TECHNO-ECONOMIC ANALYSIS OF CLOSED OTEC CYCLES FOR POWER GENERATION

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Abstract

In the present scenario of ever-increasing world electric power demand with increased attention to environmental issues, Ocean Thermal Energy Conversion (OTEC) plants, because of their enormous potential, are gaining interest.

In the present work, a techno-economic analysis of closed OTEC (CC-OTEC) cycles for electric power production is conducted. The effect of working fluid selection and number of evaporation levels from the techno-economic point of view are observed with simplified numerical models.

A more detailed analysis of the single level power plant with the selected working fluid is also reported. Sizing of heat exchangers in the configuration found by this detailed analysis is conducted and capital cost estimation for the power plant has been assessed. With the introduction of simplifying assumptions on electric energy produced by the plant, the levelized cost of energy (LCOE) is finally calculated.

Keywords: OTEC, Ocean, renewable energy, seawater, cold water pipe, CWP, closed cycles, DOW.
Sommario

Nell’odierno scenario di continua crescita della domanda elettrica mondiale e di una maggiore sensibilità per le problematiche ambientali, gli impianti che sfruttano la Ocean Thermal Energy Conversion (OTEC) per la produzione di potenza, a causa del loro enorme potenziale, stanno guadagnando un crescente interesse.

In questo lavoro è stata effettuata un’analisi tecnico-economica di impianti OTEC a ciclo chiuso (CC-OTEC) per la produzione di energia elettrica. L’effetto della scelta del fluido di lavoro e del numero di livelli di evaporazione dal punto di vista tecnico-economico sono stati osservati mediante l’uso di modelli semplificati dell’impianto.

Un’analisi più dettagliata di un impianto a un singolo livello con il fluido di lavoro migliore selezionato precedentemente è stata poi effettuata. In seguito, per la soluzione individuata da questa ultima analisi dettagliata è stato effettuato il dimensionamento degli scambiatori e una stima dei costi d’impianto. Il levelized cost of energy (LCOE) per l’impianto ottenuto è stato calcolato tramite l’introduzione di ipotesi semplificative.

Parole chiave: OTEC, Oceano, energie rinnovabili, acqua di mare, cold water pipe, CWP cicli chiusi, DOW.
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Due to the increase in global electric energy demand and the stipulation of new international climate agreements, renewable energy systems for power production are acquiring large interest. In this scenery, Ocean Thermal Energy Conversion (OTEC) power plants, which were intensively studied in the 1980s, are gaining new attention. OTEC power plants use the temperature difference between shallow warm tropical waters and cold deep seawater for energy conversion. Temperature difference between warm seawater and cold seawater for favorable sites is between 22°C and 25°C. It is estimated that the global OTEC potential can cover entirely the current world electricity demand. An area of approximately 60 million square kilometers is considered suitable for OTEC installation.

Two principal configurations have been proposed for OTEC plants: the open cycle and the closed cycle solution. In the open cycle solution warm seawater enters in a flash evaporator, it is expanded in a turbine and condensed by cold seawater at turbine outlet. Because of the flash evaporation of seawater, if a surface condenser is used, desalinated water is also produced. In the closed cycle configuration, a working fluid with favorable features is used generally in a saturated vapor Rankine cycle. The closed cycle solution has been considered the most promising in the near term because well-known components for the power block are used. For closed cycle applications, since large areas are required, titanium plate exchangers are worth of interest because of their compactness. Tests on aluminum, another promising material for heat exchangers, in seawater environment are currently underway. Despite technical feasibility has been demonstrated by the development of few pilot plants, no commercial size closed cycle OTEC plant have been developed yet because of high capital costs and uncertainties, especially related to the cold water pipe (CWP), used to collect cold deep ocean water (DOW).

In this thesis, a techno-economic analysis of closed OTEC cycles has been performed. A literature review on relevant plant components have been realized and a thermodynamic analysis of closed cycle OTEC plants have been conducted highlighting the efficiency limits of the plant. Exergy analysis of the single level Rankine cycle was performed. It emerged, as expected, that the larger destruction of exergy is due to heat transfer between seawater and working fluid at both the evaporator and condenser. An improvement in efficiency was
observed with the introduction of multiple evaporation levels. Because of high uncertainties on components’ cost and since heat exchangers cost is estimated to be a consistent part of plant cost (from 25% to 50%) the ratio between net power produced and heat transfer area was selected as objective function to be maximized in the techno-economic optimization. Such an objective function, traditionally called $\gamma$, is commonly used in literature for techno-economic optimization of closed cycle OTEC plants. According to the literature review conducted, the CWP diameter is limited because of high stresses endured by the effect of currents, waves and storms. A value of 2.5m for CWP diameter was selected for all the analyses. This value is the limit diameter for HDPE pipes. In order to reduce pressure losses in the pipe because of too high velocities, cold seawater flowrate was fixed to 8500kg/s for all the analyses. A numerical model of a single level closed cycle OTEC plant and an optimization tool, which optimized seawater temperature differences across heat exchangers and heat exchangers’ pinch points, have been developed. The plant model used simplifying assumptions on heat exchangers. A constant overall heat transfer coefficient was assumed and a simplified estimation of pressure drop on seawater side of the heat exchanger was introduced. The plant model and optimization tool have been used for working fluid selection by comparing maximum values of the objective function for different fluids. It resulted that, in accordance with most of researchers’ opinion, ammonia is the best working fluid for OTEC. The maximum $\gamma$ observed for ammonia was 0.1909kWe/m$^2$ while the maximum $\gamma$ for other fluids considered was in a range between $3\times10^{-2}$kWe/m$^2$ and 0.1342 kWe/m$^2$. A sensitivity analysis with ammonia as the working fluid was performed. A site off the coast of Puglia with a temperature difference between warm and cold seawater of 13°C was compared with tropical sites favorable for OTEC by performing a maximization of $\gamma$ by maintaining other input parameters of the model constant. The maximum $\gamma$ parameter found for tropical sites was between 0.16kW/m$^2$ and 0.20kW/m$^2$, while for the site off the coast Puglia, the maximum $\gamma$ found was lower than 0.08kW/m$^2$. The effect of turbine efficiency on the $\gamma$ parameter with ammonia as working fluid was also observed. By increasing one percentage point in turbine isentropic efficiency, the increase in maximum $\gamma$ was 2%. A multilevel closed cycle OTEC plant numerical model with the same simplifying assumptions on heat exchangers introduced for the single level model, has also been developed and a techno-economic optimization with the same optimization tool and with
ammonia as working fluid, have been carried out for a different number of evaporation levels. It was observed that the value of $\gamma$ increased with increase in number of evaporation levels. By passing from a single level to a two levels configuration, $\gamma$ parameter increased by 1%. By passing from a two level to a three level configuration the increase on $\gamma$ was only 0.16%. The possibility that a lower specific cost will be observed with a multilevel configuration (probably two levels) should be considered. However, further analyses are required to assess if specific cost of a multilevel OTEC plant is lower than a single level configuration.

Since the single level closed cycle solution is the simplest and well known solution, a detailed model of a single level closed cycle with embedded a correlation between plate heat exchangers geometrical parameters, overall heat transfer coefficient and pressure drop has been developed. The optimization tool has been modified to include plate heat exchangers’ geometry in the list of optimization parameters. The $\gamma$ parameter found by the new optimization tool was 0.184kW/m$^2$ and net power produced was 2.6MWe. Results of objective function maximization obtained with the new detailed model have been used for heat exchangers sizing and plant cost estimation. A cost of 869€/m$^2$ for titanium plate exchangers was estimated. Costs of other components have been estimated with values found in literature. Capital cost of the plant was 37.5M€ and plant specific cost was 14423€/kWe. Heat exchangers were effectively 36% of plant cost. In the plant cost estimation engineering costs were 28% of total capital costs. The engineering cost is therefore an important part of plant capital cost because the engineering procedure for the first plants is customized.

Levelized cost of energy (LCOE) has been calculated introducing simplifying assumptions on plant annual operation. It was assumed that the plant operated with constant seawater inlet temperatures for 8000h/yr producing 2.6MWe in every moment. Annual O&M costs were assumed to be 3.3% of plant cost and plant lifetime was set to thirty years. The estimated LCOE was 240€/MWh, in range with values found in literature for OTEC plants. It resulted that the LCOE of the OTEC power plant is higher than LCOE of other power generation technologies. However, large capital cost reductions that will render the OTEC closed cycle technology economically attractive in future are expected. It was observed that the mean electricity retail price for Hawaii June in 2016 was approximately 214€/MWh. If cost reductions in plant components effectively occur and engineering procedure will be standardized because of experience gained in construction of new plants, in a near future
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economical feasibility of the plant, especially for small islands with high electricity retail prices, will be demonstrated.
Introduction

In the last decades, world electricity demand increased drastically. According to the International Energy Agency (IEA), starting from 1973, electricity production increased at a rate of approximately 430TWh per year [1]. By observing Figure 0.1, it can be observed that the largest part of electric energy produced comes from fossil fuels.

![Figure 0.1: World electricity generation from 1971 to 2013 in TWh]([1]

The increase in electricity generation with fossil fuels has a negative impact on emissions, in particular increasing greenhouse gases concentration in the atmosphere. In December 2015, 186 countries at the COP 21 conference in Paris published their action plans in order to reduce global greenhouse gases emissions. According to countries involved in the COP21 conference [2] “[…] It as been widely recognized that the earth’s atmosphere is growing warmer due to greenhouse gases emissions generated by human activity”. Countries involved in the COP 21 agreement planned to increase the share of renewable energies in their energy mixes. As an example, Japan in its action plan aims to obtain 22-24% of its electricity from renewable energy sources by 2030. Even developing countries participating at the conference agreed in increasing the share of renewables in their energy mix. Cote d’Ivoire planned to include 16% of renewables in its energy mix by 2030 and Algeria planned to produce 27% of its electricity demand with renewable energy sources by 2030. In this framework, renewable energies are gaining a wide interest. Apart from well known renewable energy sources such as wind and solar energy, renewable energy from the Ocean is gaining an increasing attention. The key renewable marine energy conversion technologies
are: offshore wind, wave energy, tidal energy, ocean thermal energy conversion and salinity gradient. Renewable marine energy contribution to total electric energy produced today is very low because most of technologies are at an early development stage. Among marine energy sources however, Ocean Thermal Energy Conversion (OTEC) is well known and have been extensively studied in the past, especially in the 1980s because of an increase in the oil prices during the decade. The concept of OTEC is extremely simple: electric energy is produced by exploiting the temperature difference between warm surface seawater of tropical oceans and low deep ocean temperature. Since the temperature difference between warm seawater and cold seawater is low, the maximum efficiency obtainable by the plant is extremely low. However, unlike other energy sources such as solar and wind, the temperature difference in tropical seas between warm and cold seawater varies only slightly during the year and during day and night. Therefore, equivalent hours of operation for OTEC plant are very high [3]. Moreover, the OTEC potential according to many studies is extremely great. As an example, according to Pelc et al. [4], the estimated capacity for OTEC in the world is 10TW, large enough to satisfy world’s energy demand in 2002. Several plant configurations have been proposed for OTEC. The two most intensively studied solutions are the open cycle configuration and closed cycle configuration. The open cycle configuration uses seawater as working fluid. Warm seawater is expanded in a flash evaporator, then expanded through a turbine and finally condensed by cold seawater. In the closed cycle configuration instead, a thermodynamic cycle with a working fluid different than seawater is used and warm and cold seawater are used in heat exchangers for evaporation and condensation of the working fluid respectively.

In this thesis, since the closed cycle solution is mature and was successfully used in pilot plants, a techno-economic analysis of closed OTEC cycles have been conducted.

In Chapter 1, a literature review on main concepts of OTEC technology is presented. The different plant configurations and related advantages and disadvantages are discussed.

Since from literature review reported the previous chapters emerged that the closed cycle solution is the most mature solution, in Chapter 2 a review on components required for the closed cycle configuration and on the cold water pipe used to draw cold seawater to the plant is reported.
In Chapter 3 the methodology for techno-techno economic analysis of OTEC closed cycles is described.

In Chapter 4 a thermodynamic analysis of closed cycle OTEC plants has been performed. Ideal cycles are introduced in order to study general thermodynamic features of the plant. Rankine and multiple evaporation levels Rankine cycle for OTEC are introduced.

In Chapter 5, a numerical model of a single level closed cycle OTEC plant is described. An optimization tool used for techno-economic optimization is also presented. Working fluid selection is conducted with the optimization tool. Optimization results and sensitivity analysis for the optimized plant operating with the elected fluid are reported.

In Chapter 1, a numerical model of a multilevel closed cycle OTEC is described. With the optimization tool introduced in the previous chapter, a techno-economic analysis for a multilevel closed cycle is presented.

Since the single level closed cycle is the simplest and most studied solution, in Chapter 7 a detailed optimization of this plant is presented. According to literature review in Chapter 2, plate exchangers are considered the most favorable type of heat exchanger for closed cycle solution. Therefore, a modified optimization tool that includes plates geometry is described. The model results with this analysis are used for sizing of heat exchangers and plant cost estimation. Finally, the levelized cost of energy (LCOE) of the plant was evaluated.
1. Fundamentals of OTEC

In this chapter the main features of an OTEC power plant are highlighted. An analysis of source availability is conducted. A description of different plants configurations proposed throughout the years follows and in particular, advantages and disadvantages of the open and closed cycle configuration are underlined. Other advanced cycle configurations are listed. Finally, a list of OTEC by-products deriving from ocean water utilization after use in the plant is presented.

1.1 OTEC source availability

As stated in the introduction, OTEC uses thermal gradient between warm shallow seawater and cold deep seawater to produce electric power. The higher the temperature difference between warm and cold seawater, the higher the performance of the plant. Temperature profiles for four locations suitable for OTEC are shown in Figure 1.1. It can be seen that all the temperature profiles have a similar trend with depth. The shallow layer on the ocean surface that is able to absorb effectively the sunlight falling is called “mixed layer” [5]. In the mixed layer, which is generally comprised between the surface and 35 to 100m, temperature and salinity remain uniform because of action of winds and waves. In tropical regions included between 15°N and 15°S, the mixed layer temperature is constant during the year and during day and night in a range between 27°C and 30°C. Beneath the “mixed layer”, temperature rapidly decreases with depth until a temperature of 4.4°C is reached at a depth generally between 800-1000m. Below that limit, temperature decreases only slightly until the sea bottom is reached. Cold seawater temperature is maintained constant by currents that flow at the bottom of the sea from polar regions to the equator. Cold seawater is denser than warm seawater and remains at the bottom, while warmer seawater is displaced above. Therefore, the oceanic structure has a huge warm seawater reservoir located near the surface and huge cold reservoir in depth [5].

According to Avery et al. [5], in order to evaluate feasibility of a plant on a determined site, desirable temperature differences between warm and cold seawater are between 22°C to 25°C. The map shown in Figure 1.2 developed by Vega [6], highlights temperature
difference of ocean waters between 20m and 1000m depths. The area in which temperature difference exceeds 22°C is approximately 60 million square kilometers [7] and if OTEC plantships are sited throughout the suitable ocean area, they could produce 14TW electric power at their onboard generator [7]. This value is 19 times the total installed electric power generation capacity that was available in US in 1997 [7].

![Seawater temperature distribution with depth for various locations](image)

Figure 1.1: Seawater temperature distribution with depth for various locations [8]

Another estimation made by Pelc and Fujita [4], considers a maximum for electric power produced with OTEC without affecting the thermal structure of the ocean of 10TW. An extension of OTEC potential can be obtained when seawater temperature difference is below the aforementioned values but warm seawater is replaced with another low grade heat source such as seawater effluents leaving the condenser of conventional power plants, as reported by Soto et al. [9]. OTEC concept have been proposed also for Arctic regions where cold air in winter (at -25°C) is used as the cold reservoir and a large amount of seawater at 0°C is used as the warm thermal source [10]. This concept is referred as “Arctic OTEC”.

A US DOE study in 1981 identified ninety-eight nations and territories with access to the OTEC thermal resource within their nautical miles EEZ (Exclusive economic zone, 200 nautical miles) [11]. The following regions are considered favorable for OTEC:

- Equatorial waters between 10°N and 10°S are the first choice apart from the west Coasts of South America, whose surface temperature decreases during winter months;
- Equatorial water between 20°N and 20°S are also suitable for OTEC with the
exception of West Coasts of South America, Southern Africa, West Coast of northern Africa, Horn of Africa and off the Arabian Peninsula due to weather inconsistencies;

- Countries along the East Coast of Africa, Central and Latin American islands and Islands in the Pacific Oceans.

For countries in the Caribbean and the Pacific, the thermal resource is favorable and deep ocean water is close to the shore [11]. These regions are the most attractive regions for onshore OTEC development in the near future.

A large part of the countries with morphological and climatological interest are developing countries such as Benin, Ghana, Cuba, Haiti, Grenada, Barbados, Philippines among others. The possibility of installing an OTEC plant in these countries, to increase electrification without fuel consumption and for fresh water production, an important OTEC by-product, should be assessed.

1.2 History of OTEC

Ocean thermal energy conversion concept was proposed for the first time in 1881 by French physicist D’Arsonval. The concept regarded the introduction of a Rankine cycle with ammonia as the working fluid between the warm surface seawater and the cold deep seawater. Georges Claude, a D’Arsonval student, in 1928 demonstrated the technical feasibility of the concept by developing a demonstration plant in Belgium which used water
at 30°C from a steel plant as the warm thermal source and water from river Meuse at 10°C as the thermal sink. The test succeeded in reaching a turbine speed of 5000rpm and 50kWe of power produced. After this first success, Claude had the possibility in 1930 to develop a pilot plant 1600m offshore the coast of Mantanas Bay, Cuba. The plant operated continuously for 11 days producing 50kWe, but the cold water pipe was later destroyed by a storm. Three years later Claude obtained financing for deployment of an open cycle plant on a barge off the coast of Brazil for ice production. Gross power produced by the turbine would have been 2200kW with a net power of 1200kW. The project did not succeed in attaching the cold water pipe to the floating structure and was abandoned. Despite this failure, Claude proposed a 40MW plant offshore the coast of Abdijia, Cote d’Ivoire. The project did not obtain funding by the French government and was abandoned too.

The research and commercial development of OTEC was abandoned until the late seventies, when a rise in oil prices pushed US to start an OTEC development program [5, 11]. In 1979, Lockheed corporation, Dillingham corporation and Hawaii state government succeeded in developing an at sea test of a demonstrative OTEC closed cycle plant called “Mini OTEC” [5]. The plant operated for three months with ammonia as the working fluid and was placed on a ship. After the Mini-OTEC experience, the OTEC-1 program started in 1980 by the US Department of Energy (DOE) at Kalua-Kona, Hawaii. The program consisted in the installation of a test facility in order to test the behavior of critical OTEC components such as the cold water pipe, heat exchangers and the mooring system. A turbine was not included in the design and pumping power was provided by diesel generators. The program succeeded in developing a 670m cold water pipe, a 1370m mooring system, different types of shell and tube heat exchangers operated successfully and fouling control experiments were also conducted. However, the program stopped because of high costs of fuel for pumping seawater and high operational costs [5]. Contemporarily in 1980, US DOE started a Program Opportunity Notice (PON) inviting industry to participate in a phased cost sharing development program of a 40MWe power plant. Eight plant conceptual designs have been presented by several industries and consortiums. Only two of those designed was funded to develop a detailed design. John Hopkins University was asked to design a baseline 40MWe plant as a reference. However, after initial enthusiasm, a drop in oil prices made the project economically unattractive and funding terminated [5]. Saga University since 1980, became a great contributor to OTEC research and development.
After laboratory test, in 1980 constructed an onshore demonstration plant on the coast of Nauru island [12]. The plant was a closed cycle with R22 as working fluid and operated with shell and tube exchangers producing 100kWe gross power. Power generation continued from October to December 1981 and closed loop continued operation until 1982. Large scale development was abandoned during the eighties and nineties, even if small scale test and plant proposals continued in various parts of the world. An open cycle OTEC plant operated successfully at Hawaii from 1993 to 1998 with combined energy conversion and desalinated water production. The plant produced 103kW net power. In 2002 India started testing for development of a 1MW floating OTEC plant but the project failed in cold water pipe attachment [13]. Recently, DOE of energy awarded a $1.2 million contract for special cold water pipe development for commercially sized OTEC plants. Two $1 million grants have been awarded by Lockheed Martin in 2009 to develop a GIS-based tool to estimate the maximum energy that can be extracted from a site and to study life cycle costs to demonstrate economic feasibility of OTEC systems [11]. On April 2013 Lockheed Martin announced a collaboration with Reignwood industries to develop a 10MW offshore OTEC plant that will feed a resort on the coast of southern China [14]. In 2013 saga University in participation with other industries such as Xenesys Inc. developed a 100kW net power onshore closed cycle OTEC plant on the coast of Kumejima in the Okinawa prefecture [15]. The plant is still in operation. In 2014, the GOSEA (Global Ocean reSource and Energy Association) was founded in order to export the Kumejima model throughout the world [15]. In the same year, Akuo energy and DNCS won a EU funding to finance the NEMO (New Energy for Martinique and Overseas) project, a floating 10.7MWe OTEC facility 5.3km off the coast of Bellefontaine, Martinique [16]. Finally, in 2015 Makai celebrated the completion of a 100kW OTEC plant in Hawaii which is connected to the electric grid [17] and Bluerise continued developing an Ocean Ecopark in collaboration with Curaçao International Airport [18].

1.3 Siting characteristics of OTEC plants

OTEC power plants can be classified according to their siting characteristics. Two categories are generally considered:

- Shore based: plant is sited on the coast and seawater is brought to the plant by long
Chapter 1

- Offshore:
  - Moored: the power plant is posed on a barge moored to the sea bottom;
  - Grazing: the plant is posed on a ship able to change its position during time.

The main advantage of the offshore solution is that the plant can be positioned in favorable places where the temperature between cold and warm seawater is the highest possible even if these sites are distant from the coast. The project of the plant is, in some way, independent of site in which is positioned, and this fact can ease the series production and commercialization of offshore OTEC plants [5]. The warm seawater tubing is optional; thus warm seawater pumping power consumption is reduced. The main disadvantages are linked to plant positioning and plant resistance to mechanical loads. In case of a moored plant, an adequate mooring system must be developed, while in case of a grazing plant, auxiliary diesel engines or auxiliary electric propulsion system must be provided in order to permit the plant to move and contrast the effect of waves and currents. The entire structure must tolerate the effect of waves, currents and the effects of the one-in-a-century storm, which is the most severe storm that can be forecasted for a determined site. Another important disadvantage of offshore solution is related to the energy transfer mode to shore of the energy produced. In case of a moored plant, an underwater electric cable must be designed to connect the electric grid to the plant. However, in some sites or in case of a grazing ship, the underwater cable cannot be positioned. Electric energy produced by the power plant must be stored in some way in the power plant and transported to shore by ships. Electric energy is difficult to store efficiently, thus electric power produced by the plant is used in synthesis of fuels like hydrogen or methanol or chemicals like ammonia. Products are then stored in adequate tanks in the plant and ships periodically transport those products to final users on shores.

The onshore solution considers plant positioning on shore while seawater is drawn to the power plant by the use of seawater pipes which are generally longer than the ones used for offshore solution. This solution is simpler than the offshore solution because barge design is not necessary. Electric power produced can be directly transferred into the electric grid and losses due to electric to chemical conversion for energy storage are prevented. The main disadvantage is due to the length of seawater pipes which can be, according to [7] from two to five times longer than in the offshore configuration, with added mechanical resistance.
issues. The pipes length, in particular for cold water pipe, depends on bathymetric profile of the site selected for plant installation. Thus, design of the plant is strictly site dependent. Moreover, the problem of biofouling on heat exchangers is more emphasized than in offshore plants because concentration of organisms is higher near shore [5]. Some shelf mounted solutions have been presented during the years. A platform on the bottom of the sea is constructed and the power plant emerges from sea surface. This solution, which take the cue from oil and gas industry, has the advantage of being positioned offshore, but cannot be positioned in every favorable site because of constraints in bottom depth [5]. However, the shelf mounted plant is considered as a long-term solution.

1.4 OTEC cycles configurations

Several methods that use ocean thermal gradient for energy conversion have been proposed. Each one of these entails a particular plant configuration and specific components. The principal classification is between open and closed cycle, nevertheless, although at an early development stage, the Panchal-Bell cycle, mist lift cycle or the foam lift cycle are worth of notice too. A particular closed cycle is the cycle derived from the Kalina cycle, in which a mixture of water and ammonia is used [8]. Hybrid cycles for combined energy conversion and by-products production by using exhaust ocean water, such as desalinated water, mariculture, agriculture, cosmetics production etc., have also been proposed. Since OTEC is a capital intensive technology, these by-products are of particular interest because could help to make the plant economically feasible [19].

1.4.1 The open cycle

The open cycle (OC-OTEC) uses seawater as the working fluid. This configuration, whose schematic diagram is sketched in Figure 1.3, is defined open cycle because working fluid, which is seawater, after condensation process, is not reintroduced in the cycle, but discharged to the environment. Warm seawater flows into a deaerator where pressure is suddenly reduced to a value slightly above the saturation pressure in order to remove incondensable gases dissolved in seawater. Water flows in a flash evaporator where its pressure is reduced to the saturation pressure that corresponds to turbine inlet thermodynamic conditions. This pressure drop causes seawater to partially evaporate. Steam is expanded in a turbine, while the remaining liquid seawater is discharged to the
environment. Flow at turbine outlet is then condensed with cold seawater. If a surface condenser is used, the condensed seawater is desalinated water, a by-product of the plant that can be precious in zones with reduced access to potable water sources (i.e. isolated islands). Otherwise, a direct contact condenser can be used. The main advantage of the open cycle configuration is that if direct contact exchangers are used (the flash evaporator is a direct contact exchanger), the low available temperature difference given by warm and cold seawater can be almost completely used: steam in the vacuum chamber has the same temperature as warm seawater at the evaporator discharge, while at the condenser, temperature reached by condensed steam is equal to cold seawater discharge temperature. If a surface condenser is used, temperature difference between seawater in condensation and cold seawater exists, but in this case, the combined power and desalinated production is worth of interest.

![Figure 1.3: OTEC open cycle schematic diagram [5].](image)

The main disadvantage of the open cycle solution are the low pressure drops at which turbine operates, in the order of 2.8kPa, and large specific volumes of low pressure steam [5] thus implying a very large turbine; in particular no full scale turbine for plants 1MWe and greater have been designed and operated [5]. Vacuum pumps required for gas removal are consistent energy consumption components which negatively affect the net power output of the plant. However, their presence is required especially in the condenser. Incondensable gases in the condenser in fact, significantly degrade its performance increasing condensation pressure.
1.4.2 The closed cycle

The closed cycle configuration (CC-OTEC) has been the most studied solution by researchers throughout the years. The plant scheme is depicted in Figure 1.4. Thermodynamic cycle is generally a conventional saturated vapor Rankine cycle. Warm seawater flows into a surface evaporator, working fluid is evaporated and steam is expanded in a turbine and electric energy is produced. Flow at turbine outlet is then condensed and pumped back to the evaporator at convenient pressure by a working fluid pump.

