Ocean Thermal Energy Conversions (OTEC) Working Fluid Comparison Based on The Numerical and Analytical Analysis

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***Abstract*— Despite its considerable potential, ocean thermal energy conversion (OTEC) has yet to be developed in Indonesia. As one of the most widely developed cycles, the Rankine cycle serves as the foundation for the analysis of OTEC systems. This paper presents a dual approach to determining the performance of OTEC systems. The two approaches are based on thermodynamic-based numerical methods and chemical process-based simulation approaches, the latter of which employs ASPEN Plus software. A comparison of OTEC system performance is conducted using two identical systems with two distinct working fluids, namely ammonia and R134a. The results of the method analysis indicate that the numerical methods employed yield a net power output discrepancy of less than 1% when compared to the simulation methods and benchmark data. In comparing the two systems with different working fluids, the system utilizing ammonia as the working fluid exhibits a slightly higher cycle efficiency, though not to a statistically significant degree, at approximately 0.2%.**

Keywords—ocean thermal energy conversions, numerical, simulation, working fluids

I. INTRODUCTION

Ocean Thermal Energy Conversion (OTEC) is one of the renewable energy sources that has great potential but has not been optimally utilized in Indonesia. Theoretically, the potential electrical energy from OTEC in Indonesia reaches 57 GW [1]. As a new technology in Indonesia, OTEC development requires simulation to ensure that the developed OTEC design can operate efficiently and produce energy according to the set target.

In theory, OTEC systems generate electricity by exploiting the temperature difference between the surface of the ocean, which is heated by the sun, and the deep water [2]. Based on the type of cycle, OTEC systems can be divided into three main categories, namely closed cycle, open cycle, and hybrid cycle. The closed cycle utilizes warm surface seawater to vaporize the working fluid to drive the turbine, where there is no direct contact between the seawater and the working fluid [3]. On the other hand, Open Cycle OTEC evaporates warm seawater as the working fluid to drive the turbine [4]. While Hybrid Cycle OTEC designed to combine the previous two types of cycles to produce working fluid vapor, which is then used as desalinated water [5].

In this paper, the OTEC Rankine cycle is used as the main cycle analyzed. The Rankine cycle is one of the simple closed cycles, but is widely used in closed thermodynamic cycles. Two of the power plants using the Closed Rankine Cycle are the KRISO 1 MW power plant and the Lockheed Martin 10 MW power plant [6].

In terms of energy production and efficiency of OTEC systems, especially in closed cycles, the working fluid is a factor that has a critical influence on system performance [7]. The optimal working fluid must have suitable thermodynamic properties and a good level of stability. The selection of the working fluid also affects the overall system efficiency [8], [9] Therefore, the working fluid for OTEC systems should maximized its thermal and cycle efficiency [10].

The OTEC plant with a closed-cycle system, comprising a 2.5-meter-diameter CWP and a seawater flow rate of 8,500 kg/s, has the potential to produce 2.325 MWe at an efficiency of 2.204% [11]. Furthermore, Giostri [12] discovered that with an identical cycle and a comparable seawater flow rate (8,798 for warm and 8,500 for cold seawater), an OTEC power plant could generate 2.35 MWe at 2.05% efficiency.

A closed-cycle ocean thermal energy conversion (OTEC) system operates using a working fluid with a low boiling point. The implementation of a higher-performing working fluid could potentially enhance the system's power output, reduce the irreversible losses inherent to the cycle, and improve the effective temperature difference. A review of previous studies indicates that ammonia exhibits superior performance compared to other working fluids, including R600a, R22, R32, R143a, R410a, R152a, and R134a [[13], [14], [15], [16]]. Despite its high performance, ammonia has been identified as a substance with significant health risks [17]. In light of these concerns, R134a is employed as a point of comparison and an alternative working fluid with a reduced risk profile.

