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Thermal Efficiency Enhancement of Ocean Thermal Energy Conversion (OTEC) Using Solar Thermal Energy

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Ocean thermal energy conversion (OTEC) is a power generation system that utilizes small temperature differences between the surface and deep sea water. This paper describes the simulation results of the thermal efficiency of an OTEC plant that utilizes not only ocean thermal energy but also solar thermal energy as a heat source. This power generation system is termed SOTEC (solar and ocean thermal energy conversion). Simulation results of a 100-kWe SOTEC plant, in which the solar collector is installed in two configurations, are obtained and compared with those of an ordinary OTEC plant. The simulation result shows that the proposed SOTEC plant can enhance the annual mean thermal efficiency up to 1.5 times higher than that of the ordinary OTEC plant.

Nomenclature

A_E	=	heat transfer surface area of evaporator
A_C	=	heat transfer surface area of condenser
A_{SC}	=	aperture area of solar collector
c_{pCS}	=	specific heat of cold sea water
c_{pWS}	=	specific heat of warm sea water
g	=	acceleration of gravity
m_{CS}	=	mass flow rate of cold sea water
m_{WF}	=	mass flow rate of working fluid
m_{WS}	=	mass flow rate of warm sea water
$h_{1,2,3,4}$	=	enthalpy at the numbered points of the *T-s* diagram
ΔH_{CS}	=	total pressure difference (pressure head) of cold sea water piping
ΔH_{WF}	=	total pressure difference (pressure head) of working fluid piping
ΔH_{WS}	=	total pressure difference (pressure head) of warm sea water piping
I	=	solar radiation on the collector surface
P_G	=	turbine generate power
P_N	=	net power
P_{CS}	=	cold sea water pumping power
P_{WF}	=	working fluid pumping power
P_{WS}	=	warm sea water pumping power

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Q_C	=	heat flow rate of condenser
Q_E	=	heat flow rate of evaporator and solar collector
s	=	entropy
T_a	=	ambient temperature
T_C	=	condensing temperature
T_{CSI}	=	cold sea water inlet temperature
T_{CSO}	=	cold sea water outlet temperature
T_E	=	evaporating temperature
T_m	=	average sea water or working fluid temperature inside solar collector
T_{SCI}	=	solar collector inlet temperature
T_{SCO}	=	solar collector outlet temperature
T_{WSI}	=	warm sea water inlet temperature
T_{WSO}	=	warm sea water outlet temperature
U_C	=	overall heat transfer coefficient of condenser
U_E	=	overall heat transfer coefficient of evaporator
$(\Delta T_m)_C$	=	logarithmic mean temperature differences of the condenser
$(\Delta T_m)_E$	=	logarithmic mean temperature differences of the evaporator
η_c	=	thermal efficiency of solar collector
η_{net}	=	net Rankine cycle efficiency
η_G	=	generator efficiency
η_R	=	Rankine cycle efficiency
η_T	=	turbine efficiency
η_{CSP}	=	cold sea water pump efficiency
η_{WSP}	=	warm sea water pump efficiency
η_{WFP}	=	working fluid pump efficiency

I. Introduction

Ocean thermal energy conversion (OTEC) is a system that converts heat energy into electricity by using the temperature difference between surface water and cold deep water of the ocean[1, 2]. Considerable research effort has been directed to the development of OTEC. Uehara *et al.*[3–6] conducted numerous theoretical and experimental studies on the major components of OTEC system. Results of these studies reveal that ammonia is a suitable working fluid for a closed Rankine cycle OTEC plant. However, due to the small temperature difference (approximately 15 K to 25 K), the Rankine cycle efficiency is only 3% to 5%. This results in a high cost of electricity generated by an OTEC plant. Saitoh and Yamada [7] have proposed a concept of the multiple Rankine cycle system using both solar thermal energy and ocean thermal energy in order to improve the Rankine cycle efficiency. However, any analysis has not been conducted.

