DESIGN, FABRICATION, INSTALLATION, AND
ANALYSIS OF A CLOSED CYCLE DEMONSTRATION
OTEC PLANT

by

Mohammed Faizal

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requirements for the degree of
Master of Science in Engineering

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School of Engineering and Physics
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The University of the South Pacific

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Declaration of Originality

Statement by Author

I, Mohammed Faizal, hereby declare that the write up of this dissertation is purely my own work without the inclusion of any other research materials that has already been published or written. Any individuals' work or idea that has been included within the report has been clearly referenced and credit given to the person.

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16/05/2012

Statement by Supervisor

I hereby confirm that the work contained in this supervised research project is the work of Mohammed Faizal unless otherwise stated.

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Publications


Abstract

Ocean water covers a vast portion of the earth’s surface and is also the world’s largest solar energy collector. It plays an important role in maintaining the global energy balance as well as in preventing the earth’s surface from continually heating up due to solar radiation. The ocean also plays an important role in driving the atmospheric processes. The heat exchange processes across the ocean surface are represented in an ocean thermal energy budget, which is important because the ocean stores and releases thermal energy. The solar energy absorbed by the ocean heats up the surface water, despite the loss of heat energy from the surface due to back-radiation, evaporation, conduction and convection, and the seasonal change in the surface water temperature is less in the tropics. The cold water from the higher latitudes is carried by ocean currents along the ocean bottom from the poles towards the equator, displacing the lower density water above and creating a thermal structure with a large reservoir of warm water at the ocean surface and a large reservoir of cold water at the bottom, with a temperature difference of 22°C to 25°C between them. The available thermal energy, which is the almost constant temperature water at the beginning and end of the thermocline, in some areas of the oceans, is suitable to drive ocean thermal energy conversion (OTEC) plants. These plants are basically heat engines that use the temperature difference of the surface and deep ocean water to drive turbines to generate electricity. An overview of the heat energy budget of the ocean is presented taking into consideration all the major heat inputs and outputs. The theoretical analysis of the closed cycle OTEC system is also presented.

Experimental studies were performed on a corrugated plate heat exchanger for small temperature difference applications. Experiments were performed on a single corrugation pattern on twenty plates arranged parallelly, with a total heat transfer area of 1.16298 m². The spacing, ΔX, between the plates was varied (ΔX = 6 mm, 9 mm, and 12 mm) to experimentally determine the configuration that gives the optimum heat transfer. Water was used on both the hot and the cold channels with the flow being parallel and entering the heat exchanger from the bottom. The hot water flowrates were varied. The cold side flowrate and the hot and cold water inlet temperatures were kept constant. It is found that for a given ΔX, the average heat transfer between the two liquids increases with increasing hot water flowrates. The corrugations on the plates enhance turbulence at higher velocities, which improves
the heat transfer. The optimum heat transfer between the two streams is obtained for the minimum spacing of \( \Delta x = 6 \) mm. The pressure losses are found to increase with increasing flowrates. The overall heat transfer coefficients, \( U \), the temperature difference of the two stream at outlet, and the thermal length are also presented for varying hot water flowrates and \( \Delta x \). The findings from this work would enhance the current knowledge in plate heat exchangers for small temperature difference applications and also help in the validation of CFD codes.

A closed cycle demonstration OTEC plant was designed, fabricated, and installed in the Thermo-fluids Lab, The University of the South Pacific. An experimental study was carried out on the demonstration plant with the help of temperature and pressure readings before and after each component. An increase in the warm water temperature increases the heat transfer between the warm water and the working fluid, thus increasing the working fluid temperature, pressure, and enthalpy before the turbine. The performance is better at larger flowrates of the working fluid and the warm water. It is found that the thermal efficiency and the power output of the system both increases with increasing operating temperature difference (difference of warm and cold water inlet temperature). The performance of the system improves with increasing pressure drop across the turbine. Increasing turbine inlet temperatures also increase the efficiency and the work done by the turbine. A maximum efficiency of about 1.5 % was achieved in the system.
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>total heat transfer area, m$^2$</td>
</tr>
<tr>
<td>$A_C$</td>
<td>heat transfer area of condenser, m$^2$</td>
</tr>
<tr>
<td>$A_E$</td>
<td>heat transfer area of evaporator, m$^2$</td>
</tr>
<tr>
<td>$b$</td>
<td>plate spacing, m</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat of air (or water) at constant pressure, kJ/kg.ºC</td>
</tr>
<tr>
<td>$C_{pCW}$</td>
<td>specific heat at average cold water temperature, kJ/kg.ºC</td>
</tr>
<tr>
<td>$C_{pHW}$</td>
<td>specific heat at average hot water temperature, kJ/kg.ºC</td>
</tr>
<tr>
<td>$g$</td>
<td>gravitational acceleration, m/s$^2$</td>
</tr>
<tr>
<td>$h$</td>
<td>specific enthalpies, kJ/kg</td>
</tr>
<tr>
<td>$h_{isen}$</td>
<td>isentropic specific enthalpies, kJ/kg</td>
</tr>
<tr>
<td>$m_{CS}$</td>
<td>mass flowrate of cold seawater, kg/s</td>
</tr>
<tr>
<td>$m_{WF}$</td>
<td>mass flowrate of working fluid, kg/s</td>
</tr>
<tr>
<td>$m_{WS}$</td>
<td>mass flowrate of warm seawater, kg/s</td>
</tr>
<tr>
<td>$N$</td>
<td>precipitation, cm/year</td>
</tr>
<tr>
<td>$P$</td>
<td>operating pressures, Pa</td>
</tr>
<tr>
<td>$\dot{Q}_b$</td>
<td>rate of heat loss from the ocean by back radiation, W/m$^2$</td>
</tr>
<tr>
<td>$\dot{Q}_C$</td>
<td>heat transferred in the OTEC condenser, W</td>
</tr>
<tr>
<td>$\dot{Q}_E$</td>
<td>heat transferred in the OTEC evaporator, W</td>
</tr>
<tr>
<td>$\dot{Q}_e$</td>
<td>rate of heat loss by evaporation from the ocean surface, W/m$^2$</td>
</tr>
<tr>
<td>$\dot{Q}_h$</td>
<td>rate of sensible heat loss from ocean surface by convection and conduction, W/m$^2$</td>
</tr>
<tr>
<td>$\dot{Q}_S$</td>
<td>rate of heat added to ocean by short-wave solar radiation, W/m$^2$</td>
</tr>
<tr>
<td>$\dot{Q}_T$</td>
<td>total rate of heat gain or loss by a given area of the ocean, W/m$^2$</td>
</tr>
<tr>
<td>$\dot{Q}_v$</td>
<td>heat transport by moving currents (advection) within the ocean, W/m$^2$</td>
</tr>
<tr>
<td>$\dot{Q}_{CW}$</td>
<td>heat transferred by cold water in the heat exchanger, W</td>
</tr>
<tr>
<td>$\dot{Q}_{HW}$</td>
<td>heat transferred by hot water in the heat exchanger, W</td>
</tr>
<tr>
<td>$\dot{Q}_{Average}$</td>
<td>average heat transfer between hot and cold water in the heat exchanger, W</td>
</tr>
<tr>
<td>$S$</td>
<td>salinity, parts per thousand ($\text{‰}$)</td>
</tr>
</tbody>
</table>
\( T_s \) ocean surface temperature, °C
\( T_{WSI} \) warm seawater temperature at inlet of evaporator, °C
\( T_{WSO} \) warm seawater temperature at outlet of evaporator, °C
\( T_{CSI} \) cold seawater temperature at inlet of condenser, °C
\( T_{CSO} \) cold seawater at outlet of condenser, °C
\( T_{CWI} \) cold water temperature at inlet of heat exchanger, °C
\( T_{CWO} \) cold water temperature at outlet of heat exchanger, °C
\( T_{HWI} \) hot water temperature at inlet of heat exchanger, °C
\( T_{HWO} \) hot water temperature at outlet of heat exchanger, °C
\( U \) overall heat transfer coefficient, W/m².K
\( U_C \) overall heat transfer coefficient of condenser, W/m².K
\( U_E \) overall heat transfer coefficient of evaporator, W/m².K
\( \dot{V}_{CS} \) cold seawater flowrate, L/s
\( \dot{V}_{CW} \) cold water flowrate in corrugated plate heat exchanger, L/s
\( \dot{V}_{HW} \) hot water flowrate in corrugated plate heat exchanger, L/s
\( \dot{V}_{WF} \) working fluid flowrate, L/s
\( \dot{V}_{WS} \) warm seawater flowrate, L/s
\( V \) evaporation, cm/year
\( v_f \) specific volume of liquid working fluid, m³/kg
\( w \) amplitude or channel height, m
\( W_G \) generator power of OTEC plant, W
\( W_N \) net power of OTEC plant, W
\( W_{CSP} \) power required by cold seawater pump, W
\( W_{WSP} \) power required by warm seawater pump, W
\( W_{WFP} \) power required by working fluid pump, W
\( \Delta P_H \) pressure loss of hot water in the heat exchanger, kPa
\( \Delta T_{CW} \) temperature change of cold water in the heat exchanger, °C
\( \Delta T_{HW} \) temperature change of hot water in the heat exchanger, °C
\( \Delta T_m \) log mean temperature difference (LMTD) of the heat exchanger, °C
\( \Delta T_{outlet} \) temperature difference of hot and cold water measured at outlet of heat exchanger, °C
\( \Delta X \)  
plate spacing of the heat exchanger, mm

\( \rho_{CW} \)  
water density at average cold water temperature in heat exchanger, kg/m\(^3\)

\( \rho_{HW} \)  
water density at average hot water temperature in the heat exchanger, kg/m\(^3\)

\( \theta_{CW} \)  
thermal length of the cold water channels of the heat exchanger

\( \theta_{HW} \)  
thermal length of the hot water channels of the heat exchanger

\( \theta_{Average} \)  
average thermal length in the heat exchanger

\( \eta_{G} \)  
exticiency of generator

\( \eta_{T} \)  
exticiency of turbine

\( \eta_{CSP} \)  
exticiency of cold seawater pump

\( \eta_{WFP} \)  
exticiency of working fluid pump

\( \eta_{WSP} \)  
exticiency of warm seawater pump

\( \Delta h_{CSP} \)  
total head loss across cold water piping, m

\( \Delta h_{WSP} \)  
total head loss across warm water piping, m

\( (\Delta h_{CS})_{c} \)  
head loss in the condenser, m

\( (\Delta h_{WS})_{E} \)  
head loss in the evaporator, m

\( (\Delta h_{CS})_{d} \)  
head loss due to density differences in cold water pipe, m

\( (\Delta h_{CS})_{M} \)  
minor head loss in the cold water pipe, m

\( (\Delta h_{WS})_{M} \)  
minor head loss in the warm water pipe due to bends, m

\( (\Delta h_{CS})_{SP} \)  
frictional head loss in straight cold water pipe, m

\( (\Delta h_{WS})_{SP} \)  
frictional head loss in straight warm water pipe, m

\( (\Delta T_{m})_{C} \)  
log mean temperature difference of condenser, °C

\( (\Delta T_{m})_{E} \)  
log mean temperature difference of evaporator, °C

\( \rho \)  
seawater density, kg/m\(^3\)

\( A \)  
wavelength or pitch of corrugated plate, m

\( \rho \)  
seawater density, kg/m\(^3\)
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1.0 Introduction

1.1. Overview
Ocean Thermal Energy Conversion (OTEC) technology utilizes the temperature difference of warm surface water and deep cold water of the ocean to generate electricity. An OTEC power plant acts as a ‘heat engine’ that extracts heat energy from the warm surface water, converts part of that energy to generate electricity through a turbine, and rejects the remaining heat energy to the cold deep sea water in a cyclic process. The temperature of the ocean waters generally decreases with increasing depth, except for polar regions. This region of rapidly changing temperature is known as the thermocline. It is this region that separates the upper mixed layer of the ocean with deep ocean water. The thermocline is the deepest in the tropics and shallowest in the polar regions. Below the thermocline, is a region of deep cold ocean water where the temperature reaches an almost isothermal condition. The surface water thus acts as a large reservoir of warm water and the deep water (approximately at 1000 m) acts as a large reservoir of cold water in the tropical oceans throughout the year. This uniform temperature difference can be used to operate OTEC plants.