The main disadvantage respect to an open cycle configuration, is that large surfaces for heat exchangers are required. High heat transfer coefficients are required and a temperature drop between working fluid and seawater is necessary in order to limit heat transfer areas. Materials employed in heat exchangers must resist to a chemically aggressive environment (seawater): titanium and aluminum are considered among others [20]. Thus, heat exchangers in the closed cycle configuration are an important part of the overall plant cost. However, despite of all those criticalities, the closed cycle solution so far has been the most investigated solution because most of all required design features are commonly employed in engineering practice and can be easily adapted [5].

![Figure 1.4: OTEC closed cycle plant scheme [8]](image)

The distinctive feature of the closed cycle is that seawater is not the working fluid: a fluid with favorable properties can be selected from a wide range of fluids. Working fluid choice
is critical because impacts directly on turbine and heat exchangers. According to Avery et al. [5], desirable working fluid should have vapor pressure at 27°C in range between 700 to 1400kPa, low volume flow of working medium per kilowatt power produced, high chemical stability and compatibility with materials and structures of the power cycle, good heat transfer properties, low cost and environmental acceptability. The opportunity to choose the working fluid permits to reduce turbine and ducts sections respect to the open cycle. Conventional design procedures for these turbines can be adopted. Ammonia is considered by most researchers as the most suitable fluid for OTEC despite of its toxicity in case of leakage. However other fluids such as R22, R134a or propylene and other refrigerant fluids have been proposed. A study conducted by Marchand [21] compared different fluids have been in terms of power system costs (see Figure 1.5). As expected, ammonia resulted as the preferable fluid in terms of plant costs. Another study conducted by Yang et al. [22], comparing different working fluids for OTEC, also selected ammonia as the best working fluid for CC-OTEC.

Figure 1.5: Closed cycle OTEC power system cost according to different working fluids [21]
1.4.3 Other cycle configurations

Even if the open and closed cycle are considered as the standard plant configuration, some different layouts have been proposed.

The Panchal-Bell cycle shown in Figure 1.6, is similar to a closed cycle, but the evaporation of working fluid is driven by the condensation of warm seawater vapor created by flash evaporation. Condensed water is fresh water and it is a plant by-product. The higher heat transfer coefficient of condensing steam leads to a heat exchanger area reduction. Moreover, possible contamination and corrosion in the evaporator is prevented because contact between heat exchanger surfaces and liquid seawater is avoided.

![Figure 1.6: Panchal bell cycle with integrated desalination in the evaporation chamber [23]](image)

The mist lift cycle reported in Figure 1.7 has a different functioning. Warm seawater is drawn in a long vertical tube by means of a vacuum pump. Warm seawater partially evaporates along the tube and a mist composed by steam and liquid droplets is formed. At the top of the tube, which is theoretically 280m high, the mist is condensed by contact with a surface chilled by cold seawater. Condensed stream passes through a hydraulic turbine and finally released to the environment. Further design studies are required to define a complete system.

In the foam-lift cycle, a foaming agent raises the liquid water first and then allows it to fall and releases its stored potential energy to a hydraulic turbine [5]. A schematic of the foam lift cycle appears in Figure 1.8. Warm seawater is mixed with foaming agent and since foam bubbles reduce density of the water column, an upward flow causes the mixture to spout from the top. Foam breakers are provided and separate liquid from vapor in the foam. Vapor
is condensed by direct contact with deep ocean water, while the liquid is channeled into water columns, where it is used to drive the turbine and then discharged back into the sea. R&D is required to assess the technical feasibility of this solution and effects of the foaming agent under the economic and environmental point of view must be considered.

A closed cycle that uses zeotropic mixtures instead of pure fluids can be also implemented. A zeotropic mixture has generally a variable temperature during phase change and this fact can be used to reduce exergy losses of the heat transfer process. Some plant proposals are a
simple Rankine cycles operating with mixtures, while other proposals are based on the Kalina cycle, generally adopting a mixture of ammonia and water as working fluid. A schematic diagram of Kalina cycle for OTEC appears in Figure 1.9. High concentrations of ammonia are required to have a low mean boiling temperature. However, if condenser seawater temperature has the same mean condensing temperature of a closed cycle, the ammonia water mixture at turbine outlet will have a high pressure which reduces power extracted by the turbine [24]. A separator is introduced after the evaporator and a mixture with low ammonia concentration is extracted and bypasses the turbine. The rich ammonia mixture is expanded in the turbine and mixed with the lean solution at turbine outlet. The resulting mixture composition has lower ammonia concentration and reduces condensation pressure. Efficiency of a classical Rankine cycle for OTEC is 3%, while efficiency expected with Kalina cycle is 5% [8].

![Kalina cycle for OTEC](image)

Figure 1.9: The Kalina cycle for OTEC [23]

However, the Kalina cycle has an increase of load on heat exchangers particularly on condenser [8]. For the purpose of relieving the condenser load, the Uehara cycle have been implemented. This cycle lessens the condenser duty by means of extraction of vapor from the turbine, as seen in Figure 1.10. Therefore, an OTEC plant compact in size but with higher efficiency is possible, however, added complexity must be taken in account.
1.4.4 OTEC with solar hybridization

By increasing the temperature difference between the warm and cold thermal reservoirs, thermodynamic efficiency increases. If cold seawater temperature is fixed, constrained by the depth at which the cold water pipe intake is positioned, the only way to increase the temperature difference is to raise the maximum cycle temperature. Solar collectors are considered a good option to achieve this task and this integration is reasonable since regions with suitable conditions for OTEC have generally a good seasonal solar radiation. This solar integrated solution is called SOTEC [25]. In an open cycle configuration, solar collectors are positioned before warm seawater enters heat exchangers. Therefore, seawater enters the open cycle at higher temperature. Yamada et al. [25] proposed the closed cycle SOTEC solution shown in Figure 1.11. Solar collectors are positioned on the seawater side before seawater enters the evaporator.

Analysis conducted by Yamada et al. [25] showed that a 100kW SOTEC plant with a temperature increase of 20°C due to solar collectors, has a net efficiency 1.5 times higher than the annual net thermal efficiency of conventional OTEC plants. From the same analysis the pumping power for a 100kW SOTEC plant was 30% lower than the 100kW OTEC plant because cold seawater flowrate is reduced because of the increased Rankine cycle efficiency.
Bombarda et al. [26] proposed the plant scheme in Figure 1.12, which entails the idea that solar energy can be more profitably exploited at a higher temperature: an additional topping ORC is added to a three level OTEC plant. Heat rejected to the solar ORC is transferred to warm seawater that enters the heat exchangers of the bottoming OTEC cycle. The topping cycle is a regenerative ORC with siloxane (MM) as the working fluid. The OTEC cycle was designed according to nighttime conditions and operates in off-design conditions when topping cycle is operating. Efficiency of the total power plant is improved because seawater entering the bottoming section is warmer and because of the additional power produced by the topping cycle. However, added complexity to the design is introduced.
1.5 Energy transfer

Mechanical power produced by the turbine is converted in electrical power with an electric generator. A consistent part of this power is used to drive mainly the seawater pumps and secondarily working fluid pump (in case of a closed cycle plant) and auxiliaries required for plant operation. The remaining electric power, which is 50% to 80% electric power produced by the turbine generator\[14\], is the net output of the plant and can be transferred to an electric grid. If the plant is sited onshore, connection is easy, but in case of offshore platforms or moored plants, connection to the electric grid takes place with submarine power cables with a consequent increase of investment cost\[5\]. In case submersed cables cannot be positioned or in case of grazing plants, electric power produced is used and stored on board. According to Mura \[23\] energy produced by an OTEC plant can be used in these ways:

- Water electrolysis for hydrogen production;
- Hydrogen produced can be used for ammonia synthesis;
- Methanol production from coal by gasification using separated hydrogen and oxygen in the water electrolysis process;
- Lithium air batteries charging.

These products can be stored on the platform and carried on shore with ships at regular intervals \[5\]. However, these solutions for energy transfer are practically unfeasible at the moments because of intrinsic technological difficulties of the conversion methods proposed. The only solution practically feasible for offshore plants in the near future remains the connection to the electric grid via underwater cable.

1.6 OTEC by-products

After using seawater in the OTEC cycle, it is possible to use seawater to obtain OTEC by-products. Integration of OTEC with other plants that use deep ocean water (DOW) is interesting because can increase economic feasibility of the OTEC plant \[27\]. Since DOW is already pumped for OTEC purpose, pumping costs related to further DOW use are avoided. Research on “Multiple OTEC system” that exploit the potential of DOW combined with electric power production is in progress. Figure 1.13 resumes the most important surface water and DOW uses. One of the expected uses of seawater after use in the plant is the desalinated water production. In an open cycle configuration, warm seawater is the
working fluid and before expanding through the turbine, undergoes a flash evaporation as stated in section 1.4.1. Evaporated seawater is salt free and if condensed in a surface exchanger after expansion fresh water is obtained.

![Diagram showing multiple OTEC system](image)

**Figure 1.13: Multiple OTEC system [8]**

It is possible to obtain desalinated water even from a closed cycle configuration by positioning a flash evaporator after warm seawater heat exchanger as shown in Figure 1.14. Steam is condensed with cold seawater leaving the condenser and desalinated water can be collected. Saga university in Japan, one of the pioneering research institutes on OTEC and deep seawater uses, developed a spray flash desalination system that according to [8] has an obtainable distilled capacity of approximately 10000m$^3$/day with 1MW OTEC and 1000000m$^3$/day with a 100MW power plant. It is also possible to produce mineral water by adding an ion exchanger and a mineralizer [8].

Ocean water is rich of chloride-lithium that can be extracted for lithium industrial production. Purity of DOW reduces the cleaning burden for surfaces in contact with seawater and can make lithium extraction from seawater economically feasible [8].

Since the temperature of DOW after use in the condenser of the OTEC plant is still low, it can be used for air conditioning and chilling. Since regions suitable for OTEC are tropical regions, a massive use of DOW can reduce energy consumption for refrigeration systems.
Aquaculture is also an interesting approach to the DOW utilization. Since these waters are pure, cold and rich in nutrients, DOW can be used for fishery fertilization and massive algae production [28]. Food processing, and the cosmetic industry can also take advantage of DOW unique properties [8, 15, 28]. Finally, seawater can be used for hydrogen production by means of electrolysis using the electric power produced by the OTEC power plant.

Figure 1.14: Closed cycle OTEC with seawater desalination (left) [9], Saga University desalination plant (right) [8].

1.7 Environmental impact

Despite OTEC uses no fuel for energy production, environmental impact, forecasting a large scale diffusion of those power plants must be assessed. In case of closed cycle plants, which according to many researchers [5, 21, 22] will have the largest success in the near future, the risk of leakage of working fluid is the main environmental risk. Ammonia, elected by most of researchers as the best working fluid for OTEC conditions is toxic in case of leakage: it is highly soluble in water and concentrations of ammonia dangerous for marine life can be found at large distances from the plant [5]. Ammonia can also create a gaseous mixture which is lethal after few minutes of exposure. An ammonia leakage from a barge in Kochi in May 2016, forced authorities evacuated an area of two kilometers radius from the leakage spot [29]. Other working fluids such as hydrocarbons are dangerous in case of fluid leakage from the plant. Among refrigerants, fluids with low GWP are preferable. The effects of chemicals for biofouling control on heat exchangers surfaces must also be accounted [5]. Apart from leakage, the other concern about OTEC plants in particular for large size plants, derives from mixing of exhaust cold seawater with shallow seawater. Since large amounts
of cold seawater are used, cold seawater discharged can alter the temperature of the Ocean and alter the ecosystem. This problem can be avoided by discharging exhaust cold seawater at an appropriate depth [26]. Cold seawater is rich in nutrients and oxygen and the discharge of seawater brings to the shallowest layers of the ocean a large quantity of nutrients that can have positive effects on fishery but also negative effects derived from algae proliferation and eutrophication [30]. However, since no large plants have been constructed yet, these latter effects have not been properly evaluated and further research is needed.
2. CC-OTEC components

According to section 1.4.2, the closed cycle configuration can be readily developed with existing technologies. The closed cycle components can be adapted from off the shelf components with minimal effort. Because of these advantages, the closed cycle is the most studied configuration and the first commercial plants that will be constructed in the near future will use a closed cycle.

This chapter contains a literature review of components of closed cycle OTEC plant: cold water pipe (CWP), heat exchangers, turbine and seawater pumps. The CWP and seawater pumps for the open and closed cycle configuration have similar features. The CWP design and construction and its attachment to the plant is one of the critical issues encountered during OTEC design. According to Little [31] in order to produce 1MWe of net electric power, 4m$^3$/s of cold seawater are required. Since seawater flowrates involved are very large, large diameters in order to limit seawater velocity are desired. However, limits on maximum pipe diameter due to mechanical resistance exists. The pipe in fact, should be able to resist to currents and its connection to the plant must be flexible and resistant to the one-in-a-century storm. Connection in case of offshore plants is more critical. Consequently, CWP is one of the biggest uncertainties on plant feasibility and uncertainty on its cost is 100% [5].

Discussion on materials, design features of heat exchangers and turbine is presented. For heat exchangers in particular, a lot of research effort have been conducted during years and several innovative designs are presented. Titanium and aluminum are the elected materials for durable seawater operation [7, 20]. Shell and tube, folded tube, plate and fin, gasketed plate and brazed plate exchangers are considered as suitable for OTEC: improved design for shell and tube and plate exchangers and their testing results at the ANL (Argonne National Laboratory) and other testing facilities are reported [32]. Several turbine designs from radial inflow to axial type solutions have been proposed according to plant size. Finally, a brief description of seawater pump features typical for OTEC operation is reported.

2.1 Cold water pipe

The colder seawater used in the condenser is, the higher the thermodynamic efficiency of
the system. Therefore, CWP intake must be positioned as deep as possible in order to draw seawater at the lowest temperature possible. CWP affects directly the performance of the cycle because its dimensions have a direct effect on pressure losses and on investment costs required. The challenge of the CWP is delivering large amounts of DOW with a low pressure drop along the tube, withstanding high mechanical loads. According to Mura [23], OTEC plants of sizes between 100MW and 400MW requires seawater volume flowrates between 300m$^3$/s and 1200m$^3$/s. Power plant of those sizes are generally composed by 50-60MW modules, hence, for large sizes, more than one CWP are required.

As stated before, CWP must withstand stresses of various nature [7]:

- Stress due too suction;
- Fluctuating lateral and vertical loads caused by waves and currents acting on the ship that supports the pipe for offshore plants;
- Longitudinal stresses due to pipe weight;
- Stresses due to near shore wave and currents regimes in case of an onshore solution;
- Resistance to a hostile environment.

Several materials are considered suitable for OTEC: low density concrete, steel, aluminum, high density polyethylene (HDPE), fiber reinforced plastic (FRP), elastomer. Among these, the most important are low density concrete, HDPE and FRP [5]. The pipe structure of a concrete pipe and fabrication process of FRP pipe are shown in Figure 2.1

Concrete pipe with longitudinal reinforcing with steel bars and circumferential reinforcing with wire mesh was selected by John Hopkins University for the baseline 40MW power plant for the US DOE PON [5]. The study of John Hopkins University showed a minimum cost for concrete cold water pipe because of its massive use in a large variety of hydraulic applications such as for sewers, conduits, water distribution systems. Facilities are available for construction in a wide range of diameters, thicknesses and lengths [5].

Many researchers such as Bombarda et al [26] and companies such as TRW and SAI, consider FRP as the best material for OTEC CWP. A study of TRW in 1980 reported in [5], asserts that facilities able to manufacture a FRP 9.1m in diameter and 1000m in length pipe can be constructed. The same study reported that costs for manufacturing and installing CWP pipe on an OTEC platform were approximately 20% higher than concrete pipe, but could offer advantages in the deployment procedure. Lockheed Martin [33] is currently working and testing an advanced 4m diameter fiber composite CWP in order to develop a 10m
diameter pipe for its emerging OTEC line business.

Figure 2.1: Concrete cold water pipe structure (left) and FRP CWP fabrication (right) [5]

Although FRP and concrete are proposed for larger power plants, the Mini OTEC, the OTEC-1, the Nauru OTEC and many other plant proposals such as the Lockheed Martin 2.5MW Mini Spar [27] selected HDPE as CWP material and consider 2.5m as the maximum diameter obtainable for a HDPE CWP. Pipelife International in 2013, started commercial production of 2.5m diameter HDPE pipes [34]. HDPE is more flexible than steel and has a good adaptability to sea bottom irregularities [12]. For these two important features it was selected for the Nauru shore based plant [12]. Installing procedure of CWP adopted for the Nauru plant is reported, in Figure 2.2. Five sections of 10m length were welded together in order to form a unique 50m pipe. The first pipe was pulled in seawater by a pulling barge while at the other hand of the pipe another 50m section was welded. The pipeline has been gradually pulled in seawater and buoyancy were provided by buoys. Once the entire pipeline was pulled in water, a supporting barge has been posed on the middle of tube length to favor buoyancy. Buoys posed at opposite pipe extremities have been contemporarily gradually unfastened and the supporting barge lowered the center of the pipeline to let the pipeline to sink. Positioning operation was repeated several times until pipe was in the desired position. For the Nauru power plant, eighteen HDPE pipe sections, 50m long each, were welded and submerged. Required time for positioning and sinking operation was sixty hours [12].
Connection of the CWP to the plant is also critical especially in case of moored or grazing plants. The joint must be able to bear with CWP weight and to stresses due to currents, waves and storms. The joint could make CWP easy to detach from the plant for eventual maintenance. Three joints were considered: a ball and socket joint, a gimbal and a universal joint. Sketches of the three joints are reported in Figure 2.3. According to [23] in order to withstand stresses due to currents and waves, the CWP vertical axis can move up to 30° from the vertical direction.

Figure 2.3: CWP joints [23]
2.2 Heat exchangers

Since the efficiency of an OTEC plant is low due to low temperature difference between warm and cold seawater, large amounts of thermal power need to be exchanged in order to have acceptable power produced by the turbine. Therefore, in a closed cycle, large heat exchangers are required. Uehara et al. [35] assert heat exchangers’ cost is the 25-50% of the total plant cost. According to this, reducing heat transfer area permits to reduce plant cost. Moreover, heat exchangers not only affects plant cost but also the net power produced by the plant: large pressure drops increase the pumping power requirement and net power reduces. Low temperatures and pressures of working fluid are generally acceptable (from 10bar to 6bar for ammonia evaporator and condenser, respectively) and off the shelf heat exchangers can be used [5]. However, heat exchangers designs dedicated to OTEC have been tested in particular during experiments at the ANL in 1980s and at Saga University in Japan. These heat exchangers had surface enhancements in order to achieve higher heat transfer coefficients and reduced areas. Shell and tube heat exchangers and brazed plate heat exchangers have been tested, but also folded tube heat exchangers and plate and fin heat exchangers designs were proposed for OTEC. More recently, Makai Ocean engineering and Lockheed Martin developed a test program for heat exchangers for OTEC [36].

2.2.1 Heat exchangers materials

Since capital intensive, OTEC power systems are designed for long life, up to 30 years with minimal maintenance [20]. The aggressive environment in which heat exchangers operate requires careful selection on materials employed in order to avoid rapid deterioration and replacement costs. Cost for maintenance and cleaning from biofouling should also be taken in account. Several materials are industrially used for heat exchangers such as steel, copper-nickel alloys, aluminum and titanium. Stainless steel is commonly adopted in condensers which adopts seawater as refrigerant. Alloy 2Cr-25Ni-6Mo or austenitic stainless steels behaved well in seawater operations, but are subjected to biofouling and localized corrosion attack [20]. According to this it is clear that stainless steels require to be cleaned during operation. Mechanical cleaning with brushes or chlorination in case of 2Cr-25Ni-6Mo are tolerated. Copper-nickel alloys are attractive because of their biocidal properties, but are not able to tolerate ammonia and tests showed erosion due to turbulent flows in plate exchangers [20].
Aluminum alloys are considered suitable materials for OTEC especially for their low cost. Aluminum for OTEC uses is expected to endure for 10-15 years even if this forecast is unreliable due to lack on information [20]. Aluminum has not biocidal properties and the effects of mechanical cleaning on surfaces must be assessed. According to Eldred et al. [36] pitting is the main obstacle to a durable aluminum exchanger for OTEC and the lifetime of an aluminum heat exchangers will depend on pit growth rates. Test conducted by the same authors and by Larsen-Basse [37] showed that pitting is more severe in cold seawater than in warm seawater. According to [36], corrosion tests on aluminum alloys and research on fabrication techniques are currently underway.

Titanium emerged as the leading candidate for OTEC heat exchangers. Despite of the initial cost, maintenance and replacement is minimum and can be considered economically competitive as the other candidate materials. The mini-OTEC, the OTEC-1, the Nauru plant and Kumejima plant employed titanium for heat exchangers at least for parts in direct contact with seawater such as tubes in shell and tube heat exchangers. Titanium has a thin oxide film formed in contact with air or moisture which prevents corrosion in seawater [20]. The film is able to repair itself. Pitting is totally absent [20]. Titanium has no biocidal properties and since it is resistant to erosion, fouling can be reduced with high flow velocities. Chemical cleaning such as chlorination and mechanical cleaning are also well tolerated [20]. Titanium however is likely to be affected by fatigue caused by vibration, thus particular care in tube design for shell and tube exchangers is required [20].

2.2.2 Shell and tube heat exchangers

Shell and tube are the most diffused layout in process industry. Their design is well developed and their behavior with two phase liquids is well understood. According to this, several closed cycle plant proposals adopted the shell and tube type heat exchanger for both condenser and evaporator and this is also the case of Nauru plant. Commercially available shell and tube exchangers can be directly used on OTEC plants [5]. The working fluid generally stays on the shell side for both condensation and evaporation because the area in contact with seawater is minor on the tube side; hence, the area subject to fouling and corrosion is minor [5]. Moreover, cleaning of inside of tubes is easier than shell cleaning [5]. Carbon steel shell that tolerates well ammonia can be used while titanium tubes are employed. This solution leads to a good trade off between materials cost and corrosion resistance and was adopted for the OTEC-1 shell and tube heat exchangers. Performance of
conventional shell and tube exchangers is limited by the presence of laminar boundary layer on both sides of the tubes. Many surface enhancements have been studied to improve heat transfer in particular on the working fluid side. Smooth surfaces on evaporation side have a low number of nucleation sites and this affect negatively nucleate boiling. A surface roughening can improve heat transfer performance of the tube. During condensation, a thick condensing film forms on the tube surfaces reducing heat transfer: film breakers can stop the condensing film thickening and improve performance [5]. Another way to enhance heat transfer in vertical tubes is fluting of tubes: a liquid film flowing downward on a vertical fluted surface has different thickness in the ridges and in the flutes. Inside flutes the film is thicker because of surface tension, while on ridges the film is thinner and heat transfer on ridges increases [5]. Distribution of the film is sketched in Figure 2.4.

Figure 2.4: Liquid distribution on a vertical fluted tube [5]

In 1978 a heat exchangers test facility was set up at the ANL with the scope of testing OTEC dedicated heat exchangers designs presented by participants to the OTEC program of DOE. Heat exchangers were tested with a heat duty of 1MWt and designers were forced to furnish innovative designs [5]. Union Carbide and Carnegie Mellon University (CMU) presented improved shell and tube evaporators and condensers. Two different evaporators were presented by Union Carbide with Linde enhanced surface treatment: Linde flooded bundle evaporator and Linde sprayed bundle evaporator. In both the evaporators, titanium tubes
were treated with a proprietary surface coating called Hiflux. Both the evaporator had a horizontal configuration but while the first was flooded with liquid ammonia, in the second ammonia was sprayed on the tube bundle by specific tubes. Both designs are shown in Figure 2.5.

Figure 2.5: Linde flooded bundle evaporator (above). Linde sprayed bundle evaporator (below) [5].
In nominal operation with a water velocity of 2.1m/s and with seawater inlet and outlet temperatures of 26.67°C and 25.56°C respectively, and with ammonia temperature of 22°C, the heat transfer coefficient of the flooded bundle evaporator was 4336W/m²K, with a pressure drop of 18.8kPa. In the same nominal conditions, the sprayed bundle evaporator showed an overall heat transfer coefficient of 4313W/m²K with a pressure drop of 26.7kPa, due in particular to ducting losses [5].

Union Carbide Linde enhanced tube condenser consisted in 3.8cm OD aluminum tubes with flutes on the water side and wire wrapped on ammonia side to break the condensed film [5]. A detail of the surface enhancement is given in Figure 2.6. During tests, inlet temperature of ammonia was 8.9°C while water entered at 4.4°C and left the tubes at 5.6°C. At nominal velocity of 1.43m/s overall heat transfer coefficient was 4642W/m²K, while the pressure drop was 12.4kPa. Tests showed that ammonia heat transfer coefficient remained independent of heat duty in a range between 13620W/m²K and 22700W/m²K [5].

![Figure 2.6: Details of Linde enhanced tube condenser [5]](image)

CMU developed a vertical fluted tube evaporator and condenser. Tubes were 2.32cm ID and 3.09cm OD made of aluminum. With the same temperature of the Union Carbide exchangers, with a flow velocity of 1.98m/s the overall heat transfer coefficient was 4680W/m²K considering a cylinder with mean inner and outer diameters [5]. The pressure drop was 22kPa. When operated as a condenser with same temperature as Union Carbide condenser and water velocity of 1.98m/s the overall heat transfer coefficient was 5918W/m²K with a water side pressure drop of 23kPa [5].
2.2.3 Folded tube heat exchangers

John Hopkins University APL (Applied Physics Laboratory) developed a low cost heat exchanger for OTEC with a unique design. In the folded tube configuration, ammonia flows inside a large number of long tubes displaced horizontally and folded many times for compactness. The folded tubes are contained in a vessel and immersed in seawater that flows under gravity as depicted in Figure 2.7. Tests on this heat exchanger showed lower heat transfer coefficients than enhanced shell and tube exchangers but comparable with traditional shell and tube [5, 38]. The main advantage of this solution is that a pressure vessel to contain working fluid at high pressure is not required and this can reduce the heat exchanger’s costs. The main disadvantage of this solution is related to its unique design. Lack of manufacturing data according to [5] is perceived as a high risk and therefore, the shell and tube configuration whom manufacturing technology is mature, is still preferred.

Figure 2.7: Folded tube heat exchanger [5]
2.2.4 Plate and fin heat exchangers
Plate and fin heat exchangers are widely used in many applications from process industry to the automotive industry. According to Kakaç et al. [39], the heat transfer area per unit volume is around 2000m²/m³. This feature, in the context of OTEC where large heat transfer areas are required, is favorable in terms of space required and weight of the heat exchangers, in particular in case of a platform mounted plant. In the most common layout, the fluid streams are separated by plates and a thin corrugated plate is sandwiched and forms the fins for heat transfer enhancement [39]. Plate and fin exchangers for OTEC application are slightly different. As seen in Figure 2.8, the water passages are obtained by extruded billets while the ammonia flow channels are obtained by thin rolled sheets that are brazed between two extruded parts [5]. Seawater flows downward from a pool above the heat exchanger in order to reduce pumping power consumption. Ammonia can flow in the same direction of seawater or crossflow according to the direction in which thin sheets are brazed. Aluminum is generally used because of its favorable manufacturing features but more research on pitting corrosion resistance is required [36].