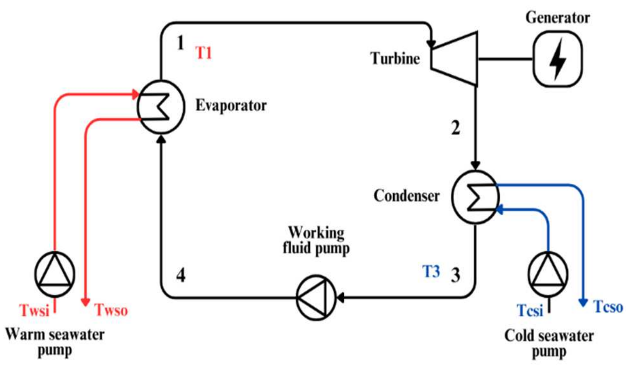

Fig. 1. OTEC Rankine cycle diagram.

As an evolving technology, the performance of OTEC systems cannot be determined by a single method. Rather, multiple methods must be employed to obtain a comprehensive understanding of the system's capabilities. This is crucial, particularly when the objective is to optimize the system design. In contrast to the aforementioned reference, this paper employed a chemical process simulation methodology utilizing ASPEN Plus software to guarantee the accuracy of the numerical methods. To compare the performance of the OTEC system, particularly in terms of system performance in power output, this paper utilizes two distinct types of working fluids, namely ammonia and R134a. The net power output and efficiency of the system are utilized to evaluate the performance of the OTEC cycle.

This paper presents an analysis of two methods, with a focus on their controlled parameters. Prior to a comparative evaluation of the methods, each result is validated against the reference data to ensure the accuracy and reliability of the methods employed. The paper then proceeds with a detailed comparison of the two methods and the two working fluids, accompanied by a discussion of the findings and their implications.

II. Methods

In some existing studies on OTEC system performance, such as those conducted by Yoon et al. [8], Giostri et al. [9], and Mao et al. [18], the analysis method is based on a single approach without comparison. Given that OTEC power plants are still a developing technology, the use of multiple approaches in system performance analysis can enhance the accuracy of the analysis results. In consideration of this, this paper employs two approaches with two distinct methods.

The analysis of the performance of the OTEC system is based on two factors: the power generated by the system and the overall efficiency of the system. The analysis is conducted using two methods: a Numerical method based on heat transfer in the working fluid throughout the cycle, and a Simulation method based on the chemical processes occurring within the cycle, which is carried out using ASPEN Plus software.

A. Numerical Methods

Rankine cycle calculation is divided into four points, as shown in Fig. 1, where each point represents its main components. In the Rankine cycle, the working fluid (in liquid state) is delivered to the evaporator by the working fluid pump (3 => 4), and heat transfer between the surface seawater and the working fluid occurs in the evaporator, producing saturated steam (4 => 1). The steam then drives the turbine and expands to a lower pressure (1 => 2). After the steam leaves the turbine, the working fluid enters the condenser, while exchanges heat with cold seawater, the working fluid's steam condenses into a liquid state to repeat the cycle (2 => 3).

The total heat transfer in heat exchanger determines the total energy produced by the Rankine cycle. The higher the heat flow rate in both heat exchangers (evaporator and condenser) the higher energy that could be produced by the entire Rankine cycle. The total heat flow rate in each heat exchanger is formulated in Eq. 1-3 for evaporator and Eq. 4-6 for condenser.

$$Q_e = m_{WF}(h_1 - h_4) \quad (1)$$

$$Q_e = UA(\Delta Tm_e) \quad (2)$$

$$Q_e = m_{WS}c_{P,WS}(T_{wsi} - T_{wso}) \quad (3)$$

$$Q_c = m_{WF}(h_2 - h_3) \quad (4)$$

$$Q_c = UA(\Delta Tm_c) \quad (5)$$

$$Q_c = m_{CS}c_{P,CS}(T_{cso} - T_{csi}) \quad (6)$$

Where:

$Q_{e,c}$ = Heat flow rate in heat exchanger (kW)

m_{WF} = Mass flow rate of working fluid (m^3/kg)