Ocean Thermal Energy Conversion plants are most suitable in tropical regions because of less variation in ambient temperature throughout the year around. Regions closer to the equator have maximum potential for OTEC systems. In tropical countries, sunlight is abundant in supply and most of the solar energy gets absorbed by the oceans. This thermal energy available in the oceans can be utilized to reduce global warming and its consequences. Research in Renewable Energy technologies creates pathways to reduce the reliance on imported fossil fuels. The pacific island countries have excellent temperature difference of surface and deep water (at approximately 1000 m) of the ocean, making this region a better place for OTEC power generation. The major advantages of OTEC power plants are that they provide a consistent power output almost throughout the year. Ocean Thermal Energy Conversion technology is environmental friendly and does not directly contribute to global warming and depletion of natural resources. It also gives a lot of useful by-products. A sea-water desalination plant can be integrated into an OTEC power plant to obtain fresh water. However, it should be noted that this field is still
under development and a lot of research still remains to be done to develop power from OTEC economically.

The current project focused on manufacturing and experimentation of a closed cycle demonstration OTEC and performing experimental studies on a corrugated plate heat exchanger. The OTEC plant was designed, fabricated, and installed in the Thermo-fluids Laboratory, The University of the South Pacific (USP).

1.2. Thesis Objectives

✓ To give an overview on the types of OTEC systems, their operational concepts, the individual components, and overall performance parameters.
✓ To perform a theoretical analysis of the closed cycle OTEC system.
✓ To perform experimental studies on corrugated plate heat exchangers for small temperature difference applications.
✓ To fabricate and install a closed cycle OTEC demonstration plant in the Thermo-fluids Lab, USP.
✓ To experimentally determine the performance of the demonstration OTEC plant under various operational conditions.

1.3. Thesis Outline

Chapter 1 gives a general introduction of Ocean Thermal Energy Conversion (OTEC) operating principles. The reasons as to why OTEC plants are suitable for Pacific Island Countries are also briefly described. The objectives of this research are also listed.

Chapter 2 gives an overview of the ocean heat budget and the different types of OTEC plants and its operational principles, the thermal structure of the oceans, the feasibility and technical limitations of OTEC plants. It also provides a detailed literature review of corrugated plate heat exchangers.

In chapter 3, a detailed theoretical analysis of the closed cycle OTEC system is presented.

In chapter 4, the system component designs, fabrication, and experimental setups are described. The corrugated plate heat exchangers and the final closed cycle OTEC demonstration plant details are given.
Chapter 5 presents the experimental results and analysis of the corrugated plate heat exchangers and the closed cycle OTEC demonstration plant.

Chapter 6 summarizes the main findings from this research.
2.0 Literature Review

The Earth’s surface is approximately covered by seventy percent of water. Ocean water makes up 97.4% of the total water available [1]. The global-ocean can be classified as a continuous body of water that separates into several major oceans and seas [2]. The major ocean divisions, according to their size, are the Pacific Ocean, Atlantic Ocean, Indian Ocean, Southern Ocean, and the Arctic Ocean [2,3]. The average temperatures of the ocean waters hardly exceed 30ºC or reduce below -2ºC [4]. It is the water in the oceans that prevents wide variations of temperature on the Earth’s surface globally [5].

The amount of heat energy required to raise the temperature of a given mass of water by 1ºC is more than that of other fluids [6]. Moreover, the ocean has the largest heat capacity compared to any single component of the climate system [7]. This property of water allows a lot of solar energy to be stored in the oceans, thus preventing the Earth’s surface from heating up [5]. The major source of thermal energy entering the ocean is from the Sun. The ocean plays an important role in maintaining the global energy balance of the Earth’s atmosphere. The ocean stores thermal energy to a much greater extent than land because of its high heat capacity [8]. The ocean can absorb heat in one region and restore it in a different place, even after decades or centuries [9]. The amount of thermal energy entering the ocean must be equal to the thermal energy leaving or the average temperature of the ocean will change [10]. Significant heat exchange processes across the ocean surface are represented in an ocean energy budget [11]. The ocean energy budget is important because the ocean stores and releases much more heat than the land over different seasons [12], thus preventing the Earth from heating up. Figure 1 shows a schematic diagram of the heat transfer processes from a given area of the ocean [13]. The rate of heat gain or loss, \( \dot{Q}_T \), by a given vertical column of ocean water with a unit horizontal cross sectional area [13] can be expressed as the difference of the total heat coming from the Sun and the total thermal energy loss from the given area. The rate of heat absorbed by the ocean from incoming solar radiation is \( \dot{Q}_s \), the rate of heat loss by back radiation is \( \dot{Q}_b \), sensible heat loss by convection and conduction is \( \dot{Q}_h \), rate of heat loss (latent heat) by evaporation from the ocean surface is \( \dot{Q}_e \), and
\( \dot{Q}_v \) is the thermal energy transported by ocean currents moving out of the given area [4,14,15]. The heat and thermal energy interactions mentioned in this dissertation are all rates of such interactions.

\[ \dot{Q}_T = \dot{Q}_s - \dot{Q}_b - \dot{Q}_h - \dot{Q}_e - \dot{Q}_v \]  

Figure 1. Schematic diagram of heat transfer processes from a given area of the ocean [13].

The heat transfer terms in Figure 1 can be represented by an equation according to the conservation of energy principle [4,13,14]:

The \( \dot{Q}_h \), \( \dot{Q}_b \), and \( \dot{Q}_v \) terms in equation 1 could be either positive or negative depending on whether thermal energy is gained or lost by the given area of the ocean [15,16]. The term in equation 1 that transfers thermal energy from one region of the ocean to another is \( \dot{Q}_v \), stating the effects of ocean currents [16]. However, for the ocean as a whole, \( \dot{Q}_v \) is taken as zero because it only accounts for the redistribution of thermal energy within the ocean [4,16]. There is a net gain of thermal energy throughout the year in the lower latitudes (positive \( \dot{Q}_T \)), but a net gain in summer (positive \( \dot{Q}_T \)) and a net loss (negative \( \dot{Q}_T \)) in winter in the higher latitudes [13,17].

The heat added to the ocean by short wave radiation is different at different latitudes and over different seasons, the maximum being at the equator. Heat lost by back
radiation from the surface of the ocean increases with decreasing altitudes of the Sun. The effective back radiation from the ocean surface is the difference of the outward radiation from the surface and the re-radiation (or down radiation) from the atmosphere. Heat lost by evaporation from the ocean surface is the largest contributing factor to the overall heat losses from the ocean. The evaporation is higher close to the equator and decreases with increasing latitudes. Heat lost by convection and conduction has seasonal and regional variations, and depends on the temperature difference of the ocean surface and the air close to the surface. A more detailed explanation of the heat budget terms are provided by Faizal et al.[18].

The thermal energy in the oceans is distributed around the globe by moving ocean currents [19]. The circulation of waters in the oceans helps to distribute the thermal energy in the lower latitudes to certain areas in higher latitudes, thus modifying climate conditions [20]. The equatorial regions, or the lower latitudes, receive much more heat from the Sun than the polar regions because of the different angles at which the sunlight strikes the Earth [5]. The major factors that drive the ocean currents are solar energy and the Earth’s rotation [21]. Solar energy that is directly absorbed by the ocean varies from region to region due to unequal heating of the Earth’s surface [4]. Ocean thermal energy conversion (OTEC) technologies can be used to extract the thermal energy in oceans.

2.1. Ocean Thermal Energy Conversion (OTEC)
Ocean thermal energy conversion (OTEC) is a technique that utilizes the temperature difference of warm surface water and deep cold water of the ocean to operate a low pressure turbine [22,23]. An OTEC power plant acts as a heat engine that extracts energy as heat from the warm surface water, converts part of that energy to generate electricity and rejects the remaining energy as heat to the cold deep sea water in a cyclic process [22,24]. It can be integrated with a desalination plant, commonly known as the hybrid cycle, to produce fresh water [25,26]. Ocean Thermal Energy Conversion plants are more suitable for low latitudes (tropical oceans) because the water temperature remains almost uniform throughout the year with few variations due to seasonal effects [23]. About 63% of the surface of the tropics between latitudes 30ºN and 30ºS is occupied by ocean water [27].

Solar energy that is absorbed by the tropical oceans maintains a relatively stable surface temperature of 26-28ºC to a depth of approximately 100 m. As the
depth increases, the temperature drops, and at depths close to 1000 m, the temperature is as low as 4°C. Below this depth, the temperature drops only a few degrees. The temperature difference of warm and cold waters is maintained throughout the year with very few variations in the tropics. From the view of a thermodynamicist, any temperature difference can be used to generate power [22]. An OTEC plant, which is similar to a heat engine governed by the first law of thermodynamics, is driven between the heat source and sink to produce work output [28], shown by a schematic diagram in Figure 2.

![Schematic diagram of an OTEC plant operating as a heat engine.](image)

The technology for OTEC was first proposed by Jacques d’Arsonval, in the year 1881 in France [29,30]. He proposed a closed cycle OTEC design that used ammonia as the working fluid [31]. However, it was his student, George Claude who built the first OTEC plant in Cuba in 1930 [32]. A low pressure turbine was used to generate 22 kW of electricity for a short while before the system got damaged [33]. Ocean thermal energy is a potential source of renewable energy and with proper designing, it could provide a source of clean renewable energy with constant power.
output with many other benefits such as pure drinking water, which can benefit many small islands and developing countries [34].

Ocean thermal energy conversion power systems are basically divided into three categories: open cycle, closed cycle, and hybrid systems. An open cycle OTEC system utilizes the warm surface water as the working fluid. The surface water is pumped into a chamber where a vacuum pump reduces the pressure to allow the water to boil at low temperature to produce steam. The steam drives a turbine coupled to a generator and then is condensed (using deep cold seawater pumped to the surface) to produce desalinated water [22,35]. A closed cycle OTEC system incorporates a working fluid, such as ammonia or ammonia/water mixture, operating between two heat exchangers in a closed cycle. A closed cycle utilizes the warm surface water to vaporize the working fluid in a heat exchanger (evaporator). The vaporized fluid drives a turbine coupled to a generator. The vapor is then condensed in a heat exchanger (condenser) using cold deep seawater pumped to the surface. The condensed working fluid is pumped back to the evaporator and the cycle is repeated. Major differences between the open and closed cycle systems are the sizes of ducts and turbines, and the surface area required by heat exchangers for effective heat transfer [22]. For a given OTEC system with a certain power output, a closed cycle system with ammonia as the working fluid requires a much smaller duct and turbine diameter compared to an open cycle system which has water as the working fluid [36]. The difference is attributed to the pressure difference across the turbine and the specific volume of the working fluids. The heat exchangers for closed cycle systems require large surface areas to minimize temperature losses and to maintain the heat transfer between the ocean water and the working fluid to obtain the required power output [22].

