

Numerical Analysis of an Ocean Thermal Energy Conversion System: Preliminary Results

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Abstract

Global warming, increasing regulation, and hazardous working conditions associated with fossil fuel consumption for energy purposes continue to push renewable energy technologies to the forefront of the discussion of meeting the world's energy needs. Recently Ocean Thermal Energy Conversion (OTEC) has been one of the leading topics in the renewable energy platform. This project used Aspen Plus modeling software in order to create functioning models of OTEC systems and analyze the data collected to determine the overall power output and efficiency of the systems. The construction of models is of vital importance to the progression of OTEC because these systems require a tremendous amount of start-up and operating costs, and it is extremely important that any scenarios that may happen in the real world application be tested in modeling conditions. OTEC systems utilize the small temperature change required of refrigerants at specified pressures and the naturally occurring temperature gradient of ocean water from the surface waters to bottom waters in order to create a vapor power cycle to produce power. This project utilized the ideal Rankine cycle, in which certain conditions remain constant, in order to predict the efficiency and overall power output of an OTEC system. Once the models were built and fully functioning, a verification of the modeling techniques was performed using numerical calculations before any data was acquired. After verification of the two models was performed and the differences in the models and their verifying counterparts were deemed acceptable, data for the heat exchange rate for the condenser and evaporator, and work input and output for the pumps and turbine, respectively, was collected and analyzed in order to determine the overall power output and efficiencies of the two working models. The overall power output of each system was approximately 32 MW and 14 MW. Once verification of the model was complete, the student's acquired data for power output and efficiency of the OTEC system with an increasing ocean surface temperature. It was found that the power output increases linearly and the efficiency increases in a polynomial manner very slightly as the ocean surface temperature increases.

Introduction

Increased regulation and societal concern about the use of fossil fuels and the harmful by-products from using them for power generation have created an abundant need for alternative energy sources. Three engineering students at West Virginia Institute of Technology (WVUIT) have chosen to model OTEC

systems using Aspen Plus software to determine how the systems efficiency and power output changes with the rise and fall of ocean surface water temperature.

The first concept of OTEC was introduced by Jacques-Arsène d'Arsonval in 1870, but it was over 100 years later in 1979 when the U.S. implemented a Mini-OTEC on a barge off the coast of Hawaii. Although the barge only had a net output of 18 kW of power, barely enough to power the ship, it is considered the first large scale application of OTEC technology [1]. Since then several experimental OTEC plants have been built to demonstrate the concept, but very few have remained active and been scaled to a size that will have any kind of substantial power output. Currently the maximum active OTEC power plant gives 1 MW of power [2]. However there is still promise for this renewable energy resource, as Lockheed Martin and Reignwood Group are currently partnering up to build a 10 MW pilot plant off the shore of southern China [3].

Technological limitations are the primary drawback to OTEC plants being widely used. Not only are the systems very expensive, but calculated efficiencies of OTEC generally range from 6% to 7% with actual systems only yielding around 3% to 4% [4]. This means that in order to produce a substantial amount of power, the OTEC has to pump a lot of water. For an open system OTEC plant to produce 100 MW of power, 400 m³/s of 26°C water would have to be pumped a distance of 20 m along the surface through a 16 m diameter pipe and then recirculated to a depth of 1000 m to 4°C water and back again [1]. With this large amount of water movement there is also the environmental effects to consider. Along with these concerns, OTEC currently need certain ocean depths and temperature difference which are only found in select areas around the equator as seen in Figure 1.

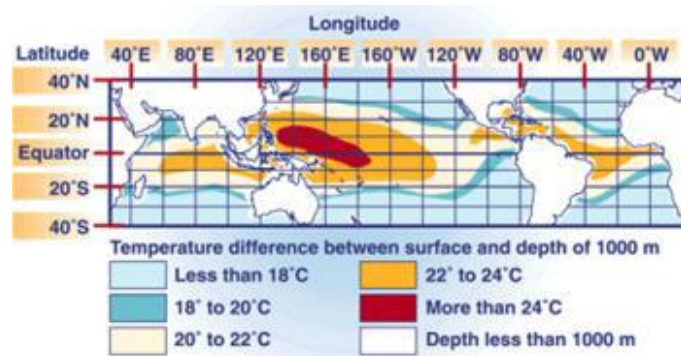


Figure 1. Distribution of global surface temperature [5]