$m_{CS,WS}$ = Mass flow rate of cold and warm seawater (m^3/kg)

h = Working fluid enthalpy at each point (kJ/kg)

$c_{P,WS,CS}$ = Specific heat capacity (J/kg.K)

$\Delta Tm_{e,c}$ = Temperature changes of the heat exchanger

$T_{csi,o}$ = Cold seawater temperature at condenser inlet and outlet (K)

$T_{wsi,o}$ = Warm seawater temperature at evaporator inlet and outlet (K)

U = Heat transfer unit of heat exchanger (W/m^2K)

A = Heat transfer area of heat exchanger (m^2)

As shown in equations 16, the heat flow rate in the heat exchanger is strongly influenced by the specifications of the components, including the mass flow rate in working fluid (m_{WF}), warm seawater (m_{WS}) or cold seawater (m_{CS}), and the heat transfer area (A) and heat transfer unit (U) of the heat exchanger. Each of those components will also influence total heat flowrate throughout the cycle loops. The enthalpy at each point (1-4), will shows how the heat flowrate happens throughout the cycle, while also determine how much energy generated by the cycle.

The enthalpy in Point 1 & 3 (evaporator and condenser) are determines by using thermodynamics table, with each point temperature calculated using Logarithmic Mean Temperature Difference (LMTD) as formulated in Eq. 7 & 8.

$$T_1 = \frac{e^{\left[\frac{T_{wsi}-T_{wso}}{\Delta Tm_e}\right]} T_{wso} - T_{wsi}}{e^{\left[\frac{T_{wsi}-T_{wso}}{\Delta Tm_e}\right]} - 1} \quad (7)$$

$$T_3 = \frac{e^{\left[\frac{T_{cso}-T_{csi}}{\Delta T m_c}\right]} T_{cso}-T_{csi}}{e^{\left[\frac{T_{cso}-T_{csi}}{\Delta T m_c}\right]}-1} \quad (8)$$

$$T_{wso,cso} = T_{wsi,csi} - \Delta T_{e,c} \quad (9)$$

The enthalpy in Point 2 & 4 (Turbine and Working fluid pump) are determined with a difference approach. In the point 2, the enthalpy is determined based on the quality of the steam (x_2). While the enthalpy of Point 4 is based on the working fluid's pressure difference between Point 4 and 3. The enthalpy in both Point 2 and 4 are formulated in Equations 11 & 14 as follows:

$$x_2 = {(s_1 - s_2)}/{s_{fg,2}} \quad (10)$$

$$h_2 = h_{f,2} + x_2 h_{fg,2} \quad (11)$$

$$P_4 = P_1 \quad (12)$$

$$wp_4 = -v(P_4 - P_3) \quad (13)$$

$$h_4 = h_3 - wp_4 \quad (14)$$

where:

$\Delta T_{e,c}$ = Seawater temperature change after getting out of evaporator and condenser

s = Working fluid's entropy at each point (kJ/mol)

s_{fg} = Working fluid's difference entropy values between saturated liquid and vapor (kJ/mol)

h_f = Working fluid enthalpy of saturated liquid (kJ/kg)

h_g = Working fluid enthalpy difference between saturated liquid and vapor (kJ/kg)

P = Working fluid pressure at each point (kPa)

v = Velocity of the working fluid (m/s)

The enthalpy changes between each working fluid then used to calculated the generated power by generator and the power used to power the working fluid pump. The generated power and parasitic power of working fluid pump are calculated using Equations 15 & 16.

$$W_G = m_{WF}(h_1 - h_2)\eta_T \eta_G \quad (15)$$

$$W_{P,WF} = {m_{WF}(h_3 - h_4)}/{\eta_{P,WF}} \quad (16)$$

Other than the parasitic power needed by the working fluid pump, the parasitic power in OTEC system also include the energy to power warm and cold seawater pumps. The power needed by the warm and cold seawater pump are formulated in Equations 17 & 18.

