BACHELOR THESIS & COLLOQUIUM – ME184841

DESIGN OF OCEAN THERMAL ENERGY CONVERSION POWER PLANT USING ISOBUTANE AS WORKING FLUID

MUHAMMAD INDRA HAZAMI
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DOUBLE DEGREE PROGRAM
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TUGAS AKHIR - ME184841

DESAIN PEMBANGKIT LISTRIK OCEAN THERMAL ENERGY CONVERSION MENGGUNAKAN ISOBUTANA SEBAGAI FLUIDA KERJA

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INSTITUT TEKNOLOGI SEPULUH NOPEMBER
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2019

APPROVAL FORM

DESIGN OF OCEAN THERMAL ENERGY CONVERSION POWER PLANT USING ISOBUTANE AS WORKING FLUID

BACHELOR THESIS

Submitted as one of Requirements to obtain Bachelor Degree in Engineering
on
Marine Fluid Machinery and System (MMS)
Bachelor Program in Marine Engineering Department
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember

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DECLARATION OF HONOR

I hereby who signed below declare that :

This final project has written and developed independently without any plagiarism act. All contents and ideas drawn directly from internal and external sources are indicated such as cited sources, literatures, and other professional sources.

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Department	: Marine Engineering

If there is plagiarism act in the future, I will fully responsible and receive the penalty given by ITS according to the regulation applied.

Surabaya, July 2019

Muhammad Indra Hazami

DESIGN OF OCEAN THERMA ENERGY CONVERSION POWER PLANT USING ISOBUTANE AS WORKING FLUID

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ABSTRACT

The search for renewable energy sources has intensifies in recent years for a number of reason. Many countries need energy to run the economic and to develop and improve human welfare. One of the research of an alternative renewable energy is Ocean Thermal Energy Conversion. This is a power cycle that is turn a heat engine, which powers a low pressure turbine. The concept of the first working models of OTEC was built and designed by French Inventor Georges Claude. The work princiole of OTEC is very simple. Surface water temperature is enough to cause the working fluid to boil, then cold water from approximately 1000 meters deeo will be pumped to the surface to condense the working fluid. The steam that comes from low pressure boiling system is enough to power a turbine thus creating work. The temperature differences between surface and deep water must be greater than 20^{o}C. Indonesia is located in tropical area that the differential temperature of the sea is high in common with low intensity of wave.

This research purpose is to know the potetential power generated by OTEC closed cycle system in Indonesia using isobutene as working fluid. This research will simulate the OTEC with software and validate with manual calculation. Beside that, this research purpose is to know the best mass flow rate of sea water and the best sea depth for OTEC system. From the simulation of the OTEC the first scenario with a depth of 1000 m and a working mass flow rate at 545.32 kg/s has the best gross power generated with 426.34 kW. OTEC design uses isobutane as the best working fluid is to use scenario 1 at a depth of 1000m with variations in the mass flow rate of seawater at 500 kg/s with a power of 13.99 kW warm sea water pump and cold sea water pump power 96.42 kW. Then get a working fluid mass flow rate of 226.13 kg s with power of 6.78 kW. Net Power generated is 52.4 kW with efficiency of 0.61%. The results of the simulation show that the deeper the depth of the sea, the greater the output power. In this study 1st scenario at a depth of 1000 m produces the largest gross power output, 85.27 kW, 170.53 kW, 255.8 kW, 341.07 kW, and 426.34 kW according to variations in the sea water mass flow rate. The results of the simulation show the greater the working fluid mass flow rate the greater the gross power output produced. But it does not apply to net power output because it has to consider the power requirements of a sea water pumps and a working fluid pump.

Key Words : Isobutane, Marine Technologies,Ocean Thermal Energy Conversion, Renewable Energy, Sea Water Mass Flow Rate.

DESAIN PEMBANGKIT LISTRIK *OCEAN THERMA ENERGY CONVERSION* MENGGUNAKAN ISOBUTANA SEBAGAI FLUIDA KERJA

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ABSTRAK

Pencarian untuk sumber energi terbarukan telah meningkat dalam beberapa tahun terakhir karena sejumlah alasan. Banyak negara membutuhkan energi untuk menjalankan ekonomi dan untuk mengembangkan serta meningkatkan kesejahteraan manusia. Salah satu penelitian energi terbarukan alternatif adalah *Ocean Thermal Energy Conversion*. Ini adalah sebuah siklus daya yang menghidupkan mesin panas, yang menggerakkan turbin tekanan rendah. Konsep model kerja pertama OTEC dibangun dan dirancang oleh Penemu Prancis Georges Claude. Prinsip kerja OTEC sangat sederhana. Suhu air permukaan cukup untuk menyebabkan fluida kerja mendidih, kemudian air dingin dari sekitar kedalaman laut 1000 meter akan dipompa ke permukaan untuk mengondensasi fluida kerja. Uap yang berasal dari sistem pendidihan bertekanan rendah sudah cukup untuk memberi daya pada turbin sehingga menghasilkan kerja. Perbedaan suhu antara permukaan dan air yang dalam harus lebih besar dari 20^{o}C. Indonesia terletak di daerah tropis yang memiliki perbedaan suhu laut yang tinggi dengan intensitas gelombang yang rendah.

Tujuan penelitian ini adalah untuk mengetahui daya potensial yang dihasilkan oleh sistem siklus tertutup OTEC di Indonesia menggunakan isobutana sebagai fluida kerja. Penelitian ini akan mensimulasikan OTEC dengan perangkat lunak dan memvalidasi dengan perhitungan manual. Selain itu, tujuan penelitian ini adalah untuk mengetahui laju aliran massa air laut terbaik dan kedalaman laut terbaik untuk sistem OTEC. Dari simulasi OTEC, skenario pertama dengan kedalaman 1000 m dan laju aliran massa fluida kerja 545,32 kg / s memiliki daya kotor terbaik yang dihasilkan dengan 426,34 kW. Desain OTEC menggunakan isobutana sebagai fluida kerja terbaik adalah dengan menggunakan skenario 1 pada kedalaman 1000m dengan variasi laju aliran massa air laut pada 500 kg / s dengan daya yang dibutuhkan 13,99 kW untuk pompa air laut hangat dan pompa air laut dingin membutuhkan daya 96.42 kW. Kemudian dapatkan laju alir massa fluida kerja 226,13 kg s dengan pompa yang membutuhkan daya 6,78 kW. Daya Bersih yang dihasilkan adalah 52,4 kW dengan efisiensi 0,61%. Hasil simulasi menunjukkan bahwa semakin dalam kedalaman laut, semakin besar daya output. Dalam studi ini, skenario 1 pada kedalaman 1000 m menghasilkan output daya kotor terbesar, 85,27 kW, 170,53 kW, 255,8 kW, 341,07 kW, dan 426,34 kW menurut variasi dalam laju aliran massa air laut. Hasil simulasi menunjukkan semakin besar laju aliran massa fluida kerja, semakin besar output daya kotor yang dihasilkan. Tapi itu tidak berlaku untuk output daya bersih karena harus mempertimbangkan kebutuhan daya pompa air laut dan pompa fluida kerja.

Kata Kunci: Energi Terbarukan, Isobutane, Konversi Energi Panas Laut, Laju Aliran Massa Air Laut, Teknologi Kelautan..

PREFACE

Praise and thank you the author said to *Allah Subhanahu Wa Ta'Ala*, because with his permission the author could complete the bachelor thesis as one of the graduation requirements of the undergraduate marine engineering department program of Faculty of Marine Technology of the Institut Teknologi Sepuluh Nopember well, smoothly and on time.

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6. To batch 2015 marine engineering friends who have helped and supported the author to complete this bachelor thesis.

The author realizes that this bachelor thesis has many obstacles and disadvantages. Therefore, constructive criticism and suggestions are very much awaited for a better learning process. Hopefully bachelor thesis can be useful for readers. thanks.

Surabaya, July 2019

Author

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CHAPTER I
INTRODUCTION

1.1 Background

The search for renewable energy sources has intensifies in recent years for a number of reason. Many countries need energy to run the economic and to develop and improve human welfare. But, awareness of the potential consequences of using carbon energy has increased, as has the price of fossil fuels, moreover, the deep water Horizon oil disaster in the gulf of Mexico that reminded us the consequences of fuel extraction and use made us motivated to search renewable energy that clean and carbon-free. Even in some countries have spurred interest in renewable energy that clean and carbon-free with financial and tax incentivies.

One of the research of an alternative renewable energy is Ocean Thermal Energy Conversion (OTEC). This is a power cycle that is turn a heat engine, which powers a low-pressure turbine. The concept of the first working models of OTEC was built and designed by French Inventor Georges Claude. His first attempt to create OTEC model was a failure and could not produce a net power output. His second attempt in 1935 was not as good as first attempt because the difficulties of installing the cold water pipe (CWP) inn order to pump the cold deep water to the surface. In 1993 a small ope cycle OTEC plant was designed and constructed by L.A. Vega in Hawaii but it was closed in 1998. It was a model of an actual working plant and did not produce enough power to funded the operation of the plant.

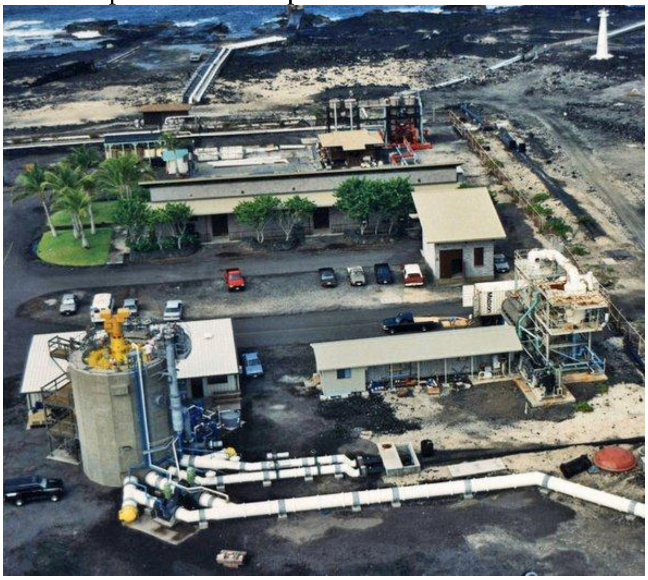

Fig 1. 1. Small open cycle OTEC plant in Hawaii

Source : scientificamerican.com

The work principle of OTEC is very simple. Surface water temperature is enough to cause the working fluid to boil, then cold water from approximately 1000 meters deep will be pumped to the surface to condense the working fluid.the steam that comes from low-pressure boiling system is enough topower a turbine thus creating work. The cold water is to used to condense the steam. The temperature differential must be greater than approximately 20°C for net power generation. Such as differentials exist between latitudes 20°-24° north and south of the equator (e.g. tropical zones of the Caribbean and the pacific). The global distribution of temperature is shown fig 2. The actual distribution of feasible sites for OTEC will depend on other factors as well, such as proximity to shore and the potential to increase the temperature gradient by other means (e.g by applying waste heat from other industrial facilities).

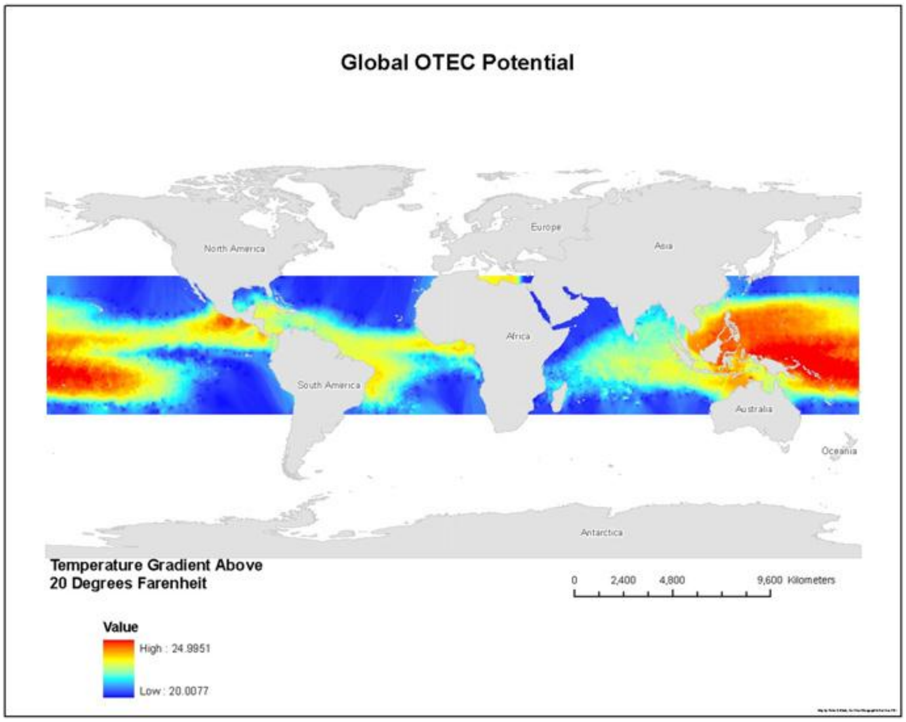

Fig 1.2. Distribution of ocean temperature gradients in excess of 20°C

Indonesia is located in tropical area that the differential temperature of the sea is high in common with low intenstity of wave. From map of distribution of ocean temperature below on fig.1.3 the temperature difference between surface and depth of 1000m in Indonesia is around 22° to 24°C that more than enough to implemented the Ocean Thermal Energy Conversion (OTEC).

The Cycle that research here is Closed cycle. In a closed cyle, a low boiling point liquid such as ammonia, propane, isobutene or another type of refrigerant is used as the working fluid in Rankine Cycle. The heat from warm seawater flowing through an evaporator vaporizes the working fluid. The vapor expands through a turbine, and then flows into a condenser where cold seawater condenses it into a liquid.

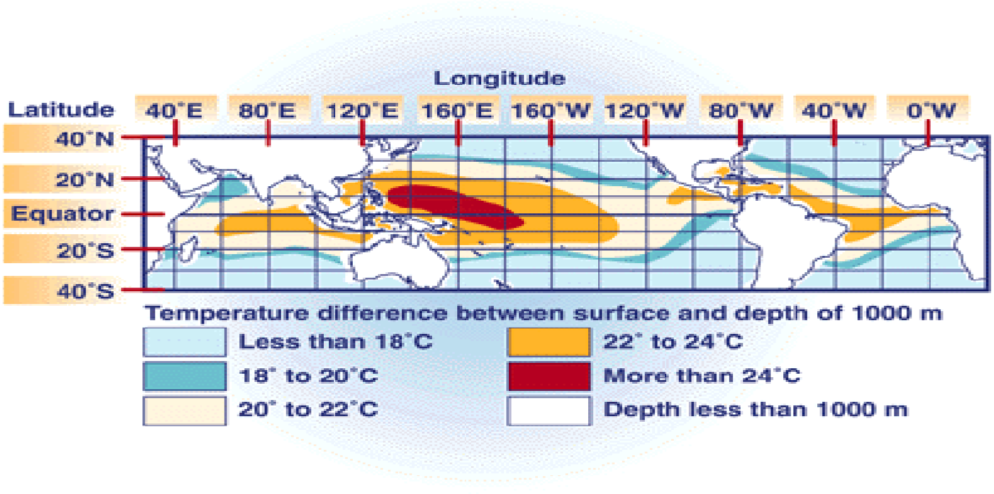

Fig.1.3 Distribution of tempereature different between surface and deep water Map

Source : Sinuhaji (2015)

1.2 Problem Statement

1. How to design an OTEC closed cycle with Isobutane as Working Fluid?
2. How does the working fluid mass flow rate against system performance?
3. How does sea depth relation against system performance?

1.3 Research purposes

1. Design an OTEC closed steam cycle with isobutene as working fluid.
2. Getting working fluid mass flow rates that are good for optimal system performance.
3. Getting sea depth that are good for optimal system performance.

1.4 Constraints

The analysis used is based on the heat and mass balance of the thermodynamic rankine cycle with the following variables and assumptions:

1. Cold sea water temperature entering the condenser is variated based on variation of sea depth.
2. Warm sea water temperature enters the evaporator and is set at 29^{o}C
3. Working fluid temperature out of the evaporator is set at 25^{o}C
4. DT for condenser is set 4oC based on condenser specification.
5. The mass flow rate of cold and warm seawater is variated to 250 kg/s, 500 kg / s, 750 kg/s, 1000 kg/s, and 1250 kg/s
6. Warm sea water is pumped at a depth of 10 meters
7. Cold sea water is pumped in variated depth of 1000 m, 900m, 800m, 700m, 600m, 500m, 400m, 300m, 200m, and 100m.
8. System in steady state (steady state)

9. Cold and warm seawater density is considered constant
10. Properties of seawater use a water propitance approach.
11. This study of OTEC design is tropical condition only.
12. Technical aspects are only limited to physical conditions.
13. There is no calculation of reviewing pump performance aspects and other equipment.

1.5 Benefits

The benefits of this thesis research are:

1. Increasing the knowledge of writers and readers about power plants that utilize Ocean Thermal Energy Conversion (OTEC).
2. Develop an environmentally friendly power generation system.
3. Optimizing the OTEC generating system by using the right working fluid.
4. Optimizing Indonesia's marine potential to increase electricity production in Indonesia.

CHAPTER II
LITERATURE REVIEW

2.1 State of the Technology

OTEC power system operate as cyclic heat engines. They receive thermal energy through heat transfer from surface sea water warmed by the sun, and transform a portion of this energy to electrical power. The second law of thermodyhamics precludes the complete conversion of thermal energy in to electricity. A portion of the heat extracted from the warm sea water must be rejected to a colder thermal sink employed by OTEC systems is sea water drawn from the ocean depths by means of a submerged pipeline. A steady-state control volume energy analysis yields result that net electrical power produced by th engine must equal the difference between the rates of heat transfer from the warm surface water and to the cold deep water. Hence, OTEC efficiency is low. Although viable OTEC systems are characterized by carnot efficiencies in the range of 6-8%, state-of-the-art combustion steam power cycles, which tap much higher temperature energy sources are theoretically capable of converting more than 60% of the extracted thermal energy into electricity (Masutani and Takahashi, 2001).

2.2 Cycle of Ocean Thermal Energy Conversion

2.2.1 Closed Cycle

The original concept employed a pure working fluid that would evaporate at the temperature of warm sea water. The vapor would subsequently expand and do work before being condensed by the cold sea water. this series of steps would be repeated continuously with same working fluid, that flow path representation closed loop. This loop process is rankine cycle. Figure below will explain the schematic diagram od a closed cycle of OTEC system in a simple way. The main component of this system are the heat exchangers, turbogenerator,and seawater supply system. So, the processes of close cycle are:

1. Heat transfer from warm surface sea water occurs in the evaporator, producing a saturated vapor from the working fluid.
2. Electricity is generated when this gas expands to lower pressure through the turbine.
3. Latent heat is transferred from the vapor to the cold sea water in condenser and resulting liquid is pressurized with a pump to repeat cycle.
4. Closed cycle is the process where heat use to evaporate the fluid on constant pressure in tank heater or evaporator which steam into turbine and piston \
5. engine or expansion does work, the steam out enters into a container where the heat is transferred from the steam to coolant causing the steam is condensed into a liquid is pumped back into the evaporator to complete the cycle

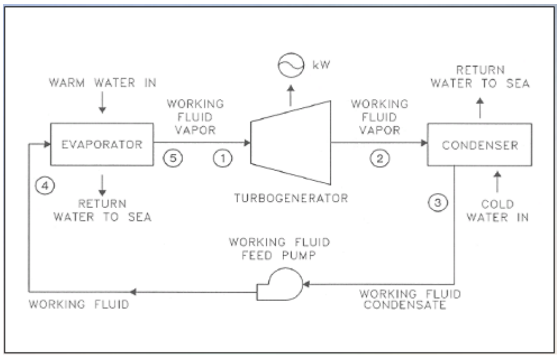

Fig 2.1. Schematic diagram of a closed-cycle OTEC System

Source:masutani et.al (2001

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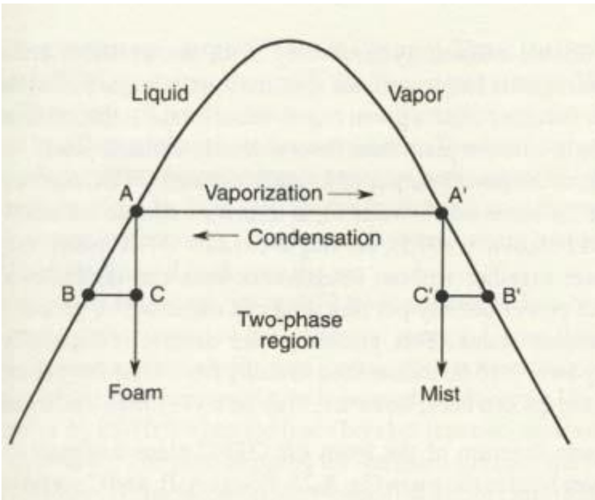

Fig.2.2 Rankine cycle for OTEC system

Source : Sinuhaji (2015)

Based on T-s diagram in fig.2.3, pressure is assumed to be constant during heat addition to the evaporator, (p1 = p4) and heat extraction from the condenser (p2 = p3).

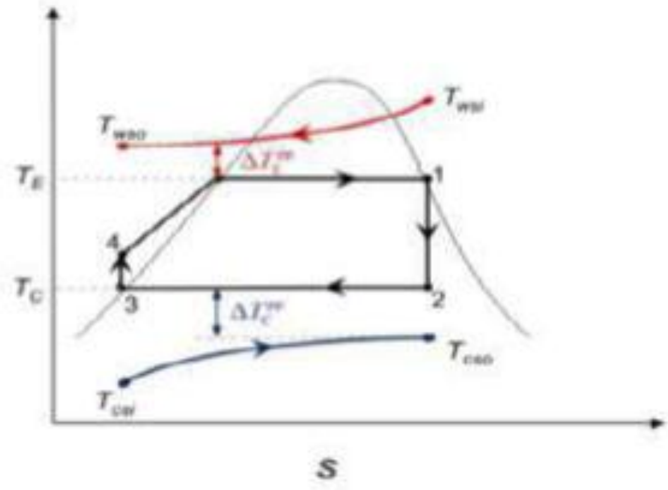

Fig 2.3. T-s diagram of the closed Rankine cycle

Table 2.1 OTEC closed cycle process

No.	Process
1	Working fluid steam entering Turbine
2	Saturated Working fluid entering condenser
3	Liquid working fluid entering working fluid pump
4	Working fluid exiting pump and entering evaporator
Twsi	Warm seawater entering evaporator
Twso	Warm seawater exiting evaporator
Tcsi	Cold seawater entering evaporator
Tcso	Cold seawater exiting evaporator

Thermodynamic Analysis and heat transfer in OTEC closed cycle system are as follows:

1. Heat exchanger (Evaporator and Condenser)
   In evaporator, working fluid is evaporated to be saturated steam after receiving hear from warm sea water. the equation of energy balance in evaporator is as follows:

$$Qe = \dot{m}_{wf} (h_1 - h_4) = \dot{m}_{ws}\, c\, (T_{wsi} - T_{wso}) \qquad (1)$$

With assumption that sea water is incompressible fluid, then heat that added to evaporator is equal to the heat that gone by warm sea water. overall heat transfer coefficient and effectice surface area from evaporator are related with heat transfer rate according to the following formula :

$$Qe = U_e\, A_e\, DTlm_e \qquad (2)$$

Where ΔTlm_e is logarithmic mean temperature difference that passes evaporator and can be written as follows :

$$\Delta Tlm e = \frac{(Twsi - Te) - (Twso - Te}{ln\frac{Twsi - Te}{Twso - Te}} \qquad (3)$$

And effective thermal conductance Ue.Ae can be written as follows:

$$\frac{1}{UeAe} = \frac{1}{hwf\ Ae} + \frac{1}{hws\ Ae} \qquad (4)$$

Equation of energy balance in condenser is basically same with the evaporator and can be written as follows :

$$Qc = \dot{m}_{wf}\ (h_2 - h_3) = \dot{m}_{cs}\ c\ (T_{csi} - T_{cso}) \qquad (5)$$

While the value of effective thermal conductance from condenser relating to heat transfer rate:

$$Qc = U_c\ A_c\ \Delta Tlm_c \qquad (6)$$

Where ΔTlm_c is logarithmic mean temperature difference that passes condenser and can be written as follows :

$$\Delta Tlmc = \frac{(Tcsi - Tc) - (Tcso - Tc}{ln\frac{Tcsi - Tc}{Tcso - Tc}} \qquad (7)$$

Effective thermal conductance in condenser can be determined with this following formula :

$$\frac{1}{UcAc} = \frac{1}{hwf\ Ac} + \frac{1}{hws\ Ac} \qquad (8)$$

2. Pump

After the working fluid is condensed, then it will be pumped to evaporator. The energy balance in pump can be written as follows :

$$W = \dot{m}_{wf}\ (h_4 - h_3) \qquad (9)$$

Change of enthalphy in pump can be calculated with this following equation:

$$h_4 - h_3 = v_4\ (P_4 - P_3) \qquad (10)$$

with assumption that temperature increasing in pump is relatively small so that can be ignored and specific volume remains the same after passing the pump v_3

$\approx v_4$. Working fluid will be pumped reach the evaporation pressure. Works of pump will be calculated with this following equation :

$$W = \frac{\dot{m}wf.v4(P4-P3)}{\eta pwf} \qquad (11)$$

Where η is efficiency of working fluid pump.

3. Turbine
   The working fluid that has evaporated will move and rotate turbine and go through adiabatic process. Steam pressure in turbin output has same value with saturated pressure in condense temperature. Power output from turbine is connected with generator. Generated power can be written with this following formula :

$$Wt = \dot{m}\, wf\ \eta t\ \eta g\ (h_{1} - h_{2s}) \qquad (12)$$

Where hws is isentropic enthalpy in turbin output and can be calculated with this following formula :

$$h_{2s} = h_{2f} + x_{2s}\, h_{2fg} \qquad (13)$$

Where h_{2f} and h_{2fg} are enthalpy of saturated liquid and enthalpy of evaporation in P_2. Quality of isentropic X_{2s} can be written as follows:

$$X_{2s} = \frac{(s1-s2f)}{S2fg} \qquad (14)$$

4. Thernal Efficiency
   Thermal Efficiency is comparison between netto work and heat energy that entering evaporator. Netto work can be calculated from difference between turbin power and pump power. From the explanation above we can calculated all of them with this following formula :

$$Wnetto = Wt - Wp \qquad (15)$$

$$\eta = \frac{Wnetto}{Qin} \qquad (16)$$

$$\eta = \frac{Wt-Wp}{Qin} \qquad (17)$$

2.2.2 Opened Cycle

The differences between opened cycle and closed cycle is the working fluid of the OTEC. In opened cycle it directly use warm sea water as OTEC working fluid, the steps of the open cycle are :

1. Flash evaporation of warm sea water in a partial vacuum.

2. Expansion of the steam through a turbine to generate power.
3. Condensation of the vapor by direct contact heat transfer to cold sea water.
4. Compression and discharge of the condensate and any residual noncondensable gases

This system called open cycle because the working fluid is discharged after singke pass and has different initial and final thermodynamic state, so the flow path and process are ‘open’.