Despite its drawbacks, OTEC systems still have promise. The energy that OTEC attempts to harvest is created by a temperature gradient between the warmer surface water of the ocean and the colder water on the bottom [1]. In order to produce any considerable energy a minimum temperature gradient of 20°C is the current standard [1]. The average year around surface temperature in tropical regions is approximately 28°C, and the temperature, at the standard used depth of 1000m, is relatively constant at 4°C [5]. In a sense this is a form of solar energy, but unlike other systems used to convert solar energy to electricity, the temperature difference between the surface and the ocean bottom is more stable from night-to-day and from season to season, meaning that OTEC systems can be run continuously where other forms of solar energy converters have prolonged down time when conditions don't provide ample sunlight [6]. It is estimated that approximately 10⁷ MW of power could be provided from ocean thermal energy. This abundance of renewable, continuous power is what makes OTEC research and development such a promising field.

Theory

In order to better understand the closed OTEC system power cycle that was modeled, it is easier to first consider the steam power plant cycle in its simplest conceptual form. In a steam power plant cycle there are four main processes: the pump, boiler, turbine, and condenser [7]. First water is pumped to the boiler in order to increase the pressure, then boiled and turned to steam [7]. The steam then enters the turbine and expands to create mechanical work, and finally the steam at the outlet of the turbine is condensed back to saturated liquid water so it can undergo the process again [7]. The steam power cycle can be seen in Figure 2.

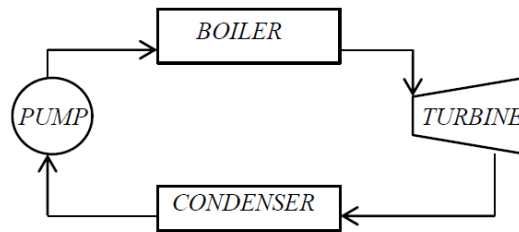


Figure 2. Steam power plant cycle

The steam power plant cycle is worth mentioning because the only difference in it and the OTEC cycle models in this report is that instead of a boiler during the heating process, an evaporator is used, which in most cases is a heat exchanger [1]. Although the concepts are very similar, there is a great difference in the working fluids, which makes application and implementation of each process very different, but for analysis purposes, the calculations and modeling techniques of ideal cycles are very similar. This is explained in greater detail in the Methods section of this report.

When the concept was first introduced by d'Arsonval it was based on a closed system where the working fluid, ammonia, is pumped from the low temperature at the bottom of the ocean to the high temperature at the surface [2]. Because ammonia has a very low boiling point, the difference in temperature of about 20 °C is enough to cause it to expand, at which point it is sent through a turbine where the energy is converted from mechanical to electrical energy. As mentioned above, the working fluid in a closed system will have to have properties that allow it to vaporize to a gaseous state at approximately 28 °C and condense back to a liquid state around 4 °C in order to produce a large enough pressure difference to create a worthwhile amount of mechanical work in the turbine. The most commonly used working fluids in the closed OTEC systems are ammonia and propylene, and although other refrigerants have been modeled, the majority of them are hazardous to the environment as well as any worker that might be exposed to them [8]. The hazardous nature of refrigerants other than ammonia has all but completely ruled them out as feasible working fluids [8].

More recently several experimental plants have been designed, and some tested, to use an open system where the ocean water is the working fluid. This process is much more complicated, however there are added benefits, because in the open system desalination is part of the process creating fresh water as a byproduct that can be pumped to main lands for irrigation and even to processing plants for clean drinking water [5].

The students decided on modeling a closed system due to the feasibility of modeling with their current thermodynamic knowledge. As previously stated, the closeness of the closed OTEC system to the steam power plant cycle makes for a very understandable thermodynamic cycle, of which the students had prior knowledge and felt confident modeling. Other complications that arise with modeling an open OTEC system are the desalination process and the even larger amount of ocean water that must be moved, the

movement of that which could have effects on the ocean wildlife in surrounding areas as well as the temperature of the ocean [9].

Method

In order to obtain accurate data from OTEC models using the Aspen Plus modeling software, first the modeling techniques had to be verified. In order to do so, two initial models were created. The first model was a Rankine power cycle, which was verified by hand calculations. The second was a Carnot power cycle, which mimicked the work of D. Bharathan's Carnot Cycle OTEC [1] model using the same software. Each model was verified based on percent error comparisons of the work and power consumptions and extractions for each process in the cycle. These verification models are discussed in further detail below.

