). Additionally, Liu *et al.* (2004) has classified the fluids into dry, isentropic or wet according to the saturation curve (dT/ds) because a dry or isentropic type of fluids is suitable for OTEC Closed Rankine cycle. The comprehension of separating the type of fluids is to make sure the fluids is totally superheated after isentropic expansion, meant to prevent liquid droplets appearing on the turbine blades.

3.4.1 Types of Working Fluids

There are two types of working fluid which can be classified as pure fluid which is pure compound and pseudo-pure fluid. Pseudo-pure fluid means several pure compounds of fluid are mixed. In this test, ammonia, R134a, R143a, Propane, R22 are classified as pure fluid and do not mix with other compound. For ammonia-water mixture, R410a, R470c, R404a, R507a is a pseudo-pure fluid. Figure 3.5 and Figure 3.6 show that the highest enthalpy difference is ammonia-water mixture, followed by pure Ammonia compound, Propane and R410a.

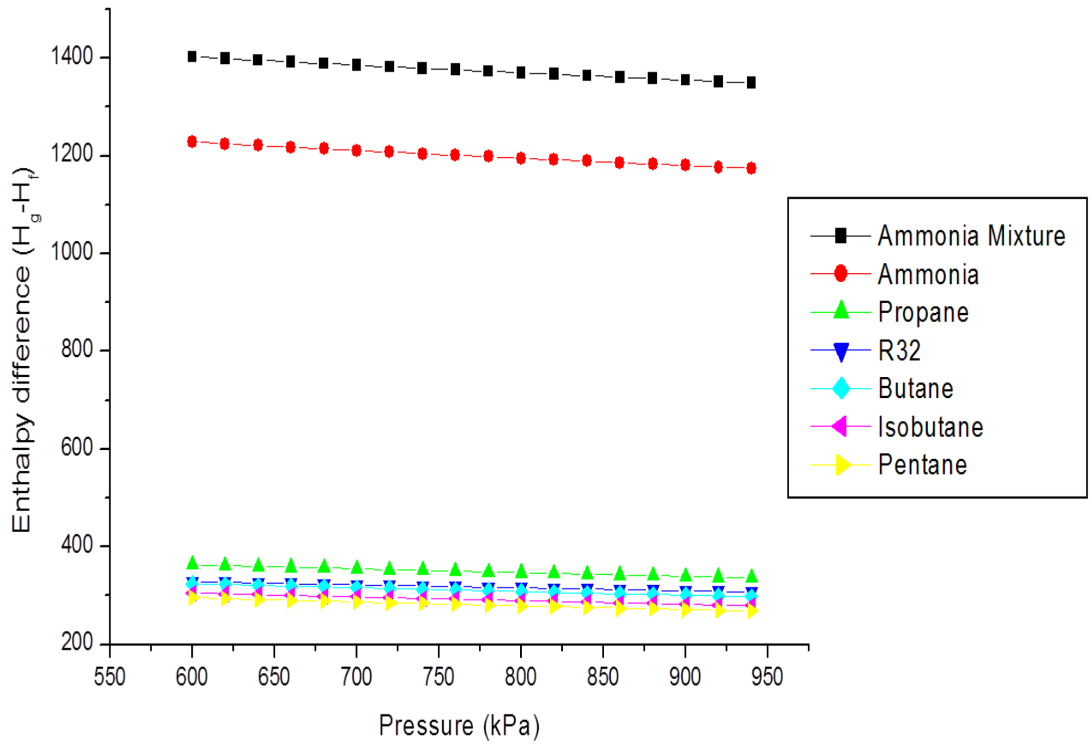


Figure 3.5 Latent heat-pressure diagram of pure fluid and pseudo-pure fluid

Based on this Figure 3.6, the amount of heat added is greater in the case of ammonia-water mixture when compared to other working fluids. This is due to its higher value of latent heat, which is the quantity of heat absorbed by a liquid to remain at a constant temperature or pressure during vaporization.

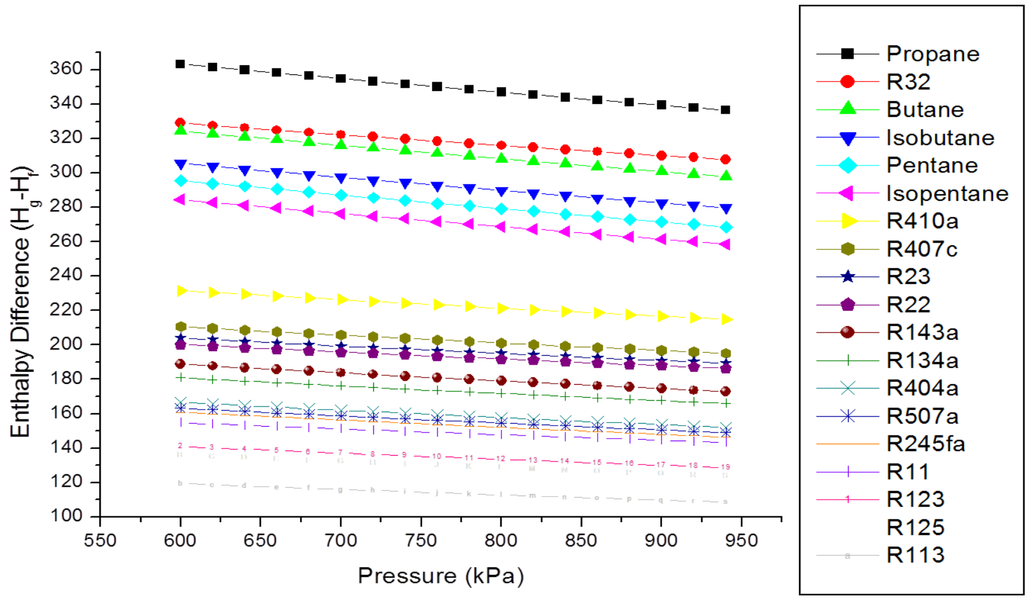


Figure 3.6 Close up Latent heat -pressure diagram of pure fluid and pseudo-pure fluid

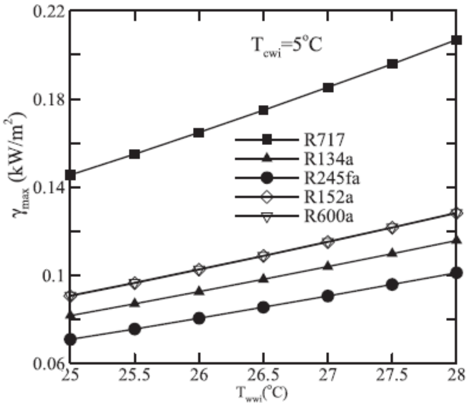
3.5 Preliminary Simulation

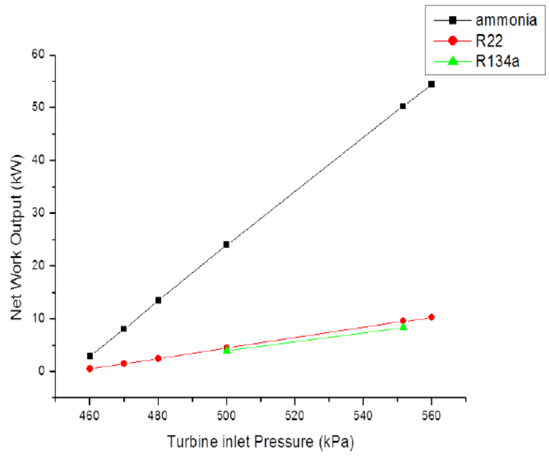
A preliminary design model for simulation of a 1 MWe OTEC Closed Rankine cycle using ammonia as working fluid. This preliminary design is to validate the model developed by Yang and Yeh (2014). Apart from that, the preliminary design model allows to estimate for 5MWe, 10MWe of the OTEC Closed Rankine cycle.