The hybrid system integrates the power cycle with desalination to produce electricity and desalinated water. Nearly 2.28 million liters of desalinated water can be obtained everyday for every MW of power generated by a hybrid OTEC system [37]. Electricity is generated in the closed cycle system circulating a working fluid and the warm and cold seawater discharges are passed through a vacuum chamber and condenser to produce fresh water [22]. The power that the pumps need to do work is supplied from the gross power output of the OTEC power generating system. The working fluids for either closed or hybrid cycles should be such that it is able to operate between the low temperatures and still give optimum efficiency. Mostly
Freon and ammonia are considered, whereas ammonia and water mixture are also accepted for use [38]. The use of mixtures instead of one component fluid improves the thermodynamic performance of power cycles [39]. Studies done by Kim et al. [40] suggests that working fluids can be selected based on the specific environment and working conditions without affecting the efficiency much. The OTEC cycles are basic Rankine cycles that operate between a heat source and sink to generate electricity [41,42] with efficiencies close to 3% [41]. To increase the thermal efficiency of the OTEC system, other kinds of energies such as solar energy, geothermal energy, industrial waste energy, and solar ponds can be introduced to increase the temperature difference [43-45].

A lot of research work has been carried out on OTEC since its discovery in 1881. The first ever OTEC plant that was successfully commissioned was in Hawaii in 1979. A 50 kW closed cycle floating demonstration plant was constructed offshore. Cold water at a temperature of 4.4 °C was drawn from a depth of 670 m. During actual operation of the plant, it was found that biofouling, effects of mixing the deep cold water with the warm surface water, and debris clogging did not have any negative effects on plant operation. The longest continuous operation was for 120 hours [46]. A 100 kW OTEC pilot plant was constructed on-land for demonstration purposes in the republic of Nauru in October 1981 by Japan. The system operated between the warm surface water and the cold heat source of 5-8°C at a depth of 500-700 m, with a temperature difference of 20°C [47]. The tests done were load response characteristics, turbine, and heat exchanger performance tests. The plant had operated by two shifts with one spare shift, and a continuous power generation record of ten days was achieved. The plant produced 31.5 kW of OTEC net power during continuous operation and was connected to the main power grid [47].

A land based open cycle OTEC experimental plant was installed in Hawaii in 1993. The turbine-generator was designed for an output of 210 kW for 26 °C warm surface water and 6 °C deep water temperature. The highest gross power achieved was 255 kWe with a corresponding net power of 103 kW and 0.4 L/s of desalinated water [25]. Saga University, Japan, is actively involved in OTEC and its byproduct studies. Experimental studies have been conducted on heat exchangers and on spray-flash evaporation desalination. Other studies done are on mineral water production using deep cold water, lithium extraction from seawater, hydrogen production, air-
conditioning and aquaculture applications using deep cold water, and using the deep cold water for food processing and medical (cosmetic) applications [48].

Uehara et al. [42] presented a conceptual design for an OTEC plant in the Philippines after taking extensive temperature readings to determine a suitable site. The ocean surface water had a temperature range of 25 to 29ºC throughout the year while the cold water remained between 4 to 8 ºC at a depth of 500 – 700 m. A total of 14 sites were suggested. A conceptual design for a 5 MW onland-type and a 25 MW floating-type were computed for. After doing cost estimates of the proposed systems, the construction of the 5 MW onland-type plant was suggested.

Uehara and Ikegami [49] performed an optimization study of a closed cycle OTEC system. They presented numerical results for a 100 MW OTEC plant with plate heat exchangers and ammonia as the working fluid. They concluded that the net power can reach up to 70.3% of the gross power of 100 MW for inlet warm water temperature of 26 ºC and inlet cold water temperature of 4 ºC. Yeh et al. [50] conducted a theoretical investigation on the effects of the temperature and flowrate of cold sea water on the net output of an OTEC plant. They found out that the maximum net output exists at a certain flowrate of the cold seawater. The output is higher for a larger ratio of warm to cold seawater flowrate. Uehara et al. [51] did a performance analysis of an integrated hybrid OTEC plant. The plant is a combination of a closed cycle OTEC plant and a spray flash desalination plant. The total heat transfer area of the heat exchangers per net power is used as an objective function. A numerical analysis was done for a 10 MW integrated hybrid plant. Straatman and Sark [45] proposed a new hybrid OTEC with an offshore solar pond to optimize costs of electricity. This proposed system would increase the OTEC efficiency from 3% to 12%. The addition of a floating offshore solar pond to an OTEC system increases the temperature difference in the power cycle.

Yamada et al. [52] did a performance simulation of a solar-boosted ocean thermal energy conversion plant, termed as SOTEC. The temperature of warm sea water used in the evaporator was increased by using a solar thermal collector. The simulation results showed that the proposed SOTEC plant can increase the overall efficiency of the OTEC system. Tong et al. [44] proposed a solar energy reheated power cycle to improve performance. They suggested that a solar collector introduced at the evaporator will greatly improve the temperature difference and thus the cycle performance. Also, it was found that without any additional loadings on the
heat exchangers, increasing the turbine inlet pressure will also improve the OTEC system performance. Ganic and Wu [53] analyzed the effect of three working fluids used in OTEC systems. The fluids studied were ammonia, propane, and Freon-114. Seven different combinations of shell-and-tube heat exchangers were considered and for each combination, a computer model of the OTEC system was used. The comparisons were made based on the total heat transfer area of the heat exchangers divided by the net power output of the plant. It was found that ammonia was the best fluid because of its relatively high thermal conductivity. Kim et al. [54] did a numerical analysis for the same conditions but with various working fluids for a closed system, a regeneration system, an open system, a Kalina system, and a hybrid system. They concluded that the regeneration system using R125 as the working fluid had better performance. They also found that using the condenser effluent of a nuclear power plant rather than ocean surface water increased the system efficiency by approximately 2%.

Moore and Martin [55] presented a general mathematical framework for the synthesis of OTEC power generating systems. They developed a systematic methodology which was demonstrated in an OTEC system with ammonia as the working fluid. The power generated was used to drive a PEM electrolyser for hydrogen production. Faizal and Ahmed [56] performed experimental studies on corrugated plate heat exchangers for small temperature applications. They varied the channel spacing. They found that the minimum channel spacing gave optimal heat transfer. However, there was no phase change involved in their experiments. Zhou et al. [57] have presented a techno-economic study on compact heat exchangers to choose an optimum heat exchanger with minimum pressure drop. They concluded that all compact heat exchangers are feasible from an energetic point of view. However, the performance differs because of the materials used. Experimental studies on heat exchangers for use in OTEC plants have also been conducted in Saga University, Japan [22]. Together with an appropriate pressure difference across the turbine, a high heat transfer rate between the working fluid and the ocean water in the heat exchangers is required for optimal power production in OTEC plants [22].

Even though the thermal resource is available to many countries, there are many factors that have to be considered before a particular country or location is selected for an OTEC plant installation. Some of them are: distance of the thermal resource from land; depth of the ocean bed; depth of the resource; size of the thermal
resource within the exclusive economic zone (EEZ); replenishment capability for both warm and cold water; ocean currents; waves; hurricanes; seabed conditions for mounting; seabed conditions for power cables of floating plants; current local power source; annual consumption; present cost per unit; local oil or coal production; scope for other renewables; aquaculture potential; potable water potential; and environmental impacts [58]. Apart from generating electricity and producing fresh water, OTEC plants can be utilized for other benefits such as production of fuels such as hydrogen, ammonia, methanol, providing air-conditioning for buildings, on-shore and near-shore mariculture, and extraction of minerals [28,59,60]. Pacific Island countries have a lot of potential for implementation of OTEC technologies because of the high ocean temperature gradient.

2.2. The Thermal Structure of the Ocean
The temperature of the ocean waters generally decreases with increasing depth, except for polar regions [6,61]. The surface layer of the oceans is usually referred to as the mixed layer, because the near-surface waters are well mixed by winds and waves and a nearly isothermal condition is maintained [4,22]. Below the mixed layer is a region of rapidly changing temperature known as the thermocline. It is this region that separates the upper mixed layer of the ocean with deep ocean water [62]. The characteristics of the thermocline vary with season, latitudes, environmental conditions and ocean currents. The thermocline is the deepest in the tropics and shallower in the polar regions [63]. Below the thermocline is a region of deep cold ocean water where the temperature reaches an almost isothermal condition [64]. The deep cold ocean water is transferred from the polar latitudes [21,22]. The surface water thus acts as a large reservoir of warm water and the deep water (approximately at 1000 m) acts as a large reservoir of cold water in the tropical oceans throughout the year [22]. This uniform temperature difference can be used to operate OTEC plants [65].

Below the ocean surface water, the water is usually divided into three zones based on the temperature structure of the ocean: an upper zone with a depth of approximately 50 to 200 m with temperatures similar to that of the surface, a zone below 200 m and extending up to 1000 m in which the temperature changes rapidly (this is the thermocline), and a zone below 1000 m in which the temperature changes are small [66]. The actual depth of the zones is difficult to determine because of the
minor irregularities in the temperature against depth profile. Figure 3 shows the temperature vs. depth profile at different latitudes.

\[ \text{Figure 3. Typical mean temperature vs. depth profiles of the open ocean at different latitudes [67].} \]

In low and middle latitudes, there is a permanent thermocline present at all the times whereas there is no permanent thermocline in polar waters [21]. For polar regions, the thermocline is shallow in spring and summer, deep in the autumn, and disappears in winter. In winter, the heat loss at the surface produces instability and the resulting convection mixes the water column to a greater depth, thus eliminating the thermocline. In the tropics, winter cooling is not strong enough to destroy the thermocline, and thus, the thermocline in the tropics is maintained throughout the year [68]. The temperature in the lower half of all the oceans is uniformly cold, with temperatures as low as 2.3°C [10]. The surface temperature of the oceans range from as high as 28 °C from the equator to -2 °C at high latitudes. The temperature is highest at low latitudes and decreases at higher latitudes [66]. Figure 4 shows the ocean surface temperature variation with latitudes.
Figure 4. Latitudinal variation of surface temperature, salinity, and density average for all oceans [66].

Figure 5. Comparison of the amount of radiation received at different latitudes [69].

In lower latitudes there is a radiation surplus, shown in Figure 5, which decreases with increasing latitude [70]. Different regions on the Earth’s surface that are equal in size receive different levels of solar radiation. The solar radiation
intensity is largest between 23.5 °N and 23.5 °S because the sunlight strikes the earth at almost right angle between these latitudes [69]. Higher latitudes receive less solar energy compared with the equator because of the decreasing angle at which the sunlight strikes the Earth’s surface [5]. Also, the sunlight has to travel a larger distance through the atmosphere at higher latitudes; thus, the atmosphere absorbs most of the solar radiation intensity before it reaches the Earth at higher latitudes [69]. It is the almost constant temperature at the beginning and end of the thermocline that can be used to drive OTEC plants. Above the thermocline, there is an almost constant source of heat and below the thermocline there is an almost constant heat sink [22].

The temperature of the ocean water can be described in two ways: in terms of in situ temperature and in terms of potential temperature [10]. In situ temperature is the observed temperature of a parcel of water at a certain depth, whereas potential temperature is defined as the temperature of a parcel of water at the sea surface if it is raised adiabatically from some depth in the ocean. Adiabatically raising the parcel of water means that it is raised in an insulated container so that there is no exchange of heat with its surroundings [71]. The water parcel, however, is not actually brought to the surface. The potential temperature is therefore always less than the in situ temperature [10,71–73].