![Figure 2.8: Plate and fin heat exchanger for OTEC application [5, 36]](image)

2.2.5 Plate and frame and brazed plate heat exchangers
Plate exchangers have started to be used from the 1930s and they widely diffused in many industrial fields. Their success is related to their high compactness respect with traditional shell and tube exchangers and their enhanced heat transfer properties. Due to these
advantages, plate exchangers have been proposed for OTEC by Alfa Laval in the 1970s [5]. The Mini-OTEC pilot plant in fact, employed successfully this type of exchangers. Plate and frame heat exchangers are composed of a series of thin plates pressed together by tightening bolts. Plates are separated each other by gaskets. A schematic view of gasketed plate heat exchanger is show in Figure 2.9. Plates have generally surface enhancements like corrugated patterns which promote turbulence and increase heat transfer area by 20 to 30% relative to projected area [5]. High turbulence however, leads to high frictional pressure losses but these can be maintained at a reasonable value because of reduced length of channels due to its good heat transfer features.

![Schematic view of gasketed plate heat exchanger](image)

Figure 2.9: Schematic view of gasketed plate heat exchanger [39]

The adoption of gaskets permits easy dismantling for mechanical cleaning. However, gaskets impose limits on the maximum pressures tolerated which are normally around 10bar, which is also the maximum pressure for an ammonia CC-OTEC [39]. The effect of seawater and working fluid on gaskets must also be verified since the risk of leakages in case of gasket failure is high [20]. Welded and brazed designs avoid this risk but could make mechanical cleaning impossible and chemical cleaning could be required [20]. Since channel flow areas are tighter than shell and tube flow areas, velocities in these exchangers are generally higher and consequently fouling is much less in plate exchangers than in tubular units [39]. Erosion problems can be significant because of high velocities and superior plate materials such as
titanium are desirable [39].

Specific plate exchangers for OTEC application have been proposed in particular by Japanese researchers [38] and companies such as Xenesys Inc. [40]. Uehara et al. [38] in particular in 1984 tested three types of shell and plate evaporator and two condensers with both ammonia and R22. Plate types tested for evaporator were the following:

- F plate: had fluted configuration on the heated surface;
- IP plate: working fluid passed through a lot of holes (3mm in diameter) drilled in the plate and impinged on the heated plates. A representation of flow passages is provided in Figure 2.10. The heated surface was covered in aluminum powder;
- P plate: plate surface on the working fluid side was covered with aluminum power and the water side had a number of flutes with drainages.

Tests showed that the P plate was superior to the others with both ammonia and R22. The overall heat transfer coefficient with ammonia was 4000-4500W/m\(^2\)K and 3500-4000 with R22. Optimum water velocities ranged between 0.8-1m/s.

![Figure 2.10: Flow configuration in the IP plates [38].](image)

Plate types tested as condensers had a number of large flutes along with several drainages on the working fluid side. The overall heat transfer coefficient for ammonia was between 3800 and 4500W/m\(^2\)K and 2000-3500W/m\(^2\)K with R22. As expected, the condenser overall
heat transfer coefficient is lower than that of evaporator because of lower heat transfer coefficient of condensing ammonia respect to ammonia evaporation. Even for condensers optimal velocities were 1m/s. Head losses on seawater side for both evaporator and condenser were around 4m.

The OTEC Okinawa power plant photographed in Figure 2.11 uses plate exchangers developed by Xenesys Inc. According to the Japanese company, lower pressure losses than conventional plate exchangers are obtained with a special design and structure that decreases unnecessary turbulence of the flow which is not related to heat transfer. The Xenesys design has both the distinctive features of a plate exchanger (high performance and compactness) and also similar characteristics of shell and tube exchanger (high pressure resistance) with a pressure drop which is approximately 1/3 than conventional plate exchangers [40]. According to Uehara et al. [38], despite enhanced surface shell and tube heat exchangers have good heat transfer properties and tolerable pressure drops, if titanium is used for their fabrication, their industrial production will be both difficult and costly. Moreover, platform space required by shell and tube heat exchangers is larger than the one required by plate exchangers and since, according to Avery et al. [5], the offshore plant configuration will be the most diffused one in future [5], low platform space required by heat exchangers is favorable.

![Figure 2.11: OTEC Okinawa plant in Kumejima with highlighted plate exchangers [15].](image-url)
2.3 Turbine

Turbine is an important component of the OTEC plant since its efficiency affects both performance and plant costs. According to Kostors et al. [39], each percentage point gained in turbine efficiency lead to 1% reduction on plant specific cost.

Two types of turbine have been traditionally considered for OTEC application: radial inflow and axial type turbines; their selection depends on the maximum efficiency potential for the turbine type at at given operating conditions [41].

In the Mini-OTEC plant a radial inflow turbine operating with 28200rpm and synchronous field generator operating at 3600rpm was used with ammonia as working fluid [41]. For the nominal warm and cold seawater temperatures, 93kW isoentropic power produced by the turbine was predicted. Turbine efficiency was 75%, however the small geared down generator caused the overall turbine-generator efficiency to be 56%, leading to 53kW gross power produced [5]. A radial inflow turbine is currently used at the 100kWe(net) Kumejima plant [15]. Nitesh et al. [41], developed a design and CFD analysis of a radial inflow turboexpander for small scale OTEC application of 2kWe capacity. The turbine worked with R22 and operated at a speed of 34000rpm. Inlet and outlet temperatures of working fluid were 24.5°C and 14°C, respectively. The small scale turbine had an efficiency of 67%.

In the Nauru pilot plant, an axial flow turbine was used with R22 as working fluid. This turbine operated with 20.55kg/s vapor flow at 24.8°C inlet temperature at 3000rpm speed. Direct coupling with an electric generator rotating at the same velocity was used and losses due to speed gear were avoided. Gross power produced by the turbine was expected to be 100kW. However, during operation turbine showed a higher efficiency than expected and 120kW gross electric power was produced leading to the historical record of 31.5kW net power produced by the plant [12]. Lockheed Martin for its 2.5MWe Mini Spar OTEC plant, also selected an axial turbine rotating at a speed of 1800rpm [27].

The radial inflow turbine is considered the preferable design in case of small units where single stage solutions can be adopted. Radial inflow turbines, with only one stage have a lower size than axial turbine and comparable efficiency; a speed gear is eventually required.

In case of large scale units, the double flow axial solution presented in Figure 2.12, reported the highest efficiencies [42]. Since the speed gear introduces approximately 2% losses it should be avoided for large turbines. In fact, as observed by Kostors et al. [42] and reported in Figure 2.13, in case of a double flow multistage turbine for a 10MWe net power module,
with ammonia flowrate of 366kg/s and inlet and outlet temperatures of 21°C and 10°C respectively, the three stage design with 1.02m base diameter reported the highest efficiency at 1800rpm speed. It was observed that higher efficiencies could be reached with higher shaft speed and higher base diameters. However, since a speed gear would be required the increase in efficiency is nullified by the presence of the speed gear. According to the same study, it resulted that a four stage turbine with 0.79m base diameter had 89.6% efficiency at 1800rpm speed and was the highest efficiency obtained according to mechanical constraints typical for turbines. In fact, the highest efficiency at 1800rpm was reached with a 5 stage solution with 0.69m base diameter. The efficiency was 0.9% better than the four stage solution, but violated the maximum blade length to diameter ratio of 0.3 imposed as a limit for the five stage solution. Moreover, adding a fifth stage at both ends of the double flow design would require adding to the plant cost with a four stage turbine 0.4% to 0.7% of total plant cost [42].

![Figure 2.12: The double flow, four stages ammonia turbine for 10MWe CC-OTEC [42].](image)

Variable inlet geometry entails benefits when the warm seawater differs from nominal conditions. The plant in fact is generally sized according to mean annual inlet seawater temperatures. Variable nozzles permit to the turbine to operate in optimal conditions even if ammonia inlet conditions differ from the design conditions. The use of variable nozzles will allow the same turbine design to be used at sites with different seawater temperatures. Because of low operating temperatures, materials proposed by McDonald et al. [43] for large
scale turbine operating with ammonia are low to medium alloys for the rotor and 12 percent chrome steel for blades also used in conventional steam turbines [42]. Kostors et al. [42], to improve stress corrosion cracking proposed annealed AISI 403 Type steel for blades.

Figure 2.13: Turbine efficiency normalized to the four stage, 1800 rpm value, versus velocity ratio for double flow axial turbine (below) [42].
2.4 Seawater pumps

Pumping power required to move seawater through the power plant is an important auxiliary electric consumption that affects directly net power output. The ratio between net electric power leaving the plant and electric power produced by the turbine is generally between 50% to 80% [14]. Therefore, seawater pumps performance, efficiency and sizing are influential parameters for overall plant design. Because of large seawater flowrates and low pressure heads required, the axial flow design is indicated [31, 44]. The report on seawater pumps for OTEC by Little [31] affirms that 4m$^3$/s of cold seawater are required to produce 1MWe net power with an OTEC plant. Warm seawater required is nearly the same. Therefore, a 100MWe OTEC plant requires 8 large pumps with 100m$^3$/s volumetric flowrate to handle cold and warm seawater flows. According to the same report, pressure head between 2 and 5 m are expected for OTEC operation.

In case of grazing or moored plans, pump is submerged at 100m depth to prevent cavitation and to lower center of gravity of the barge. In a grazing or moored plant, the submersed pumps are also subjected to plant motion, thus long shafts to connect surface electric motor to pump rotor are discouraged. Moreover, low head, low speed pumps required in OTEC operation are not common. Therefore, the best pump design for OTEC is not an off the shelf design, but special designs for the specific application should be developed [31]. An example is the self contained bulb pump arrangement for 100m$^3$/s volume flowrate and 3.5m head loss of Figure 2.14. The pump/diffuser is long 26m with a rotor tip diameter of 4.9m. The flow is straight through with minimum deviations and obstructions and diffuser reduces velocity at stator outlet from 8.5m/s to 2m/s at the diffuser outlet [31]. The diffuser consists in annular diffuser downstream rotor stator blades, followed by a conical diffuser. Blades are directly driven by the motor and no transmission is required; however, since the impeller velocity is low, a 15Hz AC synchronous motor was proposed [31]. This design has the advantage to be a plug-in unit and assembly on site of pump and diffuser is not required. Hydraulic efficiency of this pump was 76%, however according to Little [31], efficiencies of 86%-88% can be reached with further development especially in fluid dynamics. The more recent study by Bombarda et al. [26] in fact, consider seawater pump hydraulic efficiency to be 85%.
Seawater pumps are composed of moving parts operating in a hostile environment. Seawater impurities have a negative effect on moving parts of the pump. A high need of maintenance is required [44].

An innovative pumping system called “Airlift” have been proposed by Kleute et al. [44]. Air is injected at inlet flow by means of an external/internal air tube. The fluid is aerated and density reduced. The result is an increase in pressure difference over the inlet of the pipe and liquid start to flow upwards. Article’s authors affirm that this solution is 25-30% more efficient than conventional pumps and requires little maintenance. However further research is required to study the effective “Airflow” feasibility.

Figure 2.14: Bulb seawater pump for OTEC application [31]
2.5 Selection of components for near term CC-OTEC development

According to literature review conducted in the previous sections, the main features of OTEC closed cycle components are resumed in the following table:

Table 2.1: Resume of literature review on OTEC closed cycle components.

<table>
<thead>
<tr>
<th>Component</th>
<th>Type/material</th>
<th>Main Features</th>
</tr>
</thead>
<tbody>
<tr>
<td>CWP</td>
<td>HDPE</td>
<td>• Successfully employed in OTEC pilot plants;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Constraints on maximum diameter (&lt;2.5m).</td>
</tr>
<tr>
<td></td>
<td>FRP</td>
<td>• Promising material for larger diameters;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Tests currently underway.</td>
</tr>
<tr>
<td>Heat</td>
<td>Titanium S&amp;T</td>
<td>• High heat transfer coefficients with surface enhancements;</td>
</tr>
<tr>
<td>exchangers</td>
<td>exchangers</td>
<td>• Acceptable pressure drop;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Well understood 2-phase behavior;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Large volumes required;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Industrial production of titanium enhanced surface S&amp;T exchangers difficult and costly.</td>
</tr>
<tr>
<td></td>
<td>Titanium plate</td>
<td>• High heat transfer coefficients;</td>
</tr>
<tr>
<td></td>
<td>exchangers</td>
<td>• High compactness;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Promising for large scale titanium exchangers dedicated to OTEC production;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Higher pressure drops than S&amp;T exchangers. Special designs to reduce pressure drop have been recently proposed.</td>
</tr>
<tr>
<td></td>
<td>Aluminum plate</td>
<td>• Low cost solution</td>
</tr>
<tr>
<td></td>
<td>and fin</td>
<td>• Low resistance to pitting corrosion lead to reduced lifetime.</td>
</tr>
<tr>
<td></td>
<td>exchangers</td>
<td></td>
</tr>
<tr>
<td>Turbine</td>
<td>Radial inflow</td>
<td>• Compact in size;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• High rotational speeds: gearbox required;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Acceptable efficiency on small scale.</td>
</tr>
<tr>
<td></td>
<td>Axial flow</td>
<td>• Typical of large scale plants but also used on smaller sizes (100kWe net);</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Direct coupling;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• High efficiencies.</td>
</tr>
<tr>
<td>Seawater</td>
<td>Axial flow</td>
<td>• Large flowrates;</td>
</tr>
<tr>
<td>pump</td>
<td></td>
<td>• Low pressure head.</td>
</tr>
</tbody>
</table>
3. Methodology

From literature review conducted in Chapter 1 and Chapter 2 on plant configurations, history of technology development and components, the following conclusions can be deduced:

- The closed cycle solution is the most mature and it is technically feasible in the near term;
- Small scale CC-OTEC pilot plants operated successfully;
- Technical feasibility was confirmed by pilot plants, however high investment costs uncertainties, especially on cold water pipe, prevented large scale diffusion of this technology.

Because of these considerations, in this thesis work, techno-economic optimization of closed cycle OTEC plants has been conducted. Methodology used to perform the analysis is resumed in Figure 3.1. Parameters assumed as input in the techno-economic optimization were:

- Cold seawater inlet temperature;
- Warm seawater inlet temperature;
- Seawater salinity;
- Cold seawater mass flowrate;
- CWP and warm water pipe lengths and diameters;
- Turbine isentropic, mechanical and electrical efficiencies;
- Seawater and working fluid pumps hydraulic, mechanical and electrical efficiencies.

Selection of cold water pipe mass flowrate and CWP diameter were necessary because according to literature review in section 2.1, limitations on maximum CWP diameter exist. If a too large seawater flowrate is used for the CWP diameter assumed, pressure drops along CWP would be too high with a negative effect on net power produced. According to this consideration, a CWP diameter equal to 2.5m was selected for all analyses. This diameter is the limit for pipes made in HDPE that are commercially available [27, 34].
Figure 3.1: Resume of methodologies used to perform techno-economic analysis of closed cycle OTEC plants.
Cold seawater velocity in CWP is generally in a range between 0.7-2m/s in order to contain pressure drop [26]. A cold seawater flowrate of 8500kg/s was assumed for all the analyses. Since no commercial CC-OTEC plants have been constructed yet, high uncertainties on components costs exist. An approximate techno-economic optimization of CC-OTEC plants independent of economic data has been performed. Since heat exchangers’ cost is about 25-50% of the total plant cost of CC-OTEC plants the following objective function has been selected for techno-economic optimization [22, 35]:

$$\gamma = \frac{\dot{W}_{net}}{Total\ heat\ exchangers\ area}$$  \hspace{1cm} (3.1)

The analysis was divided in three main parts. In the first part, working fluid selection was conducted with a single level closed cycle model and optimization results for the best working fluid are observed. Sensitivity analysis to observe the effect of input parameters variations was also conducted for the selected working fluid. In the second part, the effect of increasing the number of evaporation levels with the previously selected working fluid on the $\gamma$ parameter was studied with a multilevel closed cycle model. In the first and second part, simplifying assumptions on heat exchangers were introduced to perform calculation. In the third part, since the single level configuration is mature for CC-OTEC plants, a maximization of the $\gamma$ parameter with the previously selected working fluid but with a detailed single level closed cycle model was conducted. Sizing of heat exchangers and an economic analysis of the power plant operating in the optimal configuration found with the detailed model was also developed.

Since detailed modeling of OTEC heat exchangers is a difficult task [5], numerical models developed for the first and second part assumed a constant overall heat transfer coefficient and an approximate method to estimate pressure drop on seawater side.

Optimization variables selected in the first part were:

- Working fluid used in the closed cycle;
- Seawater $\Delta T$ across evaporator;
- Seawater $\Delta T$ across condenser;
- Pinch point temperature difference at the evaporator;
• Pinch point temperature difference at the condenser.

A simplified model of a single level CC-OTEC plant have been developed using MATLAB. Working fluid properties were evaluated using REFPROP [45], while seawater thermodynamic properties have been evaluated with the TEOS-10 [46] equations of state for seawater assuming a constant salinity of 35g/kg for all calculations. An optimization tool which searched for the optimal seawater ΔTs and pinch point was developed by using the MATLAB function fmincon.

In order to perform working fluid selection, the optimization tool was embedded in the single level closed cycle model. The embedded model requires the value of input parameters presented at the beginning of this Chapter and a list of the working fluids to be tested. The embedded model performs maximization of the γ parameter for all fluid included in the list and provides information on Rankine cycle thermodynamic points, gross power produced by the cycle, net electric power produced and heat transfer areas required by heat exchangers. A sensitivity analysis by varying some the input parameters required by the model and assumptions on heat exchangers performance has been conducted with the selected working fluid.

Optimization variables to study the effect of a multilevel configuration with the selected working fluid were:
• Number of evaporation levels;
• Seawater ΔT across the evaporator series;
• Seawater ΔT across the condenser series;
• Pinch point temperature difference at the evaporator (assumed equal for all evaporators);
• Pinch point temperature difference at the condenser (assumed equal for all condenser).

A numerical model of a multilevel CC-OTEC plant has been developed and the same optimization tool for the single level optimization was embedded. In order to perform γ maximization, the embedded model for the multilevel CC-OTEC configuration requires input parameters presented at the beginning of this Chapter, the desired working fluid and the desired number of evaporation levels. Even in this case the model output reports, apart from the maximum γ parameter value, information on thermodynamic points of the cycles,
gross power produced by the Rankine cycles, net power, heat transfer area required etc.

Optimization results with the selected fluid for the single cycle level and the multilevel plant obtained in the first two parts have been compared.

In the third part, in order to perform a detailed techno-economic optimization of the single level CC-OTEC plant, an approximate model of plate heat exchangers was developed by using Aspen Exchangers Design and Rating (Aspen EDR) [47] and MATLAB. Correlations between heat exchangers’ geometrical parameters, seawater velocity in channels, overall heat transfer coefficient and seawater side pressure drop were developed. Plate exchangers were selected because of favorable features emerged in literature review (see section 2.2.5 for details). The approximate model of plate exchangers was then embedded in the single level closed cycle model. A new optimization tool has been developed. Optimization variables of the new optimization tool are the following:

- Seawater ΔT across evaporator;
- Seawater ΔT across condenser;
- Pinch point temperature difference at the evaporator;
- Pinch point temperature difference at the condenser;
- Plate evaporator geometrical parameters;
- Plate condenser geometrical parameters.

In order to perform optimization, parameters required by the new embedded model are resumed at the beginning of this Chapter. In this case however, the program provides also information about the heat exchangers geometries and number of plates required that maximized the γ parameter. Even in this case, the model output reports information on thermodynamic points of the cycles, gross power produced by the Rankine cycles, net power produced, heat transfer area required etc.

Plate geometry and number, mass flowrates and temperatures of streams in heat exchangers provided by the detailed numerical model of the power plant in the maximum γ configuration, has been used in Aspen EDR to perform sizing of evaporator and condenser. With the heat exchangers’ cost provided by Aspen EDR and cost estimations found in literature for other plant components, an estimation of plant costs in the configuration with maximum γ found with the detailed model has been realized.
Chapter 3

Finally, by introducing simplifying assumptions for plant operation during the year, the Levelized cost of Energy (LCOE) of the plant was determined.
4. Thermodynamics of CC-OTEC

According to advantages introduced in section 1.4.2, the closed cycle solution is the most common plant layout for OTEC plants. In this chapter thermodynamic limits of an OTEC plant are introduced. Several ideal cycles can be studied in order to have an idea of real closed OTEC cycles behavior without implementing the specific model of the real thermodynamic cycle which should be effectively used. Ideal cycles in fact, are effectively used to estimate OTEC potential of a site in the Ocean [48]. The Rankine cycle for OTEC plants is also introduced and compared with ideal cycles. Exergy analysis of the Rankine cycle is also performed. The multilevel Rankine cycle solution and a cycle that uses a working fluid with temperature glide during evaporation and condensation are finally introduced.

4.1 Ideal OTEC cycles

Ocean thermal energy conversion uses temperature difference between the warm surface seawater and the cold deep seawater to produce useful power by interposing a thermodynamic cycle between these two thermal reservoirs. Several ideal cycles can be introduced to explain thermodynamic limits of OTEC plants. These cycles, also shown in Figure 4.1, are:

- A Carnot cycle operating between warm seawater and cold seawater inlet temperatures. This cycle is the ideal cycle with the highest efficiency because uses infinite seawater flowrates and infinite heat transfer areas (Figure 4.1a);
- A triangular cycle in which a finite cold seawater flowrate is brought to the warm flowrate temperature. This cycle has infinite warm seawater flowrate and infinite heat transfer areas (Figure 4.1b);
- A parallelogram cycle with finite warm and cold seawater flowrate but infinite heat transfer areas (Figure 4.1c);
- An “Ideal CC-OTEC” with finite seawater flowrates and infinite heat transfer areas (Figure 4.1d);
- An “Ideal CC-OTEC” with finite seawater flowrates and finite heat transfer area
Chapter 4

(Figure 4.1e).

By passing from the Carnot cycle to the ideal CC-OTEC, idealities such as infinite flowrate or infinite heat transfer areas are progressively removed in order to approach real cycles behavior.

![Figure 4.1: Ideal cycles for CC-OTEC qualitative T-s diagrams](image)

In conventional thermodynamic cycles, where usually combustion takes place, the only limit in cycle maximum temperature is due to the maximum mechanical and thermal stresses tolerated by materials. This is not the case of OTEC plants or other similar cycles that use low-grade heat as thermal source. In case of OTEC, the temperature difference between warm and cold reservoir is in the order of 20°C. Therefore, since the temperature difference is low, the theoretical limit on cycle efficiency is low. If it is assumed to pump cold seawater at 4°C to the plant, and the surface temperature is between 24°C and 30°C, the efficiency of the Carnot cycle operating between these temperatures lies between 6.7% and 8.6%. Therefore, the efficiency of ideal cycles different from the Carnot cycle are lower than this benchmark.

According to basic thermodynamics, a dead state is a particular equilibrium state at which mechanical work cannot be extracted from the system. In conventional power plants, e.g. a natural gas combined cycle, the ambient temperature is a convenient dead state. In case of an OTEC power plant, a convenient dead state can be warm seawater of the upper ocean mixed layer. If such a dead state is assumed, reversible specific work that can be extracted by cold seawater, considering seawater as an incompressible fluid and neglecting pressure effect, has the following expression:

\[
L_{rev} = c_p \left( T_{c,sw} - T_{ds} \right) - T_{ds} \ln \left( \frac{T_{c,sw}}{T_{ds}} \right)
\]  

(4.1)
Where $c_p$ is seawater specific heat, $T_{c,sw}$ is cold seawater absolute temperature, while $T_{ds}$ is dead state temperature. If warm seawater at $28^\circ C$ is considered as the dead state and $4^\circ C$ is the cold seawater temperature in its initial state, considering a specific heat of 4kJ/kg [46], the reversible work per unit of cold seawater flowrate is 4.04kJ/kg. The ideal cycle which is able to obtain such a value of specific work is the triangular cycle shown in Figure 4.1b. The efficiency of the triangular cycle is described by the following equation:

$$\eta_{\text{Triangular}} = 1 - \frac{T_{ml,c}}{T_{ds}} = 1 - \frac{T_{ml,c}}{T_{w,sw}}$$ (4.2)

Where $T_{ml,c}$ is the cold seawater log mean temperature when it is heated up from $4^\circ C$ to the previously selected dead state temperature, $T_{ds}$.

However, a triangular cycle like the one introduced above, where thermal power is introduced by seawater at constant temperature cannot be effectively realized. In a closed cycle OTEC plant, seawater at ocean surface temperature is cooled by flowing through a heat exchanger, while deep ocean seawater is pumped to a heat exchanger and heated up by thermal power leaving the thermodynamic cycle.

According to this observation, an ideal cycle that better describes the seawater behavior through exchangers is the parallelogram cycle, also depicted in Figure 4.1c. The efficiency of the parallelogram cycle is:

$$\eta_{\text{par}} = 1 - \frac{T_{ml,c}}{T_{ml,w}}$$ (4.3)

Where $T_{ml,c}$ and $T_{ml,w}$ are the log-mean cold and warm seawater temperatures, respectively.

The efficiency of the parallelogram cycle is lower than the efficiency of the Carnot cycle. Compared to the triangular cycle, the parallelogram cycle shows, as seen in Figure 4.2, two important irreversible losses when warm inlet seawater temperature is used as the dead state:

- Cold seawater is discharged at a temperature lower than the dead state temperature;
- Warm seawater is cooled in the warm heat exchanger and discharged at a temperature lower than the dead state temperature.

Therefore, the efficiency of the parallelogram cycle is also lower than that of a triangular cycle. However, the parallelogram cycle is a more realistic benchmark of the maximum obtainable efficiency from a CC-OTEC power plant.
If the cold and warm seawater inlet temperatures and their respective ΔTs are imposed, the specific work per unit cold seawater flowrate obtainable with a parallelogram cycle is shown in Figure 4.3. The effect of pumping power required to move seawater is not considered. Results have been presented in terms of specific work per unit cold seawater flowrate because, as seen in section 2.1, the cold water pipe geometry limits the cold seawater flowrate which can be used in the plant. Therefore, specific work is independent of the cold water pipe geometry.