Table 3.1: Fixed condition parameters for three OTEC cycles to be investigated

Variable	Value
Warm seawater inlet temperature, T_{wsw} (^oC)	30
Cold deep seawater inlet temperature, T_{csw} (^oC)	5
Evaporating temperature, T_E (^oC)	28
Condensing temperature, T_C (^oC)	8
Turbine efficiency, η_T (%)	0.82
Generator efficiency, η_G (%)	0.95
Warm seawater pump efficiency, $\eta_{pump,wsw}$ (%)	0.80
Cold deep seawater pump efficiency, $\eta_{pump,csw}$ (%)	0.80
Working fluid pump efficiency, η_{wf} (%)	0.75

The ammonia conditions are in a steady state in the simulation of OTEC Closed Rankine cycle based on Uehara and Ikegami (1990) as shown in Table 3.1. Figure 3.7(b) shows the graphs that quantify the simulated model. When the reference case was compared to the preliminary study, it was discovered that ammonia is the working fluid that produces the highest total work output. These findings are supported by the fact that ammonia has the highest and most suitable latent heat value for the OTEC cycle system.





(a) (b)

Figure 3.7 The net work output of Closed Rankine cycle using several working fluids, where (a) is developed by (Yang & Yeh, 2014); (b) the design model simulation using LabVIEW and RefProp

Table 3.2: Analysis of OTEC Closed Rankine cycle using ammonia as working fluid

	Unit	1MWe	5MWe	10MWe
Q_{in}	kW	19724.40	81375.50	162751.00
Q_{out}	kW	18685.00	76166.90	152334.00
$W_{p(wf)}$	kW	13.22	54.53	109.06
$W_{p(wsw)}$	kW	96.74	399.12	798.24
$W_{p(cws)}$	kW	118.76	484.11	968.21
$\dot{m}_{wf}$	kg/s	15.82	65.25	130.50
$\dot{m}_{WSW}$	kg/s	1793.01	7397.29	14794.60
$\dot{m}_{CSW}$	kg/s	1587.67	6471.91	12943.80
W_T	kW	905.00	4525.00	9050.00
W_{net}	kW	676.28	3587.24	7174.48

This preliminary analysis also serves as an introduction to the visualization methods used in subsequent analysis section in Chapter 4. As can be seen from Table 3.7, the net power output initially increases greatly when scaling up the system (Upshaw, 2012)

3.6 Summary

The method to investigate the efficiency according to different working fluids and improvement of the cycle has been introduced. The methodological approaches applied in this study were mostly based on the OTEC past studies. This chapter comprised of only a part of the capability of LabVIEW and RefProp software. Simulation details and procedures explained in Section 3.1 to Section 3.4 provide a thorough process and concepts, which includes the preliminary result for a first stage of the simulation. On the other hand, the preliminary simulation that was included in Section 3.5 are shown to explain the sufficiency of the parameters used in this study. Following this finding, similar settings were then applied to the second and third stage of the simulation which concerned on the suitable techno-economic working fluid, presented in Chapter 4 with further details shown to explain the sufficiency of the parameters used in this study.

CHAPTER 4

RESULT AND DISCUSSION

In this chapter, result and discussion of the achievements of the objectives of this research are presented. In Section 4.1, the thermodynamic efficiency of the OTEC Closed Rankine Cycle adopts an interstage superheating and a modified condenser cold deep seawater cooling system is described. However, for the intention on the techno-economic topic in this study, the selected working fluid will be tested by using OTEC basic Closed Rankine Cycle. In Section 4.2, the techno-economic efficiency of each of the working fluids is plotted for the purposed of comparison. In Section 4.3, the discussion of significant reduction in the health risk and on the negative environmental effect of the selection of working fluid are presented. Finally, in Section 4.4 is a discussion to highlight the overall findings and objectives that have been achieved.

4.1 The Thermodynamic Efficiency

The basic OTEC Closed Rankine cycle has been modified to produce more output work by introducing two stage steam turbines adopting interstage superheating and a modified condenser cold deep seawater cooling system regardless the cost impact of the cycle. The assumption had been made regarding the working condition is the important factor and these considerations take place to concentrate the analysis on the effect of adopting interstage superheating and a modified condenser cold deep seawater cooling system. The same assumption and method was used for basic and proposed Closed Rankine cycle.

Based on the research on the OTEC closed Rankine cycle, a newly proposed cyclical system with interstage superheating was studied. The process 1 to 2 shows the constant pressure heat addition in the evaporator. The point 2 to 3 is the complete expansion of steam is interrupted in the high pressure turbine and steam is discharged after partial expansion. This reheated steam by using warm seawater is supplied additionally at the point 4 to a low pressure turbine for complete expansion, where the steam pressure approaches the value of the condenser pressure at point 5. The process from 5 to 6 represents the constant pressure heat rejection process in the condenser while 6-1 describes the working fluid pump compressing the working fluid to the operating pressure of the evaporator. The T-s diagram for the modified Rankine cycle in Figure 4.1 shows a slight increase in the area which is created by the introduction of reheating of steam. Thus, reheating increases the work output because of the low-pressure expansion work of the turbine. The main purpose of reheating is to protect the turbine blade by removing the moisture carried by the steam. Using two turbines also provides the advantage of improving the efficiency of the cycle, among others.

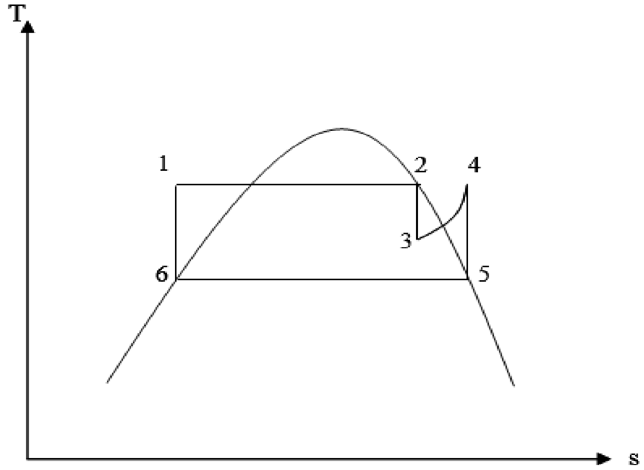


Figure 4.1 T-s diagram of the OTEC closed rankine cycle with interstage superheating and modified condenser cooling system

This happens mainly because a larger fraction of heat flow throughout the cycle occurs at the higher temperature of the working fluid. Moreover, the energy of the exhaust steam and the heat gained from the warm seawater enables the production of a large amount of work. The work output of two turbines is far greater than that of a single turbine. However, the disadvantage of using two stage turbines is the increase in manufacturing and maintenance costs. Hence, further discussion on the result of using different working fluids will focus on the basic OTEC closed Rankine cycle. Moreover, future studies will also emphasize the effect of using other working fluids on the economic efficiency.

Table 4.1: Calculated result of the proposed OTEC Closed Rankine Cycle and the basic OTEC Closed Rankine Cycle uses Ammonia as working fluid

	Unit	Analysis of 1MW OTEC power plant	
		Basic OTEC Closed Rankine Cycle	Proposed OTEC Closed Rankine Cycle
P_E	kPa	1099.30	1099.30
P_C	kPa	573.70	573.70
Q_{in}	kW	19724.40	9237.98
Q_{out}	kW	18685.00	8189.88
$W_{p(wf)}$	kW	13.22	6.17
$W_{p(wsw)}$	kW	96.74	43.80
$W_{p(cws)}$	kW	118.76	52.05
$\dot{m}_{wf}$	kg/s	15.82	7.38
$\dot{m}_{WSW}$	kg/s	1793.01	811.77
$\dot{m}_{CSW}$	kg/s	1587.67	695.89
W_T	kW	905.00	756.58
W_{net}	kW	676.28	737.22
η	%	3.43	7.98

While the output of the simulated model shows a slight increase in the net power output, the efficiency of the cycle is twice of the efficiency of a basic OTEC closed Rankine cycle. Table 4.1 clearly shows that implementing interstage superheating leads to a 4.5% increase in the basic OTEC closed Rankine cycle. Practically speaking, having two or more stages in the Rankine cycle would require significantly smaller turbines (Upshaw, 2012), which would decrease the efficiency. Therefore, the model used in this study overestimates the conversion efficiencies for power output, with an increase intechno-economic efficiency at a higher number of stages. Nevertheless, this analysis proves that the proposed OTEC closed

Rankine cycle can offer the higher work output and efficiency, but it needs additional cost because of using two turbine stages.