Thermal energy in the oceans is distributed by three processes, advection, diffusion, and vertical mixing. All these processes do not change the energy content of the ocean. Vertical mixing redistributes thermal energy within a column of the ocean whereas advection and diffusion move it horizontally as well [74]. The strength of the vertical mixing depends on the wind speeds on the ocean surface [75]. In a vertical water column in the ocean, the yearly changes in heat content are more notable in the upper layers of the ocean than the lowermost layers [76]. A vertical column of the ocean gains thermal energy from the incoming solar radiation and loses it by back radiation and evaporation. The rate of sensible heat gain or loss depends on whether the sea is warmer or colder than the air close to the ocean surface [77]. The vertical heat transfer can be thought of as being caused by very slow large-scale vertical water motion and by faster vertical motion in small eddies. Upwelling and downwelling can be considered as large-scale water motion, where upwelling reduces the energy content of the column because it brings up cold water from the bottom of the ocean, and downwelling increases the heat content [76].
2.3 Technological Issues
The proper designs of OTEC systems include the consideration of leakage of piping systems that carry the working fluid in a closed cycle. A major disadvantage of OTEC systems is the high capital cost [29,77]. Extensive research has been done on the OTEC components, for example, heat exchangers should have compact designs with optimum heat transfer and low unit cost [78]. Experimental studies on heat exchangers for use in OTEC plants have also been conducted in Saga University, Japan [22]. Biofouling in the heat exchangers provides resistance to heat transfer, therefore affecting their performance [79]. Cleaning methods such as continual circulation of close fitting balls and by chemical additives to the water are used [79].

Another major design concern is the cold water pipes that transport cold water from the ocean depths to the surface. The cold water pipes that pump deep cold ocean water to the surface require a lot of pumping power which increases the costs [50]. Approximately 4 m$^3$/s of warm surface seawater and 2 m$^3$/s of deep cold seawater (ratio 2:1), for a temperature difference of 20 °C, are required for every MW of electricity generated [80]. The cold water pipes are subjected to forces such as drag by ocean currents, oscillation forces, stresses at the connections, forces due to harmonic motion of the platform, and the dead weight of the pipe itself. Also, problems will arise in installation due to difficulties in construction and transportation to deployment site due to its very large size. The choice of materials is also debatable [22,79, 81]. The successful installations of offshore oil drilling platforms have provided technical guidance that can be directly applicable to OTEC cold water pipe design [22].

2.4 Impacts of OTEC Plants
Ocean thermal energy conversion plants can be located across about 60 million square kilometers of tropical oceans, generally at latitudes within about 20 or 25 degrees of the equator. The vast resource of cold water is constantly supplied by the deep cold water that flows from the polar regions [22,82].

The ocean thermal gradient essential for OTEC plants operation is mostly found between latitudes 20°N and 20°S [83,84]. There are at least two separate markets for OTEC plants: (i) industrial nations and islands, (ii) smaller or less industrialized islands with modest needs for power as well as desalinated water [85].
Commercial OTEC plants should be located in a stable resource environment for efficient operation of the system [86]. The country’s population, economies, policies and energy demands should also be looked at. An energy analysis that involves the environment, economy, and services should be put together for an emergy evaluation (emergy with an ‘m’) to determine the cost benefits [87]. Since capital costs are very high for OTEC plants, the by-products of these plants, such as fresh water, should be considered in a financing strategy to help overcome the initial costs [85]. Nihous et al. [88] presented a financing strategy for small land-based OTEC plants. It is based on the cost effectiveness of some OTEC by-products. The main aim of the financing strategy presented is that the by-products would gradually payback the huge amount of capital cost required to build a small OTEC plant. Studies have been done by Srinivasan et al. [89] on the cost effectiveness of OTEC plants and they designed a new OTEC system by introducing a subsea condenser. When identifying locations for OTEC plants, the thermal gradient suitable to drive the plants should not be very far away from the shore. The OTEC piping systems are a major part of the initial capital cost of OTEC plants [82].

Ocean thermal energy conversion plants can be land based, shelf mounted on platforms, or floating types on deep water [90,91]. The plants installed on or near land do not require complicated mooring, long power cables, or high maintenance costs such as with open-ocean environments. They can be installed in sheltered areas to keep it safe from storms and heavy seas. Land based or near shore located OTEC plants can be operated in combination with industries for mariculture or for desalinated water [90,92]. A shelf mounted OTEC plant can be towed to a favorable site of about 100 m depth and fixed to the sea bottom. This is done to have closer access to the cold water resource. Shelf mounted plants have to withstand the open ocean environmental conditions and the power delivery is also a concern because of the long underwater cables required to reach land [90,92]. Floating OTEC plants are designed to operate offshore, and are preferred for large power capacity plants. Offshore plants are difficult to stabilize and to moor in deep water, and the cables attached to floating plants are more vulnerable to damages in the open ocean environment. External forces such as waves, wind, and ocean currents affect the stability of the plant [90,93].

Ocean thermal energy conversion plants will have an impact on the physical characteristics of the region the plants are deployed in [94]. These plants can be used
to help improve the environment by combining it with artificial coral reef ecosystems [95]. However, changes in the climate characteristics are also possible [90]. Ocean thermal energy conversion plants can alter the ocean surface energy balance by lowering the surface temperatures. The tropical ocean environment can be affected by OTEC implemented upwelling and increase in CO₂ production due to increased mixing rate between surface and deep ocean waters. The deep water temperature can increase and the albedo of the surface can also increase due to increased phytoplankton on the surface [22, 94]. Deep cold seawater used in OTEC plants contains a lot of dissolved inorganic nutrients such as phosphate, nitrate and silicate, which could be expected to promote blooms of photosynthetic organisms if the seawater is discharged and contained within the upper ocean or in coastal waters [95,96]. The rich nutrients in deep cold water will be discharged at the ocean surface which is poor in nutrients and is much warmer compared to deep ocean water. The resulting complications due to this forced nutrient mixing are not fully understood [97]. Alterations in climate and ocean surface conditions will be more significant when multiple OTEC plants operate in a region. Also, the water intake by OTEC plants at the ocean surface would induce circulation, which could affect the coastal circulation [22]. An experimental an analytical study conducted by Jirka et al. [98] on the mixing and recirculation of surrounding ocean waters of an OTEC plant shows that large discharge velocities and plant flowrates contribute a lot to recirculation.

2.5. Corrugated Plate Heat Exchangers

Heat exchangers are heat transfer devices that exchange thermal energy between two or more media. The heat transfer between the media is purely based on temperature difference, without the use of any external energy. Some of the applications of heat exchangers are in power production industries, chemical and food industries, electronics, waste heat recovery systems, manufacturing industries, and air-conditioning and refrigeration systems. There are basically two types of heat exchangers: a direct heat exchanger and an indirect heat exchanger. In a direct heat exchanger, the two media between which heat is exchanged are in direct contact, e.g. cooling towers. In an indirect heat exchanger, the two media between which heat is exchanged are separated by a wall [99,100]. A plate heat exchanger is an indirect heat exchanger. Plate heat exchangers comprise of a stack of corrugated or embossed
metal plates with inlet and outlet ports and seals to direct the flow in alternate channels. The flow channels are formed by adjacent plates [101]. As shown in Figure 6, the hot and cold fluids flow in alternate channels and the heat transfer takes place between adjacent channels [102]. The number and size of the plates are determined by the flowrates, the physical properties of the fluids, pressure drops, and heat transfer requirements [101,103]. There are also many flow patterns that can be achieved for plate heat exchangers [101].

Figure 6. Hot and cold fluid flow in alternate passages in plate heat exchangers.

In the analysis of heat exchangers, all the thermal resistances in the path of heat flow from one fluid to another are combined into a single resistance [104], and an overall heat transfer coefficient, $U$, of the heat exchanger is determined. The overall heat transfer coefficient is a measure of the resistance to heat flow from one medium to another [100]. Phase change processes in heat exchangers have very high $U$ values due to high thermal conductivities. Because of complex physical processes, it is not generally possible to predict accurate values of $U$. Therefore, empirical formulas and $U$ values are mostly derived from experimental data [105]. One of the requirements in ocean thermal energy conversion (OTEC) plants is effective heat transfer with minimum pressure loss for small temperature difference of the hot and cold fluids. Pressure losses in heat exchangers will affect the pumping power of the pumps in OTEC plants. Studies reported by Bellas et al. [106] and Uehara et al. [107] show that pressure drop increases significantly with flowrates.

Plate heat exchangers have many advantages compared to many other heat exchangers. Plate heat exchangers can be used for high-viscosity applications, because turbulence is induced at low velocities which leads to effective heat transfer.
They also have high thermal effectiveness, large heat transfer per unit volume, low weight, possibility of heat transfer between many streams, ease of maintenance and a compact design [103, 108].

Corrugations in plate heat exchangers improve the heat transfer rates by 20% - 30% by increasing the heat transfer area and by enhancing turbulence at low flowrates [105, 109]. The corrugated plates also improve the mechanical strength of the plates [102]. Many types of enhanced surface geometries are used on plate heat exchangers. The objective is to obtain high heat-transfer coefficients without correspondingly increased pressure-loss penalties [110]. Special channel shapes, such as the wavy channels, provide mixing due to secondary flows or boundary layer separation within the channel [101]. The corrugations or wavy fins induce secondary flows (Görtler vortices) which assist in heat transfer augmentation [111]. The performance of plate heat exchangers can be improved by modifying the boundary layer and by enlarging the surfaces [112].

Since wavy surfaces have noninterrupted walls in each flow channel, the chances of fouling and particulates being caught in the channels are less. The waveform in the flow direction disrupts the flow and induces very complex flows. Görtler vortices are formed as the fluid passes over the concave wavy surfaces which enhance heat transfer. In the low-turbulence regime (Re of about 6000 to 8000), the wall corrugations increase the heat transfer by about nearly three times compared with the smooth wall channel [101]. Therefore, wavy fins are often a better choice at higher Reynolds numbers. A basic form of a corrugated or wavy geometry is shown in Figure 7. As corrugation (or wave) height to wavelength ratio increases, the separation zones in the troughs increase in relative size, giving rise to disproportionately high pressure drop [111]. A variety of corrugated or wavy patterns are proposed for plate heat exchangers [101].

![Figure 7. Schematic geometry of corrugated surfaces (\(A\) is wavelength or pitch, \(b\) is plate spacing, and \(w\) is amplitude or channel height) [113].](image)
Several studies have been carried out on heat transfer enhancement using corrugated plate heat exchangers. Picon-Nuñez et al. [103] presented a methodology on the design of compact heat exchangers. A simple approach to surface selection of the heat exchangers is based on the volume performance index. Plain-fin (wavy configuration) and louvered fins were considered in their study. They presented the volume performance index at different Reynolds numbers. Taucher and Mayinger [112] carried out numerical and experimental studies on heat transfer enhancement in plate heat exchangers with rib-roughened surfaces, which are also wavy configurations. They tested for various configurations of the ribs: shape, width, height, groove angle, spacing, angle, and arrangement patterns. They found out that the ribs show their best effects in regions where they can induce turbulence. They generalized that turbulence promoters (ribs in this case) show best performance in the transition region from laminar to turbulent flow.