$$W_{P,WS} = {m_{WS}\Delta P_{WS}}/{\rho_{WS}\eta_{P,SW}} \quad (17)$$

$$W_{P,CS} = {m_{CS}\Delta P_{CS}}/{\rho_{CS}\eta_{P,SW}} \quad (18)$$

where:

s = Working fluid's entropy at each point (kJ/mol)

s_{fg} = Working fluid's difference entropy values between saturated liquid and vapor (kJ/mol)

h_f = Working fluid enthalpy of saturated liquid (kJ/kg)

h_g = Working fluid enthalpy difference between saturated liquid and vapor (kJ/kg)

The total pressure drops of seawater, represented by $\Delta P_{CS,WS}$, occurs within the seawater pipe and heat exchanger. In the heat exchanger, the pipe positions are assumed to be horizontal. In contrast, the pipe position within the seawater pipe is inclined. The formulated equations for both pipes are presented in Eq. 19, while the friction factor is determined using Eq. 20.

$$\Delta P_H = f \frac{l_{WS,HE}\rho v^2}{2D} \quad (19)$$

$$\frac{1}{\sqrt{f}} = -2log\left[\frac{2.51}{Re\sqrt{f}} + \frac{k}{3.7D}\right] \quad (20)$$

where:

f = Friction factor

D = Pipe diameter (m)

$l_{WS,CS,HE}$ = Warm and cold seawater, and heat exchanger pipe length (m)

v = Seawater velocity (m/s)

ρ = Seawater density (kg/m3)

g = Gravity (m/s^2)

Re = Reynolds Number

k = Pipe roughness (m)

The net power output ($\bar{W}$) of OTEC system is the equivalent of the amount of power generated by the turbine (W_G) minus the amount of power required to operate the working fluid pump (W_P), warm seawater ($W_{P,WS}$), and cold seawater ($W_{P,CS}$). While, the thermal efficiency (η_{th}) is the net power output divided by the total heat flow rate in evaporator. Both parameters are formulated in Equations 21 & 22 as follows:

$$\bar{W} = W_G - W_{P,WF} - W_{P,WS} - W_{P,CS} \quad (21)$$

$$\eta_{th} = \frac{\bar{W}}{Q_e} \quad (22)$$

B. Simulation methods

ASPEN Plus software simulates the Rankine cycle OTEC cycle by approaching the chemical processes of the cycle. The use of ASPEN Plus software can simplify the complex thermodynamic modeling process. In this paper, ASPEN Plus version 11 is used with benchmark data in the form of simulation results from a single Rankine cycle.

There are several basic methods that can be used to simulate Rankine cycle chemical processes. One that is often used is the Peng-Robinson (Peng-Rob) method. The Peng-Robinson method has the ability to estimate the thermophysical behavior of complex fluid mixtures quite well, including in OTEC systems that involve phase changes of the working fluid. This method takes into account the interaction between molecules, the effects of pressure and temperature on the physical properties of the mixture, and can therefore provide fairly accurate simulation results.

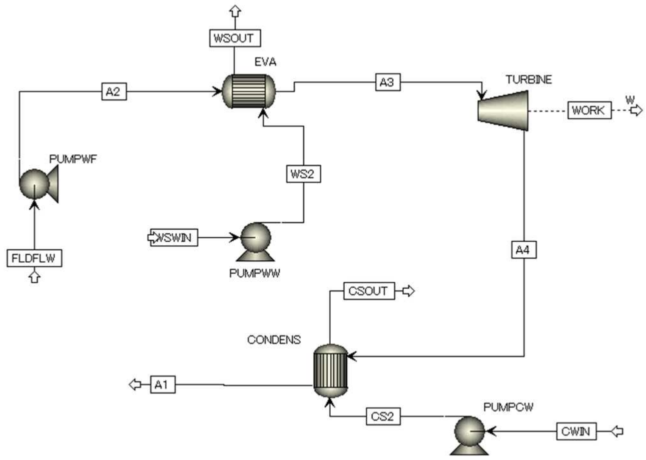

Fig. 2. Rankine cycle circuit diagram in ASPEN Plus Software.