Additionally, the system proposed in this study has also created a modified condenser cooling system. Previously, an open-loop was used for the condenser. However, the proposed system uses a closed-loop for the condenser cooling system by using ethylene glycol, which is the common working fluid in the air conditioning industry, as the chiller. Hence, it can be suggested that air conditioning technology is trying to comply with OTEC technology in this case.

The suggested working fluid uses ethylene glycol to act as the medium of heat transfer for cold seawater because it is most commonly used antifreeze fluid for standard cooling applications. Considering a safety factor of 20%, the cooling capacity of the system with ethylene glycol is 9569.88 kW, whereas the cooling capacity of a system using deep seawater is 10471.75 kW.

Therefore, using ethylene glycol in the proposed closed-loop condenser cooling system will decrease the cooling capacity required for the condenser. Nevertheless, the application of the proposed closed-loop condenser cooling system will minimize the impact of biofouling in the condenser tube.

4.2 OTEC Closed Rankine Cycle Using Different Working Fluids

The working fluid is one of the driving factors in this study. The Figure 4.2 shows the simulated result of net power generated by a deep seawater work pump for eight different working fluids. It can be observed that the net power generation is the highest for ammonia-water mixture (740 kW), and the power required to pump deep seawater for the cooling system is also low. The fact that pure ammonia has the second highest value of net power generation is a characteristic that is very well known in the OTEC power cycle. The third highest of the net power is R134a followed by R22 and Propane. R134a is the possible candidate to replace the Ammonia as it is the hisgest net power among the other 5 working fluids but it has the biggest work pump for deep seawater. Therefore, it needs a big pipe to pump from the deep seawater to condense R134a. R22 has the higher net power but lower pumping power for deep seawater compared to Propane. R32 is the fourth possible candidate to replace Ammonia. As the graph shown, R32 gives the lowest pumping power than the other workingfluids including pure Ammonia. The graph also shown that R410a and R143a have low pumping power compared to pure Ammonia, but it has the lowest net power output. In this study, a working fluid must be proposed as a replacement for ammonia because it is harmful to the environment and requires a special material to maintain. Meanwhile, this figure is used for the simulation of performance of the cycle. However, the techno-economic efficiency will discover in Sections 4.3.

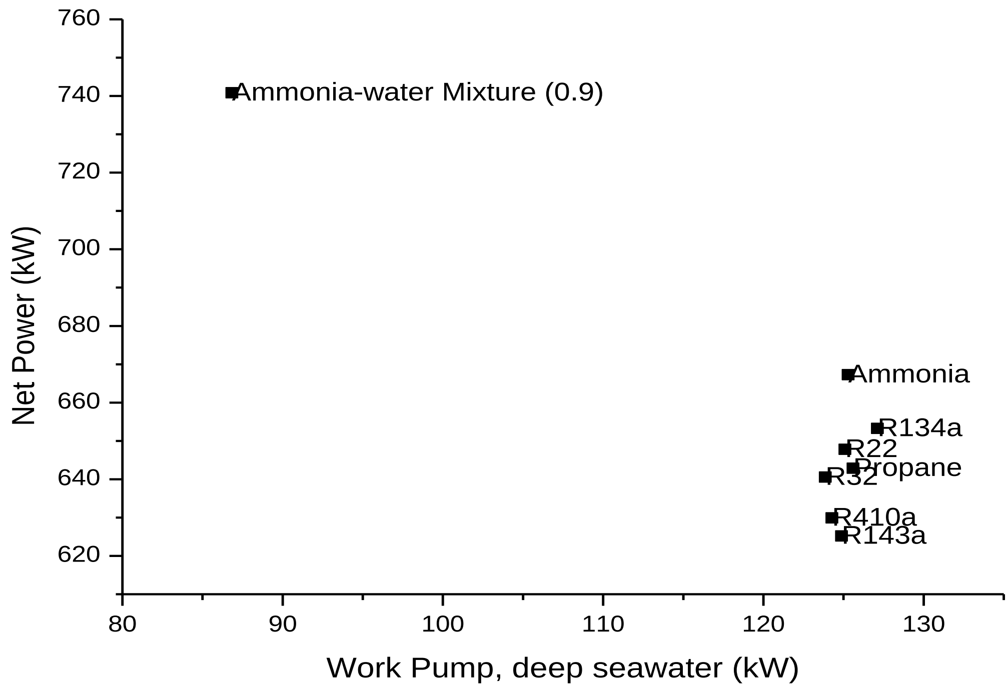


Figure 4.2 The simulated result of net power generated by a deep seawater work pump for eight different working fluids

Figure 4.3 shows the relationship between the net work output and efficiency. Though both ammonia and ammonia-water mixture have higher net work output and efficiency than the other working fluids, they require a separator to ensure the turbine blade is not affected by water vapour from the fluid (especially for ammonia-water mixture). The corresponding efficiency is lower when propane and R32 are used as working fluids. Despite this, both of them possess a relatively wider range of working pressure and a more steady working range when comaped to R410a, R22, R134a, and R143a (Gong, Gao, and Li et al., 2012).

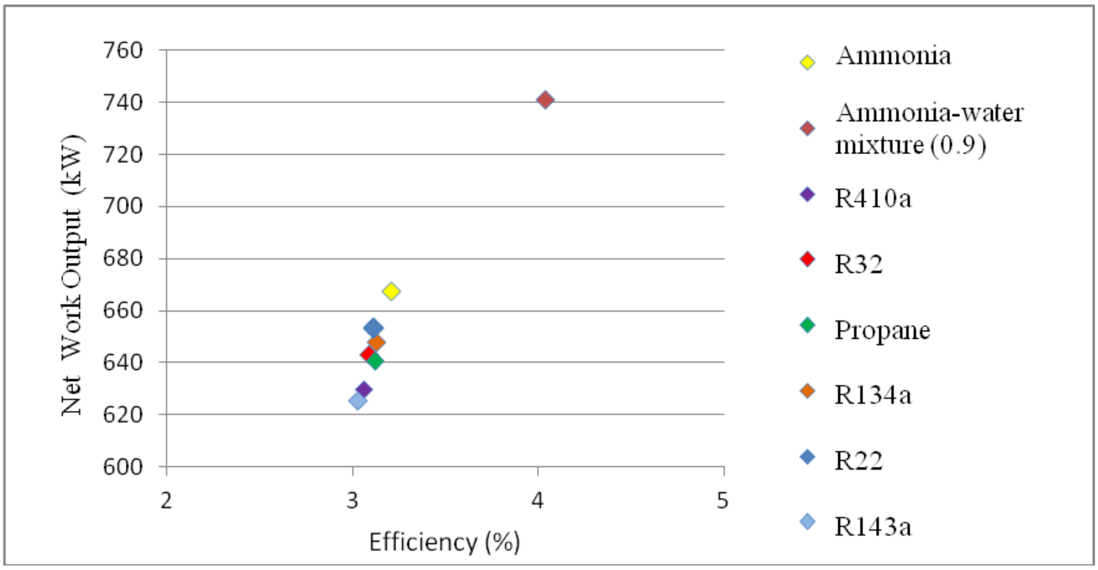


Figure 4.3 The relationship between the net work output and efficiency for eight different working fluids

4.3 The Techno-Economic Efficiency of Different Working Fluid

The evidence from the results of the simulation shows that ammonia-water mixture has a higher value of net power produced and lower cost of main product when compared to ammonia. This still makes ammonia-water mixture the best working fluid in terms of net power despite its low capital cost. It is because of these 3 factors which are; ammonia water mixture has the highest efficiency among the other fluids, a lowest pumping power needed and a lowest capital cost to built an OTEC system. However, the ammonia water mixture needs a proper maintenance on the turbine because of water droplet will be occurred in the turbine. Therefore, recommendation for further study on the maintenance cost of using ammonia water mixture is needed for further improvement. Ammonia has become the second highest of thermal efficiency but the highest capital cost because of its higher pumping power needed and the cost of special material to handle with pure ammonia. Propane has the lowest net cost of main system components when compared to the other working fluids. However, the net power output of propane is still lesser than that of ammonia and ammonia-water mixture, making it the best option to replace them.