Ciofalo et al. [113] conducted studies of flow and heat transfer in corrugated – undulated plate heat exchanges for rotary regenerators. For a particular corrugation, they varied the angle between the main flow direction and the axes of the furrows of the corrugations. They presented the Nusselt number distributions, the friction coefficient, pressure drop and heat transfer characteristics, and numerically simulated results on the flow and thermal fields induced by the wavy configurations. Kanaris et al. [114] performed CFD studies on a plate heat exchanger comprising of corrugated walls with herringbone design. They visualized the complex swirling flow in the furrows of the corrugations, and the Nusselt number and the friction factor were compared with those of smooth plates. They reported that corrugations increase the heat transfer; however, the pressure losses also increase. Elshafei et al. [115] presented heat transfer and pressure drop results in corrugated channels. They discussed the effect of channel spacing and phase shift of the corrugations on the heat transfer and the pressure drop. They showed that corrugations enhance heat transfer but with accompanying pressure drops. The results from the experiments were compared with conventional parallel plate heat exchangers and they found that corrugations enhance the heat transfer significantly. They found that the friction factor is higher for higher values of channel spacing. They also concluded that the area goodness factor decreases with increasing spacing ratio. Sparrow et al. [116] also performed experimental studies on corrugated plates and variable spacings.
Stasiek et al. [117] investigated the flow and heat transfer in corrugated passages. An experimental and numerical study of flow and heat transfer was conducted for a crossed-corrugated geometry. The effects of corrugation angle, geometry, and Reynolds number were investigated. Mitsumori et al. [118, 119] compared the performance of a closed cycle ocean thermal energy conversion (OTEC) plant using plate-type heat exchangers and tube-type heat exchangers. The results of their studies show that plate-type heat exchangers have more advantages and that they can be more compact. Test results on plate heat exchangers done at Saga University, Japan are presented by Avery and Wu [105]. It was found that the overall heat transfer coefficients and the pressure losses generally increase as the water velocity is increased. The best configurations tested at Saga University increase the overall heat transfer by a factor of 4 in comparison with smooth plates. Lyytikäinen et al. [120] performed numerical studies for varying corrugation angles and corrugation lengths and found out that both heat transfer as well as pressure drop increase as the corrugation angle is increased. They stated that it is not easy to find a specific geometry that provides both a low pressure drop and a high heat transfer simultaneously.

From the previous research carried out on heat transfer enhancement, it is obvious that wavy corrugations for plate heat exchangers are an attractive option. On the basis of the above finding, the present work is aimed at experimentally studying the heat transfer characteristics (with pressure drops) for corrugated plate type heat exchangers for use in small temperature difference applications.
3.0 Theoretical Analysis of the Closed Cycle OTEC System

The analysis of the closed cycle OTEC system is presented in this section. The equations are obtained from references [22, 38, 39, 42–44, 49–51, 79, 121–123].

Figure 8 shows a schematic of a closed cycle OTEC system and its T-S diagram.

The net power, $\dot{W}_N$, of an OTEC plant is the net power of the thermal cycle minus the pumping power required by the working fluid pump, and the warm and cold water pumps [22], given as:

$$\dot{W}_N = \dot{W}_G - (\dot{W}_{WS} + \dot{W}_{CSP} + \dot{W}_{WFP})$$

where $\dot{W}_G$ is the power available at the generator, $\dot{W}_{WS}$ is the power required for pumping warm surface water, $\dot{W}_{CSP}$ is the power required for pumping deep cold seawater, and $\dot{W}_{WFP}$ is the working fluid pumping power.
a) Generator power, $\dot{W}_G$

Since the working fluid pump, the evaporator, the condenser, and the turbine are steady flow devices, the processes of the power cycle are analyzed as steady flow processes using the steady flow energy equation (SFEE):

$$h_i + \frac{v_i^2}{2} + gz_i + q_{i-2} = h_2 + \frac{v_2^2}{2} + gz_2 + w_{i-2}$$  \hspace{1cm} (3)

The kinetic and potential energies are negligible. The SFEE simplifies to:

$$w_{i-2,isen} = h_i - h_{2,isen}$$  \hspace{1cm} (4)

where $h_i$ and $h_{2,isen}$ are enthalpies of the working fluid at inlet and exit of an isentropic/ideal turbine. The generator power is thus given as:

$$\dot{W}_G = \dot{m}_{WF} \eta_T \eta_G (h_i - h_{2,isen})$$  \hspace{1cm} (5)

where $\dot{m}_{WF}$ is the mass flowrate of working fluid, $\eta_T$ is the turbine efficiency, and $\eta_G$ is the generator efficiency.

b) Condenser

i) Heat rejection from working fluid in the condenser

The kinetic and potential energies are negligible. There is no work done. The SFEE simplifies to:

$$-q_{2-3} = h_3 - h_2$$  \hspace{1cm} (6)

where $h_3$ is the enthalpy of the working fluid at the exit of the condenser. Thus, the heat rejection from the working fluid in the condenser is:

$$\dot{Q}_C = \dot{m}_{WF} (h_2 - h_3)$$  \hspace{1cm} (7)

ii) The heat gained by cold water in the condenser

$$\dot{Q}_C = \dot{m}_{CS} C_p (T_{cs0} - T_{cSi})$$  \hspace{1cm} (8)
where $\dot{m}_{CS}$ is the mass flowrate of the deep cold sea water, $C_p$ is the specific heat, $T_{cso}$ is the temperature of cold seawater at exit of condenser, and $T_{csi}$ is the cold sea water temperature at inlet of condenser.

iii) The heat transfer in the condenser based on the heat transfer coefficient and the log mean temperature difference is:

$$\dot{Q}_C = U_C A_C (\Delta T_m)_C$$  \hspace{1cm} (9)

where $U_C$ is the overall heat transfer coefficient of the condenser, $A_C$ is the heat transfer area of the condenser, and $(\Delta T_m)_C$ is the log mean temperature difference (LMTD) of the condenser.

The log mean temperature difference is calculated as:

$$(\Delta T_m)_C = \frac{(T_2 - T_{csi}) - (T_3 - T_{cso})}{\ln \left( \frac{T_2 - T_{csi}}{T_3 - T_{cso}} \right)}$$  \hspace{1cm} (10)

where $T_2$ and $T_3$ are temperatures of the working fluid at the inlet and outlet of the condenser.

c) Working fluid pump power, $W_{WFP}$

The kinetic and potential energies are negligible. Work is done on the pump, therefore negative work output. SFEE simplifies to:

$$-w_{3-4,ise} = h_3 - h_{4,isen}$$  \hspace{1cm} (11)

where $h_{4,isen}$ is the enthalpy of the working fluid at the exit of an isentropic/ideal pump. The working fluid pump power is thus calculated as:

$$\dot{W}_{WFP} = \frac{\dot{m}_{WFP} (h_{4,isen} - h_3)}{\eta_{WFP}}$$  \hspace{1cm} (12)

where $\eta_{WFP}$ is the working fluid pump efficiency. For real life analysis, the pump efficiency should include the efficiency of the electric motor that runs the pump. The shaft work for a steady flow device (pump) is:
\[ W = - \int_{3}^{4} v dp \]  

(13)  

The working fluid pumping power is also given as:

\[ \dot{W}_{WFP} = \frac{\dot{m}_{WF} v_f (P_4 - P_3)}{\eta_{WFP}} \]  

(14)  

where \( v_f \) is the specific volume of the working fluid, and \( P_3 \) and \( P_4 \) are operating pressures.

d) Evaporator
i) Heat absorption by the working fluid in the evaporator

The kinetic and potential energies are negligible. There is no work done. The SFEE simplifies to:

\[ q_{4-1} = h_4 - h_4 \]  

(15)  

The heat rejection from the working fluid in the evaporator is thus given as:

\[ \dot{Q}_E = \dot{m}_{WF} (h_4 - h_4) \]  

(16)  

ii) The heat loss by warm water in the evaporator

\[ \dot{Q}_E = \dot{m}_{WS} C_p (T_{ws} - T_{wso}) \]  

(17)  

where \( \dot{m}_{WS} \) is the mass flowrate of the warm surface sea water, \( C_p \) is the specific heat, \( T_{ws} \) is the temperature of warm seawater at inlet of evaporator, and \( T_{wso} \) is the warm sea water temperature at outlet of evaporator.

iii) The heat transfer in the evaporator based on the heat transfer coefficient and the log mean temperature difference is:
\[ \dot{Q}_E = U_E A_E (\Delta T_m)_E \]  

(18)

where \( U_E \) is the overall heat transfer coefficient of the evaporator, \( A_E \) is the heat transfer area of the evaporator, and \( (\Delta T_m)_E \) is the log mean temperature difference (LMTD) of the evaporator. The log mean temperature difference is calculated as:

\[ (\Delta T_m)_E = \frac{(T_{wsi} - T_1) - (T_{wso} - T_4)}{\ln\left(\frac{T_{wsi} - T_1}{T_{wso} - T_4}\right)} \]  

(19)

where \( T_1 \) and \( T_4 \) are temperatures of the working fluid at the inlet and outlet of the evaporator.

e) **Cold sea water pumping power, \( \dot{W}_{CSP} \)**

The cold seawater pumping power is given as:

\[ \dot{W}_{CSP} = \frac{\dot{m}_{CS} g \Delta h_{CSP}}{\eta_{CSP}} \]  

(20)

where \( \eta_{CSP} \) is the pump efficiency, \( g \) is the gravitational acceleration, and \( \Delta h_{CSP} \) is the total head loss in the cold water pipe. The total head loss across the cold water piping system is:

\[ \Delta h_{CSP} = (\Delta h_{CS})_{SP} + (\Delta h_{CS})_M + (\Delta h_{CS})_C + (\Delta h_{CS})_d \]  

(21)

where \( (\Delta h_{CS})_{SP} \) is the head loss due to friction in the straight cold water pipe, \( (\Delta h_{CS})_M \) is the minor head losses due to bends, \( (\Delta h_{CS})_C \) is head loss of cold water in the condenser, and \( (\Delta h_{CS})_d \) is the head loss due to density differences. The cold seawater pumping power is thus given as:

\[ \dot{W}_{CSP} = \frac{\dot{m}_{CS} g [(\Delta h_{CS})_{SP} + (\Delta h_{CS})_M + (\Delta h_{CS})_C + (\Delta h_{CS})_d]}{\eta_{CSP}} \]  

(22)

e) **Warm sea water pumping power, \( \dot{W}_{WSP} \)**

The warm surface seawater pumping power is given as:
\[ \dot{W}_{WSP} = \frac{\dot{m}_{WS} g \Delta h_{WSP}}{\eta_{WSP}} \]  

(23)

where \( \eta_{WSP} \) is the pump efficiency, \( g \) is the gravitational acceleration, and \( \Delta h_{WSP} \) is the total head loss in the warm water pipe. The total head loss across the warm water piping system is:

\[ \Delta h_{WSP} = (\Delta h_{WS})_{SP} + (\Delta h_{WS})_{M} + (\Delta h_{WS})_{E} \]  

(24)

where \((\Delta h_{WS})_{SP}\) is the frictional headloss in the straight warm water pipe, \((\Delta h_{WS})_{M}\) is the minor head losses in the pipe due to bends, and \((\Delta h_{WS})_{E}\) is the head loss of warm water in the evaporator. The warm seawater pumping power is thus given as:

\[ \dot{W}_{WSP} = \frac{\dot{m}_{WS} g [(\Delta h_{WS})_{SP} + (\Delta h_{WS})_{M} + (\Delta h_{WS})_{E}]}{\eta_{WSP}} \]  

(25)
4.0 Device Designs, Fabrication, and Experimentation

Experimental studies were conducted on a corrugated plate exchanger with varying channel spacing, and a closed cycle OTEC demonstration plant. All the fabrications and experiments were carried out in the thermo-fluids laboratory, the University of the South Pacific.

4.1. Corrugated Plate Heat Exchanger

The current design is chosen based on the enhancement of heat transfer characteristics due to the incorporation of wavy configurations in plate exchangers. The traditional geometry of the wavy configurations is retained to reduce the number of variables in the present work and to study the effect of the flow rate and plate spacing. The hot water flowrates, $\dot{V}_{HW}$, and the spacing between the plates, $\Delta X$, are varied while the corrugation pattern remain the same. The focus of the experiments is on the measurements of the temperatures of the two fluids at inlet and exit of the heat exchanger and then to determine which $\Delta X$ value gives optimum heat transfer. A detailed physical explanation of the flow and the enhanced turbulence by the corrugations in the channels is also presented.