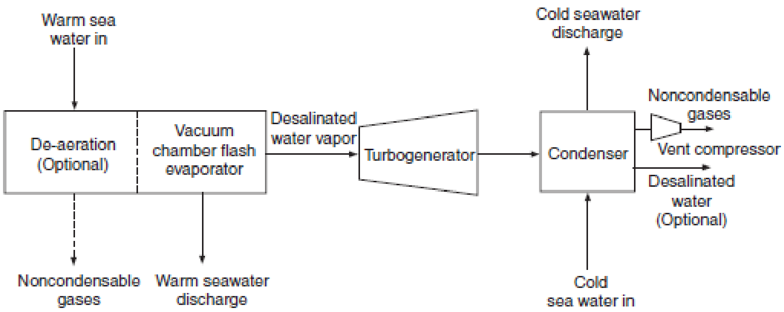

Fig 2.4. Schematic diagram of an open cycle OTEC system

Source:masutani (2001)

Open cycle OTEC eliminates expensive heat exchangers at the cost of low system pressures. Partial vacuum operation has the disadvantage of needing large components to accommodate flow rates as a consequence of the low steam density, volumetric flow rates are very high per unit of electricity generated.

2.2.3 Hybrid Cycle

The hybrid system cycle is uses parts of both open-cycle to produce electricity and closed-cyle to produce fresh water. so it may be able to penetrate the marketplace more readily than plants dedicated just to poduce electricity.

Usually, like in claude cyle, warm surface sea water is flash evaporated in a partial vacuum. Little bit different from closed cyle the heat exchanger use this low pressure steam to evaporate low-boiling point fluid like isobutene. So it reduce the potential for biofouling in the ammonia evaporator.

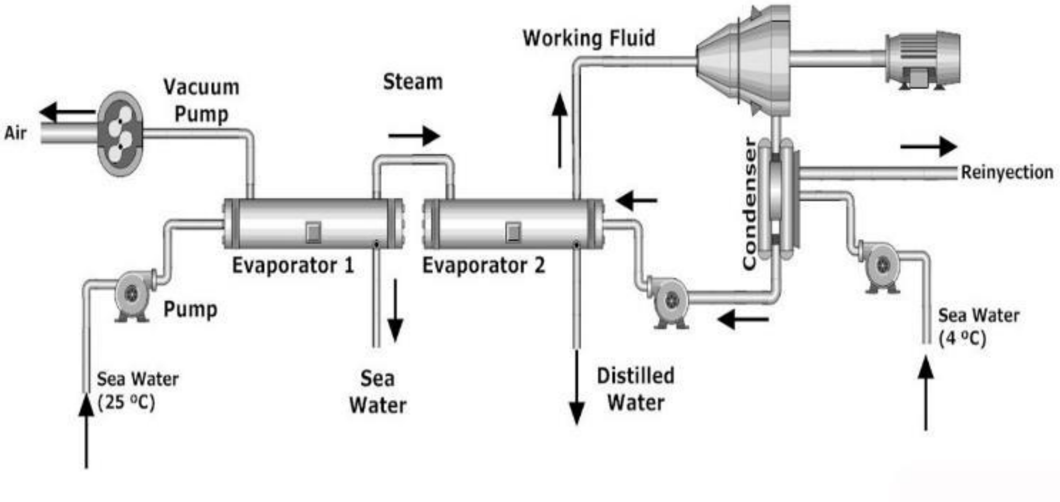

Fig 2.5. Schematic of hybrid cycle

Source : http://proyectos2.iingen.unam.mx/IIDEA/otec_plants.html

2.3 Main Characteristics of Working Fluid

2.3.1 Isobutane C_4H_{10}

Isobutane C_4H_{10}, also known as methylpropane is an alkane with four carbons originally called Butane. Alkanes are chains of carbons atom has as many hydrogen atoms attached as possible. This means that all of the bonds between carbon atoms are single bonds (no double bonds). Such a molecule is said to be saturated (Alkhailidi, M. Qandil, and H. Qandil, 2014).

The advantage of isobutene is a colorless, odorless gas. It is easily liquefied under pressure, and the loquid becomes a gas immediately when the pressure is released. So it commonly use in lighters and camp stoves as a fuel. It also used as propellant in some hair spray breath freshners.

Dry isobutene or R-600a use as refrigant has negligible ozone depletion potential and very low global warming potential. It can use as a good replacement for R-12, R-22, R-134a, and other chloro-fluoro-carbon or hydro-fluoro-carbon refrigants in most conventional stationary refrigation and air conditioning systems.

Table 2.2 Main properties for isobutene

PC	• 58.123 g/mol
Solid Phase	• Latent heat of fusion (1,013 bar, at triple point) : 78.115 kJ/kg
Liquid Phase	• Liquid density (1.013 bar at boiling point) : 593.4 kg/m^3 • Liquid/gas equivalent (1.013 bar and 15^{o}C (59^{o}F)) :

	236 vol/vol • Boiling point (1.013 bar) : -11.7^{o}C
Critical point	• Critical temperature : 134.9^{o}C • Critical pressure : 36.48 bar
Gaseous phase	• Gas density (1.013 bar at boiling point): 2.82 kg/m3 • Gas density (1.013 bar and 15^{o}C) : 2.51 kg.m3 • Compressibility factor (z) (1.013 bar and 15^{o}C) : 0.9675 • Specific gravity (air=1)(1.013 bar and 21^{o}C) : 2 • Specific volume (1.013 bar and 21^{o}C): 0.406 m3/kg • Heat capacity at fixed P (C_p) (1.013 bar and 15^{o}C) ; 0.095kJ/(mol.K) • Heat capacity at fixed V (C_v) (1.013 bar and 15^{o}C): 0.086 kJ.(mol.K) • Ratio of specific heats (γ: Cp/Cv) (1.013 bar and 15^{o}C) : 1.095845 • Viscosity (1.013 bar and 0^{o}C : 0.0000689 Poise • Thermal conductivity (1.013 bar and 0^{o}C ; 13.97 mW/(m.K)
Miscellanous	• Solubility in water (1.013 bar and 20^{o}C) : 0.0325 vol (isobutene)/vol(water) • Auto-ignition temperature : 460^{o}C

2.4 OTEC Power Plant Projects

2.4.1 OTEC Projects in Okinawa, Japan

Kumejima Island, Okinawa Prefecture is located at the southernmost tip of Japan, and features high seawater temperatures that make it an ideal spot for OTEC. A demonstration plant for OTEC went into operation there in June 2013. The temperature on warm seawater is 23.5^{o}C and the temperature of cold seawater is 8.7^{o}C. But since the temperature of ocean water is lower than that of hot springs water a high performance heat exchanger known as titanium plate heat exchanger is used.

The demonstration facilities that recently went into operation have a generating capacity 50 KW. The plan is to verify them through the end of 2014 and to develop a 1000 KW – class plant by applying the data from this. At Kumejima Islanda the plan is not only use the deep ocen water that is drawn up to generate power, but also to use it in an integrated for the aquaculture of shrimp and seaweed and for air condinitioung in plant factories and so forth.

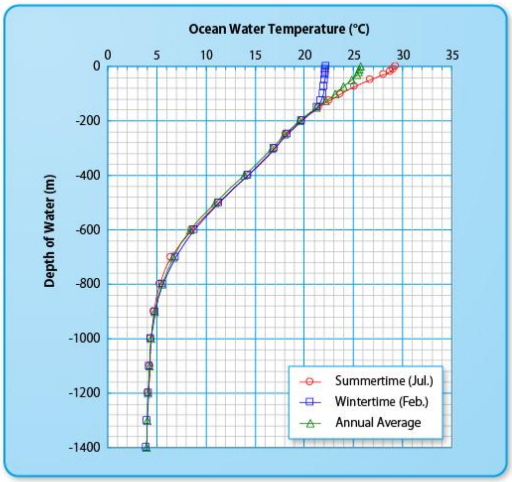

Fig 2.6 Ocean water temperature distribution by season in the ocean regions around Kumejima Island.

Source : http://otecokinawa.com

Fig 2.7 OTEC Power Plant 50KW in Okinawa, Japan

Source : http://otecokinawa.com

2.4.2 Makai Ocean Engineering's OTEC in Hawaii, USA

Makaii Ocean Engineering's OTEC power plant is the world's biggest operational facility of its kind. With an annual power generation capacity of 100 KW, which is sufficient to power 120 homes in Hawaii.

Makai's OTEC plant is a closed-cycle facility that uses an ammonia fluid to drive the turbine-generator. Makai's Otec plant dorms part of its OTEC heat exchanger test facility and marine corrosion lab, named Ocean Energy Research Center (OERC), located at the NELHA site, which was opened in 2011 following the award of a fund by the US Navy in 2009.

Fig 2.8 Makai's OTEC Power Plant in Hawaii, USA
Source : *https://www.makai.com/ocean-thermal-energy-conversion*

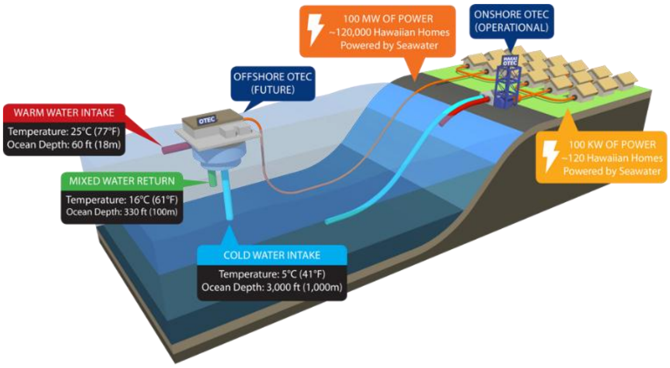

Fig 2.9 Makai's OTEC model
Source : https://www.makai.com/ocean-thermal-energy-conversion

2.5 Previous Research

2.5.1 Potential Ocean Thermal Energy Conversion (OTEC) in Bali

This research is written by Adrian Rizki Sinuhaji in 2015. This paper aim to determine the potential and the provision of new and renewable energy in Indonesia. It choose closed cyle OTEC system with ammonia as a working fluid. The maximum efficiency of carnot engine is obtained in the North Bali Sea by 0,788813. Based on data, surface water in north Bali sea X0 = 30.3^{o}(From *: Balai Riset dan Observasi Kelautan)* and the calculated maximum depth is 600 meters.

Table 2.3 Data od surface water in North Bali

No	Depth	B	X0	Xn
1	0	0	30,3	30,3
2	100	-0.047	30,3	25,63

3	200	-0.068	30,3	16,74
4	300	-0.061	30,3	12,06
5	400	-0.054	30,3	8,78
6	500	-0.047	30,3	6,89
7	600	-0,04	30,3	6,4

From this research the result of net power is 69,4 kW with efficiency 3.1%

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2.5.2 Simulasi Sistem Pembangkit OTEC Siklus Tertutup dengan Variasi Fluida Kerja Ammonia (NH_3) dan Refrigerant (R-12. R-22, R-23, R-32, R134a) Menggunakan Cycle Tempo

This research is written by Kevin Kurniawan S. this research will be simulated OTEC powerplant closed cycle using variated working fluid. The result of this research shows that ammonia is the best working fluid to be applied on OTEC power plant closed cycle.

Table 2.4. Net Power of variant working fluid

No	Working Fluid	Mwf (kg/s)	W Turbine (MW)	W Parasitic (MW)	W Netto (MW)
1	Ammonia	412	18,3	0,49	17,8
2	R-22	1590	11,8	0,56	11,19
3	R-32	1038	11,4	0,69	10,7
4	R-12	1401	10,3	0,56	9,8
5	R-134a	2029	9,8	0,65	9,39
6	R-23	550	3,14	0,94	2,2

2.5.3 Analysis of OTEC Power Plant using Isobutane as the Working Fluid

This researches is written by A.Alkhalidi, M. Qandil, H. Qandil. This research is about use of organic isobutene for closed-cylcle OTEC onshore plant that delivers 110 MW electric powers. In isobutene cycle, hot ocean surface water is used to vaporize and to superheat isobutene in a heat exchanger. Isobutene vapor then expands through a turbine to generate useful power. The exhaust vapor is condensed afterwards, using the cold deeper ocean water, and pumped to a heat exchanger to complete a cycle.

Overall results showed in table 4 represent total values for the whole suggested OTEC plant (consisting of the combined 24 sub-plants each with separate evaporator, turbine, condenser and circulating pump)

Table 2.5 Results Summary for Isobutane 110 MW OTEC Power Plant

Output work	121.8 MW
Isobutane MFR (Total)	7164 kg/s
Warm water VFR (Total)	112.3 m^3/s
Cold Water VFR (Total)	167.2 m^3/s
Isobuane Pump Work (24 pumps Total)	1.73 MW
Cycle efficiency	5.1 %

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CHAPTER III
METHODOLOGY

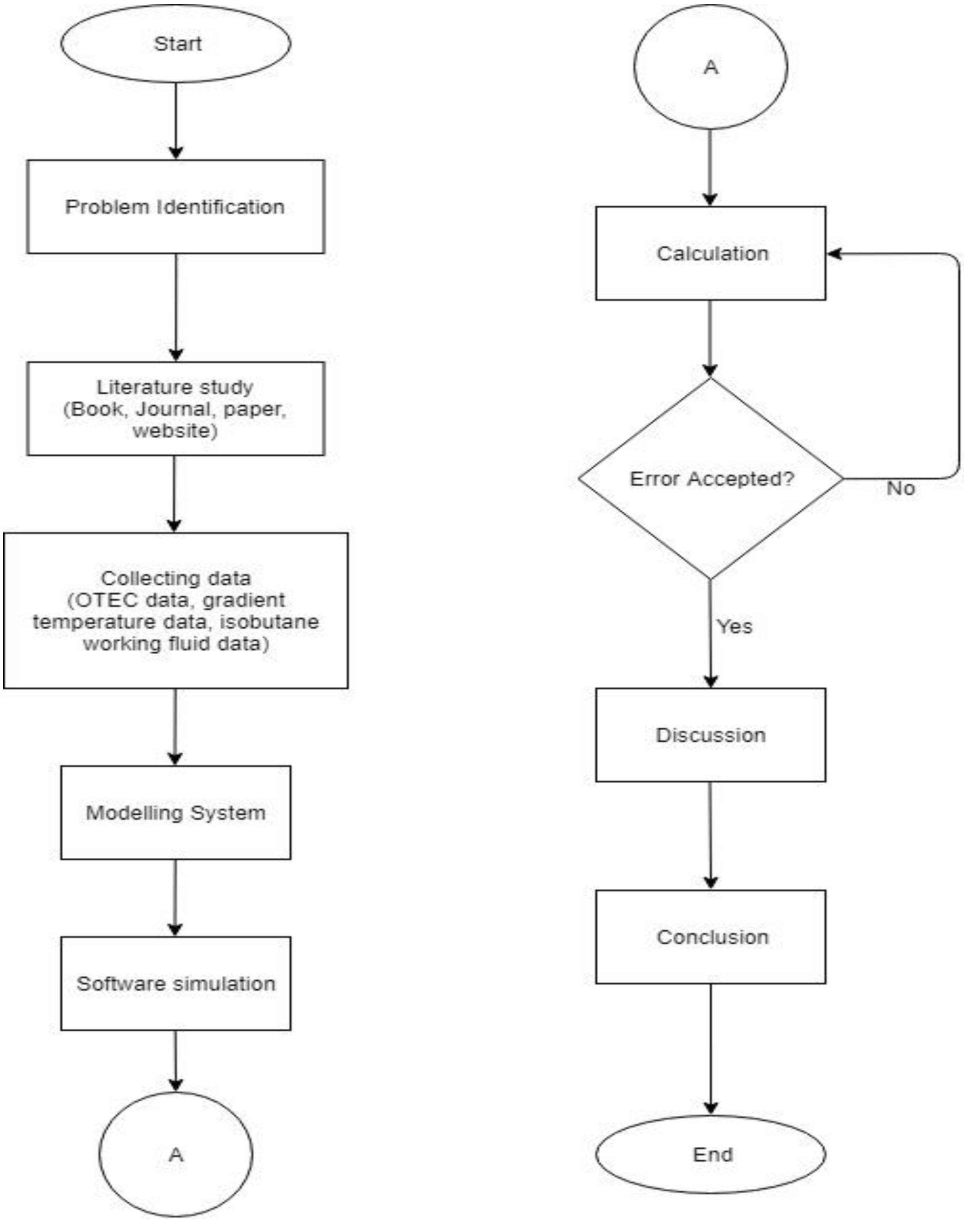

Fig 3.1 Methodology Flowchart

3.1 Problem Identification

The choice of work fluid must be adjusted to the various limitations that exist in the system. The physical properties of the working fluid such as boiling point, density, critical pressure, etc. greatly determine the performance of a generating system. In addition, the application of OTEC technology must be adapted to the climatic conditions and the marine waters of the area to be applied. This is related to sea surface temperature, water conditions, depth of the sea, and environmental conditions.

3.2 Literature Study

Literature study is done after decided the problem. The study is done with gathering and studying material that related to this research. The materials are books, journals, papers, bachelor thesis, and article on the internet.

3.3 Data Collection

The choice of work fluid must be adjusted to the various limitations that exist in the system. The physical properties of the working fluid such as boiling point, density, critical pressure, etc. greatly determine the performance of a generating system. In addition, the application of OTEC technology must be adapted to the climatic conditions and the marine waters of the area to be applied. This is related to sea surface temperature, water conditions, depth of the sea, and environmental conditions. Data collection conducted was obtained from various journals and research on OTEC that had been done previously, both OTEC in the country and abroad. From the collection of data obtained characteristics such as pressure and OTEC work temperature in general. In addition, various characteristics of seawater are obtained at certain surfaces and depths which greatly determine the success of the operation of the system. Seawater temperature taken is based on data obtained from research conducted by Adrian Rizki Sinuhaji on "Potential Ocean Thermal Energy Conversion (OTEC) in Bali."

3.4 Modelling System

This step is to design the OTEC closed cycle system with software called cycle tempo and from other references OTEC design which has existed.

3.4.1 System Model

System modeling will be designed with the closed cycle OTEC system. Components that will be used include condensers, evaporators, turbines, generators, working fluid pumps, pumps for sea water, and warm sea water inlets and outlets and cold sea water.

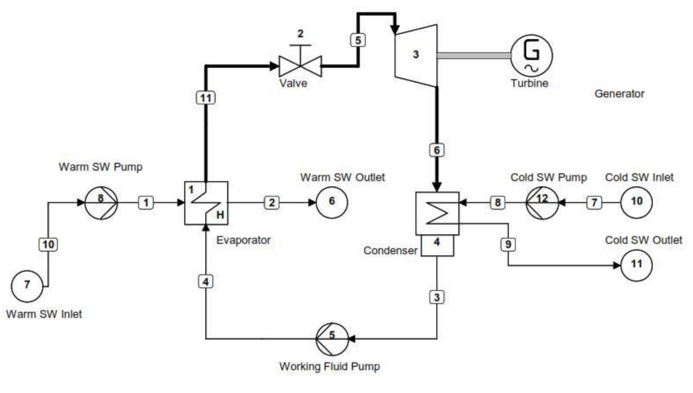

Figure 3.2 OTEC System Model

3.4.2 Seawater Pump System

Seawater pumps on the OTEC system have two pumps. The first pump for deep sea water and the second pump for surface sea water. Pipe diameter is 0.5m for warm sea water and 0.7m for cold sea water

With the assumption used is the flow in a steady state, the flow in the form of incompressible flow, dynamic viscosity and constant density of seawater.

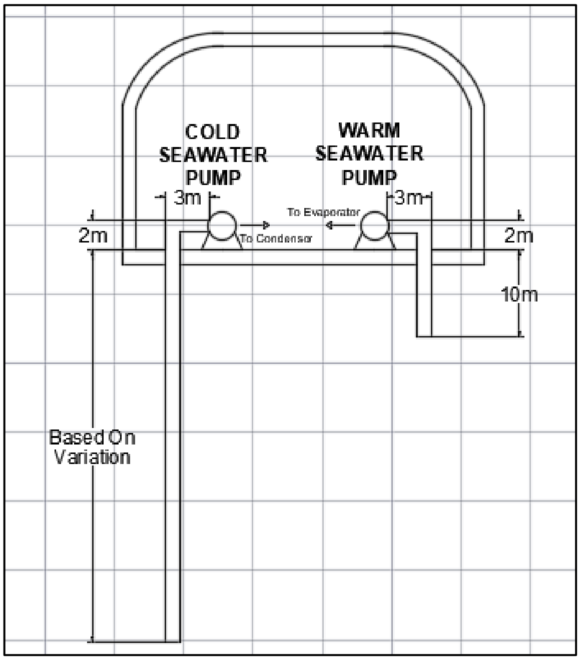

Figure 3.3 OTEC Sea water pump system

3.4.3 Prameter set

Table 3.1 Design Set Parameter

Operational Parameter	Value	Unit
Fixed Variable		
Depth of warm seawater	10	m
Input Temperature of warm seawater	29	^oC
Output Temperature of warm seawater from evaporator	25	^oC
Output Temperature of working fluid from evaporator	25	^oC
Independent Variable		
Cold and warm seawater mass flow rate	250;500;750;1000;1250	kg/s
Depth of seawater bed	1000;900;800;700;600;500;400;300;200;100	m
Produced		
Working fluid mass flow rate	*	kg/s
Power Generated	*	kW
Power input for CW,SW, and WF Pump	*	kW
Efficiency	*	

Generator		
Generator eff.	100%	
Turbine		
Efficiency	97%	
Working Fluid Circulating Pump		
Efficiency	85%	
Ocen Water Pump		
Efficiency	90%	

3.4.4 Turbine Temperature Drop

Based on the research "A Preliminary Assessment of OTEC Resources" by Nihous. The decreasing temperature of working fluid after passing through the turbine can be seen in the OTEC temperature ladder stage namely DT/2. Where y is the ratio between cold sea water mass flow rates and warm sea water mass flow rates.

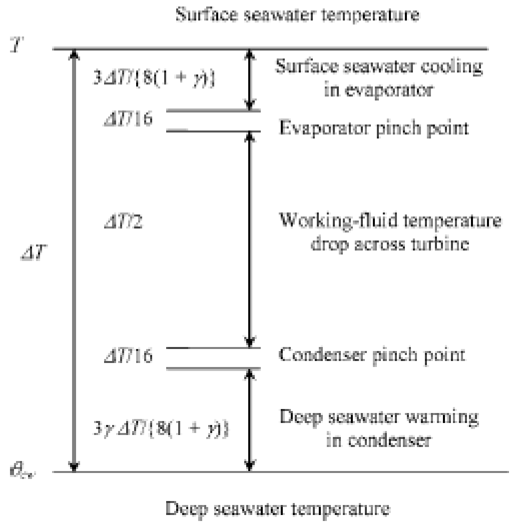

Fig.3.4 OTEC Temperature ladder

From the table above, the temperature drop can be calculated from each determined scenario. For the example is scenario 1.

T warm sea water = 29^{o}C
T cold sea water = 5^{o}C
DT = 24^{o}C
Working fluid temperature drop across turbine = DT/2 = 12^{o}C

We have set the temperature input of turbine is 25^{o}C so the temperature out of turbine is temperature input of turbine minus working fluid temperature drop across turbine.

Tin turbine = 25^{o}C
DTwf across turbine = 12^{o}C
Tout Turbine = Tin turbine - DTwf across turbine = 13^{o}C

Table 3.2 Tout Turbine for each scenario

Scenario	Twsw (^{o}C)	Tcsw (^{o}C)	DTsw (^{o}C)	DTwf across turbine (^{o}C)	Tout Turbine (^{o}C)
1	29	5	24	12	13.00
2	29	5.5	23.5	11.75	13.25
3	29	6.5	22.5	11.25	13.75
4	29	7.75	21.25	10.625	14.38
5	29	8.25	20.75	10.375	14.63
6	29	9	20	10	15.00

Scenario	Twsw (^oC)	Tcsw (^oC)	DTsw (^oC)	DTwf across turbine (^oC)	Tout Turbine (^oC)
7	29	10	19	9.5	15.50
8	29	11	18	9	16.00
9	29	13	16	8	17.00
10	29	22	7	3.5	21.50

3.5 Analysis and Processing Results Data

After all the data is collected, it then performs the calculation and analysis process. The results of these calculations will be processed in the form of tables and graphs. The results obtained are calculated manually using thermodynamic equations and heat transfer to obtain accurate results.

3.6 Compilation of Reports

All results of calculations and analysis will be poured into a report that is arranged systematically and in accordance with the final report in general. This step is to analyze the results such as efficiency system, the power produced, and the energy needed set out in tables and graphs. Results of all work previously will be discussed. Discussion based on calculations and simulations that have been carried out include pressure and power requirements sea water pump, variations in working fluid against the ratio of the flow rate of sea water warm and cold sea water. This will lead to differences working pressure on the evaporator and condenser for each variation do. This will have an impact on the performance of the generating system such as turbine power produced, pump work from fluid work, net work, and system thermal efficiency.

3.7 Conclusion

This is the last step that conclude all the process and become tha answer for problem statement.

CHAPTER IV
ANALYSIS STUDY

4.1 OTEC Location Selection

Indonesia has a sea surface that is always exposed to the sun in constant from year to year. This situation is an advantage when you want to make an OTEC installation that utilizes temperature differences between sea levels and deep sea water temperatures. Therefore, when determining the location for construction OTEC must have temperature data for each depth of the sea.

Based on the Temperature Distribution Modeling research on the depth of the sea in the south of the island of Java with the Gauss Method - Newton by Maruta Prima Jaya et al. Obtained secondary data measured by expendable bathytherograph (XBT) covering temperature data on two coordinate points. The first coordinates are at 8 ° 13 '1.20 "oLS and 105 ° 32' 60.0" oBT. The second coordinate point is located at 7 ° 35 '60.0 "oLS and 105 ° 19' 58.8" oBT. The first point was taken on December 31, 2014, April 12, 2011, and April 11, 2004. The second point was taken on January 1, 2012 and January 29, 1992. The author chose the first coordinate point as the OTEC location.

