Ocean Thermal Energy Conversion: Final Design Report

Zeynep Ay, Liam O'Buachalla,
Athanasios Pantazopoulos, Sofie Roger

Chemical Engineering Thermodynamics 1
12/11/2022

Part A: Designing an Ideal Nuclear Cycle

Table 1: Ideal Nuclear Cycle Stream Data

State	P (Bar)	T (°C)	Phase(s)	f_v	$\hat{v}$ (m^3/kg)	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	115	322	Superheated Vapor	--	.015719	-13267.7	-3.89255
2	65	280.86	Saturated Liquid-Vapor	.886	.026502	-13366.8	-3.89255
3	65	280.86	Saturated Liquid	0	.001336	-14730.8	-6.35462
4	115	282.648	Subcooled Liquid	--	.001326	-14724.2	-6.25462

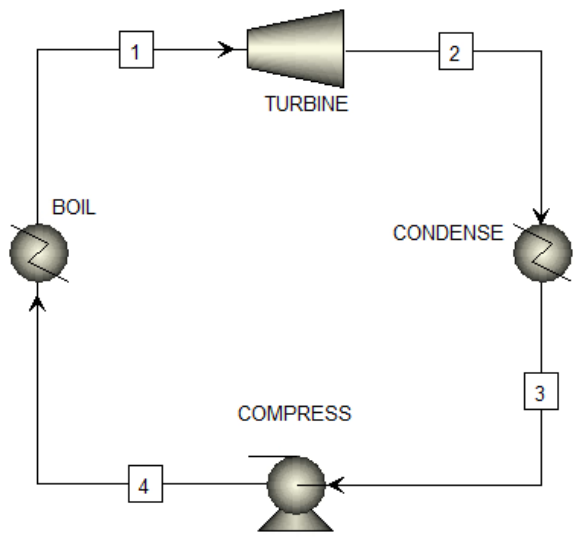


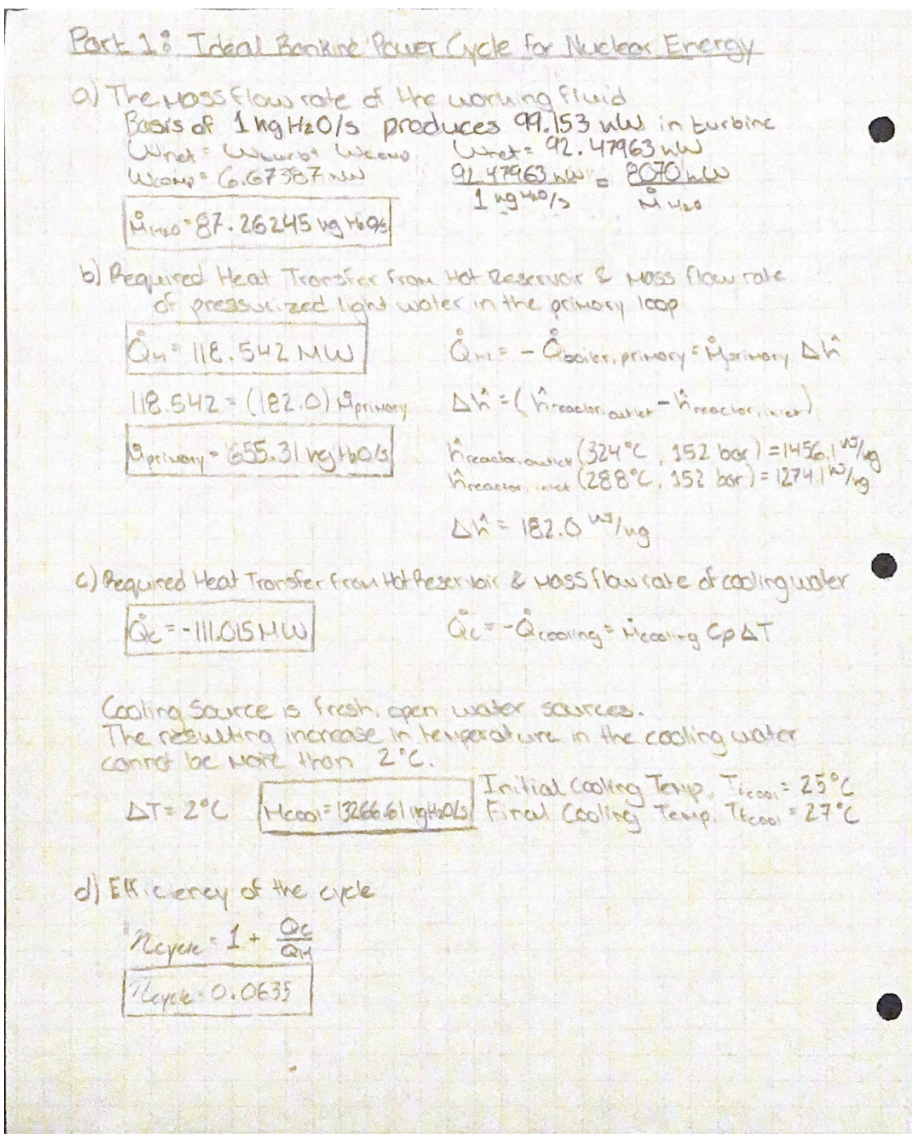
Figure 1: AspenPlus flowsheet for Rankine Power Cycle

All values were determined by modeling the process in Aspen Plus. The turbine and compressor are isentropic, operating at the respective efficiency of 100%.