Experiments were performed on a single corrugation pattern on twenty plates arranged parallelly. The spacing between the plates, $\Delta X$, was varied to experimentally determine the spacing that gives the optimum heat transfer. Water was used on both the hot and the cold channels with the flow being parallel. Both the hot and cold water entered the heat exchanger from the bottom. This allowed the water to fully fill the heat exchanger channels before exiting into the atmosphere, thus utilizing the full area of the plates for effective heat transfer and preventing the formation of hydraulic diameters. The flowrates, $\dot{V}_{HW}$, for the hot side were varied from 0.18L/s to 0.63 L/s, while the cold side flowrate, $\dot{V}_{CW}$, was kept constant at 0.16L/s. The inlet temperatures for both the hot and cold water were kept constant at 49 ºC and 26 ºC respectively. This gives a temperature difference of 23 ºC at the inlet of the heat exchanger. The plates used are corrugated galvanized sheets, with a thickness of 0.4 mm. The other geometric details of the plates and the heat exchanger are provided in Figure 9 and Table 1.
A steam generator is used to maintain a constant temperature of 49 °C in the hot water tank. The hot water is directed into the heat exchanger by a centrifugal pump with a rated capacity of 81 L/min at a total head of 21 m and driven by a 0.5 HP variable speed motor. The inlet temperature of the cold water is maintained at a constant temperature of 26 °C. CABAC T6201 digital thermometers, with a resolution of 0.1 °C and a temperature range of -50 °C to +250 °C, were mounted at the inlets and outlets of the heat exchangers. WIKA EN 837-1 pressure gauges, with an accuracy of 1%, pressure range of 0 – 100 kPa, and a temperature range of -20 °C to 60 °C, mounted at the inlet of the hot and cold water streams measure the gauge pressure at which the fluids enter the heat exchanger. Figure 10 shows a schematic of the experimental setup.
Figure 10. Schematic diagram of the experimental setup.

The fluids exit into the atmosphere from the heat exchanger. The flowrate of a particular stream of water is equally divided in all the channels, as shown in Figure 11. The pipes that carried water to and away from the heat exchanger had its ends equally divided. This is done to achieve similar velocities and pressure of water in their respective channels. There are a total of nine channels for hot water, \( N_H \), and ten channels for cold water, \( N_C \). Figure 12 shows the three fabricated heat exchangers.

The repeatability of the temperature measurements was within 4% and that of pressure measurements was within 2.4%. The accuracies of measurement or estimation of \( \rho, C_p, \dot{V} \) and temperatures were taken into consideration for estimating the uncertainty of \( \dot{Q} \), considering the fact that always the temperature change was used for estimating \( \dot{Q} \) (from which \( U \) was obtained directly). The maximum error in the estimation of \( \dot{Q} \) was found to be 3.3%.

Figure 11. A schematic diagram of the heat exchanger showing exit ports and the flow dividers used at inlet and exit (blue for cold water and red for hot water).
4.2. Closed Cycle Demonstration OTEC Plant
A closed cycle demonstration OTEC plant with refrigerant R134-a as the working fluid was built and experimented on. Figure 13 shows a schematic of the demonstration plant. Figure 14 shows the final set-up.

![Schematic diagram of the OTEC demonstration plant (P = pressure gauges, T = Temperature sensors).](image)

Figure 13. Schematic diagram of the OTEC demonstration plant (P = pressure gauges, T = Temperature sensors).
Copper tubes with a total length of 5 m and external diameter of 15.88 mm (wall thickness = 1.24 mm) are used in the system. Pressure and temperature gauges are placed before and after each component of the system. MINGZHU pressure gauges (model: MZ-B9028), with an accuracy of 1%, are used to record pressure changes. The high side gauge has a pressure range of 0 – 3447 kPa and the low side gauge has a pressure range of 0 – 1517 kPa. CABAC T6201 digital thermometers, with a resolution of 0.1 ºC and a temperature range of -50 ºC to +250 ºC are used to record the temperature. A storage tank with a capacity of 6 liters is placed just before the refrigerant pump to ensure that the pump receives a continuous supply of refrigerant and is not starved. A National Refrigeration Products LP22E refrigerant pump is used to circulate the working fluid (R134-a) in the system. The capacity of the pump is 0.15 kg/s with a power rating of 372.8 W. A voltage regulator is used to vary the pump rpm to regulate the working fluid flowrate. A GPI commercial grade flowmeter (model: A109A025LM low flow Aluminum flowmeter) with a flow range of 1 – 11 LPM is installed between the pump and evaporator to record the flowrate of the working fluid.

The water pumps (model: CP200SN) used has a rating of 550 watts, flow of 130 LPM, and a head of 23 meters. They are used to pump the warm and cold waters through the heat exchangers. Shut valves are used to control the flowrate. Both the
warm and cold water are at atmospheric pressure. The temperature of the water at inlet and outlet of the heat exchangers are recorded using CABAC T6201 digital thermometers, with a resolution of 0.1 °C and a temperature range of -50 °C to +250 °C. The warm water temperatures were 24 °C, 27 °C, and 30 °C. The cold water temperature was kept constant between 4.5–5 °C. The warm water flowrates, $\dot{V}_{WS}$, were varied from 0.38 – 0.46 L/s. The cold water flowrate, $\dot{V}_{CS}$, was kept constant at 0.16 L/s. The working fluid flowrates, $\dot{V}_{WF}$, were 2.5 L/s and 4.5 L/s.

The heat exchangers that were experimented on are corrugated plate heat exchangers, as described in section 4.1. However, the heat exchangers used in the final setup were shell and tube type for both the evaporator and the condenser. During manufacturing of the corrugated copper plate heat exchangers for the final system, it was found out that the plates bulged at 400 kPa of air pressure. The plates are too thin and the corrugation pattern that was experimented on did not provide a strong reinforcement. The plates were then reinforced by brazing rods at some portions in the channels. But the bulging problem was still not solved and the reinforcements led to leakages. It was then decided to use shell and tube heat exchangers.

Three spiraled tubes are used in the heat exchangers. The first tube outer diameter is 15.88 mm with a wall thickness of 1.24 mm and the other tubes have an outer diameter of 9.52 mm and wall thickness of 0.89 mm. The shell diameter is 115 mm with a height of 560 mm. Both the warm and cold water enter the heat exchangers from the bottom. This allows the water to fully fill the shells for effective heat transfer and prevent the formation of hydraulic diameters. Figure 15 shows a picture of the coils.

![Figure 15. A picture of the spiraled tubes used in the heat exchangers.](image)
An eight bladed mini, impulse turbine with a diameter of 130 mm enclosed in a metal casing of diameter of 140 mm is used in the system. The turbine is used to study the pressure and enthalpy drop of the working fluid. Figure 16 shows a picture of the mini turbine.

Figure 16. The mini turbine used in the set-up.

The pressure and temperature values read from the gauges were fed into a program in the Engineering Equation Solver (EES) [104]. All the thermodynamic properties were calculated using EES which were then used to calculate the efficiency and the power output. The codes used to do the computations are as follows:

<table>
<thead>
<tr>
<th>T1</th>
<th>P1</th>
<th>T2</th>
<th>P2</th>
<th>T3</th>
<th>P3</th>
<th>T4</th>
<th>P4</th>
</tr>
</thead>
<tbody>
<tr>
<td>24</td>
<td>551.5805832</td>
<td>20.2</td>
<td>482.6330103</td>
<td>13.5</td>
<td>455.0539811</td>
<td>16.5</td>
<td>551.5805832</td>
</tr>
</tbody>
</table>

R$='R134a' "string variable used to hold name of refrigerant 134a"

"Evaporator - properties for state 1"
T1=24 "recorded temperature after evaporator and before turbine"
p1=551.5805832 "recorded pressure after evaporator and before turbine"

h1=enthalpy(R$,T=T1,P=p1)  "enthalpy"
s1=entropy(R$,T=T1,P=p1)  "entropy"
x1=quality(R$,T=T1,P=p1)  "quality"
v1=volume(R$,T=T1,P=p1)  "specific volume"
u1=intenergy(R$,T=T1,P=p1)  "internal energy"
rhol=density(R$,T=T1,P=p1)  "density"
Tsat1=T_sat(R134a,P=p1)  "saturated temperature"

"Turbine - properties for state 2"
T2=20.2 "recorded temperature after turbine and before condenser"
p2=482.6330103 "recorded pressure after turbine and before condenser"
\( h_2 = \text{enthalpy}(R$, $T=T_2, P=p_2) \)
\( s_2 = \text{entropy}(R$, $T=T_2, P=p_2) \)
\( x_2 = \text{quality}(R$, $T=T_2, P=p_2) \)
\( v_2 = \text{volume}(R$, $T=T_2, P=p_2) \)
\( u_2 = \text{intenergy}(R$, $T=T_2, P=p_2) \)
\( \rho_2 = \text{density}(R$, $T=T_2, P=p_2) \)
\( T_{\text{sat}2} = T_{\text{sat}}(\text{R134a}, P=p_2) \)

"Condenser - properties for state 3"

\( T_3 = 13.5 \) "recorded temperature after condenser and before pump"
\( p_3 = 455.0539811 \) "recorded pressure after condenser and before pump"

\( h_3 = \text{enthalpy}(R$, $T=T_3, P=p_3) \)
\( s_3 = \text{entropy}(R$, $T=T_3, P=p_3) \)
\( x_3 = \text{quality}(R$, $T=T_3, P=p_3) \)
\( v_3 = \text{volume}(R$, $T=T_3, P=p_3) \)
\( u_3 = \text{intenergy}(R$, $T=T_3, P=p_3) \)
\( \rho_3 = \text{density}(R$, $T=T_3, P=p_3) \)
\( T_{\text{sat}3} = T_{\text{sat}}(\text{R134a}, P=p_3) \)

"Pump - properties for state 4"

\( T_4 = 16.5 \) "recorded temperature after pump and before evaporator"
\( p_4 = 551.5805832 \) "recorded pressure after pump and before evaporator"

\( h_4 = \text{enthalpy}(R$, $T=T_4, P=p_4) \)
\( s_4 = \text{entropy}(R$, $T=T_4, P=p_4) \)
\( x_4 = \text{quality}(R$, $T=T_4, P=p_4) \)
\( v_4 = \text{volume}(R$, $T=T_4, P=p_4) \)
\( u_4 = \text{intenergy}(R$, $T=T_4, P=p_4) \)
\( \rho_4 = \text{density}(R$, $T=T_4, P=p_4) \)
\( T_{\text{sat}4} = T_{\text{sat}}(\text{R134a}, P=p_4) \)

\$\text{TabWidth} 2 \text{ cm} \$
5.0 Experimental Results and Analysis

5.1. Corrugated Plate Heat Exchangers

The experimental data for the corrugated plate heat exchangers is given in Appendix 1-4. Figure 17 shows the change in the temperature of the hot and cold water (i.e. the difference of inlet and outlet temperatures of the respective streams) with varying hot water flowrates, $\dot{V}_{HW}$, for $\Delta X = 12$ mm. The $\Delta T_{HW}$ decreases with increasing flowrate, and is a minimum at the highest flowrate. The $\Delta T_{HW}$ is a maximum at the lowest flowrate because the hot water gets more time to exchange heat with the cold water. The $\Delta T_{CW}$ is a maximum at the maximum $\dot{V}_{HW}$ because the hot water stream continuously supplies heat energy to the cold water stream at a higher rate without losing much heat energy. At higher $\dot{V}_{HW}$, the temperature change of the hot water from inlet to outlet is very small. Therefore, the hot water acts as a continuous heat source to the cold water stream. Similar trends are observed for $\Delta X = 6$ mm and 9 mm.