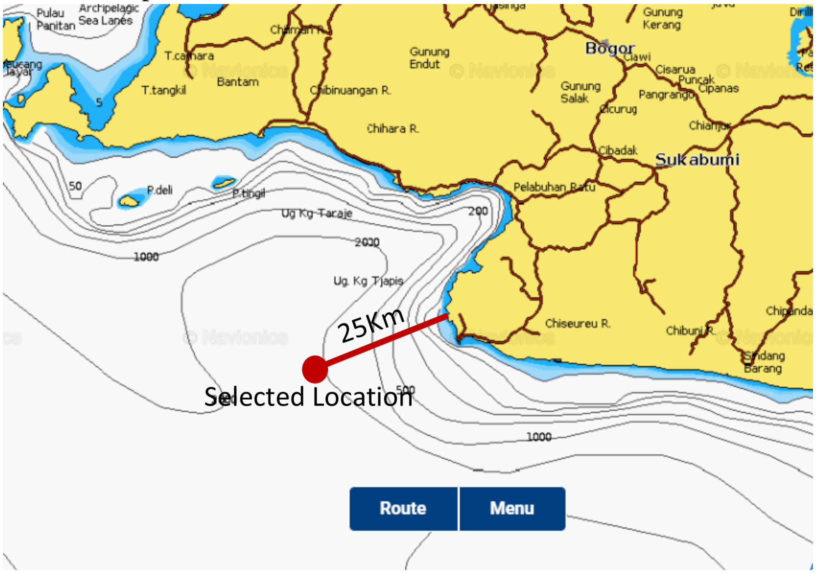

Figure 4.1 Selected Location, Teluk Cikepuh

Source: webapss.navionics.com

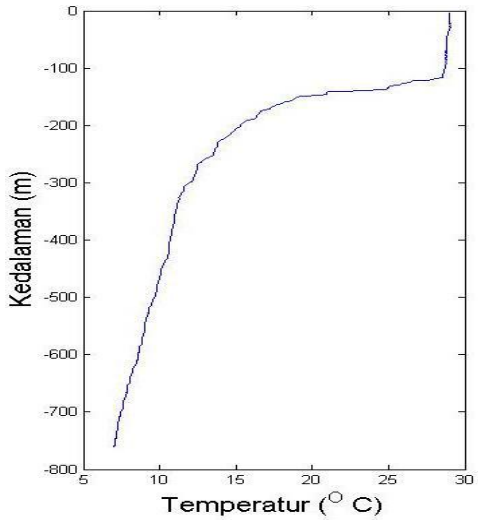

Fig 4.2 Observation Curve of First Coordinate in 31st December 2014

Source: Maruta Prima Jaya (2017)

In addition to retrieving secondary data from research above the temperature of each ocean depth data can be taken using software called Global Marine Agro Atlas released by Argo Company, an American oceanographic company. The Global Marine Argo Atlas application uses a location survey system and periodic recording of world regions.

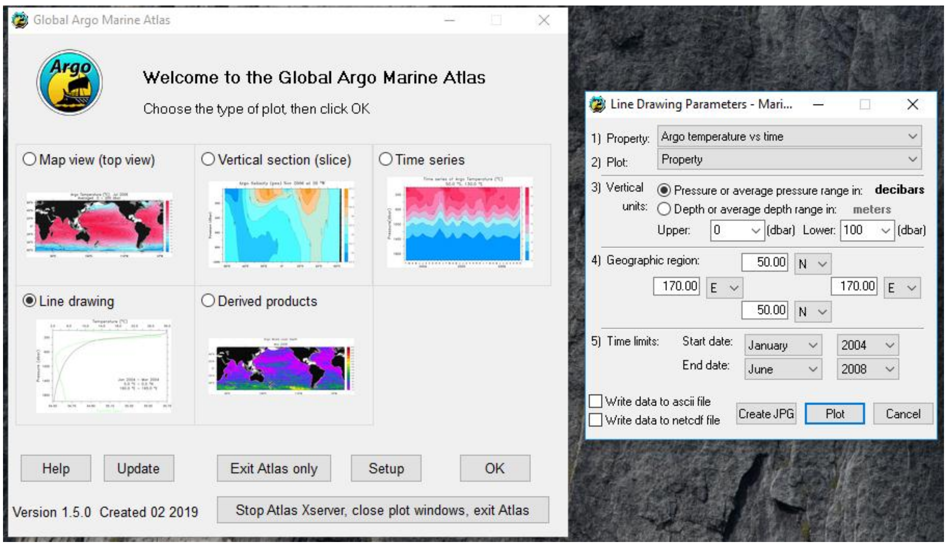

Fig 4.3 Main menu of Global Marine Agro Atlas Software

Source: Global Marine Agro Atlas Software

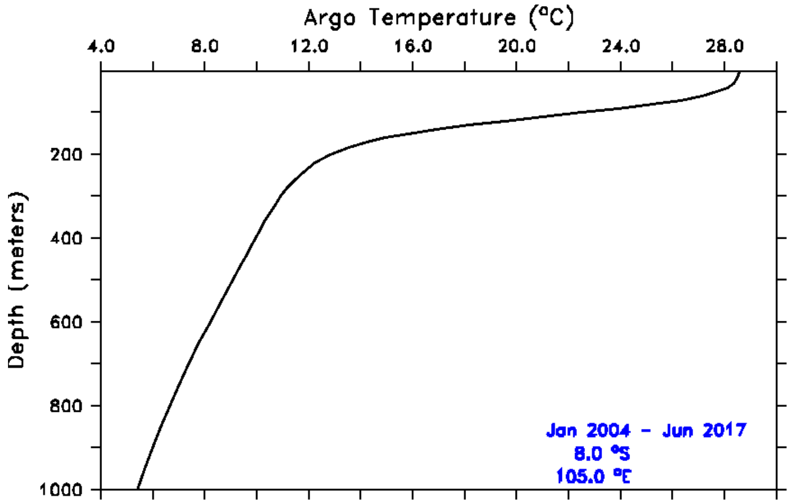

Fig 4.4 Sea depth temperature data from Global Marine Agro Atlas Software

Source: Global Marine Agro Atlas Software

After determining the next location, determine the surface sea water temperature and deep sea water according to the specified variables, namely 1000m, 900m, 800m, 700m, 600m, 500m, 400m, 300m, 200m and 100m for the depth of deep sea water and 10m for the depth of surface sea water . To determine the temperature variations in the depth of sea water can use graphs of temperature data for each depth of sea water from Global Marine Agro Atlas Software.

Table 4.1 Selected Location Temperature Data

Scenario	Surface depth (m)	Twsw (°C)	Seabed depth (m)	Tcsw (°C)	DTsw (°C)
1	10	29	1000	5	24
2	10	29	900	5.5	23.5
3	10	29	800	6.5	22.5
4	10	29	700	7.75	21.25
5	10	29	600	8.25	20.75
6	10	29	500	9	20
7	10	29	400	10	19
8	10	29	300	11	18
9	10	29	200	13	16
10	10	29	100	22	7

4.2 Simulation Results

OTEC simulation in this study uses Tempo 5.0 Cycle. In this discussion chapter, the simulation results of 10 scenarios will be displayed in a variation of the sea water mass flow rate at 250 kg / s with predetermined design parameters. after that the author will compare the power output of each scenario.

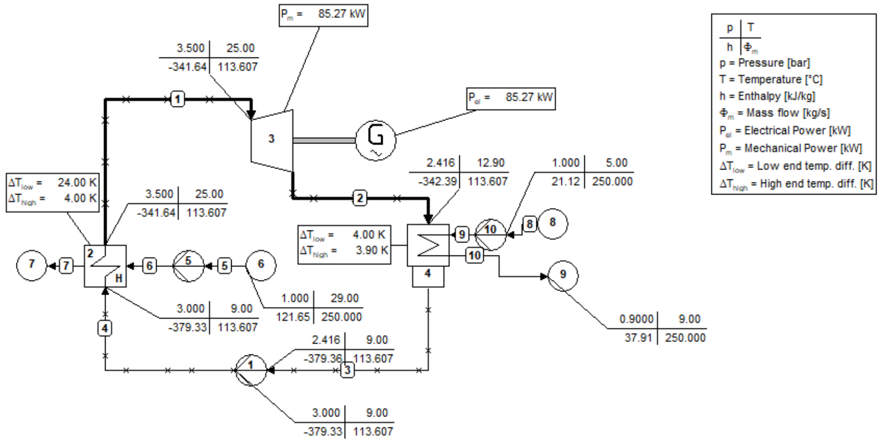

Fig 4.5 Simulation scenario 1 d=1000m, seawater flowrate=250 kg/s

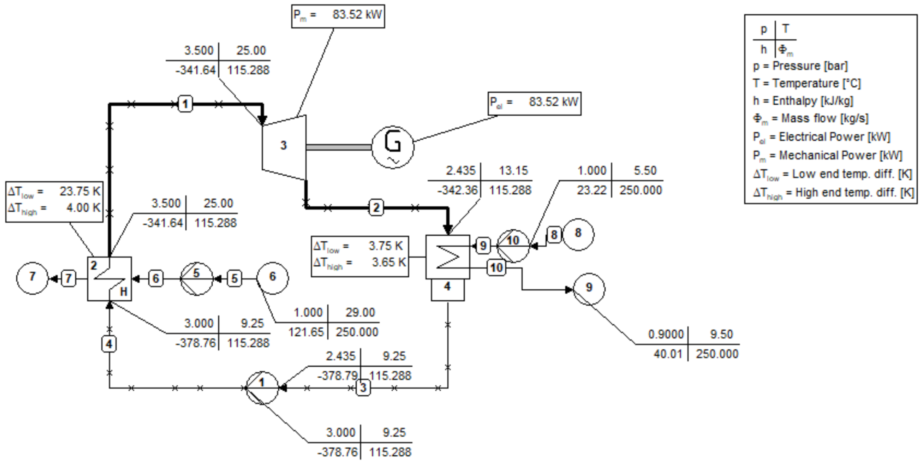

Fig 4.6 Simulation scenario 2 d=900m, seawater flowrate=250 kg/s

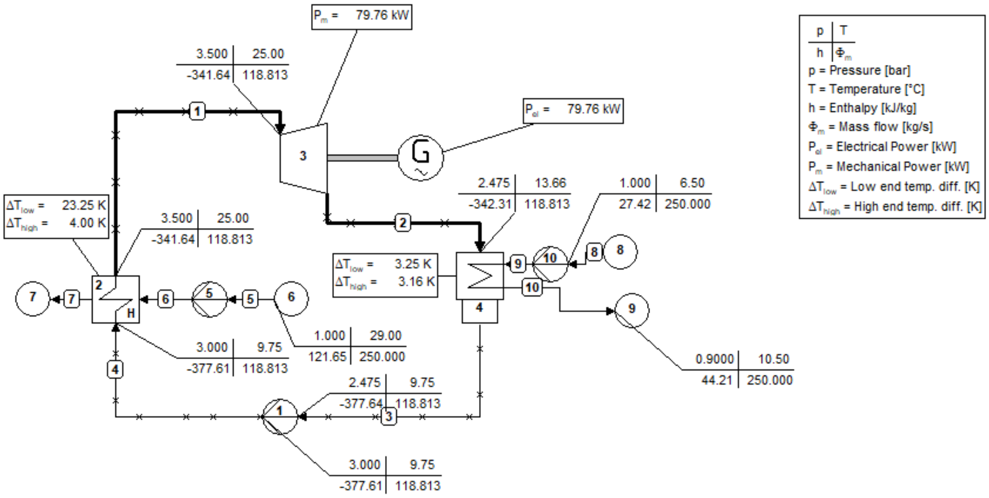

Fig 4.7 Simulation scenario 1 d=800m, seawater flowrate=250 kg/s

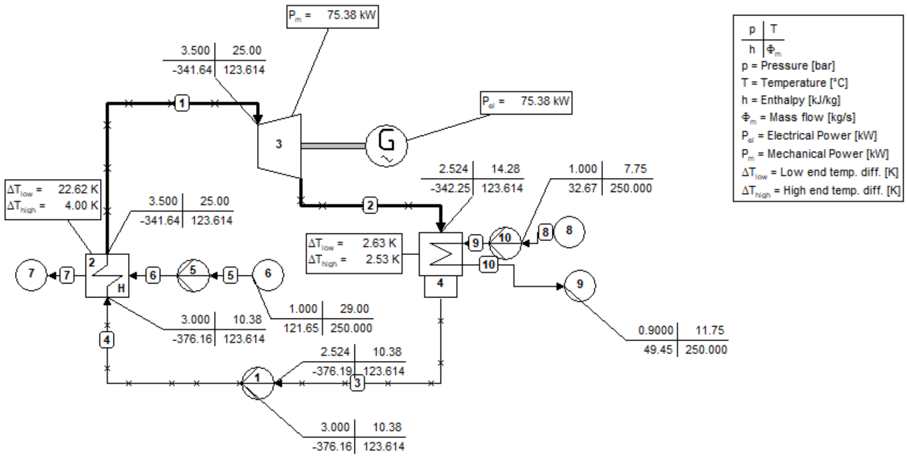

Fig 4.8 Simulation scenario 1 d=700m, seawater flowrate=250 kg/s

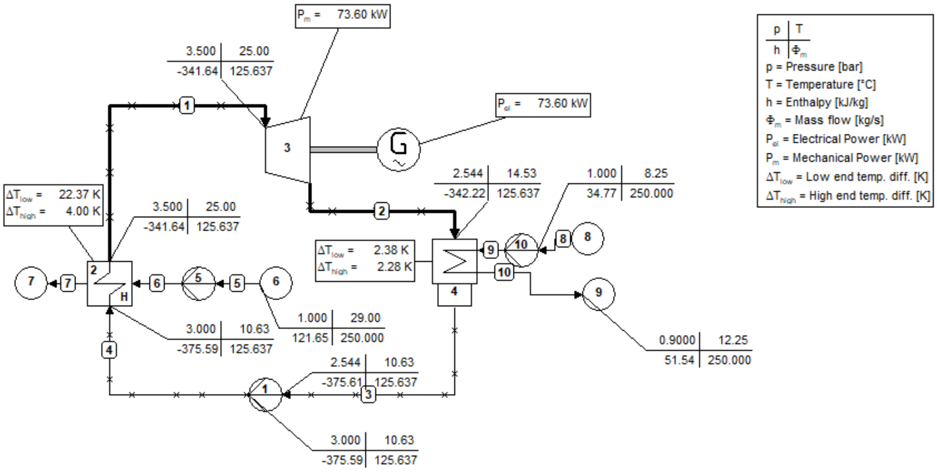

Fig 4.9 Simulation scenario 5 d=600m, seawater flowrate=250 kg/s

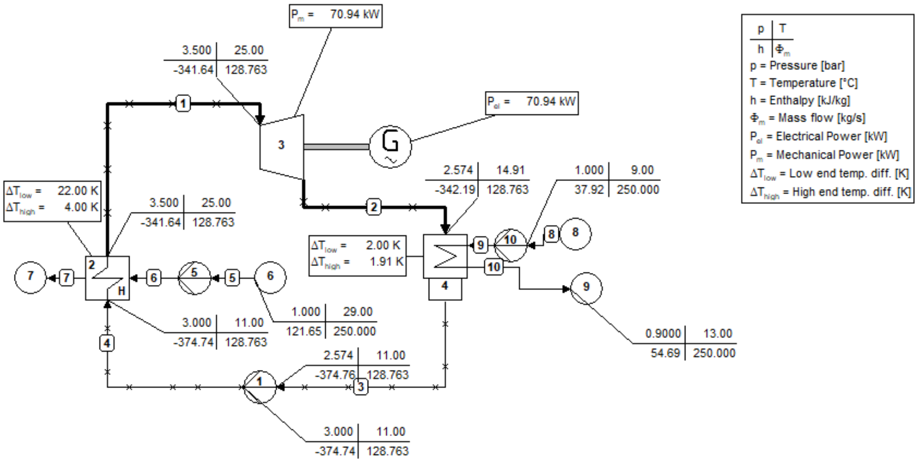

Fig 4.10 Simulation scenario 6 d=500m, seawater flowrate=250 kg/s

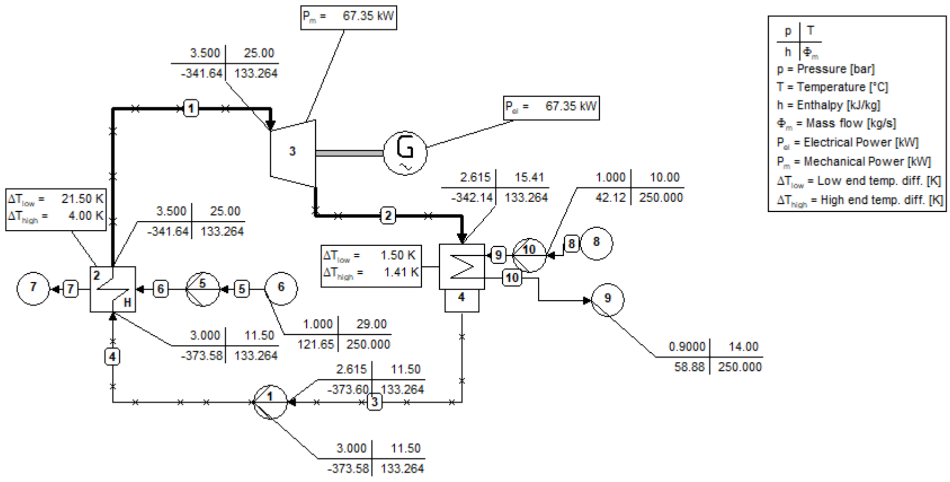

Fig 4.11 Simulation scenario 7 d=400m, seawater flowrate=250 kg/s

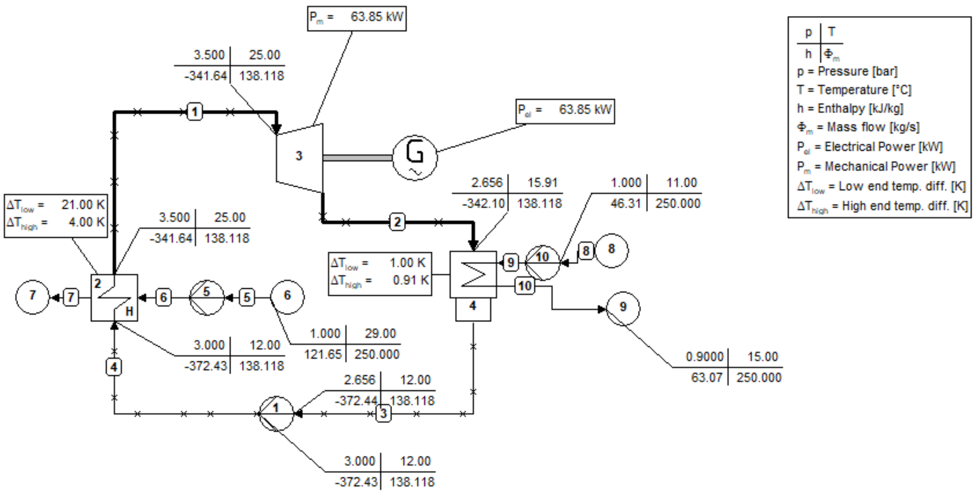

Fig 4.12 Simulation scenario 8 d=300m, seawater flowrate=250 kg/s

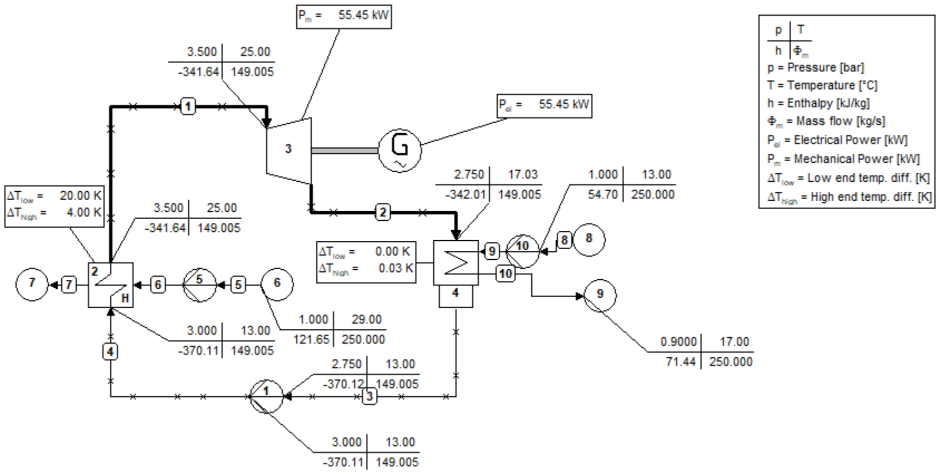

Fig 4.13 Simulation scenario 9 d=200m, seawater flowrate=250 kg/s

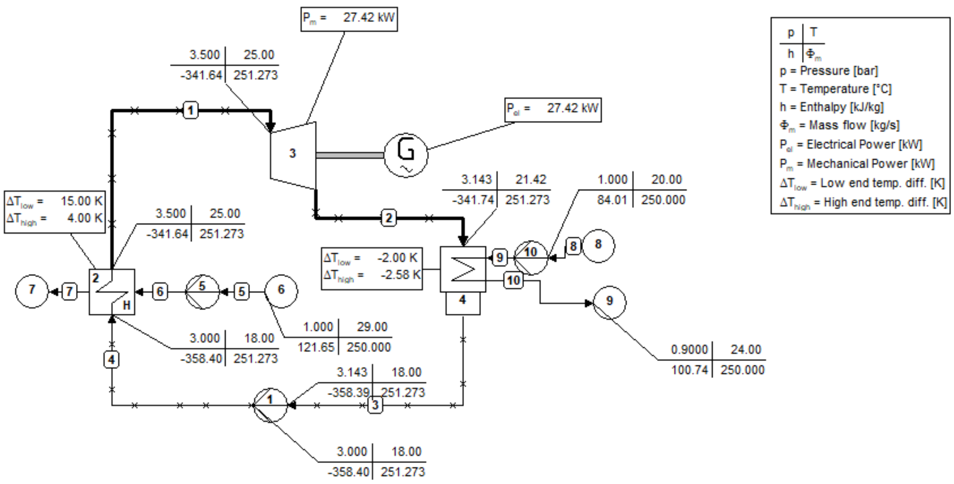

Fig 4.14 Simulation scenario 10 d=100m, seawater flowrate=250 kg/s

From the simulation results can be obtained the output power of each scenario. The following table will display the output power comparison of each scenario.

Table 4.2 Power output for each simulation of scenario

Scenario	Sea Depth (m)	msw (kg/s)	mwf (kg/s)	Simulation Gross Power Output (kW)
1	100	250	113.06	85.27
		500	226.13	170.53
		750	339.19	255.8
		1000	452.26	341.07
		1250	565.32	426.34
2	900	250	115.29	83.52
		500	230.58	167.03
		750	345.86	250.55
		1000	461.15	334.06
		1250	576.44	417.58
3	800	250	118.81	79.76
		500	237.63	159.52
		750	356.44	239.29
		1000	475.25	319.05
		1250	594.07	398.81
4	700	250	123.61	75.38
		500	247.23	150.77
		750	370.84	226.15
		1000	494.46	301.54
		1250	618.07	376.92
5	600	250	125.64	73.6
		500	251.27	147.2
		750	376.91	220.8
		1000	502.55	294.4
		1250	628.18	368
6	500	250	128.76	70.94
		500	257.53	141.88
		750	386.29	212.81
		1000	515.05	283.75
		1250	643.81	354.69
7	400	250	133.26	67.35
		500	266.53	134.7

Scenario	Sea Depth (m)	msw (kg/s)	mwf (kg/s)	Simulation Gross Power Output (kW)
		750	399.79	202.05
		1000	533.06	269.4
		1250	666.32	336.75
8	300	250	138.12	63.85
		500	276.24	127.7
		750	414.36	191.55
		1000	552.47	255.4
		1250	690.59	319.25
9	200	250	149.01	55.45
		500	298.01	110.89
		750	447.01	166.34
		1000	596.02	221.78
		1250	745.02	277.23
10	100	250	251.27	27.42
		500	502.55	54.85
		750	753.89	82.27
		1000	1005.09	109.7
		1250	1256.37	139.21

From the table above, power output disebut gross power output karena belum dikurangi kebutuhan power pompa.perhitungan net power output akan dibahas di sub bab selanjutnya

From the table above, it can be seen that the closer to the sea surface the smaller the power produced. we can take an example by taking one sample from each scenario. We take samples when the mass flow rate of seawater is 250 kg / s from each scenario like following below :

- The first scenario at a depth of 1000 m with a mass flow rate of sea water 250 kg / s produces a gross power output of 85.27 kW.
- The second scenario at a depth of 900 m produces a gross power output of 83.52 kW.
- The third scenario at a depth of 800 m produces a gross power output of 79.76 kW.
- The fourth scenario at a depth of 700 m produces a gross power output of 75.38 kW.
- The fifth scenario at a depth of 600 m produces a gross power output of 73.6 kW.

- The sixth scenario at a depth of 500 m produces a gross power output of 70.94 kW.
- The seventh scenario at a depth of 400 m produces a gross power output of 67.35 kW.
- The eighth scenario at a depth of 300 m produces a gross power output of 63.85 kW.
- The ninth scenario at a depth of 200 m produces a gross power output of 55.45 kW.
- The tenth scenario at a depth of 100 m produces a gross power output of 27.42 kW.

Furthermore, if we examine further the greater the mass flow rate of working fluid, the greater the gross power output produced. We take the example in scenario 1 like following below:

- When the mass flow rate of the working fluid is 113.06 kg / s gross power output is 85.27 kW.
- When the mass flow rate of working fluid is 226.13 kg / s gross power output is 170.53 kW.
- When the mass flow rate of working fluid is 339.19 kg / s the output gross power is 255.8 kW.
- When the working fluid mass flow rate is 452.26 kg / s output gross power is 341.07 kW.
- When the mass flow rate of working fluid is 565.32 kg / s gross power output is 426.34 kW.

It can be concluded that in the first scenario at a depth of 1000m with mass low rate working fluid at 565.32 kg / s it produced the largest gross power output with 426.34 kW.

4.3 Calculation

4.3.1 OTEC Power output ,Working Fluid Pump Power Input, and Heat Absorbed by Heat Exchanger

First we look at OTEC Closed cycle system schematic diagram below

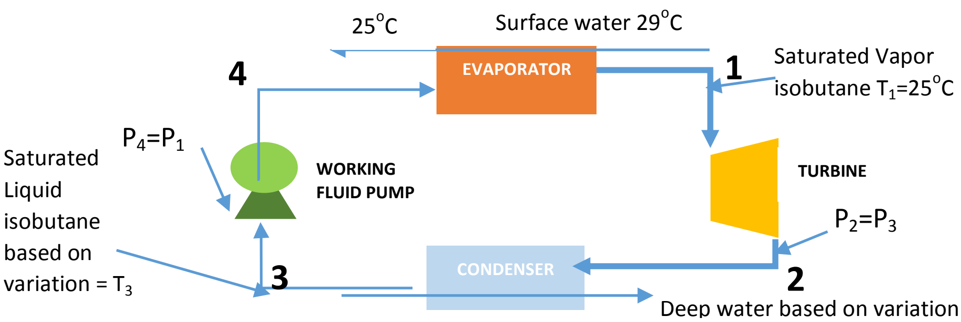

Figure 4.15 OTEC closed cycle system schematic

First we decided the temperature input and ouput of cold sea water then saturated liquid isobutene temperature based on variation of depth of sea water. the depth variation of sea water will have an effect on the temperature of deep sea water because the deeper the sea the lower the temperature. The following are the results of deep sea temperature input and output and temperature saturated liquid isobutene based on variations in ocean depth.