The pressure of state one was determined by maximizing the pressure of the stream without the water condensing from a vapor to a liquid. The maximization of pressure would also contribute to larger work outputs from the turbine since the work is proportional to the changes in pressure of the working fluid. The temperature of state one was determined by the given temperatures of the primary loop that heats the working fluid in the boiler. The primary loop has a maximum temperature of 324 °C, which will heat the working fluid to an almost identical value and will not exceed it. Therefore, the temperature of the working fluid in state one should be a little less than the inlet temperature for the primary loop in the boiler.

Using Aspen Plus and the given pressure drop minimum of 50 bar, the conditions of the second stream were determined. The condenser operated isobarically and isothermally, conducting a phase change to create a stream with just liquid. Streams one and four see the same pressure, meaning that the pressure increase of the compressor needs to be the same value as the pressure decrease of the turbine, 50 bar. Using the aforementioned conclusions, Aspen Plus was able to determine any intermediate values for this cycle.

For the condenser, cooling water is needed to perform a phase change. Using the determined value of the required cooling heat and an estimated change in temperature of 2ºC, the flow rate of the coolant was determined. Considering that the coolant needs to be accessible and that the use of water would prevent any cross contamination or pollution. Most water sources sit at 25ºC in equatorial territories, considering that the design cycle occurs in a setting that is similar to OTEC. To prevent any damage to oceanic ecosystems, the temperature change of the coolant was limited to 2ºC. Therefore, the inlet temperature of the coolant is 25ºC



Part 1: Ideal Rankine Power Cycle for Nuclear Energy

a) The mass flow rate of the working fluid

Basis of 1 kg H_2O/s produces 99.153 kW in turbine

$W_{net} = W_{turbine} + W_{comp}$ $\quad W_{net} = 92.47963$ kW

$W_{comp} = 6.67587$ kW $\quad \frac{92.47963 \text{ kW}}{1 \text{ kg } H_2O/s} = \frac{8070 \text{ kW}}{\dot{m}_{H_2O}}$

$\dot{m}_{H_2O} = 87.26245$ kg H_2O/s

b) Required Heat Transfer from Hot Reservoir & mass flow rate of pressurized light water in the primary loop

$\dot{Q}_H = 118.542$ MW $\quad \dot{Q}_H = -\dot{Q}_{boiler,primary} = \dot{m}_{primary} \Delta\hat{h}$

$118.542 = (182.0)\dot{m}_{primary}$ $\quad \Delta\hat{h} = (\hat{h}_{reactor,outlet} - \hat{h}_{reactor,inlet})$

$\dot{m}_{primary} = 655.31$ kg H_2O/s $\quad \hat{h}_{reactor,outlet}(324°C, 152 \text{ bar}) = 1456.1$ kJ/kg

$\hat{h}_{reactor,inlet}(288°C, 152 \text{ bar}) = 1274.1$ kJ/kg

$\Delta\hat{h} = 182.0$ kJ/kg

c) Required Heat Transfer from Hot Reservoir & mass flow rate of cooling water

$\dot{Q}_C = -111.015$ MW $\quad \dot{Q}_C = -\dot{Q}_{cooling} = \dot{m}_{cooling} C_p \Delta T$

Cooling Source is fresh, open water sources.
The resulting increase in temperature in the cooling water cannot be more than 2°C.

$\Delta T = 2°C$ $\quad \dot{m}_{cool} = 1326661$ kg H_2O/s $\quad$ Initial Cooling Temp, $T_{i,cool} = 25°C$ Final Cooling Temp, $T_{f,cool} = 27°C$

d) Efficiency of the cycle

$\eta_{cycle} = 1 + \frac{Q_C}{Q_H}$

$\eta_{cycle} = 0.0635$

Part B: Design for Improved Cycle for Renewable Energy Source

Strategy 1: Changing the Working Fluid

Closed Cycle OTEC (CC-OTEC) power systems operate as cyclic heat engines wherein a pure working fluid ideally evaporates at the temperature of the warm sea water. The gas subsequently expands to a lower pressure as it passes through the system's turbine to produce work. The net electrical power produced by the system must equal the difference between the rates of heat transfer from the warm surface water to the cold deep water. Given CC-OTECs low theoretical efficiency, successful implementation of OTEC power generation requires deliberate engineering to recover more energy when the vapor expands through the turbine than is consumed in re-pressurizing the liquid. To improve efficiency and to reduce system costs working fluids that have a low boiling temperature are used in the process.

The preliminary design for the CC-OTEC cycle located in Iba, Philippines was modeled in Aspen Plus to test the various working fluids: Ammonia, R32, R600a, R245fa, R152a, R134a. The power output from the turbine was compared. The plant's location constrains evaporation and condensation temperatures to 27°C and 7°C respectively. The size of the OTEC was also constrained to the initial design, holding the flow rate constant at 89.567 kg/s. The change in mass flow rate from the idealized mass flow rate of 100.41kg/s that was hand calculated as part of the preliminary report can be attributed to the fact that Aspen uses different changes in enthalpy in their database.

Part 1:

Using the NIST Database, pressure, enthalpy and entropy values were obtained for each working fluid. Pressure values were manipulated to maximize the theoretical work produced by the turbine in Aspen for each working fluid.