There are two types of working fluids that can be classified as pure fluids: a pure compound and a pseudo-pure fluid. In this simulation, ammonia, R134a, R143a, propane, R22, and R32 are classified as pure fluids which do not mix with other compounds. For the ammonia-water mixture, R410a acts as a pseudo-pure fluid, i.e. a mixture of several pure compounds of fluid.

As the result of the simulation at Figure 4.4, clearly displays that propane and R32 are potential fluids that can act as a replacement for ammonia. This is because they have a lower cost and are non-toxic when compared to ammonia. Propane dissolves easily in mineral oils. Additionally, as shown in Table 4.3, propane is a highly flammable fluid.

However, this is not a significant issue in its application in OTEC closed Rankine cycle system since the highest temperature only reaches 40 ^{o}C .

R32 is the second best substitute for ammonia. It has a high value of specific heat at normal boiling point and an instantaneous effect of saturated vapour pressure on temperature. R32 is also characterized by its high productivity at cold temperature and high energy effectiveness, even if it is slightly inferior to R22 and ammonia in this properties. Propane and R32 have the potential to economically satisfy the safety and environmental requirements of the system. They are likely the best options to be used as refrigerants for the OTEC working fluid that will replace ammonia or ammonia-water mixture in the future.

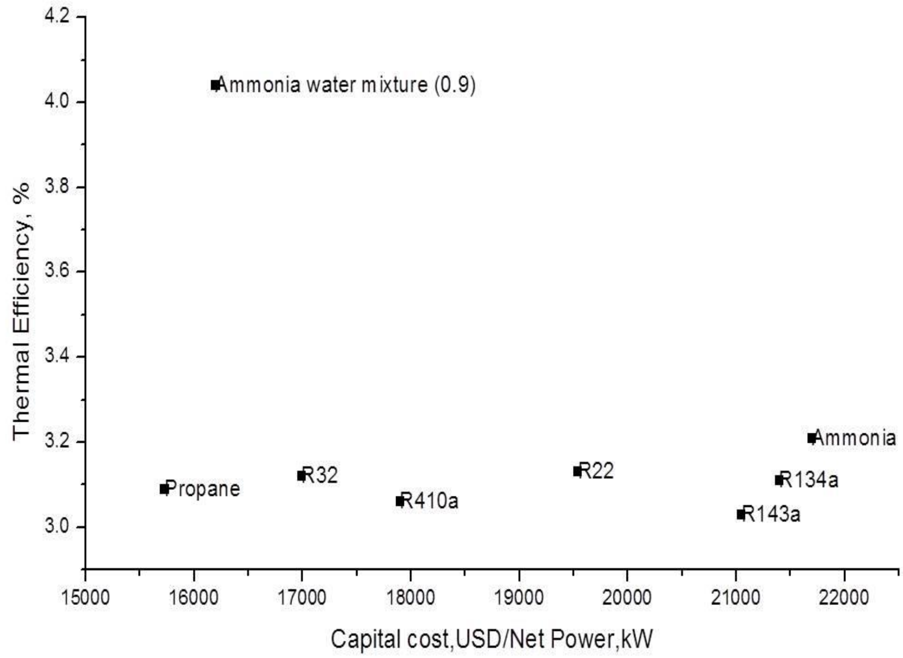


Figure 4.4 The simulated result of thermal efficiency and capital cost (USD/kW) for eight different working fluids

Table 4.2: Calculated result of the different working fluid with the capital cost per net power ($/kW)

Working Fluids	Capital Cost/Net Power ($/kW)
Ammonia	21700
Ammonia-water mixture	16201
R410a	17900
Propane	15730
R22	19540
R32	16990
R134a	20025
R143a	21400

The Table 4.2 is generated from the Figure 4.4 for the exact value of the capital cost per net power output.

4.4 Environmental Criteria

By focusing on the thermodynamic and environmental feasibility, this study has enhanced the important role of working fluid in the OTEC power cycle, as shown in Table 4.3.

Table 4.3: Environment criteria for each working fluid by (Nouman, 2012)

	Flammability	**Toxicity**	**ODP**	**GWP**
Ammonia	Low	High	0	1.00
Ammonia-water mixture (0.9)	Low	High	0	1.00
R410a	Non	Low	0	2340
Propane	High	Low	0	3
R32	Low	Low	0	675
R22	Non	Low	0.055	1900
R134a	Non	Non	0	1300
R143a	Non	Non	0	4470

4.5 Summary

In this Chapter, the study has evaluated the proposed OTEC Closed Rankine cycle which is to improve system efficiency of an OTEC power cycle. It can be seen that for the turbine gross power of proposed OTEC Closed Rankine cycle is decreased because of the amount of the working fluid flowing into the low stage turbine is reduced. With regards to the techno-economic analysis has evaluated using basic OTEC Closed Rankine cycle. As for the working fluid, the Propane and R32 was chosen to replace ammonia as its non-corrosive, lower toxic characteristic and being more economic efficient. R32 has a characteristic of smaller turbine size than ammonia used that can reduce the capital cost(Kim, Ng, & Chun, 2009).

CHAPTER 5

CONCLUSION

The simulation was conducted to investigate the performance of OTEC cycle using different working fluids. At the initial stage of the study, preliminary simulation was conducted to confirm the simulation model with the reference from past OTEC studies. Second stage simulation was then executed for the proposed OTEC Closed Rankine cycle adopting an interstage superheating and modified evaporator warm surface sea water heating and condenser cold deep sea water cooling systems. The third stage of simulation was generated on OTEC Closed Rankine cycle using different working fluids. In Section 5.1, concluding remarks which were based on the analysis in Chapter 4 were presented, and the outlined objectives of this study (in Chapter 1) were revisited to summarize and conclude what has been achieved. Subsequently, recommendations were included in Section 5.2.

5.1 Concluding Remarks

The current state of the OTEC Closed Rankine cycle technology was described. A model which incorporates Labview and Refprop softwares was successfully developed and used for a preliminary evaluation of a performance of an OTEC cycle. The preliminary result of a test run on 1MW net power output shows a close agreement with that of exiting data. The model was later used as a tool to evaluate the performance of the OTEC basic closed rankine cycle using six different working fluids.

The work was also extended to study the performance of a proposed modified OTEC Rankine cycle in which interstage reheating was introduced. The same developed model was used to evaluate the performance of this proposed modified OTEC Rankine cycle. The result shows an increase of thermal cycle efficiency from 3.43% to 7.98%. Therefore, this study has proved that there are improvement on the performance of closed rankine cycle by adopting an interstage reheating using warm surface water.

The result also shows that although none of the other fluids perform as good as that of ammonia water mixture, in term of thermal cycle efficiency, the propane is found to be the best option to substitute it, in term of economic efficiency and safer to the environment.

5.2 Recommendations

There are many working fluids that have interesting thermodynamic properties and need to examine their use in OTEC Rankine cycle with consideration of environmental and safety features. Especially the mixture of working fluids to gain the possible close thermodynamic properties that suitable for OTEC system. For the recommendation future work should also focus on studying the proposed cycle to be evaluated with various working fluids and compared due to its economic efficiency. Additionally, future works can discover other software which is more intelligent, powerful and user friendly such as CyclePad, Cycle-Tempo and HYSYS software. The future work should include practical studies for most interesting working fluids in order to compare the theoretical and practical results. The practical testing is a necessary issue, however there is uncertainty in databases for some working fluids.

