HYBRID OCEAN THERMAL ENERGY CONVERSION SYSTEM

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Abstract

Ocean Thermal Energy Conversion (OTEC) systems are based on utilizing the temperature difference between the warm surface water around 27ºC and the cold, 1000 meter deep bottom water at around 5ºC to produce energy. There have been several investigations in the past which focused on stand-alone OTEC systems. The low temperature difference between the surface and bottom waters limits the power output of such systems and make them impractical due to requirement of enormous flow rates for the warm and cold fluids making such applications not feasible. This project focuses on conceptual design of a hybrid OTEC system for a location 39 km off the shores of Jamaica that is augmented with other forms of energy to make it economically feasible. With proper balancing of various energy sources, hybrid power generation can make significant contributions to meet the demand for energy.

Acknowledgement

We would like to thank the Professor Selcuk Guceri for going above and beyond in our project by providing us with the knowledge, support, guidance, enthusiasm, and positivity throughout the process. We would also like to thank Professor Stephen Kmiotek for his expertise in chemical analysis as well as his invaluable knowledge in Aspen Plus. Finally, we would like to thank Jon Van Blarcum for creating the foundation needed for this project to be successful.

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Introduction

Ocean Thermal Energy Conversion (OTEC) has been previously investigated for decades since its possibility to improve clean and renewable energy appeals to researchers. An OTEC system utilizes the temperature difference between the warm surface ocean water and the cold deep ocean water to produce power. This temperature difference in addition to a working fluid that has a low boiling temperature is typically ran in a Rankine Cycle. A Rankine Cycle is composed of a pump, boiler (evaporator), turbine and condenser. These four components allow the working fluid to phases throughout the cycle, which maximize the power produced, since there are changing in enthalpy at each state. By establishing a system in the Tropics or Subtropics where annually the change in ocean temperature is consistently 20C and above, various direct and indirect benefits could be obtained.

In this paper, the OTEC system is modelled 39 km off the shores of Montego Bay, Jamaica, where annually the difference in surface water and deep water temperatures remains in the range of 22ºC-24ºC. A country like Jamaica would benefit greatly from this system, since it will have a positive economic, social, and environmental impact.

However, the low thermal efficiency of a pure OTEC system does not make it a feasible investment. As a result, the use of auxiliary heat as provided to be a solution to this disadvantage. The combination of OTEC with another renewable energy source would create a hybrid cycle. This cycle has been proven to increase the thermal effect thus improving the power production. In this paper, the use of a hybrid cycle with ammonia as a working fluid would be investigated to reach an optimal temperature that the working fluid should be increase to using the auxiliary energy source.

Background

Carnot Cycle

The Carnot cycle was proposed by a French physicist, Sadi Carnot in 1824, which have later been improved on by several scientists [16]. This was a theoretical construct and was used to describe the highest limit in the efficiency of a classical thermodynamic engine when energy is converted. The Carnot cycle is described as having reversibility that absorbs high temperature heat from the boiler and releases the low temperature to the condenser as seen in figure 1 [1]. The processes of the Carnot cycle are described below [2].

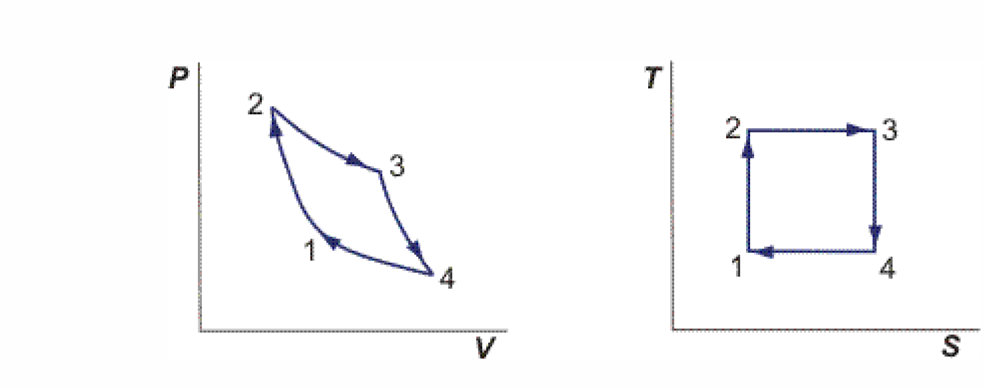


Figure 1: P-V and T-S Diagrams [6]

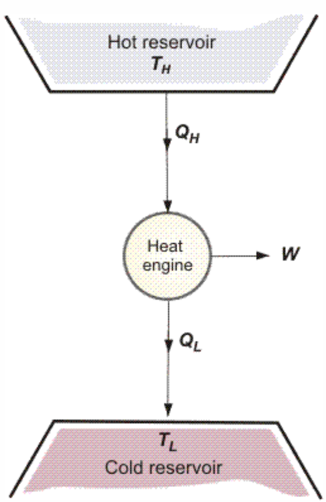


Figure 2: Carnot cycle [1]

- 1-2 Isothermal expansion. Heat is absorbed where gas is expanded reversibly and hot temperature is constant.
- 2-3 Adiabatic expansion. Insulated engine with no heat loss, where the gas expands slowly until the temperature is converted from hot temperature to cold temperature.
- 3-4 Gas is compressed when it reduces to the cold temperature. An ideal gas will have constant temperature so no change in internal energy.
- 4-1 Insulated engine with no heat loss, where gas compresses slowly until the temperature is converted from cold temperature to hot temperature. Cycle is complete

The efficiency of the Carnot cycle does not consider external factors and only the highest temperature and lowest temperature are considered. To increase the efficiency of the cycle, the parameters that can be manipulated are to increase the highest temperature or to decrease the lowest temperature, with the efficiency always being smaller than 1 [2].

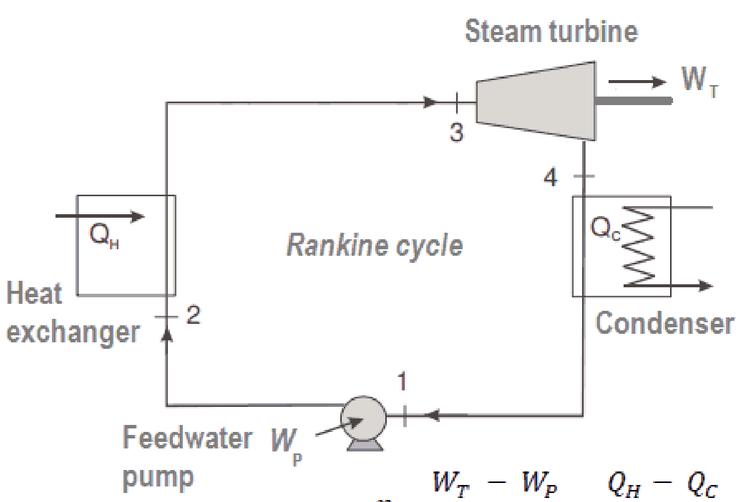
For a Carnot cycle, the thermal efficiency is as follows:

- Irreversible engine $\eta_{th} < \eta_{th,carnot}$
- Reversible engine $\eta_{th} = \eta_{th,carnot}$
- Unrealistic engine $\eta_{th} > \eta_{th,carnot}$

Rankine Cycle

The Rankine cycle is a fundamental power plant that predicts the performance of steam turbine systems. It was named after a Scottish polymath and Glasgow University professor, William John Macquorn Rankine [7]. Rankine developed an idealized thermodynamic cycle utilizing an operating fluid that is continuously evaporated and condensed within the system to convert heat to mechanical work. It describes the process in which steam engines generate power. The power output of a Rankine cycle depends on the temperature difference between the hot source and the cold source. Carnot's theorem proved that a higher temperature difference is need to produce more mechanical power to be efficiently extracted.

The most common Rankine cycles are operated with steam, which typically runs using coal, liquid fuel, biomass and solar thermal power plants. Water is typically the working fluid of choice since it has favorable properties such as its non-toxic and unreactive chemistry, abundance, and low cost as well as its thermodynamic properties [6]. Heat is supplied to raise the temperature of the water to produce steam in a closed loop cycle. The Rankine cycle has four basic components as shown in figure 3. These include the pump, boiler, turbine, and condenser. Each components of the cycle are operated at steady state and with control volume. The figure below shows the schematics of a Rankine cycle.



$$\eta = \frac{W_T - W_P}{Q_H} = \frac{Q_H - Q_C}{Q_H}$$

Figure 3: Schematic of a Rankine Cycle [6]

The Rankine cycle undergoes four processes in series: two isentropic processes, which are alternated with two isobaric processes. The working principle of a basic Rankine cycle is described below.

- 1-2 Isentropic pressurization to compressed liquid
- 2-3 Liquid is converted to saturated steam with no pressure change
- 3-4 Saturated vapor is expanded inside the steam turbine, which causes it to rotate, producing mechanical energy. This energy is converted into electricity by the generator. The pressure and temperature of the steam is decreased so that the fluid inside the turbine is approximately entirely gas. No change in entropy.
- 4-1 Isobaric condensation process

For an ideal case, 1-2 and 3-4 are the isentropic processes. Figure 4 shows the graph of temperature vs entropy in a Rankine cycle.

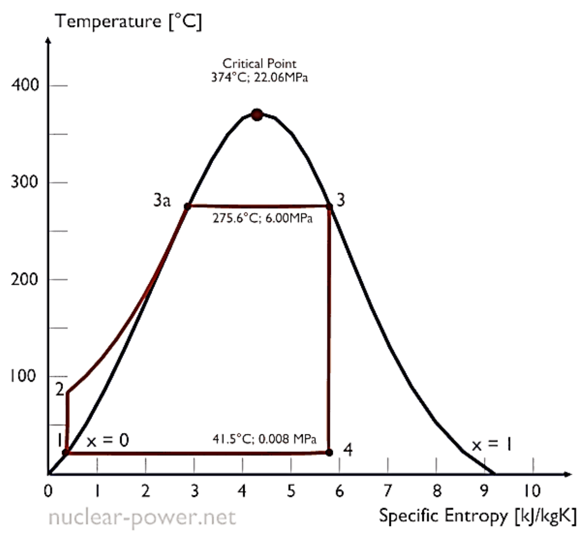


Figure 4: Temperature vs Entropy [6]

The efficiency of Rankine cycles is described to be close to that of the Carnot cycle. However, the phases of the Rankine cycle are correlated with irreversible processes, which means that the overall efficiency is decreased. Even with high temperature in the boiler, only 40% of the fuel converted is usable energy [1].

Ocean Thermal Energy Conversion (OTEC)

The OTEC system is a clean and renewable energy cycle that utilizes the temperature difference between the warm surface ocean water and the cold deep ocean water to produce power. This system typically uses the Rankine cycle processed with a working fluid in the closed loop. The diagram below shows the schematic of an OTEC system.

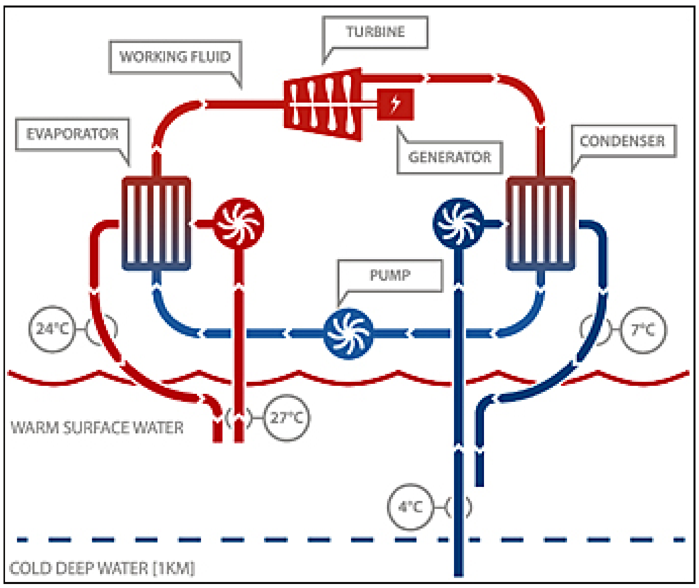


Figure 5: Schematic of OTEC Cycle [11]

The conventional OTEC system has a closed loop where the working fluid is heated and cooled in the boiler and condenser, respectively. The warm ocean water evaporates the working fluid while the cold ocean water condenses the working fluid. At a potential OTEC site, the warm surface ocean water temperature typically varies seasonally at 24ºC-30ºC, while the cold deep ocean water typical remains about 5ºC-9ºC [11]. In order for the OTEC system to operate at producing constant base-load power, the temperature difference in the ocean water needs to be approximately 20ºC [11]. Some primary motivations for an OTEC system are seen in the diagram below.

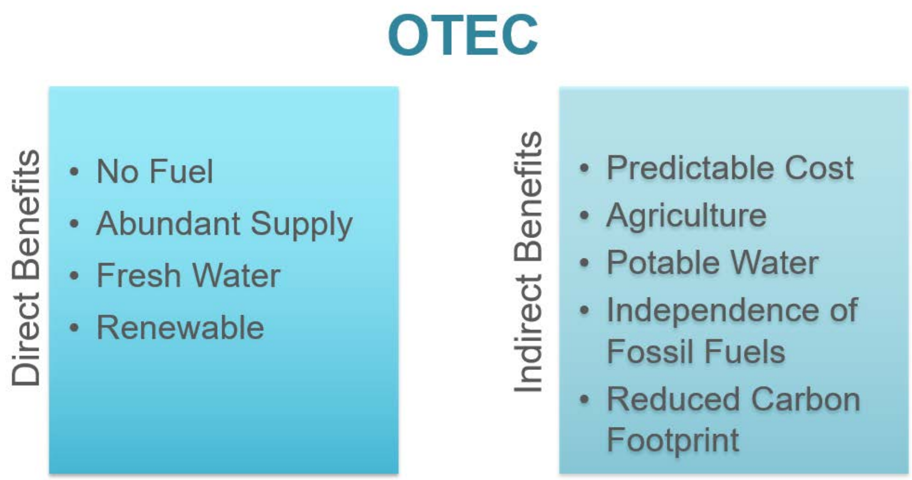


Figure 6: Benefits of an OTEC system

OTEC system uses no fuel. The ocean water itself fuels the system, which makes cost predictable. About 70% of our planet is covered with water, making it a constant and abundant supply of the fuel for the system [9].The by-product of the OTEC system can be applicable to industries outside of power generation. Since water is being boiled in the Rankine cycle, fresh water is a by-product of running the system and can be used in the agricultural sector and the public sector. The fresh water can be used as renewable resource of irrigation for agriculture, and potable water can be supplied to the inhabitants of the shores. The OTEC system is a renewable energy source and can help promote sustainability to reduce the carbon footprint as well as our independence of fossil fuels.

This type of renewable energy is one of the world's largest clean technology and can be available in the tropics and subtropics of the Earth. Currently, most of the OTEC systems built are used for research and development as seen in table 1 [15].