Water taken from the surface of the ocean = 28˚C
Ocean water at depth of 1000m = 6˚C

The warm surface water of the ocean (28 ˚C) transfers heat to the ammonia working fluid, raising its temperature to 27˚C to make it a superheated vapor as it enters the turbine. Reasonable temperature changes in operating OTEC systems require minimum $\Delta T = 20$˚C, therefore:
State 1 and 4: 27 ˚C
State 2 and 3: 7 ˚C

The system is adiabatic between State 2 and State 3: $P_2 = P_3$
The turbine is isentropic, therefore $s_1 = s_2$
The compressor is isentropic, therefore $s_3 = s_4$
The quality of the working fluid at State 2 was calculated using the following formula:

Constraints of State 1: Maximize pressure without the superheated working fluid condensing to maximize work produced by the turbine

Constraints of State 2: The required change in temperature requires that the vapor be compressed to a pressure that gives 7 ˚C

Table 2: Working Fluid as Ammonia

State	P (Bar)	T(˚C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	10.65	27	Superheated Vapor	--	1630.2	5.7796
2	5.5363	7	Saturated Liquid-Vapor	0.947	1549.02	5.7796
3	5.5363	7	Saturated Liquid	0	378.16	1.6002
4	10.65	7.11	Subcooled Liquid	--	378.97	1.6002

Table 3: Working Fluid as R32

State	P (Bar)	T(˚C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	17.819	27	Superheated Vapor	--	516.25	2.0580
2	10.115	7	Saturated Liquid-Vapor	0.93448	490.34	2.0580
3	10.115	7	Saturated Liquid	0	212.36	1.0440
4	17.819	7.11	Subcooled Liquid	--	212.62	1.0440

Table 4: Working Fluid as R600a

State	P (Bar)	T(˚C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	3.716	27	Superheated Vapor	--	590.57	2.31
2	1.9976	7	Saturated Liquid-Vapor	1	564.624	2.31
3	1.9976	7	Saturated Liquid	0	216.14	1.0581
4	3.716	7.11	Subcooled Liquid	--	216.40	1.0581

Table 5: Working Fluid as R245fa

State	P (Bar)	T(˚C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	17.819	27	Superheated Vapor	--	424.96	1.7547
2	10.115	7	Saturated Liquid-Vapor	1	402.07	1.7547
3	10.115	7	Saturated Liquid	0	208.89	1.0321
4	17.819	7.11	Subcooled Liquid	--	209.12	1.0321

Table 6: Working Fluid as R152a

State	P (Bar)	T(˚C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	17.819	27	Superheated Vapor	--	524.25	2.0864
2	10.115	7	Saturated Liquid-Vapor	0.97486	510.052	2.0864
3	10.115	7	Saturated Liquid	0	211.99	1.0431
4	17.819	7.11	Subcooled Liquid	--	212.13	1.0431

Table 7: Working Fluid as R134a

State	P (Bar)	T(˚C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	17.819	27	Superheated Vapor	--	413.34	1.7155
2	10.115	7	Saturated Liquid-Vapor	0.988397	398.32	1.7155
3	10.115	7	Saturated Liquid	0	209.47	1.0340
4	17.819	7.11	Subcooled Liquid	--	209.62	1.0340

Part 2 – Properties of Ideal Cycle

Part 3: Assume Ideal cycle $\eta_{turbine,isen} = \eta_{comp,isen} = 100\%$.

a) $\dot{m}_{all} = 89.567\ kg/s$ ← held constant in ASPEN simulation so NH_3 as working fluid would produce ~8MW in turbine.

☆ For all other working fluids, $\dot{m}$ to produce 8MW would be much greater than 89.567kg/s because $\dot{W}_{net} = -8MW = \dot{m}(\hat{W}_{turbine} + \hat{W}_{comp})$

where $\hat{W}_{turbine} = \Delta\hat{h} = \hat{h}_2 - \hat{h}_1$

and $\hat{W}_{comp} = \Delta\hat{h} = \hat{h}_4 - \hat{h}_3$

$\Delta\hat{h}_{1-2\ NH_3}$ and $\Delta\hat{h}_{3-4\ NH_3}$ > other working fluids that were considered

~~Most cost effective and environmentally friendly to have lower lifetime~~

b) heat transfer rate from hot reservoir to working fluid

$\dot{Q}_H = \dot{m}(\hat{h}_1 - \hat{h}_4)$

$\dot{Q}_{H\ ammonia} = 112.07MW$

$\dot{Q}_{H\ R32} = 27.20MW$

$\dot{Q}_{H\ R600a} = 33.51MW$

$\dot{Q}_{H\ R245} = 19.33MW$

$\dot{Q}_{H\ R152a} = 27.96MW$

$\dot{Q}_{H\ R134a} = 18.25MW$

mass flow rate of water supplied to boiler (kg/s)

$-\dot{Q}_H = \dot{m}_{H_2O} \int_{T_i}^{T_f} \bar{C}_{p\ H_2O}$) $\bar{C}_{p\ H_2O} = 4.184\ kJ/kg\text{-}K$

NH_3 : $\dot{m}_{H_2O} = 2.68 \times 10^4\ kg/s$

R32 : $\dot{m}_{H_2O} = 6.5 \times 10^3\ kg/s$

R600a : $\dot{m}_{H_2O} = 8.01 \times 10^3\ kg/s$

R245 : $= 4.62 \times 10^3\ kg/s$

R152a : $= 6.68 \times 10^3\ kg/s$

R134a : $= 4.36 \times 10^3\ kg/s$

c) heat transfer rate from ~~hot~~ cold reservoir to working fluid (MW) & mass flow rate of cooling water (kg/s)