Numerical Calculation Verification

As previously stated, the OTEC closed system power cycle is very similar to that of the steam power plant cycle. With this in mind, the first verification test was performed by making hand calculations based on an ideal Rankine OTEC cycle with ammonia as the working fluid. In the ideal Rankine cycle the processes occur as follows, along with the T-s diagram of the cycle, Figure 3 [7]:

- 1-2: Isentropic compression of working fluid to compressed liquid.
- 2-3: Heat transfer to the working fluid at a constant pressure as the fluid moves from compressed liquid to saturated vapor.
- 3-4: Isentropic expansion of the working fluid to a liquid-vapor mixture.
- 4-1: Heat transfer from the fluid at a constant pressure as the fluid moves from a liquid-vapor mixture to saturated liquid.

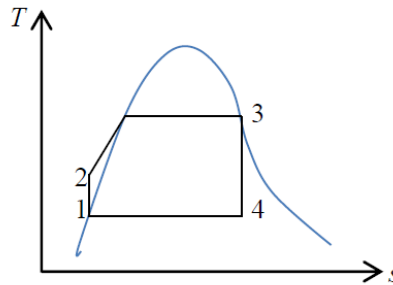


Figure 3. Temperature (T) vs. entropy (s) diagram of ideal Rankine OTEC power cycle

Roughly following the basic parameters set out in D. Bharathan's OTEC model [2], it was assumed that the pump inlet pressure was 6 bar, the turbine inlet pressure was 9 bar, and the mass flow rate of ammonia through the system was 580 kg/s. Also in doing the numerical calculations all the ideal Rankine cycle assumptions listed above were used.

The first step in this process is to identify the defining characteristics at each point (1, 2, 3, and 4) in the cycle. Included below are these defining characteristics along with a brief description of how they were found and/or calculated. Values from the ammonia saturated vapor-liquid and superheated steam tables were taken from [7].

Point 1:

The pressure at 1 is specified in the assumptions, and T, h, v, and s are all values from the tables for saturated liquid ammonia at a pressure of 6 bar.

$$\begin{aligned}
P_1 &= 6 \text{ bar} \\
T_1 &= 9.27 \text{ }^\circ\text{C} \\
h_1 &= 223.32 \text{ kJ/kg} \\
v_1 &= 1.5982 \cdot 10^{-3} \text{ m}^3/\text{kg}
\end{aligned}$$

Point 2:

$$\begin{aligned}
P_2 &= 9 \text{ bar} \\
h_2 &= 223.800 \text{ kJ/kg}
\end{aligned}$$

The pressure at 2 is specified from the assumptions at 9 bar. In order to calculate h at 2, it was considered that the specific work of the pump is characterized by the following equation for the specific work of a control volume process (1).

$$w = \int_1^2 v dp = v(p_2 - p_1) \quad (1)$$

Maintaining the specific volume at point one and considering the given pressures at points 1 and 2, work can be calculated as $w = 0.4795 \text{ kJ/kg}$. Using this value in equation (2) where both work and h_1 are known, the enthalpy at point 2 is found.

$$w = h_2 - h_1 \quad (2)$$

$$\begin{aligned}
P_2 &= 9 \text{ bar} \\
h_2 &= 223.800 \text{ kJ/kg}
\end{aligned}$$

Point 3:

The pressure at 2 and 3 is equal because of the ideal Rankine cycle assumption of heat transfer to the working fluid at constant pressure. The values for s_3 and h_3 are both found using the superheated vapor tables and interpolation. Although the temperature at 3 is not necessary for the work and heat transfer calculations, it is noted here because it had to vary slightly from the ideal Rankine cycle assumption. In the ideal Rankine cycle, it is assumed that the state of the working fluid at 3 is saturated vapor, however, because the Aspen Plus software value for saturated vapor was different from the value in [8], the students chose to use a value of $T_3 = 22 \text{ }^\circ\text{C}$ instead of $21.52 \text{ }^\circ\text{C}$, the value listed in the table. Even though this is a slight deviation from the intended cycle, it is still a valid argument for the verification with a difference in temperature of less than $0.5 \text{ }^\circ\text{C}$.