Table 1: Current OTEC Technologies Worldwide

Country	Power output	Purpose	Year
Saga, Japan	30 kW	Research and development	1980
Gosung, Korea	20 kW	Research and development	2012
Reunion Island, France	15 kW	Research and development	2012
Kumejima, Japan	100 kW	Research and development, electricity production	2013
Hawaii, USA	105 kW	Electricity production	2015

Advantages of OTEC

- **OTEC is operated 24/7**
  Since the system is operated by the ocean, which is available 24 hours a day, 7 days a week, there will always be constant supply of clean energy producing a source of power. This proves to be a great advantage over other renewable sources, which only produce intermittent power such as solar or wind energy.
- **Many potential sites for OTEC**
  The National Renewable Energy Laboratory (NREL) of the United States Department of Energy (DOE) lists at least 68 countries and 29 territories as a potential OTEC site where the annual temperature difference is approximately 20C. This means that the technology is accessible for the tropics and subtropics where approximately 3 billion people live [23]
- **Humanitarian Benefits**
  An OTEC system is capable of producing enormous amounts of drinking water, which can be beneficial to nearby communities where fresh water is a limited resource. A small hybrid 1MW OTEC is capable of producing 4,500 cubic meters of fresh water per day, which could supply potable water for 20,000 people [15]. Additionally, nutrients can be brought up from the deep waters that can be useful to the fishing grounds. The nutrient rich water can also be discharged into the water bodies on land such as ponds, where it can help promote growth of marine species.

Disadvantages of OTEC

- **Capital Investment is very high**

  The initial investment for OTEC system requires expensive, large diameter pipes submerged deep in the ocean waters. Some potential OTEC sites lack the economic resources to manufacture this system. Additionally, since OTEC technology has not proven to have a high efficiency, governments and companies are less likely invest in this system since OTEC-produced electricity costs more than electricity generated from fossil fuels at the same costs [22]. Equipment for OTEC systems are huge so a large capital is required upfront.
- **Harmful Effects on the Environment**

  There may be damage to coral reefs by the pipes that must be long enough to extract cold water from the depths of the ocean. With closed cycle OTEC systems, there could be a possibility of ammonia leakage, which even in small concentrations can be toxic to marine life. Additionally, the discharge of cold or warm water from the OTEC system may negatively impact the marine ecosystems so disposal needs to be carried out a few meters away from the shores.
- **Lifespan of OTEC System**

  OTEC systems are located in the ocean and are subject to environmental impacts. The weather could affect the lifespan of the system since natural disasters such as hurricanes are frequent in the tropics and subtropics. This could lead to catastrophic damage to a system that has such high initial cost. Additionally, marine life could impact the materials of the system. The mechanical properties of the material may be weakened by marine life such as algae and will cause corrosion over time.

Kalina Cycle

In November 23, 2004 Alexander I. Kalina patented a new thermal cycle that uses a binary working fluid which was coined "Kalina Cycle". [14] The cycle is composed of a mixture where one component has a high boiling point while the other has a low boiling point. The difference in thermodynamic properties of each component allows the mixture to utilize an enriched liquid stream, and an enriched vapor stream at the condenser and turbine, respectively. The ideal binary working fluid is an ammonia-water mixture, since the mixture concentration could be manipulated to maximize the enthalpy, entropy, energy and exergy at each stage. [20] The figure below show the original model propose by Alexander Kalina where there are separators placed throughout the system to change the mixtures reaction to heat or cooling. For example, if a mixture enters the separator, which is placed before the boiler, then more ammonia could be added to the mixture to have the boiling temperature decrease since ammonias boiling temperature is significantly smaller than the boiling temperature of water.

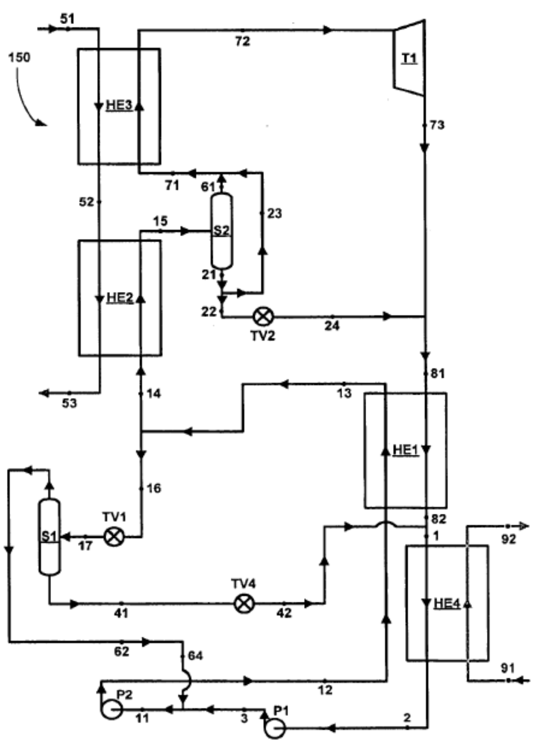
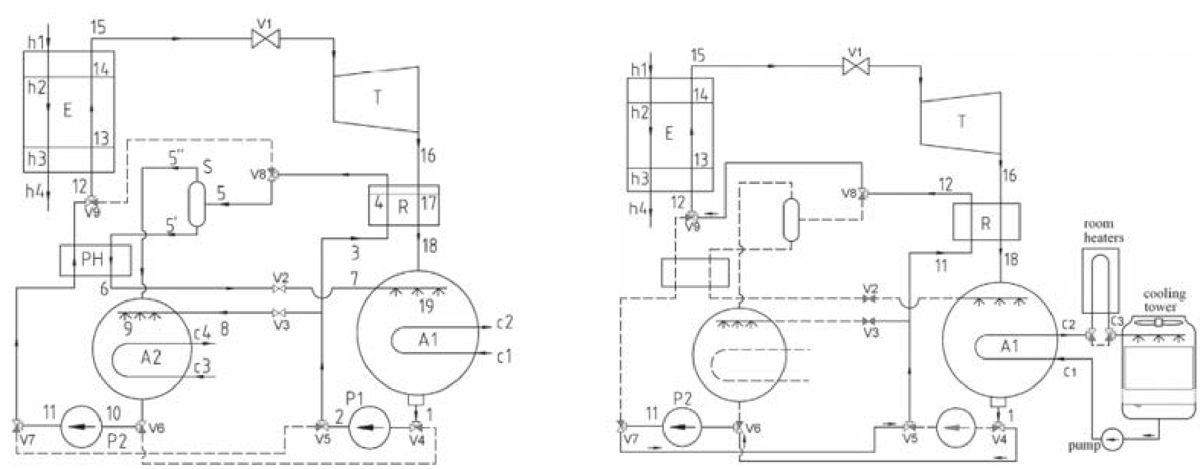


Figure 7: Alexander Kalina's Thermal System using a Kalina Binary Fluid [14]

The benefit of using a Kalina cycle is the increase in thermal efficiency that can be obtained from low temperature waste heat such as OTEC, geothermal and solar. However, it is important to understand why this cycle is not implemented to a higher degree. Various studies have been conducted to understand the relationship between the improvements in the performance of the Kalina cycle and thermo-economics behind the system. In a case study conducted in China, a Rankine Cycle was analyzed using ammonia-water mixture as its working fluid and compared the thermal and power efficiencies seasonally. The two systems were different based on whether the system was non-heating season vs. heating as shown in the figure below. [25]



A B

Figure 8: Rankine Cycle system, A is the system under non-heating season where B is the system under heating seasons.

The system above would switch from a Kalina cycle (ammonia-water) during the non-heating season where there was a low temperature heat waste to a regular Rankine Cycle (water) during the heating season where the high temperatures would be enough to run water by itself. As a result of the experiment, it was shown that there was an increase in efficiency from the Rankine to the Kalina from 18% to 24.7%, however, the power recovery efficiency was higher in the Rankine cycle. This is important because more power was produced overtime with the Rankine cycle compared to the Kalina, thus yielding a higher thermoeconomic scale. [25]

Kalina Cycle and OTEC

Our project is focusing on using an OTEC system as our energy source, therefore we will investigate the combination of OTEC and a Kalina Cycle to understand its feasibility. In an OTEC system, the Kalina cycle would help increase the heat transfer irreversibility's in a closed-cycle between the boiler and condenser. However, a pure OTEC system would not be able to meet the demands of an ammonia-water mixture hence an auxiliary heat must be used in order to run the system. Due to the increase in heat, the overall system would have to have higher requirements such as mass flow rate of the ocean water, surface area and pipe diameter, which would all increase the cost to build. [17] Therefore for the purpose of this paper, the use of a Kalina cycle in the OTEC system will be disregarded. A Hybrid OTEC will be investigated.

Hybrid OTEC Using Auxiliary Energy Source

There are various forms of renewable energy that could be considered as an auxiliary energy source for the hybrid OTEC system such as solar, geothermal, and waste energy. We will discuss the hybrid OTEC system using an auxiliary energy source of solar energy.

Solar energy

The application of solar thermal electricity is one of the most feasible forms of renewable energy since the sun is constantly providing solar radiation on Earth daily. Similar to an OTEC system, solar thermal electricity is produced through the use of a working fluid running a Rankine cycle power system, as a result of the temperature difference power can be produced through the thermodynamic properties of a fluid. The current issue in this application is the high cost. However, by creating a hybrid system, which combines solar thermal electricity with an OTEC system could increase the economic efficiency of solar and the power efficiency of OTEC. The conceptual design of the OTEC-offshore solar pond system is described below:

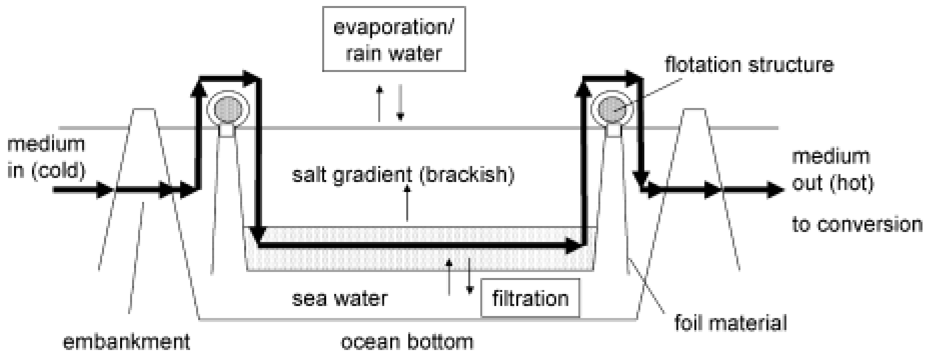


Figure 9: Offshore OTEC Solar Pond System [20]

With the system above, it is estimated that the theoretical thermal efficiency would be 12% which is much greater than a stand-alone OTEC with a thermal efficiency of 3%. There would also be a decrease in the estimated kWh price from 12 c/kWh to 4 c/kWh. Therefore, a hybrid OTEC-solar would be a feasible investment. [20]

Methods

In this section the approach taken to perform the analysis of a pure and hybrid OTEC system will be discussed in depth. The diagram below outlines the steps taken to achieve our final analysis.

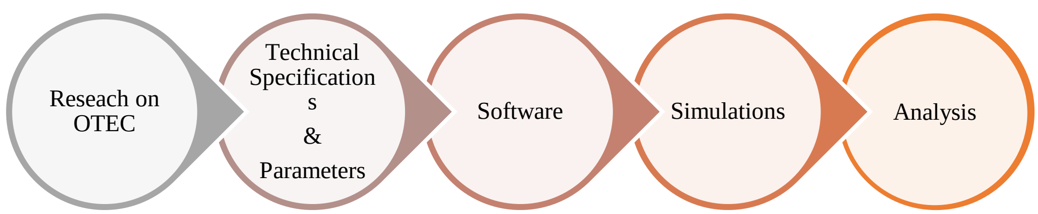
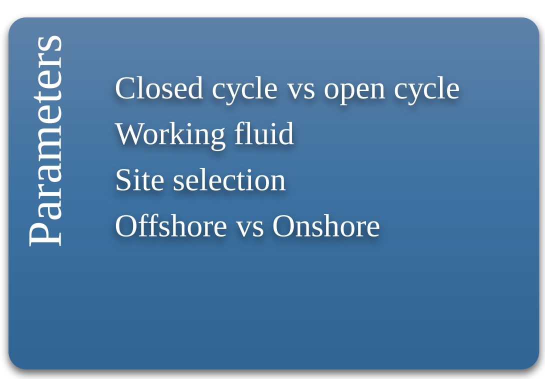


Figure 10: Flow Chart showing the Approach Taken in the Development of the Project

Design Parameters

In designing an OTEC cycle, the following parameters were established for our hybrid system. The type of cycle, working fluid, site selection, and offshore vs onshore system.



Closed Cycle vs Open Cycle

The selection of the type of cycle is one of the key component in the design process of our OTEC system. It represents the foundation of our project as well as the fundamental mechanisms. The OTEC cycle can be designed in two principle options, an open cycle, and a closed cycle. In both cycles, a high water flow rate and extraction of cold water from the depths of the ocean induces a significant consumption of energy.

An open cycle uses warm water at approximately 26ºC, which is expanded in a low pressure flash or chamber allowing it to evaporate with a small fraction of about 5% (Gicquel, 2006). A low pressure chamber condenses the steam that is produced to drive the turbine by a heat exchanger with the cold deep ocean water at approximately 4ºC. The product of this heat exchanger is pure water and can be reused for applications such as potable water for the public sector, and irrigation for the agricultural sector. The open cycle OTEC system has the advantage of producing electricity and fresh water. It uses water as opposed to using a type of refrigerant for its working fluid, which will not be a hazard to the environment. However, a disadvantage of the open cycle is the large turbines that is needed to compensate for the low expansion ratios. This means that the low steam density will require a very large volumetric flow rate to produce only one unit of electricity. Open cycles must be carefully sealed to prevent in-leakage of atmospheric air since it could possibly degrade the operation of the system. According to Koerner, gases that do not condense in this cycle including oxygen, nitrogen, and carbon dioxide dissolved in seawater could possibly be released in the vacuum so they need to be maintained by removal. The figure below shows an open cycle, which consists of a flash evaporator, turbine, condenser, basin to collect reused sea water, and vacuum pump.