By neglecting pressure effects, the thermal power specific to cold seawater flowrate leaving the cycle is expressed as:

\[ Q_{out} = c_p \Delta T_{c,sw} \]  

(4.4)

If the ΔT endured by warm seawater is also imposed, the efficiency of the parallelogram cycle can be defined according to equation (4.3). Specific work obtained by the parallelogram cycle can be finally calculated:

\[ L_{id} = \frac{Q_{out}}{1 - \eta_{par}} \frac{1}{\eta_{par}} \]  

(4.5)

According to Figure 4.3, the maximum specific work can be obtained for a null ΔT_{w,sw} and for a ΔT_{c,sw} equal to the temperature difference between warm and cold inlet temperatures: this condition corresponds to the triangular cycle discussed above. In fact, if warm seawater
inlet temperature is 28°C and cold seawater inlet temperature is 4°C, the maximum specific work for the optimal seawater \( \Delta T_s \) is equal to 4.04 kJ/kg.

Figure 4.3: Specific work of parallelogram cycle according to seawater \( \Delta T_s \). Pressure losses on seawater side not considered.

Although the parallelogram cycle follows well the cooling and heating seawater paths through heat exchangers, a fluid that is actually able to perfectly follow both the warm and cold seawater temperature variations does not exist. According to section 1.4.2, the saturated vapor Rankine cycle is the most common closed cycle used for real closed cycle OTEC plants. A Carnot cycle as the one sketched in Figure 4.1d, can be considered as the ideal corresponding of a saturated vapor Rankine cycle with unitary turbomachinery efficiencies and without the working fluid preheating section [26]. The efficiency of this ideal CC-OTEC is:

\[
\eta_{id,CC-OTEC,infiniteHX} = 1 - \frac{T_{min,cycle}}{T_{max,cycle}} = 1 - \frac{T_{in,csw} + \Delta T_{c,sw}}{T_{in,w,sw} - \Delta T_{w,sw}} \tag{4.6}
\]

Therefore, according to equation (4.6), the maximum temperature reached in this ideal closed cycle is warm seawater outlet temperature, while minimum temperature reached of the closed cycle is the cold seawater outlet temperature. An irreversible loss is introduced because this
ideal CC-OTEC introduces a temperature difference between seawater and working fluid in both the cold and warm exchangers. Warm and cold seawater outlet temperatures correspond with working fluid maximum and minimum temperatures, respectively. In order to avoid infinite heat transfer surfaces, a finite temperature difference between seawater and working fluid at seawater heat exchangers outlet is required. The cycle shown in Figure 4.1e is an ideal CC-OTEC with finite heat transfer areas. The ideal CC-OTEC efficiency with finite heat transfer surfaces is:

\[ \eta_{id,CC-OTEC} = 1 - \frac{T_{\text{min,cycle}}}{T_{\text{max,cycle}}} = 1 - \frac{T_{\text{in,c,sw}} + \Delta T_{c,sw} + \Delta T_{pp}}{T_{\text{in,w,sw}} - \Delta T_{w,sw} - \Delta T_{pp}} \]  

(4.7)

Where \( \Delta T_{pp} \) is the pinch point temperature difference and \( T_{\text{max,cycle}} \) and \( T_{\text{min,cycle}} \) are maximum and minimum temperature of working fluid in the ideal CC-OTEC. The pinch point temperature difference value generally comes from a techno-economic optimization between the power output and the heat exchangers area. A value of 2°C [26] for both warm and cold seawater pinch point is assumed in the following calculations. As seen in Figure 4.1e, another irreversible loss is introduced because of pinch points presence, which reduce the temperature difference between maximum and minimum temperature of the ideal CC-OTEC.

The same procedure described by equations (4.4) and (4.5) is performed for the ideal CC-OTEC cycles by substituting the parallelogram cycle efficiency in equation (4.5) with the ideal CC-OTEC efficiencies of equations (4.6) and (4.7). The specific work according to both warm and cold seawater \( \Delta T_s \)s can be calculated. Results appears in Figure 4.4 compared with the parallelogram cycle results. Specific work obtained with the ideal CC-OTEC cycles is lower than specific work obtained by the parallelogram cycle because of the irreversible losses due to temperature difference between seawater and working fluid introduced. Moreover, as expected, the specific work obtained by the cycle with pinch points different than zero is lower than the specific work obtained with CC-OTEC with null pinch points. In fact, the temperature difference between seawater and working fluid in the heat transfer processes of the ideal CC-OTEC with finite heat transfer area is higher than the ideal CC-OTEC with infinite heat transfer area. Therefore, the ideal CC-OTEC with pinch points operates with higher irreversible losses than the same cycle with infinite surfaces, resulting in a lower temperature difference between its maximum and minimum temperatures.
Figure 4.4: Specific work of the ideal-CC OTEC compared with specific work obtained by a parallelogram cycle. Pressure losses on seawater side are not considered. Pinch point value is reported in figure.
As observed in Figure 4.4. Both the ideal CC-OTEC with infinite and finite heat exchangers’ surfaces show the same trends. By increasing $\Delta T_{w,sw}$, the specific work of both the ideal CC-OTEC cycles is reduced because the ideal cycle maximum temperature becomes lower. For a fixed $\Delta T_{w,sw}$, according to equations (4.4) and (4.5), if $\Delta T_{c,sw}$ is low, specific thermal power leaving the cycle is low and thus the specific work obtained is low. If $\Delta T_{c,sw}$ becomes greater, the specific thermal power leaving the ideal CC-OTEC cycles is proportionally higher. However, by increasing $\Delta T_{c,sw}$, the minimum temperature of working fluid in the ideal CC-cycles becomes higher and, according to the efficiency definition of equations (4.6) and (4.7), the ideal CC-OTEC efficiency is reduced. The increase in specific work due to higher thermal power leaving the cycle, is contrasted by a lower cycle efficiency. Therefore, the optimal $\Delta T_{c,sw}$ is a trade off between these two contrasting aspects.

The performance of the parallelogram cycle can be approximated by using a certain number of ideal CC-OTEC in series. A graphical description of this ideal multilevel closed cycle OTEC appears in Figure 4.5. The obtained specific work is the sum of specific works produced by each one of the cycles. The higher the number of cycles adopted, the higher the specific work of the overall system because temperature differences between seawater and working fluid in the ideal CC-OTEC are reduced. If the same concept of varying both the cold and warm seawater $\Delta T$s in order to obtain the maximum specific work is applied to this multilevel solution, results appear in Figure 4.6. By increasing the number of cycles the surface of the multilevel ideal CC-OTEC approaches the Lorentz cycle surface.

For an infinite number of levels and with null pinch point the Lorentz cycle surface can be reached.

![Figure 4.5: Multilevel ideal CC-OTEC](image-url)
Figure 4.6: Maximum specific work with multilevel ideal CC-OTEC. Pinch point was assumed 2°C for both exchangers and seawater pressure drop was not considered.
Chapter 4

The discussion developed so far, neglected the effect of seawater pumps consumption. As
clarified in section 2.4, seawater pumps are required to bring warm and cold seawater to
their respective heat exchangers. The net specific work per unit cold seawater flowrate
therefore becomes:

\[ L_{\text{net}} = \frac{\dot{W}_{\text{gross,ideal cycle}} - \dot{W}_{c,sw,pump} - \dot{W}_{w,sw,pump}}{m_{c,sw}} \] (4.8)

Where \( \dot{W}_{c,sw,pump} \) and \( \dot{W}_{w,sw,pump} \) are ideal cold and warm seawater pumps consumptions
respectively, obtained both from the following equation:

\[ \dot{W}_{sw,pump} = \frac{\dot{m}_{sw}}{\rho} \Delta p_{sw} \] (4.9)

Where \( \dot{m}_{sw} \) is warm or cold seawater flowrate, \( \rho \) is seawater density and \( \Delta p_{sw} \) the overall
pressure drop on warm or cold seawater side. If cold seawater flowrate \( \dot{m}_{c,sw} \) is assumed,
\( \dot{W}_{\text{gross,ideal cycle}} \) is gross power produced by thermodynamic cycle, obtained with the
following equation:

\[ \dot{W}_{\text{gross,ideal cycle}} = \frac{\dot{Q}_{\text{out}}}{1 - \eta_l} = \frac{\dot{m}_{c,sw} c_p \Delta T_{c,sw}}{1 - \eta_l} \] (4.10)

Warm seawater flowrate required to compute warm seawater pump consumption is obtained
by the following equation:

\[ \dot{m}_{w,sw} = \frac{\dot{Q}_{\text{in}}}{c_p \Delta T_{w,sw}} = \frac{\dot{Q}_{\text{out}} + \dot{W}_{\text{gross,ideal cycle}}}{c_p \Delta T_{w,sw}} \] (4.11)

If a pressure drop is assumed for both warm and cold seawater, net specific work can be
obtained with equation (4.8).

Referring to Figure 4.7, if a seawater pressure drop of 0.3 bar is assumed for both cold and
warm seawater [26] and both \( \Delta T_{sw} \) are varied in order to search for the maximum net specific
work using an ideal CC-OTEC, it appears that the optimum \( \Delta T_{w,sw} \) is greater than zero
because, in case of a null \( \Delta T_{w,sw} \), the requested warm seawater flowrate would be infinite
and the pumping power related to warm seawater would consequently be infinite too. By
increasing \( \Delta T_{w,sw} \) however, the ideal CC-OTEC efficiency calculated with equation (4.7) is
reduced and the optimal warm seawater temperature difference results from a trade off
between a high seawater pump consumption and a low cycle efficiency. As for the maximum
specific work of an ideal CC-OTEC neglecting pressure losses (see Figure 4.4), the optimal
\( \Delta T_{c,sw} \) is the result of a trade off between high thermal power leaving the cycle (which is
proportional to gross specific work in this case) and a high ideal cycle efficiency.

Figure 4.7: Maximum net specific Ideal CC-OTEC work according to seawater $\Delta T$s. Pinch point was 2°C for both exchangers and 0.3bar was assumed as pressure drop for both warm and cold seawater.

4.2 Single level Rankine cycle OTEC plant

4.2.1 Maximum net specific work
The single level Rankine cycle is the most common plant layout for closed cycle OTEC
plants. According to section 1.4.2, ammonia is considered as one of the fluids with desirable features for CC-OTEC. An ammonia Rankine cycle T-s diagram is reported in Figure 4.8. The Rankine cycle however, differs from the ideal CC-OTEC because of these non idealties:

- Preheating is required to bring working fluid temperature from the pump outlet temperature to the evaporation temperature;
- Turbomachinery have an isentropic efficiency lower than unity;
- As shown in Figure 4.9, in the Rankine cycle, the pinch point is not positioned at warm seawater outlet but at the end of the preheating section. If the same $\Delta T_{w,sw}$ and the $\Delta T_{pp}$ are selected for both the ideal CC-OTEC and Rankine cycle, the maximum temperature of the Rankine cycle will be higher than that of the ideal CC-OTEC.

![Figure 4.8: Ammonia Rankine cycle for OTEC T-s diagram](image)

The ideal CC-OTEC have been substituted with an ammonia Rankine cycle and assuming the same pinch points of 2°C at both evaporator and condenser, the search for the optimal combination of seawater $\Delta T$s have been conducted. The pressure drop was maintained equal to 0.3bar for both the warm and cold seawater pump. The same procedure has been also performed introducing turbomachinery efficiencies in the Rankine cycle. According to Bombarda et al. [26] constant isentropic turbine efficiency of 89% and constant working
fluid pump efficiency of 80% was selected. Results are reported in Figure 4.10. The ideal CC-OTEC and the Rankine cycle with unitary turbomachinery efficiencies surfaces are so similar that cannot be distinguished: the irreversible loss due to the presence of preheating in the Rankine cycle is compensated by the higher evaporation temperature reached because of different pinch point positioning. Hence, it can be stated that the ideal CC-OTEC, which in reality is a simple Carnot cycle interposed between two variable temperature sources, is a good approximation of the Rankine cycle behavior operating between the same sources. Thus, the Carnot cycle can be used for a preliminary estimation of the maximum performance achievable with a closed cycle with ammonia. If turbomachinery efficiencies are imposed, the Rankine cycle surface is lowered because gross power produced by the Rankine cycle is reduced. With the turbomachinery efficiencies introduced, the maximum net specific work for the Rankine cycle occurs for \( \Delta T_{w,sw}=1.75^\circ C \) and \( \Delta T_{c,sw}=8.25^\circ C \) and its value is 0.826 kJ/kg\(_{c,sw}\), while for the ideal cycle in the same conditions net specific work is 0.970 kJ/kg\(_{c,sw}\). If a cold seawater flowrate of 8500 kg/s is assumed, the net power produced by the plant with the ammonia cycle, operating in the aforementioned conditions, would be approximately 7 MWe.

![Figure 4.9: TQ diagram at the evaporator of the Ideal CC-OTEC and the Rankine cycle for OTEC](image-url)
Chapter 4

4.2.2 Exergy analysis

The first law efficiency of the optimal ammonia closed cycle operating in the optimal conditions of section 4.2.1 neglecting the seawater pumps consumption has the following expression:

Figure 4.10: Maximum net specific work of a Rankine cycle for OTEC compared with the ideal CC-OTEC. A pinch point of 2°C was selected for both the cycles. Pressure drop was assumed to be 0.3 bar for both warm and cold seawater. Isentropic turbine efficiency for the turbine was 89% and working fluid pump efficiency was 80%
\[ \eta_{1,\text{gross}} = \frac{W_{\text{gross,Rankine cycle}}}{\dot{Q}_{\text{in}}} \]  

(4.12)

Where \( W_{\text{gross,Rankine cycle}} \) is the gross power produced by the ammonia Rankine cycle and \( \dot{Q}_{\text{in}} \) is thermal power entering the ammonia cycle. The value of first law efficiency in the aforementioned conditions is 2.94\%. Thus, only 2.94\% of thermal power exchanged with warm seawater is converted in gross power by the Rankine cycle. The remaining 97.06\% is rejected to the condenser.

Exergy analysis of the ammonia Rankine cycle operating in the optimal conditions of section 4.2.1 have been conducted in order to compare the Rankine cycle performance with that of a Lorentz cycle operating with the same seawater \( \Delta T_s \). Ambient temperature has been set to the lower mean log temperature of cold seawater:

\[
T_{\text{amb}} = \frac{\Delta T_{c,sw}}{\ln\left(\frac{T_{\text{in,c,sw}} + \Delta T_{c,sw} + 273.15}{T_{\text{in,c,sw}} + 273.15}\right)}
\]  

(4.13)

Exergy analysis results are depicted in Figure 4.11. The second principle efficiency of the Rankine cycle is 44.76\%, which means that the Rankine cycle produces 44.76\% of power produced by a Lorentz cycle operating with the same seawater temperature differences. More than 86\% of exergy destroyed is due to heat transfer. Exergy destroyed by turbomachinery depends on the values of isentropic efficiencies selected: in this case turbine and pump isentropic efficiencies of 89\% and 80\% were assumed, respectively.

**OTEC Rankine cycle exergy analysis**

![Exergy analysis of Rankine cycle for OTEC](image)

Figure 4.11: Exergy analysis of Rankine cycle for OTEC
Since exergy losses in heat transfer processes are due to finite temperature differences occurring between seawater and working fluid, it appears evident that if the temperature difference between seawater and working fluid in the heat transfer sections would be lower, the exergy efficiency would grow and a performance similar to an ideal cycle can be approached. A better performance with real cycles, from the exergy efficiency standpoint, can be obtained by implementing a multiple evaporation levels solution or by using fluid with temperature glide during phase change as working fluid. The use of advanced cycles such as the Kalina and Uehara cycles (see section 1.4.3 for details) can also improve exergy efficiency.

4.3 Multilevel Rankine cycle OTEC plant

A numerical model of a multilevel Rankine cycle OTEC plant, as the one shown in Figure 4.12, has been implemented. The model was created in such a way that the same input data of a single level solution are required. As can be seen in Figure 4.13(left), by increasing the number of evaporation levels it was observed that, by maintaining a constant pressure drop of 0.3bar as for the single level solution, the net specific work obtained increased. In fact, by increasing the number of evaporation levels, the Rankine cycle in each level operates with a lower mean log temperature difference in the heat exchangers, reducing exergy destruction in the heat transfer process.

![Figure 4.12: Multilevel Rankine cycle for OTEC plant scheme](image-url)
In fact, as observed in Figure 4.13(right), the cumulated Rankine cycle efficiency, obtained as the ratio between the sum of the gross power produced by each Rankine cycle and overall thermal power entering the plant, becomes higher and the efficiency of the ideal parallelogram cycle, with a large number of evaporation levels is approached. However, the slope of the cumulated Rankine cycles efficiency decreases as the number of levels becomes higher and even if null pinch points and unitary turbomachinery efficiencies are assumed, the efficiency of the parallelogram cycle cannot be reached. The slope reduction observed, is due to the presence of working fluid preheating. In fact, as shown in Figure 4.14, while by increasing the number of evaporation levels the exergy destroyed by the evaporation section becomes lower, the exergy destroyed by the preheating section becomes more and more relevant because adding an evaporation level means adding a preheating section with a finite temperature difference between seawater and working fluid that cannot be avoided.

Figure 4.13: Specific work produced by the Rankine cycles series (left). Efficiency of cumulated Rankine cycle compared with efficiency of the parallelogram cycle (right). Unitary turbomachinery efficiencies and null pinch points were assumed.

Figure 4.14: Exergy destroyed by the evaporation and preheating sections according to number of evaporation levels
4.4 Ideal OTEC cycle with generic zeotropic mixture as working fluid

According to Chys et al. [49] and as demonstrated in section 4.2.2, the largest shortcoming of a Rankine cycle operating with a pure fluid is the fact that evaporation and condensation processes occur isothermally, leading to large exergy losses. Another way to maximize power output from an OTEC cycle and consequently improve exergy efficiency is the adoption of zeotropic mixtures as working fluid. The use of mixture broadens the choice of working fluids suitable for OTEC, and efficiency and output improvement are reached because of a temperature glide during phase change, which reduces the temperature difference between two fluids involved in the heat transfer process and exergy losses due to evaporation and condensation are reduced. Research on zeotropic mixtures used as working fluid in Rankine cycles is limited, however, as reported by [49], several works proved the increase in efficiency in ORC application when used instead of pure fluids.

Since the number of possible zeotropic mixtures is high, in order to maintain the maximum abstraction possible, the effect of a temperature glide in a Rankine cycle operating in OTEC conditions have been assessed without selecting a specific mixture. Parallelogram cycles as the ones drawn in Figure 4.15, similar to the ideal parallelogram cycle shown in Figure 4.1c, were assumed to model the generic mixture behavior. For the generic zeotropic mixture ideal cycle however, an equal $\Delta T_{\text{glide}}$ for the warm and cold heat transfer section and a pinch point higher than zero was selected instead.

![Figure 4.15: Ideal cycles with generic zeotropic mixture qualitative T-s diagrams with different pinch points positioning highlighted with a green circle.](image-url)
The assumption of an equal glide for both the evaporation and condensation process have been found reasonable because of the generally low pressure difference between evaporation and condensation. If $\Delta T_{\text{glide}} < \Delta T_{w,\text{sw}}$, the pinch point was positioned in point 2 while if $\Delta T_{\text{glide}} > \Delta T_{w,\text{sw}}$, pinch point is positioned in point 3. The same happens at the condenser: if temperature glide is higher than $\Delta T_{c,\text{sw}}$, pinch point is positioned at point 1, vice versa pinch point occurs at point 4. No preheating and unitary turbomachinery efficiency was assumed.

Once cold seawater flowrate and $\Delta T_{c,\text{sw}}$ are selected, gross power obtained by the cycle is calculated with equation (4.10). The efficiency used in the equation is:

$$\eta_{\text{glide}} = 1 - \frac{T_{ml4.1}}{T_{ml3.2}}$$  \hspace{1cm} (4.14)

For each combination of seawater $\Delta T$s, the optimal $\Delta T_{\text{glide}}$ for maximum net specific work obtainable is found. A pinch point of 2°C was selected and a constant pressure drop of 0.3bar has been imposed as for the previous analyses. Results are shown in Figure 4.16. According to the procedure to evaluate net specific work produced by ideal OTEC cycles of equations from (4.8) to (4.11), it was observed that the optimal $\Delta T_{\text{glide}}$ for each combination of warm and cold seawater $\Delta T$s was equal to $\Delta T_{c,\text{sw}}$. The maximum net specific work is reached when $\Delta T_{c,\text{sw}} = \Delta T_{w,\text{sw}}$ and its value is higher than that of the ideal CC-OTEC introduced in section 4.1. When $\Delta T_{c,\text{sw}} = \Delta T_{w,\text{sw}}$, the $\Delta T_{\text{glide}}$ that maximizes net specific work is equal to the seawater $\Delta T$s, in order to maximize the cycle area of the parallelogram cycle on the T-s diagram. In this way, in each point of the heat exchangers there is a temperature difference between seawater and working fluid equal to the pinch point assumed.

However, this condition is optimal from the point of view of net specific work, but the worst from the point of view of heat transfer area required; which can be calculated with the following equation:

$$A_{hx} = \frac{\dot{Q}_{hx}}{U \Delta T_{ml,\text{hx}}}$$  \hspace{1cm} (4.15)

Where $\dot{Q}_{hx}$ is thermal power exchanged, $U$ is overall heat transfer coefficient and $\Delta T_{ml,\text{hx}}$ the mean log temperature difference between seawater and working fluid involved in the heat transfer process. By considering a constant overall heat transfer coefficient, the log mean temperatures will be at their minimum and the areas required are the largest. However,
glide mixtures can have some advantages in off-design conditions of heat exchangers because of biofouling.

Figure 4.16: Maximum net specific work of an ideal CC-OTEC with generic mixtures with $\Delta T_{\text{glide}}$ that maximizes net specific work in each point. A pressure drop of 0.3 bar was assumed for both warm and cold seawater and a pinch point of 2°C was assumed for both warm and cold heat exchanger.
5. Techno-economic analysis of a single level CC-OTEC plant

In this Chapter, the numerical model of a single level closed cycle OTEC plant is presented. Assuming several input variables, the model is able to calculate gross power produced by the Rankine cycle, seawater pumps consumptions and heat transfer areas required. Then, net power produced by the plant and the $\gamma$ parameter can be obtained. An optimization tool is developed to perform a maximization of the $\gamma$ parameter by varying the seawater $\Delta T$s and condenser and evaporator pinch point temperature differences. A list of pure fluids proposed in literature for CC-OTEC plants was used as working fluid in the Rankine cycle and optimization results have been compared. The working fluid with the higher value of $\gamma$ parameter was selected. A detailed optimization of the power plant with the elected working fluid have been conducted and a sensitivity analysis of significant parameters assumed in the model have been performed.

5.1 Single level closed cycle OTEC plant model

The numerical model of single level CC-OTEC plant is described in Figure 5.1. Input data required by the model are listed the same figure. As stated in section 2.1, constraints on maximum allowable diameters and pipe lengths are imposed by mechanical resistance of the pipe. The model considers cold seawater flowrate as a fixed input parameter because its value is related to the cold water pipe effectively adopted in the plant. With these input data, thermodynamic points of the Rankine cycle are determined and warm seawater and working fluid flowrate are derived from energy balances at the heat exchangers. Then, electric power produced by the Rankine cycle and heat exchangers heat transfer area is obtained. The effect of seawater pressure for the calculation of energy balances at condenser and evaporator was neglected because the pressure drop across heat exchangers has little effect on enthalpy of seawater. A similar assumption was used by Yang et al. [22] and Soto et al. [9]. Seawater pressure drop in pipes and heat exchangers is evaluated and seawater pumps electric power consumption can be obtained. Consequently, net electric power produced can be calculated
and the value of $\gamma$ parameter can be determined. Constant turbomachinery efficiencies were assumed according to [9, 22, 26, 50].

5.1.1 Rankine cycle model

A numerical model of a single level Rankine cycle for OTEC have been developed. The model has been implemented with MATLAB and thermodynamic properties of working fluid have been evaluated with REFPROP [45]. The model was developed for pure fluids. Fluids suitable for OTEC can have positive or negative slope of saturated vapor curve in

![Diagram of numerical model of a CC-OTEC plant](image)
their T-s diagram. If a fluid has negative slope of saturated vapor curve, like ammonia or water, the expansion in the turbine will be wet and vapor quality at the end of expansion will be lower than unity. The model scheme adopted in this case is reported in Figure 5.2. Since cold seawater inlet temperature, its ΔT across condenser and relative pinch point are assumed to be known, condensation temperature can be obtained:

\[ T_{\text{cond}} = T_{\text{in,sw}} + \Delta T_{\text{c,sw}} + \Delta T_{\text{pp,cond}} \] (5.1)

No subcooling was assumed since the preheating and evaporation generally occurs in the same exchanger. In Figure 5.2 the preheating section was drawn separately from the evaporation section in order to highlight the position of the pinch point in the evaporator which corresponds to:

\[ \Delta T_{\text{pp, evap}} = T_b - T_3 = T_b - T_{\text{evap}} \] (5.2)

Since the pinch point at the evaporator and warm seawater ΔT is known while \( T_b \) and \( T_{\text{evap}} \) are unknown, a terminal temperature difference is assumed and evaporation temperature is calculated:

\[ T_{\text{evap}} = T_{\text{in,sw}} - \Delta T_{\text{w,sw}} - TTD \] (5.3)

Once evaporation and condensation temperatures are defined, since turbine and working fluid isentropic efficiency are input parameters, thermodynamic properties of working fluid points 1 to 5 of the Rankine cycle of Figure 5.2 are determined. Working fluid flowrate comes from an energy balance at the condenser:

\[ \dot{m}_{\text{wf}} = \frac{\dot{Q}_{\text{cond}}}{h_5 - h_1} = \frac{\dot{m}_{\text{c,sw}} c_{p,sw} \Delta T_{\text{c,sw}}}{h_5 - h_1} \] (5.4)

Warm seawater flowrate instead, is obtained by an energy balance at the complete evaporator:

\[ \dot{m}_{\text{w,sw}} = \frac{\dot{m}_{\text{wf}} (h_4 - h_2)}{c_{p,sw} \Delta T_{\text{w,sw}}} \] (5.5)

The pinch point at the evaporator can finally be calculated:

\[ \Delta T_{\text{pp, evap, calc}} = T_{\text{in,sw}} - \frac{\dot{m}_{\text{wf}} (h_4 - h_3)}{\dot{m}_{\text{w,sw}} c_{p,sw}} - T_{\text{evap}} \] (5.6)

If \( \Delta T_{\text{pp, evap, calc}} \) is different from the pinch point assumed at the evaporator, TTD is varied until the difference between assumed and calculated pinch point is zero.

Gross power obtained by the Rankine cycle finally is:
\[
W_{gross} = W_{el,turbine} - W_{el,pump} = \dot{m}_{wf}(h_4 - h_5)\eta_{m,t}\eta_{el,t} - \frac{\dot{m}_{wf}(h_2 - h_1)}{\eta_{m,p}\eta_{el,p}} \tag{5.7}
\]

Where \(\eta_{m,t}\) and \(\eta_{el,t}\) are mechanical and electric efficiencies of the turbogenerator, respectively and \(\eta_{m,p}\) and \(\eta_{el,p}\) are mechanical and electric efficiencies of the working fluid pump respectively.