$$\begin{aligned}
P_3 &= 9 \text{ bar} \\
s_3 &= 5.0722 \text{ kJ/kg-K} \\
h_3 &= 1462.40 \text{ kJ/kg-K} \\
T_3 &= 22 \text{ }^\circ\text{C}
\end{aligned}$$

Point 4:

The pressure at 1 and 4 is equal because of the ideal Rankine cycle assumption of heat transfer from the working fluid at constant pressure. It is visible from Figure 3 that the process from 3 to 4 is at a constant entropy, therefore $s_3 = s_4$. Considering the state of the working fluid is a liquid-vapor mixture at 4, the quality of the mixture can be calculated using equation (3), and using that quality along with values from the tables and equation (4), the enthalpy at 4 is calculated.

$$s = s_f + \chi(s_g - s_f) \quad (3)$$

$$h = h_f + \chi(h_g - h_f) \quad (4)$$

$$\begin{aligned}
P_4 &= 6 \text{ bar} \\
s_4 &= 5.0722 \text{ kJ/kg-K} \\
\chi_4 &= 0.9658 \\
h_4 &= 1409.12 \text{ kJ/kg-K}
\end{aligned}$$

Once each point was fully defined, the work input to the pump, the work output of the turbine, and the heat transfer to and from the working fluid were calculated using the following equations.

$$\dot{W}_{pump} = \dot{m}(h_2 - h_1) \quad (5)$$

$$\dot{W}_{turbine} = \dot{m}(h_3 - h_4) \quad (6)$$

$$\dot{Q}_{evaporator} = \dot{m}(h_3 - h_2) \quad (7)$$

$$\dot{Q}_{condensor} = \dot{m}(h_4 - h_1) \quad (8)$$

Once the values for power and heat transfer rate were calculated from the numerical calculation, an OTEC model was built in Aspen plus following the same cycle with the same assumptions made for temperature, pressure, and ideal Rankine cycle and the results of each were compared. A detailed schematic of the model can be found Figure 4. Table 1 shows this comparison using percent difference, the equation for which is listed below.

$$\% \text{ Difference} = \frac{|\text{Actual} - \text{Calculated}|}{\text{Actual}} \times 100\% \quad (9)$$

Table 1. Verification of OTEC model with numerical calculations

Verification of OTEC Model with Numerical Calculations				
		Numerical	Aspen Model	Percent difference
Heat Exchangers	Evaporator (kW)	717558	727634	1.40
	Condensor (kW)	687764	695146	1.07
Mechanical Devices	Pump (kW)	278	278	0.00
	Turbine (kW)	30902	32766	6.03

From Table 1, it is clear that the percent difference from the model to the numerical calculations is acceptable. The only percent difference that is larger than 5 % is that of the turbine. This is still acceptable though, and it is important to consider that the method that the Aspen plus software uses to calculate the power output could be a very complicated iterative solution that could not be duplicated by hand calculations in a timely manner.

The only item not taken into consideration in this verification is the flow rate of warm and cold water to the evaporator and condenser, respectively. Although these values are important when considering the overall power output, since this verification focuses solely on the thermodynamic cycle of the working fluid, these flow rates are not necessary to include in the verification process. The flow rates were chosen by finding a value that ensured enough heat could be transferred to and from the working fluid in the evaporator and condenser.

OTEC Model Verification

The second verification process that was performed used the OTEC model created in [2] and a duplicate model that was created to ensure that the values were reasonably close. In order to do so, the students used temperature and pressure values from [2] and the knowledge of the OTEC power cycle in order to create a model that mimicked that of [2]. Once the model was created, a similar table to the numerical calculation was created in order to ensure that the values were matched up properly. This table, Table 2, is included below, along with Figure 5, the OTEC model created by [2]. The students Aspen Plus model and detailed schematic is seen in Figure 6.