In this study, we presently propose an OTEC system that utilizes not only ocean thermal energy but also solar thermal energy as heat sources. This power generation system is termed as SOTEC (solar and ocean thermal energy conversion). A fundamental simulation of the SOTEC system with a typical low-cost flat plate type solar collector, which increases the turbine inlet temperature of the working fluid by 20 K, is performed and the result is compared with that of an ordinary OTEC system.

II. Simulation of 100-kWe SOTEC plant

A. SOTEC System and Cycle

Figures 1 and 2 show schematics of the ordinary closed-cycle OTEC plant and the proposed SOTEC plants; they show the general arrangement of the heat exchangers, pumps, piping, turbine generator, and solar collector. In this simulation the solar collector is installed in two configurations. In the case of SOTEC-1, the warm sea water is heated by solar collector as shown in Fig.2 (a), while in the case of SOTEC-2, the working fluid after the evaporator is heated by the solar collector as shown in Fig.2 (b). The authors assumed an ideal saturated Rankine cycle to determine the theoretical thermal efficiency of the Rankine cycle. The corresponding *T*-*s* diagram is shown in Fig.3. Here, T_E is the evaporating temperature, T_C the condensing temperature, T_{WSI} the warm sea water inlet temperature, T_{WSO} the warm sea water outlet temperature, T_{CSI} the cold sea water inlet temperature, T_{CSO} the cold sea water outlet temperature, T_{SCO} the solar collector outlet temperature, Q_E the heat flow rate at the evaporator, and Q_C the heat flow rate at the condenser.

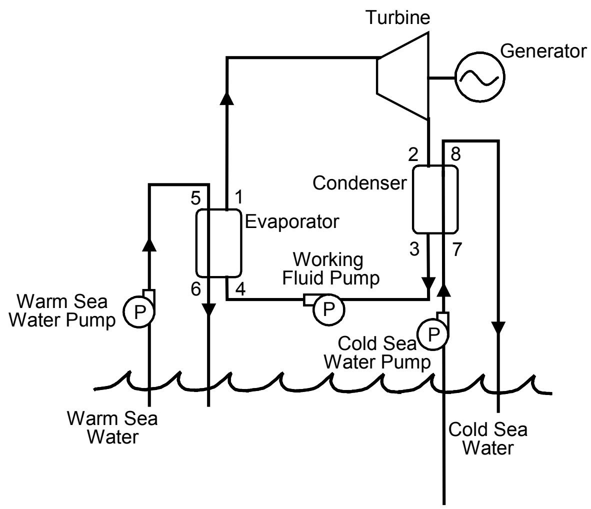


Figure 1. Components of ordinary closed-Rankine cycle OTEC and their arrangement.