TABLE I. PARAMETERS CONFIGURATION.

Parameters	Symbols	Value
Warm seawater inlet temperature (°C)	T_{wsi}	26
Cold seawater inlet temperature (°C)	T_{csi}	4.5
Seawater pressure (kPa)		100
Generator efficiency (%)	η_G	94
Turbine efficiency (%)	η_T	75
Working fluid pump efficiency (%)	$\eta_{P,WF}$	72
Seawater pump efficiency (%)	$\eta_{P,SW}$	72
Thermal conductivity of heat exchanger (kW/m^2K)	U	4.693
Total heat transfer area of evaporator (m^2)	A_e	41500
Total heat transfer area of condenser (m^2)	A_c	42500
Evaporator minimum approach temperature (°C)	ΔT_e	1.2
Condenser minimum approach temperature (°C)	ΔT_c	1.0
Mass flowrate of working fluid (kg/s)	m_{WF}	580
Mass flowrate of warm seawater (kg/s)	m_{WF}	50000
Mass flowrate of cold seawater (kg/s)	m_{WF}	28450
Seawater velocity through the pipe (m/s)	v_{sw}	1.5
Seawater pipe diameter (m)	D	1
Length of warm seawater pipe (m)	l_{WS}	100
Length of cold seawater pipe (m)	l_{CS}	2800
Seawater pipe roughness (m)	k_{SW}	0.061
Heat exchanger pipe roughness (µm)	k_{WF}	0.25
Warm water loop loos (kPa)		30
Cold water loop loss (kPa)		72
Evaporator loss/stage (kPa)		6
Condenser loss/stage (kPa)		6

TABLE II. THERMODYNAMICS PROPERTIES OF WORKING FLUIDS.

Parameters	NH3	R134a
Molecular weight (g/mol)	17.031	102.032
Boiling temperature (°C)	-33.33	-26.30
Melting Temperature (°C)	-77.73	-103.30
Critical temperature (°C)	132.41	101.21
Specific Heat (kJ/kg·K)	4.776	1.421
Liquid Thermal Conductivity (mW/mK @24 °C)	488.40	81.60
Evaporation Enthalpy (kJ/kg, @24 °C)	1169.95	178.7
Evaporation Entropy (kJ/kg-K, @24 °C)	3.9372	0.6014
ASHRAE	B2L	A1

TABLE III. COMPARISON OF SIMULATION AND NUMERICAL RESULTS WITH JOURNAL ON AMMONIA WORKING FLUID.

	Reference [17]	**Simulation**	**Numerical**
Net Power (kW)	18389	18377	18367
Parasitic Power (kW)	5344	5296	5590.5

The Rankine cycle simulation stage using ASPEN Plus software begins by compiling the Rankine cycle circuit block, as shown in Fig. 2. The blocks used are pump, heat exchanger, and turbine just like the Rankine cycle in general. The system can be said to be in balance if the main properties such as temperature, pressure, volume, and phase at the time of input are the same as the stream properties at the end of the system. After the circuit is created, several parameters can be entered from the stream in the working fluid, warm sea water input, and cold sea water input.

III. PARAMETERS

As the basis of calculations and simulations, to compare the results of the two methods, the same parameter values are used between each method, the values of each parameter used are listed in Table 1. However, not all parameters are used in each method due to the different needs between calculations and simulations. However, the main parameters such as seawater temperature, working fluid type, efficiency of each component and mass flow rate have the same values between the two methods.

In addition to the technical parameters, the same type of working fluid is used in this paper. As a benchmarking requirement between the two methods and the reference, ammonia (NH3) working fluid is used. While as a comparison to determine the most optimal working fluid to be applied to OTEC systems, R134a working fluid is used. The thermodynamic properties of the two working fluids are shown in Table 2.