Figure 5.2: Plant scheme adopted to describe Rankine cycle numerical model

In case of a fluid with dry expansion, the expansion ends in the superheated vapor region and the effect of desuperheating is taken into account. The TQ diagram at the condenser in this case is reported in Figure 5.3. For these fluids, the condensation temperature is defined in a similar way as for the evaporator. A terminal temperature difference for condenser is assumed and condensation temperature is calculated:

\[
T_{\text{cond}} = T_{\text{in,c,sw}} + \Delta T_{c,sw} + TTD_{\text{cond}} \tag{5.8}
\]

Evaporation temperature is determined according to equation (5.3). Once evaporation and condensation temperature are known, thermodynamic properties of points of the saturated Rankine cycle can be determined. Working fluid mass flowrate can be determined with an energy balance at the condenser. With an energy balance at the condensation only section, seawater temperature at outlet of condensation section \(T_d\) is calculated. Since the pinch point
is positioned at the end of desuperheating section in correspondence of $T_d$, the value of the pinch point with the assumed $TTD_{cond}$ can be determined:

$$\Delta T_{pp,cond,calc} = T_{cond} - T_d$$  

(5.9)

A numerical method is used in order to have calculated pinch point equal to the desired imposed pinch points for both the condenser and evaporator.

![TQ diagram at the condenser in case of a working fluid with dry expansion](image)

**Figure 5.3**: TQ diagram at the condenser in case of a working fluid with dry expansion

### 5.1.2 Heat transfer area evaluation

The Rankine cycle model gives as output thermal power exchanged and all temperatures of streams involved in heat transfer processes. The heat transfer area of a generic heat exchanger is:

$$A_{hx} = \frac{\dot{Q}_{hx}}{U \Delta T_{ml,hx}}$$  

(5.10)

Where $\dot{Q}_{hx}$ is thermal power, $\Delta T_{ml,hx}$ is the log mean temperature difference and $U$ is the overall heat transfer coefficient. The following heat transfer areas has been taken into account separately applying equation (5.10) for each section each one with the relative $\Delta T_{ml,hx}$:

- Heat transfer area required by the working fluid evaporation section;
- Heat transfer area required by the working fluid preheating section;
- Heat transfer area required by the working fluid condensation section;
- In case of a fluid with dry expansion, the heat transfer area required by the desuperheating section is accounted.

The heat transfer process in an OTEC evaporator with ammonia as the working fluid is described in Figure 5.4. For the condenser a similar scheme can be adopted but heat flux in this case goes from ammonia to seawater. The overall heat transfer coefficient for both evaporator and condenser have the following expression [5]:

$$U = \left( \frac{1}{h_{sw}} + R''_{\text{biofouling}} + R''_{\text{wall}} + R''_{\text{corr}} + \frac{1}{h_{wf}} \right)^{-1}$$  \hspace{1cm} (5.11)

Where $h_{sw}$ and $h_{wf}$ are seawater and working fluid convective heat transfer coefficients, respectively, $R''_{\text{biofouling}}$ is thermal resistance per surface area due to biofouling, $R''_{\text{wall}}$ is metal wall conductive resistance per unit surface area and $R''_{\text{corr}}$ is thermal resistance per unit surface area due to corrosion film, generally neglected.

![Figure 5.4: Heat transfer process scheme [5]](image)

In Table 5.1, values of heat transfer coefficients and thermal resistances for typical state of the art titanium evaporator with ammonia as working fluid are reported [5]. For an evaporator with no biofouling, the overall heat transfer coefficient calculated from data in
Table 5.1 is 3198W/m²K.

In the analyses conducted in the following sections of this chapter, the overall heat transfer coefficient was imposed and maintained constant for both the evaporator and condenser. The evaporator and condenser convective heat transfer coefficients were used also for the preheating and desuperheating sections, for sake of simplicity. This assumption is not a rigorous, however since thermal power exchanged in the single phase sections are the 5% of the inlet and outlet overall thermal power, their contribution to the overall heat transfer area is minimal. These assumptions were necessary because the physical process of heat transfer with phase change is difficult to model and detailed model of an evaporator and a condenser for OTEC goes beyond the scope of this thesis. Moreover, since an optimization tool was used with the CC-OTEC model, if such a detailed model of heat exchangers was implemented, computational time required for the analyses conducted would have been unbearable.

Table 5.1: Baseline heat transfer coefficients for ammonia OTEC evaporator [5]

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<tbody>
<tr>
<td>( h_{sw} )</td>
<td>5556 W/m²K</td>
</tr>
<tr>
<td>( R”_{biofouling} )</td>
<td>0-8.8x10⁻⁵ m²K/W</td>
</tr>
<tr>
<td>( R”_{wall} )</td>
<td>0.0000447 m² K/W</td>
</tr>
<tr>
<td>( h_{amm} )</td>
<td>11364 W/m²K</td>
</tr>
</tbody>
</table>

From conceptual designs made available in literature [5] it was observed that the overall heat transfer coefficient for condenser was 85-98% of the evaporator heat transfer coefficient. If it is assumed that the ammonia convective heat transfer coefficient for condensation is 80% of that of the evaporator, if the same seawater convective heat transfer coefficient and wall conductive resistance of Table 5.1 are maintained and no biofouling resistance is introduced, the overall heat transfer coefficient for a condenser is 2987W/m²K, the 93% of that of the evaporator.

5.1.3 Seawater pressure drop evaluation

Seawater pressure drop assessment is crucial to determine net power leaving the plant. According to Ikegami [14], seawater pumps consumption is the 20-50% of gross power produced by Rankine cycle. Overall pressure drop is composed of two contributions:

- Pressure drop in seawater pipes;
Chapter 5

- Pressure drop in the heat exchangers.

Localized pressure drops contributions, such as pressure drop due to valves and heat exchanger manifolds, were neglected. The same assumption was used by Yang et al. [22]. Pressure drop for cold seawater pipe is the sum of frictional pressure drop and pressure loss caused by the difference in density between the water in the CWP and the surrounding warm seawater [5, 35]. The frictional pressure drop is computed with an empirical formula proposed by Uehara et al. [35] in terms of pressure head:

$$\Delta H_{CWP} = 6.82 \frac{L_{CWP}}{D_{CWP}^{1.17}} \left( \frac{v_{CWP}}{100} \right)^{1.85}$$  \hspace{1cm} (5.12)

Since CWP geometry and cold seawater flowrate is fixed, velocity and frictional pressure head remains constant in the optimization. Pressure head due to density difference is determined according to the following equation [35]:

$$\Delta H_{p,c} = L_{CWP} - \frac{1}{\rho_{c,sw}} \left( \frac{1}{2} \left( \rho_{c,sw} + \rho_{w,sw} \right) L_{CWP} \right)$$  \hspace{1cm} (5.13)

The cold water pipe length is assumed equal to the depth at which cold seawater is drawn. The only pressure drop considered for warm water pipe is frictional pressure drop only:

$$\Delta H_{WWP} = 6.82 \frac{L_{WWP}}{D_{WWP}^{1.17}} \left( \frac{v_{WWP}}{100} \right)^{1.85}$$  \hspace{1cm} (5.14)

Since warm seawater flowrate is not imposed but comes from the energy balance at the evaporator, a seawater velocity in warm pipe was assumed. Pipe diameter results from the following equation:

$$D_{WWP} = \sqrt[4]{\frac{4 \dot{m}_{w,sw}}{\rho_{w,sw} v_{WWP} \pi}}$$  \hspace{1cm} (5.15)

If diameter of warm water pipe exceeds the maximum allowable diameter, the diameter is imposed to this limit and velocity was calculated as a consequence:

$$v_{WWP} = \frac{\dot{m}_{w,sw}}{\rho_{w,sw} \pi \frac{D_{lim}^{2}}{4}}$$  \hspace{1cm} (5.16)

According to section 5.1.2, constant overall heat transfer coefficient was assumed for heat exchangers.

The overall heat transfer coefficient by neglecting the effect of biofouling and wall resistance has the following form:
Techno-economic analysis of a single level CC-OTEC plant

\[ U = \left( \frac{1}{h_{wf}} + \frac{1}{h_{sw}} \right)^{-1} \]  

(5.17)

Since working fluid is subjected to phase change, as observed in Table 5.1, its heat transfer coefficient is generally higher than the seawater heat transfer coefficient. Thus, \( h_{wf} \) is maintained constant, assuming that the heat transfer coefficient of seawater is the only limiting factor to heat transfer. The same assumption has been used by Upshaw [3]. Since a constant overall heat transfer coefficient is assumed, as a consequence, even the seawater heat transfer coefficient is maintained constant. Single phase heat transfer of a liquid flowing in channels is proportional to Nusselt number, which is proportional to Reynolds number and Prandtl number elevated at a definite power according to the following relationships:

\[ h_{sw} \propto Nu_{sw} \propto Re_{sw}^{\alpha} Pr_{sw}^{\beta} \]  

(5.18)

If the variation of seawater thermophysical properties is negligible, the heat transfer coefficient is proportional to the product of velocity in channels and channels hydraulic diameter at a certain power:

\[ h_{sw} \propto (v_{sw} D_{h_{hx}})^{\gamma} \]  

(5.19)

If hydraulic diameter of channels is maintained constant, since seawater heat transfer coefficient remains constant, even velocity through channels remain constant.

Heat exchangers pressure drop can also be expressed in the following form:

\[ \Delta p_{hx} = f \rho \frac{v_{sw}^2}{2} \left( \frac{L}{D_{h_{hx}}} \right) \]  

(5.20)

According to assumptions introduced so far and affirming that friction factor remains constant, it results that the only parameter whom pressure drop depends on is channel length. Thermal power in a heat exchanger can be calculated with the following equation:

\[ \dot{Q}_{hx} = m_{sw} c_p \Delta T_{sw} \]  

(5.21)

The ideal pumping power required to let seawater flow in the heat exchanger is:

\[ \dot{W}_{id,hx} = \frac{m_{sw}}{\rho} \Delta p_{hx} \]  

(5.22)

The ideal pumping power required for heat exchangers is assumed to be proportional to thermal power:

\[ \dot{W}_{id,hx} = k_p \dot{Q}_{hx} \]  

(5.23)
Where the proportionality constant \( k_p \) derives from a reference heat exchanger performance:

\[
k_p = \left( \frac{\dot{W}_{id,\text{hx}}}{\dot{Q}_{\text{hx}}}_{\text{ref}} \right) = \left( \frac{\dot{m}_{\text{sw}} \Delta p_{\text{hx}}}{\rho \dot{m}_{\text{sw}} c_p \Delta T_{\text{sw}}}_{\text{ref}} \right) = \left( \frac{\Delta p_{\text{hx}}}{\rho c_p \Delta T_{\text{sw}}}_{\text{ref}} \right)
\] (5.24)

By substituting equation (5.24) in equation (5.23), if seawater density and seawater specific heat are maintained constant:

\[
\dot{W}_{id,\text{hx}} = \left( \frac{\Delta p_{\text{hx}}}{\rho c_p \Delta T_{\text{sw}}}_{\text{ref}} \right) \dot{Q}_{\text{hx}} = \left( \frac{\Delta p_{\text{hx}}}{\rho c_p \Delta T_{\text{sw}}}_{\text{ref}} \right) \dot{m}_{\text{sw}} c_p \Delta T_{\text{sw}}
\] (5.25)

By rearranging the terms of the latter:

\[
\dot{W}_{id,\text{hx}} = \frac{\dot{m}_{\text{sw}}}{\rho} \left( \frac{\Delta p_{\text{hx}}}{\Delta T_{\text{sw}}}_{\text{ref}} \right) \Delta T_{\text{sw}}
\] (5.26)

By comparing equation (5.26) with equation (5.22) it is found that:

\[
\Delta p_{\text{hx}} = \left( \frac{\Delta p_{\text{hx}}}{\Delta T_{\text{sw}}}_{\text{ref}} \right) \Delta T_{\text{sw}}
\] (5.27)

According to equation (5.26), if seawater mass flowrate varies, the pumping power varies proportionally and the same happens if \( \Delta T_{\text{sw}} \) varies.

Therefore, by comparing equation (5.27) with equation (5.20), if seawater \( \Delta T \) increases, channel length increases proportionally and so do the pressure drop and the final ideal pumping power required by each heat exchanger is obtained with equation (5.23).

Ideal seawater pumping power required can then be calculated:

\[
\dot{W}_{id,\text{sw,pump},c} = \dot{m}_{\text{c,sw}} g \left( \Delta H_{\text{CWP}} + \Delta H_{\rho,c} \right) + \dot{W}_{id,\text{hx},c}
\] (5.28)

\[
\dot{W}_{id,\text{sw,pump},w} = \dot{m}_{\text{w,sw}} g \Delta H_{\text{CWP}} + \dot{W}_{id,\text{hx},w}
\] (5.29)

Electric seawater pumps consumption is finally computed:

\[
\dot{W}_{el,\text{sw,pump}} = \frac{\dot{W}_{id,\text{sw,pump}}}{\eta_{\text{hydr}} \eta_{\text{mech}} \eta_{\text{el}}}
\] (5.30)

Where \( \eta_{\text{hydr}} \) is hydraulic efficiency, \( \eta_{\text{mech}} \) and \( \eta_{\text{el}} \) are mechanical and electric efficiency, respectively.

### 5.1.4 Net power produced by the plant and the \( \gamma \) parameter evaluation

The net power produced by the plant is finally computed:

\[
\dot{W}_{el,\text{net}} = \dot{W}_{el,\text{turbine}} - \dot{W}_{el,\text{w,pump}} - \dot{W}_{el,\text{w,sw,pump}} - \dot{W}_{el,c,\text{sw,pump}}
\] (5.31)
The total heat transfer area is the sum of required area for evaporator and condenser including the preheating and desuperheating sections (the latter one only if working fluid has a dry expansion):

\[ A_{tot,hx} = A_{evap} + A_{cond} + A_{preH} + A_{desh} \] (5.32)

The \( \gamma \) parameter can finally be determined:

\[ \gamma = \frac{W_{el,net}}{A_{tot,hx}} \] (5.33)

### 5.1.5 The optimization tool

In order to perform techno-economic optimization of the single level CC-OTEC plant, an optimization tool was developed with MATLAB in order to maximize the \( \gamma \) parameter, which was selected as the objective function of the techno-economic analysis (see Chapter 3 for details). The optimization tool used the MATLAB function `fmincon` to perform optimization of the following variables required as input by the CC-OTEC numerical model described in the previous section:

- \( \Delta T_{c,sw} \): seawater temperature difference across the condenser (with included eventual desuperheating section);
- \( \Delta T_{w,sw} \): seawater temperature difference across the evaporator (with included preheating section);
- \( \Delta T_{pp,evap} \): pinch point temperature difference at the evaporator;
- \( \Delta T_{pp,cond} \): pinch point temperature difference at the condenser

If a maximization of the \( \gamma \) parameter is performed by using the single stage CC-OTEC model with embedded the optimization tool, the introduction of these four variables in the input parameters of the single level closed cycle model is not required because the optimization tool search for the optimum value of the four variables considered in the optimization process.
5.1.6 Parameters assumed in the analyses

A list of the constant parameters assumed in the model is presented in the following table:

<table>
<thead>
<tr>
<th>Inlet conditions</th>
<th>Ref</th>
</tr>
</thead>
<tbody>
<tr>
<td>Warm seawater inlet temperature</td>
<td>A</td>
</tr>
<tr>
<td>Cold seawater inlet temperature</td>
<td>A</td>
</tr>
<tr>
<td>Seawater salinity</td>
<td>A</td>
</tr>
<tr>
<td>Cold seawater mass flowrate</td>
<td>A</td>
</tr>
<tr>
<td>Working fluid turbomachinery efficiencies</td>
<td></td>
</tr>
<tr>
<td>( \eta_{\text{is,turbine}} )</td>
<td>89</td>
</tr>
<tr>
<td>( \eta_{\text{mech,turbine}} )</td>
<td>97</td>
</tr>
<tr>
<td>( \eta_{\text{el,turbine}} )</td>
<td>99.5</td>
</tr>
<tr>
<td>( \eta_{\text{is,pump}} )</td>
<td>80</td>
</tr>
<tr>
<td>( \eta_{\text{mech,pump}} )</td>
<td>96</td>
</tr>
<tr>
<td>( \eta_{\text{el,pump}} )</td>
<td>98</td>
</tr>
<tr>
<td>Seawater pumps</td>
<td></td>
</tr>
<tr>
<td>( \eta_{\text{hydr,pump}} )</td>
<td>85</td>
</tr>
<tr>
<td>( \eta_{\text{mech,pump}} )</td>
<td>97</td>
</tr>
<tr>
<td>( \eta_{\text{el,pump}} )</td>
<td>97</td>
</tr>
<tr>
<td>( k_{p_{\text{sw,sw,pump}}} )</td>
<td>0.001667</td>
</tr>
<tr>
<td>( k_{P_{\text{sw,sw,pump}}} )</td>
<td>0.001601</td>
</tr>
<tr>
<td>Hydraulic data</td>
<td></td>
</tr>
<tr>
<td>Cold water pipe length</td>
<td>C</td>
</tr>
<tr>
<td>Warm water pipe length</td>
<td>D</td>
</tr>
<tr>
<td>Limit diameter</td>
<td>C</td>
</tr>
</tbody>
</table>

| References: A=Bombarda et al. [26], B= Avery et al. [5], C=Lockheed Martin Mini Spar [27] |
| D=assumption of the author |

The cold seawater flowrate assumed was similar to value assumed by Bombarda [26]. The limit diameter of 2.5m, was imposed according to the maximum acceptable diameter for a HDPE pipe considered by Lockheed Martin Mini Spar OTEC plant [27]. HDPE was selected since it was successfully adopted for cold water pipe in the Mini OTEC, OTEC-1 and Nauru plants [5]. Since no data are available in literature, a 200m length pipe was selected for warm water pipe. The proportionality constants adopted to compute ideal pumping power required for evaporator and condenser have been obtained by the Lockheed Martin plate exchangers whose data have been reported in Table 5.3. The exchangers were selected in the preliminary
Techno-economic analysis of a single level CC-OTEC plant

design of the 40MWe baseline plant for DOE’s PON (see section 1.2).

Table 5.3: Lockheed Martin plate exchanger data for the 40Mwe OTEC preliminary design baseline for DOE’s PON (Adapted from [5])

<table>
<thead>
<tr>
<th></th>
<th>Evaporator</th>
<th>Condenser</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water flow [kg/s]</td>
<td>105600</td>
<td>102900</td>
</tr>
<tr>
<td>NH₃ flow [kg/s]</td>
<td>1722</td>
<td>1722</td>
</tr>
<tr>
<td>Water inlet/outlet temps [°C]</td>
<td>27.78/23.81</td>
<td>4.44/8.36</td>
</tr>
<tr>
<td>U [W/(m²K)]</td>
<td>3259</td>
<td>3118</td>
</tr>
<tr>
<td>Head loss [m]</td>
<td>2.7</td>
<td>2.56</td>
</tr>
</tbody>
</table>

5.2 Working fluid selection

The following pure fluids have been examined: ammonia, R134a, R152a, R22, methanol, propane, propylene, R32, R227ea, butane, R600a, R245fa, R1234ze, R1234yf. For each fluid, the maximum γ was researched by finding the optimal combination of seawater ΔTs and pinch points at condenser and evaporator. Data in Table 5.2 where used as constant parameters in the model.

The following equation of heat transfer coefficient was used:

$$ U = \left( \frac{1}{h_{sw}} + R^w_{wall} + \frac{1}{h_{wf}} \right)^{-1} = \left( \frac{1}{h_{wf}} + \frac{1}{h_{sw}^*} \right)^{-1} \quad (5.34) $$

Where $h_{wf}$ is working fluid heat transfer coefficient and $h_{sw}^*$ includes the effect of seawater heat transfer coefficient and wall conductive resistance. The value of $h_{sw}^*$ was maintained constant at 4450W/m²K by using seawater heat transfer coefficient and wall resistance in Table 5.1, while working fluid heat transfer coefficient was obtained as:

$$ h_{wf} = h_{ref} \beta = h_{ref} \frac{h_{wfi}}{h_{ref,2phase}} \quad (5.35) $$

Where $h_{ref}$ was heat transfer coefficient for a reference fluid while β is the ratio between $h_{wfi}$, the heat transfer coefficient of a generic working fluid during phase change and $h_{ref,2phase}$, the reference fluid heat transfer coefficient obtained in the same conditions as $h_{wfi}$.

Ammonia was considered the reference fluid and overall heat transfer coefficient for
evaporator was assumed to be 3198W/m² K, obtained with heat transfer coefficients provided as a baseline in Table 5.1. Reference ammonia convective heat transfer coefficient for evaporation was 11364W/m²K, while ammonia convective heat transfer coefficient for condenser was assumed to be 80% of the ammonia heat transfer coefficient during evaporation according to considerations in section 5.1.2. With the reduction of ammonia convective heat transfer coefficient during condensation, which becomes 9091W/m²K, the overall heat transfer coefficient for condenser in the reference case is 2987W/m² K.

Boiling and condensation correlations for plate exchangers were selected to determine $\beta$, because according to section 2.2.5 plate exchangers have several advantages for OTEC. Several correlations for boiling in plate exchangers have been proposed in literature. In a recent review of available correlations accomplished by Eldeeb et al. [51], it emerged that correlations proposed are strictly dependent on fluid adopted, geometry, experimental conditions in which they were determined and different correlations lead to different results if employed in the same heat transfer problem. Eldeeb et al. also reported that under certain experimental conditions, nucleate boiling occurs and the Cooper correlation [52, 53] predicted well heat transfer performance. Since Cooper correlation is a simple correlation and depends on few parameters, it was found suitable for the calculation of the $\beta$ coefficient. The same correlation was used by Yang et al. [22] for predicting heat transfer coefficient. The Cooper correlation is the following:

$$h_{nucleate\ boiling} = \{55 \Delta T_e^{0.67} p_{red}^{0.12} (-\log_{10} p_{red})^{-0.55} M^{0.5}\}^{3.03}$$

(5.36)

Where $p_{red}$ is the reduced pressure, $M$ is molecular weight and $\Delta T_e$ is wall superheating. If the same wall superheating was assumed, $\beta$ becomes:

$$\beta_{boiling} = \left\{\frac{p_{red, wfi}^{0.12} (-\log_{10} p_{red, wfi})^{-0.55} M_{wfi}^{0.5}}{p_{red, amm}^{0.12} (-\log_{10} p_{red, amm})^{-0.55} M_{amm}^{0.5}}\right\}^{3.03}$$

(5.37)

According to [51] there are few correlations for condensation in plate exchangers and most of them are specific to a definite fluid. The Shah correlation, which was developed for condensation in horizontal vertical and inclined pipes, is widely accepted for condensation in plate heat exchangers [51]. The correlation for mean vapor quality is reported by Kakaç et al. [39]:

$$h_{cond} = h_t \left(0.55 + \frac{2.09}{P_{red}}\right) = 0.023 Re^{0.8} Pr^{0.4} k_{liq} \left(0.55 + \frac{2.09}{P_{red}}\right)$$

(5.38)
Where $Re$ is the Reynolds number, $Pr$ is the Prandtl number $k_{liq}$ is the liquid conductivity and $p_{red}$ is reduced pressure.

The $\beta$ coefficient for condensation can be determined assuming the same velocity and the same hydraulic diameters for both the generic and the reference fluids:

$$\beta_{\text{cond}} = \left( \frac{\rho_{liq,wfi}}{\rho_{liq,wfi}} \right)^{0.8} \frac{Pr^{0.4} k_{liq,wfi}}{Pr_{0.4} k_{liq,amm}} \left( 0.55 + \frac{2.09}{Pr_{0.38}} \right)$$

(5.39)

Thermodynamic properties for $\beta$ have been calculated at saturation temperature provided by the Rankine cycle model.

The maximum $\gamma$ was researched for each one of the selected working fluids with the optimization tools and assuming input parameters of Table 5.2. The Rankine cycle was operated with each one of the working fluids in order to obtain net electric power produced. The overall heat transfer coefficients were corrected according to equation (5.35) for both evaporator and condenser and the total heat transfer area was obtained. The $\gamma$ parameter was obtained and its value was maximized by varying seawater $\Delta Ts$ and pinch points.

Optimization results are shown in Figure 5.5 and Figure 5.6. It appears that optimal net powers for all working fluids are included in a range between 1900 and 2000kW. The highest net power is reported for ammonia and R22 follows. The heat transfer areas instead, are very different and since net powers are similar, the higher the heat transfer area, the lower the $\gamma$ parameter. The little gap observed in net power between ammonia and R22 becomes larger when their respective heat transfer areas are compared.

According with the assumptions introduced and from the analysis conducted it can be concluded that ammonia is effectively the working fluid with desirable characteristics for OTEC as concluded by [7, 21, 22, 26].

Turbine isentropic efficiency was maintained constant for all investigated fluids. Anyway, the effect of fluids on turbine efficiency was observed: a correlation developed by Astolfi [54] for ORC axial turbines isentropic efficiency, which is function of the size parameter (SP) and volume ratio ($V_r$) only, was used with the size parameters and volume ratios obtained in the Rankine cycle model:

$$\eta_{is,turbine} = \sum_{i=0}^{15} A_i(V_r,SP) F_i(V_r,SP)$$

(5.40)

Where coefficients $A_i$ and $F_i$ depends on number of turbine stages selected. As observed in
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Figure 5.7, ammonia still remained the working fluid with the highest value of $\gamma$ parameter. The higher value of the $\gamma$ parameter depends on turbine efficiency value calculated with equation (5.40), which was higher than the value assumed in Table 5.2. As an example for a three stage turbine working with ammonia, turbine efficiency with equation (5.40) is 93\% while the assumed value was 89\%. However, since correlation (5.40) was developed for turbines with dry expansion only its adoption in the model would be an improper use for large parts of the fluid investigated which, as ammonia, do not have a dry expansion. Therefore, since it was observed that turbine efficiency would be similar even with different working fluids, the assumption of constant turbine isentropic efficiency was found reasonable.

The same analysis was also conducted by substituting the Cooper correlation for $\beta_{\text{boiling}}$ calculation with the Almalfi correlation. Although according to Eldeeb et al. [51] the Almalfi correlation shows a good potential in estimating the heat transfer coefficient for different refrigerants, this correlation introduced more parameters such has chevron angle, mass flux and heat flux. These parameters were assumed to be the same between the i-th working fluid and reference working fluid in order to perform $\beta_{\text{boiling}}$ calculation. Results, even with this variation introduced, showed a better performance of the plant with ammonia as working fluid. The Shah correlation for $\beta_{\text{cond}}$ was also substituted with the Nusselt correlation for condensation outside tubes considering the eventuality to adopt a shell and tube heat exchanger. The Cooper correlation was maintained for $\beta_{\text{boiling}}$ assuming nucleate boiling outside tubes as assumed by Yang et al. [22]. Even in this case, ammonia resulted the working fluid with highest value of $\gamma$ parameter.