$\dot{Q}_c = \dot{m}(\hat{h}_3 - \hat{h}_2)$

$\dot{Q}_{c\,NH_3} = -104.87\,MW$

$\dot{Q}_{c\,R32} = -24.90\,MW$

$\dot{Q}_{c\,R600a} = -31.21\,MW$

$\dot{Q}_{c\,R245} = -17.30\,MW$

$\dot{Q}_{c\,R152a} = -26.70\,MW$

$\dot{Q}_{c\,R134a} = -16.91\,MW$

$-\dot{Q}_c = \dot{m}_{H_2O} \int_{T_1}^{T_2} \bar{C}_{p\,H_2O}\, dT$

NH_3 : 2.51×10^4 kg/s

R32 : 5.95×10^3 kg/s

R600a : 7.46×10^3 kg/s

R245 : 4.135×10^3 kg/s

R152a : 6.38×10^3 kg/s

R134a : 4.04×10^3 kg/s

d) efficiency of cycle

$\eta_{cycle} = 1 + \dot{Q}_c / \dot{Q}_H$

$\eta_{NH_3} = 0.0642$

$\eta_{R32} = 0.0845$

$\eta_{R600a} = 0.0686$

$\eta_{R245} = 0.105$

$\eta_{R152a} = 0.045$

$\eta_{R134a} = 0.0734$

Table 7: Work Produced from ASPEN, Boiling Temperatures from NIST and Change in Specific Enthalpy from State 1 to 2

Working Fluid	Work Produced by Turbine (kW)	Boiling Temp (˚C)	$\Delta\hat{h}$ (kJ/kg)
Ammonia	-8142.33	-33.34	81.18
R 32	-2320.91	-51.7	25.91
R600a	-2323.9	-11.7	25.95
R245fa	-2050.29	15.3	22.89
R152a	-1271.67	-24.9	14.148
R134a	-1345.68	-26.3	15.02

Despite the fact that ammonia has relatively low efficiency rates, its heat transfer to and from the coolant is dramatically higher than other working fluids. Since the theoretical yield of an idealized CC-OTEC plant ranges between 4-6%, obtaining high yields of work is difficult. For the other working fluids that were considered in this analysis, to obtain the same high yield of 8MW, the working fluid flow rate would be difficult to accommodate structurally and economically. It can therefore be concluded that it was the strongest decision to choose ammonia as our working fluid for the preliminary design. Ammonia is the best working fluid due to the fact it has the largest change in enthalpy between states one and two. Between states one and two, ammonia is a saturated liquid-vapor mixture with a $f_v = 0.947$ that expands throughout the turbine to produce W_s. Since the turbine is assumed to be stationary and isentropic, a mechanical energy balance gives:

<u>turbine:</u> stationary, isentropic $\Rightarrow$ adiabatic, reversible, steady state

$$\Delta H + \cancel{\Delta E_k}^{0} + \cancel{\Delta E_p}^{0} = \cancel{Q}^{0} + W_s \qquad W_{turbine} = \Delta H \Rightarrow \hat{W}_{turbine} = \Delta\hat{h} = \hat{h}_2 - \hat{h}_1$$

The working fluid that gives the largest change in enthalpy will give the largest workout from the turbine.

Strategy 2: Addition of Reheater & Second Turbine After First Turbine

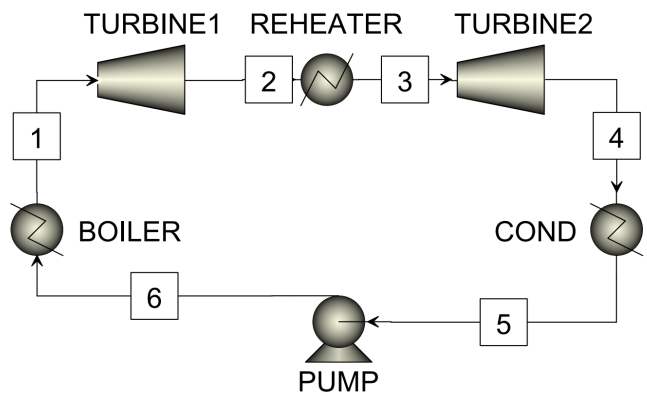


Figure 2: AspenPlus flowsheet for addition of reheater & 2nd turbine

State	P (Bar)	T (°C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg-K)
1	10.6266	27	superheated vapor	-	-2690.86	-6.94277
2	5.5363	7.06077	saturated liquid-vapor	0.960196	-2781.47	-6.94277
3	10.6266	27	superheated vapor	1	-2690.86	-6.94277
4	5.5363	7.06077	saturated liquid-vapor	0.96016	-2781.47	-6.94277
5	5.52445	7	saturated liquid	0	-3969.47	-11.1824
6	10.65	7.14131	subcooled liquid	-	-3968.66	-11.1797

Table 1: Data for streams through unit operations of cycle with reheater & 2nd turbine

Turbines and compressors are isentropic for ideal cycle.