![Figure 17. Temperature change of the fluids (difference of inlet and outlet temperatures of the respective streams) ($\Delta X = 12$ mm).](image)

Figure 17 shows the temperature difference of the hot and cold water, $\Delta T_{outlet}$, measured at the exit of the heat exchanger for all $\Delta X$ values. The temperature difference increases slightly and then decreases as $\dot{V}_{HW}$ is increased. The minimum temperature difference at the exit is obtained at the highest $\dot{V}_{HW}$ for all $\Delta X$ values. The inlet temperature difference is 23 °C for all $\dot{V}_{H}$ and $\Delta X$ values, and the minimum
\[ \Delta T_{\text{outlet}} \text{ is obtained for } \Delta X = 6\text{mm. Therefore, the optimum heat transfer between the two streams is obtained for } \Delta X = 6\text{mm.} \]

Figure 18. Temperature difference of hot and cold water at the exit of the heat exchanger against \( \dot{V}_{\text{HW}} \).

The average heat transferred between the two streams is shown in Figure 19. The heat transfer is calculated as:

\[ \dot{Q}_{\text{HW}} = \rho_{\text{HW}} C_{\text{PHW}} \dot{V}_{\text{HW}} (\Delta T_{\text{HW}}) \tag{26} \]

\[ \dot{Q}_{\text{CW}} = \rho_{\text{CW}} C_{\text{PCW}} \dot{V}_{\text{CW}} (\Delta T_{\text{CW}}) \tag{27} \]

\[ \dot{Q}_{\text{Average}} = \left( \frac{\dot{Q}_{\text{HW}} + \dot{Q}_{\text{CW}}}{2} \right) \tag{28} \]

where \( \dot{Q}_{\text{HW}} \) and \( \dot{Q}_{\text{CW}} \) are heat transferred by hot and cold water streams respectively, \( \dot{Q}_{\text{Average}} \) is the average heat transfer between the two streams. As seen from Figure 19, \( \dot{Q}_{\text{Average}} \) increases with increasing \( \dot{V}_{\text{HW}} \) for all values of \( \Delta X \) because of high turbulence at high velocities, causing a much higher heat transfer. The optimum heat transfer is obtained for \( \Delta X = 6\text{mm} \), because for a given \( \dot{V}_{\text{HW}} \), the hot water velocity will always be higher in the \( \Delta X = 6\text{mm} \) channels because of the reduced area. Similar trends for heat transfer (but presented as Nusselt number) with increasing Reynolds numbers for corrugated plates are reported by Tauscher and Mayinger [112].
The inlet gauge pressures at which the two fluids flowed in the heat exchanger were recorded. Since the flowrate of the cold water was kept constant, the pressure variation of the cold water was much less for all values of $\Delta X$, approximately 8 -10 kPa. The pressure loss of the hot water varied a lot with $\dot{V}_{HW}$, as shown in Figure 20. The pressure loss increased with increasing $\dot{V}_{HW}$. The highest pressure loss of 45 kPa is recorded for $\Delta X = 6$ mm. The minimum pressure losses are recorded for $\Delta X = 12$ mm. Similar trends for pressure losses are reported by Bellas et al. [106] and Elshafei et al. [115]. The pressure losses are however presented against the Reynolds numbers in their case. The pressure losses are due to the promotion of
unstable vortices due to the corrugations. The increase in pressure losses with increasing Reynolds numbers for corrugated plates are also reported by Tauscher and Mayinger [112].

The variations of the overall heat transfer coefficient, $U$, for different $\dot{V}_{HW}$ and $\Delta X$ are shown in Figure 21. The $U$ value is calculated as:

$$U = \frac{\dot{Q}_{Average}}{A\Delta T_M} \quad (29)$$

$$\Delta T_M = \frac{(T_{HW1} - T_{CW1}) - (T_{HW0} - T_{CW0})}{\ln \left( \frac{T_{HW1} - T_{CW1}}{T_{HW0} - T_{CW0}} \right)} \quad (30)$$

where $\dot{Q}_{Average}$ is the arithmetical mean of $\dot{Q}_{HW}$ and $\dot{Q}_{CW}$, $A$ is the total heat transfer area and $\Delta T_M$ is the log mean temperature difference. The overall heat transfer coefficient takes into account all the resistances that are present in the path of the heat transfer. As shown in Figure 21, $U$ increases with $\dot{V}_{HW}$ for all values of $\Delta X$. The $U$ value is higher for $\Delta X = 6$ mm because the fluid velocities are higher in the 6 mm channels, thus higher turbulence which enhances heat transfer. A similar trend for the overall heat transfer coefficients against water velocities has been reported by Avery and Wu [105] and Uehara et al. [107]. Sparrow and Comb [116] found that the heat transfer coefficient for the larger plate spacing was slightly smaller than that of the lower plate spacing, but the pressure drop was also lower.

![Figure 21. The variation of the overall heat transfer coefficient, $U$, with varying $\dot{V}_{HW}$.](image)

40
The variations of the average thermal length, $\theta_{\text{Average}}$, for varying $\dot{V}_{HW}$ and $\Delta X$ are shown in Figure 22. The thermal length represents the performance and is the relationship between the temperature difference in one stream and the LMTD. A higher thermal length means that the heat transfer and the pressure drop are large, whereas a lower thermal length means that heat transfer and pressure drops are low [124]. The thermal lengths are calculated as:

$$\theta_{HW} = \frac{\Delta T_{HW}}{\Delta T_m}$$  \hspace{1cm} (31)

$$\theta_{CW} = \frac{\Delta T_{CW}}{\Delta T_m}$$  \hspace{1cm} (32)

$$\theta_{\text{Average}} = \frac{\theta_{HW} + \theta_{CW}}{2}$$  \hspace{1cm} (33)

where $\theta_{HW}$ and $\theta_{CW}$ are the thermal lengths of the hot water and cold water channels respectively, and $\theta_{\text{Average}}$ is the arithmetic mean of $\theta_{HW}$ and $\theta_{CW}$. As seen from Figure 22, $\theta_{\text{Average}}$ increases with $\dot{V}_{HW}$ for all $\Delta X$ values, and is higher for $\Delta X = 6$ mm compared to other $\Delta X$ values. Therefore, the heat exchanger with $\Delta X = 6$ mm has better performance.

![Figure 22. The variation of the average thermal length, $\theta_{\text{Average}}$, with varying $\dot{V}_{HW}$.](image)

The pumping costs of heat exchangers will be higher if the pressure losses are significant. The overall heat transfer coefficient, $U$, is also considered when
designing or choosing heat exchangers. Figure 23 shows a relationship between the U value and the pressure losses of the warm water for all values of \( \Delta X \). A similar criterion for the selection of heat exchangers based on the heat transfer coefficient and the pressure losses is reported by Rafferty and Culver [108]. There is an increase in the U value with increasing pressure loss. For \( \Delta X = 6 \text{ mm} \), there is a high pressure loss, therefore, the heat exchanger would have higher operational costs. However, the heat exchanger with \( \Delta X = 6 \text{ mm} \) is appropriate because of significant heat transfer coefficients and effective heat transfer, even though the pressure losses are higher. The operational cost could be higher due to high pressure losses, but the main objective is to obtain an effective heat transfer rate between the two streams for such a low temperature difference.

![Figure 23](image)

Figure 23. The overall heat transfer coefficient, U, presented against the pressure loss of the hot water, \( \Delta P_H \).

Both the hot and cold water streams are single-phase flows that undergo mainly forced convection and conduction in the heat exchanger channels. A hydrodynamic and a thermal boundary layer begin to develop as soon as the fluids enter the channels. The convex and concave surfaces in the closed channels cause instabilities in the flow, which enhance turbulence. The flow over the convex surface is more stable because the velocity gradient maintains a constant sign across the boundary layer [125], and any fluid element that gets displaced outward to a higher velocity region gets pushed back to a lower radius region due to higher radial pressure gradient [126]. The flow over the concave surface is unstable because the velocity gradient changes sign in the boundary layer [125], and any fluid element
that gets displaced to a greater radius moves into a region of low velocities where the pressure gradient is too low to push it back to a lower radius (Görtler instability) [126]. The secondary flows or the Görtler vortices induced by the corrugations cause the partial restarts of the boundary layer [111], and prevent it from being fully developed.

The boundary layer over the convex surface has a point of inflection which slows down the flow near the surface and changes the flow direction under a strong adverse pressure gradient. When the incoming flow meets the reversed flow at some point, the fluid near the surface is transported into the mainstream, or separated from the surface. Since the flow is in a closed channel, and the plate geometries are same, most of the fluid elements are pushed back to the surface. However, due to initial separation, there are vortices formed in the wake region and their characteristics depend on the Reynolds numbers. When on the concave surface, the flow gets unstable due to Görtler instability. As the flow moves forward, it encounters a rising wall (the next convex surface) and as a result, the flow close to the wall slows down and disturbs the incoming flow. As a result, turbulence is enhanced and this continues upto the end of the channel. Metwally and Manglik [127] performed a numerical study on sinusoidal plate channels and concluded that flow separation and attachment generates vortices that cause mixing which enhances the heat transfer. The corrugations on the plates always cause turbulence in the channels regardless of the flow being laminar or turbulent at the entrance of the channels. Turbulence in the channels leads to wall shear stresses which also reduces fouling on the plate surfaces. The heat transfer is a result of the disruptions of both the hydrodynamic and the thermal boundary layers.

Smooth plates are not so effective because once the hydraulic boundary layer is fully developed, the central region of the fluids do not receive much heat from the adjacent channel compared to the fluid elements close to the wall. Also, as the wall spacing is increased, the heat received by the central region decreases. In contrast, corrugations on the plate surface lead to continuous disruptions in the boundary layer across the length of a channel from inlet to exit. The secondary flow causes turbulent mixing of the fluids in the channels from one wall to another. This allows almost all the fluid elements to have effective heat transfer from adjacent channels. Therefore, it is advisable to always prefer corrugated plates over smooth plates for plate type heat exchangers.
5.2. Closed Cycle Demonstration OTEC Plant

The efficiency and power output were calculated using the enthalpy values from EES. The other properties calculated were density, saturation temperature, and quality. The power output was calculated using the enthalpy drop across the turbine multiplied by the working fluid flowrate. The thermal efficiency was calculated by dividing the enthalpy drop across the turbine by the enthalpy difference of the outlet and inlet of the evaporator.

Figures 24 and 25 show the thermal efficiencies and the power output of the demonstration plant against the difference of the warm and cold water inlet temperatures for varying $\dot{V}_{WS}$ and for both $\dot{V}_{WF}$. It is generally seen that the thermal efficiency and the power output increases with increasing temperature difference. The results are presented against the temperature difference because it is an important parameter in choosing actual plant installation sites and system design. Optimum power will be produced when the total temperature difference is sufficient to promote heat transfer in the heat exchangers as well as to provide a pressure drop across the turbine [22]. The efficiencies are higher for higher $\dot{V}_{WS}$. There is more heat transfer in the evaporator at higher flowrates because the warm water continuously supplies heat energy to the working fluid without losing much energy through the length of the heat exchanger, thus more heat transfer to the working fluids and better turbine performance. Yamada et al. [52] presented similar trends in efficiencies against the operating temperature difference. Hettiarachichi et al. [128] also presented the efficiencies against the operating temperature difference and obtained similar trends. The efficiencies for $\dot{V}_{WF} = 4.5$ L/s are higher compared to $\dot{V}_{WF} = 2.5$ L/s. Higher $\dot{V}_{WF}$ leads to a higher pressure at the turbine inlet and reduces heat loss to the surrounding on the higher temperature side. The range of thermal efficiencies for $\dot{V}_{WF} = 2.5$ L/s is 0.8 – 1.15% and 0.8 – 1.5% for $\dot{V}_{WF} = 4.5$ L/s.