![Graph](image_url)

**Figure 5.5:** Net power and heat transfer area for optimum $\gamma$ with different working fluids
Techno-economic analysis of a single level CC-OTEC plant

5.3 Optimum configuration results for a single level CC-OTEC plant with ammonia as working fluid

In this section, optimization results obtained with the optimization tool with ammonia as working fluid are presented. The effect of the pinch points on the $\gamma$ parameter and the effect of the seawater $\Delta T$ across heat exchangers are observed. Finally, the effect of seawater inlet temperatures, turbine efficiency, overall heat transfer coefficient and proportionality
constant for pressure drop on seawater side of heat exchangers (see section 5.1.3 for details) are assessed.

5.3.1 Optimization results
The effect of seawater $\Delta T$s and pinch points on the $\gamma$ parameter was observed for a single level ammonia cycle. Assumptions for cycle parameters such as turbomachinery efficiencies, inlet temperatures etc., are the ones highlighted in Table 5.2. These parameters remained constant during the optimization process. Overall heat transfer coefficient for evaporator was $3198\text{W/(m}^2\text{K)}$ and condenser overall heat transfer coefficient was $2987\text{W/m}^2\text{K}$ according to section 5.1.2.

The effect of pinch point on $\gamma$ was observed for an OTEC plant operating with $\Delta T_{w,sw} = \Delta T_{c,sw}=5^\circ\text{C}$. Results are plotted in Figure 5.8. From the figure, it can be observed that an optimum exists. A lower value of pinch points favors electric power produced since difference between evaporation and condensation in the Rankine cycle becomes higher, but impacts negatively on heat transfer area since temperature difference in heat exchangers becomes lower and, assuming constant overall heat transfer coefficients, larger heat transfer areas are required.

The trend of the objective function $\gamma$ with seawater temperature differences was observed by imposing a combination of $\Delta T_{c,sw}$ and $\Delta T_{w,sw}$ and searching for pinch point temperature differences at the evaporator and condenser that maximized $\gamma$ parameter. The surface in

![Figure 5.8: The effect of pinch point on $\gamma$ with imposed seawater temperature differences of 5°C at both the evaporator and condenser.](image)
Figure 5.9, shows the trend of the maximum $\gamma$ parameter obtainable for an imposed combination of warm and cold seawater temperature difference. A similar surface was obtained by Yang et al. [22] with different assumptions. The trend shows the presence of an optimum combination of seawater $\Delta T$s. The maximum $\gamma$ with the assumptions introduced is 0.1908 kW/m$^2$ and the optimal warm and cold seawater temperatures are 1.64°C and 2.20°C, respectively. The optimum pinch point at the evaporator is 3.89°C and the optimum pinch point at the condenser is 3.67°C. Model output in optimal conditions are resumed in Table 5.4. As can be observed by the table net power produced in optimal conditions is nearly 2MWe and plant electric efficiency is 2.59%. The total heat transfer area is 10434m$^2$.

Since a large part of electric power produced by the turbine is consumed within the plant to drive working fluid and seawater pumps, the ratio between net power delivered and gross power produced by the turbine is:
\[
\frac{\dot{W}_{\text{net}}}{\dot{W}_{\text{gross}}} = \frac{\dot{W}_{\text{gross}} - (\dot{W}_{\text{w,f,pump}} + \dot{W}_{\text{w,sw,pump}} + \dot{W}_{\text{c,sw,pumps}})}{\dot{W}_{\text{gross}}} \tag{5.41}
\]

For OTEC plant this value falls between 50% to 80% and in optimal conditions the ratio obtained by the model is 70.9%.

The optimization results can be interpreted in the following way: in order to reduce heat transfer area large mean log temperature differences are required and thus high pinch point temperature differences are desirable. However, since the difference between inlet seawater temperatures is low, high seawater ΔTs combined with high pinch points have a negative effect on net power produced because pressure difference across the turbine is reduced and gross power produced by the Rankine cycle is reduced. Thus, optimal seawater ΔTs are low in order to increase the difference between evaporation and condensation pressure. A secondary effect is that, according to the pressure drop in heat exchangers evaluation presented in section 5.1.3, pumping power required by heat exchangers is also reduced because of low seawater temperature differences across them.

| Table 5.4: Model results for a single level OTEC plant in optimal conditions |
|-----------------------------------------------|---------|---------|-----------------|
| ΔTw,sw,opt                                   1.64  °C | ΔTpp,evap,opt  3.89  °C |
| ΔTc,sw,opt                                   2.2  °C | ΔTpp,cond,opt  3.67  °C |
| Inlet thermal power                          77.295 MW | Warm water pipe  pressure loss  0.0452 bar |
| Outlet thermal power                         74.424 MW | Cold water pipe  pressure loss  0.3875 bar |
| Warm seawater mass flowrate                  11773 kg/s | Evaporator pressure loss  0.1119 bar |
| Cold seawater mass flowrate                  8500 kg/s | Condenser pressure loss  0.1441 bar |
| Working fluid mass flowrate                  62.6 kg/s | Warm seawater pump consumption 226.3 kW |
| Evaporation temperature                      22.55 °C | Cold seawater pump consumption 549.8 kW |
| Condensation temperature                     9.86 °C | Net electric power produced 1.9913 MW |
| Turbine electric power                       2.809 MW | η_{\text{plant}}  2.59 % |
| Working fluid pump consumption               42.24 kW | Total heat transfer area required 10434 m² |
| Gross power produced from the cycle          2.767 MW | γ  0.1908 kW/m² |
5.3.2 Effect of inlet temperatures
Since the performance of an OTEC plant is strictly correlated to temperatures at which seawater enters the plant, the effect of inlet seawater temperatures on $\gamma$ was observed. By keeping constant the other parameters, the optimal seawater $\Delta T$s and pinch points have been researched to maximize $\gamma$ parameter for the inlet seawater temperatures considered as input. By observing Figure 5.10, it appears that the higher the difference between seawater inlet temperatures the higher the optimal $\gamma$ parameter. The simulation with inlet temperatures used in the Nauru plant, the Kumejima plant and a location in the Indonesian sea considered favorable for OTEC installation [48] was included in figure. The temperature of a site in the Adriatic Sea off the coast of Puglia was also observed. Warm seawater temperature was 26°C and the cold seawater temperature at 1000m was 13°C [55]. With such a temperature difference the Adriatic Sea is not a favorable location for OTEC installation.

![Figure 5.10: $\gamma$ optimization conducted with different inlet temperatures](image)

5.3.3 Effect of turbine efficiency
The turbine isentropic efficiency used in the previous optimization was 89%. This value was selected according to Bombarda et al. [26]. A similar value, 90.7% have been calculated by Uehara et al. [35] for a single flow axial turbine. Constant isentropic efficiency of 90% was
selected by Soto et al. [9]. The effect of turbine efficiency in the $\gamma$ optimization have been caught by performing the optimization with the same parameters as listed in Table 5.2 but with different values of turbine efficiency. As described by Figure 5.11, when a higher efficiency turbine is selected, the maximum of the surface portrayed in Figure 5.9 moves to lower seawater $\Delta T$s. The evaporation and condensation temperatures remain nearly the same, hence pinch point temperature differences for evaporator and condenser become larger as observed in Figure 5.12. Therefore, mean log temperature differences at heat exchangers is increased.

![Figure 5.11: Effect of turbine efficiency on optimal seawater $\Delta T$s](image)

![Figure 5.12: Effect of turbine efficiency on optimal pinch points](image)

Gross power produced by the Rankine cycle becomes higher and since seawater $\Delta T$s are reduced with increased turbine efficiency, the seawater pumps electric consumption becomes lower according to the pressure drop estimation rule for heat exchangers introduced
in section 5.1.3. Net power produced by the plant becomes higher with higher turbine efficiency. Since optimum cold seawater ΔT is reduced with increased efficiency and cold seawater flowrate remains constant, thermal power leaving the cycle is reduced and consequently, thermal power entering the cycle is also reduced. This fact, coupled with the larger mean log temperature differences at heat exchangers brings to lower heat transfer area required. As confirmed by Figure 5.13 and Figure 5.14 if a turbine with higher efficiency is used, the γ parameter becomes higher because the optimal net power becomes greater and the heat transfer area reduced. An increase of 1% in turbine efficiency brings an increase of 2% on the value of γ.

![Figure 5.13: Optimal net power and heat transfer area according to turbine efficiency](image1)

![Figure 5.14: The effect of turbine efficiency on γ parameter.](image2)
5.3.4 Effect of overall heat transfer coefficient

Since overall heat transfer coefficient is independent of others parameters in the model, its value did not influence the optimal seawater ΔTs or pinch points. The only effect is on the value of the γ parameter, which, as shown in Figure 5.15, varied linearly with increasing overall heat transfer coefficient because area required by heat exchangers, with higher heat transfer coefficients, is smaller. No effects have been observed on the required heat exchangers pumping power because in the present model, a relationship between overall heat transfer coefficient and related seawater pressure drop have not been developed.

![Figure 5.15: Optimal γ with increasing overall heat transfer coefficient](image)

5.3.5 Effect of \( k_p \) for heat exchangers pressure drop estimation

Pressure drop in heat exchangers was estimated considering ideal pumping power required to be proportional to thermal power exchanged:

\[
\dot{W}_{id,hx} = k_p \dot{Q}_{hx}
\]  \hspace{1cm} (5.42)

The value of proportionality constants used so far were obtained by data on a plate evaporator and condenser for an OTEC plant designed by Lockheed Martin and reported by Avery et al. [5]. The values of \( k_p \) for both evaporator and condenser were obtained with the equation:

\[
k_p = \left( \frac{\dot{W}_{id,hx}}{\dot{Q}_{hx}} \right)_{ref}
\]  \hspace{1cm} (5.43)

The values calculated in this way referring to the Lockheed exchangers were
0.001667 W_{el}/W_{th} for the evaporator and 0.001601 W_{el}/W_{th} for condenser. The value of $k_p$ observed for other heat exchangers for OTEC found in literature [5] was between 0.0013 and 0.0017 W_{el}/W_{th}.

The effect of $k_p$ on the maximum $\gamma$ parameter was observed in Figure 5.16. By varying $k_p$ from 0.001 W_{el}/W_{th} to 0.002 W_{el}/W_{th} for both condenser and evaporator, it was observed that $\gamma$ diminished when $k_p$ grew. However, $k_p$ had no effect on the optimal seawater temperatures. By observing Figure 5.17, the pinch point reduced when $k_p$ became larger. This fact can be explained by the fact that since seawater pumps consumption is larger when $k_p$ is larger, power produced by turbine is increased, and to do so, the difference between evaporation and condensation becomes larger.

Figure 5.16: optimal $\gamma$ parameter with variable $k_p$

Figure 5.17: Effect of $k_p$ on the optimal pinch points
6. Techno-economic analysis of multilevel CC-O PAC TEC plants

A multilevel cycle optimization was conducted in order to study the effect of the introduction of more than one evaporation level on the $\gamma$ parameter. The model of a multilevel CC-O PAC TEC plant have been developed and optimization of seawater $\Delta T$s and pinch points to maximize the $\gamma$ parameter was carried out for five plants from one to five evaporation levels. Results have been finally compared with the single level solution.

6.1 Multilevel closed cycle OTEC plant model

6.1.1 Model description

A model for a multilevel OTEC plant have been constructed. The model has been developed in order to have the same input parameters as the single level model with the addition of desired number of evaporation levels as a new parameter. The procedure to perform $\gamma$ parameter calculation is the same of Figure 5.1 with the only difference that the Rankine cycle model is different for the multilevel Rankine cycle.

Flow chart of the algorithm used to perform calculation of the multilevel Rankine cycle is shown in Figure 6.1. The warm and cold seawater overall temperature differences are divided in equal parts according to the number of selected levels:

$$\Delta T_{w,sw,i} = \frac{\Delta T_{w,sw}}{N_{stages}}$$ \hfill (6.1)

$$\Delta T_{c,sw,i} = \frac{\Delta T_{c,sw}}{N_{stages}}$$ \hfill (6.2)

The first level has the following seawater inlet and outlet temperatures:

$$T_{out,w,sw,1} = T_{in,w,sw} - \Delta T_{w,sw}$$ \hfill (6.3)

$$T_{in,w,sw,1} = T_{in,w,sw} - \Delta T_{w,sw} + \Delta T_{w,sw,i}$$ \hfill (6.4)
ASSIGN:
\(T_{\text{in},w,sw}, \Delta T_{w,sw}, T_{\text{in},c,sw}, \Delta T_{c,sw}, m_{c,sw}, \Delta T_{pp}, N\)

\[\Delta T_{c,sw,i} = \frac{\Delta T_{c,sw}}{N} \quad \text{and} \quad \Delta T_{w,sw,i} = \frac{\Delta T_{w,sw}}{N}\]

\[T_{\text{out},w,sw,i} = T_{\text{in},w,sw} - \Delta T_{w,sw}\]
\[T_{\text{in},w,sw,i+1} = T_{\text{out},w,sw,i} + \Delta T_{w,sw,i}\]
\[T_{\text{out},c,sw,i} = T_{\text{in},c,sw} + \Delta T_{c,sw,i}\]
\[T_{\text{out},c,sw,i+1} = T_{\text{in},c,sw,i+1} + \Delta T_{c,sw,i}\]

\(\dot{m}_{c,sw,\text{calc}}\)

Repeat until \(\dot{m}_{c,sw,\text{calc}} = \dot{m}_{c,sw}\)

\(Q_{\text{out}} = 0\)

For \(i = 1 \rightarrow N\)

Solve \(i\)-th Rankine cycle

\(Q_{\text{out}} = Q_{\text{out}} + Q_{\text{out},i}\)

\(T_{\text{out},w,sw,i+1} = T_{\text{in},w,sw,i+1}\)
\(T_{\text{in},w,sw,i+1} = T_{\text{out},w,sw,i+1} + \Delta T_{w,sw,i}\)
\(T_{\text{in},c,sw,i+1} = T_{\text{out},c,sw,i}\)
\(T_{\text{out},c,sw,i+1} = T_{\text{in},c,sw,i+1} + \Delta T_{c,sw,i}\)

\(\dot{m}_{w,sw}\)

\(\dot{m}_{c,sw}\)

STOP

Figure 6.1: Multilevel Rankine cycle model flowchart
Techno-economic analysis of multilevel CC-OTEC plants

\[ T_{\text{in,c,sw},1} = T_{\text{in,c,sw}} \]  (6.5)

\[ T_{\text{out,c,sw},1} = T_{\text{in,c,sw},1} + \Delta T_{\text{c,sw},i} \]  (6.6)

For the next levels, the seawater temperatures assignments are the following:

\[ T_{\text{out,w,sw},i+1} = T_{\text{in,w,sw},i} \]  (6.7)

\[ T_{\text{in,w,sw},i+1} = T_{\text{out,w,sw},i+1} + \Delta T_{\text{w,sw},i} \]  (6.8)

\[ T_{\text{in,c,sw},i+1} = T_{\text{out,c,sw},i} \]  (6.9)

\[ T_{\text{out,c,sw},i+1} = T_{\text{in,c,sw},i+1} + \Delta T_{\text{c,sw},i} \]  (6.10)

With both the warm and cold, inlet and outlet temperatures assigned for each level, a Rankine cycle model slightly different from the one conceived in section 5.1.1 is developed and performed for each level. The warm seawater flowrate is considered as known. Since ammonia is selected as working fluid, as in the model developed in section 5.1.1, a terminal temperature difference at the evaporator is assumed and condensation and evaporation temperatures are calculated according to equations (5.1) and (5.3), respectively. Thermodynamic properties of Rankine cycle points can then be calculated.

Working fluid for the i-th level mass flowrate is equal to:

\[ m_{wf,i} = \frac{\dot{Q}_{\text{in},i}}{(h_4 - h_2)_i} = \frac{m_{w,sw} c_{p,sw} (T_{\text{in,w,sw},i} - T_{\text{out,w,sw},i})}{(h_4 - h_2)_i} \]  (6.11)

Seawater temperature at the evaporator pinch point for the i-th cycle is:

\[ T_{b,i} = T_{\text{in,w,sw},i} - \frac{\dot{Q}_{\text{evap},i}}{m_{w,sw} c_{p,sw}} = T_{\text{in,w,sw},i} - \frac{m_{wf,i} (h_4 - h_2)_i}{m_{w,sw} c_{p,sw}} \]  (6.12)

The value of TTD at the evaporator is changed in order to obtain a pinch point equal to the desired one.

Thermal power leaving the i-th cycle is:

\[ \dot{Q}_{\text{out},i} = m_{wf,i} (h_5 - h_1)_i \]  (6.13)

And the overall thermal power discharged to cold seawater is:

99
An energy balance at the series of condensers permits to determine the cold seawater flowrate:

\[ \dot{Q}_{\text{out}} = \sum_{i=1}^{N} \dot{Q}_{\text{out},i} \]  

(6.14)

An iterative calculation changes the value of warm seawater flowrate assumed in equation (6.11) in order to obtain cold seawater calculated with equation (6.15) equal to the assumed value of cold seawater in the input parameters.

\[ m_{c,\text{sw,calculated}} = \frac{\dot{Q}_{\text{out}}}{c_p \rho_{c,\text{sw}}(T_{\text{out},c,\text{sw}} - T_{\text{in},c,\text{sw}})} \]  

(6.15)

Seawater pressure drop evaluation was conducted as explained in section 5.1.3: no additive contributions to the pressure drop have been introduced, though Bombarda et al. [26] stated that an increase in the evaporation levels number causes a slight increase in pumping power consumption due to the sequence of heat exchangers needed in the multilevel solution. Net power output of the power plant is obtained while total heat transfer area is the sum of all areas required by all the heat exchangers involved. Finally, the value of the \( \gamma \) parameter for the multilevel CC-OTEC plant is calculated.

6.1.2 The optimization tool

An optimization tool similar to the one used of the single level closed cycle techno-economic analysis has been used. The same objective function and the same optimization variables were selected (see section 5.1.5 for details). In this case, the seawater \( \Delta T \)s optimized by the tool are the total \( \Delta T \)s endured by seawater in the heat exchangers cascade.

6.2 Optimization results

The optimization procedure was similar to the one adopted in Chapter 5: the seawater \( \Delta T \)s and evaporator and condenser pinch points that maximize the \( \gamma \) parameter were searched with the optimization tool. However, in this case the number of evaporation levels varied. Pinch points researched were the same for all evaporation levels. Input data for optimization were the same used in Chapter 5, and resumed in Table 5.2. The same cold seawater mass flowrate was assumed and the same inlet temperatures, the same \( k_p \) for seawater pumps and the same turbomachinery efficiencies and overall heat transfer coefficients were selected. The effect of seawater \( \Delta T \)s on the \( \gamma \) parameter on three plant layouts from one to three
evaporation levels were observed. For each layout, several combinations of seawater ΔTs were imposed and the optimization process researched the optimum evaporator and condenser pinch points in order to obtain the maximum γ. Three surfaces $γ = f(ΔT_{w,sw}, ΔT_{c,sw})$ have been developed and reported in Figure 6.2. The three surfaces are similar to the one depicted in Figure 5.9 for the single level Rankine cycle and are very close one another especially in the region where the maximum γ takes place. Moreover, the locus of maximum γ occurs for a similar combination of seawater ΔTs for all the three evaporation levels investigated.

Figure 6.2: Maximum γ parameter for imposed seawater ΔTs for multilevel configurations.
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An optimization for maximizing $\gamma$ by changing the seawater $\Delta T$s and the pinch points has been conducted for one to five levels configurations. Optimization results are resumed in Table 6.1. The combination of $\Delta T$s and $\gamma$ parameter for the single level Rankine cycle are the same found in the single cycle optimization in section 5.3.1 and the position and value of the maximum $\gamma$ for one, two and three levels layouts, is the same observed in Figure 6.2.

Table 6.1: Optimization results with multilevel Rankine cycle

<table>
<thead>
<tr>
<th>No. levels</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta T_{w,sw}$ [°C]</td>
<td>1.64</td>
<td>1.71</td>
<td>1.72</td>
<td>1.72</td>
<td>1.72</td>
</tr>
<tr>
<td>$\Delta T_{c,sw}$ [°C]</td>
<td>2.19</td>
<td>2.31</td>
<td>2.32</td>
<td>2.33</td>
<td>2.34</td>
</tr>
<tr>
<td>$\Delta T_{pp, evap}$ [°C]</td>
<td>3.89</td>
<td>4.24</td>
<td>4.36</td>
<td>4.43</td>
<td>4.47</td>
</tr>
<tr>
<td>$\Delta T_{pp, cond}$ [°C]</td>
<td>3.67</td>
<td>4.12</td>
<td>4.30</td>
<td>4.39</td>
<td>4.44</td>
</tr>
<tr>
<td>$W_{net}$ [kW]</td>
<td>1991</td>
<td>2110</td>
<td>2131</td>
<td>2139</td>
<td>2142</td>
</tr>
<tr>
<td>$A_{tot}$ [m²]</td>
<td>10432</td>
<td>10949</td>
<td>11040</td>
<td>11079</td>
<td>11093</td>
</tr>
<tr>
<td>$\gamma$ [kW/m²]</td>
<td>0.19085</td>
<td>0.19268</td>
<td>0.19299</td>
<td>0.19309</td>
<td>0.19313</td>
</tr>
</tbody>
</table>

In conclusion, as observed in section 4.3, an increase in number of evaporation levels brings benefits to the maximum power produced and efficiency. By observing Table 6.1, from the $\gamma$ parameter point of view, it appears that increasing the number of evaporation levels brings an increase of $\gamma$. As shown in Figure 6.3, the increase on value of the maximum $\gamma$ becomes lower as the number of evaporation levels increases. By passing from a single level to a two level configuration, $\gamma$ increase by 1% and by passing from a two level to a three level configuration the increase on $\gamma$ was only 0.16%. Adding an evaporation level increases the plant complexity and its operation. For example, adding a second evaporation level requires doubling the number of heat exchangers, the turbine, ducts and working fluid pumps. These components, apart from the heat exchangers, are not accounted in the objective function selected for optimization, but affects plant costs, of which $\gamma$ is only a rough indicator. Moreover, adding an evaporation level requires adding a heat exchanger on both the warm and cold seawater side. The pressure losses due to valves and pressure drop at the inlet and outlet of heat exchangers increase as the number of heat exchangers in series becomes higher, negatively affecting net power produced because seawater pumps consumption becomes higher. The effect of the higher pumping power required by the heat exchangers cascade in a multilevel configuration has not been evaluated in this analysis. Therefore, according to these considerations, using more than two evaporation levels may not be
economically attractive because the increase in plant costs due to added components and the increase in required pumping power could have a negative effect on plant specific cost. However, a detailed analysis of components costs and an evaluation of the increase in pressure drop due to the series of heat exchanger are required to verify if at least two evaporation levels bring advantages in terms of specific plant cost.

Figure 6.3: Optimum $\gamma$ parameter according to number of evaporation levels
7. Sizing of CC-OTEC heat exchangers and economic analysis

In chapters 5 and 6, approximate models to simulate closed cycle OTEC plants have been developed. In particular, a single level CC-OTEC plant model operating with different working fluids has been used, by means of an optimization tool, to select the working fluid with desirable characteristics for closed cycle operation. The fluid with highest $\gamma$ parameter value was ammonia and therefore was used in the analyses that followed. Optimization results of the single level closed cycle working with ammonia have been presented. A sensitivity analysis to evaluate the effect of some of the input parameters on the optimal configuration have been carried out. A model of a multilevel CC-OTEC plant has also been developed and maximization of $\gamma$ for plants operating with the same input parameters but with different number of evaporation levels has been performed. It resulted that the $\gamma$ parameter increased as the number of evaporation levels increased. However, it was observed that for a number of evaporation levels higher than two, the maximum $\gamma$ parameter increased only slightly.

The main limit of the models presented so far is the absence of a correlation between overall heat transfer coefficient and seawater pressure drop. Since heat transfer area for CC-OTEC plant is extremely large, high heat transfer coefficients are desirable in order to reduce the heat transfer area required. However, higher heat transfer coefficients generally entail higher pressure losses in the seawater side, resulting in an increase of pumping power required. Since seawater pumping power is an important auxiliary consumption that affects directly power plant performance (20%-50% of power produced by the Rankine cycle is consumed by seawater pumps), a trade off between high heat transfer coefficients and low pressure drops exists.

All pilot plants developed so far adopted the single level configuration because it is the simplest layout. In this Chapter, the single level OTEC plant model described in section 5.1
has been modified in order to include a correlation between overall heat transfer coefficient and seawater pressure drop in heat exchangers. An approximate model of a plate heat exchanger that correlates the heat transfer coefficient, plate geometry and seawater velocities, with heat exchangers pressure drop has been obtained with Aspen EDR. The heat exchanger model was then embedded in the single level CC-OTEC plant numerical model. The optimization tool has been modified to include the effect of heat exchangers geometry on the $\gamma$ parameter. Results obtained by this new optimization have been used in Aspen EDR to effectively size the titanium evaporator and condenser.

Finally, an economic analysis of the CC-OTEC plant with maximized $\gamma$ has been performed.

### 7.1 Approximate model of a plate heat exchanger for OTEC application

As described in section 2.2.5, plate exchangers are considered one of the most favorable designs for OTEC plant, in particular for their ease of fabrication and compactness. According to Qiao et al. [56], very few investigations have been carried out on the simulation of plate exchangers, especially with phase change, and only few plate exchangers models with phase transition have been developed so far. The development of a detailed model for a plate exchanger is beyond the scope of this thesis, thus, in order to include the effect of heat exchangers geometry on plant performance, an approximate model of plate exchanger has been developed. The model is able to capture the interaction between heat transfer and pressure drop of a plate exchanger with a simplified geometry. Input parameters of the approximate model of a plate exchanger are:

- Plate exchanger geometry;
- Seawater flowrate;
- Thermal power exchanged;
- Mean log temperature difference between seawater and ammonia.

The model was then included in the OTEC plant model by substituting the pressure drop for heat exchangers evaluation of section 5.1.3 and the constant overall heat transfer coefficient adopted in the previous analyses with specific correlations.

#### 7.1.1 The overall heat transfer coefficient power law function

The plate exchanger model assumed that the overall heat transfer coefficient for both evaporator and condenser is a function of seawater velocity only (assuming that heat transfer
Sizing of CC-OTEC heat exchangers and economic analysis

coefficient of ammonia during phase change remains constant). The relation between seawater velocity and global heat transfer is represented by the following equation:

\[ U_{hx} = C v_{sw,hx}^\alpha \]  

(7.1)

Determination of the constants of the power law has been done with Aspen EDR. Twenty simulations with gasketed titanium plate heat exchangers geometries operating with seawater and ammonia in typical conditions for OTEC have been performed separately for evaporator and condenser by assigning seawater flowrate and inlet and outlet conditions of the streams. The design of the exchanger consisted in finding the number of heat exchangers and plates required to satisfy the constraint:

\[ \frac{A_{\text{effective}}}{A_{\text{required}}} = 1 \]  

(7.2)

Where \( A_{\text{effective}} \) is the effective area of the heat exchangers with the number of plates selected, and \( A_{\text{required}} \) is the area obtained by the equation:

\[ A_{\text{required}} = \frac{\dot{Q}_{hx}}{U \Delta T_{ml}} \]  

(7.3)

The heat exchangers geometries used were APV heat exchangers of the series SR23 and SR14 made available by Aspen EDR. These heat exchangers had a chevron angle to horizontal of 42° and 45° and different plate length and widths, but a similar plate pitch and thickness, 4.6mm and 0.6mm respectively. Seawater flowrate was maintained constant at 300kg/s and different combinations of inlet and outlet seawater temperatures and ammonia evaporation and condensation temperatures were tested for each heat exchanger geometry. For the evaporator, the ammonia single phase section was neglected because according to simulation with Aspen EDR, the heat transfer coefficients and areas provided by the program in case of preheating and evaporation where related to a weighted mean log temperature difference. Moreover, according to Aspen EDR results, only approximately 5-8% of plate length is interested by ammonia single phase heat transfer.