W_{net} = -8.07 MW = $W_{s,\ turbine1} + W_{s,\ turbine2} + W_{s,\ comp} = \dot{m}_{NH3}*((\hat{h}_2 - \hat{h}_1) + (\hat{h}_4 - \hat{h}_3) + (\hat{h}_6 - \hat{h}_5))$ => **$\dot{m}_{NH3}$ = 44.7315 kg/s**

$\dot{Q}_H = \dot{m}_{NH3}*((\hat{h}_3 - \hat{h}_2) + (\hat{h}_1 - \hat{h}_6))$ = **61.211 MW = $\dot{Q}_H$**

$$-\dot{Q}_H = \dot{m}_{H_2O,\,heating}\left(\Delta\hat{h}_{reheater} + \Delta\hat{h}_{boiler}\right) = \dot{m}_{H_2O,\,heating}\left(\int_{T_i}^{T_F} C_p\,dT + \int_{T_i}^{T_F} C_p\,dT\right) \quad T_F = 27°C,\ T_i = 28°C$$

$$= \dot{m}_{H_2O,\,heating}\left(2C_p(27-28°C)\right) \quad C_{p,H_2O} = 4.184\ \frac{kJ}{kg\text{-}K} \Rightarrow \boxed{\dot{m}_{H_2O,\,heating} = 7314.9\ \frac{kg}{s}}$$

Warm water enters the boiler at 28°C, the temperature of the hot source, the surface of the ocean, and it provides heat to the working fluid to boil it to a superheated vapor. The warm water's

temperature decreases to 26°C, a ΔT of -1°C to minimize disruption to the marine environment once the warm water is returned to the ocean.

$\dot{m}_{H2O,\ heating}$ **= 7.3149×10³ kg/s**

$\dot{Q}_C = \dot{m}_{NH3} * (\hat{h}_5 - \hat{h}_4)$ = **-53.141 MW =** $\dot{Q}_C$

$$-\dot{Q}_C = \dot{m}_{H_2O,\ cooling}(\Delta \hat{h}_{condenser}) = \dot{m}_{H_2O,\ cooling}\left(\int_{T_i}^{T_f} C_{p,H_2O}\, dT\right) \quad T_f = 7°C,\ T_i = 6°C$$

$$= \dot{m}_{H_2O,\ cooling}\left(C_{p,H_2O}(7-6°C)\right) \Rightarrow \boxed{\dot{m}_{H_2O,\ cooling} = 12701\ \frac{kg}{s}}$$

Cooling water enters the condenser at 6°C, the temperature of the cold source at ~1000 meters below sea level, and it absorbs heat from the working fluid to condense it to a saturated liquid. The cooling water's temperature rises to 7°C, a ΔT of 1°C to minimize disruption to the marine environment once the cooling water is returned to the ocean.

$\dot{m}_{H2O,\ cooling}$ **= 1.2701×10⁴ kg/s**

$\eta_{cycle} = |W_{net}| / \dot{Q}_H$ = **0.13184 =** η_{cycle}

<u>Strategy 3: Decreasing Mass Flow Rate & Decreasing ΔT to Minimize Environmental Impacts</u>

Decreasing Mass Flow Rate:

In order to maximize the power output of an OTEC system, a minimum ΔT value of 20°C between the warm and cold water will be necessary. For this to be possible, in an idealized cycle, 30028 kg/s will have to be extracted at a constant rate from the warm reservoir and 28099 kg/s will have to be extracted from the cold reservoir for an 8.142 MW OTEC power plant. Although the cold-water intake will have low impacts to the biological environment due to the extraction occurring below the euphotic layer (1-200 m depth) in the South China Sea, the warm water intake has the chance to disrupt populations of phytoplankton, which are considered to be the most important marine primary producer at the bottom of the food web.

Although most modern-day OTEC systems do include screens to prevent marine species and debris from getting sucked into the plant, some organisms would get trapped in the intake screen (impingement) and organisms small enough to pass through the intake screen would end up trapped in the system (entrapment). This causes damage to the marine biological environment surrounding the OTEC system, as well as the OTEC itself. The warm-water intake velocity is a critical factor to the impingement and entrapment of vital organisms such as phytoplankton, with higher velocities causing more disruption to marine biological environments. The general guideline regarding velocities for conventional OTEC plant intakes is to keep the velocity below 15 cm/s so as to allow fish to escape the screen wells (U.S. Environmental Protection Agency, 1973). The standard pipe size for an OTEC plant tends to be 12 m in inner diameter. (Utsunomiyab, 2022)

Determining Volumetric Flow rate & Mass flowrate:

Typical OTEC Plant Pipe Inner Diameter: 12 m
Velocity recommended: 15 cm/s or 0.15 m/s

Determin Cross-sectional Area:

$A = \pi r^2$ ← Assuming perfect cylinder

$\downarrow$

$A = \pi (6\ m)^2$

$\downarrow$

$A = 113.097\ m^2$

Determine Volumetric flowrate:

$Q_v = \text{Velocity} \times \text{Area}$

$\downarrow$

$Q_v = 0.15\ m/s \times 113.097\ m^2$

$\downarrow$

$Q_v = 16.964\ m^3/s$

Determining Mass Flowrate:

$$\dot{m}_{H_2O} = \frac{16.964\ m^3}{s} \times \frac{997\ kg}{m^3} = \boxed{1.69 \times 10^4\ kg/s}$$

density of water

Decreasing ΔT:

At a mass flow rate of 16900 kg/s from the warm water reservoir and 28099 kg/s from the cold reservoir, the OTEC plant is projected to displace millions of gallons of water. Most of the water taken up by the OTEC is to be discharged back into the South China Sea in the form of a mixed discharge ranging around 14-16°C. Although the impacts of large water volume displacement is yet to be studied, there will be clear implications to surrounding ecosystems if the discharged water is to not fall within the ideal range (Myers, 1986). An example of an OTEC system off the coast of Hawaii using a discharge stream at 15°C is shown below:

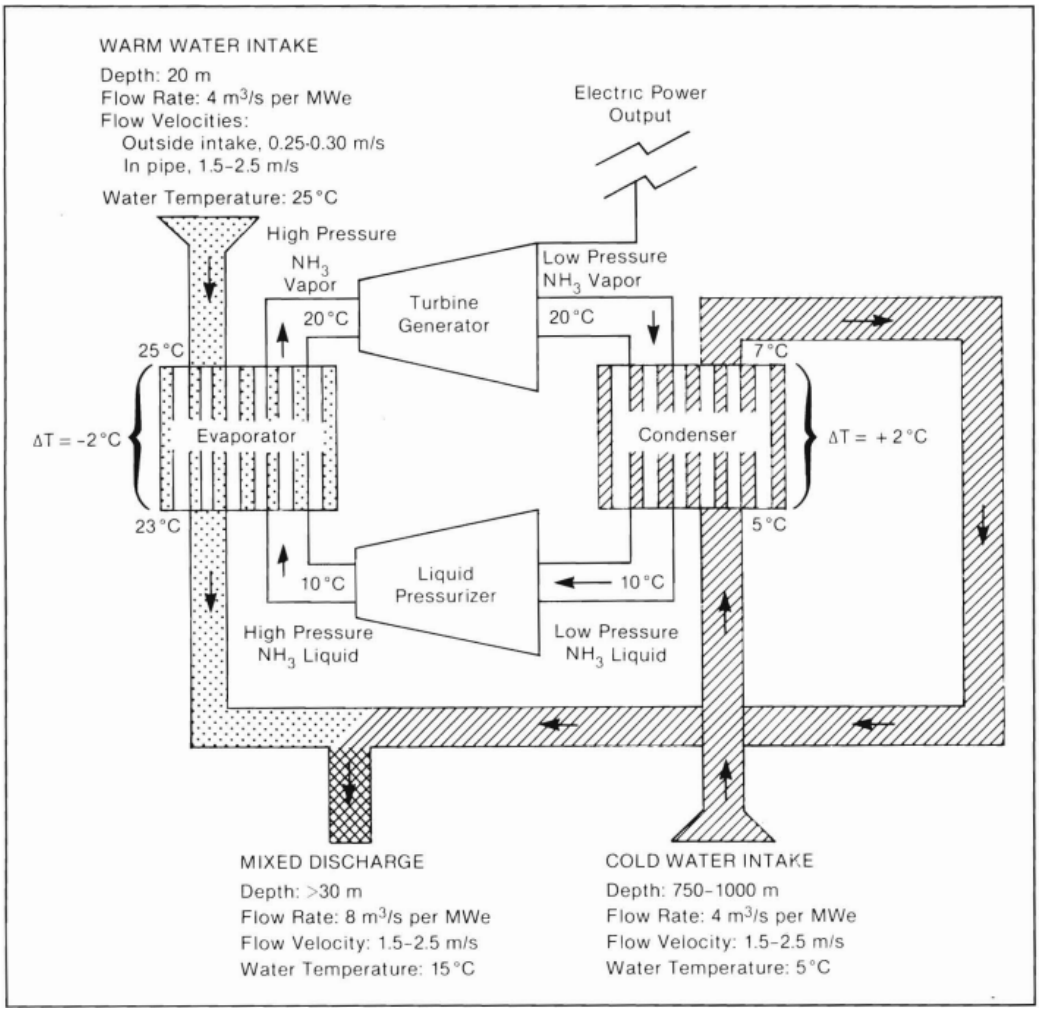


Figure 3: Example Hawaii OTEC System w/ Mixed Discharge (Myers, 1986)

Calculating out the discharge water temperature of the current OTEC system at a temperature of 27°C mass flow rate of 16900 kg/s discharged from the boiler and a temperature of 7°C and 28099 kg/s discharged from the condenser (assuming perfect heat transfer), the mixed discharge temperature would come out to be 19.48°C, much above ideal. To be able to minimize environmental damages and to have a mixed discharge value closer to the ideal range, the ΔT value can be decreased from 20°C to 19°C. Doing this gives us a discharge water temperature of

14.635°C, which is much more ideal.

Calculating Mixed Discharge Water temperature:

At $T_H = 27°C$ & $T_L = 7°C$, $\dot{m}_{H_2O,H} = 16900\ kg/s$ & $\dot{m}_{H_2O,C} = 28099\ kg/s$

Assuming perfect heat transfer:

$$WT_{mixed} = \frac{\dot{m}_{H_2O,H} T_H + \dot{m}_{H_2O,C} T_C}{\dot{m}_{H_2O,H} + \dot{m}_{H_2O,C}}$$

$$\downarrow$$

$$WT_{mixed} = 19.48\ °C$$

At $T_H = 26.5°C$ & $T_L = 7.5°C$, $\dot{m}_{H_2O} = 16900\ kg/s$ & $\dot{m}_{H_2O,C} = 28099\ kg/s$

$$WT_{mix} = \frac{\dot{m}_{H_2O,H} T_H + \dot{m}_{H_2O,C} T_C}{\dot{m}_{H_2O,H} + \dot{m}_{H_2O,C}}$$

$$\downarrow$$

$$\boxed{WT_{mix} = 14.635°C}$$

falls in 14°C–16°C ideal mixed temperature range

These new conditions are then applied to an ideal cycle in AspenPLUS, same assumptions Strategy 1 & 2 with a change in the temperature at each state.