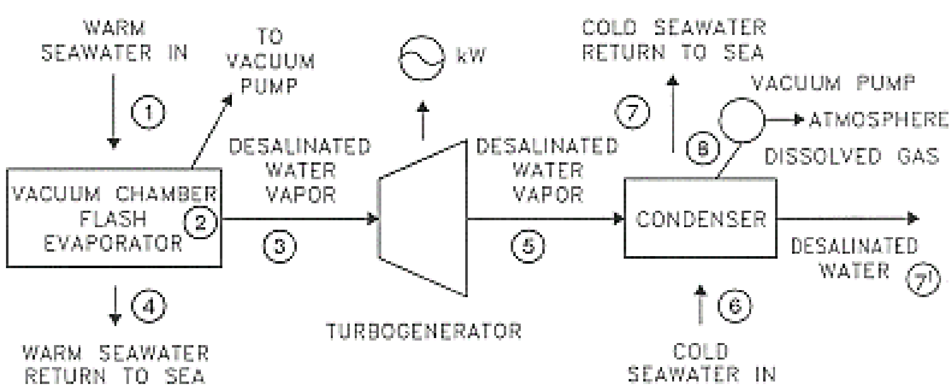


Figure 11: Open Cycle OTEC System

Closed cycle OTEC systems use a pure working fluid that evaporates at the temperature of the warm surface ocean water. In our application, ammonia is the working fluid. Including a refrigerant as the working fluid rearranges the plant equipment of the open cycle. In a closed cycle, ammonia is evaporated by the warm surface ocean water, and condensed by the cold deep ocean water. Since the boiling point of ammonia is high, when it comes into contact with the warm ocean water the pressure becomes high in the evaporator and condenser. It is this pressure difference that allows the ammonia vapor to expand in the turbine creating electricity. However, the disadvantages of closed cycle is the possibility of biofouling of the heat exchangers. Ammonia is toxic even at low concentrations and may affect marine life negatively. The closed cycle includes components such as the heat evaporator, condenser, turbine, and generator. The figure below shows a schematic of a closed cycle OTEC system.

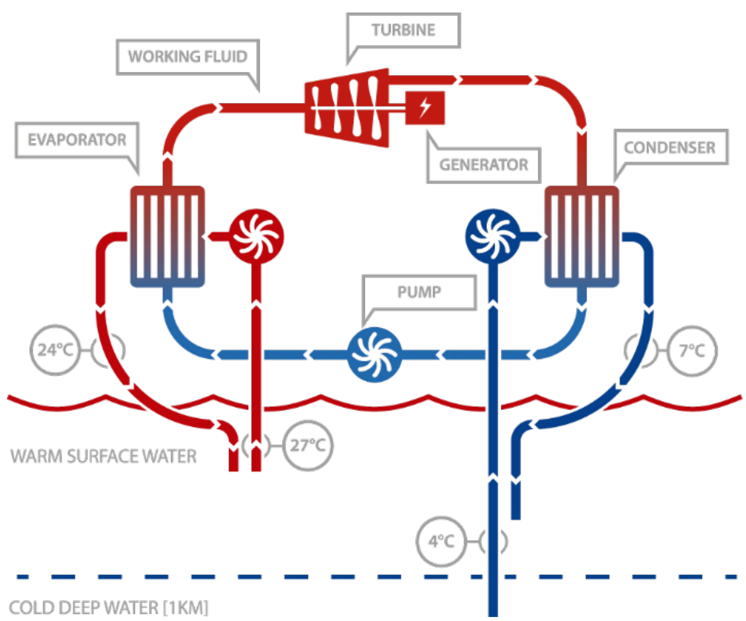


Figure 12: Closed Cycle OTEC System

Our project focuses on using a closed cycle OTEC system. As previously mentioned, the size of the closed cycle system is lowered because of the high running pressure when compared to open cycle. The cost of energy generation is reduced in a closed cycle system due to a lower maintenance cost of corrosion of the turbine blades when compared to open cycle. Additionally, the closed cycle is simpler since there is no need for a vacuum pump as the working fluid of ammonia boils at the temperature of the surface warm ocean water.

Working Fluid

We have chosen ammonia as the working fluid of our closed cycle system. The working fluid of our system was primarily chosen based on the boiling point, critical point, and pressure of the refrigerants. Other factors included cost, availability, and toxicity as an environmental hazard. The table below shows the boiling points, and temperature and pressures of ammonia, ethane, methane, and R-134a compared to water.

Table 2: Properties of Potential Working Fluids [19]

Substance	Boiling point	Critical Point	
	Temperature (K)	Temperature (K)	Pressure (MPa)
Ammonia	239.82	405.4	11.33
Ethane	184.57	305.322	4.87
R-134a	247.08	374.21	4.05
Water	373.12	647.1	22.06

The leading candidates for our system was ammonia and R-134a. Both ammonia and R-134a have a very low boiling point of 239.83K and 247.08K, respectively. However, the temperature at which critical point is observed for ammonia is 405.4K, which is higher than R-134a of 374.21K. This allows us to introduce an additional heat source at the evaporator to increase work output at the turbine. Simulations of our optimized model with each of these working fluids were compared to analyze the total worked produced from the turbine. The working fluid was then compared with the pressure at state 3, while having low pressure as an important consideration.

Site selection

We investigated an appropriate site for our optimized OTEC system. For this parameter, our important considerations were a high annual temperature difference between the surface and deep water, accessibility of deep cold water, and distance of the OTEC system from the shore. Our OTEC site of choice was the north shores of Jamaica. Figure x obtained from OTEC news shows a map of the annual temperature difference between the surface and deep ocean water.

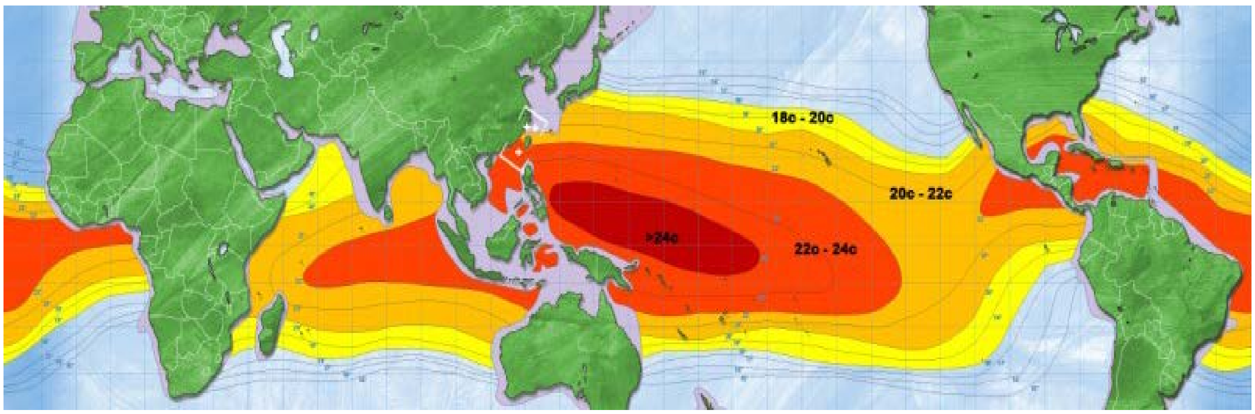


Figure 13: Potential OTEC Sites showing Temperature Difference between Surface and Deep Ocean Water [7]

The map show the annual temperature differences in red, orange, mustard, and yellow. These regions depict areas on the map that have an annual temperature difference of >24ºC, 22ºC-24ºC, 20ºC-2º2C, and 18ºC-20ºC respectively. The red region will provide the highest thermal efficiency while the yellow region will provide the lowest thermal efficiency. The depth of the ocean was an important factor in our OTEC site selection since easy accessibility to deep cold ocean water will help to improve thermal efficiency in our cycle. Just north of Jamaica is the Cayman Trough where the deepest point in the Caribbean Sea is located. It was formed by the tectonic boundary between the North American plate and the Caribbean plate, with the maximum depth of approximately 7,686m [7].

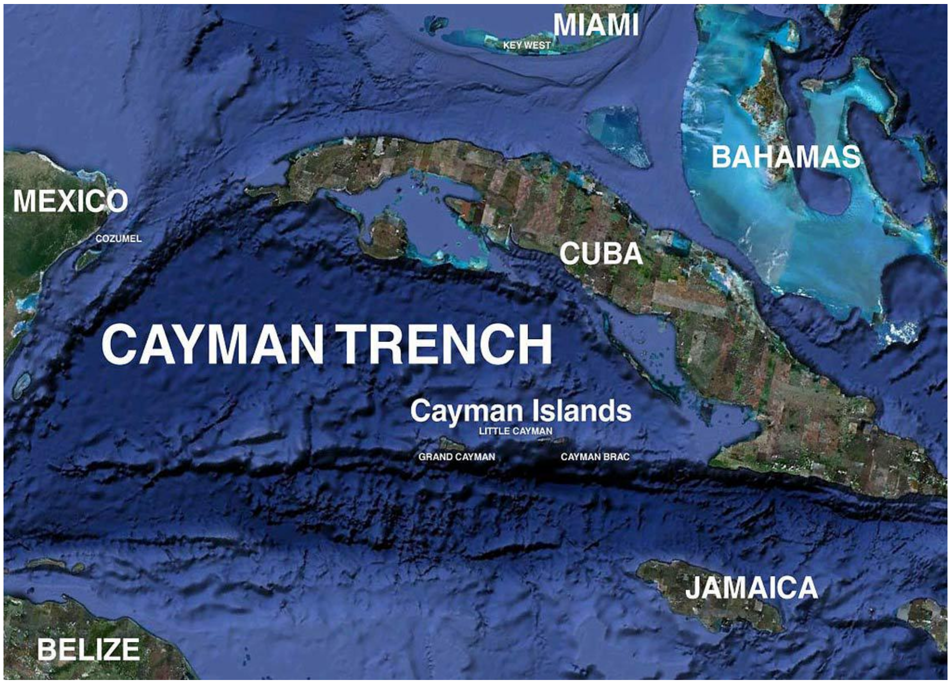


Figure 14: Cayman Trench located North of Jamaica [3]

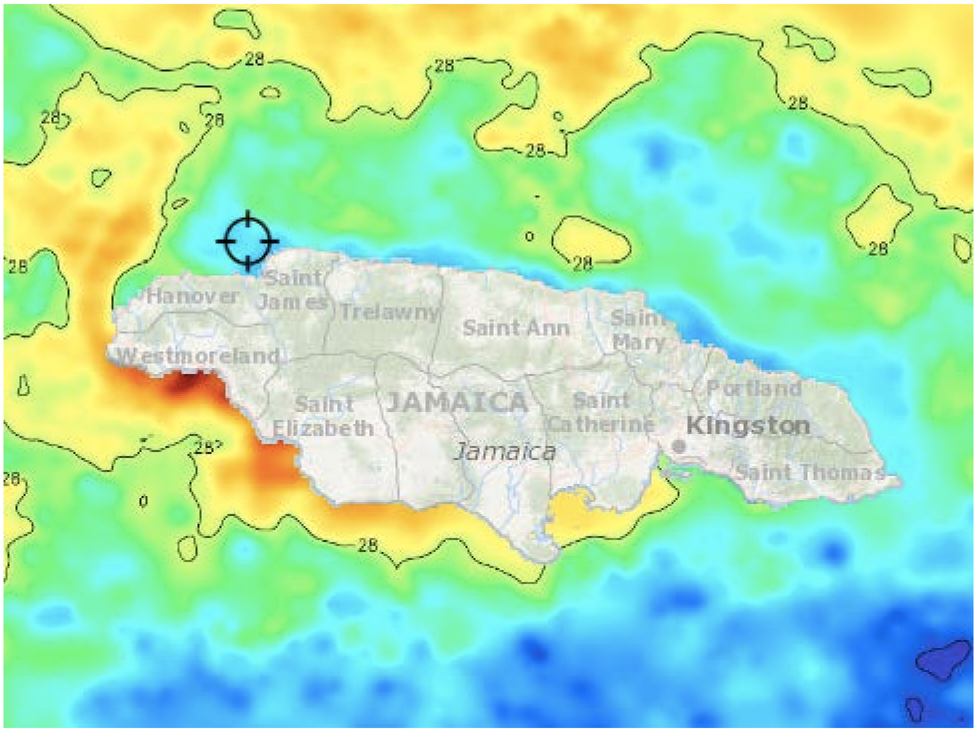


Figure 15: Location of Hybrid OTEC System [3]

Accessibility of deep cold ocean water allows for the existence of an adequate thermal resource. In considering the north shore of Jamaica, we see that as the distance from the islands to the trench increases, the depth increases. However, this pose as a complication from the perspective of heat loss in transferring the energy of our OTEC system to the shore. Additionally, a distance far from the shore will have a longer transit time for the vessels to transport energy. The distance of our hybrid OTEC system is calculated to be approximately 39 km from the shores of Montego Bay, Jamaica. This location was chosen because it is a city that would be more likely to provide infrastructure and resources needed to run the plant. Figure 15 show the location of our OTEC system at 18.3253N/77.5886W from the coast of Montego Bay, Jamaica at 18.467N/77.9235W. Other points were discussed in evaluating the potential OTEC sites. Although OTEC technology is a renewable energy that may possibly provide remote and isolated communities a source of energy, logistical problems with respect to operations, construction, and installation of the plant. Many islands in the Caribbean is underdeveloped, which means they are likely to lack the infrastructure, capital and suitable manpower to operate the plant.

Offshore vs onshore

The two types of OTEC plants we considered were offshore and onshore systems. Below figure x and x visualizes the concepts of the offshore and onshore OTEC systems, respectively.

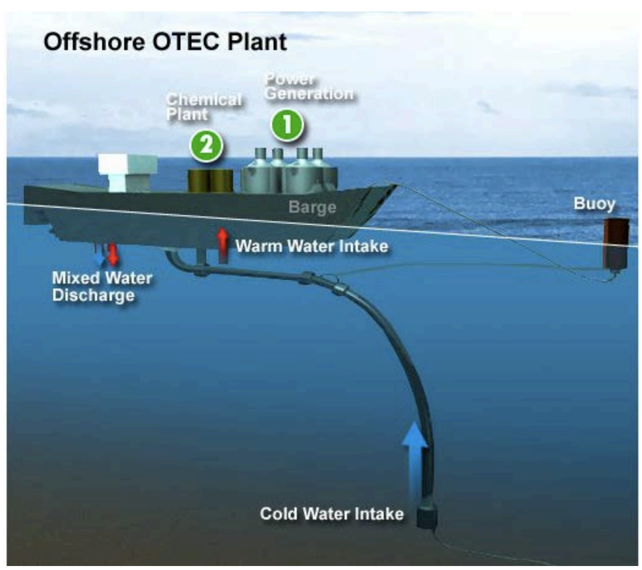


Figure 16: Offshore OTEC Plant [15]

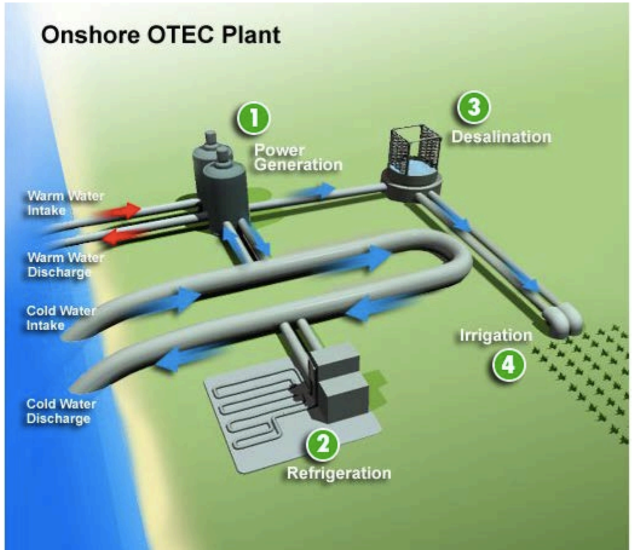


Figure 17: Onshore OTEC Plant [15]

Figure 16 show an offshore system where the OTEC plant is floating in the ocean and anchored. Long pipes draws cold ocean water from the depths while a shorter pipe draws warm ocean water from the surface. The plant is situated on a floating platform where a power generation and chemical plant is located. Figure 17 show an onshore system that is situated on the shores of land. Long pipes are utilized for both the warm and cold ocean waters. More importantly, the pipes for the cold water intake is considerably longer since it has to cover the distance of cold water that is found only at deeper levels of the ocean. The refrigeration, power generation, and desalination components of the system are all located on land.