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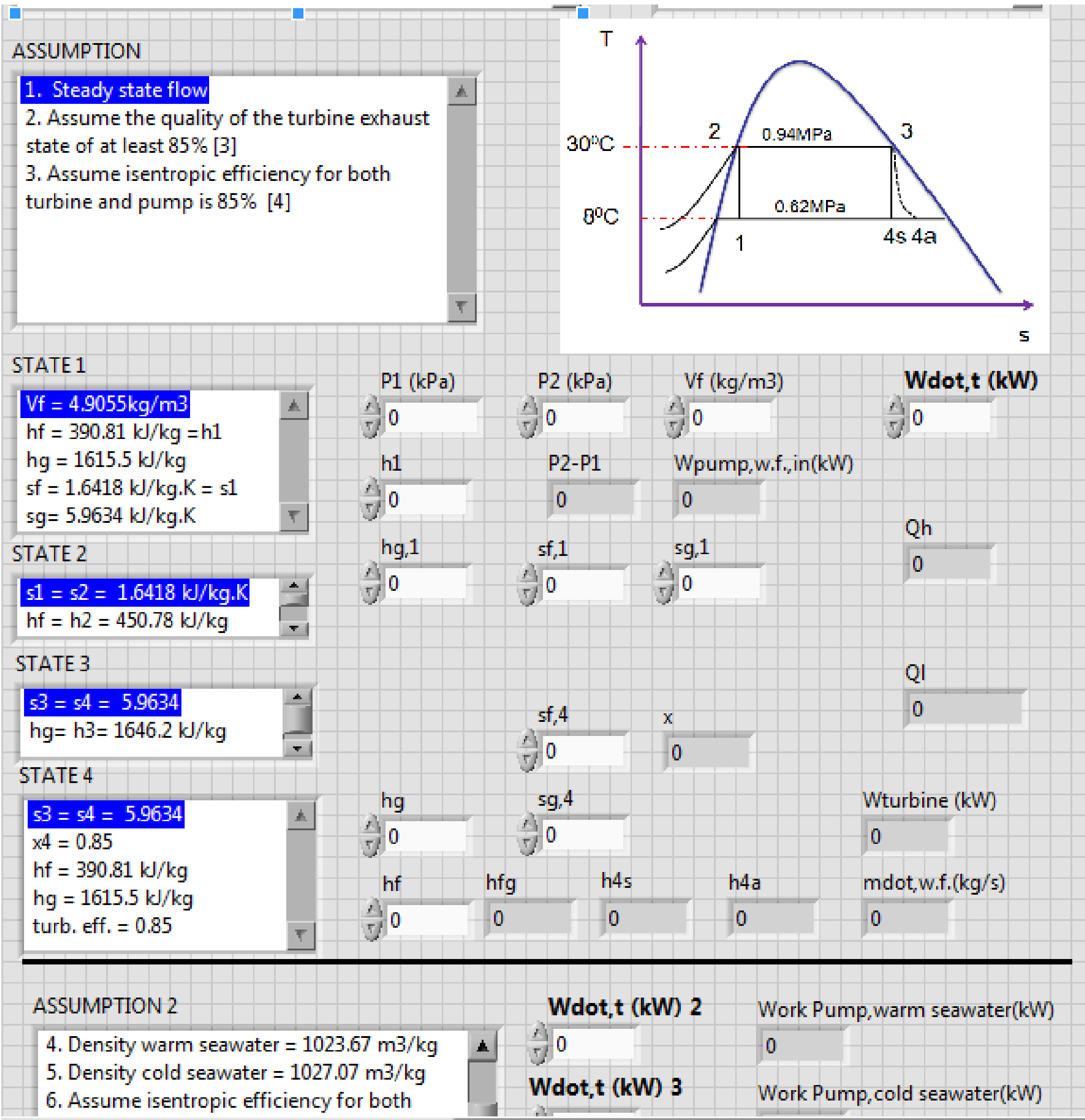
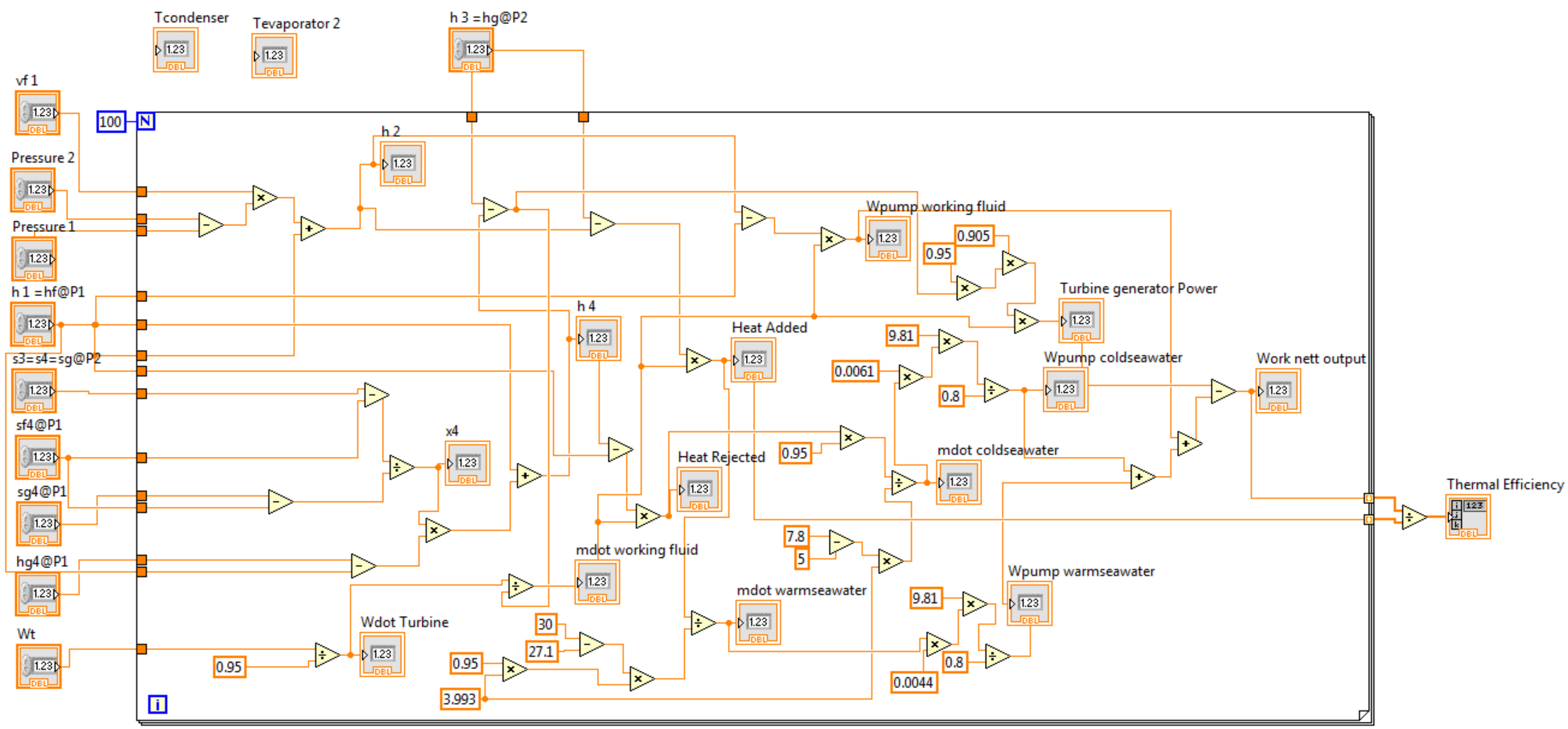
APPENDIX A

OTEC SIMULATION LABVIEW



ASSUMPTION
1. Steady state flow
2. Assume the quality of the turbine exhaust state of at least 85% [3]
3. Assume isentropic efficiency for both turbine and pump is 85% [4]
T
30°C
8°C
2
0.94MPa
3
0.62MPa
1
4s 4a
s
STATE 1
Vf = 4.9055kg/m3
hf = 390.81 kJ/kg = h1
hg = 1615.5 kJ/kg
sf = 1.6418 kJ/kg.K = s1
sg= 5.9634 kJ/kg.K
STATE 2
s1 = s2 = 1.6418 kJ/kg.K
hf = h2 = 450.78 kJ/kg
STATE 3
s3 = s4 = 5.9634
hg= h3= 1646.2 kJ/kg
STATE 4
s3 = s4 = 5.9634
x4 = 0.85
hf = 390.81 kJ/kg
hg = 1615.5 kJ/kg
turb. eff. = 0.85
P1 (kPa) 0
P2 (kPa) 0
Vf (kg/m3) 0
Wdot,t (kW) 0
h1 0
P2-P1 0
Wpump,w.f.,in(kW) 0
Qh 0
hg,1 0
sf,1 0
sg,1 0
Ql 0
sf,4 0
x 0
hg 0
sg,4 0
Wturbine (kW) 0
hf 0
hfg 0
h4s 0
h4a 0
mdot,w.f.(kg/s) 0
ASSUMPTION 2
4. Density warm seawater = 1023.67 m3/kg
5. Density cold seawater = 1027.07 m3/kg
6. Assume isentropic efficiency for both
Wdot,t (kW) 2 0
Wdot,t (kW) 3
Work Pump,warm seawater(kW) 0
Work Pump,cold seawater(kW)