Table 4.3 Tin Cold Seawater and T_1 Saturated Liquid Isobutane

Scenario	**Seabed depth (m)**	**T_{in} csw (°C)**	**T_3 Saturated Liquid Isobutane (°C)**
1	1000	5	9
2	900	5.5	9.25
3	800	6.5	9.75
4	700	7.75	11.75
5	600	8.25	10.38
6	500	9	10.63
7	400	10	11.5
8	300	11	12
9	200	13	13
10	100	22	18

At point 1 in figure 4.7, the temperature of working fluid is reduced to 25°C due to heat exchange from the condensed isobutene to the warm surface. It cannot be the exact temperature of the surface water temperature.

For example in this calculation we will analyze scenario 1 (d=1000m). the enthalpy value at each point can be determined through the simulation results. The following is the enthalphy result of scenario 1 simulation.

h1 = -341.64 kJ/kg

h2 = -342.39 kJ/kg

h3 = -379.36 kJ/kg

h4 = -379.33

Setelah itu dilanjutkan dengan menghitung mass flow rate dari fluida kerja dengan rumus persamaan heat and mass balance pada condenser:

$$\dot{m}_{cold\ water} \times Cp_{cold\ water} \times DT_{cold\ water} = \dot{m}_{isobutane} \times q_{out}$$

Cp_{ww} = 4.18

DT = $4^{o}C$
Q_{out} = h_2-h_3= -342.39 kJ/kg – (-379.36 kJ/kg) = 36.97 kJ/kg

Table 4.4 Isobutane mass flow rate

$\dot{m}_{\text{cold sea water}}$ (kg/s)	$\dot{m}_{\text{isobutane}}$ (kg/s)
250.00	113.06
500.00	226.13
750.00	339.19
1000.00	452.26
1250.00	565.32

Now we can solve for the power output of the turbine with the following equation.

$$\mathbf{WT = \dot{m}_{isobutane} (h_1\text{-}h_2)}$$

Table 4.5 Power Output from Turbine Scenario 1

$\dot{m}_{\text{isobutane}}$ (kg/s)	WT (KW)
113.06	84.79849
226.13	169.60
339.19	254.40
452.26	339.19
565.32	423.99

Next the pump power input and the total pump power will be calculated. The pump power input (Wp) will be calculated along with the enthalphy at point 4.

Total pump power input (Wpump-total) will be calculated along with enthalpphy at point 4.

$$\mathbf{Wpump\text{-}total = \dot{m}_{isobutane} (h_3\text{-}h_4)}$$

Table 4.6 Table of Working Fluid Pump Power Input Scenario 1

$\dot{m}_{isobutane}$ (kg/s)	Wpump-total (KW)
113.06	3.39
226.13	6.78
339.19	10.18
452.26	13.57
565.32	16.96

Next we will be calculated heat supplied to the exchanger (Qh) can be written as:

$$\mathbf{Qh = \dot{m}_{isobutane}\,(h_1\text{-}h_4)}$$

Table 4.7 Heat Absorbed Table Scenario 1

$\dot{m}_{isobutane}$ (kg/s)	Qh (kW)
113.06	4261.41
226.13	8522.81
339.19	12784.22
452.26	17045.63
565.32	21307.03

Before we calculated efficiency of this system we have to calculate power input for sea water pump to get the Wnet. So in the next step we will calculate cold sea water and warm sea water pump power input.

4.3.2 Sea Water Pump Power Input

A. Warm Sea Water Pump Power Input

From the book "Pumps and Compressors" by Ir. Soelarso, to look for energy pumps in kW are obtained from the equation as follows:

$$\dot{W} = ϒ\, x\ Q\ x\ H$$

Where,

$\dot{W}$: Pump Power (kW) H: Pump Total Head (m)
ϒ: Fluid Specific Gravity (kN / m3)
Q: Flow Capacity (m3 / s)

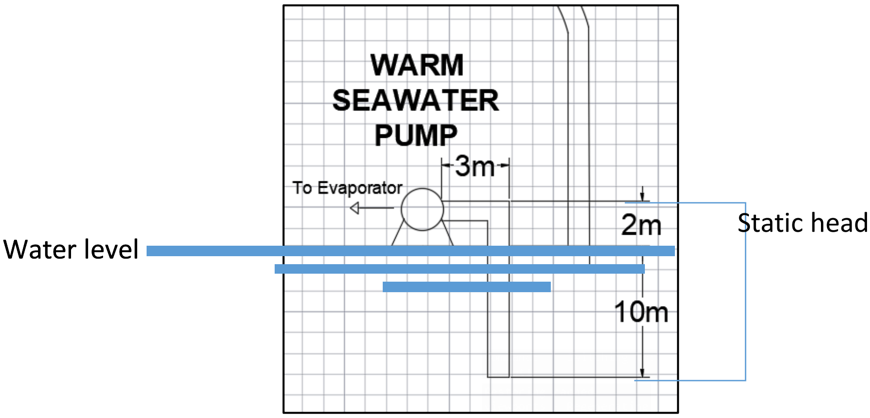

Figure 4.15 Warm Sea Water Pump Scheme

to calculate pump power, you must first know:

1. Sea Water Specific Gravity

   $\rho = 1025 \text{ kg / m}^3$

   $g = 9.81 \text{ m / s}^2$

   then,

   $\gamma = \rho \text{ x } g$

   $\gamma = 10055.25 \text{ N / m}^3$

   $= 10.06 \text{ kN / m}^3$

2. Capacity

   We use 250 kg/s as an example for this calculation. The diameter of the pipe is set in 0.5 m. so we can calculate the fluid velocity as follows:

   $A = \frac{1}{4}(p D^2)$

   $A = 0.25 \text{ x } 3.14 \text{ x } 0.6^2$

   $A = 0.283 \text{ m}^2$

   So,

   $\dot{m}ww = . v . A$

   $v = \dot{m}ww / . A$

   $v = 250/(1025 . 0{,}283)$

   $V = 0.863 \text{ m/s}$

   So, the capacity can be calculate as follows:

   $Q = . A$

   $Q = 0.863 \text{ m}^2\text{/s x } 0.283 \text{ m}^2$

   $Q = 0.244 \text{ m}^3\text{/s}$

Table 4.8 Capacity of Warm sea water

$\dot{m}ww$ (kg/s)	A (m^2)	v (m^2/s)	Q (m/s)
250	0.283	0.863	0.244
500	0.283	1.726	0.488
750	0.283	2.589	0.732
1000	0.283	3.452	0.976
1250	0.283	4.315	1.220

3. Head Total
   a. Pressure Head
   D pressure head is differences pressure between suction and discharge side. We can calculate pressure head with following formulas.

$$Hp = Pdisc – P\ suc / (r \times g)$$

DP = -1 bar = -1×10^5 N/m^2
r = 1025 kg/m^3
g = 9,8 m/s^2

Hp = -1×10^5 N/m^2 / (025 kg/m^3 x 9,8 m/s^2)
= -10 m

   b. Velocity head
   because of the flow velocity on the discharge side and the suction side of the value the same, then the velocity head = 0 m

   c. Static Head
   Hs = hdisc – hsuction
   Where.
   Hdischarge = 0 m (because in a line with pump)
   Hsuction = 12 m (from suction to pump)

   d. Head Losses
   Suction side
   a. Losses caused by the Head Loss Major
      I. Reynolds number
      $Rn = v . D/ \upsilon$
   Where,
   υ = Kinematic viscosit

   y of seawater at 29 °C = $0.859.10^{-6}$ m2/s

   v = 0.863 m / s

D = 0.6 m

Rn = 603165.412 (turbulence)

Table 4.9 warm sea water Reynolds number

mww	v	Rn	Flow
250	0.863	603165.412	Turbulence
500	1.726	1206330.824	Turbulence
750	2.589	1809496.236	Turbulence
1000	3.452	2412661.648	Turbulence
1250	4.315	3015827.060	Turbulence

II. Friction Losses

Because of the turbulence flow, friction factor in suction pipe cen be calculated with following formula :

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

We can use Colebrook Equation Formulas Calculator online developed by AJ Design to shorten calculate time. We use stainless steel pipe so the relative roughness is 0.015.

Table 4.10 Friction Factor for warm sea water pipe

mww	Rn	f
250	723798.494	0.057216442
500	1447596.989	0.057195224
750	2171395.483	0.057188149
1000	2895193.978	0.05718461
1250	3618992.472	0.057182487

III. Head Loss Major

$$Hf1 = f \ x \ L \ x \ v2 \ / \ (D \ x \ 2g)$$

Where,
f = Friction factor
L = Length of pipe (15 m)
V = Velocity (m/s^2)
g = 9.8 m/s^2
so,

Table 4.11 Suction Head Loss Major warm sea water

mww	f	v^2	hf
250	0.0440	0.74	0.07
500	0.0440	2.98	0.27
750	0.0440	6.70	0.60
1000	0.0440	11.92	1.07
1250	0.0440	18.62	1.67

e. Losses causes by accessories (Head Loss Minor)

Table 4.12 Loss Coefficient Calculation Warm Pipe SW Suction Side

No	Type	n	k	n.k
1	Gate valve	1	0.86	0.86
2	elbow 90o	1	0.45	0.45
3	strainer	1	1.5	1.5
Total				2.81

We can calculate head loss minor with this following formula :

Hl = ktotal x (v^2/2g)

Table 4.13 Head loss minor warm SW suction side

mww	v2	k	Hmin
250	0.74	2.81	0.107
500	2.98	2.81	0.427
750	6.70	2.81	0.961
1000	11.92	2.81	1.709
1250	18.62	2.81	2.670

f. head loss Suction Total

Head loss suction total = Hf + Hl

Table 4.14 warm SW Head loss suction total

mww	hmay	hmin	head loss total
250	0.07	0.107	0.174

mww	hmay	hmin	head loss total
500	0.27	0.427	0.695
750	0.60	0.961	1.563
1000	1.07	1.709	2.779
1250	1.67	2.670	4.342

Discharge side

g. Losses caused by the Head Loss Major

IV. Reynolds number

$Rn = v \cdot D / \upsilon$

Where,

υ = Kinematic viscosit

y of seawater at 29 °C = $0.859.10^{-6}$ m2/s

v = 0.863 m / s

D = 0.6 m

Rn = 603165.412 (turbulence)

Table 4.15 warm sea water Reynolds number

mww	v	Rn	Flow
250	0.863	603165.412	Turbulence
500	1.726	1206330.824	Turbulence
750	2.589	1809496.236	Turbulence
1000	3.452	2412661.648	Turbulence
1250	4.315	3015827.060	Turbulence

h. Friction Losses

Because of the turbulence flow, friction factor in suction pipe cen be calculated with following formula :

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

We can use Colebrook Equation Formulas Calculator online developed by AJ Design to shorten calculate time. We use stainless steel pipe so the relative roughness is 0.015.

Table 4.16 Friction Factor for warm sea water pipe

mww	Rn	f
250	723798.494	0.057216442
500	1447596.989	0.057195224
750	2171395.483	0.057188149
1000	2895193.978	0.05718461
1250	3618992.472	0.057182487

i. Head Loss Major

$$Hf1 = f \; x \; L \; x \; v2 \, / \, (D \; x \; 2g)$$

Where,
f = Friction factor
L = Length of pipe (2 m)
V = Velocity (m/s^2)
g = 9.8 m/s^2
so,

Table 4.17 Suction Head Loss Major warm sea water

mww	f	v^2	hf
250	0.0440	0.74	0.01
500	0.0440	2.98	0.03
750	0.0440	6.70	0.06
1000	0.0440	11.92	0.11
1250	0.0440	18.62	0.17

j. Losses causes by accessories (Head Loss Minor)

Table 4.18 Loss Coefficient Calculation Warm Pipe SW discharge Side

No	Type	n	k	n.k
1	Gate valve	1	0.86	0.86
Total				0.86

We can calculate head loss minor with this following formula :

$$\mathbf{Hl = ktotal \ x \ (v^2/2g)}$$

Table 4.19 Head loss minor warm SW discharge side

mww	v^2	k	Hmin
250	0.74	0.86	0.033
500	2.98	0.86	0.131
750	6.70	0.86	0.294
1000	11.92	0.86	0.523
1250	18.62	0.86	0.817

k. Head loss Suction Total

Head loss suction total = Hf + Hl

Table 4.20 warm SW Head loss discharge total

mww	hf	hl	head loss total
250	0.01	0.033	0.039
500	0.03	0.131	0.157
750	0.06	0.294	0.354
1000	0.11	0.523	0.630
1250	0.17	0.817	0.984

l. Head Total

H = Hp + Hv + Hs + Hf + Hl

Table 4.21 Warm SW Head Total

mww	Head total
250	2.213
500	2.852
750	3.917
1000	5.409
1250	7.326

m. Pump Power

From the equations described above, you can calculated pump power in kW. That is:

$$\dot{W} = ɣ\, x\; Q\; x\; H$$

Table 4.22 Power Input Warm SW Pump

h	Q (m^3/s)	ϒ ($kN.m^3$)	W (kW)
2.213	0.244	10.055	5.43
2.852	0.488	10.055	13.99
3.917	0.732	10.055	28.82
5.409	0.976	10.055	53.06
7.326	1.220	10.055	89.84

B. Cold Sea Water Pump Power Input

After that, we calculate the power input on the cold sea water pump. Because the depth variation of sea water, as an example of calculation is used scenario 1, which is 1000m depth.

From the book "Pumps and Compressors" by Ir. Soelarso, to look for energy pumps in kW are obtained from the equation as follows:

$$\dot{W} = ɣ\, x\; Q\; x\; H$$

Where,

$\dot{W}$: Pump Power (kW)
H: Pump Total Head (m)
ϒ: Fluid Specific Gravity (kN / m3)
Q: Flow Capacity (m3 / s)
to calculate pump power, you must first know:

a. Sea Water Specific Gravity

ρ = 1025 kg / m^3
g = 9.81 m / s^2
then,
ɣ = ρ x g
ɣ = 10055.25 N / m^3
= 10.06 kN / m^3

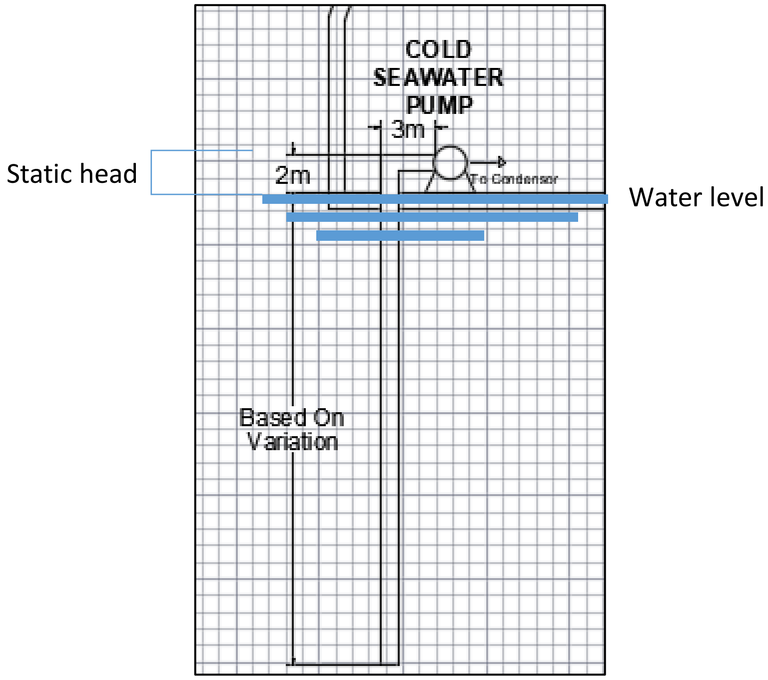

Fig 4.15 Cold sea water pump scheme

b. Capacity

We use 265 kg/s as an example for this calculation. The diameter of the pipe is set in 0.7 m. so we can calculate the fluid velocity as follows:

A = ¼(p D^2)

A = 0.25 x 3.14 x 0.7^2

A = 0.385 m^2

So,

$\dot{m}ww \quad = \ .\ v\ .\ A$

$v \quad = \dot{m}ww/\ .\ A$

$v \quad = 250/(1025\ .\ 0.385)$

V = 0.634 m/s

So, the capacity can be calculate as follows:

Q = . A

Q = 0.385 m^2 x 0.634 m/s

Q = 0.244 m^3/s

Table 4.23 Capacity of Warm sea water

Mcw	A (m^2)	v (m/s)	Q (m^3/s)
250	0.385	0.634	0.244
500	0.385	1.268	0.488
750	0.385	1.902	0.732
1000	0.385	2.536	0.976
1250	0.385	3.170	1.220

a. Head Total

Pressure Head

D pressure head is differences pressure between suction and discharge side. We can calculate pressure head with following formulas.

$$Hp = Pdisc – P\ suc / (r \times g)$$

DP = - 99.437 bar = -99.437 x 10^5 N/m^2

r = 1025 kg/m^3

g = 9,8 m/s^2

Hp = -99.437 x 10^5 N/m^2 / (025 kg/m^3 x 9,8 m/s^2)

= -989.915 m

b. Velocity head

because of the flow velocity on the discharge side and the suction side of the value the same, then the velocity head = 0 m

c. Static Head

Hs = hdisc – hsuction

Where.

Hdischarge = 0 m (because in a line with pump)

Hsuction = 1002m (from water level to pump)

d. Head Losses

a. Losses caused by the Head Loss Major

Reynolds number and friction losses

$Rn = v\ .\ D/\ \upsilon$

Where,

υ = Kinematic viscosit

y of seawater at 5 ^oC = 1.58.10^{-6} m2/s

v = 0.634 m / s

D = 0.7 m

Rn = 200599.230 (turbulence)

Because of the turbulence flow, friction factor in suction pipe cen be calculated with following formula :

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{Re\sqrt{f}}\right)$$

We can use Colebrook Equation Formulas Calculator online developed by AJ Design to shorten calculate time. We use stainless steel pipe so the relative roughness is 0.015.

Table 4.24 warm sea water Reynolds number

mcw	v	Rn	f
250	0.634	200599.230	0.0440
500	1.268	401198.460	0.0440
750	1.902	601797.690	0.0440
1000	2.536	802396.920	0.0440
1250	3.170	1002996.150	0.0440

e. Head Loss Major Suction and discharge

$$Hf1 = f \ x \ L \ x \ v2 \ / \ (D \ x \ 2g)$$

Where,
f = Friction factor
L suct = Length of pipe (1005 m)
Ldisch = 2 m
V = Velocity (m/s^2)
g = 9.8 m/s^2
so,

Table 4.25 Total Head Loss Major Cold sea water Scenario 1 (d = 1000m)

L suction (m)	L disch (m)	f	v^2 (m^2/s)	hf suction	hf discharge	hf total
1005	2	0.0440	0.40	1.81	0.00	1.82
1005	2	0.0440	1.61	7.26	0.01	7.27

L suction (m)	L disch (m)	f	v^2 (m^2/s)	hf suction	hf discharge	hf total
1005	2	0.0440	3.62	16.33	0.03	16.36
1005	2	0.0440	6.43	29.03	0.06	29.09
1005	2	0.0440	10.05	45.36	0.09	45.45

f. Losses causes by accessories (Head Loss Minor)

Table 4.23 Loss Coefficient Calculation Cold Pipe SW Suction Side

No	Type	n	k	n.k
1	Gate valve	1	0.86	0.86
2	elbow 90o	1	0.45	0.45
3	strainer	1	1.5	1.5
Total				2.81

Table 4.24 Loss Coefficient Calculation Cold Pipe SW Discharge Side

No	Type	n	k	n.k
1	Gate valve	1	0.86	0.86
Total				0.86

We can calculate head loss minor with this following formula :

$$\mathbf{Hl = ktotal \times (v^2/2g)}$$

Table 4.25 Total Head loss minor Cold SW suction and discharge side scenario 1 (d=1000m)

mww	hmin disch	hmin suct	h min total
250	0.06	0.018	0.075
500	0.23	0.071	0.301
750	0.52	0.159	0.678
1000	0.92	0.282	1.205
1250	1.44	0.441	1.882

g. Head Total

H = Hp + Hv + Hs + Hf + Hl

Table 4.26 Cold CW Head Total

mww	Head Total
250	13.978
500	19.657
750	29.123
1000	42.375
1250	59.413

h. Pump Power
From the equations described above, you can calculated pump power in kW. That is:

$$\dot{W} = \Upsilon\, x\, Q\, x\, H$$

Table 4.27 Power Input Warm SW Pump Scenario 1

h	**Q (m^3/s)**	**ϒ ($kN.m^3$)**	**W (kW)**
13.978	0.244	10.055	34.28
19.657	0.488	10.055	96.42
29.123	0.732	10.055	214.27
42.375	0.976	10.055	415.70
59.413	1.220	10.055	728.55

4.3.3 Simulation and Manual Comparation

After calculate the OTEC system manually, a comparison is made between power output simulation results and manual calculation as validation. Validation is used to calculate errors whether they are within reasonable limits or exceed limits. Here is how to calculate errors.

% error = ((Simulation – manual)/simulation) x 100%

For example, scenario 1 is used when the mass flow rate of seawater is 250 kg / s with the simulation output power of 85.27 kW and the manual output is 84.8 kW. Then the error is obtained.

% error = ((85.27-84.8)/85.27) x 100%
= 0.55%

With the results of the 0.55% error percentage, it can still be said to be reasonable because it is below 5%. This shows that the simulation with Cycle Tempo is quite accurate. Next, the results of the comparison of the overall scenario will be displayed along with the error size.

Table 4.28 Scenario 1 Power output manual and simulation comparation

WT net manual	WT net simulation	error
84.80	85.27	0.55%
169.60	170.53	0.55%
254.40	255.8	0.55%
339.19	341.07	0.55%
423.99	426.34	0.55%

Others scenario data comparation between manual and simulation will be in attachment section

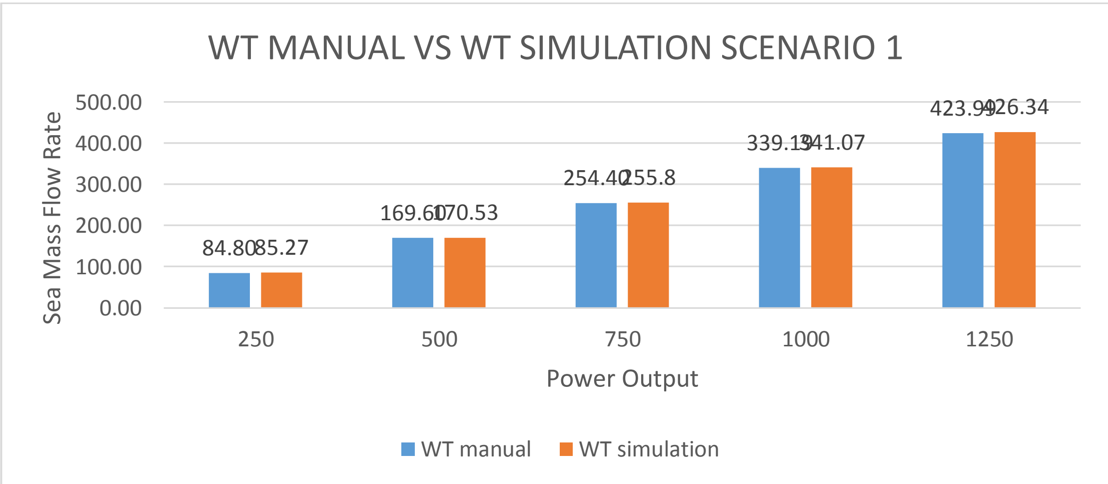

Fig.4.16 Scenario 1 Power output manual and simulation comparation

4.3.4 Net Power and Efficiency

Net power is the amount of electricity generated by a OTEC that is transmitted and distributed for consumer use. So we have to take into account the input power of working fluid and sea water pump along with efficiency of turbine and generator. We con use the formula below to calculate net power.

WTnet = (WT x ht x hg) – Wpwf – Wpwm –Wpcw

W_T = Power Turbine output (kW)

Wpwf = Power input working fluid pump (kW)

Wpwm = Power input warm sea water pump (kW)

Wpcw = Power input cold sea water pump (kW)

ht = Turbine efficiency = 98%

hg = Generator Efficiency = 100%

For the example we use scenario 1 sea depth 1000m

Table 4.29 Net Power Scenario 1 (d=1000m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT (kW)
113.06	84.79849	3.39	5.43	34.28	41.70
226.13	169.60	6.78	13.99	96.42	52.40
339.19	254.40	10.18	28.82	214.27	1.13
452.26	339.19	13.57	53.06	415.70	-143.13
565.32	423.99	16.96	89.84	728.55	-411.36

Table 4.30 Net Power Scenario 2 (d=900m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
115.29	83.01	3.46	5.43	33.84	40.28
230.58	166.01	6.92	13.99	92.88	52.23
345.86	249.02	10.38	28.82	202.32	7.51
461.15	332.03	13.83	53.06	387.36	-122.23
576.44	415.04	17.29	89.84	673.21	-365.30

Table 4.31 Net Power Scenario 3 (d=800m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
118.81	79.60	3.56	5.43	33.39	37.22
237.63	159.21	7.13	13.99	89.33	48.76
356.44	238.81	10.69	28.82	190.36	8.93
475.25	318.42	14.26	53.06	359.03	-107.93
594.07	398.02	17.82	89.84	617.87	-327.51

Table 4.32 Net Power Scenario 4 (d=700m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
123.61	75.40	3.71	5.43	32.95	33.32
247.23	150.81	7.42	13.99	85.79	43.61
370.84	226.21	11.13	28.82	178.41	7.86
494.46	301.62	14.83	53.06	330.69	-96.97
618.07	377.02	18.54	89.84	562.53	-293.88

Table 4.33 Net Power Scenario 5 (d=600m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
125.64	72.87	2.51	5.43	32.51	32.42
251.27	145.74	5.03	13.99	82.25	44.47
376.91	218.61	7.54	28.82	166.46	15.79
502.55	291.48	10.05	53.06	302.36	-73.99
628.18	364.35	12.56	89.84	507.19	-245.24

Table 4.34 Net Power Scenario 6 (d=500m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
128.76	70.82	2.58	5.43	32.07	30.75
257.53	141.64	5.15	13.99	78.71	43.79
386.29	212.46	7.73	28.82	154.50	21.41
515.05	283.28	10.30	53.06	274.02	-54.11
643.81	354.10	12.88	89.84	451.85	-200.46

Table 4.35 Net Power Scenario 7 (d=400m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
133.26	66.63	2.67	5.43	31.62	26.92
266.53	133.26	5.33	13.99	75.17	38.78
399.79	199.90	8.00	28.82	142.55	20.53
533.06	266.53	10.66	53.06	245.69	-42.88
666.32	333.16	13.33	89.84	396.50	-166.51

Table 4.36 Net Power Scenario 8 (d=300m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
138.12	63.53	1.38	5.43	31.18	25.54
276.24	127.07	2.76	13.99	71.63	38.69
414.36	190.60	4.14	28.82	130.60	27.04
552.47	254.14	5.52	53.06	217.35	-21.80
690.59	317.67	6.91	89.84	341.16	-120.24

Table 4.37 Net Power Scenario 9 (d=200m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
149.01	55.13	1.49	5.43	30.74	17.48
298.01	110.26	2.98	13.99	68.08	25.21
447.01	165.40	4.47	28.82	118.64	13.46
596.02	220.53	5.96	53.06	189.02	-27.51
745.02	275.66	7.45	89.84	285.82	-107.45

Table 4.38 Net Power Scenario 10 (d=100m)

mwf (kg/s)	WT (KW)	WF Pump (KW)	WW Pump (KW)	CW Pump (KW)	Net WT
251.27	23.89	2.51	5.43	30.30	-14.35
502.55	50.25	5.03	13.99	64.54	-33.30
753.89	75.39	7.54	28.82	106.69	-67.66
1005.09	100.51	10.05	53.06	160.68	-123.29
1256.37	125.64	12.56	89.84	230.48	-207.25

In this scenario we can’t use red line isobutene mass flow rate because the power output/generated by turbine can’t meet pumps power input requirement. From the table above we know that scenario 10 that sea depth in 100 m is impossible to be implemented because all of the power output from variation of working fluid mass flow rate cant’t meet pumps power input needs.