Our OTEC system was an offshore system, which was chosen primarily because of closeness to the ocean and relatively lower cost to onshore system. According to Multon, offshore OTEC designs have little impact to the land and minimizes the impact of leakage of ammonia. [18] Since offshore designs are floating in the ocean kilometers away from shore, there is no need to find land as an added resource for the system. Ammonia is a hazardous working fluid, even in small concentrations. Having a plant offshore, far away from inhabited areas will likely reduce impact of the system to communities on land. Having the system offshore reduces the complication of long pipes and energy loss that is needed to bring the ocean water to land in an onshore design. While the offshore design have many benefits, it has portrayed some drawbacks. Weather and corrosion are factors that will likely shorten the lifespan of an offshore OTEC system. Onshore OTEC designs have some advantages in that the initial investment is smaller than for offshore. Offshore designs require a huge investment for the construction, transport, and installation of the platform [18]. These types of systems could possibly have less of an impact on the marine sector (ie. Fishermen) since a foreign structure is not built in their fishing territories.

Technical Specifications

Closed Cycle OTEC System [1,16]

A closed cycle utilizes a heat source and a cold source throughout the system. The system uses ammonia as the working fluid that will evaporate when it comes into contact with the warm ocean water. The difference in the temperature of the warm surface ocean water and the cold deep ocean water must be converted from thermal energy to produce maximum work output. The thermal efficiency of OTEC systems can be calculated using the Carnot efficiency equation below.

$$\eta_{Carnot} = 1 - \frac{T_H}{T_C}$$

η_{Carnot} is the Carnot efficiency, T_H is the temperature of the warm surface ocean water, and T_C is the temperature of the cold deep ocean water. The warm ocean water of approximately 25ºC is pumped into the evaporator, where it comes into contact with closed cycle with the working fluid of ammonia. This warm ocean water that has higher temperature than the ammonia was used to transfer heat to the ammonia, which has a lower boiling temperature. The work by the pump in the closed cycle was calculated using the following equation:

$$W_{pump} = \vartheta \times (P_2 - P_1)$$

ϑ is the specific volume of the ammonia, P_1 is the pressure of the ammonia at the condenser, and P_2 is the pressure of the ammonia at the boiler. This warm ocean water evaporates the ammonia into vapor expanding the turbine and turning the blades to drive the turbine. A generator that is connected to the turbine converts mechanical energy to electrical energy. The work of the turbine was calculated using the following equation:

$$W_{turbine} = h_3 - h_4$$

The enthalpy $h_{1,2,3,4}$ of the system is described as state 1, state 2, state 3, and state 4. The net work output was calculated using the following equation:

$$W_{net} = W_{turbine} - W_{pump}$$

When the vaporized ammonia passes through the turbine, it then enters the condenser, where it comes into contact with the cold ocean water of approximately 5ºC. The vaporized ammonia is then condensed back into a liquid state where it is then pumped back into the evaporator to complete the closed cycle. The enthalpy at state 2 was calculated using the following equation:

$$h_2 = h_f + x_2 h_{fg}$$

$$x_2 = \frac{s - s_f}{s_{fg}}$$

h_f is the enthalpy of the saturated liquid, x_2 is the isentropic quality, h_{fg} is the enthalpy of the vaporization, s_f is the entropy of the saturated liquid, and s_{fg} is the entropy of the vaporization. The work of the pump input at state 4 was calculated using the following equation:

$$W_p = -\vartheta \times (P_4 - P_3)$$

ϑ is the specific volume of the ammonia, and $P_4,\ P_3$ is the pressure at state 3 and state 4. When the warm ocean water and the cold ocean water passes through the evaporator and condenser respectively, they are discharged back into the ocean. It can be seen that the efficiency of the system is determined by the temperature difference in the ocean water, the greater the temperature difference, the greater the efficiency, which results in a higher work output.

In designing a feasible closed cycle OTEC system, iterations of the mass flow rate, velocity, and diameter of the pipes were calculated in Excel to determine realistic optimal performance. To calculate the mass flow rate of ammonia, we used the following equations:

$$Q_{net} = m \times (h_2 - h) + W$$

$$m = \rho \times A \times Vel$$

Software

In this project, the incorporation of various software were used to simulate and analysis our Rankine cycle with varying parameters. Four different software were considered to model and analyze the cycle: excel, Aspen Plus V8.8, NIST RefProp 9.1, and Matlab. All four provided a high potential in modeling the system, however not all were feasible in the use for our model. For the simulations we combined the use of RefProp 9.1 and excel.

Matlab

Of these, Matlab was originally used to run a base model, however, due to the vast diversity in thermodynamic properties in working fluids with varying temperatures and pressure, we realized that the Matlab code was not the most efficient method when we wanted to manipulate the parameters of the working fluid.

ASPEN Plus V 8.8

ASPEN is chemical engineering software used to create a simulation model of a system with your own input specification. The capabilities of the software were the ones that would help us in the analysis section of the cycle at various states. Ideally, we wanted to use the software as a verification tool, where it would validate the design to be a feasible model based on the varying parameters. However, when we modeled our cycle there were many errors in the output thermal calculations. As a result, we realized due to time constraints the use of Aspen would not be accurate, therefore, we used Aspen as a way to model a schematic of our system, as shown in the figure below.

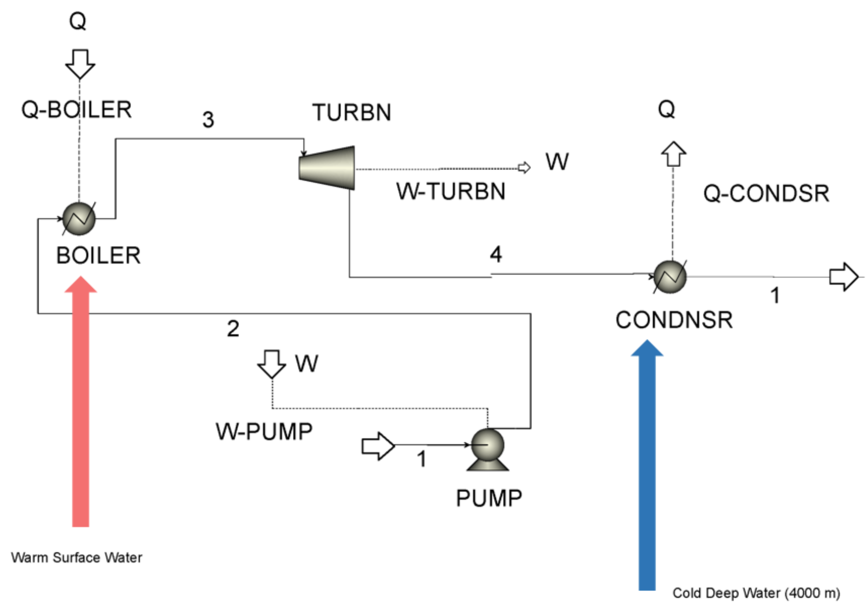


Figure 18: Schematic of the Rankine Cycle used in our system with each state labeled.

NIST RefProp [19]

NIST RefProp, reference fluid thermodynamic and transport properties, was the source for the working fluid thermodynamic properties. This was arguably the most important software used to runs the simulations. Based on the working fluid chosen, which can be pure or a mixture, RefPop will give the output of its saturation table, isoproperty, state table at equilibrium, and specific state tables that allow you to input 2 properties and give you the remaining outputs. It was also used to graph the state tables at each simulations providing the T-s and T-h diagram.

Excel

Excel was the source for our mathematical calculations that are mentioned in the technical specifications, such as thermal efficiency, power efficiency, mass flow rate, cross-sectional area, pipe diameter, heat into the boiler, heat out of the boiler, work into the pump, work out of the turbine, as well as the properties of the ocean water such as enthalpy and mass flow rate.

Simulations

As mentioned, we ran various simulations of our Rankine Cycle with different working fluids and varying parameters. The simulations consisted of the following:

1. A base model of a Rankine Cycle using steam to verify our excel spreadsheet provided accurate results.
2. A base model of a pure OTEC system where the working fluid was the varying factor. The working fluids include: ammonia, ethane, R-134a and water. At each working fluid T_1=279K (6 C) and T_3=298K (25 C) remained the same.
3. Four simulations of a hybrid OTEC system using ammonia as a working fluid. Each simulation increase T_3 by 10K, where the first simulation was at T_3=308K.
4. An optimized hybrid OTEC system simulation having ammonia as a working fluid, where T_3= 370K.

Prior to running simulations 3 and 4 which were used to analysis the feasibility of a hybrid OTEC system, there were two methods considered to run all of the simulations which were tested with Simulations 1 and 2.

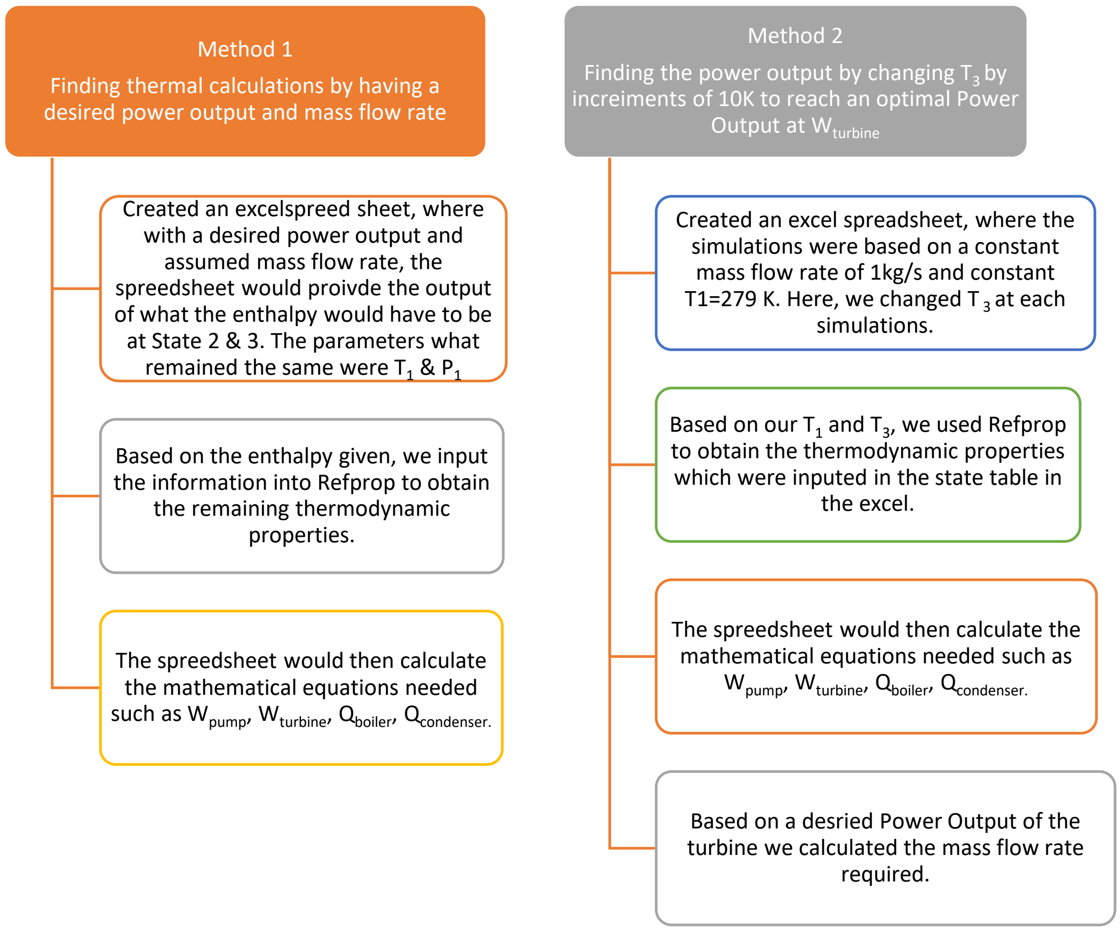


Figure 19: Flow chart showing two methods used in conducting the simulations of the working fluid with an additional energy source.

Method 1 provided an ideal power generated however we ran into the issue of having a high mass flow rate of 1000 kg/s of the working fluid which would not be a feasible number. Therefore, we realized we needed another method. Method 2 was used as a result since it gave us a baseline calculation for each simulation conducted at a constant mass flow rate, thus allowing us to understand how changing the inlet temperature of the turbine, T_3, would affect our systems state tables. As well as, understanding the relationship between ΔT_3 and Δh_{3-4}, and its effects on our systems outputs. By creating the state table's baseline, we were able to create a feasible mass flow rate at each desired power output in MW.

Once we finalized using method 2 as our approach for each simulations, we ran ammonia each time with the parameters of increasing T_3 each time by 10K. Therefore, we ran six different simulations at temperatures of 298, 308, 328, 338, 348 & 370K. The last simulation of 370 K was chosen because there was a linear relationship between the first five temperature sets, therefore we realized that 370 K provided the most optimized calculations (this will be discussed more in the results section). Each simulation had a corresponding state table, such as the one below, where the X are the items that change.

Table 3: Rankine Cycle State Table used for each simulation.

State	Temperature	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	Specific volume (v)	Quality (x)	State
1 Inlet of Pump/Outlet of Condenser	279	X	X	X	X	X		Sat. Liq.
2 Outlet of Pump/ Inlet of Boiler	$\approx T_1$	$\approx P_3$	X	2s $\approx s_1$	X	X		Sat. Vap
3 Outlet of Boiler/Inlet of Turbine	X	X	X	X	X	X		Super-Heated
4 Outlet of Turbine/Inlet of Condenser	X	$\approx P_1$	X	4s $\approx s_3$	X	X	X	Mix.