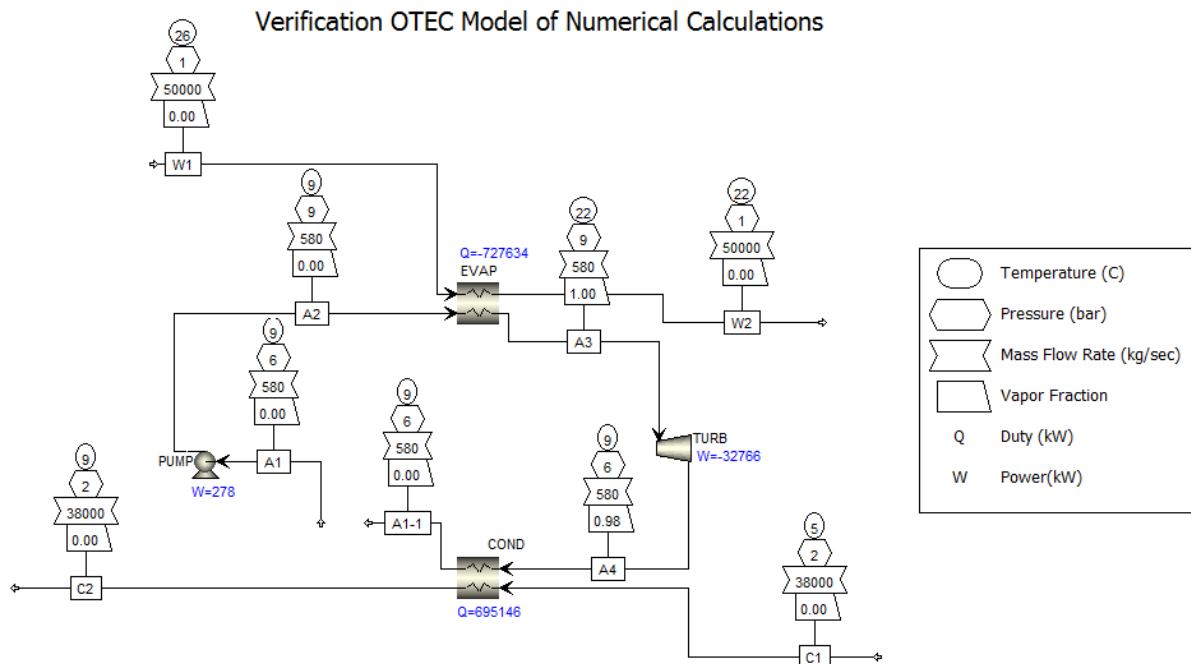


Figure 4. Numerical Calculation OTEC Model

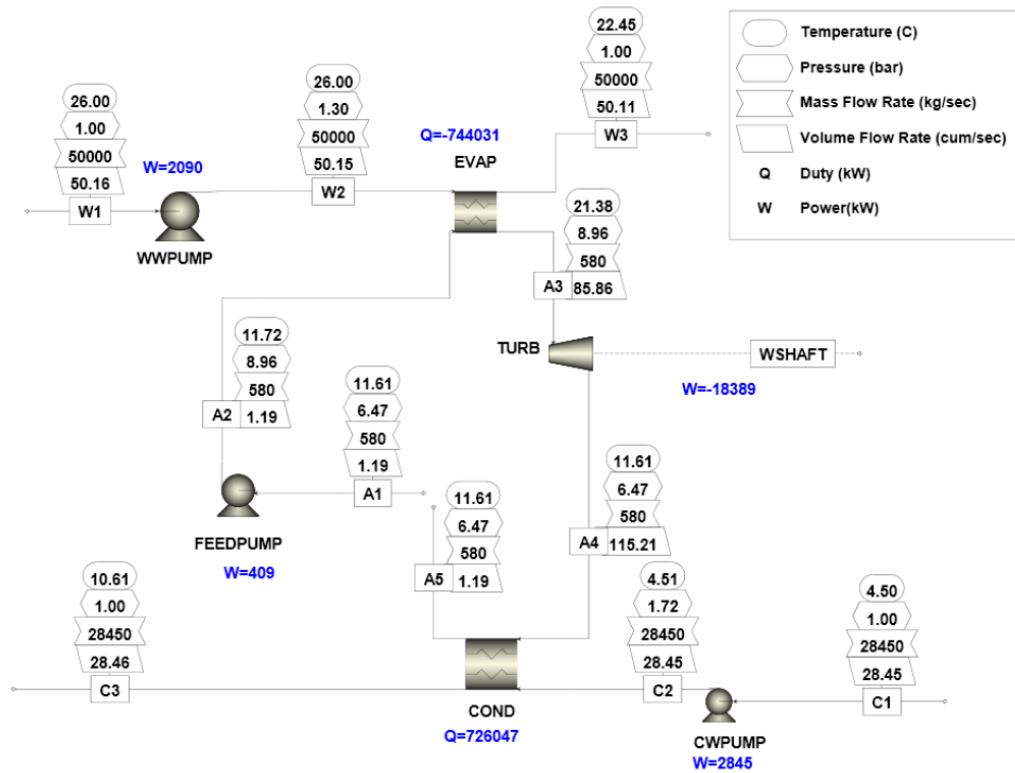


Figure 5. D. Bharathan's OTEC model [2]

Verification OTEC Model of D. Bharathan's Results

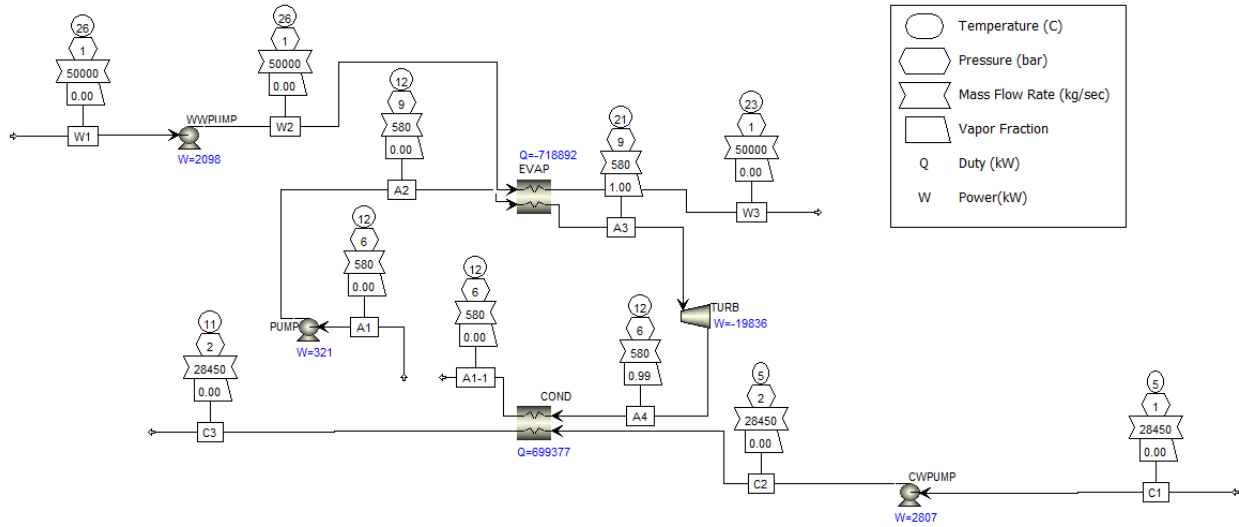


Figure 6. Verification OTEC model for [2]

Table 2. Verification of OTEC model with D. Bharathan's Results

Verification of OTEC Model with D. Bharathan's Results				
		D. Bharathan	Students	Percent difference
Water Pumps	Warm water pump (kW)	2090	2098	0.38
	Cold water pump (kW)	2845	2807	1.34
Heat Exchangers	Evaporator (kW)	744031	718892	3.38
	Condenser (kW)	726047	699377	3.67
Mechanical Devices	Pump (kW)	409	321	21.52
	Turbine (kW)	18389	19836	7.87

In order to create the model the students had to fix the pump inlet pressure and temperature as well as the condenser outlet pressure. Also fixed in this model was the ocean surface and bottom temperatures and the flow rate of ocean water. In doing so, the rest of the blocks in the model could run smoothly and the validity of the model could be determined.

As is evident in Table 2, once again the mechanical work processes of the pump and the turbine are troubling. The input work to the pump is 21.52%, however, the work in the pump is so small compared to the other work and heat transfer calculations that if this model were to be used for data acquisition, it would still be a reasonable fit. Overall these numbers are acceptable for the desired data that can be acquired from the models, and that the modeling techniques used in this report are valid for data acquisition.