Then calculate the efficiency of the closed-cyle OTEC system. The efficiency of the cyle can be calculated through the following formula:

$$\mathbf{h_{cycle} = (Wnet / Qh) \times 100\%}$$

for example we use scenario 1

1. $\mathbf{h_{cycle}}$ = (41.7 kW/ 4261.41 kW) x 100%
2. $\mathbf{h_{cycle}}$ = **0.979%**

Table 4.39 Efficiency cyle of Scenario 1 (d=1000m)

WT net (KW)	Qh(KW)	Efficiency
41.70	4261.41	0.98%
52.40	8522.81	0.61%
1.13	12784.22	0.01%
-143.13	17045.63	-0.84%
-411.36	21307.03	-1.93%

Table 4.40 Efficiency cyle of Scenario 2 (d=900m)

WT net (KW)	Qh(KW)	Efficiency
40.28	4279.49	0.94%
52.23	8558.98	0.61%
7.51	12838.47	0.06%
-122.23	17117.96	-0.71%
-365.30	21397.45	-1.71%

Table 4.41 Efficiency cyle of Scenario 3 (d=800m)

WT net (KW)	Qh(KW)	Efficiency
37.22	4273.70	0.87%
48.76	8547.41	0.57%
8.93	12821.11	0.07%
-107.93	17094.81	-0.63%
-327.51	21368.52	-1.53%

Table 4.42 Efficiency cyle of Scenario 4 (d=700m)

WT net (KW)	Qh(KW)	Efficiency
33.32	4267.16	0.78%
43.61	8534.35	0.51%
7.86	12801.50	0.06%
-96.97	17068.66	-0.57%
-293.88	21335.85	-1.38%

Table 4.43 Efficiency cyle of Scenario 5 (d=600m)

WT net (KW)	Qh(KW)	Efficiency
32.42	4265.38	0.76%
44.47	8530.72	0.52%
15.79	12796.09	0.12%
-73.99	17061.44	-0.43%
-245.24	21326.81	-1.15%

Table 4.44 Efficiency cyle of Scenario 6 (d=500m)

WT net (KW)	Qh(KW)	Efficiency
30.75	4262.06	0.72%
43.79	8524.11	0.51%
21.41	12786.17	0.17%
-54.11	17048.22	-0.32%
-200.46	21310.24	-0.94%

Table 4.45 Efficiency cyle of Scenario 7 (d=400m)

WT net (KW)	Qh(KW)	Efficiency
26.92	4256.45	0.63%
38.78	8512.94	0.46%
20.53	12769.39	0.16%
-42.88	17025.84	-0.25%
-166.51	21282.29	-0.78%

Table 4.46 Efficiency cyle of Scenario 8 (d=300m)

WT net (KW)	Qh(KW)	Efficiency
25.54	4252.65	0.60%
38.69	8505.34	0.45%
27.04	12757.99	0.21%
-21.80	17010.64	-0.13%
-120.24	21263.33	-0.57%

Table 4.47 Efficiency cyle of Scenario 9 (d=200m)

WT net (KW)	Qh(KW)	Efficiency
17.48	4242.17	0.41%
25.21	8484.34	0.30%
13.46	12726.49	0.11%
-27.51	16968.66	-0.16%
-107.45	21210.83	-0.51%

Table 4.48 Efficiency cyle of Scenario 10 (d=100m)

WT net (KW)	Qh(KW)	Efficiency
-14.35	4211.34	-0.34%
-33.30	8422.67	-0.40%
-67.66	12635.23	-0.54%
-123.29	16845.34	-0.73%
-207.25	21056.68	-0.98%

From the table above it can be concluded that in Scenario 1 with a Power output of 40.28 kW has the greatest efficiency value with 0.98%.

4.4 Graphic and Analysis

4.4.1. Depth vs Power

This relationship graphic between sea water depth and power output uses the largest output power data from each scenario:

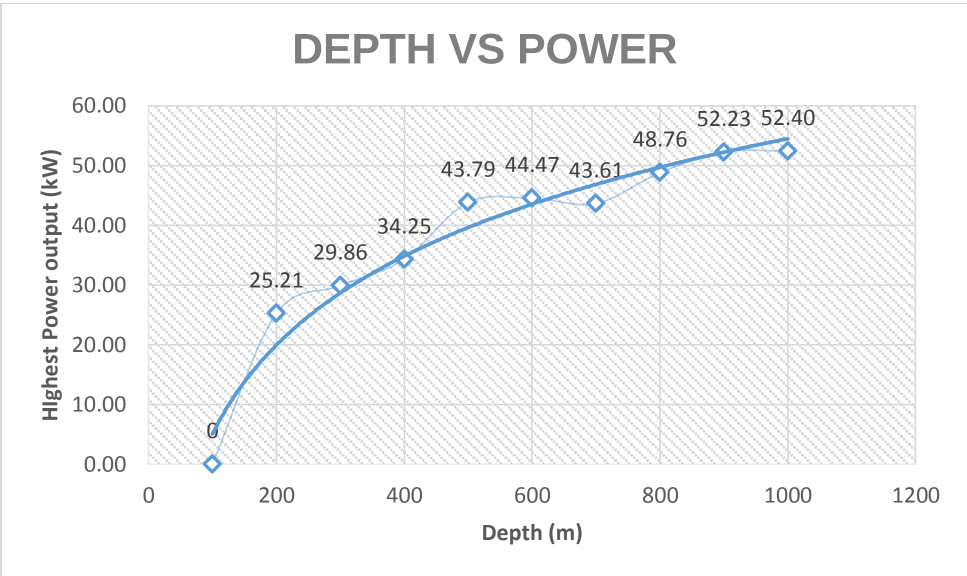

Figure 4.17 Depth vs Power Graphic

From the graph above it can be concluded that the deeper the depth of the sea, the greater the power output produced by the OTEC closed cycle. The highest power produced is by scenario 1 with 52.4 kW at a depth of 1000m with a mass flow rate of isobutene of 226.13 kg/s. For the OTEC system on the 100 m level, it can be concluded that it cannot produce power because the net Power is under 0.

4.4.2 DT Sea Water vs Power

This relationship graphic DT Sea Water vs Power output uses the largest output power data from each scenario. In that graph (fig.4.18) it can be concluded that the greater the difference in surface sea water temperature and deep sea water the greater the output power produced.

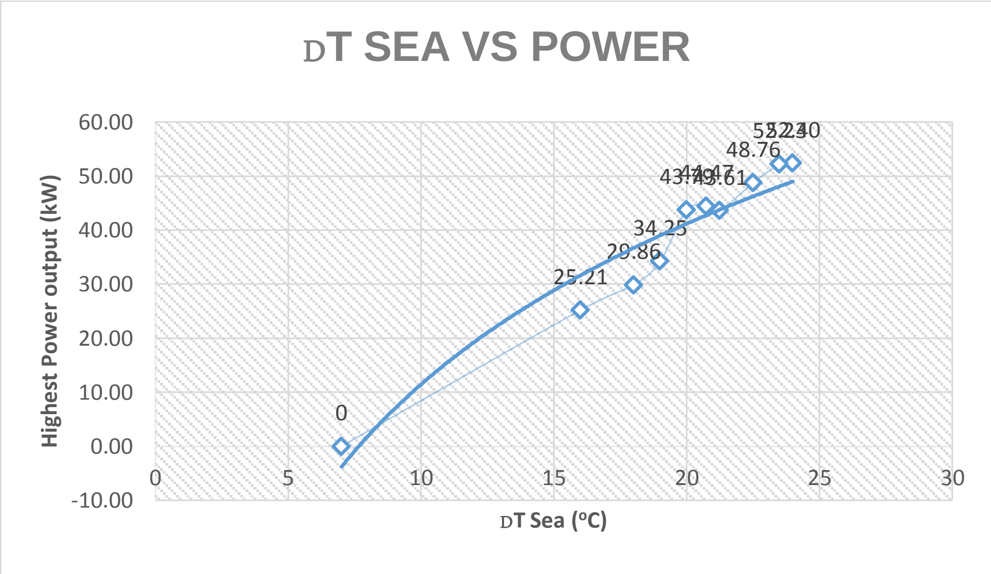

Figure 4.18 DT Sea Water vs Power Graph

This can be influenced by the depth of the sea because the deeper the sea the temperature will decrease. In the graph above it can be concluded that scenario 1 (d = 1000m) has the largest output power of 52.4 kW because it has the highest temperature difference of 24^{o}C.

4.4.3 Mass flow rate vs Power

In this graph comparing the relationship of the isobutene mass flow rate to power and the overall data from the scenario will be presented in the next page.

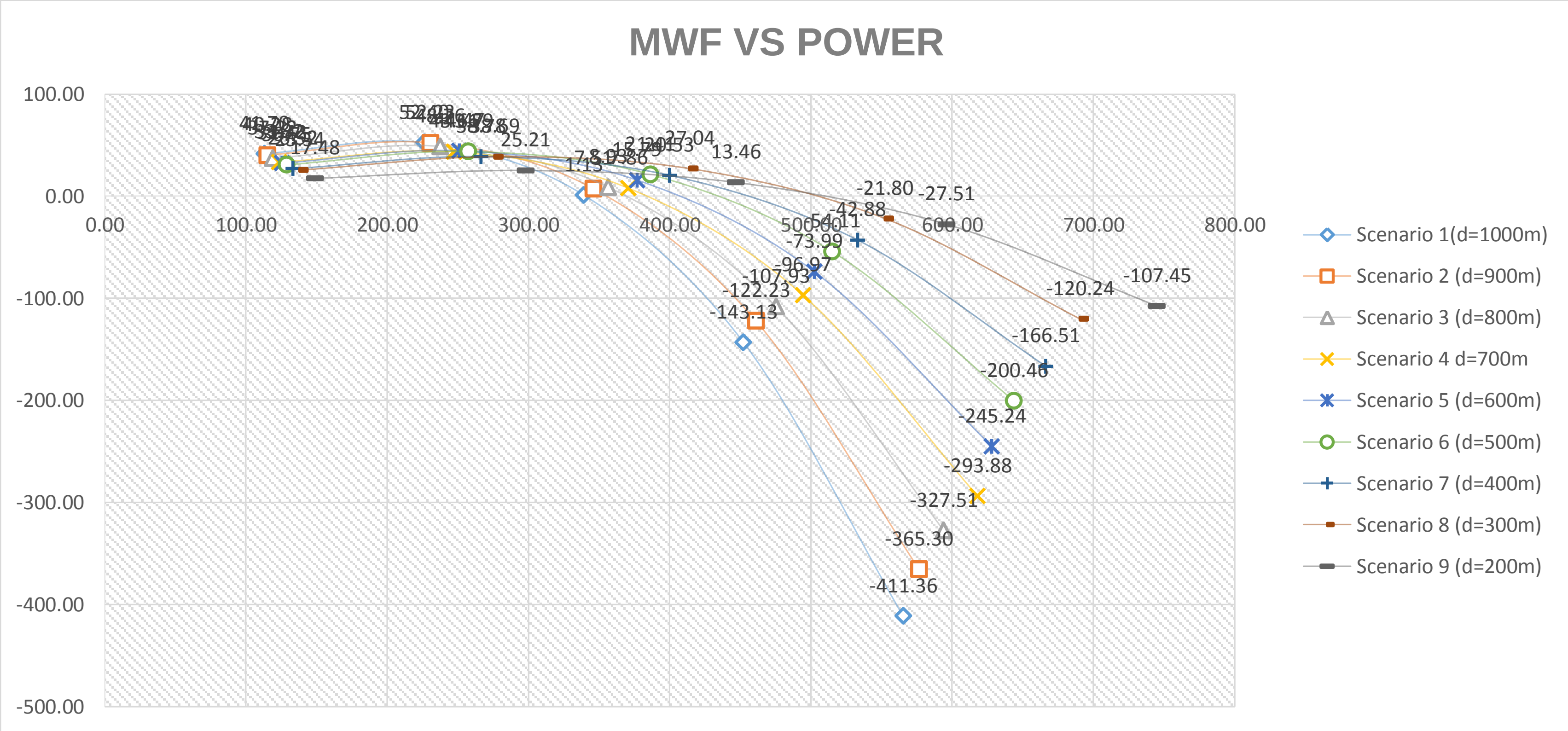

Figure 4.19 Working fluid mass flow rate vs Power

From the graph above, it can be seen that each scenario has its own optimum point which produces the highest output power. The optimum point in the 1st scenario lies in the working fluid mass flow rate 226.13 kg / s with a net power output of 52.4 kW. In the 2ndscenario the optimum point lies in the working fluid mass flow rate of 230.58 kg / s with a net power output of 52.23 kW. . In the 3r^{d} scenario the optimum lies in the working fluid mass flow rate of 237.63 kg / s with a net power output of 48.76 kW. . In the 4thscenario the optimum point lies in the mass flow rate of working fluid 247.23 kg / s with a net power output of 43.61 kW. . In the 5th scenario the optimum point lies in the working fluid mass flow rate 251.27 kg / s with a net power output of 44.47 kW. . In 6th scenario the optimum point lies in the working fluid mass flow rate of 257.53 kg / s with a net power output of 43.79 kW. . In the 7th scenario the optimum point lies in the working fluid mass flow rate 266.53 kg / s with a net power output of 38.78 kW. . In the 8th scenario the optimum point lies in the working fluid mass flow rate of 276.24 kg / s with a net power output of 38.69 kW. . In 9th scenario the optimum point lies in the working fluid mass flow rate of 298.01 kg / s with a net power output of 25.21 kW. For the 10th scenario it cannot be implemented because it does not produce net power.

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CHAPTER V
CONCLUSION

5.1 Conclusion

Some results can be concluded from closed cycle OTEC design analysis using isobutene as working fluid with simulations using the following cycle cycle:

1. OTEC sea water depth for warm sea water is 10 m and for cold sea water is variated from 1000 m until 100 m. The input temperature of warm sea water is 29°C and the output is 25 °C. The isobutane temperature input turbine is 25 °C. The temperature differences in condenser is set in 4 °C .The cold and warm seawater mass flow rate variation is from 250 kg/s until 1250 kg/s. The results from OTEC uses isobutane as the best working fluid is to use scenario 1 at a depth of 1000m with the mass flow rate of seawater at 500 kg/s with a power of 13.99 kW rewuired for warm sea water pump and required cold sea water pump power 96.42 kW. Then get a working fluid mass flow rate of 226.13 kg s with pump required power of 6.78 kW. Net Power generated is 52.4 kW with efficiency of 0.61%.
2. The results of the simulation show the greater the working fluid mass flow rate the greater the gross power output produced. But it does not apply to net power output because it has to consider the power requirements of a sea water pumps and a working fluid pump.
3. The results of the simulation show that the deeper the depth of the sea, the greater the output power. In this study 1st scenario at a depth of 1000 m produces the largest gross power output, 85.27 kW, 170.53 kW, 255.8 kW, 341.07 kW, and 426.34 kW according to variations in the sea water mass flow rate.

5.2 Suggestion

To perfect this research, the authors have several suggestions, including:

1. It is necessary to examine variations in the mass flow rate of warm seawater and cold seawater with a non-constant ratio. For example, when the mass flow rate of warm sea water is 1000 kg / s, mass flow rate of cold sea water is 500 kg / s. this method is needed to find out more about the potential of OTEC.
2. It is necessary to examine the modifications to the cold sea water pump so that it does not need to require too much power.

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ATTACHMENT

Attachment List :
Attachment I Simulation results
Attachment II Summary results of manual calculation
Attachment III Error Comparation between simulation and manual calculation

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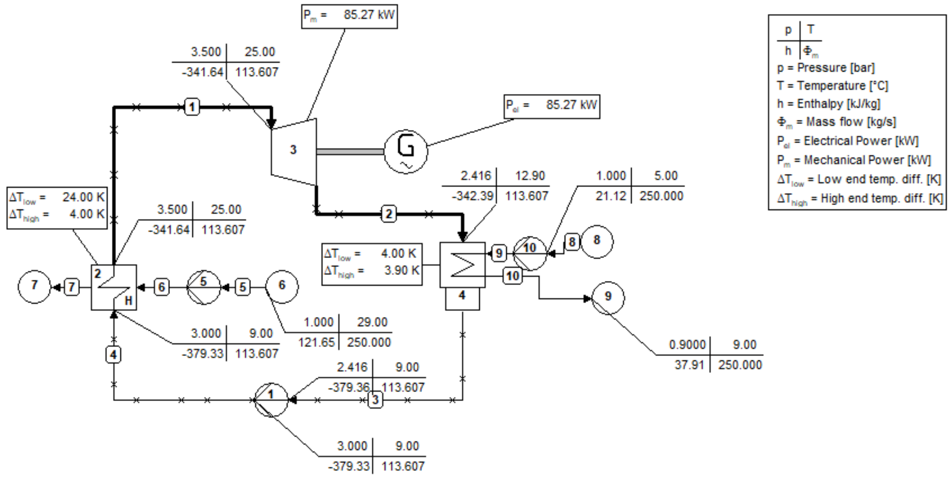
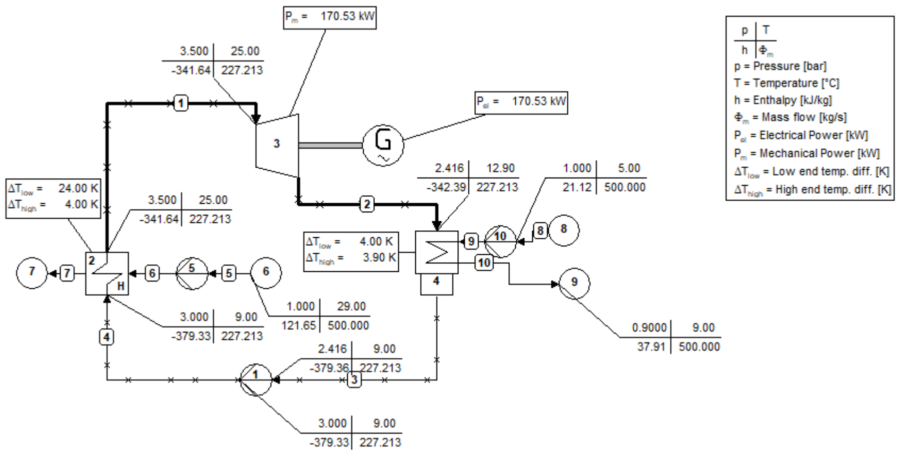
ATTACHMENT I



Scenario 1 d=1000m sea water mass flow rate 250 kg/s



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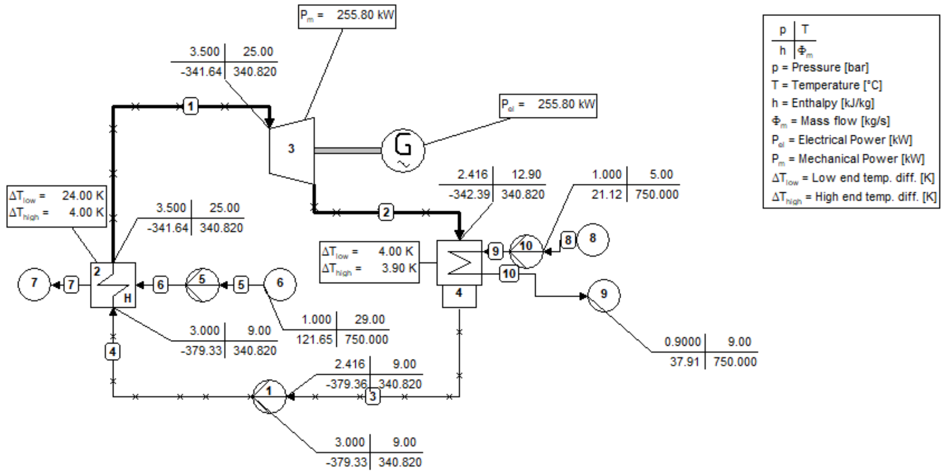

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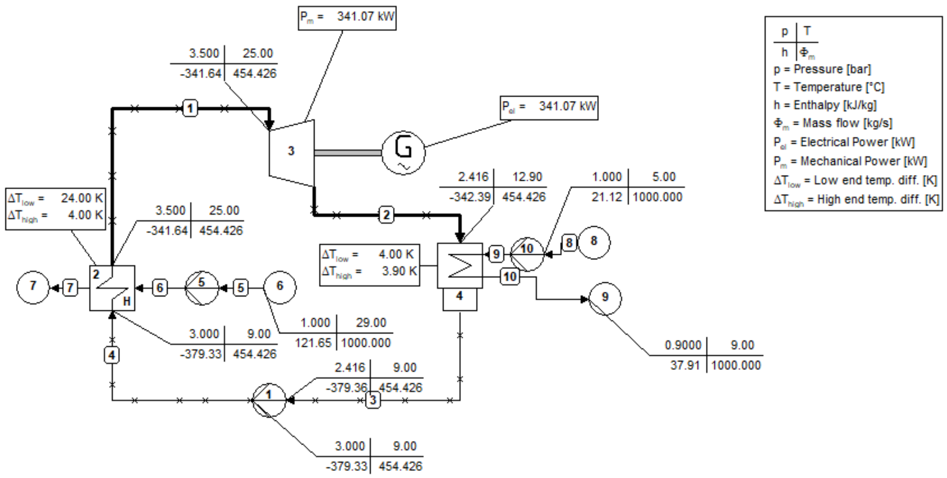

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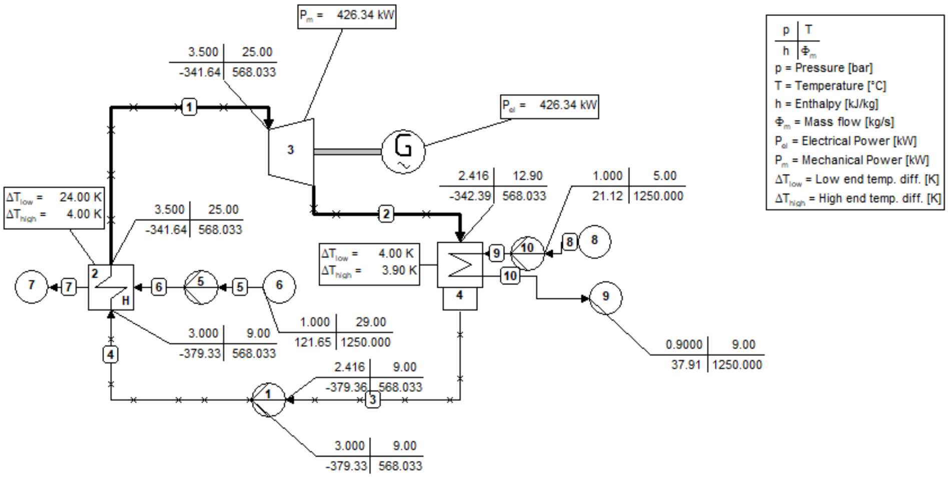

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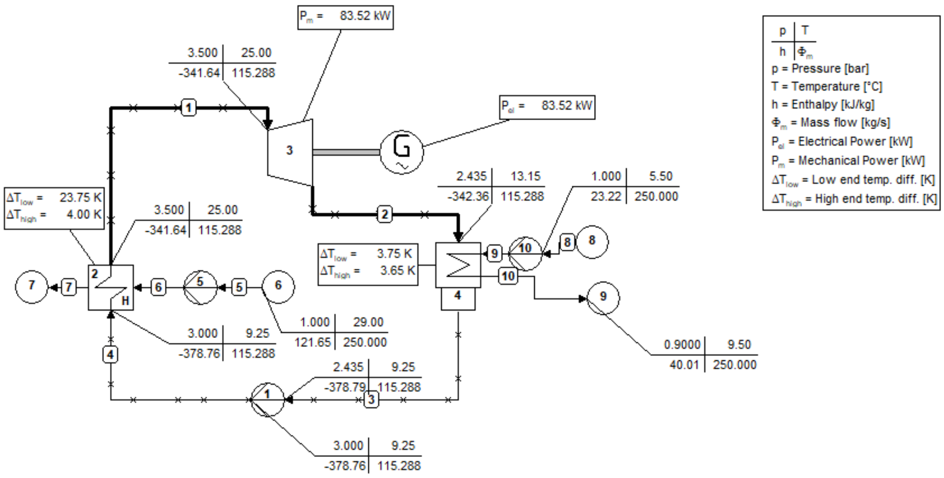