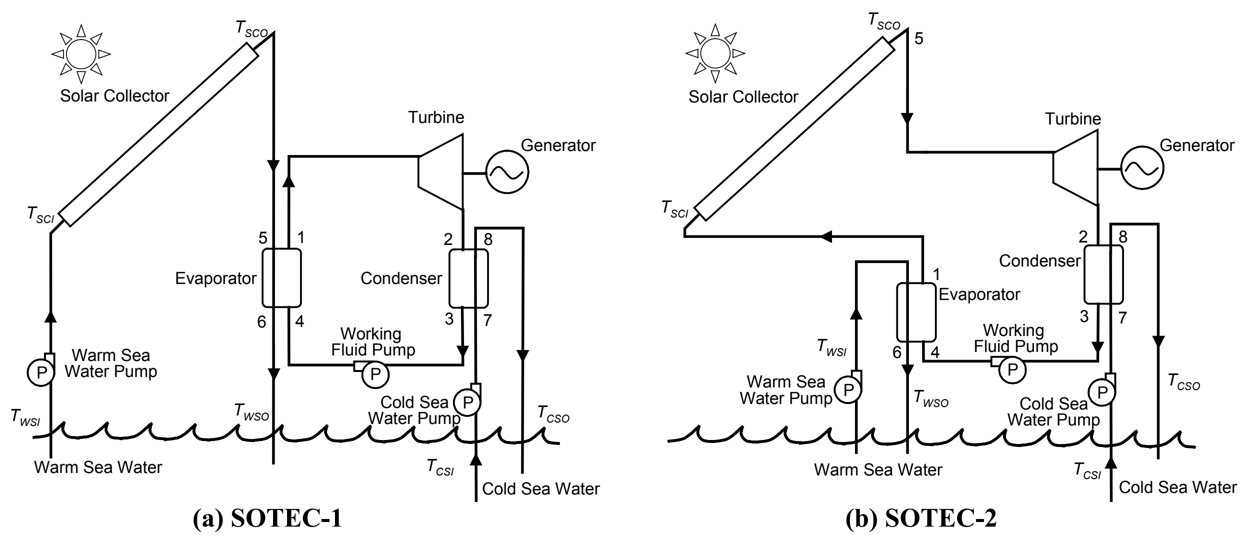


Figure 2. Components of the proposed SOTEC and their arrangement.

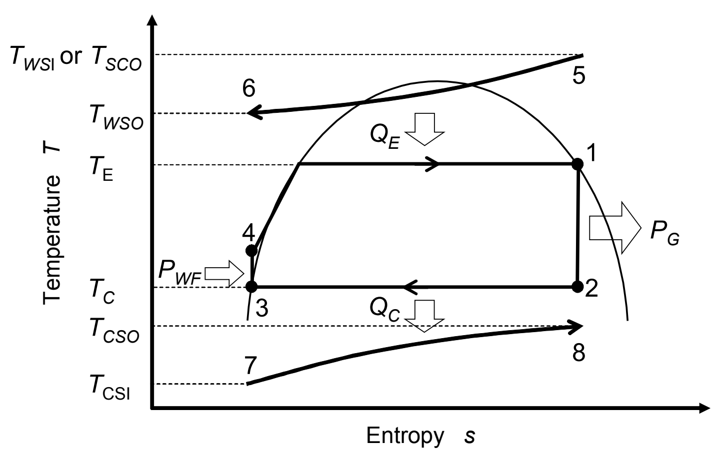


Figure 3. *T-s* diagram of the closed Rankine cycle.

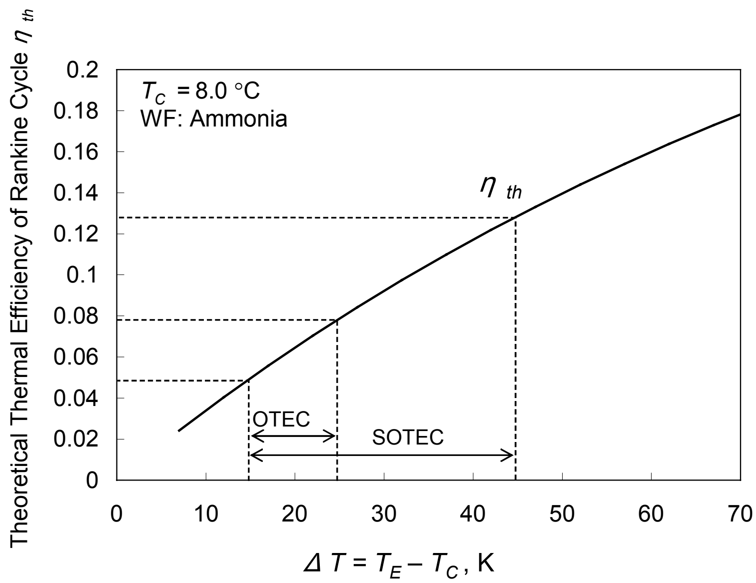


Figure 4. Theoretical thermal efficiency of saturated Rankine cycle in the operating temperature range ΔT of OTEC and SOTEC.