Analysis Procedure

The Effect of our Ocean Water Temperature Change to the Systems Calculations

In our excel sheet, the properties of the warm surface water and deep cold water was analyzed to understand the mass flow rate (kg/s) for each inlet flow. The information was also used to generate the enthalpy (kJ/kg) for the warm water into the boiler and the cold water into the condenser. These values remained constant throughout each simulation, since it is the only source for heating and cooling in the pure OTEC system. The hybrid OTEC would be created based off of the original heat into the boiler.

A Base Model of a Pure OTEC system with Working Fluid as a Varying Factor

An analysis of the each working fluids boiling temperature and critical point properties, temperature and pressure, as well as the work produced at each simulation ran was conducted to choose the working fluid most suitable for our model.

Base Model of Pure OTEC with Ammonia, Four Simulations and the Optimized Hybrid OTEC Simulations Analysis

For the analysis of each simulation based on the information given in the state table, an excel spreadsheet (Appendix IV) was used where specific cells calculated the following simulations technical specifications for 2 parameters. The table below shows the calculations found using the base model of each simulations at the mass flow rate 1 kg/s, a working fluid velocity 5m/s, and the calculations of the simulations when we change our work output of the turbine. The calculations were then analyzed using charts to understand the following relationships:

- Temperature vs. Enthalpy
- Thermal Efficiency
- Enthalpy vs. $W_{turbine}$
- Mass Flow Rate vs. $W_{turbine}$

Table 4: OTEC system calculations of Ammonia as the working Fluid

<table>
<tr><th colspan="2">Parameters</th></tr>
<tr><td>Analysis 1 based on:
Mass Flow Rate 1 kg/s
Working Fluid Velocity 5m/s</td><td>Analysis 2 based on:
$W_{turbine}$ desired based on</td></tr>
<tr><td>Calculations obtained:
• W_{pump} (kJ/kg)
• $W_{tubrine}$ (kJ/kg) & (MW)
• W_{net} (kJ/kg)
• Q_{in} (kJ/kg)
• Q_{out} (kJ/kg)
• Q_{net} (kJ/kg)
• n_{th} (thermal efficiency)
• Area (m^2)
• Pipe Diameter (m)</td><td>• Mass Flow Required (kg/s)
• Area Required
• Pipe Diameter required
• $W_{required,turbine}$ in order to achieved $W_{desired,turbine}$</td></tr>
</table>

Results and Discussion

Working Fluid Results [19]

As mentioned in our methods, choosing a working fluid for our closed cycle was an important part of understanding the optimal parameters of our hybrid OTEC system. Therefore, we ran ammonia, ethane, R-134a and water at a $T_{1\ 279K}$ and a T_{3298K} to understand how each working fluid worked under a pure OTEC system. The following information demonstrates how we choose ammonia as the most feasible working fluid based on the state tables of each fluid. We also based our choice on each substance critical points temperature, pressure and density as it would increase the maximum T_3 with auxiliary energy source in our system. The table below shows the normal boiling point (K), and critical point of temperature (K) and pressure (MPa) of each substance. It is important to understand the difference between a substance normal boiling point and their critical temperature.

In thermodynamics, the critical points refer to the boundaries where two phases become indistinguishable from one another. Therefore, it is important in our project to understand how the substance would become a supercritical fluid past the critical point of the liquid-vapor phase, as shown in the figure below. A supercritical fluid would not work in our system because we must know the phase at each state of the working fluid. For example, our system has to operate at a specific quality at state 4 to reduce damage to our turbine blades. Therefore, when we analyzed the working fluid chosen we took into account the critical point of each one [8].

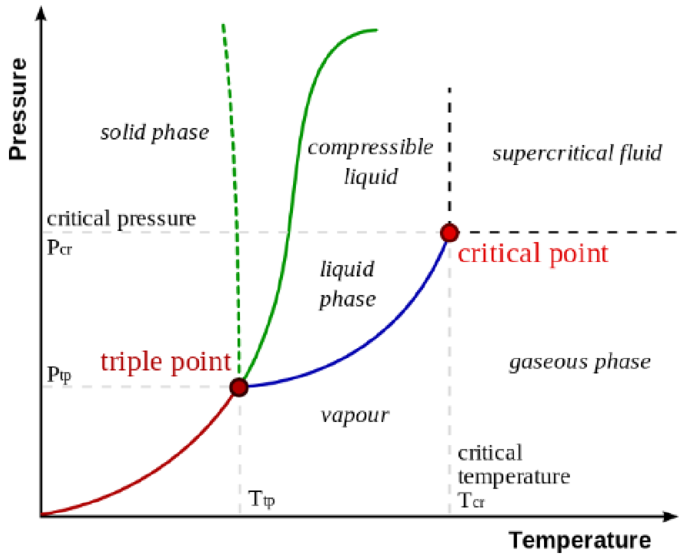


Figure 20: Pressure vs Temperature showing Critical Point [12]

Ammonia (NH3)

Ammonia provided to be a potential working fluid due to its low boiling point of 239.82K and a relatively high critical point. Temperature of 405.4K and even higher critical point pressure of 11.33 MPa.

Table 5: Pure OTEC Ammonia State Table

State	T (K)	P (MPa)	Density (kg/m3)	Volume (m3/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg)	Quality (kg/kg)	Phase
1	279	0.53168	630.48	0.001586	370.32	1.5694		Liq
2	281.75	0.85337	626.83	0.001595	383.32	1.6139		Liq
3	298	0.85	6.4669	0.154633	1638.8	5.9035		Super – Heated
4	279.00	0.53168	4.3464	0.230075	1579.5	5.9035	0.97	Mix

Ethane (CH_3CH_3)

Ethane provided to be a potential working fluid due to its low boiling point of 184.57K and a relatively high critical point. Temperature of 305.32K (compared to its boiling point), however it fell short of having a low pressure at its critical point of 4.8722 MPa.

Table 6: Pure OTEC Ethane State Table

State	T (K)	P (MPa)	Density (kg/m3)	Volume (m3/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg)	Quality (kg/kg)	Phase
1	279	2.7423	385.45	0.0025944	269.40	1.1394		Liq
2	279.31	2.9423	386.18	0.0025894	269.91	1.1394		Liq
3	298	2.9423	49.284	0.020290	595.75	2.2904		Super – Heated
4	279.00	2.7423	57.172	0.017491	535.09	2.0917	0.95	Mix

R-134a (CF_3CH_2F)

R-134a provided to be a potential working fluid due to its low boiling point of 247.08K and a relatively high critical point. Temperature of 374.21K (compared to its boiling point), however it fell short of having a low pressure at its critical point of 4.0593 MPa.

Table 7: Pure OTEC R-134a State Table

State	T (K)	P (MPa)	Density (kg/m3)	Volume (m3/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg)	Quality (kg/kg)	Phase
1	279	0.40	1275.2	0.00078420	207.91	1.0285		Liq
2	279.03	0.40	1275.3	0.00078415	207.95	1.0285		Liq
3	298	0.60	28.597	0.034968	413.86	1.7286		Super – Heated
4	294.75	0.60	30.651	0.032626	401.53	1.6868	0.95	Mix

Water

Although water has a high boiling point of 373.12K, and high critical point properties of 647.1K and 22.064MPa, we used it as a base to compare it to the other possible working fluids. However, our pure OTEC environment would not be suitable to change the phase of water from subcooled liquid to a super-heated vapor.

Working Fluids Compared

After running the different simulations using various working fluids, we analyze the total work produced from the turbine, W_{3-4}. Since each working fluid operated at relatively similar temperatures at each state, we compared the W_{3-4} to the pressure required in State 3.

Substance	P_3 (MPa)	W_{3-4} (KJ/Kg)
Ammonia	0.85	59
Ethane	2.94	60.66
R134a	0.60	12.56

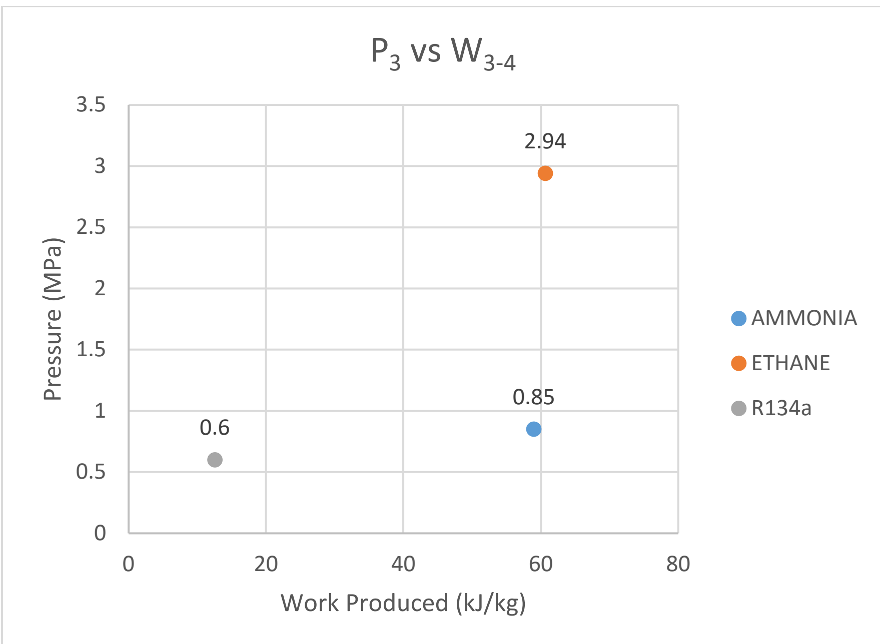


Figure 21: Pressure vs Work Produced

As shown by the graph above, ammonia provided the highest work produced out of the turbine while remaining at a low pressure. Therefore, for our system we choose ammonia as the most suitable working fluid.

Simulation Results for the Closed-Cycle using Ammonia as the working fluid

As previously stated, we produced six simulations of our system with ammonia as our working fluid in order to understand the effects of increasing the temperature at the inlet of the turbine, T_3, on the total $W_{out,turbine}$. During these initial simulations, we used the same T1 279, mass flow rate 1 kg/s, and working fluid velocity 5 m/s. In simulations 1-5, the temperature was increase in increments of 10K, while our 6^{th} simulation the T_3 was increased 20K. Our 6^{th} simulation was increased differently, because we found a trend from simulations 1-5, where there was a linear correlation between T_3 and $h_{3-4,}$ thus increasing $w_{out,turbine.}$ Since we initially ran the simulations at a mass flow rate of 1kg/s, we wanted to understand how the mass flow rate would change at a desirable power output. This section will discuss the results from all six simulations, in demonstrating the following relationships:

- Temperature vs. Enthalpy
- Thermal Efficiency
- Enthalpy vs. Power Produced
- Mass Flow Rate vs. Power Generated

Appendix ###, demonstrates the state tables of all six simulations, but for the purpose of this section we have included the state tables of our base NH_3 model, with the simulations running as a pure OTEC system without any additional heat added to the boiler, and our optimized NH_3 model with auxiliary energy source added to the boiler.

Table 8: Ammonia Base Closed Cycle without Auxiliary Energy

	NH_3 Baseline Closed Cycle Without Auxiliary Energy							
State	Temperature	Pressure	Enthalpy	Entropy	Density	Volume	Quality	Phase
	K	Mpa	kJ/kg	kJ/kg-K	kg/m^3	m^3/kg	x	
1	279	0.53168	370.32	1.5694	630.48	0.00159		Sat. Liq
2	281.75	0.85337	383.32	1.6139	626.83	0.00159		Sat. Liq
3	298	0.85	1638.8	5.9035	6.4669	0.15463		Superheated
4	279	0.53168	1579.5	5.9035	4.3464	0.23008	0.97434	2-Phase

Table 9: Ammonia Optimized Closed Cycle using Auxiliary Energy Source

NH_3 Optimized Closed Cycle Using Auxiliary Energy Source								
State	Temperature	Pressure	Enthalpy	Entropy	Density	Volume	Quality	Phase
	K	Mpa	kJ/kg	kJ/kg-K	kg/m^3	m^3/kg	x	
1	279	0.53168	370.32	1.5694	630.48	0.001586	Subcooled	Sat. Liquid
2	281.12	2.6056	380.52	1.5972	628.81	0.0015903	Subcooled	Sat. Liquid
3	370	2.6056	1761.12	5.7881	16.597	0.06025	Superheated	Gas
4	283	0.61186	1507.6	5.7881	16.597	0.06025	0.9123	2-Phase

Temperature vs. Enthalpy

After running the simulations, there was a correlation between enthalpy and temperature. As the temperature of the working fluid at state 3 was increased, there was an increase in enthalpy, h. The chart, shows the increase of enthalpy in state 3 at each of the temperatures used in the six simulations.

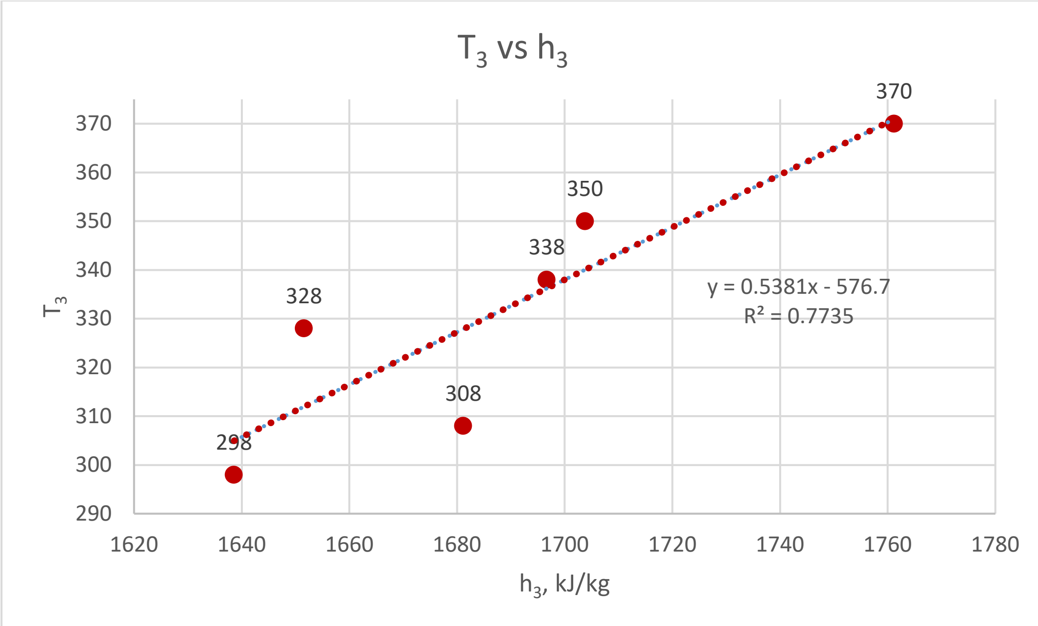


Figure 22: State 3 Temperature vs Enthalpy

As shown in the chart, there was a linear increase in enthalpy that can be generalized by the line: y= 0.5381x-576.7. As the temperature increased, there was an increase in temperature of approximately 1.23 %. This increase is important because as our enthalpy increases, there is a change in entropy at state 3. Thus since our system has $s_3 \approx s_4$ & $P_1 \approx P_4$ then at state 4 the entropy decreases from simulation 1 to six as shown in figure 23.