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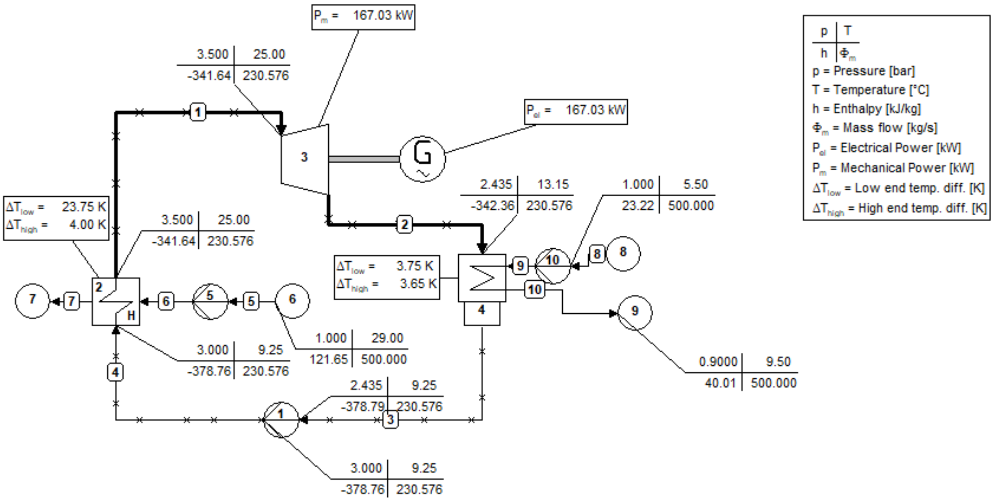

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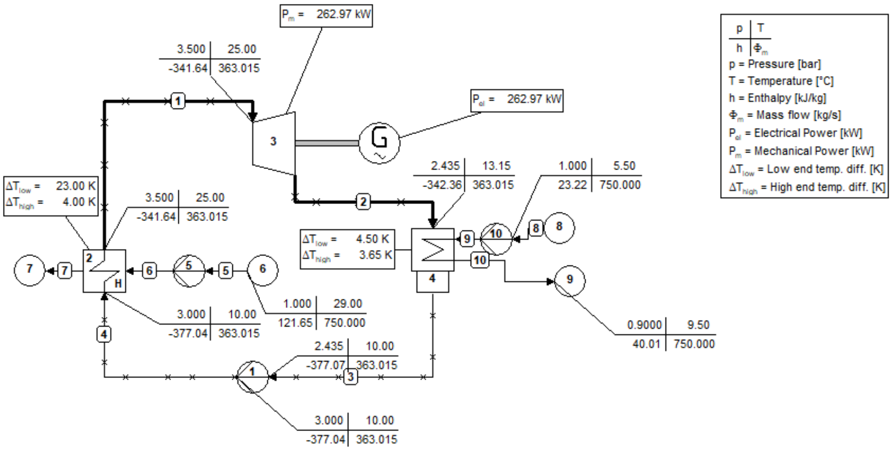
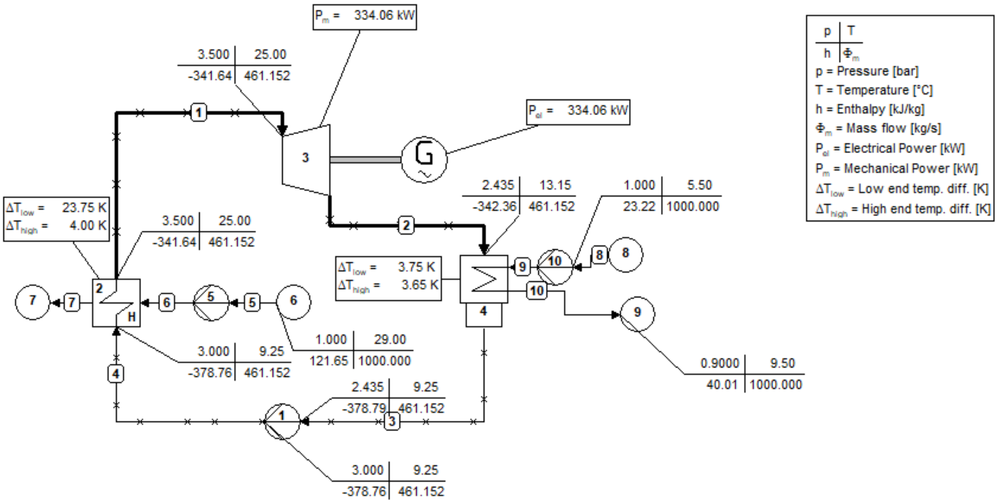
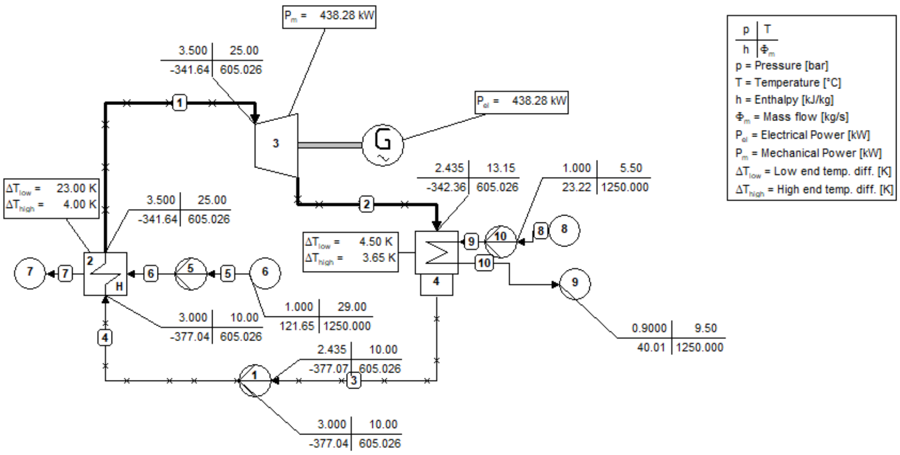


Scenario 2 d=900m sea water mass flow rate 750 kg/s



Scenario 2 d=900m sea water mass flow rate 1000 kg/s



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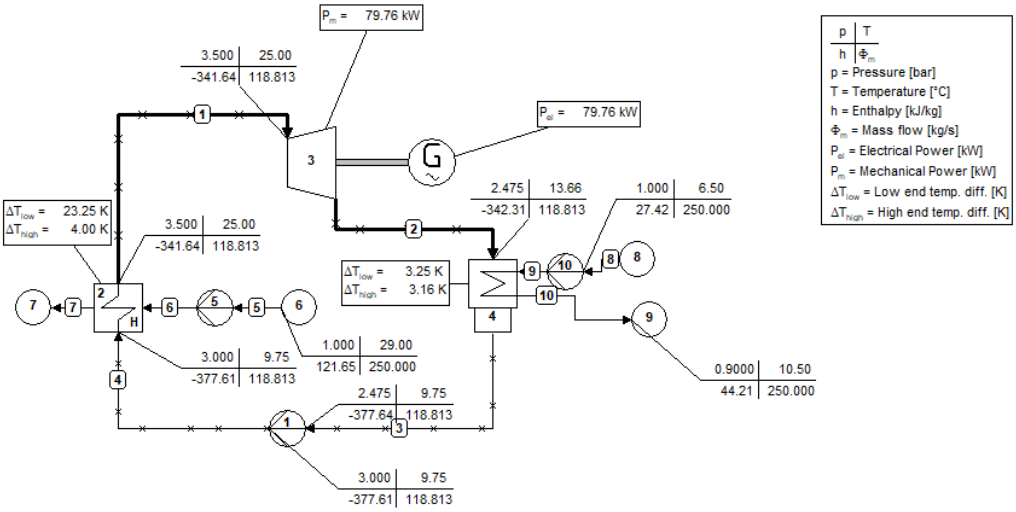

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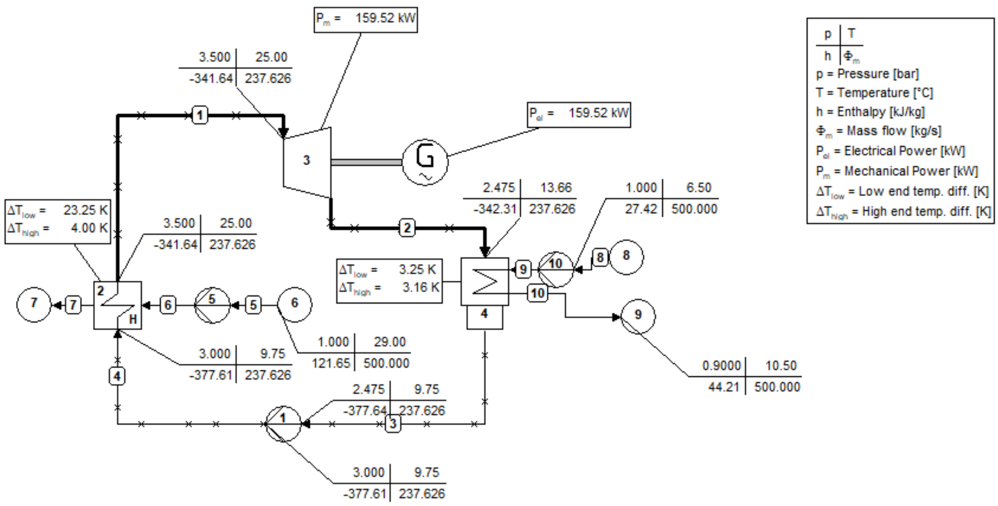

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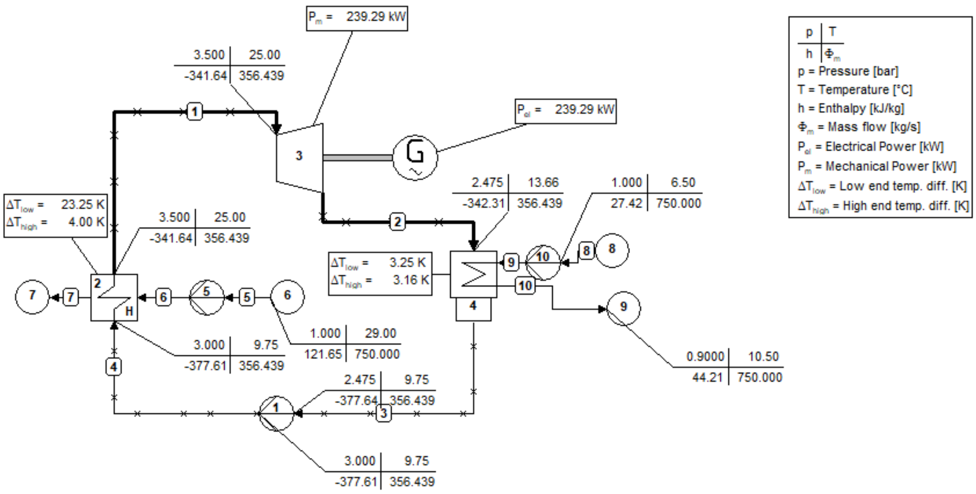

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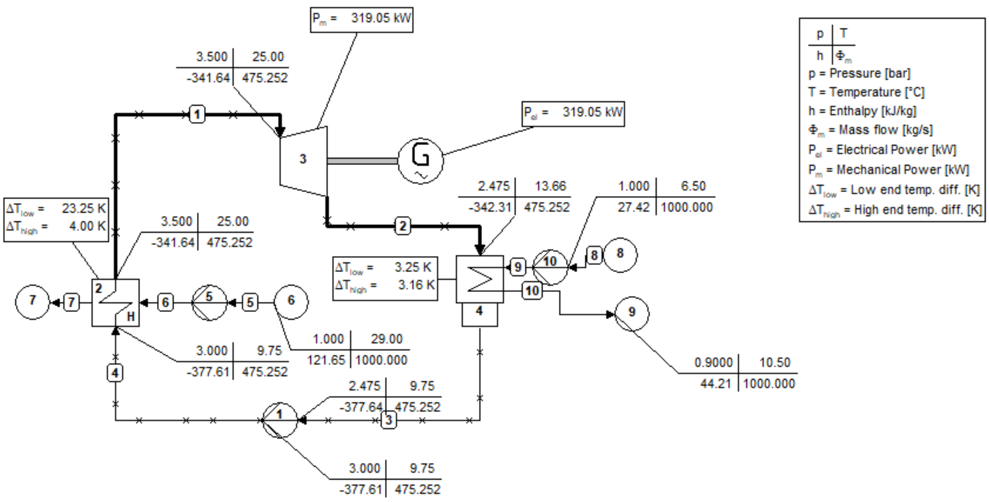

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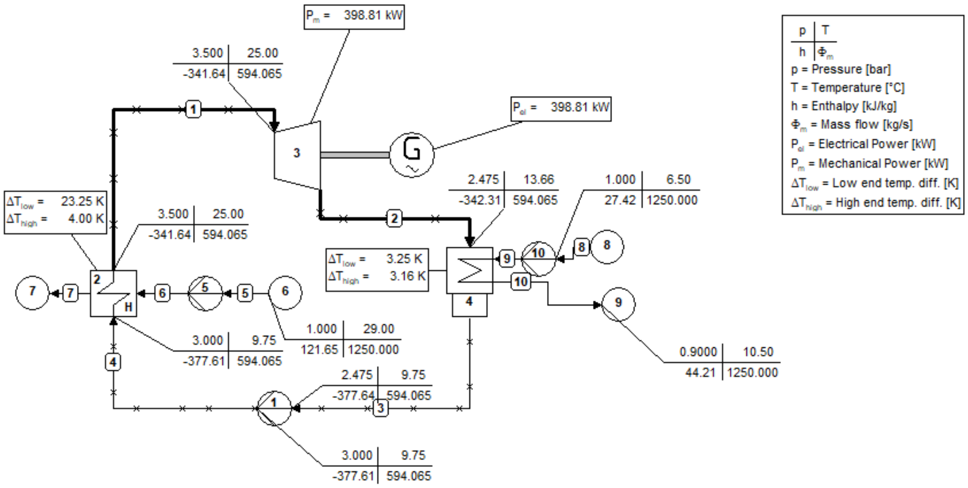
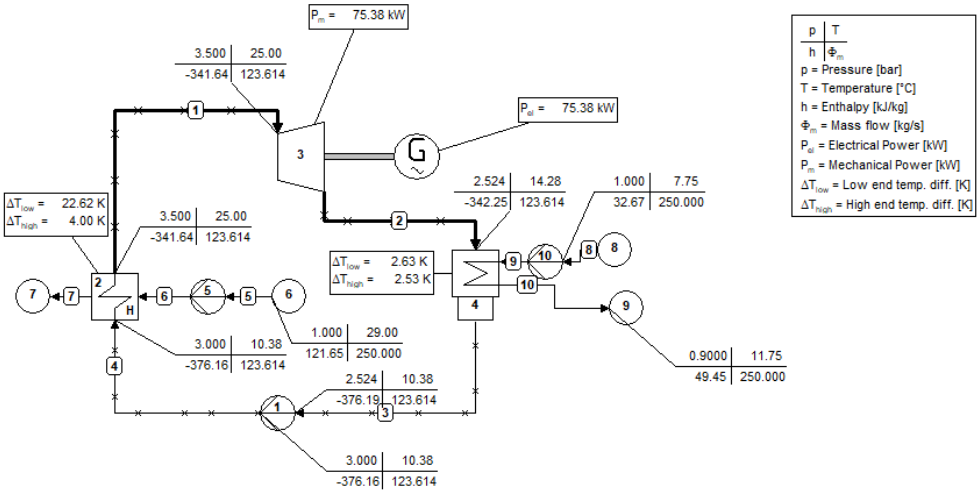


Scenario 3 d=800m sea water mass flow rate 1250 kg/s



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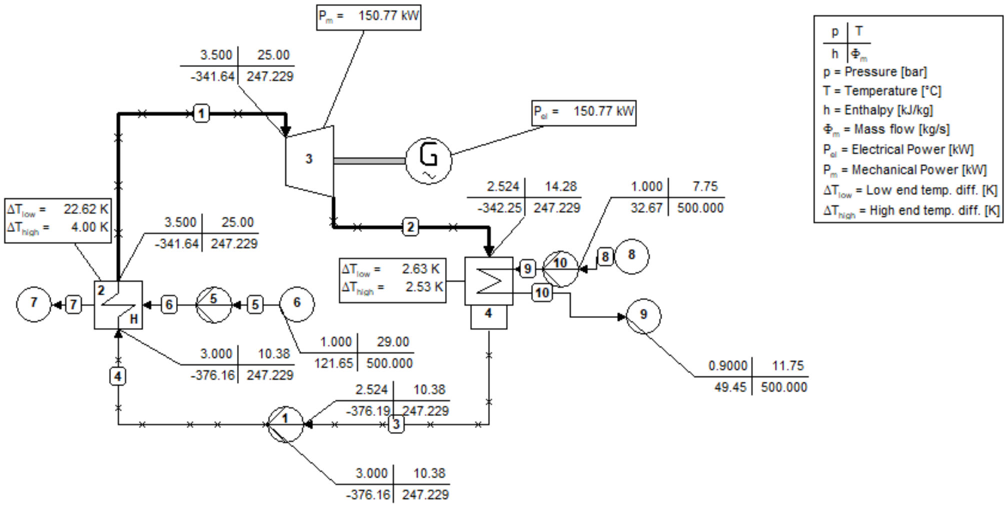
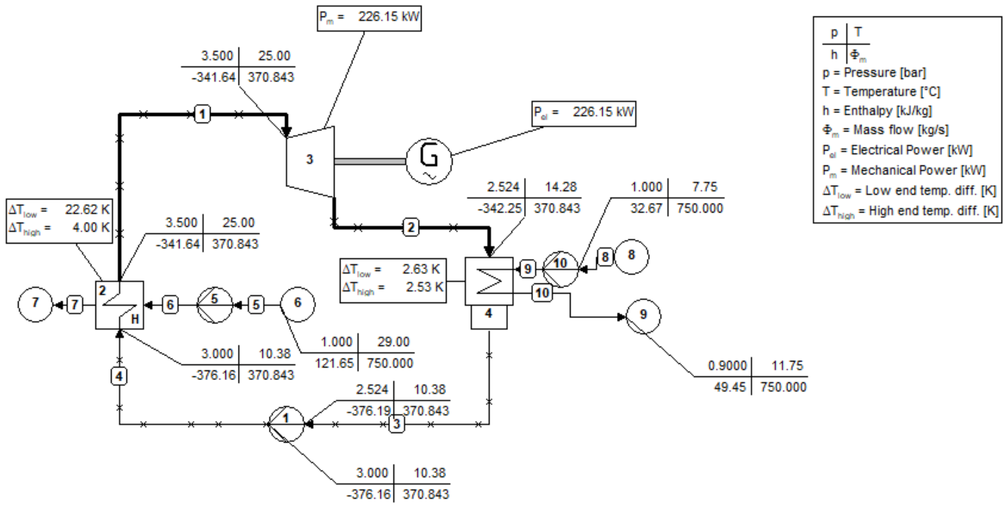
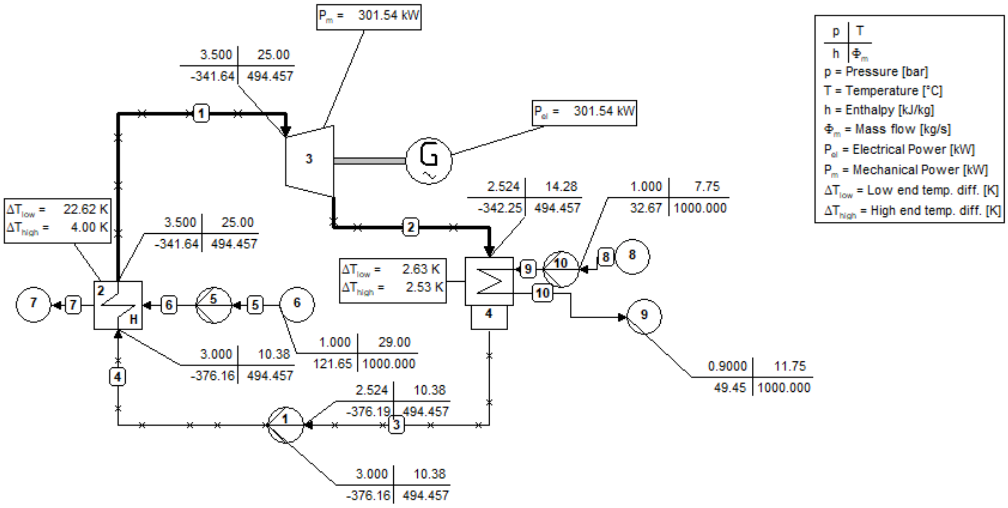


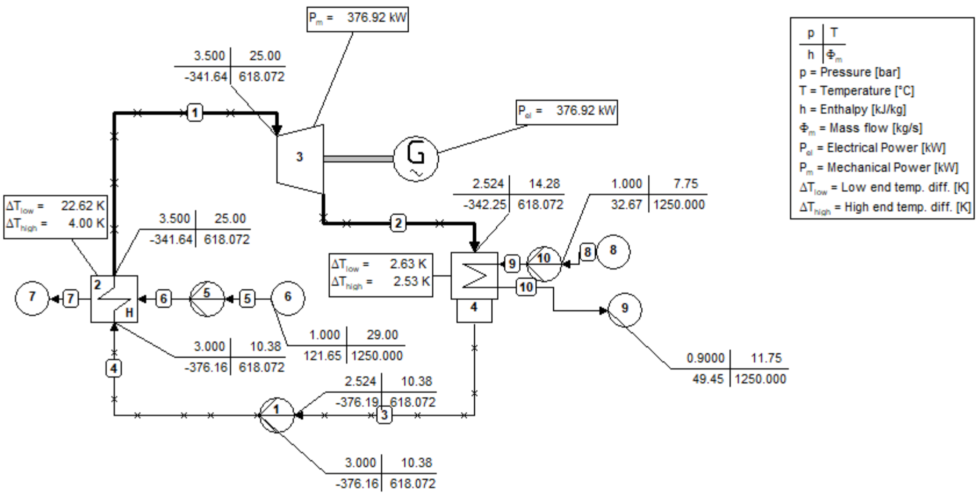
Scenario 4 d=700m sea water mass flow rate 500 kg/s



Scenario 4 d=700m sea water mass flow rate 750 kg/s



Scenario 4 d=700m sea water mass flow rate 1000 kg/s



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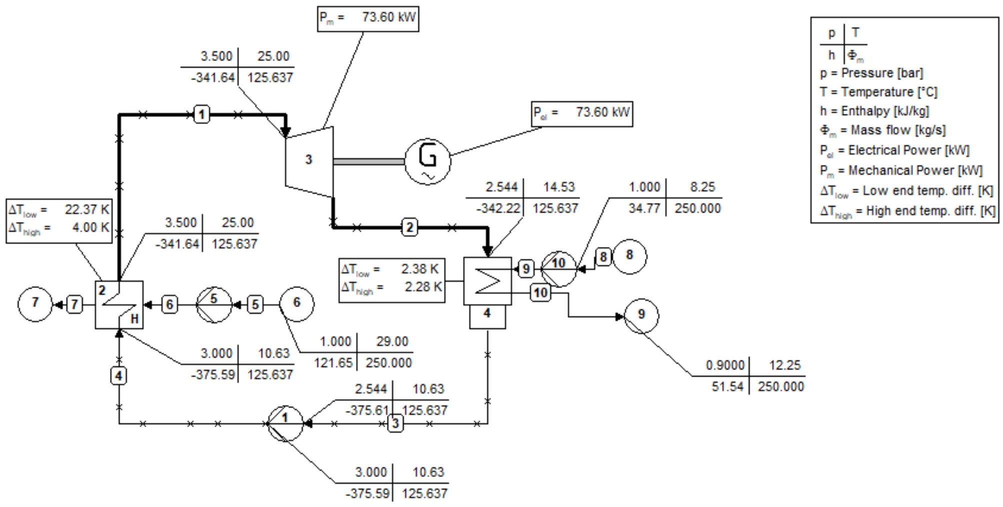

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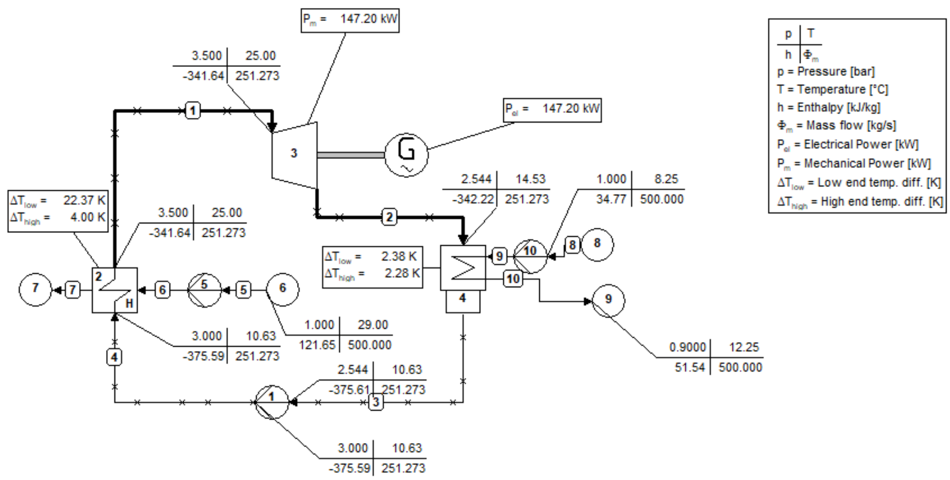
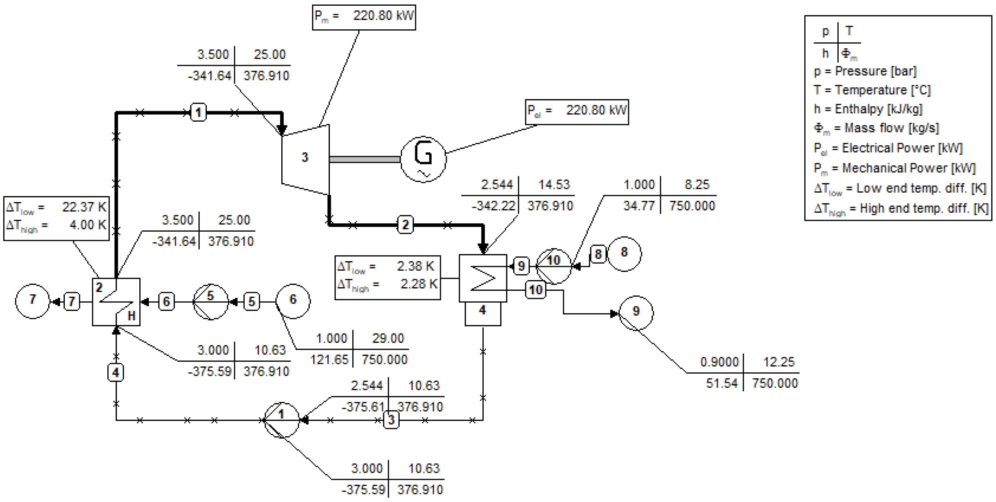
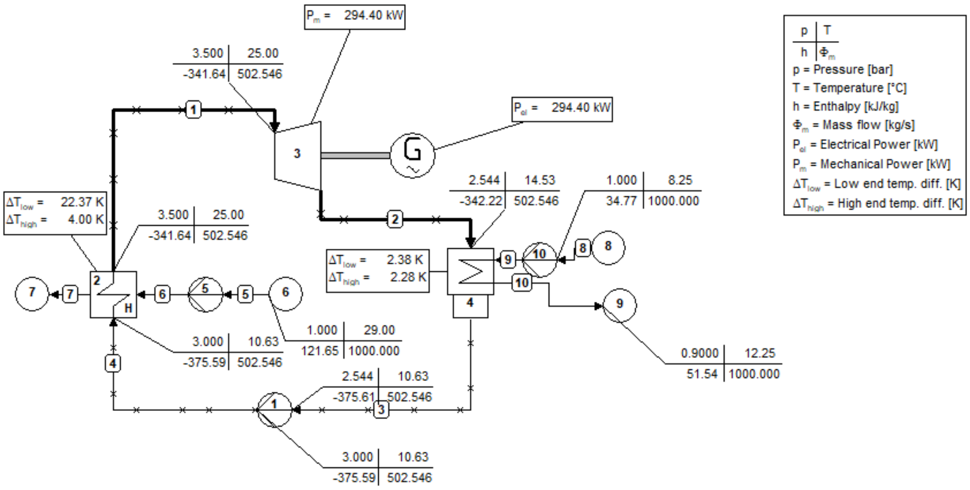