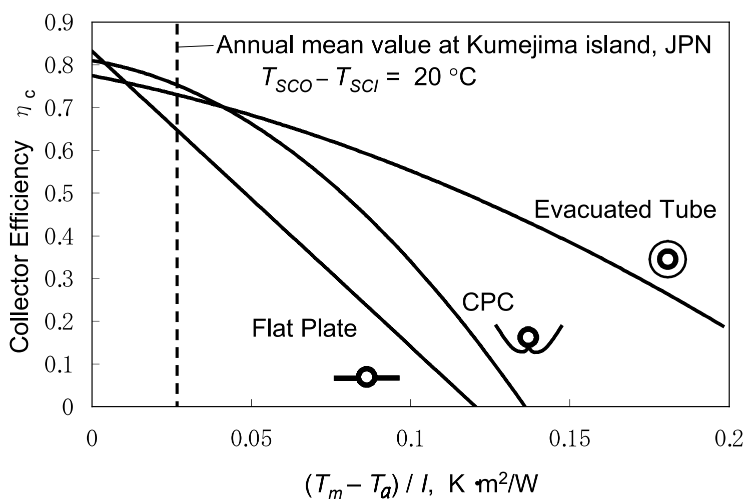


Figure 5. Collector efficiency of flat plate, evacuated tube, and CPC solar collectors.

Figure 4 shows the relationship between theoretical thermal efficiency of the Rankine cycle and the temperature difference $\Delta T = T_E - T_C$. The conventional OTEC has ΔT between 15 K and 25 K, so that the maximum theoretical thermal efficiency is approximately 8%. If the solar collector can additionally increase T_E by 20 K in the SOTEC, then the thermal efficiency of the SOTEC can be improved up to 13%, and the turbine inlet pressure increases up to 2.3 MPa.

Figure 5 shows collector efficiency curves of typical flat plate type solar collector, evacuated tube type solar collector, and compound parabolic concentrator (CPC) type solar collector. These collectors are typically used for residential water heating and the cost of these collectors is much lower than high temperature solar concentrating collectors due to a mass production. Annual mean collector efficiencies of these solar collectors were estimated to be high (approximately 65% to 75% in the case of SOTEC-1) for the weather condition in Kumejima island (Lat. 26-20N, Lng. 126-48E) near Okinawa island in Japan, which is the planned construction site of the OTEC plant. For the low temperature operating range the flat plate type solar collector seems to be efficient and be the most cost-effective way to enhance the thermal efficiency of OTEC at this time. In the present simulation, each of these solar collectors is incorporated as the additional heating device in the SOTEC system and the required collector area is calculated. The flat plate type solar collector has already been used as the evaporator of an organic working fluid, which has a lower boiling temperature than water, in the solar heat-pump water heater developed by Morrison, G.L [8].

The following assumptions are applied to the present simulation.

1. Solar collector efficiency η_c is calculated using the collector efficiency curve shown in Fig.5. Here, average sea water or working fluid temperature inside solar collector $T_m = (T_{SCI} + T_{SCO})/2$ in the case of SOTEC-1 and $T_m = T_E$ in the case of SOTEC-2.
2. Thermodynamic cycle of OTEC, SOTEC-1, and SOTEC-2 is ideal saturated Rankine cycle using pure Ammonia as working fluid.
3. Heat losses from piping and other auxiliary components are negligible.

B. Equations and Conditions for Simulation

The net power P_N of the OTEC or SOTEC plant is defined as

$$P_N = P_G - \left(P_{WS} + P_{CS} + P_{WF}\right) \tag{1}$$

where P_G is the turbine generator power; P_{WS} the warm sea water pumping power; P_{CS} the cold sea water pumping power; and P_{WF} the working fluid pumping power. The value of P_G is calculated from the product of the working fluid mass flow rate m_{WF} and the adiabatic enthalpy difference between the evaporator and the condenser, and is calculated as follows:

$$P_G = m_{WF}\eta_T\eta_G\left(h_1 - h_2\right) \tag{2}$$

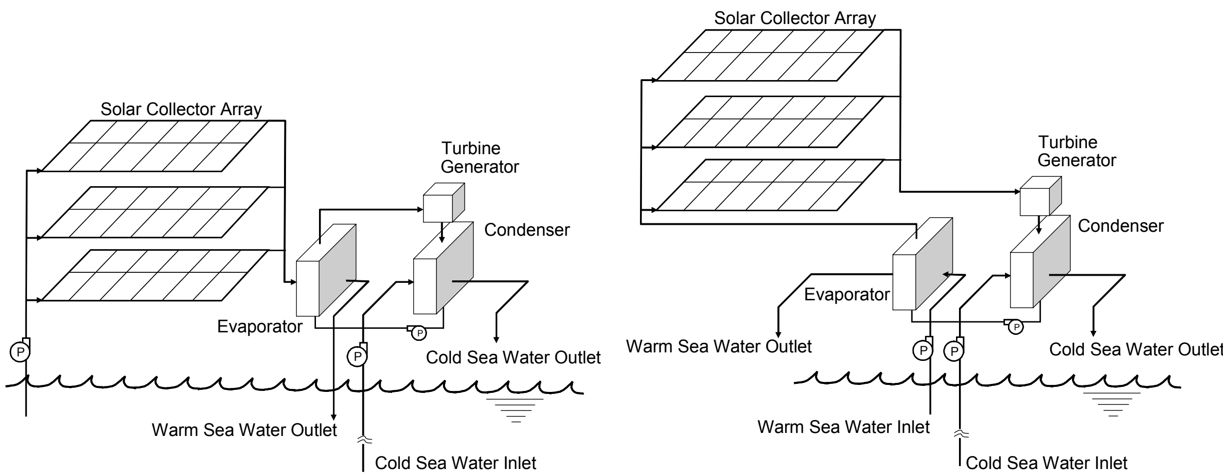
where η_T is the turbine efficiency and η_G is the generator efficiency. The values of enthalpies h_1 and h_2 are calculated using PROPATH [9]. The values of P_{WS}, P_{CS}, and P_{WF} are calculated as follows:

$$P_{WS} = m_{WS}\Delta H_{WS}\, g / \eta_{WSP} \tag{3}$$

$$P_{CS} = m_{CS}\Delta H_{CS}\, g / \eta_{CSP} \tag{4}$$

$$P_{WF} = m_{WF}\Delta H_{WF}\, g / \eta_{WFP} \tag{5}$$

where g is the acceleration of gravity and m_{WS}, m_{CS}, m_{WF}, ΔH_{WS}, ΔH_{CS}, ΔH_{WF}, η_{WSP}, η_{CSP}, and η_{WFP} are mass flow rate, total pressure difference, and pump efficiency of the piping of warm sea water, cold sea water, and working fluid, respectively. The pressure difference of each piping is calculated using equations derived by Uehara and Ikegami[4]. They performed a detailed optimization of a closed-cycle 100-MW OTEC plant with plate-type heat exchangers using ammonia as the working fluid. Figure 6 shows a piping diagram of the proposed SOTEC plant. Table 1 shows the piping condition of the SOTEC plant. Specification of the plate type heat exchanger is as same as that of Uehara and Ikegami[4]. Total pressure head of the solar collector array including piping is assumed to be 15m.



(a) SOTEC-1 **(b) SOTEC-2**

Figure 6. Piping diagram of the SOTEC plant

Table 1. Piping for the simulation.

Warm sea water pipe		
Length	m	50
Diameter	m	0.7
Cold sea water pipe		
Length	m	1000
Diameter	m	0.7
Plate type heat exchanger (evaporator and condenser)		
Plate length	m	4.0
Plate width	m	1.5
Plate thickness	mm	1.0
Clearance of sea water side	mm	5.0
Clearance of working fluid side	mm	5.0
Total pressure head of solar collector array and piping	m	15