Simulation of Changing Ocean Surface Temperature

Once the students had verified their modeling techniques the next step in the process was data acquisition. The students selected the numerical calculation verification model as a base and gathered data for ocean surface temperatures from 25 °C to 35 °C in increments of 1 °C. In creating this model, the students fixed the bottom ocean temperature at 5 °C, the outlet of condenser and inlet of pump

temperature and pressure at 9.27 °C and 6 bar, respectively, and the difference in temperature of ammonia and ocean water at the outlet of the evaporator at 1.2 °C. This value of temperature difference was cited in [1] as a standard heat exchanger value. Also important to note here is that the isentropic and mechanical efficiencies of the pump and turbine are set to 1, so the model portrays an ideal Rankine cycle with no losses due to irreversibility during expansion or compression, and the outlet pressures of the pump and turbine are fixed at 9 bar and 6 bar respectively in accordance with the ideal Rankine cycle.

RESULTS

Verification

Once the verification process was complete, the students were able to analyze their models based on overall power output and efficiency of each of the models created. In order to calculate the efficiency of each cycle, Equation (10) was used. The results are included in Table 3.

$$\eta = \frac{W_{net}}{Q_H} \quad (10)$$

Energy Analysis of OTEC Models		
	W _{net} (kW)	Efficiency (%)
Numerical Calculations Model	32488	4.53
OTEC Verification Model	14610	2.03

Table 3. Energy Analysis Results of OTEC Models

These results are normal compared to the information given in the Introduction and Theory sections of this report. However it is important to note a couple of things here.

The first noteworthy item is that the OTEC model used in the numerical calculations verification does not include pumps that push the warm and cold ocean waters into the evaporator and condenser. Had these values been included in this calculation, the efficiency of the numerical calculation model would have been lower.

Simulation of Changing Ocean Surface Temperature

Using Equations 5 through 10 and simulation data from the previously mentioned OTEC model, the students found that power output increases linearly and constantly as the ocean surface temperature increases, and, although very slightly, efficiency of the cycle also increases. The results have been included in Figures 7 and 8, charting power output and efficiency, respectively, as a dependent variable and ocean surface temperature as the independent variable.

The simulation data shows that both power output and efficiency increase slightly as the ocean surface temperature rises. Included in the plots is a best fit curve. These curves show that where power output increases linearly with an increase in ocean surface temperature, the efficiency of the OTEC system increases as a polynomial function with respect to the rising ocean surface temperature. Both best fit curves show and R² value of greater than 0.99 showing that they are good fits.