Tcondenser
Tevaporator 2
h 3 =hg@P2
vf 1
Pressure 2
Pressure 1
h 1 =hf@P1
s3=s4=sg@P2
sf4@P1
sg4@P1
hg4@P1
Wt
100
N
h 2
Wpump working fluid
0.905
0.95
Turbine generator Power
h 4
Heat Added
9.81
0.0061
0.8
Wpump coldseawater
Work nett output
x4
0.95
Heat Rejected
mdot coldseawater
Thermal Efficiency
7.8
5
mdot working fluid
mdot warmseawater
Wpump warmseawater
9.81
Wdot Turbine
0.95
30
27.1
0.95
0.8
0.0044
3.993
i

APPENDIX B

LIST OF WORKING FLUIDS

Working Fluid	Molar mass (g/mol)	Boiling Point(°C)
Ammonia	17.03	-33.327
Ammonia-water mixture (90:10)	17.124	
Benzene	78.11	-0.5
Butane	58.122	-0.49
Cyclohexane	84.16	353.9
Ethane	30.07	-88.6
Ethylbenzne	106.2	136
Hexane	86.18	69
Isobutane	52.12	-11.7
Isopentane	72.15	27.7
Octane	114.2	125.52
Pentane	72.15	36
Propane	44.1	-43
O-Xylene	106.2	144.4
m-Xylne	106.2	139
p-Xylene	106.2	138.4
R11	137.4	23.6
R113	187.4	47.6
R114	170.9	3.7
R115	154.5	-38.9
R116	138	-78.3
R12	120.9	-29.8
R123	152.9	27.9
R124	136.5	-12.1
R125	120	-48.3
R134a	102	-26.2
R143a	84.041	-47.241
R152a	66.05	-24.02
R22	86.45	-41.2
R23	70	-82.1
R236	152	6.19
R245fa	134.1	14.9
R32	52.02	-51.7
R401a	94.44	-32.97
R401b	92.8	-34.67
R401c	101	-28.4
R402b	94.71	-47.4
R404a	97.6	-46.45
R407c	86.2	-43.56
R410a	72.59	-51.44
R500	99.31	-33.5
R502	111.6	-45.4
R507a	98.86	-46.74
Toluene	92.14	110.6

APPENDIX C

AMMONIA TABLE

1: ammonia: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Pressure (kPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	429.38	638.57	3.4567	343.15	1605.4	1.4716	6.0926
2	2.0000	462.46	635.82	3.7090	352.42	1607.5	1.5052	6.0667
3	4.0000	497.48	633.06	3.9757	361.71	1609.6	1.5386	6.0410
4	6.0000	534.53	630.27	4.2573	371.02	1611.5	1.5719	6.0158
5	8.0000	573.70	627.46	4.5545	380.36	1613.4	1.6050	5.9908
6	10.000	615.05	624.64	4.8679	389.72	1615.3	1.6380	5.9662
7	12.000	658.66	621.79	5.1983	399.11	1617.0	1.6708	5.9419
8	14.000	704.63	618.93	5.5461	408.52	1618.7	1.7034	5.9179
9	16.000	753.03	616.04	5.9123	417.97	1620.3	1.7359	5.8941
10	18.000	803.95	613.13	6.2975	427.44	1621.9	1.7682	5.8707
11	20.000	857.48	610.20	6.7025	436.94	1623.3	1.8005	5.8475
12	22.000	913.69	607.24	7.1281	446.47	1624.7	1.8326	5.8245
13	24.000	972.68	604.26	7.5751	456.03	1626.0	1.8645	5.8017
14	26.000	1034.5	601.26	8.0443	465.62	1627.2	1.8963	5.7792
15	28.000	1099.3	598.23	8.5368	475.25	1628.3	1.9281	5.7569
16	30.000	1167.2	595.17	9.0533	484.91	1629.3	1.9597	5.7347
17	32.000	1238.2	592.08	9.5950	494.61	1630.3	1.9911	5.7128
18	34.000	1312.4	588.97	10.163	504.34	1631.1	2.0225	5.6910
19	36.000	1390.0	585.82	10.758	514.12	1631.9	2.0538	5.6693
20	38.000	1470.9	582.65	11.381	523.93	1632.5	2.0850	5.6479
21	40.000	1555.4	579.44	12.034	533.79	1633.1	2.1161	5.6265
22	42.000	1643.5	576.20	12.717	543.69	1633.5	2.1472	5.6053
23	44.000	1735.3	572.92	13.432	553.64	1633.9	2.1781	5.5841
24	46.000	1831.0	569.61	14.181	563.63	1634.1	2.2090	5.5631
25	48.000	1930.5	566.25	14.965	573.68	1634.2	2.2398	5.5422
26	50.000	2034.0	562.86	15.785	583.77	1634.2	2.2706	5.5213

APPENDIX D

AMMONIA-WATER MIXTURE TABLE

7: ammonia/water: V/L sat. T=0.0 to 50.0 °C (90/10)

	Temperature (°C)	Liquid Phase Pressure (kPa)	Vapor Phase Pressure (kPa)	Liquid Phase Density (kg/m³)	Vapor Phase Density (kg/m³)	Liquid Phase Enthalpy (kJ/kg)	Vapor Phase Enthalpy (kJ/kg)	Liquid Phase Entropy (kJ/kg-K)	Vapor Phase Entropy (kJ/kg-K)
1	0.00000	387.31	5.3271	680.62	0.040205	230.18	1729.1	1.2773	8.4770
2	2.0000	417.09	6.1339	678.09	0.045964	239.51	1733.1	1.3111	8.4233
3	4.0000	448.61	7.0462	675.54	0.052426	248.85	1737.2	1.3448	8.3705
4	6.0000	481.95	8.0756	672.98	0.059662	258.21	1741.2	1.3783	8.3188
5	8.0000	517.17	9.2345	670.40	0.067749	267.60	1745.2	1.4116	8.2681
6	10.000	554.33	10.537	667.81	0.076768	277.00	1749.2	1.4447	8.2184
7	12.000	593.52	11.996	665.20	0.086806	286.43	1753.2	1.4777	8.1696
8	14.000	634.80	13.630	662.57	0.097957	295.88	1757.2	1.5105	8.1217
9	16.000	678.23	15.454	659.93	0.11032	305.35	1761.2	1.5431	8.0747
10	18.000	723.90	17.487	657.27	0.12401	314.85	1765.1	1.5756	8.0285
11	20.000	771.87	19.749	654.59	0.13912	324.37	1769.1	1.6079	7.9832
12	22.000	822.22	22.261	651.89	0.15580	333.91	1773.1	1.6401	7.9387
13	24.000	875.01	25.045	649.18	0.17416	343.47	1777.1	1.6722	7.8951
14	26.000	930.34	28.127	646.45	0.19433	353.06	1781.0	1.7040	7.8522
15	28.000	988.26	31.531	643.70	0.21648	362.68	1784.9	1.7358	7.8101
16	30.000	1048.9	35.286	640.93	0.24074	372.32	1788.9	1.7674	7.7687
17	32.000	1112.2	39.420	638.14	0.26729	381.99	1792.8	1.7988	7.7280
18	34.000	1178.4	43.966	635.33	0.29628	391.68	1796.7	1.8302	7.6881
19	36.000	1247.4	48.955	632.49	0.32792	401.41	1800.6	1.8614	7.6488
20	38.000	1319.5	54.425	629.64	0.36237	411.16	1804.5	1.8924	7.6102
21	40.000	1394.6	60.411	626.76	0.39985	420.94	1808.4	1.9234	7.5722
22	42.000	1472.8	66.953	623.87	0.44057	430.75	1812.2	1.9542	7.5349
23	44.000	1554.2	74.093	620.94	0.48475	440.59	1816.1	1.9849	7.4982
24	46.000	1638.9	81.876	618.00	0.53262	450.46	1819.9	2.0155	7.4621
25	48.000	1727.0	90.347	615.03	0.58443	460.36	1823.7	2.0460	7.4265
26	50.000	1818.5	99.555	612.03	0.64044	470.30	1827.5	2.0764	7.3916