Table 2 shows the conditions for the simulation. The overall heat transfer coefficients of the evaporator U_E and the condenser U_C are assumed to be 4000 W/m^{2}K and 3500 W/m^{2}K, respectively. These values have been experimentally obtained by Uehara and Nakaoka[3]. The heat transfer areas of the evaporator A_E and the condenser A_C are defined as

$$A_E = Q_E / (U_E (\Delta T_m)_E) = m_{WS} c_{pWS} (T_{WSI} - T_{WSO}) / (U_E (\Delta T_m)_E) \tag{6}$$

$$A_C = Q_C / (U_C (\Delta T_m)_C) = m_{CS} c_{pCS} (T_{CSO} - T_{CSI}) / (U_C (\Delta T_m)_C) \tag{7}$$

where, $(\Delta T_m)_E$ and $(\Delta T_m)_C$ are the logarithmic mean temperature differences of the evaporator and the condenser, respectively. Q_E and Q_C are the heat flow rate of the evaporator and the condenser, respectively, defined as

$$Q_E = m_{WF} (h_1 - h_4) \tag{8}$$

$$Q_C = m_{WF} (h_2 - h_3) \tag{9}$$

Rankine cycle efficiency η_R and the net Rankine cycle efficiency η_{net} are given as follows:

$$\eta_R = P_G / Q_E \tag{6}$$

$$\eta_{net} = P_N / Q_E \tag{7}$$

The annual mean weather and sea water data at Kumejima Island are used for the simulation [10].

Table 2. Conditions for the simulation.

Turbine generator power	P_G	kW	100
Turbine efficiency	η_T	-	0.80
Generator efficiency	η_G	-	0.96
Sea water pump efficiency	η_{WSP} η_{CSP}	-	0.80 0.80
Working fluid pump efficiency	η_{WFP}	-	0.75
Evaporator (plate-type heat exchanger)			
Overall heat transfer coefficient	U_E	W/m^2K	4000
$T_{WSI} - T_E$ (OTEC, SOTEC-2) and $T_{SCO} - T_E$ (SOTEC-1)		K	4.0
Condenser (plate-type heat exchanger)			
Overall heat transfer coefficient	U_C	W/m^2K	3500
$T_C - T_{CSI}$		K	4.0
Solar collector			
Tilt angle		°	30
Azimuth angle		°	0 (facing South)
Weather condition (Annual mean value in Kumejima, Japan)			
Ambient temperature	T_a	°C	22.6
Solar radiation on collector surface	I	W/m^2	457
Sea water temperature (Annual mean value in Kumejima, Japan)			
Warm sea water temperature at depth 0 m		°C	25.7
Cold sea water temperature at depth 1000 m		°C	4.4

C. Simulation Results

The authors assumed that the ordinary OTEC plant is running during nighttime and the daytime when the weather condition is not suitable for SOTEC operation, in other words, SOTEC plant is selectively operated instead of OTEC only when the weather condition satisfies the available heat collection of solar collector.

First, the ordinary 100-kWe OTEC plant without solar collector was simulated and the heat transfer areas of the evaporator and condenser were determined. Next, the SOTEC plant was simulated by optimizing the flow rates of warm and cold sea water in order to ensure that the heat transfer capacities of the evaporator and condenser for the OTEC are not exceeded. In the case of SOTEC, the turbine inlet temperature i.e. the evaporating temperature increase due to the solar collector is assumed to be 20 K, and the solar collector area which achieves this temperature increase was estimated.

Table 3 shows the simulation results of the 100-kWe plants of OTEC, SOTEC-1, and SOTEC-2. As a result, in the present condition, the net Rankine cycle efficiency η_{net} increases from 2.3% for OTEC to 6.4% for SOTEC-1, and 6.9% for SOTEC-2. Total annual heat collectable time of solar collector at Kumejima island is 2440 hours (102 running days), so that the annual mean net Rankine cycle efficiency of SOTEC-1 and SOTEC-2 is improved by 150% and 153% of ordinary OTEC, respectively.