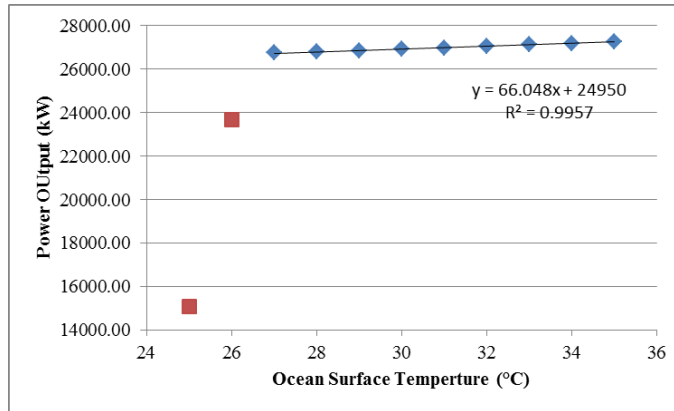


Figure 7. Power Output vs. Ocean Surface Temperature

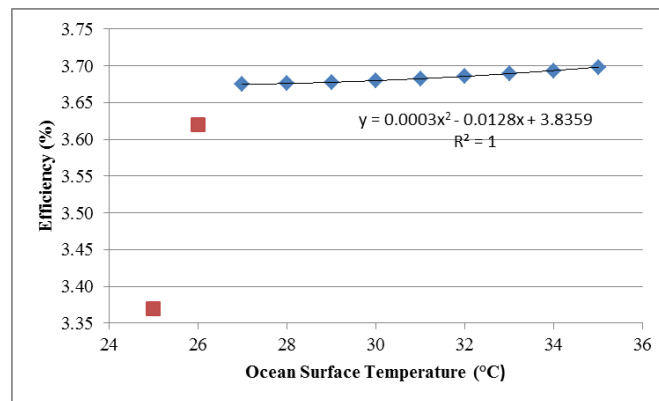


Figure 8. Efficiency vs. Ocean Surface Temperature

It must also be noted that the two data points for 25 °C and 26 °C are denoted using different markers and are not included in the best fit curves. This is because the quality of ammonia entering the turbine at 25 °C and 26 °C is 0.591 and 0.893 respectively. Sending an ammonia liquid steam mixture of these qualities into the turbine is unrealistic because the damaging effects of the liquid droplets of ammonia on the turbine blades and surface would cause the turbine to fail in a short amount of time and would be inefficient, as seen in Figure 8, for the short period of time the turbine could function. Despite these facts the points are included with the data to show that for this model with the fixed parameters mentioned previously, it is not feasible to run the OTEC system with an ocean surface temperature of less than 27 °C.

It is important to consider the nature at which each the power output and efficiency increase with an increase in ocean surface temperature. For the power output the best fit curve shows a linear increase, where the efficiency increases in a polynomial manner. If it is assumed that the enthalpy increases linearly with temperature, the linear increase for power output makes sense because as temperature of the ocean surface increases so too does the enthalpy of ammonia at the inlet of the turbine. The efficiency, on the other hand does not increase in a linear fashion. If taken into account the effects of ocean surface temperature on the evaporation process as well as the turbine output power, it makes sense that the best fit curve for efficiency is more complicated because both the heat transfer rate into the evaporator and the power produced in the turbine increase, each at its own rate with respect to the increase in ocean surface temperature.

CONCLUSIONS

The students were able to create a working model of an OTEC system using the Aspen Plus software and were able to use the model in order to collect data as to how the system would operate with changing surface water temperatures. The conclusions they were able to draw from the data gathered were pertaining to the nature of the increase in power output and efficiency of the system as well as the lower limit of operating conditions for a system with the given pressures and flow rates.

Although the power output and efficiency of the OTEC system increase with rising ocean surface temperatures, this increase is very slight and especially for efficiency. It seems to be a good assumption that given these conditions that the OTEC system would be able to handle the power output increases displayed in Figure 7 because they are so slight. Also the quality of the vapor gas mixture exiting the turbine was never found to be less than 0.98, which is reasonable for the turbine to operate properly without extensive mechanical damage occurring on the inside of the turbine due to the formation of liquid ammonia droplets inside the turbine as expansion occurs.

However, most notable from these findings is the apparent lower limit of the ocean surface temperature of 27 °C. This value could prove to be a large contributing factor to the sites at which the OTEC system given similar parameters could be placed as well as the type of temperature drop that the system could handle effectively. These drops in temperature could be due to severe weather or seasonal change, but they must be taken into account so as not to damage or even destroy the turbine.

DISCUSSION

The importance of these types of modeling techniques and data acquisition is crucial in the current OTEC design field. As stated in the introduction, OTEC systems can be very expensive and complicated, and maintenance on these systems can be difficult, especially on elements of the system that are completely submerged and fixed at great depths in the ocean. For this project the students chose to explore how the changing surface temperature of the ocean can affect the efficiency and power output of the OTEC system.

This shows only one aspect of the capabilities of modeling an OTEC system and how these models can affect design and implementation of a system. Other considerations and data that could be collected for future projects could include, but may not be limited to, change in bottom temperature, introduction of isentropic and mechanical efficiencies to the expansion and compression processes, and varying heat exchange rates and flow rates of ocean water and ammonia for better technologies. The possibilities for this type of modeling are endless and could have a major effect on how an OTEC system is ultimately designed and ultimately implemented.

Also important to note here is that the limited time frame the students had to work with the models and the Aspen Plus software made it hard to implement everything they would have liked to in order to present a more accurate model of a fully functioning system. The software has the capability of introducing pump and turbine efficiency curves for more a more accurate assessment of the mechanical work required or produced in these systems, as well as several heat transfer models that could be explored and used in order to more accurately assess the evaporator and condenser components. The modeling possibilities are endless. Although the students did not have the time to explore all these options, the fact that they were able, with the help of their advisor, to learn the basic functions of Aspen Plus and create a working model, all outside of the scheduled class time, shows the type of achievement undergraduate students can have when they are motivated and have a motivated faculty behind them. It is the hope of the students that these types of projects are continued at WVU TECH and institutions all over the country, because these types of independent learning activities that show the full potential of undergraduate engineering students.

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