Once the constraint of equation (7.2) was satisfied for all geometries and combinations of temperatures considered, seawater velocities through plates and overall heat transfer coefficients for both evaporator and condenser were plotted as in Figure 7.1. The constants of the power law function that correlates seawater velocity to overall heat transfer coefficient for evaporator and condenser have been determined:
Chapter 7

\[ U_{\text{evap}} = 4797.7 \, v_{\text{sw, evap}}^{0.5353} \]  \hspace{1cm} (7.4)

\[ U_{\text{cond}} = 4144.5 \, v_{\text{sw, hx}}^{0.3693} \]  \hspace{1cm} (7.5)

The correlations developed, have been used in the plate evaporator and condenser approximate model.

![Graph showing correlations between seawater velocity and overall heat transfer coefficient obtained with Aspen EDR](image)

**Figure 7.1:** Correlations between seawater velocity and overall heat transfer coefficient obtained with Aspen EDR

### 7.1.2 Heat transfer area evaluation

With the correlations found with Aspen EDR, a MATLAB model of plate exchangers is developed. The plate geometry considered in the plate exchanger model is shown in Figure 7.2.

![Plate exchanger geometry](image)

**Figure 7.2:** Plate exchanger geometry, adapted from [39]
The chevron angle was considered constant at 45° and the model assumed as input parameters the heat transfer area length $L_p$, the plate width $L_w$, the port diameter $D_p$ and the channel spacing $b$, obtained by subtracting the plate thickness to the plate pitch:

$$b = p - t$$  \hspace{1cm} (7.6)

A surface enlargement factor $\phi$ of 1.1 was also assumed [39]. The enlargement factor is defined in the following equation:

$$\phi = \frac{A_{\text{plate}}}{L_p L_w}$$  \hspace{1cm} (7.7)

According to equation (7.7), with the plate length and plate width selected, the plate area can be determined:

$$A_{\text{plate}} = \phi L_p L_w$$  \hspace{1cm} (7.8)

Considering a single pass heat exchanger, the number of channels per flow is:

$$N_{ch} = \frac{N_{plates} - 1}{2}$$  \hspace{1cm} (7.9)

The number of channels can be determined also by dividing the total seawater flowrate per the seawater flowrate flowing in each channel, which can be calculated by assuming a first attempt seawater velocity:

$$N_{ch} = \frac{\dot{m}_{sw,\text{hx}}}{\dot{m}_{sw,\text{ch}}} = \frac{\dot{m}_{sw,\text{hx}}}{\rho v_{sw} b L_w}$$  \hspace{1cm} (7.10)

Where the product of $b$ per $L_w$ is the channel flow area. By combining equation (7.9) and (7.10), the number of plates can be evaluated:

$$N_{plates} = 2 \frac{\dot{m}_{sw,\text{hx}}}{\rho v_{sw} b L_w} + 1$$  \hspace{1cm} (7.11)

Therefore, the total heat transfer area of the heat exchanger is:

$$A_{\text{hx}} = A_{\text{plate}} N_{plates}$$  \hspace{1cm} (7.12)

With the same velocity assumed to evaluate the number of channels, the value of the overall heat transfer coefficient is obtained with the power law of equation (7.1) with the constants experimentally found in equation (7.4) and (7.5). The product of overall heat transfer and heat transfer area for the assumed velocity is calculated:

$$UA'_{\text{hx}} = U_{\text{hx}} A_{\text{hx}} = C v_{sw,\text{hx}}^\alpha A_{\text{plate}} N_{plates}$$  \hspace{1cm} (7.13)

The product of heat transfer area per overall heat transfer coefficient can be obtained also by
dividing thermal power by the mean log temperature difference:

\[ UA''_{hx} = \frac{\dot{Q}_{hx}}{\Delta T_{ml}} \quad (7.14) \]

The seawater velocity is finally varied in order to obtain:

\[ UA'_{hx} = UA''_{hx} \quad (7.15) \]

### 7.1.3 Pressure drop evaluation

Pressure drop in a plate exchanger is composed of two terms:

\[ \Delta p_{tot} = \Delta p_{friction} + \Delta p_{port} \quad (7.16) \]

The frictional pressure drop along the seawater channels has the following form:

\[ \Delta p_{friction} = 4 f \left( \frac{L_w}{D_h} \right) \rho \frac{v_{sw}^2}{2} \quad (7.17) \]

Where the term \( D_h \) is the hydraulic diameter of channels of the plate exchanger [39]:

\[ D_h = \frac{4 \text{ channel flow area}}{\text{wetted perimeter}} = \frac{4 b L_w}{2(b + L_w \phi)} \quad (7.18) \]

The Fanning friction factor \( f \) in equation (7.17) is given by [39]:

\[ f = \frac{K_f}{R e^m} \quad (7.19) \]

Where \( Re \) is the Reynold number and values of constant \( K_f \) and \( m \) are functions of Reynold number and chevron angle [39]. The value of these constants have been chosen according to data reported by Kakaç et al [39] for Reynold number greater than 300 and chevron angle of 45°. In this case, the value of \( K_f \) and \( m \) are 1.441 and 0.206, respectively.

The pressure drop in ports can be roughly estimated [39]:

\[ \Delta p_{port} = 1.4 \frac{G_p^2}{2 \rho} \quad (7.20) \]

Where \( G_p \) is the mass flux in the port ducts:

\[ G_p = \frac{\dot{m}_{sw}}{\pi D_p^2} \quad (7.21) \]

Where \( D_p \) is the port diameter. From equation (7.20) and (7.21) it is clear that port pressure drop is inversely proportional to its diameter.
7.1.4 Plate exchanger approximate model validation

The approximate model for plate exchangers have been validated by selecting a number of commercial plate geometries and comparing Aspen EDR results with approximate model results in typical condition for OTEC evaporator and condenser. Seawater flowrate equal to 300kg/s was selected for all simulations. Part of comparison result is reported in Table 7.1. Error of approximate model results respect to Aspen EDR for both overall heat transfer coefficient and pressure drop was comprised within ±10% except for a case in which the error on pressure drop was 12%. The discordance of pressure drop between Aspen EDR results and approximate model results in particular is due to pressure drop in ports estimation because equation (7.20), according to [39] is a rough correlation for pressure drop in ports. If the frictional pressure drop only, was compared, the error was lower. Results of the approximate plate exchanger model were considered acceptable for a rough estimation of heat transfer coefficient and pressure drop according to inlet and outlet conditions of streams and plate geometry.

<table>
<thead>
<tr>
<th>Plate</th>
<th>T_{in,sw} [°C]</th>
<th>T_{out,sw} [°C]</th>
<th>T_{amm} [°C]</th>
<th>U_{Aspen} [W/m²K]</th>
<th>U_{model} [W/m²K]</th>
<th>ErrU %</th>
<th>ΔP_{Aspen} [bar]</th>
<th>ΔP_{model} [bar]</th>
<th>ErrΔP %</th>
</tr>
</thead>
<tbody>
<tr>
<td>APVSR23AO</td>
<td>28</td>
<td>21</td>
<td>19.7</td>
<td>1421.5</td>
<td>1538.8</td>
<td>-8.25</td>
<td>0.016</td>
<td>0.017</td>
<td>-9.88</td>
</tr>
<tr>
<td>APVSR23AN</td>
<td>28</td>
<td>23.2</td>
<td>20.5</td>
<td>4096.9</td>
<td>4082.7</td>
<td>0.35</td>
<td>0.590</td>
<td>0.544</td>
<td>7.83</td>
</tr>
<tr>
<td>APVSR14AP</td>
<td>28</td>
<td>23.2</td>
<td>21.2</td>
<td>3771.3</td>
<td>3813.6</td>
<td>-1.12</td>
<td>0.586</td>
<td>0.610</td>
<td>-4.29</td>
</tr>
<tr>
<td>APVSR23AO</td>
<td>4</td>
<td>8</td>
<td>11</td>
<td>3143.7</td>
<td>3372.1</td>
<td>-7.26</td>
<td>0.302</td>
<td>0.323</td>
<td>-6.92</td>
</tr>
<tr>
<td>APVSR23AO</td>
<td>4</td>
<td>10</td>
<td>11.6</td>
<td>2023.7</td>
<td>2060.9</td>
<td>-1.84</td>
<td>0.027</td>
<td>0.025</td>
<td>6.60</td>
</tr>
<tr>
<td>APVSR23AN</td>
<td>4</td>
<td>9</td>
<td>11</td>
<td>2964.7</td>
<td>3055.9</td>
<td>-3.08</td>
<td>0.265</td>
<td>0.257</td>
<td>2.94</td>
</tr>
</tbody>
</table>

7.2 The new model of a CC-OTEC plant

The approximate plate exchangers model has been integrated in the single level OTEC plant model presented in section 5.1. Input parameters for the new model are the following:

- Cold seawater mass flowrate;
- Warm and cold seawater inlet temperatures;
- Seawater ΔTs across the exchangers;
- Pinch point temperature differences at both the evaporator and condenser;
- Turbomachinery efficiencies;
Chapter 7

- CWP and warm water pipe geometry;
- Seawater pumps efficiencies;
- Plate exchangers geometries of both evaporator and condenser. Plate exchanger geometric parameters required per each heat exchanger are \( L_p, L_w, b \) and \( D_p \).

The process followed by the new CC-OTEC plant model is the following:

Once input parameters are assigned, the model performs computes the thermodynamic properties in Rankine cycle points, warm seawater mass flowrate and gross power produced by the Rankine cycle according to equations presented in section 5.1.1. After these calculations, temperatures of inlet and outlet streams and seawater and ammonia flowrates
in all heat exchangers are known. Thermal powers exchanged at both the evaporator and condenser are also known. Therefore, all input parameters required by the plate exchanger approximate model are known. The heat exchanger model for evaporator and condenser with the assumed geometries is used to estimate heat transfer area required and pressure drop on seawater side of heat exchangers according to procedure explained in sections 7.1.2 and 7.1.3. For the evaporator, the preheating section was neglected in the heat transfer area calculation since its contribution to total heat transfer area is low. After heat exchanger pressure drop on seawater side have been evaluated the overall pressure drop can be calculated by adding heat exchangers pressure losses to pressure losses in pipes. Seawater pumping power is obtained according to equation (5.30) and net electric power produced by the plant is finally obtained. The $\gamma$ parameter is finally calculated by diving net electric power produced per the total heat transfer area required by heat exchangers.

7.2.1 The new optimization tool
An optimization of a single level closed cycle for OTEC have been performed with the new model described in the previous section. The optimization tool was modified in order to include plate exchangers geometry in the list of optimization variables. The maximization of the $\gamma$ parameter has been performed by varying the following input parameters of the model:

- Seawater $\Delta$Ts across the exchangers;
- Pinch point temperature differences at both the evaporator and condenser;
- Plate exchangers geometries of both evaporator and condenser. In particular, $L_{p,\text{evap}}, L_{w,\text{evap}}, D_{p,\text{evap}}, L_{p,\text{cond}}, L_{w,\text{cond}}, D_{p,\text{cond}}$.

In order to limit the pressure drop in ports which increases proportionally with seawater flowrate entering the heat exchanger, the seawater flowrate in each heat exchanger was limited to 300kg/s. This value was the seawater flowrate used to develop the power law functions for evaporator and condenser in section 7.1.2 and to validate the plate exchangers approximate model in section 7.1.4. The number of heat exchangers required in parallel has been obtained by rounding out the following equation:
Chapter 7

\[ N_{hx\text{ in parallel}} = \frac{m_{sw}}{300 kg/s} \] (7.22)

Plate exchanger geometry was subjected to the following constraints. Since, according to approximate model, in order to increase heat transfer area without increasing pressure drop, the optimization process tends to search for the largest plate width possible. In order to avoid flow maldistribution, according to Kakaç et al. [39] plate width should be limited. Maximum permitted plate width was assumed to be equal to plate length \( L_p \). The maximum port diameter was assumed to be 400mm, that is the limit diameter for gasketed plate exchangers [39]. In the optimization process, port diameter was also assumed to be at least lower than half the plate width; otherwise plate exchangers cannot be effectively built because port diameters are intersected.

### 7.2.2 Parameters assumed in the analysis

Input parameters for the optimization are the same reported in Table 5.2 except for seawater pumps proportionality constants \( k_p \), which are not required with the new model developed because a correlation between plate geometry, heat transfer coefficient and related pressure drop has been established. Turbomachinery efficiencies were maintained constant during optimization. Channel spacing of heat exchangers was maintained equal to 4mm: the heat exchangers used to develop the power law functions of overall heat transfer coefficient had the same channel spacing.

### 7.3 Optimization results

The maximization of the \( \gamma \) parameter lead to the following results:

<table>
<thead>
<tr>
<th>( \Delta T_{w,sw} )</th>
<th>3.26°C</th>
<th>( \Delta T_{c,sw} )</th>
<th>3.26°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta T_{pp,\text{evap}} )</td>
<td>2.84°C</td>
<td>( \Delta T_{pp,\text{cond}} )</td>
<td>2.84°C</td>
</tr>
<tr>
<td>( m_{w,sw} )</td>
<td>8798 kg/s</td>
<td>( m_{\text{amm}} )</td>
<td>92.9 kg/s</td>
</tr>
<tr>
<td>( T_{\text{evap}} )</td>
<td>22.05°C</td>
<td>( T_{\text{cond}} )</td>
<td>10.11°C</td>
</tr>
<tr>
<td>( W_{el,\text{turbine}} )</td>
<td>3.936 MW</td>
<td>( \dot{W}_{el,\text{amm, pump}} )</td>
<td>58.9 kW</td>
</tr>
<tr>
<td>( \Delta p_{w,sw} )</td>
<td>0.458 bar</td>
<td>( \Delta p_{w,sw} )</td>
<td>0.758 bar</td>
</tr>
<tr>
<td>( W_{el,\text{sw,pump}} )</td>
<td>493.3 kW</td>
<td>( W_{el,\text{c,sw,pump}} )</td>
<td>783.8 kW</td>
</tr>
<tr>
<td>( W_{el,\text{net}} )</td>
<td>2.6 MW</td>
<td>( \eta_{1st \text{ law}} )</td>
<td>2.27%</td>
</tr>
<tr>
<td>( A_{hx, \text{tot}} )</td>
<td>14118 m²</td>
<td>( \gamma )</td>
<td>0.184 kW/m²</td>
</tr>
</tbody>
</table>
Optimized geometry, number of exchangers and performance of plate exchangers are resumed in Table 7.3. As expected, the port diameter and the plate width approached both the limits imposed. The overall heat transfer coefficient of the condenser is 86% of that of the evaporator. Since seawater pressure drop on cold water pipe is higher than that in the warm water pipe, in order to reduce cold seawater pump consumption, the optimal seawater velocity for condenser is lower than the evaporator one and frictional pressure drop is lower for condenser.

Optimization results are different from results found in the optimization performed in Chapter 5 with simplified assumptions for heat exchangers.

Table 7.3: Plate exchanger model results for maximum $\gamma$

<table>
<thead>
<tr>
<th></th>
<th>Evaporator</th>
<th></th>
<th>Condenser</th>
</tr>
</thead>
<tbody>
<tr>
<td>$N_{plates}$</td>
<td>211</td>
<td>$N_{plates}$</td>
<td>244</td>
</tr>
<tr>
<td>$L_p$</td>
<td>978mm</td>
<td>$L_p$</td>
<td>981mm</td>
</tr>
<tr>
<td>$L_w$</td>
<td>978mm</td>
<td>$L_w$</td>
<td>981mm</td>
</tr>
<tr>
<td>$D_p$</td>
<td>391mm</td>
<td>$D_p$</td>
<td>392mm</td>
</tr>
<tr>
<td>$v_{sw}$</td>
<td>0.72m/s</td>
<td>$v_{sw}$</td>
<td>0.62m/s</td>
</tr>
<tr>
<td>$m_{sw}$</td>
<td>293.3kg/s</td>
<td>$m_{sw}$</td>
<td>293.1kg/s</td>
</tr>
<tr>
<td>$U_{evap}$</td>
<td>4019.4W/(m²K)</td>
<td>$U_{cond}$</td>
<td>3464.7W/(m²K)</td>
</tr>
<tr>
<td>$A_{evap}$</td>
<td>221m²</td>
<td>$A_{cond}$</td>
<td>258m²</td>
</tr>
<tr>
<td>$\Delta p_{friction}$</td>
<td>0.3382bar</td>
<td>$\Delta p_{friction}$</td>
<td>0.2882bar</td>
</tr>
<tr>
<td>$\Delta p_{port}$</td>
<td>0.0838bar</td>
<td>$\Delta p_{port}$</td>
<td>0.0822bar</td>
</tr>
<tr>
<td>$\Delta p_{tot}$</td>
<td>0.4221bar</td>
<td>$\Delta p_{tot}$</td>
<td>0.3704bar</td>
</tr>
</tbody>
</table>

### 7.4 Sizing of Heat Exchangers

Optimal geometries of plate exchangers estimated by the model and reported in Table 7.3, have been used as baseline for sizing of heat exchanger operating in the optimized conditions for the 6.2MWe OTEC plant resumed in Table 7.2.

#### 7.4.1 Sizing of evaporator

Sizing of evaporator have been performed with Aspen EDR. Geometric parameters required by the program to perform sizing are:

- Horizontal port centers distance;
- Vertical port centers distance;
Plate thickness;
Compressed plate pitch;
Port diameter;
Plate width;
Chevron angle (to horizontal);
Plate area;
Number of channels.

The approximate model reported plate length $L_p$, plate width $L_w$ and port diameters $D_p$. However, the other geometric parameters can be rapidly obtained by observing Figure 7.2. The vertical and horizontal port centers distances are [39]:

\[
L_v = L_p + D_p \tag{7.23}
\]
\[
L_h = L_w - D_p \tag{7.24}
\]

Plate thickness was assumed 0.6mm according to limits for titanium heat exchangers reported by [20]. If plate thickness and channel spacing are defined, compressed plate pitch is obtained by manipulating equation 5.6. Plate spacing $b$ was assumed to be 4mm, as the value of the commercial heat exchangers adopted in the power law determination. The chevron angle was assumed to be 42°. The number of channels was obtained by the number of plates suggested by the approximate model by using equation 5.9. Plate geometry adopted in the simulation with Aspen EDR is the following:

<table>
<thead>
<tr>
<th>Table 7.4: Evaporator geometric parameters derived from approximate model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chevron angle to horizontal</td>
</tr>
<tr>
<td>Horizontal port centers distance</td>
</tr>
<tr>
<td>Vertical port centers distance</td>
</tr>
<tr>
<td>Plate thickness</td>
</tr>
<tr>
<td>Compressed plate pitch</td>
</tr>
<tr>
<td>Port diameter</td>
</tr>
<tr>
<td>Plate width</td>
</tr>
<tr>
<td>Plate area</td>
</tr>
</tbody>
</table>

Sizing of the evaporator was conducted first by sizing a heat exchanger with the two phase section only. Then the complete evaporator with also the single phase ammonia section has been designed. Sizing of the two-phase only was conducted to compare results of the
approximate model with the Aspen EDR results.

By considering the ammonia two-phase section only, the inlet and outlet conditions of seawater and ammonia are the following:

<table>
<thead>
<tr>
<th>Table 7.5: Inlet conditions for the two phase section only sizing</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Seawater</strong></td>
</tr>
<tr>
<td>$m_{sw}$</td>
</tr>
<tr>
<td>$T_{in}$</td>
</tr>
<tr>
<td>$T_{out}$</td>
</tr>
</tbody>
</table>

The approximate model and Aspen EDR results are then compared in Table 7.6.

<table>
<thead>
<tr>
<th>Table 7.6: Approximate model and Aspen EDR results considering two phase section only.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Approximate model</strong></td>
</tr>
<tr>
<td>$N_{plates}$</td>
</tr>
<tr>
<td>$U_{evap}$</td>
</tr>
<tr>
<td>$\Delta p_{friction}$</td>
</tr>
<tr>
<td>$\Delta p_{port}$</td>
</tr>
<tr>
<td>$\Delta p_{tot}$</td>
</tr>
</tbody>
</table>

According to Table 7.6, the approximate model prediction of number of channels overall heat transfer coefficients and frictional pressure drop lead to errors lower than 5% respect to Aspen EDR results. As previously observed in model validation, the port pressure drop obtained by the approximate model differs from Aspen EDR results. The error on the overall pressure drop however is 10%.

The inlet and outlet conditions of the complete evaporator are resumed in the following table:

<table>
<thead>
<tr>
<th>Table 7.7: Inlet and outlet conditions for the complete evaporator</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Seawater</strong></td>
</tr>
<tr>
<td>$\dot{m}$</td>
</tr>
<tr>
<td>$T_{in}$</td>
</tr>
<tr>
<td>$T_{out}$</td>
</tr>
<tr>
<td>$p_{in}$</td>
</tr>
</tbody>
</table>

The overall heat transfer coefficient determined by Aspen EDR in this case was referred to a mean log temperature difference weighted on thermal power exchanged during single phase and two phase heat transfer. If plate geometry of Table 7.4 is used, because of a
reduction in overall heat transfer coefficient respect to the evaporation only section, the number of plate required was 267. Since seawater flowrate was the same as the evaporation section only sizing, seawater velocity in channel is reduced, reducing the pressure drop to a lower value than that estimated by the OTEC plant model. In order to obtain a pressure drop similar to the one obtained in the plant model, the vertical port centers distance of the plates has been raised from 1369mm to 1490mm in the Aspen EDR input geometry. In this case, the number of plates required was 219 and the pressure drop of the heat exchanger was similar to the pressure drop obtained by the $\gamma$ optimization developed in the preceding section.

A sketch of the plate evaporator geometry sized with Aspen EDR is shown in Figure 7.4.

Figure 7.4: Aspen EDR sketch of the evaporator.
Sizing of CC-OTEC heat exchangers and economic analysis

Sizing results provided by Aspen EDR are the following:

<table>
<thead>
<tr>
<th>Table 7.8: Aspen EDR sizing results for evaporator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total heat exchanged</td>
</tr>
<tr>
<td>Overall heat transfer coefficient</td>
</tr>
<tr>
<td>Overall effective MTD</td>
</tr>
<tr>
<td>Effective surface area</td>
</tr>
<tr>
<td>Number of plates</td>
</tr>
<tr>
<td>Frictional pressure drop</td>
</tr>
<tr>
<td>Port pressure drop</td>
</tr>
<tr>
<td>Total pressure drop</td>
</tr>
<tr>
<td>Plate pack weight (empty)</td>
</tr>
<tr>
<td>Total cost (1 heat exchanger)</td>
</tr>
</tbody>
</table>

7.4.2 Sizing of condenser

Sizing of condenser was simpler because ammonia flow leaving the turbine is already in the two phase section with 0.97 vapor quality, complete condensation occurs and saturated liquid is obtained at the exchanger outlet. Inlet and outlet conditions are resumed in the following table:

<table>
<thead>
<tr>
<th>Table 7.9: Condenser inlet and outlet conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td>$\dot{m}$</td>
</tr>
<tr>
<td>$T_{in}$</td>
</tr>
<tr>
<td>$T_{out}$</td>
</tr>
<tr>
<td>$p_{in}$</td>
</tr>
<tr>
<td>Inlet vapor quality</td>
</tr>
</tbody>
</table>

Plate geometry results from approximate model have been used to define geometric parameters required by Aspen EDR that are presented in Table 7.10.
Table 7.10: Condenser geometric parameters obtained from the approximate model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chevron angle to horizontal</td>
<td>42°</td>
</tr>
<tr>
<td>Horizontal port centers distance</td>
<td>589mm</td>
</tr>
<tr>
<td>Vertical port centers distance</td>
<td>1373mm</td>
</tr>
<tr>
<td>Plate thickness</td>
<td>0.6mm</td>
</tr>
<tr>
<td>Compressed plate pitch</td>
<td>4.6mm</td>
</tr>
<tr>
<td>Port diameter</td>
<td>392mm</td>
</tr>
<tr>
<td>Plate width</td>
<td>983mm</td>
</tr>
<tr>
<td>Plate area</td>
<td>1.06m²</td>
</tr>
</tbody>
</table>

With this geometry, sizing of condenser has been performed with Aspen EDR. The approximate model and Aspen EDR results are then compared:

Table 7.11: Approximate model and Aspen EDR results for condenser

<table>
<thead>
<tr>
<th></th>
<th>Approximate model</th>
<th>Aspen EDR</th>
</tr>
</thead>
<tbody>
<tr>
<td>$N_{plates}$</td>
<td>244</td>
<td>265</td>
</tr>
<tr>
<td>$U_{cond}$</td>
<td>3464W/(m²K)</td>
<td>3295.3W/(m²K)</td>
</tr>
<tr>
<td>$\Delta p_{friction}$</td>
<td>0.2882bar</td>
<td>0.2678bar</td>
</tr>
<tr>
<td>$\Delta p_{port}$</td>
<td>0.0822bar</td>
<td>0.0291bar</td>
</tr>
<tr>
<td>$\Delta p_{tot}$</td>
<td>0.3704bar</td>
<td>0.2969bar</td>
</tr>
</tbody>
</table>

As previously observed in the model validation and sizing of evaporator, the estimation of pressure drop in ports of the approximate model is different from the Aspen EDR results obtained with the same geometry. Error committed by approximate model on overall heat transfer coefficient for condenser was 5% compared to the Aspen EDR results. The effective area required by Aspen involved 21 plates more than expected for the selected geometry. Consequently, since 10 channels are added, the frictional pressure drop is lower than expected by the approximate model because seawater flowrate was divided in a larger number of channels. The same procedure as for the evaporator is proposed. Vertical port centers distance was increased from 1373mm to 1450mm in order to meet the pressure drop and number of plates suggested by the approximate model. Simulation with increased plate length leads to results of Table 7.12. The solution proposed has a number of plates and pressure drop similar to the values provided by the approximate model in Table 7.3 and error on overall heat transfer between approximate model and Aspen EDR results is lower than 3%.
Table 7.12: Aspen EDR sizing results for condenser with modified vertical port centers distance

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total heat exchanged</td>
<td>3910.8kW</td>
</tr>
<tr>
<td>Overall heat transfer coefficient</td>
<td>3370W/(m²K)</td>
</tr>
<tr>
<td>Overall effective MTD</td>
<td>4.27°C</td>
</tr>
<tr>
<td>Effective surface area</td>
<td>272.5m²</td>
</tr>
<tr>
<td>Number of plates</td>
<td>241</td>
</tr>
<tr>
<td>Frictional pressure drop</td>
<td>0.3419bar</td>
</tr>
<tr>
<td>Port pressure drop</td>
<td>0.0276bar</td>
</tr>
<tr>
<td>Total pressure drop</td>
<td>0.3695bar</td>
</tr>
<tr>
<td>Plate pack weight (empty)</td>
<td>963.7kg</td>
</tr>
<tr>
<td>Total cost (1 heat exchanger)</td>
<td>265728$</td>
</tr>
</tbody>
</table>

A sketch of plate condenser geometry appears in the following figure:

Figure 7.5: Aspen EDR sketch of the condenser.
The heat exchangers cost estimated by Aspen EDR for both evaporator and condenser have been used for heat exchangers’ cost estimation in the following section.