According to this result, the required area of solar collectors of SOTEC-1 are approximately 1.6 times as large as that of SOTEC-2. This result is mainly caused by the lack of heat transfer area of evaporator. If the heat transfer area of the SOTEC-1 evaporator is assumed to be 310 m^2, which is about 80 m^2 larger than that required by OTEC, then SOTEC-1 needs the same collector area as SOTEC-2. The flat plate solar collector efficiency of SOTEC-2 has the lowest value because the T_m of SOTEC-2 becomes higher than that of SOTEC-1 due to the constant temperature during evaporation. The SOTEC-2 with an evacuated tube solar collector has the minimum solar collector area of $3878m^2$, while the flat plate solar collector area of SOTEC-2 is $5017m^2$. Due to the high pressure in this application, the cost of the flat plate solar collector will be lower than that of evacuated tubes at this time. However, the evacuated tube solar collector could be feasible for SOTEC because the cost of the latest evacuated tube solar collector is being dramatically reduced due to development of mass production technology. However special plumbing is required in SOTEC-2 due to the high pressure of the working fluid.

Since the heat exchanger cost (approximately 25% to 50% of the plant cost) is one of the major costs of the OTEC plant [6], SOTEC-2 is better than SOTEC-1 from this point of view. The problem with SOTEC-2 is, however, corrosion resistance and pressure tightness of the solar collector which directly heats the ammonia working fluid. Although flat type solar collectors with stainless absorber pipes are available commercially and some types of the evacuated tube collector have the required corrosion resistance and pressure tightness, further research and cost estimation is necessary to verify the collector efficiency as an evaporator of the ammonia working fluid and to judge the feasibility of SOTEC-2.

III. Conclusion

A combined ocean thermal energy and solar thermal energy (SOTEC) system is proposed and fundamental simulation was performed. The results reveal that the installation of solar collector enhances the thermal efficiency of an OTEC system. For future studies, the authors intend to perform a transient annual system simulation including a heat storage system, estimate the detailed cost of SOTEC plant, and experimentally verify the collector efficiency in case that the collector is used as the evaporator for ammonia working fluid.

Table 3. Simulation results of 100-kWe OTEC, SOTEC-1, and SOTEC-2 plants.

			OTEC	SOTEC-1	SOTEC-2
Warm sea water inlet temperature	T_{WSI}	°C	25.7	25.7	25.7
Warm sea water outlet temperature	T_{WSO}	°C	22.9	33.1	20.4
Cold sea water inlet temperature	T_{CSI}	°C	4.4	4.4	4.4
Cold sea water outlet temperature	T_{CSO}	°C	7.1	8.5	8.5
Solar collector inlet temperature	T_{SCI}	°C	-	25.7	41.7
Solar collector outlet temperature	T_{SCO}	°C	-	45.7	41.7
Evaporation temperature	T_E	°C	21.7	41.7	41.7
Condensate temperature	T_C	°C	8.4	8.4	8.4
Net power	P_N	kW	69.9	84.0	91.4
Warm sea water pumping power	P_{WS}	kW	9.5	7.7	0.1
Cold sea water pumping power	P_{CS}	kW	18.6	5.4	5.4
Working fluid pumping power	P_{WF}	kW	2.1	2.9	3.2
Warm sea water flow rate	m_{WS}	kg/s	260	27.2	3.2
Cold sea water flow rate	m_{CS}	kg/s	260	76.0	76.0
Working fluid flow rate	m_{WF}	kg/s	2.4	1.0	1.0
Rankine cycle efficiency	η_R	-	3.4	7.6	7.6
Net Rankine cycle efficiency	η_{net}	-	2.3	6.4	6.9
Heat transfer area of evaporator	A_E	m^2	232	221	12
Heat transfer area of condenser	A_C	m^2	390	288	288
Flat plate solar collector					
Collector efficiency	η_c	-	-	0.63	0.54
Collector area	A_{SC}	m^2	-	7574	5017
Evacuated tube collector					
Collector efficiency	η_c	-	-	0.73	0.70
Collector area	A_{SC}	m^2	-	6588	3878
CPC solar collector					
Collector efficiency	η_c	-	-	0.74	0.69
Collector area	A_{SC}	m^2	-	6458	3933

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