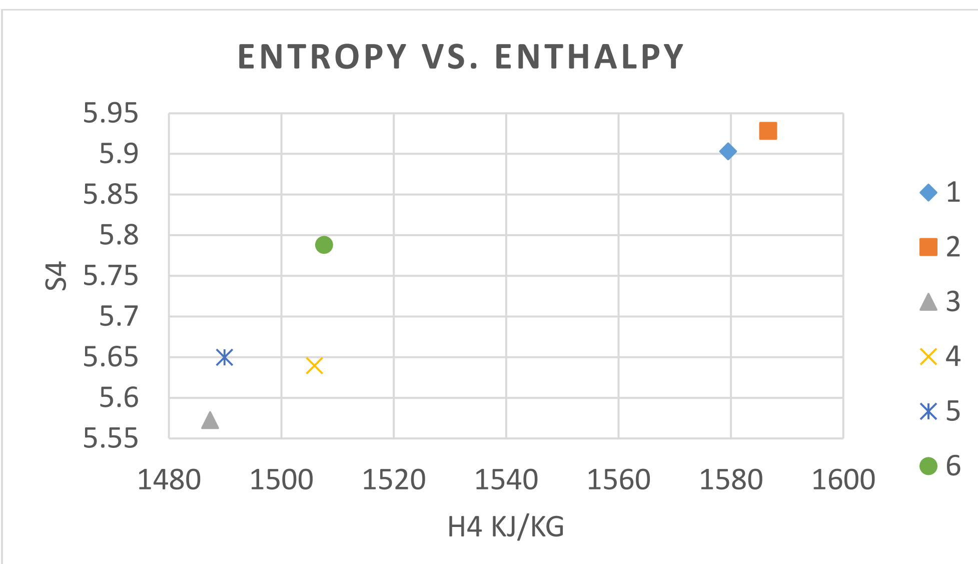


Figure 23: Entropy vs Enthalpy

This correlation between the effects of increasing T_3 on h_3 and h_4, is important because in our system, the work produced from the turbine is based on the formula:

$$W_{turbine3} = \dot{\mathrm{m}} \times (\mathrm{h}_3 - \mathrm{h}_4)$$

Therefore, our results show that there is a linear relationship in our system in regards to increase $T_{3,}$

$$T_3 \xrightarrow{yields} (h_3 - h_4) \xrightarrow{yields} W_{turbine}$$

Which changes our overall T-h diagram based on each simulation, figure 24. In the figure, each simulation has an increase in area under the curve. This increase in area led to a larger range of enthalpy between states, which will led to a higher power produced by the system.

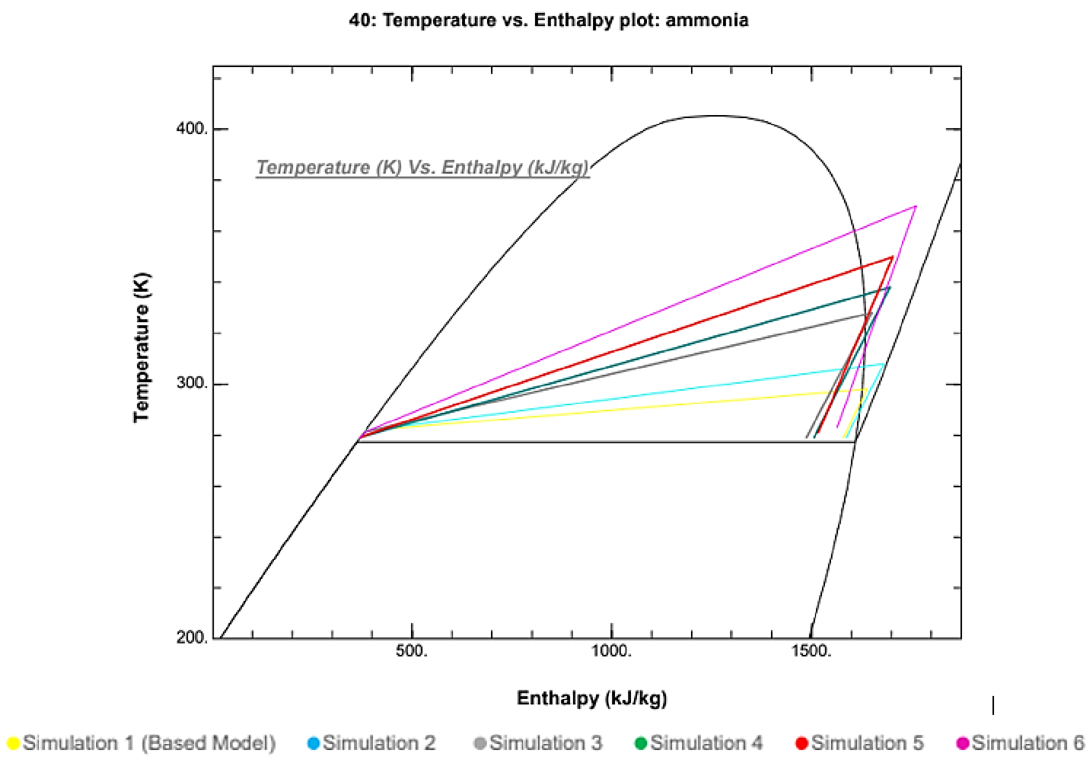


Figure 24: Varying temperature with Resulting Enthalpy

Based on the relationship between temperature and enthalpy, we found that the manipulation of T_3 provided us with the most effective approach of increasing the $W_{turbine}$, which would affect other factors such as thermal efficiency, mass flow rate, and area which will be discussed in the following chapters.

Enthalpy vs Power Output

As discussed above, as the enthalpy increases in the state 3 there is an increase in power output, $W_{turbine}$. Since our objective was to find an optimal system that would produce the max power while using a reasonable supplementary heat into the boiler from an auxiliary energy source, then it was important to analyze the trend between increase enthalpy and power output.

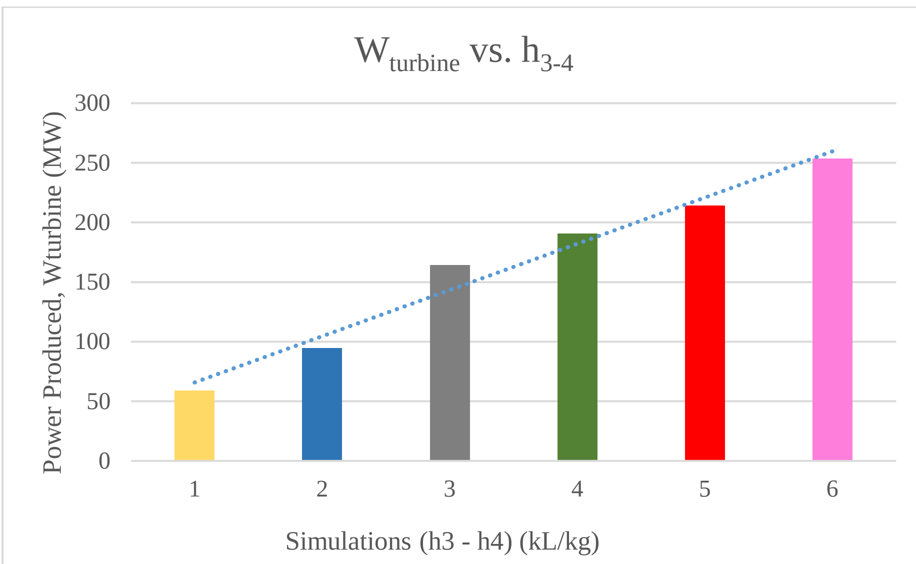


Figure 25: Work vs Enthalpy at State 3

In our system, each simulation increased the power output by a certain percentage however as the temperature was increased each time the increase in power began to stabilize. From our pure OTEC simulation 1 to our increase of 10K in simulation 2 there was a 60.07% in power generated, from simulation 2 to 3 there was a 73.68% increase, the remaining simulations began to decrease and reach an equilibrium. Where from simulation 3 to 4, there was only a 16.13% increase, from simulation 4 to 5 there was a 12.11% increase and from simulation 5 to 6 there was an 18.56% in power generation. It was assumed that based on this trend, if we were to continue to increase our temperature at T_3, the factor in which the power produced increases will reduce in amount since our fluid is reaching its critical temperature of 405.4 K. Therefore, if we maxed out our T_3 at 400 K the power output, $W_{turbine}$, would not have a large change from our optimized power of 253.6 kJ/kg at T_3 of 370K. As a result, the final optimized temperature is T_3=370 at simulation 6, because it was an acceptable enthalpy at T_3 therefore a reasonable amount of additional heat into the boiler could be added while producing a high power produced.

Mass Flow Rate vs Power

An important aspect in our design process was to consider alternative ways to reduce mass flow rate, which would produce an OTEC system that was more realistic. The mass flow rate of ammonia was compared with power to evaluate the relationship for optimal power produced. The power produced by the turbine was calculated at each simulation, which was then used to calculate the mass flow rate of ammonia in the pipes. A linear relationship between power produced by turbine and mass flow rate can be seen in figure x.

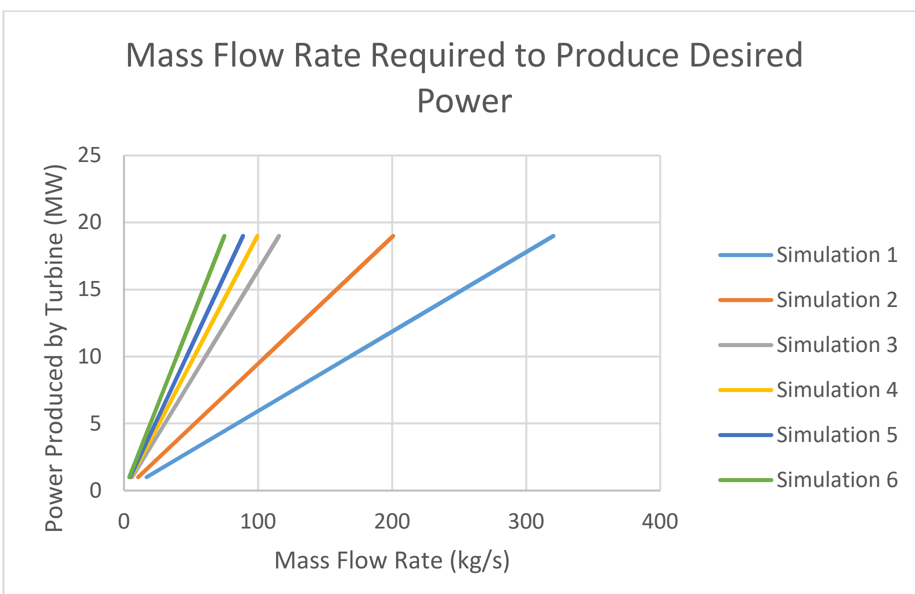


Figure 26: Power Produced vs Mass Flow Rate

It can be seen that the mass flow rate can be varied by increasing the auxiliary heat that is added into the evaporator. Each simulation was varied by increasing the temperature at the evaporator with auxiliary energy source by 10K. With a higher temperature in the evaporator, a higher power will be produced with decreasing mass flow rate of the ammonia.

$$W_{turbine} \propto \dot{m}$$

We compared our base model with our optimized model. With each model, we compared the mass flow rates at a specified desired power output of 10MW. In our base model, the mass flow rate is 168.34 kg/s. Contrastingly, our optimized model produces a mass flow rate of only 39.43 kg/s. Since the evaporating and condensing temperature of the working fluid are fixed, a mass flow rate of the working fluid needs to be larger for a low temperature (when compared to the auxiliary OTEC system) of the base model. With a lower mass flow rate, it is more feasible since it allows for a system that can be manufactured more realistically. Additionally, with such a large mass flow rate for ammonia in the system, it will have consequences with further designs such as the diameter of the pipes in the closed loop of the cycle.

Mass Flow Rate vs Diameter

Although we want to have a system that produces maximum power output, a realistic design for our OTEC system is necessary for it to be feasible. The diameter of the pipes in the closed cycle of the OTEC system was analyzed with mass flow rate of ammonia. Comparing the base model and the optimized model, we choose the mass flow rate of ammonia at each simulation to be at a power produced of 10MW. The base model shows the mass flow rate to be 168.63 kg/s with a pipe diameter of 16.66m. However, the optimized model shows the mass flow rate to be 39.43 kg/s with a pipe diameter of 6.63m. Reducing the size of the system with lower mass flow rate and pipe diameter will not only yield a higher power output, but it will also reduce cost of manufacturing. When temperature is increased in the evaporator with each simulation, the power output increases, and the mass flow rate and diameter decreases. The graph show the base model that has the least desirable outcome.

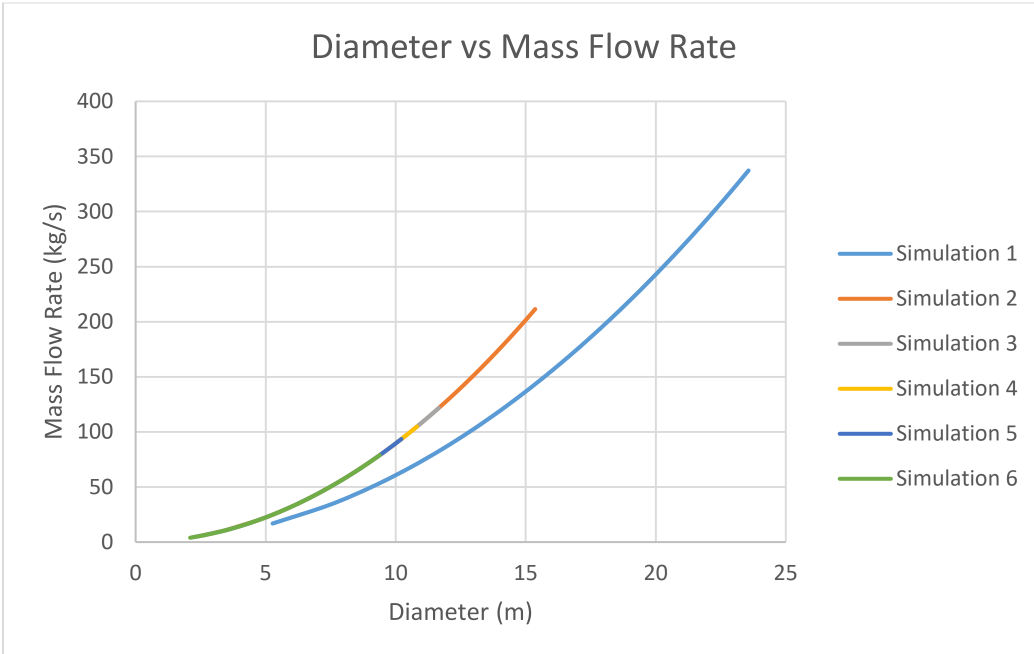


Figure 27: Diameter vs Mass Flow Rate of Ammonia

Thermal Efficiency vs Work

The thermal efficiency determines the power output of the system. With adding auxiliary energy at the evaporator, we need to ensure that this is a plausible investment by comparing the thermal efficiency with the work produced for base model versus optimized model. For our analysis, we have three parameters: thermal efficiency, work desired, and work required. In any engineering system, efficiency plays a huge role in the outcome of work. The thermal efficiency will be calculated by the following equation:

$$\eta_{thermal} = 1 - \frac{q_{in}}{q_{out}}$$

The work desired is the work theoretical work of our OTEC system, and the work required is the calculated work, which projected to be the accurate work based on the efficiency. The relationship among the three parameters can be seen in the equation:

$$Work_{required} = \frac{Work_{desired}}{\eta_{thermal}}$$

For our analysis, we compare the base model with the optimized model with a fixed $Work_{required}$ of 15 MW. The thermal efficiency, $\eta_{thermal}$ was 3.68% and 17% for our base model and optimized model, respectively. The table below show thermal efficiency vs work for each model.