Water taken from the surface of the ocean = 28˚C
Ocean water at depth of 1000m = 6˚C

The warm surface water of the ocean (28 ˚C) transfers heat to the ammonia working fluid, raising its temperature to 26.5˚C to make it a superheated vapor as it enters the turbine. Temperature change in OTEC systems for minimal environmental impacts requires $\Delta T = 19˚C$, therefore:
State 1 and 4: 26.5 ˚C
State 2 and 3: 7.5 ˚C

The system is adiabatic between State 2 and State 3: $P_2 = P_3$
The turbine is isentropic, therefore $s_1 = s_2$
The compressor is isentropic, therefore $s_3 = s_4$

State	P (Bar)	T (°C)	Phase(s)	f_v	$\hat{h}$ (kJ/kg)	$\hat{s}$ (kJ/kg*K)
1	10.499	26.5	Superheated Vapor	--	-2691.9	-6.939
2	5.6348	7.56	Saturated Liquid-Vapor	0.961	-2778.46	-6.939
3	5.6348	7.56	Saturated Liquid	0	-3966.23	-11.167
4	10.499	7.69	Subcooled Liquid	--	-3965.45	-11.167

Table 2: Stream Properties Results from AspenPlus for most Environmentally Beneficial OTEC Plant

Calculating Heat Transfer + Efficiency:

Heat transfer rate from cold reservoir to working fluid (MW):

$$\dot{Q}_C = \dot{m}_{NH_3}(\hat{h}_3 - \hat{h}_2) = (89.567\ kg/s)(-3966.23 - (-2778.46)\ kJ/kg)$$

$$\boxed{\dot{Q}_C = -106.41\ MW}$$

Mass flow rate is kept constant for means of comparison

Heat transfer rate from Hot reservoir to Working fluid (MW):

$$\dot{Q}_H = \dot{m}_{NH_3}(\hat{h}_1 - \hat{h}_4) = (89.567\ kg/s)(-2691.9 - (-3965.45)\ kJ/kg)$$

$$\boxed{\dot{Q}_H = 114.06\ MW}$$

Calculating Efficiency of cycle:

$$\eta_{cycle} = 1 + \frac{\dot{Q}_C}{\dot{Q}_H}$$

$$\boxed{\eta_{cycle} = 0.0674}$$

The work output of the turbine amounted to 7.7 MW.

Part C: Project Summary

Table 1: Comparing Performances of Modified & Improved Designs

Cycle	Working fluid flow rate (kg/s)	Heat transfer from hot source (MW)	Heat transfer from cold source (MW)	Primary loop flow rate (kg/s)	Coolant flow rate (kg/s)	Efficiency of cycle
Nuclear	87.26245	118.542	-111.015	655.31	13266.61	0.0635
OTEC w/ Ammonia	89.567	112.07	-104.87	26800	25100	0.0642
OTEC w/ R32	89.567	27.20	-24.90	6500	5950	0.0845
OTEC w/ R600a	89.567	33.51	-31.21	8010	7460	0.0686
OTEC w/ R245fa	89.567	19.33	-17.30	4620	4135	0.105
OTEC w/ R152a	89.567	27.96	-26.70	6680	6380	0.045
OTEC w/ R134a	89.567	18.25	-16.91	4360	404	0.0734
OTEC w/ 2 turbines	44.7315	61.211	-53.141	7314.9	12701	0.13184
OTEC w/ lower ΔT	89.567	114.06	-106.41	16900	28099	0.0674

<u>Conclusions from Analysis of All Designs</u>

Through considering potential improvements to our OTEC plant design and testing them to evaluate their efficacy, modifications to the preliminary design were identified that can increase the effectiveness of the power plant. Viable improvements to our cycle would increase its efficiency, meaning more work would be produced for a lower working fluid flow rate. Less working fluid would also mean heat transfers to and from it would decrease, so less coolant and interaction with a hot source would be required, facilitating operation of the OTEC plant. Finally, the quality of the stream from any turbine should be as close to 1 as possible to maximize the longevity of the turbine, as liquid in the outlet stream reduces a turbine's lifespan.

In Strategy 1, changing the working fluid for the cycle, ammonia was identified as the current best option, as it produces the highest power output, $W_{net} = W_{turbine} + W_{comp}$, for a given flow rate. This was demonstrated in Table 7 of Strategy 1, where for the flow rate of ammonia needed to generate 8.07 MW, the required energy for our location, the same flow rate did not produce nearly enough power with the other working fluids considered, R32, R600a, R245fa,

R152a, R134a, not even reaching half of the desired energy output. Any working fluid for the OTEC plant would also need to operate between the temperatures of 7 and 27°C, where a minimum ΔT of 20°C is needed to drive an OTEC plant to generate energy. The cold source, ocean water at a depth of ~1000 m, has temperature 6°C, and the hot source, the surface ocean water, is 28°C. Thus, a working fluid for our cycle would need to be able to be easily brought to a superheated vapor at 27°C and a subcooled liquid at 7°C, which these potential working fluids all satisfy. Ammonia then becomes the best current option for a working fluid, as it produces the most work for a given mass flow rate among the viable working fluids identified.