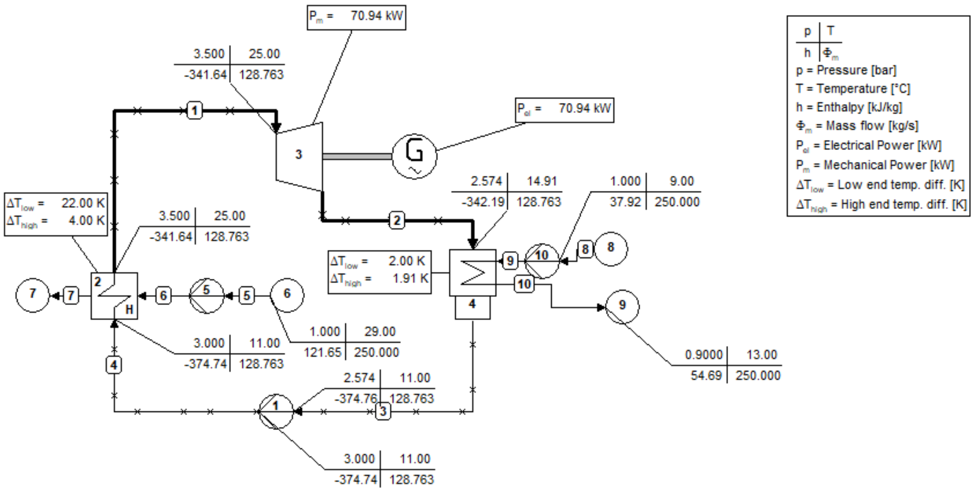
Scenario 5 d=600m sea water mass flow rate 500 kg/s



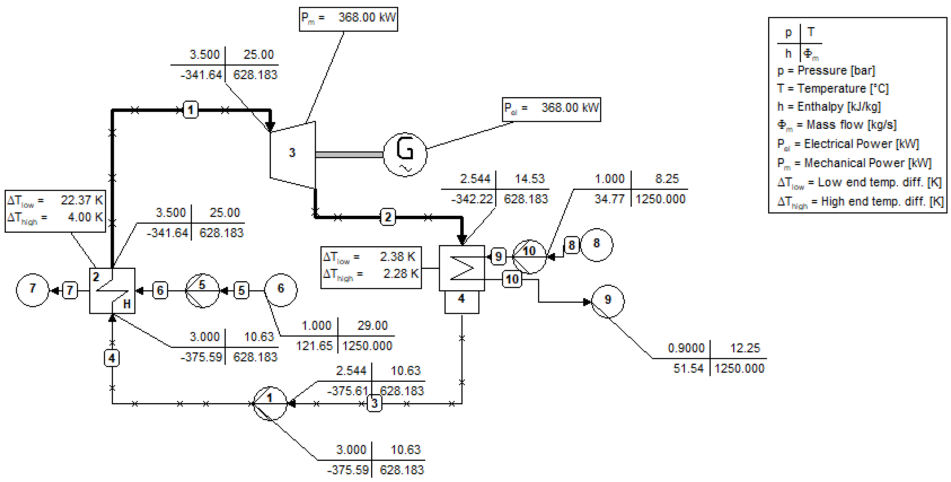
Scenario 5 d=600m sea water mass flow rate 750 kg/s



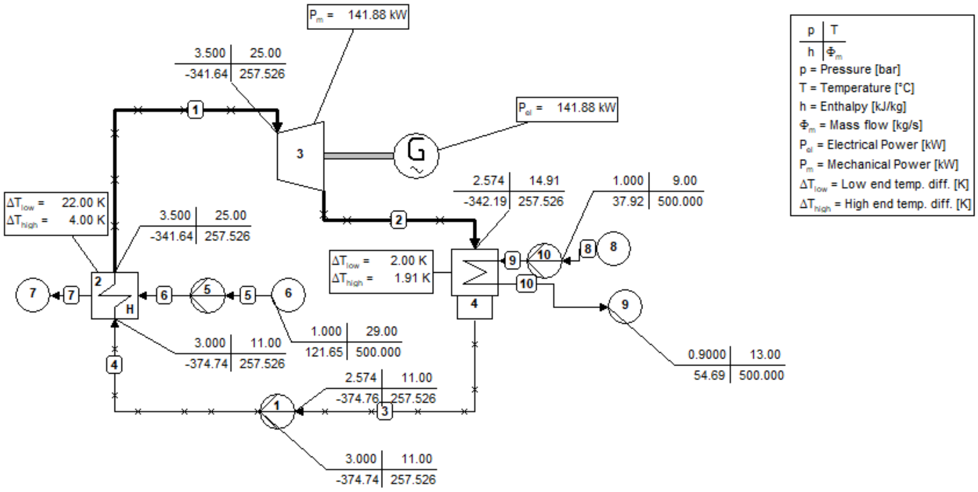
Scenario 5 d=600m sea water mass flow rate 1000 kg/s



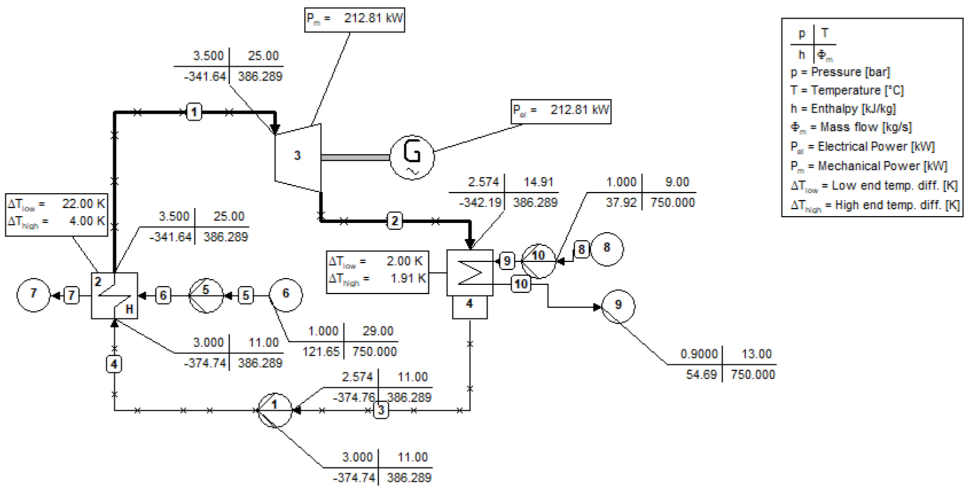
Scenario 5 d=600m sea water mass flow rate 1250 kg/s



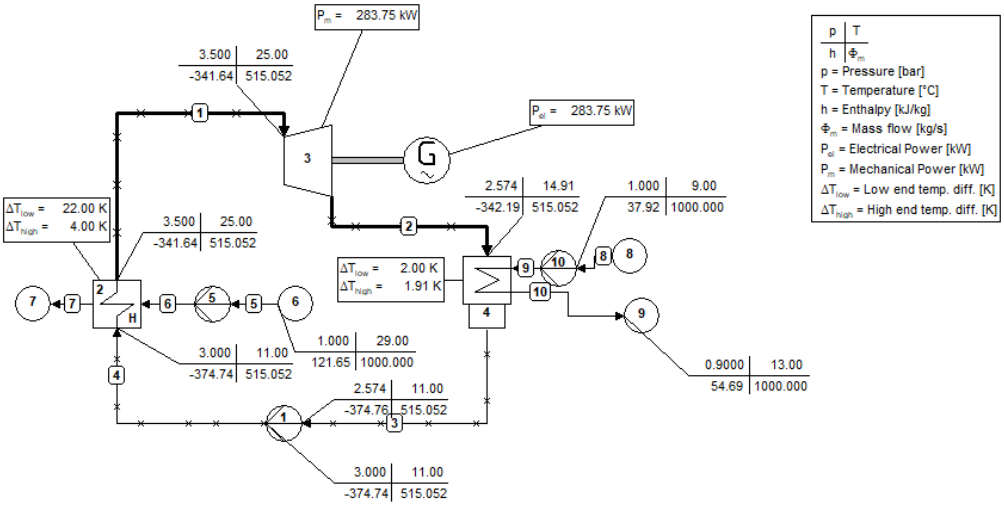
Scenario 6 d=500m sea water mass flow rate 250 kg/s



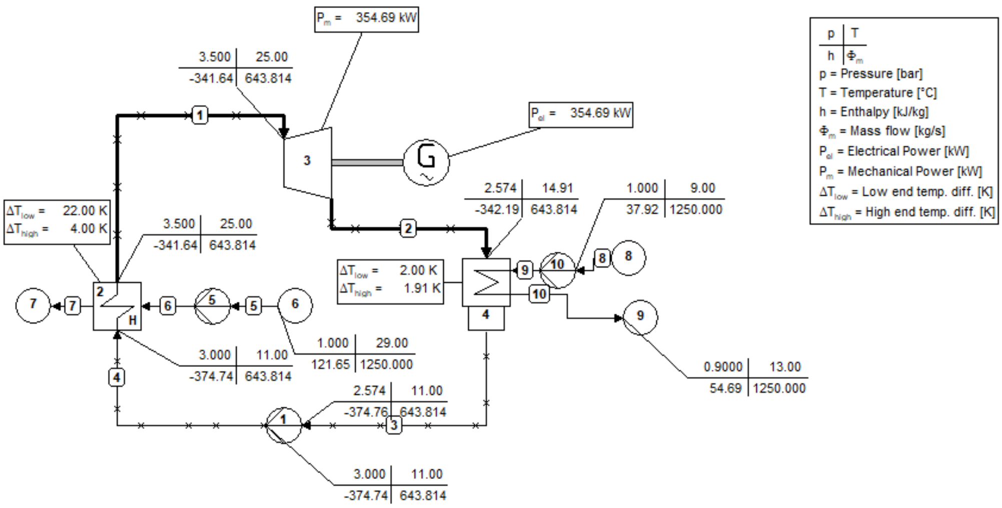
Scenario 6 d=500m sea water mass flow rate 500 kg/s



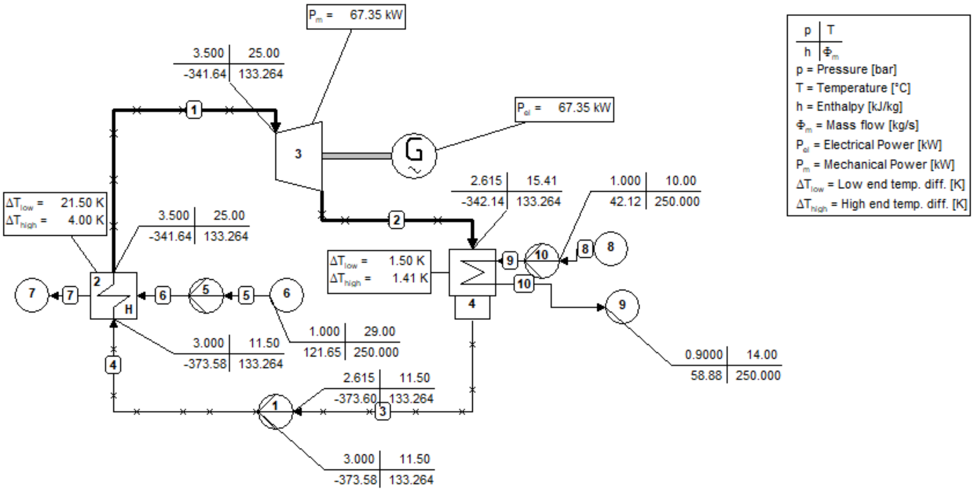
Scenario 6 d=500m sea water mass flow rate 750 kg/s



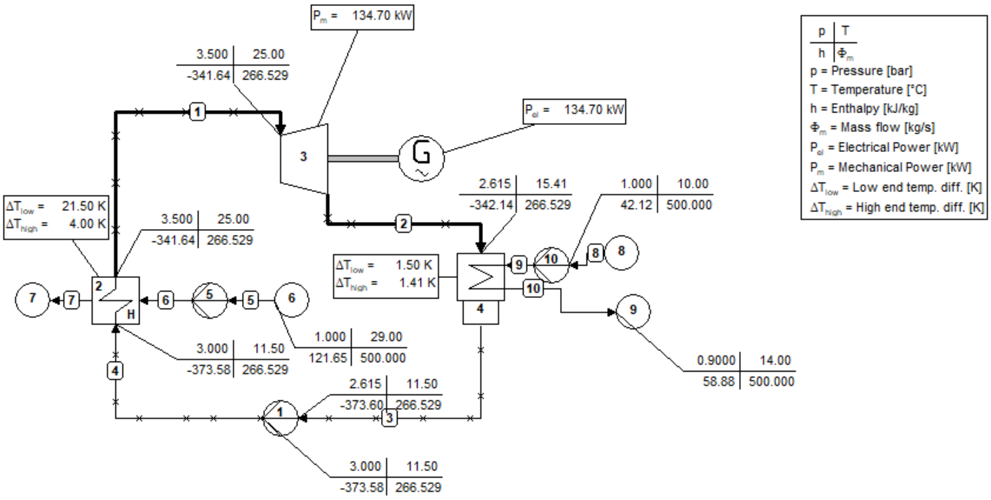
Scenario 6 d=500m sea water mass flow rate 1000 kg/s



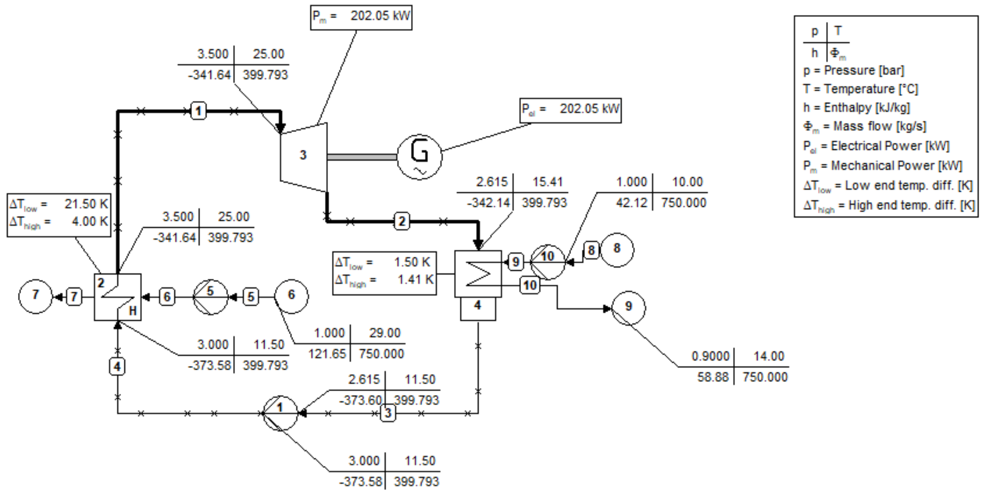
Scenario 6 d=500m sea water mass flow rate 1250 kg/s



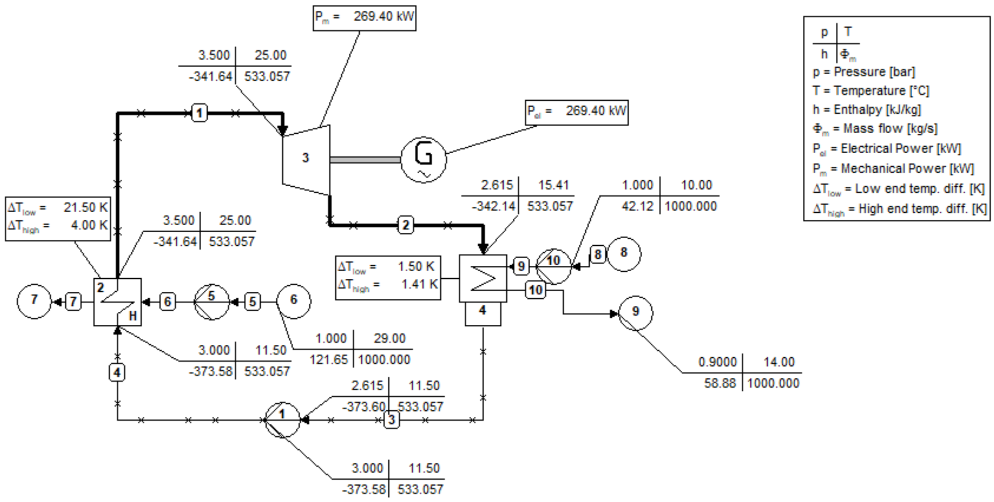
Scenario 7 d=400m sea water mass flow rate 250 kg/s



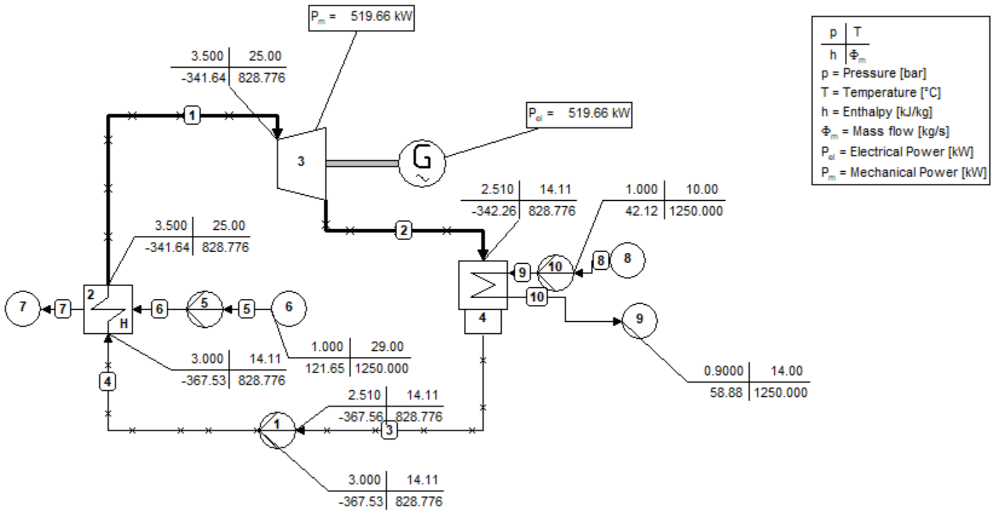
Scenario 7 d=400m sea water mass flow rate 500 kg/s



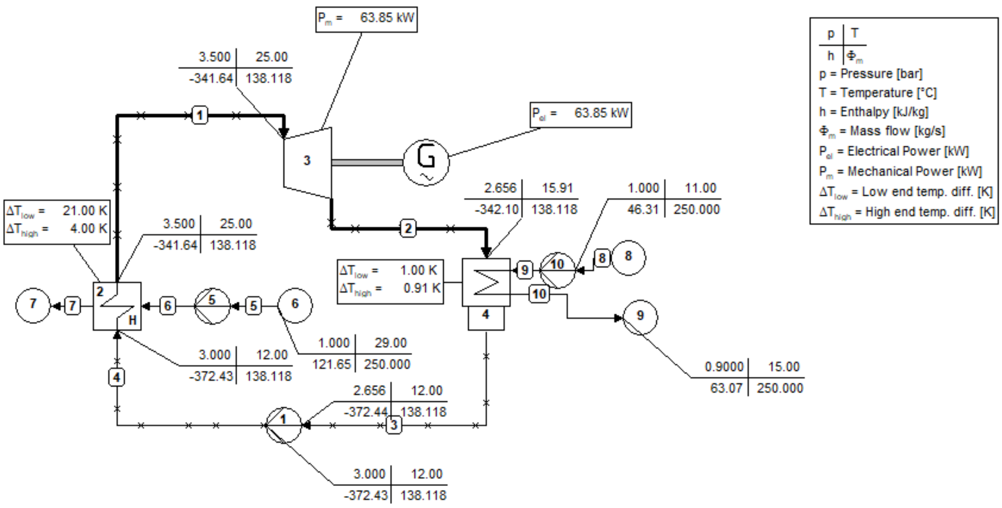
Scenario 7 d=400m sea water mass flow rate 750 kg/s



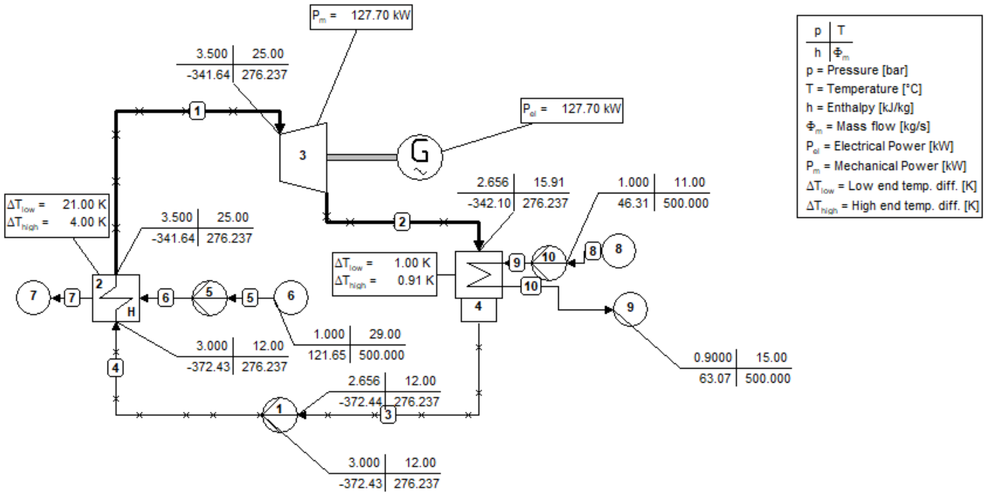
Scenario 7 d=400m sea water mass flow rate 1000 kg/s



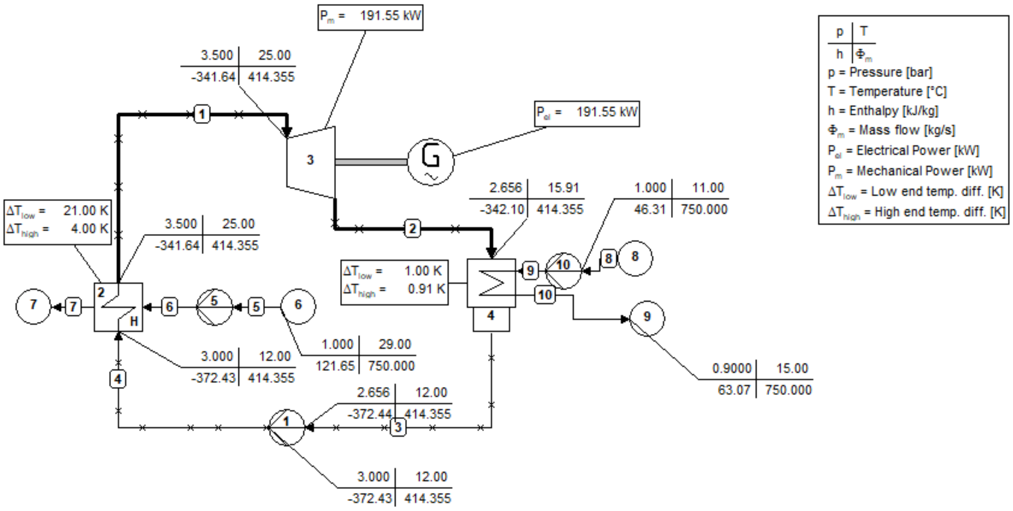
Scenario 7 d=400m sea water mass flow rate 1250 kg/s



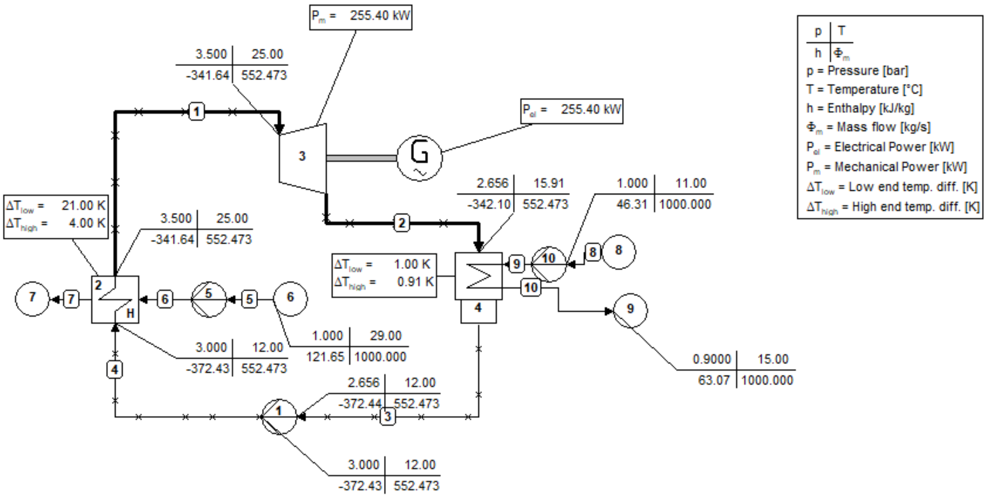
Scenario 8 d=300m sea water mass flow rate 250 kg/s