APPENDIX E

PROPANE TABLE

1: propane: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Pressure (MPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	0.47446	528.59	10.351	200.00	574.87	1.0000	2.3724
2	2.0000	0.50410	525.88	10.975	205.02	577.06	1.0181	2.3703
3	4.0000	0.53510	523.13	11.630	210.06	579.24	1.0362	2.3682
4	6.0000	0.56749	520.36	12.315	215.14	581.41	1.0542	2.3663
5	8.0000	0.60131	517.56	13.032	220.25	583.55	1.0722	2.3644
6	10.000	0.63660	514.73	13.783	225.40	585.67	1.0902	2.3626
7	12.000	0.67340	511.86	14.568	230.57	587.77	1.1082	2.3608
8	14.000	0.71175	508.97	15.388	235.79	589.85	1.1261	2.3592
9	16.000	0.75168	506.03	16.247	241.03	591.91	1.1440	2.3575
10	18.000	0.79324	503.06	17.144	246.32	593.94	1.1620	2.3560
11	20.000	0.83646	500.06	18.082	251.64	595.95	1.1799	2.3544
12	22.000	0.88139	497.01	19.063	256.99	597.93	1.1978	2.3529
13	24.000	0.92807	493.92	20.088	262.39	599.88	1.2157	2.3514
14	26.000	0.97653	490.79	21.160	267.83	601.80	1.2336	2.3500
15	28.000	1.0268	487.62	22.280	273.31	603.68	1.2515	2.3486
16	30.000	1.0790	484.39	23.451	278.83	605.54	1.2695	2.3471
17	32.000	1.1331	481.12	24.675	284.40	607.35	1.2874	2.3457
18	34.000	1.1891	477.79	25.956	290.01	609.13	1.3053	2.3443
19	36.000	1.2472	474.41	27.295	295.68	610.87	1.3233	2.3429
20	38.000	1.3072	470.96	28.697	301.39	612.57	1.3413	2.3414
21	40.000	1.3694	467.46	30.165	307.15	614.21	1.3594	2.3399
22	42.000	1.4337	463.89	31.701	312.96	615.81	1.3774	2.3384
23	44.000	1.5002	460.25	33.312	318.83	617.36	1.3955	2.3368
24	46.000	1.5690	456.54	35.000	324.76	618.86	1.4137	2.3352
25	48.000	1.6400	452.75	36.771	330.75	620.29	1.4319	2.3335
26	50.000	1.7133	448.87	38.630	336.80	621.66	1.4502	2.3317

APPENDIX F

R22 TABLE

3: R22: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Pressure (kPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	497.99	1281.5	21.229	200.00	405.05	1.0000	1.7507
2	2.0000	531.20	1274.7	22.602	202.35	405.78	1.0085	1.7478
3	4.0000	566.05	1267.8	24.044	204.71	406.50	1.0169	1.7450
4	6.0000	602.59	1260.8	25.559	207.09	407.20	1.0254	1.7422
5	8.0000	640.88	1253.8	27.150	209.47	407.89	1.0338	1.7395
6	10.000	680.95	1246.7	28.820	211.87	408.56	1.0422	1.7368
7	12.000	722.86	1239.5	30.572	214.28	409.21	1.0505	1.7341
8	14.000	766.68	1232.2	32.410	216.70	409.85	1.0589	1.7315
9	16.000	812.44	1224.9	34.337	219.14	410.47	1.0672	1.7289
10	18.000	860.20	1217.4	36.358	221.59	411.07	1.0755	1.7263
11	20.000	910.02	1209.9	38.477	224.06	411.66	1.0838	1.7238
12	22.000	961.95	1202.3	40.698	226.54	412.22	1.0921	1.7212
13	24.000	1016.0	1194.6	43.027	229.04	412.77	1.1004	1.7187
14	26.000	1072.4	1186.7	45.467	231.55	413.29	1.1086	1.7162
15	28.000	1130.9	1178.8	48.024	234.08	413.79	1.1169	1.7136
16	30.000	1191.9	1170.7	50.705	236.62	414.26	1.1252	1.7111
17	32.000	1255.2	1162.6	53.515	239.19	414.71	1.1334	1.7086
18	34.000	1321.0	1154.3	56.461	241.77	415.14	1.1417	1.7061
19	36.000	1389.2	1145.8	59.551	244.38	415.54	1.1499	1.7036
20	38.000	1460.1	1137.3	62.792	247.00	415.91	1.1582	1.7010
21	40.000	1533.6	1128.5	66.193	249.65	416.25	1.1665	1.6985
22	42.000	1609.8	1119.6	69.762	252.32	416.55	1.1747	1.6959
23	44.000	1688.7	1110.6	73.511	255.01	416.83	1.1830	1.6933
24	46.000	1770.4	1101.4	77.451	257.73	417.07	1.1913	1.6906
25	48.000	1855.1	1091.9	81.593	260.47	417.27	1.1997	1.6879
26	50.000	1942.7	1082.3	85.952	263.25	417.44	1.2080	1.6852

APPENDIX G

R32 TABLE

2: R32: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Pressure (kPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	813.10	1055.3	22.091	200.00	515.30	1.0000	2.1543
2	2.0000	866.47	1048.3	23.550	203.50	515.65	1.0126	2.1471
3	4.0000	922.45	1041.3	25.090	207.03	515.96	1.0252	2.1399
4	6.0000	981.13	1034.2	26.714	210.58	516.24	1.0377	2.1327
5	8.0000	1042.6	1027.0	28.426	214.15	516.47	1.0503	2.1256
6	10.000	1106.9	1019.7	30.232	217.74	516.66	1.0628	2.1185
7	12.000	1174.2	1012.2	32.137	221.36	516.80	1.0753	2.1114
8	14.000	1244.5	1004.7	34.145	225.01	516.90	1.0878	2.1043
9	16.000	1317.9	997.06	36.264	228.68	516.95	1.1003	2.0972
10	18.000	1394.6	989.28	38.498	232.39	516.95	1.1128	2.0902
11	20.000	1474.6	981.38	40.856	236.12	516.90	1.1253	2.0831
12	22.000	1557.9	973.34	43.344	239.89	516.79	1.1378	2.0760
13	24.000	1644.8	965.16	45.971	243.69	516.62	1.1503	2.0688
14	26.000	1735.3	956.82	48.745	247.53	516.39	1.1629	2.0616
15	28.000	1829.5	948.31	51.676	251.40	516.09	1.1755	2.0544
16	30.000	1927.5	939.62	54.776	255.32	515.72	1.1881	2.0471
17	32.000	2029.4	930.75	58.056	259.28	515.29	1.2007	2.0397
18	34.000	2135.3	921.67	61.530	263.28	514.77	1.2134	2.0322
19	36.000	2245.4	912.37	65.211	267.34	514.17	1.2262	2.0246
20	38.000	2359.7	902.83	69.118	271.45	513.49	1.2391	2.0169
21	40.000	2478.3	893.04	73.268	275.61	512.71	1.2520	2.0091
22	42.000	2601.4	882.96	77.684	279.84	511.82	1.2650	2.0011
23	44.000	2729.2	872.58	82.389	284.13	510.83	1.2781	1.9929
24	46.000	2861.6	861.86	87.412	288.50	509.72	1.2914	1.9845
25	48.000	2998.9	850.77	92.786	292.95	508.48	1.3048	1.9759
26	50.000	3141.2	839.26	98.550	297.49	507.10	1.3183	1.9670