The work done by the turbine for both $\dot{V}_{WF}$ generally increases with increasing operating temperature difference, and is higher for larger $\dot{V}_{WS}$. The turbine uses most of the energy from the working fluid to do work, and as a result there is a pressure drop across the turbine which leads to an enthalpy drop. The larger the pressure (and enthalpy drop) across the turbine, the more work is done by the turbine. The power output for $\dot{V}_{WF} = 4.5$L/s is higher compared to $\dot{V}_{WF} = 2.5$ L/s. A higher $\dot{V}_{WF}$ gives a
higher pressure at the turbine inlet and thus a higher pressure and enthalpy drop across the turbine. The power output for $\dot{V}_{WF} = 2.5$ L/s is between $5 - 6.8$ W and $8.5 - 15.8$ W for $\dot{V}_{WF} = 4.5$ L/s.

Figure 24. Thermal efficiency and power output of the system against operating temperature difference, for $\dot{V}_{WF} = 2.5$ L/s, and varying $\dot{V}_{WS}$.

Figure 25. Thermal efficiency and power output of the system against operating temperature difference, for $\dot{V}_{WF} = 4.5$ L/s, and varying $\dot{V}_{WS}$.

Figure 26 shows the turbine inlet pressure and turbine pressure drop against operating temperature difference, for $\dot{V}_{WS} = 0.46$ L/s and both the $\dot{V}_{WF}$. The inlet pressure and pressure drop across the turbine increased as the operating temperature
(difference of warm water and cold water inlet temperature) difference increased. For \( \dot{V}_{WF} = 2.5 \text{ L/s} \), the maximum pressure at the turbine inlet (exit of evaporator) was 551.58 kPa and after the condenser was 455.05 kPa, for a warm water inlet temperature of 30ºC. For \( \dot{V}_{WF} = 4.5 \text{ L/s} \), the maximum pressure at the turbine inlet was 586.05 kPa and after the condenser pressure was 482.63 kPa, for the same warm water inlet temperature of 30ºC. Thus, it can be seen that the pressure at the evaporator and condenser increased with increasing warm water inlet temperatures.

The variations in the \( \dot{V}_{WS} \) did not affect the pressure.

![Graph](image)

Figure 26. Turbine inlet pressure and turbine pressure drop against operating temperature difference, for \( \dot{V}_{WS} = 0.46 \text{ L/s} \) and both \( \dot{V}_{WF} \).

Figures 27 and 28 show the thermal efficiencies and the power output against the pressure drop across the turbine, for both \( \dot{V}_{WF} \). The pressure drop across the turbine achieved in this demonstration system is between 40 – 75 kPa. Even though the results are presented against the pressure drop, the superheat at the turbine inlet will make a significant difference in the system performance, since phase change in the cycle ideally occurs at constant pressure. The superheat in the present system for both the working fluid flowrates is between 4.3 – 6.09 ºC. Without any major focus on superheating, it is seen that the thermal efficiencies increase with increasing pressure across the turbine. Higher warm water flowrate give higher efficiencies. Also, \( \dot{V}_{WF} = 4.5 \text{ L/s} \) has higher efficiencies compared to \( \dot{V}_{WF} = 2.5 \text{ L/s} \). The power output increases with increasing pressure drop in a manner similar to the thermal efficiencies. The higher values for \( \dot{V}_{WS} \) and \( \dot{V}_{WF} \) give higher power. For \( \dot{V}_{WF} = 4.5 \text{ L/s} \),
there is a significant jump in the pressure drop across the turbine which leads to a
sudden increase in the efficiency.

Figure 27. Thermal efficiency and power output of the system against the pressure
drop across the turbine, for $\dot{V}_{WF} = 2.5$ L/s and varying $\dot{V}_{WS}$.

Figure 28. Thermal efficiency and power output of the system against the pressure
drop across the turbine, for $\dot{V}_{WF} = 4.5$ L/s and varying $\dot{V}_{WS}$. 
Figures 29 and 30 show the thermal efficiencies and the power output against the turbine inlet temperature for all $V_{WS}$ and both $V_{WF}$. The temperature values at the inlet of the turbine in this demonstration system are similar to those of actual systems. The turbine inlet temperature, achieved after the working fluid passes through the evaporator, is higher for higher values of the warm seawater inlet temperature (because of the high heat transfer due to higher temperature difference of the working fluid and
warm water). The efficiencies for both cases increase with increasing turbine inlet temperature. The higher the inlet temperature (for a given pressure), the higher will be the superheat and the enthalpy, thus more energy available to drive the turbine. Tong et al. [44] and Hettiarachichi et al. [128] had achieved similar trends for efficiency against turbine inlet temperature. The higher efficiencies are obtained for \( \dot{V}_{WF} = 4.5 \) L/s and for larger \( \dot{V}_{WS} \). The power output increases with increasing turbine inlet temperature and has similar trends to those of the thermal efficiencies. There is more work done by the turbine when the turbine inlet temperature is higher. The power is higher for \( \dot{V}_{WF} = 4.5 \) L/s and for larger \( \dot{V}_{WS} \).

Figures 31 and 32 show the thermal efficiencies and the power output against the ratio of the water flowrates, \( \dot{V}_{ws}/\dot{V}_{cs} \), for both \( \dot{V}_{WF} \). Both the efficiency and the power increase with increasing \( \dot{V}_{ws}/\dot{V}_{cs} \). The highest efficiency and power for both \( \dot{V}_{WF} \) are obtained for the maximum water temperature of 30°C. The higher flowrate of the working fluid (\( \dot{V}_{WF} = 4.5 \) L/s) gives higher efficiencies and power output. Yeh et al. [50] presented similar trends of the net work against the ratio of the water flowrates. They also stated that it is always economical to increase the warm water flowrates since the pipe length of the warm water pipes are much smaller than the cold water pipes.

![Figure 31. Thermal efficiency and power output of the system against the ratio of the water flowrates, \( \dot{V}_{ws}/\dot{V}_{cs} \), for \( \dot{V}_{WF} = 2.5 \) L/s and all warm water temperatures.](image-url)
Figure 32. Thermal efficiency and power output of the system against the ratio of the water flowrates, $\dot{V}_{ws}/\dot{V}_{es}$, for $\dot{V}_{WF} = 4.5$ L/s and all warm water temperatures.
6.0 Conclusions

The heat exchange processes across the ocean surface and the technology for ocean thermal energy conversion are presented. The heat exchange processes across the ocean surface are represented in an ocean energy budget. Ocean currents transfer thermal energy from the lower latitudes to cooler regions in the higher latitudes. The ocean energy budget quantifies the amount of heat gained and lost by the ocean, and this can be used to determine the overall temperature change of the system over a certain period of time. The accurate measurements and predictions of the ocean energy budget terms are difficult and some errors and imbalances are still present. The transport of cold water from the higher latitudes towards the equator along the ocean bottom results in the displacement of the lower density water above and creates a thermal structure with a large reservoir of warm water at the ocean surface and a large reservoir of cold water at the bottom. The temperature difference of these reservoirs can be used to drive OTEC systems. Ocean thermal energy conversion (OTEC) plants operate using this temperature difference to run a turbine with efficiencies close to 3%. The thermal structure of the oceans, or the thermocline, varies with different latitudes and is permanent for lower latitudes. There are many technological issues for OTEC plant implementation, such as getting cold water from the ocean depths, which is a major concern. Many technological problems are however solved, such as fouling and compact designs of heat exchangers. The case studies presented by many researchers clearly show that OTEC technology can be successfully implemented. However, proper design and planning is required. The initial capital cost for OTEC plants is very high, but once the plant is operational, the costs will be recovered in the long run through power generation and by-products. Ocean thermal energy conversion plants can alter the ocean surface energy balance by altering the surface temperatures and increased CO₂ production due to increased mixing of surface and deep waters. But no such issues were faced during the actual operation of the demonstration OTEC plants in Hawaii and Nauru. However, very large plants or a cluster of OTEC plants may have some effects on the environment and ocean surface energy balance.

The heat transfer and pressure drops in a corrugated plate heat exchanger with variable spacing and variable warm water flowrates were studied with the help of temperature measurements at the inlet and exit of the plate heat exchanger. It is
found that for a given plate spacing, \( \Delta X \), with increasing hot water flowrate, \( \dot{V}_{HW} \), the average heat transfer, \( Q_{\text{Average}} \), between the two streams increases due to high turbulence at higher velocities. The overall heat transfer coefficient, \( U \), the pressure losses, and the average thermal length are found to increase with increasing \( \dot{V}_{HW} \), and are higher for \( \Delta X = 6 \text{ mm} \) heat exchanger compared to other \( \Delta X \) values. The plate heat exchanger with \( \Delta X = 6 \text{ mm} \) is found to be appropriate due to effective heat transfer and higher thermal length even though the pressure losses are higher. The corrugations on the plate surfaces induce secondary flows in the channels and cause turbulent mixing which allows all the fluid elements in a particular channel to have effective heat transfer with the adjacent channels.

A closed cycle OTEC demonstration plant was built and its performance was experimentally studied with the help of temperature and pressure readings before and after each component. A higher warm water temperature increases the heat transfer between the warm water and the working fluid, thus increasing the working fluid temperature, pressure, and enthalpy before the turbine. The performance is better at larger flowrates of the working fluid and the warm water. It is found that the thermal efficiency of the system and the work done by the turbine both increases with increasing operating temperature difference (difference of warm and cold water temperature). The turbine inlet pressure and the pressure drop across the turbine both increase with increasing operating temperature difference. The performance of the system improves with increasing pressure drop across the turbine. Increasing turbine inlet temperatures also increase the efficiency and the work done by the turbine. The efficiency and the power output increase with increasing ratio of warm water flowrates to cold water flowrates.
References


43. Yamada N, Hoshi A, Ikegami Y. Thermal efficiency enhancement of ocean thermal energy conversion (OTEC) using solar thermal energy. 4th International Energy Conversion Engineering Conference and Exhibit (IECEC), San Diego, California, American Institute of Aeronautics and Astronautics, 2006-4130.


75. Hamilton K. Numerical resolution and modeling of the global atmospheric circulation: a review of our current understanding and outstanding issues. In


Appendix
Appendix 1: The recorded values of temperatures in the corrugated heat exchanger experiments

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<th>Tho (deg)</th>
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<th>Tco (deg)</th>
<th>ΔTc (deg)</th>
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### Appendix 2: Calculations done in Excel (part 1)

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<th>$\Delta T_m$</th>
<th>$U$ (W/m$^2$K)</th>
<th>Velocity (m/s)</th>
<th>$\text{thermal length (hot)}$</th>
<th>$\text{thermal length (cold)}$</th>
<th>$\text{Average thermal length}$</th>
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## Appendix 4: Calculations done in Excel (part 3)

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Appendix 5: The recorded values of temperatures and pressures in the demonstration OTEC plant experiments for $\dot{V}_{HF} = 2.5$ L/s.

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### Appendix 6: Calculations done in EES and Excel (part 1)

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<th>P3</th>
<th>P1-P2</th>
<th>P1/P2</th>
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<th>P2</th>
<th>P3</th>
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71
Appendix 8: The recorded values of temperatures and pressures in the demonstration OTEC plant experiments for \( \dot{V}_{WF} = 4.5 \text{ L/s} \).

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<th>3 (T(deg) P (kpa))</th>
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Appendix 9: Calculations done in EES and Excel (part 1)
Appendix 10: Calculations done in EES and Excel (part 2)

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