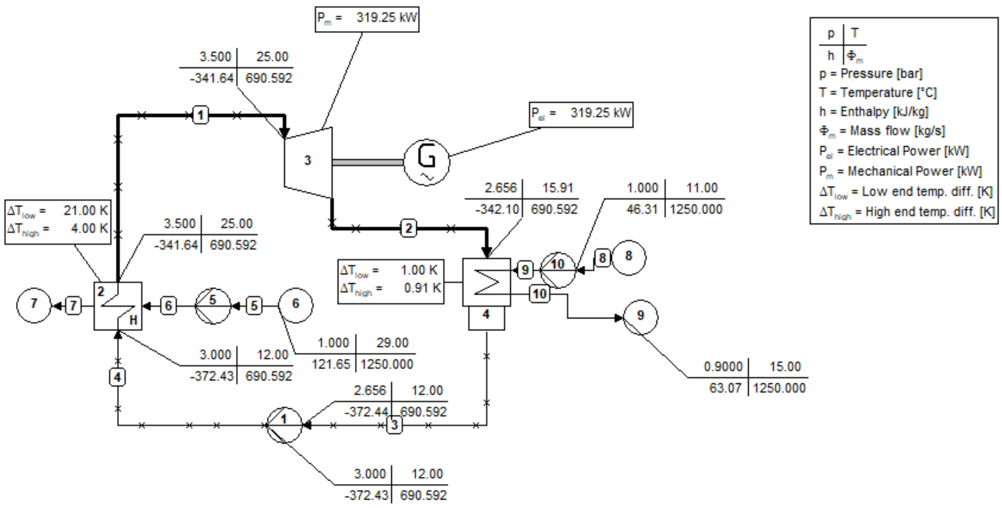
Scenario 8 d=300m sea water mass flow rate 500 kg/s



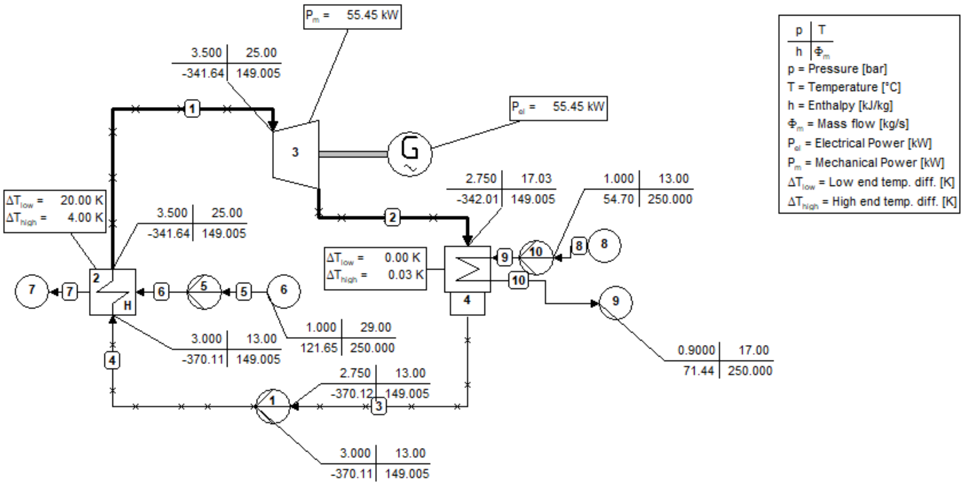
Scenario 8 d=300m sea water mass flow rate 750 kg/s



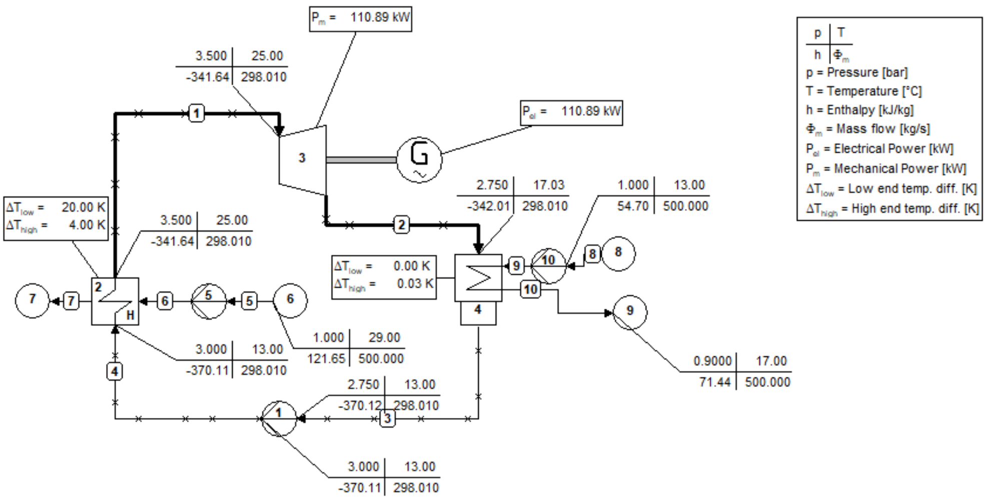
Scenario 8 d=300m sea water mass flow rate 1000 kg/s



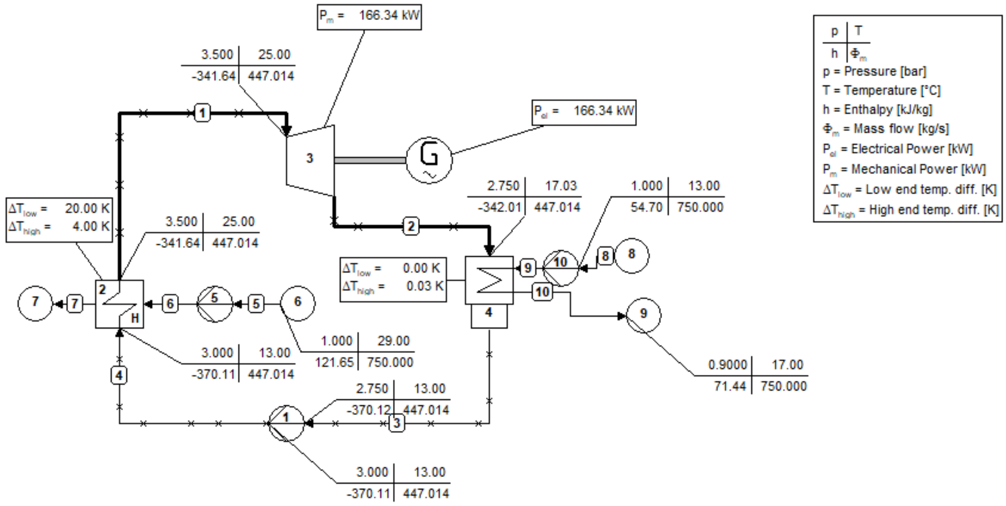
Scenario 8 d=300m sea water mass flow rate 1250 kg/s



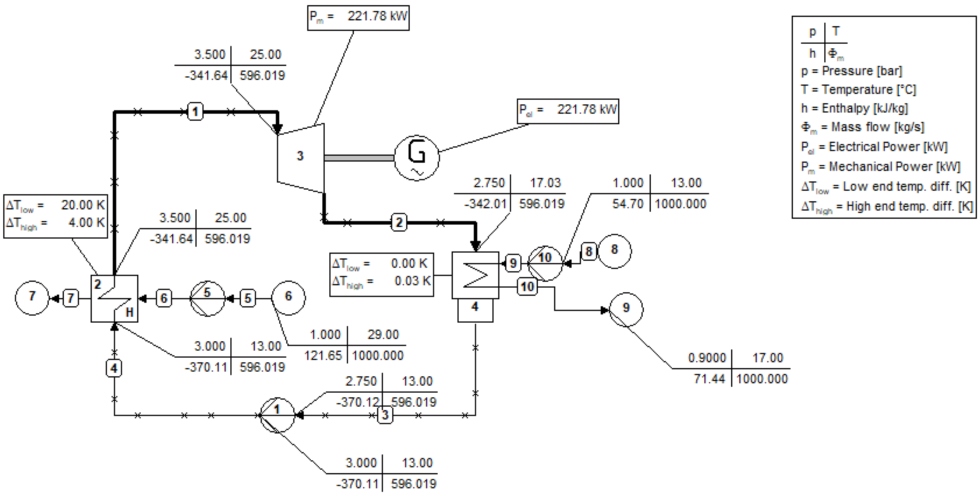
Scenario 9 d=200m sea water mass flow rate 250 kg/s



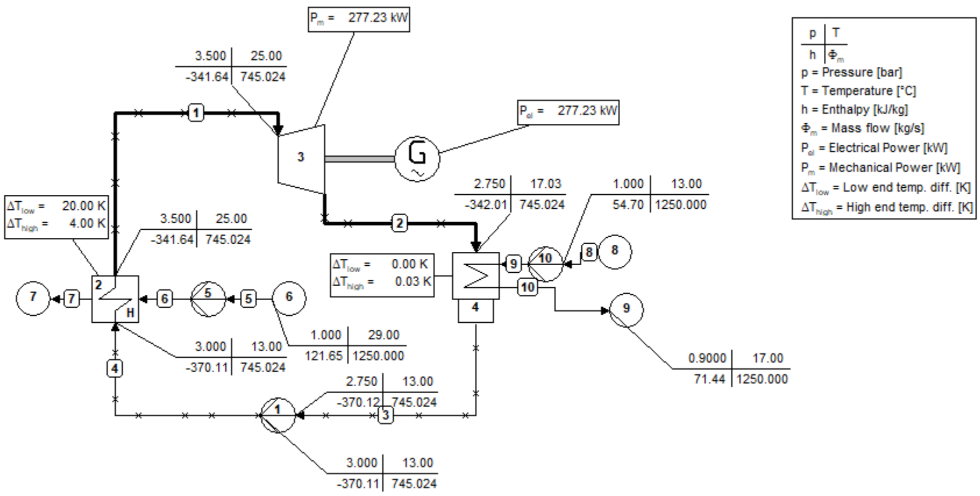
Scenario 9 d=200m sea water mass flow rate 500 kg/s



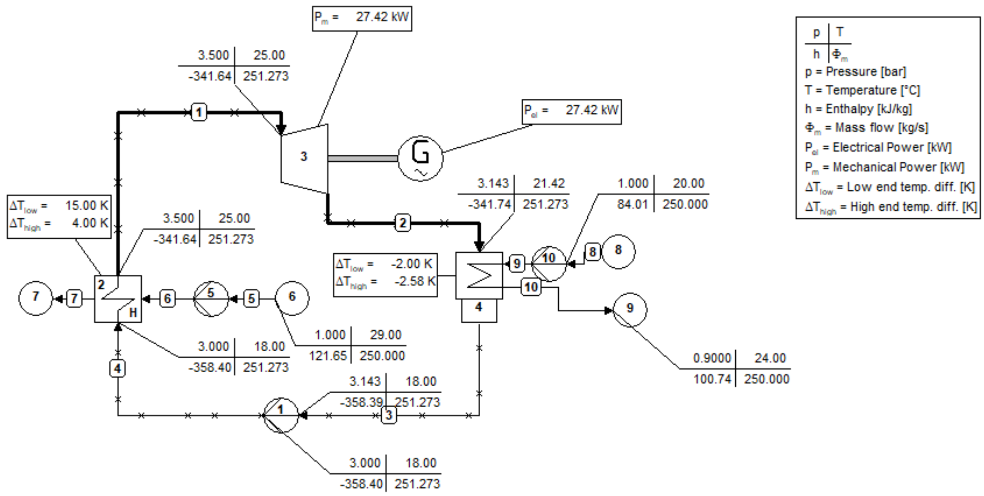
Scenario 9 d=200m sea water mass flow rate 750 kg/s



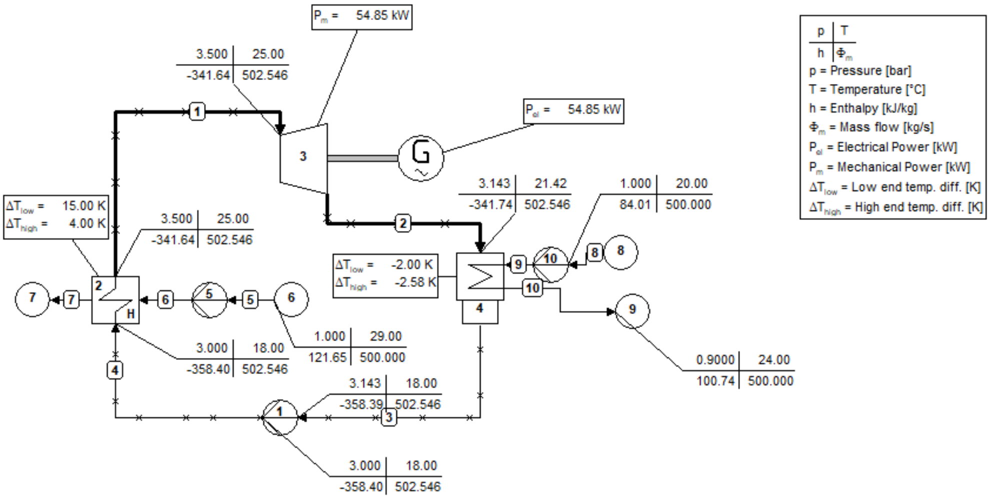
Scenario 9 d=200m sea water mass flow rate 1000 kg/s



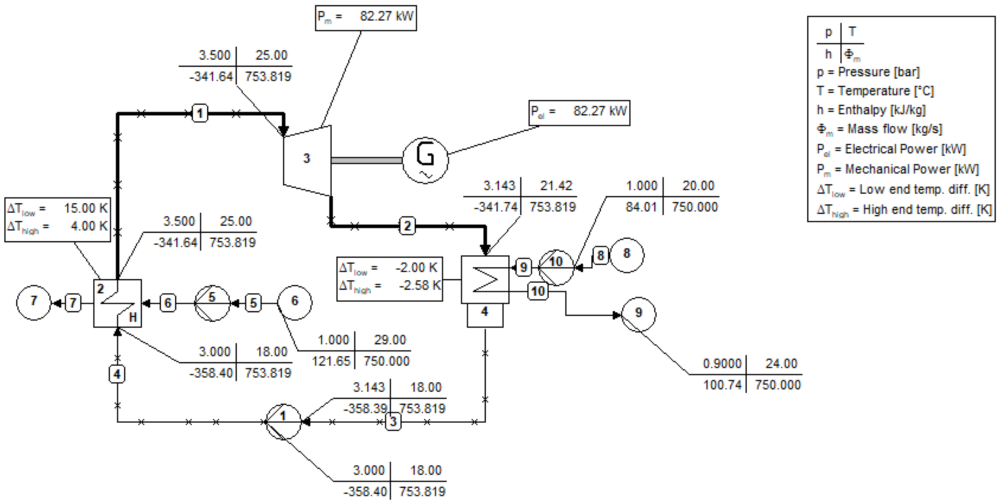
Scenario 9 d=200m sea water mass flow rate 1250 kg/s



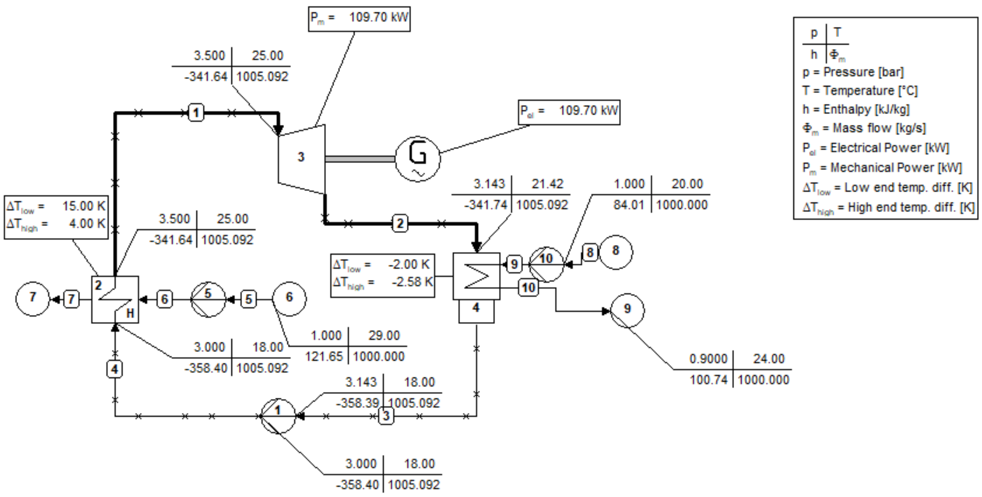
Scenario 10 d=100m sea water mass flow rate 250 kg/s



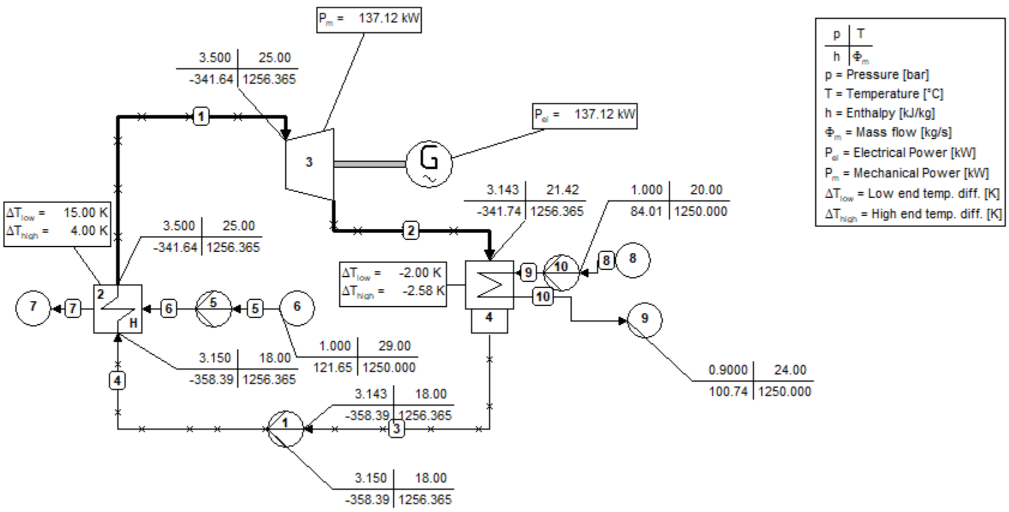
Scenario 10 d=100m sea water mass flow rate 500 kg/s



Scenario 10 d=100m sea water mass flow rate 750 kg/s



Scenario 10 d=100m sea water mass flow rate 1000 kg/s



Scenario 10 d=100m sea water mass flow rate 1250 kg/s

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ATTACHMENT II

Scenario 1 d= 1000 m Tc= 5 °C Tw= 29°C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
113.06	250	250	5.43	34.28	3.39	41.70	4261.41	0.98%
226.13	500	500	13.99	96.42	6.78	52.40	8522.81	0.61%
339.19	750	750	28.82	214.27	10.18	1.13	12784.22	0.01%
452.26	1000	1000	53.06	415.70	13.57	-143.13	17045.63	-0.84%
565.32	1250	1250	89.84	728.55	16.96	-411.36	21307.03	-1.93%

Scenario 2 d= 900 m Tc= 5.5 **°C** Tw= 29°C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
115.29	250	250	5.43	33.84	3.46	40.28	4279.49	0.94%
230.58	500	500	13.99	92.88	6.92	52.23	8558.98	0.61%
345.86	750	750	28.82	202.32	10.38	7.51	12838.47	0.06%
461.15	1000	1000	53.06	387.36	13.83	-122.23	17117.96	-0.71%
576.44	1250	1250	89.84	673.21	17.29	-365.30	21397.45	-1.71%

Scenario 3 d= 800 m Tc= 6.5 **°C** Tw= 29°C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
118.81	250	250	5.43	33.84	3.56	37.22	4273.70	0.87%
237.63	500	500	13.99	92.88	7.13	48.76	8547.41	0.57%
356.44	750	750	28.82	202.32	10.69	8.93	12821.11	0.07%
475.25	1000	1000	53.06	387.36	14.26	-107.93	17094.81	-0.63%
594.07	1250	1250	89.84	673.21	17.82	-327.51	21368.52	-1.53%

Scenario 4 d= 700 m Tc= 7.75 **°C** Tw= 29°C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
123.61	250	250	5.43	32.95	3.71	33.32	4267.16	0.78%
247.23	500	500	13.99	85.79	7.42	43.61	8534.35	0.51%
370.84	750	750	28.82	178.41	11.13	7.86	12801.50	0.06%
494.46	1000	1000	53.06	330.69	14.83	-96.97	17068.66	-0.57%
618.07	1250	1250	89.84	562.53	18.54	-293.88	21335.85	-1.38%

Scenario 5 d= 600 m Tc= 8.25 °C Tw= 29°C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
125.64	250	250	5.43	32.51	2.51	32.42	4265.38	0.76%
251.27	500	500	13.99	82.25	5.03	44.47	8530.72	0.52%
376.91	750	750	28.82	166.46	7.54	15.79	12796.09	0.12%
502.55	1000	1000	53.06	302.36	10.05	-73.99	17061.44	-0.43%
628.18	1250	1250	89.84	507.19	12.56	-245.24	21326.81	-1.15%

Scenario 6 d= 500 m Tc= 9 °C Tw= 29°C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
128.76	250	250	5.43	32.07	2.58	30.75	4262.06	0.72%
257.53	500	500	13.99	78.71	5.15	43.79	8524.11	0.51%
386.29	750	750	28.82	154.50	7.73	21.41	12786.17	0.17%
515.05	1000	1000	53.06	274.02	10.30	-54.11	17048.22	-0.32%
643.81	1250	1250	89.84	451.85	12.88	-200.46	21310.24	-0.94%

Scenario 7 **d=** **400** **m** **Tc=** **10** **^{o}C** **Tw=** 29^{o}C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
133.26	250	250	5.43	31.62	2.67	26.92	4256.45	0.63%
266.53	500	500	13.99	75.17	5.33	38.78	8512.94	0.46%
399.79	750	750	28.82	142.55	8.00	20.53	12769.39	0.16%
533.06	1000	1000	53.06	245.69	10.66	-42.88	17025.84	-0.25%
666.32	1250	1250	89.84	396.50	13.33	-166.51	21282.29	-0.78%

Scenario 8 **d=** **300** **m** **Tc=** **11** **^{o}C** **Tw=** 29^{o}C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
138.12	250	250	5.43	31.18	1.38	25.54	4252.65	0.60%
276.24	500	500	13.99	71.63	2.76	38.69	8505.34	0.45%
414.36	750	750	28.82	130.60	4.14	27.04	12757.99	0.21%
552.47	1000	1000	53.06	217.35	5.52	-21.80	17010.64	-0.13%
690.59	1250	1250	89.84	341.16	6.91	-120.24	21263.33	-0.57%

Scenario 9 **d=** **200** **m** **Tc=** **13** **^{o}C** **Tw=** 29^{o}C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
149.01	250	250	5.43	30.74	1.49	17.48	4242.17	0.41%
298.01	500	500	13.99	68.08	2.98	25.21	8484.34	0.30%
447.01	750	750	28.82	118.64	4.47	13.46	12726.49	0.11%
596.02	1000	1000	53.06	189.02	5.96	-27.51	16968.66	-0.16%
745.02	1250	1250	89.84	285.82	7.45	-107.45	21210.83	-0.51%

Scenario 10 **d=** **100** **m** **Tc=** **22** **^{o}C** **Tw=** 29^{o}C

mwf (kg/s)	mww (kg/s)	mcw (kg/s)	ww Pump (KW)	cw Pump (KW)	wf Pump (KW)	WT net (KW)	Qh(KW)	Efficiency
251.27	250	250	5.43	30.30	2.51	-14.35	4211.34	-0.34%
502.55	500	500	13.99	64.54	5.03	-33.30	8422.67	-0.40%
753.89	750	750	28.82	106.69	7.54	-67.66	12635.23	-0.54%
1005.09	1000	1000	53.06	160.68	10.05	-123.29	16845.34	-0.73%
1256.37	1250	1250	89.84	230.48	12.56	-207.25	21056.68	-0.98%

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ATTACHMENT III

Scenario 1

WT manual	WT net simulation	error
84.80	85.27	0.55%
169.60	170.53	0.55%
254.40	255.8	0.55%
339.19	341.07	0.55%
423.99	426.34	0.55%

Scenario 2

WT manual	WT net simulation	Error
83.01	83.52	0.61%
166.01	167.03	0.61%
249.02	250.55	0.61%
332.03	334.06	0.61%
415.04	417.58	0.61%

Scenario 3

WT manual	WT net simulation	error
79.60	79.76	0.19%
159.21	159.52	0.19%
238.81	239.29	0.20%
318.42	319.05	0.20%
398.02	398.81	0.20%

Scenario 4

WT manual	WT net simulation	error
75.40	75.38	0.03%
150.81	150.77	0.03%
226.21	226.15	0.03%
301.62	301.54	0.03%
377.02	376.92	0.03%

Scenario 5

WT manual	WT net simulation	error
72.87	73.6	0.99%
145.74	147.2	0.99%
218.61	220.8	0.99%
291.48	294.4	0.99%
364.35	368	0.99%

Scenario 6

WT manual	WT net simulation	error
70.82	70.94	0.17%
141.64	141.88	0.17%
212.46	212.81	0.16%
283.28	283.75	0.17%
354.10	354.69	0.17%

Scenario 7

WT manual	WT net simulation	error
66.63	67.35	1.07%
133.26	134.7	1.07%
199.90	202.05	1.07%
266.53	269.4	1.07%
333.16	336.75	1.07%

Scenario 8

WT manual	WT net simulation	error
63.53428	63.85	0.49%
127.069	127.7	0.49%
190.6033	191.55	0.49%
254.1376	255.4	0.49%
317.6723	319.25	0.49%

Scenario 9

WT manual	WT net simulation	error
55.13	55.45	0.57%
110.26	110.89	0.56%
165.40	166.34	0.57%
220.53	221.78	0.56%
275.66	277.23	0.57%

Scenario 10

WT manual	WT net simulation	error
26.55	27.42	3.17%
53.11	54.85	3.17%
79.66	82.27	3.17%
106.22	109.7	3.17%
134.80	139.21	3.17%

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Muhammad Indra Hazami is a student of Marine Engineering Department (Joint Degree : Hochschule Wismar, Germany), Faculty of Marine Technology, Institut Teknologi Sepuluh Nopember, and Specialized in Marine Machinery and System. He was born in Jakarta on 20th May 1997, He completed his Elementary School at SDIT Darul Abidin Depok in 2009, Junior High School at SMPIT As-Syifa Boarding School of Subang and SMPN 98 Jakarta in 2012, High School at SMAN 1 Depok in 2015. During his study in bachelor degree, he was active in several organizations and projects. He took 3 years experiences in ITS EXPO the biggest event in ITS as a Distrik Kreatif staff in ITS EXPO 2016, Distrik Kreatif Coordinator in ITS EXPO 2017 and Seni Budaya Conceptor in ITS EXPO 2019. He also took 2 years experiences in BEM ITS as staff of Ministry of Economic Affair in BEM ITS Wahana Juang 2017/2018 and as a Fundraising General Director of Ministry of Economic Affair in BEM ITS Gelora Aksi 2018/2019. For Futher discussion and suggestion regarding this bachelor thesis, the author can be reached through his email : indrahazami@gmail.com