APPENDIX H

R134A TABLE

4: R134a: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Pressure (kPa)	Liquid Density (kg/m^3)	Vapor Density (kg/m^3)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	292.80	1294.8	14.428	200.00	398.60	1.0000	1.7271
2	2.0000	314.62	1288.1	15.465	202.69	399.77	1.0098	1.7260
3	4.0000	337.66	1281.4	16.560	205.40	400.92	1.0195	1.7250
4	6.0000	361.98	1274.7	17.717	208.11	402.06	1.0292	1.7240
5	8.0000	387.61	1267.9	18.938	210.84	403.20	1.0388	1.7230
6	10.000	414.61	1261.0	20.226	213.58	404.32	1.0485	1.7221
7	12.000	443.01	1254.0	21.584	216.33	405.43	1.0581	1.7212
8	14.000	472.88	1246.9	23.015	219.09	406.53	1.0677	1.7204
9	16.000	504.25	1239.8	24.522	221.87	407.61	1.0772	1.7196
10	18.000	537.18	1232.6	26.109	224.66	408.69	1.0867	1.7188
11	20.000	571.71	1225.3	27.780	227.47	409.75	1.0962	1.7180
12	22.000	607.89	1218.0	29.539	230.29	410.79	1.1057	1.7173
13	24.000	645.78	1210.5	31.389	233.12	411.82	1.1152	1.7166
14	26.000	685.43	1202.9	33.335	235.97	412.84	1.1246	1.7159
15	28.000	726.88	1195.2	35.382	238.84	413.84	1.1341	1.7152
16	30.000	770.20	1187.5	37.535	241.72	414.82	1.1435	1.7145
17	32.000	815.43	1179.6	39.799	244.62	415.78	1.1529	1.7138
18	34.000	862.63	1171.6	42.180	247.54	416.72	1.1623	1.7131
19	36.000	911.85	1163.4	44.683	250.48	417.65	1.1717	1.7124
20	38.000	963.15	1155.1	47.316	253.43	418.55	1.1811	1.7118
21	40.000	1016.6	1146.7	50.085	256.41	419.43	1.1905	1.7111
22	42.000	1072.2	1138.2	52.998	259.41	420.28	1.1999	1.7103
23	44.000	1130.1	1129.5	56.064	262.43	421.11	1.2092	1.7096
24	46.000	1190.3	1120.6	59.292	265.47	421.92	1.2186	1.7089
25	48.000	1252.9	1111.5	62.690	268.53	422.69	1.2280	1.7081
26	50.000	1317.9	1102.3	66.272	271.62	423.44	1.2375	1.7072

APPENDIX I

R143A TABLE

5: R143a: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Pressure (kPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	619.67	1024.3	27.306	200.00	387.81	1.0000	1.6876
2	2.0000	659.16	1017.4	29.066	203.00	388.81	1.0108	1.6861
3	4.0000	700.51	1010.5	30.921	206.03	389.79	1.0216	1.6846
4	6.0000	743.78	1003.5	32.875	209.07	390.75	1.0324	1.6832
5	8.0000	789.01	996.32	34.936	212.13	391.68	1.0432	1.6818
6	10.000	836.28	989.06	37.107	215.22	392.60	1.0539	1.6804
7	12.000	885.64	981.68	39.395	218.33	393.48	1.0647	1.6790
8	14.000	937.14	974.17	41.808	221.47	394.35	1.0755	1.6775
9	16.000	990.85	966.53	44.351	224.63	395.18	1.0863	1.6761
10	18.000	1046.8	958.74	47.034	227.81	395.98	1.0970	1.6747
11	20.000	1105.2	950.80	49.864	231.02	396.76	1.1078	1.6732
12	22.000	1165.9	942.70	52.852	234.27	397.50	1.1186	1.6717
13	24.000	1229.0	934.43	56.008	237.54	398.20	1.1295	1.6701
14	26.000	1294.7	925.97	59.343	240.84	398.87	1.1403	1.6685
15	28.000	1363.0	917.31	62.871	244.18	399.49	1.1512	1.6669
16	30.000	1434.0	908.45	66.605	247.56	400.07	1.1621	1.6652
17	32.000	1507.7	899.35	70.561	250.97	400.61	1.1730	1.6634
18	34.000	1584.2	890.00	74.759	254.42	401.09	1.1840	1.6616
19	36.000	1663.6	880.39	79.218	257.91	401.52	1.1951	1.6596
20	38.000	1746.0	870.48	83.962	261.45	401.89	1.2062	1.6575
21	40.000	1831.4	860.25	89.018	265.04	402.19	1.2174	1.6553
22	42.000	1920.0	849.67	94.418	268.68	402.42	1.2286	1.6530
23	44.000	2011.7	838.70	100.20	272.39	402.56	1.2400	1.6505
24	46.000	2106.8	827.30	106.40	276.15	402.62	1.2515	1.6478
25	48.000	2205.3	815.41	113.08	279.98	402.58	1.2631	1.6448
26	50.000	2307.3	802.97	120.31	283.90	402.43	1.2748	1.6416

APPENDIX J

R410A TABLE

6: R410A: V/L sat. T=0.0 to 50.0 °C

	Temperature (°C)	Liquid Pressure (kPa)	Vapor Pressure (kPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)
1	0.00000	800.71	798.08	1169.9	30.575	200.00	421.39	1.0000	1.8106
2	2.0000	852.99	850.21	1161.9	32.603	203.05	421.98	1.0110	1.8067
3	4.0000	907.82	904.87	1153.7	34.744	206.11	422.54	1.0219	1.8029
4	6.0000	965.26	962.14	1145.4	37.005	209.20	423.08	1.0328	1.7991
5	8.0000	1025.4	1022.1	1137.0	39.392	212.31	423.58	1.0437	1.7952
6	10.000	1088.3	1084.8	1128.5	41.911	215.45	424.05	1.0546	1.7914
7	12.000	1154.1	1150.4	1119.7	44.571	218.61	424.48	1.0656	1.7876
8	14.000	1222.8	1218.9	1110.9	47.380	221.79	424.88	1.0765	1.7838
9	16.000	1294.5	1290.4	1101.9	50.348	225.01	425.24	1.0874	1.7800
10	18.000	1369.4	1365.1	1092.7	53.483	228.25	425.56	1.0983	1.7761
11	20.000	1447.5	1442.9	1083.3	56.798	231.52	425.83	1.1093	1.7722
12	22.000	1528.8	1524.1	1073.7	60.303	234.82	426.06	1.1202	1.7683
13	24.000	1613.6	1608.6	1063.9	64.012	238.15	426.24	1.1312	1.7643
14	26.000	1701.8	1696.6	1053.9	67.940	241.53	426.37	1.1423	1.7603
15	28.000	1793.6	1788.2	1043.6	72.103	244.93	426.44	1.1533	1.7561
16	30.000	1889.1	1883.4	1033.1	76.520	248.38	426.46	1.1644	1.7519
17	32.000	1988.4	1982.4	1022.3	81.211	251.87	426.41	1.1756	1.7476
18	34.000	2091.6	2085.4	1011.2	86.199	255.41	426.28	1.1868	1.7432
19	36.000	2198.8	2192.3	999.71	91.511	258.99	426.09	1.1981	1.7387
20	38.000	2310.1	2303.3	987.91	97.179	262.63	425.81	1.2095	1.7340
21	40.000	2425.6	2418.6	975.72	103.24	266.32	425.45	1.2209	1.7291
22	42.000	2545.5	2538.2	963.08	109.73	270.08	424.98	1.2325	1.7241
23	44.000	2669.9	2662.4	949.95	116.70	273.91	424.41	1.2442	1.7188
24	46.000	2798.8	2791.1	936.28	124.21	277.81	423.72	1.2560	1.7133
25	48.000	2932.5	2924.6	921.98	132.33	281.80	422.90	1.2680	1.7075
26	50.000	3071.1	3063.0	906.97	141.15	285.88	421.92	1.2802	1.7013

APPENDIX K

MATERIAL AND WORKING FLUIDS COMPATIBILITY ANALYSIS

No.	Material	Consideration	Working Fluid	Piping Cost (USD/ton)
1.	Copper	• May be used with hot or cold water • Hard copper is used for water applications and is joined by soldering and brazing	R22, R32, R410a	700
2.	Galvanized Malleable Iron/Cast Iron	• Used to transport natural and propane gas	Propane	750
3.	Stainless Steel	• High resistance to corrosion • Can be used to reduce material thickness, weight and cost	R134a & R143a	1500
4.	ASTM A106 (carbon steel pipe)	• May be used in power plants, boilers, oil and gas refineries, and ships • The piping transport fluids and gases that exhibit high pressures and temperatures	Ammonia & Ammonia-water Mixture	1000