### 7.5 Economic analysis

Determination of specific cost for OTEC plants is an extremely difficult task because of high uncertainties. Since a commercial size plant has not been developed yet, most of cost estimates are obtained by plant conceptual designs. According to Avery et al. [5]: “[…]

industrial experience shows that the estimate cost to complete development of a new technology generally increases as development proceeds from conceptual design to final commercial manufacture. The difference between initial design estimated and final cost is dependent on the extent to which the manufacturing process employs already development equipment, procedures and facilities”. Most of CC-OTEC plant components are off the shelf or adapted components from other applications. Heat exchangers, turbine, and other components of the power generation block are well known components in power generation industry. The CWP instead is the highest source of uncertainty in determination of plant cost. According to Upshaw, the CWP cost for a 10MWe plant can vary from 10M$ to 50M$ and this great uncertainty, which is related to the material selection and installation process (see section 2.1 for details), has an important impact on final plant cost. Avery et al. [5] introduced a 100% uncertainty on CWP cost in its cost estimate.

A significant variation in OTEC plant costs is also related to plant siting. In case of a moored plant the cost of the hull, mooring system and power transmission cable should be accounted. CWP attachment to the barge is also difficult in case of moored plants. As observed in section 1.2, one of the major causes of failure in developing an offshore OTEC plant is related to CWP attachment. In case of an onshore plant instead, a barge is not required, but the CWP can be up to five times the length required in case of an offshore plant because of bathymetric profile of site in which the plant is positioned [7]. Because of all these uncertainties the optimization of the plant according to the $\gamma$ parameter was considered reasonable as underlined by Uehara et al. and Yang et al. [22, 35].

An evaluation of plant costs for the 2.6MWe power plant obtained by the new CC-OTEC plant model of section 7.3 has been performed.

The following components costs have been approximately assessed considering an onshore plant in which CWP length was assumed equal to 1000m:

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- CWP;
- Evaporator;
- Condenser;
- Turbine;
- Seawater pumps.

Reference cost for components have been determined according to Lockheed Martin 2.5MWe Mini Spar report [27]. Since the cost provided by this report were only a rough order of magnitude estimate, and report was dated 2011, since it was observed that the CEPCI ratio between 2016 and 2011 was only 0.95 [57], no inflation rates were assumed and USD have been converted in € with a 0.8917 coefficient [58].

The limit diameter assumed for the CWP was 2.5m, according to the limit of production for HDPE. The CWP in the 2.6MWe OTEC plant is 1000m long and 2.5m in diameter, equal to the Lockheed Martin CWP. According to Lockheed Martin [27], for this plant size the HDPE pipe is a cost effective material, however in case of larger plants, FRP is considered the best cost effective material, but it is currently under test for large diameters. Therefore, the CWP cost was assumed equal to the cost provided for the Lockheed Martin Mini-Spar plant, which was roughly 4.89M€. The CWP cost specific to cold seawater volumetric flowrate assumed in the plant modeled in this thesis is roughly 590k€/(m³/s).

For the 2.5MWe OTEC Mini-spar plant power produced by the turbine was 4.4MWe. An axial turbine rotating at 1800rpm speed was selected for the Mini Spar and the cost of the turbogenerator was 1.87M€. The turbogenerator cost of the 2.6MWe plant, which produced 3.94MWe, has been obtained with the sixth tenth rule [59]:

\[
C_{\text{turbogen}} = C_{\text{turbogen,ref}} \left( \frac{W_{\text{turbogen}}}{W_{\text{turbogen,ref}}} \right)^{0.6} \quad (7.25)
\]

The turbogenerator cost was therefore equal to 1.75M€, 442€/kWe of electric power produced by the turbogenerator.

The heat exchangers costs have been obtained from Aspen EDR results of section 7.4. Thirty evaporators as the one sized in section 7.4.1 are required. The cost of a titanium plate evaporator is 224k€, therefore, the cost of thirty evaporators is 6.709M€. Twenty-nine condensers as the one sized in section 7.4.2 are required. The cost of a single titanium plate condenser is 237k€, therefore the cost of all condensers required is 6.872M€. The heat exchanger cost obtained was similar to the heat exchangers cost provided by Lockheed
Martin for the 2.5MWe Mini Spar plant [27]. The cost of titanium plate exchangers specific to their surface is 869€/m².

Finally, the warm and cold seawater pumps costs have been determined considering a cost specific to pump electric consumption of 890€/kWe as reported by Upshaw [3]. Therefore, the cost of the warm water pump is 439k€ while the cost of the cold water pump is 698k€.

The remaining components of power block such as working fluid loop, mechanical and electrical installation and plant controls have been considered to be 26% of the sum of components cost estimated so far, according to a cost breakdown structure provided by Vega [6].

Assuming that the plant will be an onshore plant, a rough estimate of engineering and project management costs of 10.6M€ have been introduced according to costs provided by Lockheed Martin [27]. This cost was not included in many economic analyses found in literature [3, 23] but included in many conceptual designs of the 1980s [5]. According to the author, since no commercial power plants have not been constructed yet, and the engineering procedure is not standardized, design and project management costs of the power plant cannot be neglected.

The results of cost estimates of components for the 2.6MWe power plant resulting from the numerical model developed are reported in the following table:

<table>
<thead>
<tr>
<th>Component</th>
<th>Cost [M€]</th>
<th>Specific costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>CWP</td>
<td>4.887</td>
<td>590k€/(m³/s)cold,sw</td>
</tr>
<tr>
<td>Turbogenerator</td>
<td>1.745</td>
<td>442€/kW_{el,turbine}</td>
</tr>
<tr>
<td>Evaporator</td>
<td>6.709</td>
<td>869€/m²</td>
</tr>
<tr>
<td>Condenser</td>
<td>6.872</td>
<td>869€/m²</td>
</tr>
<tr>
<td>Warm seawater pump</td>
<td>0.439</td>
<td>890€/kW_{el,pump}</td>
</tr>
<tr>
<td>Cold seawater pump</td>
<td>0.698</td>
<td>890€/kW_{el,pump}</td>
</tr>
<tr>
<td>Other costs</td>
<td>5.55</td>
<td>1408€/kW_{el,gross}</td>
</tr>
<tr>
<td>Engineering and project management</td>
<td>10.6</td>
<td>4076€/kW_{el,net}</td>
</tr>
</tbody>
</table>

The effective cost of the 2.6MWe power plant with the assumptions introduced is therefore 37.50M€ and the specific cost obtained is 14423€/kWe (specific to net electricity output). As observed in Figure 7.6, it can be seen that the evaporator and condenser are responsible of 36% of the plant cost, in accordance with the the range identified by Uehara et al. [35].
The Levelized Cost of Energy (“LCOE”) of the power plant have been estimated through the fixed charge ratio FCR [60]. According to Binotti [60], the FCR is “[…] the fraction of the total investment cost that the investor has to cover every year to face the yearly depreciation of return of the capital, tax expense, insurance expense associated with the installation of a particular generating unit […].” Assuming a plant life of thirty years [20], and assuming a debit share of 60%, a cost of debt of 60%, an equity share of 40% and a cost of equity of 13%, an FCR equal to 10.05% is obtained [60]. The LCOE has the following expression:

\[
LCOE = \frac{CC}{EE} FCR + \frac{C_{O&M}}{EE}
\]  

Where CC is plant capital cost, FCR is the fixed charge ratio, \(C_{O&M}\) is the annual operation and maintenance costs and \(EE\) is energy produced. The operation and maintenance costs selected for the estimation is 3.3% of the plant cost, assuming that the first OTEC plant could have lifetime O&M costs roughly equivalent to replacing the entire plant [3]. Electric energy produced by the plant was calculated assuming 8000h/year of plant operation [23] and assuming that inlet seawater temperatures remained constant during the year. With inlet temperatures of 28°C and 4°C for warm and cold seawater respectively, electric energy
produced is 20.8GWhe. With the capital cost estimated equal to 37.5M€, the estimated LCOE for the OTEC plant is 240€/MWhe. The main assumptions for LCOE calculation are resumed in Table 7.14.

<table>
<thead>
<tr>
<th>Warm seawater inlet temperature</th>
<th>28°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cold seawater inlet temperature</td>
<td>4°C</td>
</tr>
<tr>
<td>Net electric power produced</td>
<td>2.6MWe</td>
</tr>
<tr>
<td>Hours of operation</td>
<td>8000h/yr (at constant power)</td>
</tr>
<tr>
<td>Plant cost</td>
<td>37.50M€</td>
</tr>
<tr>
<td>Operation and maintenance</td>
<td>3.3% of plant cost</td>
</tr>
<tr>
<td>FCR</td>
<td>10.05%</td>
</tr>
<tr>
<td>Plant life</td>
<td>30 years</td>
</tr>
</tbody>
</table>

According to Upshaw, the LCOE of an OTEC plant fall in a range between 97€/MWhe and 550€/MWhe [3]. This LCOE is only an estimate because the effective equivalent hours depend on the seawater temperature which may change during the year. If seawater tropical seas like the one listed in section 1.1 are considered, the assumption of constant surface temperature is considered reasonable.

2016 estimates of average LCOE for selected plants have been provided by Upshaw [3] and IRENA [61] and reported in Table 7.15.

Table 7.14: assumptions for LCOE estimation for the OTEC plant

Table 7.15: Average LCOE of other power production technologies [3, 61]

<table>
<thead>
<tr>
<th>Power Production Technology</th>
<th>LCOE €/MWh</th>
<th>Ref</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conv coal</td>
<td>89</td>
<td>[3]</td>
</tr>
<tr>
<td>Coal with CCS</td>
<td>128</td>
<td>[3]</td>
</tr>
<tr>
<td>NGCC</td>
<td>62</td>
<td>[3]</td>
</tr>
<tr>
<td>Adv Nuclear</td>
<td>117</td>
<td>[3]</td>
</tr>
<tr>
<td>Offshore wind</td>
<td>152</td>
<td>[3]</td>
</tr>
<tr>
<td>Geothermal</td>
<td>95</td>
<td>[3]</td>
</tr>
<tr>
<td>Hydro</td>
<td>81</td>
<td>[3]</td>
</tr>
<tr>
<td>CSP</td>
<td>161</td>
<td>[61]</td>
</tr>
</tbody>
</table>
The estimated LCOE for the OTEC plant is higher than that of the other technologies reported. The higher LCOE was expected because any plant built will be the first of its kind at that size, and manufacturing and installation would be a highly customized and engineered process [3]. If the engineering and project management cost is halved, the resulting LCOE for the OTEC plant would be 206€/MWh which is 85% of the estimated LCOE. Higher economic feasibility can be reached if the OTEC plant is coupled with facilities for OTEC by products production (e.g. desalinated water), as stated in section 1.6.

High economies of scale are expected for OTEC technology [6]. In fact, most of conceptual designs presented so far were for sizes up to 40MWe. However, these designs included components such as large diameters FRP CWP (up to 8m [5]) which testing is currently underway [5, 27, 33]. The CWP diameter (and related cold seawater flowrate) assumed in the present thesis work is a diameter that can be commercially developed with the existing technologies and facilities [34]. If as expected, new materials with larger diameters will be developed in the near future, an increase in cold seawater flowrate employed, which is proportional to the plant size, will bring a reduction in plant specific cost.

According to section 2.2, if industrial production of plate exchangers for OTEC technology will take-off an important cost reduction of manufacturing cost of this components is expected. Moreover, if aluminum exchangers with special features to resist pitting corrosion in seawater are developed, another important reduction in costs is also expected. Efficiency improvements are also an important lever in cost reduction. If higher efficiencies are reached, the net power output increase could cause a reduction in plant specific cost.

In states where fuel costs for conventional power production are high the OTEC technology could be economically sustainable in future. In case of Hawaii, the mean electricity price for June 2016 converted in EUR was 0.214€/kWh [62]. If plant cost reductions expected in future effectively occur, the economical feasibility of OTEC in the context of small islands will be demonstrated.
Conclusions and future developments

In the present thesis work, a techno-economic analysis of OTEC cycles for power production has been conducted. A detailed literature analysis on the OTEC technology has been conducted on plant configurations and plant components. The attention was focused on closed cycle OTEC plants because, from literature review, emerged that technical feasibility of these plants has been already demonstrated and since components required by the closed cycle are well known in the power generation industry, can be readily adapted for OTEC application. Thermodynamic analysis of closed cycle OTEC plants have been performed. Exergy analysis highlighted that exergy destroyed during heat transfer processes was the largest part. Therefore, any improvement to reduce the temperature difference between seawater and working fluid in heat exchangers, such has the adoption of a multilevel cycle, increase the closed cycle power output and efficiency.

Since no commercial size power plant has not been developed yet, high uncertainties on cost of plant components exist. The cold water pipe especially has been detected has the highest source of cost uncertainties. Since it was observed that heat exchangers in a closed cycle OTEC plant are a relevant part of plant cost (from 25% to 50%, because titanium is used for heat exchangers), according to techno-economic optimizations of OTEC plants found in literature, an objective function which was independent of components’ costs have been used in this-techno economic optimization. This objective function, traditionally called $\gamma$, is the ratio between net power produced by the plant and heat exchangers area required. Techno-economic optimization of an OTEC plant is a generally a multivariable problem (e. g. seawater temperature across heat exchanger, pinch points at the evaporator and condenser, heat exchangers geometry). In order to perform techno-economic optimization, numerical models of closed cycle OTEC plants with one evaporation level and with multilevel configuration have been developed. Simplifying assumptions for heat exchangers were assumed. An optimization tool for maximization of $\gamma$ parameter by optimizing seawater temperature difference across heat exchangers and pinch points have been also developed.
Conclusions and future developments

and used to assess optimal parameters of the plant.

The single level CC-OTEC numerical model with embedded optimization tool was used to study the effect of the working fluid choice on the $\gamma$ parameter. Performance of fourteen working fluids among refrigerant fluids and hydrocarbons has been assessed. It resulted that for seawater inlet temperatures of 28°C and 4°C and with other input parameters equal, the value of maximum $\gamma$ for ammonia was 0.1909kWe/m², while maximum $\gamma$ for the other fluids was between $3x10^{-2}$kWe/m² and 0.1342 kWe/m². Hence, ammonia, in accordance with most of researchers’ opinion, resulted the best working fluid for CC-OTEC. With a cold seawater flowrate of 8500kg/s, the plant with ammonia as working fluid produced 2MWe of net power with an efficiency of 2.6%, a typical value for single level CC-OTEC plants. A sensitivity analysis to evaluate the effect of some input parameters of the model (inlet and outlet seawater temperatures, turbine efficiency) have been conducted with ammonia as the working fluid. It was observed that the higher the temperature difference between warm and cold inlet seawater, the higher the value of $\gamma$. A site off the coast of Puglia in the Adriatic Sea was compared with other sites favorable for OTEC in tropical seas maintaining equal the other parameters required by the model. The maximum $\gamma$ parameter found for tropical sites was between 0.16kW/m² and 0.20kW/m², while for the Italian site the maximum $\gamma$ found was lower than 0.08kW/m². In the sensitivity analysis it was also observed that an increase of 1% in turbine efficiency lead to an increase of 2% on the maximum value of $\gamma$.

The optimization of $\gamma$ by changing the number of evaporation levels has been performed by using the multilevel CC-OTEC numerical model with ammonia as working fluid coupled with optimization tool. A slight advantage in $\gamma$ was observed for the multilevel configuration. The increase on maximum $\gamma$ from the single level to the two level configuration is 1%, and for a higher number of evaporation levels the increase is lower. It was observed in fact that by passing from a two level to a three level configuration the increase on $\gamma$ was only 0.16%.

The multilevel CC-OTEC model however did not take in account the effect of adding heat exchangers in series on the seawater side pressure drop. Moreover, from the economic point of view adding an evaporation level could result in an increase in plant cost because of added complexity of the plant and the larger number of components required. Even if an increase in net power produced was observed, adding an evaporation level requires a more detailed analysis from the economic point of view.

Since the single level CC-OTEC is the simplest configuration and was used in the pilot plants
developed so far, a detailed techno-economic optimization of a single level closed cycle was conducted by adding an approximate model of a plate evaporator and a plate condenser in the power plant numerical model and by modifying the optimization tool adding the plate geometry of both condenser and evaporator as optimization variables. The $\gamma$ parameter found by the new optimization tool was 0.184kW/m$^2$ and net power produced was 2.6MWe. Optimal geometries obtained with the new optimization tool were used to size titanium heat exchangers. The heat exchangers specific cost per unit surface was estimated to be 869€/m$^2$. Other plant costs have been estimated by scaling values found in literature. The estimated plant cost was 37.5M€ and the specific cost was 14423€/kWe. It resulted that the largest part of plant components cost as expected are the cold water pipe and heat exchangers. Titanium heat exchangers effectively resulted to be the 36% of plant costs according to estimates found in literature. In the plant cost estimation, engineering costs were 28% of total capital costs. The engineering cost is therefore an important part of plant capital cost. In fact, if the plant modeled would be effectively constructed, it will be the first plant of his size and the engineering procedure requires to be customized.

An LCOE comparison between the closed cycle OTEC plant and other technologies for power generation has been fulfilled. It was assumed that the plant operated for 8000h/yr at the net power of 2.6MWe. If it is assumed to neglect the effect of seasonal variations on the warm seawater surface temperature this assumption was found reasonable. The resulting LCOE considering a plant life of thirty years and an annual cost for O&M equal to 3.3%, was 240€/MWhel, in range with values found in literature. It appears that the OTEC plant is nowadays not competitive with other power generation technologies. However, with the validation of new materials for CWP, since cold seawater flowrate is directly proportional to net power produced, plants of higher size can be developed and economies of scale can be exploited. If aluminum heat exchangers resistant to pitting corrosion will be developed an important reduction in costs will be observed. Moreover, if a large number of plants will be constructed effectively the engineering procedure for plant design will be standardized, reducing engineering costs required for plant development. If this reduction of cost will effectively happen in future, the closed cycle OTEC will become economically feasible, in particular for small islands where electricity prices are high. As an example, by comparing the Hawaii electricity price in June 2016 which was 214€/MWhel with the estimated LCOE for the OTEC plant, economical feasibility of CC-OTEC plants seems not so far.
Conclusions and future developments

During the realization of this thesis work, several interesting issues for further investigations emerged:

- An optimization tool which accounts the effect of heat exchangers geometry as the one developed for the single level CC-OTEC can be developed for the multilevel configuration in order to estimate specific cost of a plant with more than one evaporation level;
- Working fluids different than pure fluids evaluated in this work, such as zeotropic mixtures, can be used as working fluid in the OTEC plant closed cycle. The temperature glide observed during phase change in these mixtures can improve efficiency by reducing exergy destruction in the heat transfer process and can have some advantages in off-design conditions of heat exchangers because of biofouling;
- In order to enhance economic feasibility of the plant, the effect of desalinated water production and sale could be observed. This solution is of particular interest in case of small islands were large size OTEC plants are not required. The economic return of selling seawater can overcome the disadvantage of not exploiting economies of scale;
- In order to assess power plant behavior with variable seawater inlet temperature conditions due to seasonal variations and for a better estimation of electrical production during the year, it would be interesting to develop an off-design simulation model of the plant.
References


References


References


https://www.aspentech.com/.


[54] M. Astolfi and E. Macchi, "Efficiency correlations for axial flow turbines working with non-conventional fluids".


References


## Abbreviation index

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>ANL</td>
<td>Argonne National Laboratory</td>
</tr>
<tr>
<td>CC-OTEC</td>
<td>Closed cycle OTEC</td>
</tr>
<tr>
<td>CWP</td>
<td>Cold water pipe</td>
</tr>
<tr>
<td>DOW</td>
<td>Deep Ocean water</td>
</tr>
<tr>
<td>EEZ</td>
<td>exclusive economic zone</td>
</tr>
<tr>
<td>FCR</td>
<td>Fixed charge ratio</td>
</tr>
<tr>
<td>FRP</td>
<td>Fiber reinforced plastic</td>
</tr>
<tr>
<td>GOSEA</td>
<td>Global Ocean reSource and Energy Association</td>
</tr>
<tr>
<td>GWP</td>
<td>Global Warming Potential</td>
</tr>
<tr>
<td>HDPE</td>
<td>High density polyethylene</td>
</tr>
<tr>
<td>LCOE</td>
<td>Levelized cost of energy</td>
</tr>
<tr>
<td>MM</td>
<td>Siloxane</td>
</tr>
<tr>
<td>NEMO</td>
<td>New Energy for Martinique and Overseas</td>
</tr>
<tr>
<td>O&amp;M</td>
<td>Operation and maintenance</td>
</tr>
<tr>
<td>OC-OTEC</td>
<td>Open cycle OTEC</td>
</tr>
<tr>
<td>ORC</td>
<td>Organic Rankine cycle</td>
</tr>
<tr>
<td>OTEC</td>
<td>Ocean thermal energy conversion</td>
</tr>
<tr>
<td>PON</td>
<td>Program opportunity notice</td>
</tr>
<tr>
<td>R&amp;D</td>
<td>Research and Development</td>
</tr>
<tr>
<td>SOTEC</td>
<td>Solar Ocean Thermal Energy Conversion</td>
</tr>
<tr>
<td>US DOE</td>
<td>US Department of Energy</td>
</tr>
</tbody>
</table>
# List of symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{amb}$</td>
<td>Ambient temperature</td>
</tr>
<tr>
<td>$R''_{biofouling}$</td>
<td>Biofouling thermal resistance</td>
</tr>
<tr>
<td>$b$</td>
<td>Channel spacing</td>
</tr>
<tr>
<td>$L_{CWP,WWP}$</td>
<td>Cold or warm water pipe length</td>
</tr>
<tr>
<td>$\dot{m}_{c,sw}$</td>
<td>Cold seawater mass flowrate</td>
</tr>
<tr>
<td>$T_{c,sw}$</td>
<td>Cold seawater temperature</td>
</tr>
<tr>
<td>$\Delta T_{c,sw}$</td>
<td>Cold seawater temperature difference</td>
</tr>
<tr>
<td>$\Delta T_{w,sw}$</td>
<td>Cold seawater temperature difference</td>
</tr>
<tr>
<td>$R''_{corr}$</td>
<td>Corrosion film thermal resistance</td>
</tr>
<tr>
<td>$T_{ds}$</td>
<td>Dead state temperature</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Efficiency</td>
</tr>
<tr>
<td>$W_{el}$</td>
<td>Electric power</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Enlargement factor</td>
</tr>
<tr>
<td>$h_1$</td>
<td>Enthalpy (number on subscript)</td>
</tr>
<tr>
<td>$f$</td>
<td>Friction factor</td>
</tr>
<tr>
<td>$W_{gross}$</td>
<td>Gross power produced by the closed cycle</td>
</tr>
<tr>
<td>$D_h$</td>
<td>Hydraulic diameter</td>
</tr>
<tr>
<td>$L_{id}$</td>
<td>Ideal specific work</td>
</tr>
<tr>
<td>$k_{liq}$</td>
<td>Liquid conductivity</td>
</tr>
<tr>
<td>$\mu_{liq}$</td>
<td>Liquid viscosity</td>
</tr>
<tr>
<td>$T_{ml}$</td>
<td>Mean log temperature</td>
</tr>
<tr>
<td>$\Delta T_{ml}$</td>
<td>Mean log temperature difference</td>
</tr>
<tr>
<td>$W_{el,net}$</td>
<td>Net electric power</td>
</tr>
<tr>
<td>$L_{net}$</td>
<td>Net specific work</td>
</tr>
<tr>
<td>$N_{ch}$</td>
<td>Number of Channels</td>
</tr>
<tr>
<td>$N_{plates}$</td>
<td>Number of plates</td>
</tr>
<tr>
<td>$N_u$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$U$</td>
<td>Overall heat transfer coefficient</td>
</tr>
<tr>
<td>$\Delta T_{pp}$</td>
<td>Pinch point temperature difference</td>
</tr>
<tr>
<td>$A_{plate}$</td>
<td>Plate area</td>
</tr>
<tr>
<td>$L_p$</td>
<td>Plate length</td>
</tr>
<tr>
<td>$p$</td>
<td>Plate pitch</td>
</tr>
<tr>
<td>$t$</td>
<td>Plate thickness</td>
</tr>
<tr>
<td>$L_w$</td>
<td>Plate width</td>
</tr>
<tr>
<td>$D_p$</td>
<td>Port diameter</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>Pressure drop</td>
</tr>
<tr>
<td>$\Delta H$</td>
<td>Pressure head</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Ratio between convective heat transfer coefficients</td>
</tr>
</tbody>
</table>
List of symbols

\[ \begin{align*}
 p_{\text{red}} & \quad \text{Reduced pressure} \\
 h_{\text{ref}} & \quad \text{Reference convective heat transfer coefficient} \\
 L_{\text{rev}} & \quad \text{Reversible specific work} \\
 Re & \quad \text{Reynolds number} \\
 h_{\text{sw}} & \quad \text{Seawater convective heat transfer coefficient} \\
 \rho & \quad \text{Seawater density} \\
 W_{\text{sw,pump}} & \quad \text{Seawater pumping power} \\
 k_p & \quad \text{Seawater pumps proportionality constant} \\
 \text{SP} & \quad \text{Size parameter} \\
 c_p & \quad \text{Specific heat} \\
 \Delta T & \quad \text{Temperature difference} \\
 \Delta T_{\text{glide}} & \quad \text{Temperature glide of a mixture} \\
 TTD & \quad \text{Terminal temperature difference} \\
 \dot{Q} & \quad \text{Thermal power} \\
 v_{\text{CWp,WWp}} & \quad \text{Velocity in cold or warm water pipe} \\
 V_r & \quad \text{Volume ratio} \\
 R_{\text{wall}}^w & \quad \text{Wall conductive resistance} \\
 m_{\text{w,sw}} & \quad \text{Warm seawater mass flowrate} \\
 T_{\text{w,sw}} & \quad \text{Warm seawater temperature} \\
 h_{\text{wf}} & \quad \text{Working fluid heat transfer coefficient} \\
 \dot{m}_{\text{wf}} & \quad \text{Working fluid mass flowrate} \\
 \gamma & \quad \text{Net power/Heat exchanger area}
\end{align*} \]