Table 10: Thermal Efficiency vs. Work of a pure OTEC compared to the optimized OTEC

Pure OTEC: Ammonia Working Fluid	Hybrid OTEC: Optimized Ammonia Working Fluid
$\boldsymbol{Work_{required} = 407.6\ MW}$	$Work_{required} = 88.24\ MW$
$\boldsymbol{Work_{desired} = 15MW}$	$Work_{desired} = 15MW$
$\boldsymbol{\eta_{thermal} = 0.0368}$	$\eta_{thermal} = 0.17$

Ammonia was the working fluid used in the comparison above. For the base model, it can be seen that for the same work desired of 15MW, the system actually requires a work required of 407.6MW. However, for our optimized system, the work required is only 88.24MW. This shows that if the system was not optimized, a significant amount of power is needed to achieve the same outcome of 15MW.

Open-Cycle Hybrid OTEC w/ Water as the Working Fluid vs. Closed-Cycle Hybrid OTEC w/ Ammonia as the Working Fluid

To understand the true benefit of a hybrid OTEC system with ammonia as the working fluid, we decided to compare the state table and power output at the turbine of our optimized NH_3 with axillary energy source with an open cycle hybrid system with H_2O as the working fluid. Each cycle was run under the same conditions of T_1 of 279 and our high T_3 of 370 K. One concern we faced was water's high boiling point of 373 K and our system running at a high of 370 K, however due to low pressures, the fluid was able to change states from saturated liquid to superheated vapor. The state tables below show the thermodynamic properties each system has.

NH_3 Optimized Closed Cycle Using Auxiliary Energy Source								
State	Temperature	Pressure	Enthalpy	Entropy	Density	Volume	Quality	Phase
	K	Mpa	kJ/kg	kJ/kg-K	kg/m^3	m^3/kg	x	
1	279	0.53168	370.32	1.5694	630.48	0.001586	Subcooled	Sat. Liquid
2	281.12	2.6056	380.52	1.5972	628.81	0.0015903	Subcooled	Sat. Liquid
3	370	2.6056	1761.12	5.7881	16.597	0.06025	Superheated	Gas
4	283	0.61186	1507.6	5.7881	16.597	0.06025	0.9123	2-Phase
H_2O Open Cycle Using Auxiliary Energy Source								
State	Temperature	Pressure	Enthalpy	Entropy	Density	Volume	Quality	Phase
	K	Mpa	kJ/kg	kJ/kg-K	kg/m^3	m^3/kg	x	
1	279	0.05	24.642	0.089081	999.92	0.0010001	Subcooled	Sat. Liquid
2	339.48	0.08	277.71	0.91	979.82	0.0010206	Subcooled	Sat. Liquid
3	370	0.08	2672.1	7.4526	0.47448	2.1076	Superheated	Gas
4	354.47	0.05	2549.5	7.3229	0.32201	3.1055	0.95846	2-Phase

Figure 28: Comparing Performance of Ammonia vs Water

It could be inferred from above, that water has a higher enthalpy at T_3 compared to ammonia. Therefore, the $W_{turbine,H2O}$ for this state table is 122.6 kJ/kg, which is However, in order to compress water from state 4 back to state 1, the work needed to be added into the pump is 253.068 kJ/kg.

Conclusion

Our design parameters consisted of a closed cycle OTEC system located 39km off the shores of Jamaica, with ammonia selected as the working fluid. The surface temperature of the water was 25ºC, and the deep water temperature at 1000m was 6ºC. Based on the project results, it was concluded that a conventional OTEC system would not be feasible. A conventional closed cycle OTEC system using ammonia as the working fluid has a low efficiency of 3.68% due to low temperature difference between the warm surface ocean water and the cold deep ocean water. The inlet temperature into the turbine produces a low change in enthalpy from state 3 to 4, which produces a low power output from the turbine.

A hybrid OTEC system was a more suitable solution to increase the thermal efficiency of the system. By using an auxiliary energy source at the boiler, we increase temperature at state 3, which increases power output by 430% at an optimized temperature of 370K of ammonia. With this increase in power output, other parameters in our analysis were affected such as work required, work desired, mass flow rate, and pipe diameter. At the optimized power output of 15MW, the work required was 60MW, the mass flow rate was 236.59 kg/s, and the pipe diameter was 1.91m. These calculations were significantly lower than the values obtained from the base model.

Although the thermal efficiency was increased with each simulation, the results confirmed that OTEC systems produce low power output. The thermal efficiency only increased by 13.32%. This supports previous claims of altering the conceptual design to yield better results in power generation. Thus, improvements on this model would greatly impact the performance of the system.

Recommendations

To improve the overall performance of the system, various recommendations can be made. A more advanced Rankine cycle can be utilized, such as the regenerative cycle. This would allow the working fluid to flow through the boiler multiple times to increase temperature inlet of the turbine. A regenerative cycle in addition to an auxiliary energy source will ensure a consistently high temperature at state 3. Further investigations can be made into various working fluids such as other refrigerants that have similar properties to ammonia. The possibility of utilizing various binary fluids in the Kalina cycle would increase the thermal efficiency by a small percentage.

An important consideration in taking into account of the development of an OTEC system are the effects on the environment as well as the environmental effects on the system. The OTEC system may potentially have negative effects on marine life and the inhabitants of the nearby island. Ammonia even in small concentrations is toxic to the environment. Therefore, a safety regulation should be investigated in the production of an OTEC system, should an incident where the working fluid is released into the ocean. Additionally, the materials of the system should be considered as location and marine life can impact its mechanical properties and life span. For example, the growth of algae on the pipes and platform may be corrosive to the material selected. Locations where there are higher annual temperature difference in the ocean water such as Indonesia should be considered to optimize the system. Since the OTEC cycle is situated on a floating platform offshore, the weather in which it is located may affect its lifespan. Potential OTEC sites are typically located in the Tropics, where frequency of natural disasters are high, therefore maintenance of the operations would be costly.

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Appendices

Appendix I: Excel Simulations of Each Potential Working Fluid at a Pure OTEC

Ammonia

State	T (K)	P (MPa)	Density (kg/m3)	Volume (m3/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg)	Quality (kg/kg)	Phased
1	279	0.53168	630.48	0.001586	370.32	1.5694		Liq
2	281.75	0.85337	626.83	0.001595	383.32	1.6139		Liq
3	298	0.85	6.4669	0.154633	1638.8	5.9035		Super – Heated
4	279.00	0.53168	4.3464	0.230075	1579.5	5.9035	0.97	Mix

Ethane

State	T (K)	P (MPa)	Density (kg/m3)	Volume (m3/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg)	Quality (kg/kg)	Phased
1	279	2.7423	385.45	0.0025944	269.40	1.1394		Liq
2	279.31	2.9423	386.18	0.0025894	269.91	1.1394		Liq
3	298	2.9423	49.284	0.020290	595.75	2.2904		Super – Heated
4	279.00	2.7423	57.172	0.017491	535.09	2.0917	0.95	Mix

R-134a

State	T (K)	P (MPa)	Density (kg/m3)	Volume (m3/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg)	Quality (kg/kg)	Phased
1	279	0.40	1275.2	0.00078420	207.91	1.0285		Liq
2	279.03	0.40	1275.3	0.00078415	207.95	1.0285		Liq
3	298	0.60	28.597	0.034968	413.86	1.7286		Super – Heated
4	294.75	0.60	30.651	0.032626	401.53	1.6868	0.95	Mix

Appendix II: Excel Open-Cycled Water with an Auxiliary Energy Source

	T (K)	P (MPa)	D (kg/m^3)	V (m^3/kg)	H (kJ/kg)	S (kJ/kg)	Quality	Phase
1	279	0.05	999.92	0.0010001	24.642	0.089081	Subcooled	Liquid
2	339.48	0.08	979.82	0.0010206	277.71	0.91	Subcooled	Liquid
3	370	0.08	0.47448	2.1076	2672.1	7.4526	Superheated	Gas
4	354.47	0.05	0.32201	3.1055	2549.5	7.3229	0.95846	2-Phase

Appendix III: Excel Ocean Water information

This information below was used throughout each simulation conducted in this research

Ocean Info.	**Hot H2O**	**Enthalpy (kJ/kg)**	**Cold H2O**	**Enthalpy (kJ/kg)**
Inlet Temperature (C)	25	104.2	7	28.765
Outlet Temperature (C)	20	83.286	5	20.389
Mass Flow Rate (kg/s)	189,746.80		77,417.32	

Appendix IV: Excel Simulation of a Pure OTEC Using Ammonia as the Working Fluid

Calculations

Simulation 1								
Working Fluid	***Pure Ammonia***		**w, Work In (kJ/kg)**	**w, Work Out (kJ/kg)**	**qin, Heat In (kJ/kg)(**	**qout, Heat Out in (kJ/kg)**	***Ammonia***	
Mass Flow Rate (Inlet of Turbine kg/s)		Pump	-370.32				*cv (kJ/kg K)*	***cp (kJ/kg K)***
Turbine Inlet Temperature (K)	298	Boiler			1638.8		1.66	2.19
Ideal Power Out of Turbine MW	15	Turbine		59				
nth, Cycle Efficiency		Condenser				1209.18		
Thermal Eff., nth	*%*	Power Eff.	%					
0.2621552355	26.21552355	7.244856661	724.4856661					

State Table for T3 298K

STATE	Temperature (K)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	specific volume (v)	Quality (x)	State
Inlet of Pump/Outlet of Condenser	279	0.53168	370.32	1.5694	630.48	0.001586093135	0	Sat. Liq
Outlet of Pump/ Inlet of Boiler	281.75	0.85337	0	1.6139	626.83	0.001595328877		Sat. Liq
Outlet of Boiler/Inlet of Turbine	298	0.85	1638.8	5.9035	6.4669	0.1546335957		Superheated
Outlet of Turbine/Inlet of Condenser	279	0.53168	1579.5	5.9035	4.3464	0.2300754648	0.97434	2-Phase

Mass Flow Rate Influence on System

Using Mass Flow Rate as 1 kg/s

mass flow rate (kg/s)	vel (m/s)	
1	5	
Area Based on Mass Flow Rate of 1 kg/s & Vel (5 m/s)		
0.03092671914	**m^2**	
Diameter of Pipe Based on Information Above (m)		
0.1984366 m	19.84 cm	
Work Out of Turbine Based on Mass Flow Rate of 1kg/s		
59	kJ/sec	
59	kW	
0.0593	MW	

Desired MW vs. Mass Flow Rate vs. Area vs. Diameter

Therefore,			
Desired MW	*Mass Flow Needed (kg/s)*	Desired MW	Mass flow rate
1	16.86340641	11	185.4974705
2	33.72681282	12	202.3608769
3	50.59021922	13	219.2242833
4	67.45362563	14	236.0876897
5	84.31703204	15	252.9510961
6	101.1804384	16	269.8145025
7	118.0438449	17	286.6779089
8	134.9072513	18	303.5413153
9	151.7706577	19	320.4047218
10	168.6340641	20	337.2681282
278	4688.026981		

Area Based on Mass Flow Rate of Desired MW					
Desired MW	*Area Needed*	Pipe Diameter	Desired MW	*Area Needed*	Pipe Diameter
1	21.81079258	5.269761576	11	239.9187184	17.47782188
2	43.62158516	7.452568292	12	261.729511	18.25498959
3	65.43237774	9.127494794	13	283.5403035	19.00039557
4	87.24317032	10.53952315	14	305.3510961	19.71764233
5	109.0539629	11.78354511	15	327.1618887	20.40969882
6	130.8647555	12.90822693	16	348.9726813	21.07904631

7	152.6755481	13.9424786	17	370.7834739	21.7277836
8	174.4863406	14.90513658	18	392.5942664	22.35770488
9	196.2971332	15.80928473	19	414.405059	22.97035817
10	218.1079258	16.66444931	20	436.2158516	23.56709022

Appendix V: Excel Simulation of a Hybrid OTEC Using Ammonia as the Working Fluid (Simulations 2-6)

Calculations Simulation 2

Simulation 2						
Working Fluid	***Pure Ammonia***		**w, Work In (kJ/kg)**	**w, Work Out (kJ/kg)**	**qin, Heat In (kJ/kg)(**	**qout, Heat Out in (kJ/kg)**
Mass Flow Rate (Inlet of Turbine kg/s)		Pump	12.61			
Turbine Inlet Temperture (K)	308	Boiler			1298.27	
Ideal Power Out of Turbine MW	15	Turbine		95		
nth, Cycle Efficiency		Condenser				1216.28
Thermal Eff., nth	**%**	Power Eff.	%			
0.0631532732	6.31532732	0.8667019027	86.67019027			

Ammonia		
cv (kJ/kg K)	***cp (kJ/kg K)***	
1.66	2.19	

State Table of T3 308

STATE	Temperture (K)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	specfic volume (v)	Quailty (x)	State
Inlet of Pump/Outlet of Condenser	279	0.53168	370.32	1.5694	630.48	0.001586093135	0	Sat. Liq
Outlet of Pump/ Inlet of Bolier	281.7	0.62	382.93	1.6139	626.72	0.001595608884		Sat. Liq
Outlet of Boiler/Inlet of Turbine	308	0.62	1681.2	6.1954	4.3844	0.2280813794		SuperHeated
Outlet of Turbine/Inlet of Condenser	279	0.53168	1586.6	5.9287	4.3215	0.2314011339	0.98	2-Phase