IV. RESULTS

A. Benchmarking

Prior to the performance comparison between NH3/ammonia and R134a working fluids, benchmarking was carried out to ensure that the Rankine cycle scheme used in the ASPEN Plus software was correct according to the reference of Bharathan D [19]. In addition, the calculation method was also benchmarked to ensure that the two methods gave similar results to each other.

TABLE IV. RESULTS COMPARISON OF DIFFERENCE WORKING FLUIDS.

Parameters	Ammonia	R134a
Mass flowrate of working fluid (kg/s)	580	3970
Net power output (kW)	12776.5	12568.5
Cycle efficiency (%)	1.8	1.6
Generated power (kW)	18367	18522.1
Cold seawater pump parasitic power (kW)	3092.8	3092.8
Warm seawater pump parasitic power (kW)	2158.1	2158.1
Working fluid pump parasitic power (kW)	339.6	702.7

As illustrated in Table 3, the discrepancy between the two analytical approaches is in alignment with the established reference values. With regard to the net power output parameter, the difference between the two methods and the results presented in [19] is found to be just under 1%. While the parasitic power difference in the numerical method is slightly larger than the reference, reaching approximately 250 kW, said difference remains within the error range of 5%. Based on these findings, it can be concluded that both methods have been validated for use in comparing the performance of OTEC systems utilizing two difference working fluids.

B. Comparation

A comparative analysis of the performance of the OTEC system was conducted between two configurations, one utilizing ammonia as the working fluid and the other employing R134a. As illustrated in Table 4, the two systems yield comparable outcomes when operating at the identical target net power output of 1.2 MW. As both systems utilize the identical seawater flow rate of 50000 kg/s for warm and 28450 kg/s for cold seawater, the seawater pumping power is identical for both. Consequently, there is a considerable discrepancy in the power required by the working fluid pump and the generator to achieve the target net power output, taking into account working fluid pump parasitic power.

In accordance with the considerable disparity in enthalpy values between ammonia and R134a, as evidenced in Table II, the requisite flow rate of the working fluid between the two systems also exhibits a notable divergence. The flow rate requirement of the system with R134a as the working fluid is approximately 685% of the working fluid flow rate of the system with ammonia. The discrepancy in the flow rate of the working fluid also results in a twofold increase in the power requirement for the pump in the system utilizing R134a as the working fluid. These findings align with those of Samsuri et al. [17] and Chen et al. [14], who also observed superior performance of a closed-cycle OTEC system with ammonia as the working fluid compared to other options.

Furthermore, with regard to cycle efficiency, the system utilising an ammonia working fluid exhibits a 0.2% higher efficiency value than R134a. This value is lower than the results of Giostri et al. [12], where the closed cycle system efficiency can reach 2%. However, the discrepancy is still within the acceptable range, given the differences in component specifications, particularly the seawater pipe and heat exchanger utilized. Both components exert a considerable influence on the parasitic power of the OTEC system.

V. CONCLUSIONS

This paper presents a comparative analysis of two working fluids, namely NH3/ammonia and R134a, with a focus on their performance in the Rankine cycle in terms of power output and efficiency. A numerical calculation method was employed to facilitate a comparison between the two working fluids. The formulation was initially standardized by comparing the calculation results with the simulation results obtained from ASPEN Plus software and reference journals. The net power results of the calculation exhibit a discrepancy of less than 1% in comparison to the results of the reference. The parasitic power results exhibit a discrepancy of less than 5%.

A comparison of the two working fluids reveals that the cycle utilizing ammonia as the working fluid necessitates a significantly lower working fluid mass flow rate than R134a. The discrepancy in mass flow rate is nearly sixfold. Furthermore, a notable discrepancy is observed in the parasitic power consumption of the working fluid pump, which is approximately twice as high in one cycle compared to the other.

Moreover, a numerical method that has been validated through a chemical processes simulation approach can be employed to ascertain the optimal combination of component and working fluid specifications for OTEC systems. Particularly, in the context of performance enhancement with a predetermined net power output target.

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