In Strategy 2, a reheater and second turbine were added following the first turbine, where the reheater would bring the ammonia working fluid back to a superheated vapor as it exited the first turbine to again be expanded in the second turbine, thus generating more work per cycle. Because more work was then produced for the same mass flow rate of ammonia, the amount of working fluid could be decreased. In the calculations for Strategy 2, the working fluid flow rate needed to generate 8.07 MW was 44.7315 kg/s, 49.94% of the original mass flow rate, 89.567 kg/s. The plant could then be operated more easily, as less working fluid, and therefore less coolant and interaction with a hot source for heat transfers to and from ammonia, would be required. From the calculations for strategy 2, to condense the smaller flow rate of ammonia, 1.2701×10^4 kg/s of coolant water was needed, 50.8% of the original value, 2.51×10^4 kg/s. Similarly, the amount of warm water to superheat the ammonia decreased by 72.71% from 2.68×10^4 kg/s to the improved cycle's value of 7.3149×10^3 kg/s. Reflected in the smaller $\dot{m}_{H2O,\ heating}$ value, heat transfer to superheat the working fluid, $\dot{Q}_H$, for strategy 2 was lower due to the reduced working fluid flow rate to heat. Thus, with the same power output, W_{net}, and smaller $\dot{Q}_H$, 54.62% its original value, the efficiency, $|W_{net}|/\dot{Q}_H$, increased 105.4% from 0.0642 to 0.13184. This improved design is over twice as efficient, meaning more work is produced for less heat transfer and working fluid. Additionally, the qualities of the turbine outlet streams increased 1.39% from that of the preliminary design, indicating this model has greater longevity as well.

For Strategy 3, to be able to minimize impacts to the biological marine environment surrounding the OTEC plant, the mass flow rate of the water extracted from the warm water reserve was decreased from 30028 kg/s to 16990 kg/s based on the recommended velocity rates from the U.S. Environmental Protection Agency. The ΔT was then decreased from 20˚C to 19˚C in order to decrease the mixed discharge temperature from 19.48˚C to 14.6˚C, decreasing potential environmental disruptions even further. The overall efficiency of the system increased from 0.0642 to 0.067, a 4.68% increase. The work output due to these changes decreased from 8.07 MW to 7.7 MW, a decrease of about 4.58%. Therefore, a higher working flow rate would be needed to reach 8.07 MW, which would also require more heat transfer and greater warm and cooling water flow rates. Although the system itself is not as highly performing as the other two strategies, minimizing the environmental impacts of the OTEC system will make it more viable for communities, and can help ensure that the benefits of the OTEC system outweigh the cons.

Finally, nuclear power was explored to be used as a comparison for what OTEC could replace as a common source of energy. The nuclear cycle yields an efficiency of 0.0635, which is

slightly lower than the efficiency of our original OTEC cycle, 0.0642. Nuclear power produces radioactive waste, but it is emission free. With some modifications, like those of Strategy 2, OTEC can yield even larger efficiencies that prove it a more viable energy source. It also has zero waste in addition to no emissions.

Overall, considering the potential modifications to our preliminary design and their performance data, an OTEC plant with ammonia as the working fluid and a reheater and second turbine after the first turbine appear to currently satisfy our community's energy demands best, while also remaining environmentally-friendly and cost-effective. With this design, our desired energy output is achieved more efficiently, using less materials and requiring less heat transfer, meaning disturbance of the surrounding environment is less likely. Additionally, the design will likely have a longer lifespan due to the higher quality of the turbine outlet streams.

Reflection & Future Directions

Ocean Thermal Energy Conversion is an environmentally-friendly, great alternative that can yield benefits beyond sufficient energy production for our town. The process is waste and emissions free and has limited potential impact on the surrounding environment, especially with our environmentally-conscious design.

The OTEC cycle requires few modifications from the preliminary design. It is able to use warm, sea level ocean water as an energy source for heating and cold ocean water at depth ~1000 m as a coolant. OTEC doesn't require an additional cycle to heat and cool the working fluid, resulting in a more efficient process. These conditions are especially optimal for an ammonia working fluid.

OTEC has other potential benefits beyond energy. The working fluid could also be used as a refrigerant for any external cooling, like refrigeration and air conditioning. Also, OTEC has desalination capabilities for the ocean water passing through the boiler and condenser. If OTEC is most optimal for an energy independent community, it could also provide the community with an additional source of freshwater. Generally, OTEC produces less energy than a nuclear power cycle, but this is the trade off of an environmentally friendly design. OTEC is ideal for smaller communities and can yield many more benefits beyond energy.

References:

Myers, E. P. . “The Potential impact of ocean thermal energy conversion (OTEC) on fisheries” U.S. Dept. of Commerce, National Oceanic and Atmospheric Administration, National Marine Fisheries Service. 1986.

U.S. ENVIRONMENTAL PROTECTION AGENCY. “Reviewing environmental impact statements - Power plant cooling systems, engineering aspects.” Pacific Northwest Environmental Research Lab., U.S. EPA, Corvallis, OR, 1973.

Utsunomiyab T., Hisamatsua R., “Coupled response characteristics of cold water pipe and moored ship for floating OTEC plant”, June 2022, https://www.sciencedirect.com/science/article/pii/S0141118722001006