Mass Flow Rate Influence on System

Using Mass Flow Rate as 1 kg/s

Area Based on Mass Flow Rate of 1 kg/s & Vel (5 m/s)		
0.04561627589	**m^2**	
Diameter of Pipe Based on Information Above (m)		
0.1984366 m	19.84 cm	
Work Out of Turbine Based on Mass Flow Rate of 1kg/s		
95	kJ/sec	
95	kW	
0.0946	MW	

Desired MW vs. Mass Flow Rate vs. Area vs. Diameter

Therefore,			
Desired MW	*Mass Flow Needed (kg/s)*	Desired MW	mass flow
1	10.57082452	11	116.2790698
2	21.14164905	12	126.8498943
3	31.71247357	13	137.4207188
4	42.2832981	14	147.9915433
5	52.85412262	15	158.5623679
6	63.42494715	16	169.1331924
7	73.99577167	17	179.7040169
8	84.56659619	18	190.2748414
9	95.13742072	19	200.845666
10	105.7082452	20	211.4164905

Area Based on,					
Desired MW	*Area Needed*	Pipe Diameter	Desired MW	*Area Needed*	Pipe Diameter
1	9.269344609	3.435419345	11	101.9627907	11.39399697
2	18.53868922	4.858416631	12	111.2321353	11.9006417
3	27.80803383	5.950320852	13	120.5014799	12.3865806
4	37.07737844	6.870838691	14	129.7708245	12.85416217
5	46.34672304	7.681831187	15	139.0401691	13.30532191
6	55.61606765	8.415024449	16	148.3095137	13.74167738

7	64.88541226	9.089265237	17	157.5788584	14.16459683
8	74.15475687	9.716833261	18	166.848203	14.57524989
9	83.42410148	10.30625804	19	176.1175476	14.97464576
10	92.69344609	10.86374985	20	185.3868922	15.36366237

Calculations for simulations 3

Simulation 3						
Working Fluid	***Pure Ammonia***		**w, Work In (kJ/kg)**	**w, Work Out (kJ/kg)**	**qin, Heat In (kJ/kg)(**	**qout, Heat Out in (kJ/kg)**
Mass Flow Rate (Inlet of Turbine kg/s)		Pump	14.2			
Turbine Inlet Temperture (K)	328	Boiler			1267.08	
Ideal Power Out of Turbine MW	15	Turbine		164		
nth, Cycle Efficiency		Condenser				1116.98
Thermal Eff., nth	***%***	Power Eff.	%			
0.1184613442	11.84613442	0.9135727328	91.35727328			

Ammonia	
cv (kJ/kg K)	*cp (kJ/kg K)*
1.66	2.19

State Table with T3 328K

STATE	Temperature (K)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	specific volume (v)	Quality (x)	State
Inlet of Pump/Outlet of Condenser	279	0.53168	370.32	1.5694	630.48	0.001586093135	0	Sat. Liq
Outlet of Pump/ Inlet of Boiler	281.83	2.3024	384.52	1.6	627.83	0.001592787857		Sat. Liq
Outlet of Boiler/Inlet of Turbine	328	2.0435	1651.6	5.5728	15.394	0.06496037417		SuperHeated
Outlet of Turbine/Inlet of Condenser	279	0.53168	1487.3	5.5728	4.7027	0.2126438004	0.9	2-Phase

Mass Flow Rate Influence on System

Using Mass Flow Rate as 1 kg/s

Mass Flow Rate	Vel (m/s)
1	5
Area Based on Mass Flow Rate of 1 kg/s & Vel (5 m/s)	
0.01299207483	**m^2**
Diameter of Pipe Based on Information Above (m)	
0.1984366 m	19.84 cm
Work Out of Turbine Based on Mass Flow Rate of 1kg/s	
164	kJ/sec
164	kW
0.1643	MW

Desired MW vs. Mass Flow Rate vs. Area vs. Diameter

Therefore,			
Desired MW	*Mass Flow Needed (kg/s)*	Desired MW	mass flow
1	6.086427267	11	66.95069994
2	12.17285453	12	73.03712721
3	18.2592818	13	79.12355447
4	24.34570907	14	85.20998174
5	30.43213634	15	91.29640901
6	36.5185636	16	97.38283628
7	42.60499087	17	103.4692635
8	48.69141814	18	109.5556908
9	54.7778454	19	115.6421181
10	60.86427267	20	121.7285453

Area Based on,					
Desired MW	*Area Needed*	Pipe Diameter	Desired MW	Area Needed	Pipe Diameter
1	5.337066342	2.606792984	11	58.70772976	8.645754233
2	10.67413268	3.686561992	12	64.0447961	9.030195785
3	16.01119903	4.515097892	13	69.38186245	9.398925767
4	21.34826537	5.213585967	14	74.71892879	9.753726223
5	26.68533171	5.828966315	15	80.05599513	10.09606581
6	32.02239805	6.385312675	16	85.39306147	10.42717193

7	37.35946439	6.896925954	17	90.73012781	10.74808282
8	42.69653074	7.373123983	18	96.06719416	11.05968598
9	48.03359708	7.820378951	19	101.4042605	11.36274718
10	53.37066342	8.243403217	20	106.7413268	11.65793263

Calculations for Simulation 4

Simulation 4						
Working Fluid	***Pure Ammonia***		**w, Work In (kJ/kg)**	**w, Work Out (kJ/kg)**	**qin, Heat In (kJ/kg)(**	**qout, Heat Out in (kJ/kg)**
Mass Flow Rate (Inlet of Turbine kg/s)		Pump	2.09			
Turbine Inlet Temperture (K)	338	Boiler			1324.29	
Ideal Power Out of Turbine MW	15	Turbine		191		
nth, Cycle Efficiency		Condenser				1135.58
Thermal Eff., nth	*%*	Power Eff.	%			
0.1424989995	14.24989995	0.9890461216	98.90461216			
Ammonia						
cv (kJ/kg K)	***cp (kJ/kg K)***					
1.66	2.19					

State Table of T3 338

STATE	Temperature (K)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	specific volume (v)	Quality (x)	State
Inlet of Pump/Outlet of Condenser	279	0.53168	370.32	1.5694	630.48	0.001586093135	0	Sat. Liq
Outlet of Pump/ Inlet of Boiler	279.28	1.8482	372.41	1.5694	627.83	0.001592787857		Sat. Liq
Outlet of Boiler/Inlet of Turbine	338	1.8482	1696.7	5.75	12.902	0.0775073632		Superheated
Outlet of Turbine/Inlet of Condenser	279	0.53168	1505.9	5.6396	4.6262	0.2161601314	0.915	2-Phase

Mass Flow Rate Influence on System

Using Mass Flow Rate as 1 kg/s

Mass Flow Rate	Vel (m/s)	
1	5	
Area Based on Mass Flow Rate of 1 kg/s & Vel (5 m/s)		
0.01550147264	**m^2**	
Diameter of Pipe Based on Information Above (m)		
0.1984366 m	19.84 cm	
Work Out of Turbine Based on Mass Flow Rate of 1kg/s		
191	kJ/sec	
191	kW	
0.1908	MW	

Desired MW vs. Mass Flow Rate vs. Area vs. Diameter

Therefore,			
Desired MW	*Mass Flow Needed (kg/s)*	Desired MW	mass flow
1	5.241090147	11	57.65199161
2	10.48218029	12	62.89308176
3	15.72327044	13	68.13417191
4	20.96436059	14	73.37526205
5	26.20545073	15	78.6163522
6	31.44654088	16	83.85744235
7	36.68763103	17	89.09853249
8	41.92872117	18	94.33962264
9	47.16981132	19	99.58071279
10	52.41090147	20	104.8218029

Area Based on,					
Desired MW	*Area Needed*	Pipe Diameter	Desired MW	Area Needed	Pipe Diameter
1	4.595807128	2.419001513	11	50.55387841	8.022920384
2	9.191614256	3.420984746	12	55.14968553	8.379667047
3	13.78742138	4.189833523	13	59.74549266	8.721833989
4	18.38322851	4.838003025	14	64.34129979	9.051074878
5	22.97903564	5.40905182	15	68.93710692	9.368752573
6	27.57484277	5.925319393	16	73.53291405	9.67600605
7	32.1706499	6.400076423	17	78.12872117	9.973798745

8	36.76645702	6.841969493	18	82.7245283	10.26295424
9	41.36226415	7.257004538	19	87.32033543	10.54418314
10	45.95807128	7.649554443	10	91.91614256	10.81810364

Calculations for Simulation 5

Simulation 5						
Working Fluid	***Pure Ammonia***		**w, Work In (kJ/kg)**	**w, Work Out (kJ/kg)**	**qin, Heat In (kJ/kg)(**	**qout, Heat Out in (kJ/kg)**
Mass Flow Rate (Inlet of Turbine kg/s)		Pump	3.09			
Turbine Inlet Temperture (K)	350	Boiler			1330.39	
Ideal Power Out of Turbine MW	15	Turbine		214		
nth, Cycle Efficiency		Condenser				1119.58
Thermal Eff., nth	***%***	Power Eff.	%			
0.1584572945	15.84572945	0.9855539972	98.55539972			
Ammonia						
cv (kJ/kg K)	***cp (kJ/kg K)***					
1.66	2.19					

State Table for T3 350K

STATE	Temperature (K)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	specific volume (v)	Quality (x)	State
Inlet of Pump/Outlet of Condenser	279	0.53168	370.32	1.5694	630.48	0.001586093135	0	Sat. Liq
Outlet of Pump/ Inlet of Boiler	279.41	2.48	373.41	1.5694	631.36	0.001583882413		Sat. Liq
Outlet of Boiler/Inlet of Turbine	350	2.4675	1703.8	5.65	17.136	0.058356676		Superheated
Outlet of Turbine/Inlet of Condenser	281	0.57068	1489.9	5.65	5.0311	0.1987636898	0.9	2-Phase

Mass Flow Rate Influence on System

Using Mass Flow Rate as 1 kg/s

Mass Flow Rate	Vel (m/s)	
1	5	
Area Based on Mass Flow Rate of 1 kg/s & Vel (5 m/s)		
0.0116713352	**m^2**	
Diameter of Pipe Based on Information Above (m)		
0.1984366 m	19.84 cm	
Work Out of Turbine Based on Mass Flow Rate of 1kg/s		
214	kJ/sec	
214	kW	
0.2139	MW	

Desired MW vs. Mass Flow Rate vs. Area vs. Diameter

Therefore,			
Desired MW	*Mass Flow Needed (kg/s)*	Desired MW	
1	4.675081814	11	51.42589995
2	9.350163628	12	56.10098177
3	14.02524544	13	60.77606358
4	18.70032726	14	65.4511454
5	23.37540907	15	70.12622721
6	28.05049088	16	74.80130902
7	32.7255727	17	79.47639084
8	37.40065451	18	84.15147265
9	42.07573633	19	88.82655446
10	46.75081814	20	93.50163628

Area Based on,					
Desired MW	*Area Needed*	Pipe Diameter	Desired MW	Area Needed	Pipe Diameter
1	4.099485741	2.284651345	11	45.09434315	7.577331288
2	8.198971482	3.230984917	12	49.19382889	7.914264414
3	12.29845722	3.957132207	13	53.29331463	8.23742757
4	16.39794296	4.56930269	14	57.39280037	8.548382581
5	20.49742871	5.108635712	15	61.49228612	8.84841661
6	24.59691445	5.596230035	16	65.59177186	9.138605379
7	28.69640019	6.044619291	17	69.6912576	9.419858812
8	32.79588593	6.461969834	18	73.79074334	9.692954751

9	36.89537167	6.853954034	19	77.89022908	9.958564333
10	40.99485741	7.224701909	20	81.98971482	10.21727142

Calculations for simulation 6

Simulation 6						
Working Fluid	***Pure Ammonia***		**w, Work In (kJ/kg)**	**w, Work Out (kJ/kg)**	**qin, Heat In (kJ/kg)(**	**qout, Heat Out in (kJ/kg)**
Mass Flow Rate (Inlet of Turbine kg/s)		Pump	10.2			
Turbine Inlet Temperture (K)	370	Boiler			1380.68	
Ideal Power Out of Turbine MW	15	Turbine		254		
nth, Cycle Efficiency		Condenser				1137.28
Thermal Eff., nth	*%*	Power Eff.	%			
0.1762899441	17.62899441	0.9597791798	95.97791798			
Ammonia						
cv (kJ/kg K)	*cp (kJ/kg K)*					
1.66	2.19					

State Table for T3 370K

STATE	Temperature (K)	Pressure (MPa)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Density (kg/m^3)	specific volume (v)	Quality (x)	State
Inlet of Pump/Outlet of Condenser	279	0.53168	370.32	1.5694	630.48	0.001586093135	0	Sat. Liq
Outlet of Pump/ Inlet of Boiler	281	2.6056	380.52	1.5972	628.81	0.001590305498		Sat. Liq
Outlet of Boiler/Inlet of Turbine	370	2.6056	1761.2	5.7881	16.597	0.06025185274		Superheated
Outlet of Turbine/Inlet of Condenser	283	0.61186	1507.6	5.7881	16.597	0.06025185274	0.9123	2-Phase

Mass Flow Rate Influence on System

Using Mass Flow Rate as 1 kg/s

Mass Flow Rate	Vel (m/s)	
1	5	
Area Based on Mass Flow Rate of 1 kg/s & Vel (5 m/s)		
0.01205037055	**m^2**	
Diameter of Pipe Based on Information Above (m)		
0.1238669499	0.001238669499	
Work Out of Turbine Based on Mass Flow Rate of 1kg/s		
254	kJ/sec	
254	kW	
0.2536	MW	

Desired MW vs. Mass Flow Rate vs. Area vs. Diameter

Therefore,			
Desired MW	*Mass Flow Needed (kg/s)*	Desired MW	mass flow
1	3.943217666	11	43.37539432
2	7.886435331	12	47.31861199
3	11.829653	13	51.26182965
4	15.77287066	14	55.20504732
5	19.71608833	15	59.14826498
6	23.65930599	16	63.09148265
7	27.60252366	17	67.03470032
8	31.54574132	18	70.97791798
9	35.48895899	19	74.92113565
10	39.43217666	20	78.86435331
60	236.5930599		

Area Based on					
Desired MW	*Area Needed*	Pipe Diameter	Desired MW	Area Needed	Pipe Diameter
1	3.457728707	2.098218445	11	38.03501577	6.95900331
2	6.915457413	2.967328982	12	41.49274448	7.268441904
3	10.37318612	3.634220952	13	44.95047319	7.565234191
4	13.83091483	4.19643689	14	48.40820189	7.850814544
5	17.28864353	4.691759075	15	51.8659306	8.126365094
6	20.74637224	5.139564559	16	55.32365931	8.39287378
7	24.20410095	5.551364202	17	58.78138801	8.651176274

8	27.66182965	5.934657963	18	62.23911672	8.901986945
9	31.11955836	6.294655335	19	65.69684543	9.145922163
10	34.57728707	6.635149315	10	69.15457413	9.383518149
60	2.851034042	1.905269627			